CHAPTER 01

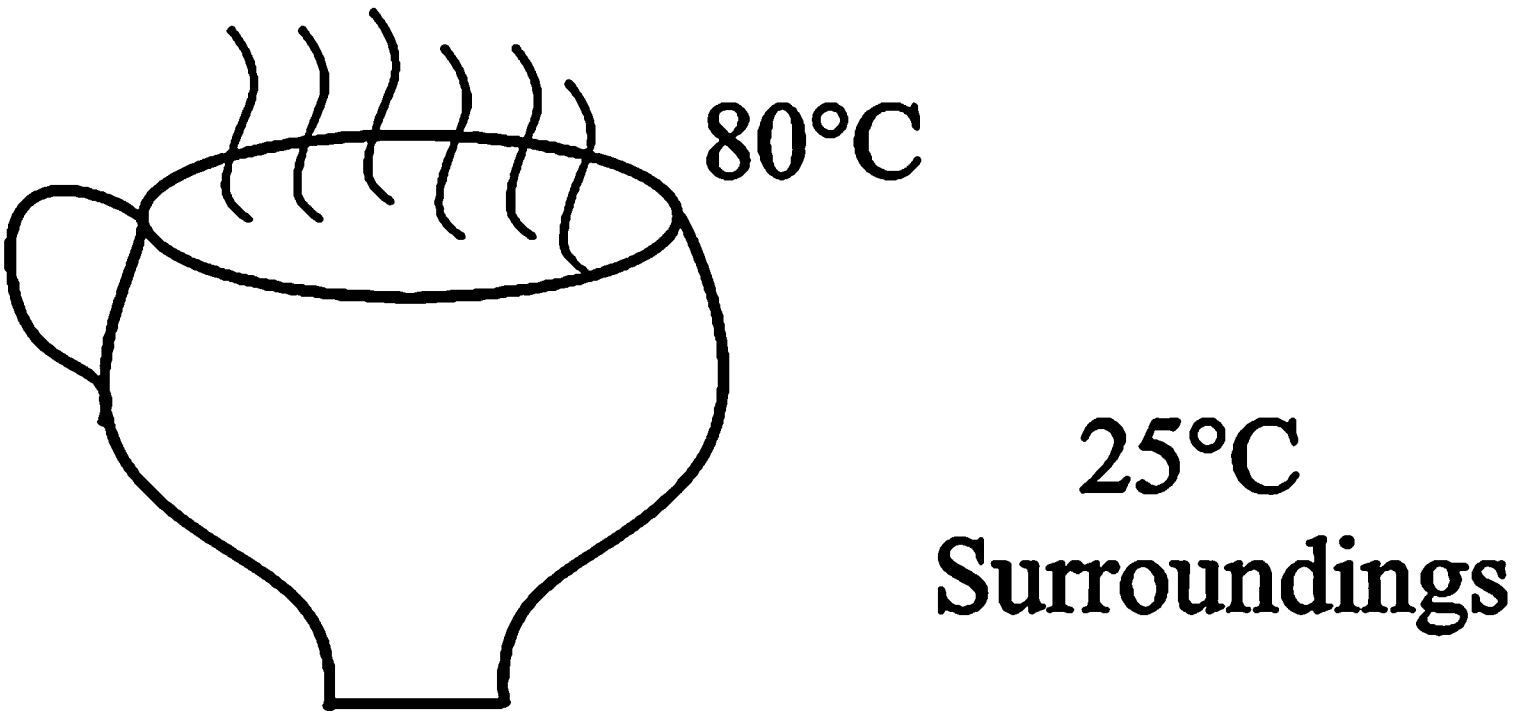
SOLUTIONS

Question: 1.002P

An office worker claims that a cup of cold coffee on his table warmed up to 80°C by picking up energy from the surrounding air, which is at 25°C. Is there any truth to his claim? Does this process violate any thermodynamic laws?

Step 1 of 1

A cup of hot coffee is kept in the living room; after a certain period of time, the coffee gets cold. To heat the coffee we have to spend some energy.



In our wash room, we get water by opening the tap (Valve). This water comes from the overhead tank which is located at an elevated place. To transfer this water back to the elevated place we need to spend some energy.

So, from the above naturally occurring processes, we can say that heat flows from a higher temperature to lower temperature. Water flows from an elevated place to a lower place.

Therefore, all the spontaneous processes occurring in nature are in one direction. To reverse the direction of process we need to spend energy.

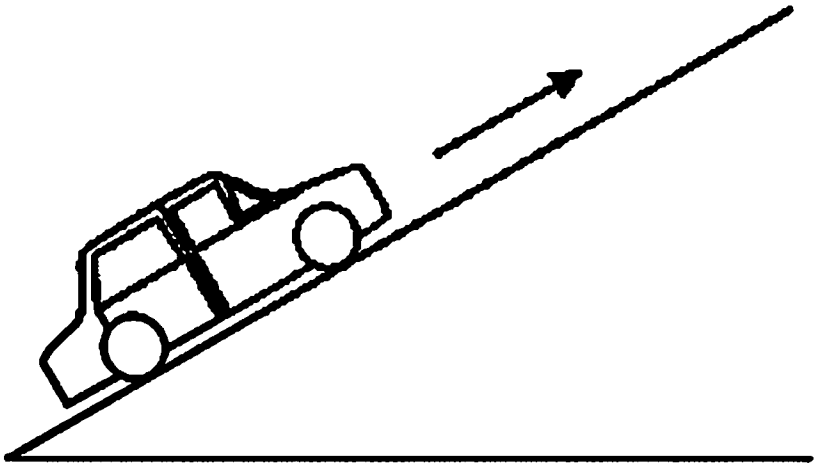
So, without any external work, it is impossible to raise the temperature of a body higher than the surrounding temperature.

Therefore, there is no truth in his claim. It violates the second law of thermodynamics.

Question: 1.003P

One of the most amusing things a person can experience is that in certain parts of the world a still car on neutral going uphill when its brakes are released. Such occurrences are even broadcast on TV. Can this really happen or is it a bad eyesight? How can you verify if a road is really uphill or downhill?

Step 1 of 2



Step 2 of 2

A car cannot go uphill without the engine is running. Such things cannot happen. It violates the laws of physics. On TV shows it just a camera trick.
Using a spirit level or any level meters we can verify the steepness of a road

Question: 1.004P

How does the science of heat transfer differ from the science of thermodynamics?

Step 1 of 1

The science of thermodynamics deals with the amount of heat transfer as a system when it undergoes a process change from one equilibrium state to another state. While heat transfer on the other hand, deals with the rate of heat transfer as well as the temperature distribution within the system at a specified time.

For example when designing a radiator in an automobile, thermodynamics helps to determine the amount of energy transfer to be taken place by using the properties of the coolant used in the radiator. While heat transfer helps to design the radiator tubes like diameter of the tubes and shape of the tubes.

Question: 1.005P

What is the driving force for (*a*)heat transfer, (*b*) electric current, and (*c*) fluid flow?

Step 1 of 3

a)

Temperature difference is the driving force for heat transfer because without temperature difference heat can not to be flow from one body to the other.

Step 2 of 3

b)

The driving force for electric current flow is the electric potential difference (voltage).

Step 3 of 3

c)

The driving force for fluid flow is the pressure difference that exists in it.

Question: 1.006P

Why is heat transfer a nonequilibrium phenomenon?

Step 1 of 1

The science of Heat transfer deals with the amount of heat transfer as well as the temperature distribution within the system at a specified time. Two bodies are in thermal equilibrium means two bodies temperature are same. But in heat transfer without temperature difference heat can not to be flow from one body to the other. Hence in heat transfer state of the bodies are changed with respect to time. Thus it is a non-equilibrium phenomenon.

Question: 1.007P

Can there be any heat transfer between two bodies that are at the same temperature but at different pressures?

Step 1 of 1

If two bodies temperature are same then there is no driving force to transfer of heat because the driving force exists when there is temperature difference between them. So there will be no heat transfer.

Question: 1.008P

Define stress, normal stress, shear stress, and pressure. Mass, Force, and Units

Step 1 of 3

Stress: It is defined as the force acting per unit area.

Equation of stress:

$$\sigma = \frac{F}{A}\left(\mathrm{N/m^2}\right)$$

Here, F is the force acting and A is the area of cross-section.

Step 2 of 3

Force applied is acting in the normal direction to the cross section, then the stress is defined as **normal force.**

If the force applied is acting in the tangential direction to the cross-section, then the corresponding stress is called **shear stress.**

Step 3 of 3

Pressure: It is define as the force which is acted continuously on or against an object by which the force is in contact with it.

Question: 1.009P

In a news article, it is stated that a recently developed geared turbofan engine produces 15,000 pounds of thrust to propel the aircraft forward. Is "pound" mentioned here lbm or lbf? Explain.

Step 1 of 1

Pound force (lbf) is the force unit whereas pound-mass (lbm) is the mass unit in English system. One pound force is the force required to accelerate a mass of 32.174 lbm by $1 ft/s^2$. The 'pound' mentioned here is 'lbf' because of geared turbofan engine produces 15000 pounds of thrust (force) to propel the aircraft to move forward (accelerate).

Question: 1.010P

Explain why the light-year has the dimension of length.

Step 1 of 1

In space the distance between stars, planet – planet, planet – sun is usually measured in light – years.

The word 'light' refers to speed at which the light travels

So, light – year is then the product of velocity and time

$$\text{Velocity} = \frac{\text{displacement}}{\text{time}} \quad \frac{\text{m}}{\text{s}}$$

$$1\ \text{Light} - \text{year} = \text{Velocity} \times \text{time}$$

$$= \text{m/s} \times \text{s}$$

$$= \text{m}$$

Hence the unit of light – year is m

One light-year is equivalent to 9.461×10^{12} km, or 5.879×10^{12} mi

Question: 1.011P

What is the net force acting on a car cruising at a constant velocity of 70 km/h (*a*) on a level road and (*b*) on an uphill road?

Step 1 of 3

Given:
Constant velocity $= 70\,\text{km/h}$

Step 2 of 3

Since velocity is constant, acceleration becomes zero.
So,
(a)
On a level road:
Net force acting on car:
$F = \text{mass} \times \text{acceleration}$
$F = 0$

Step 3 of 3

(b)
On an uphill road:
Net force acting on car:
$F = \text{mass} \times \text{acceleration}$
$F = 0$

In both cases, net force acting on the car is zero as force depends upon acceleration but not on the type of surface i.e on a level road or on an uphill track.

Question: 1.012P

A man weighs 210 lbf at a location where g=32.10 ft/s2. Determine his weight on the moon, where g=5.47 ft/s2.

Step 1 of 1

Write the Newton's second law of motion.

$$W = mg$$

Here, W is the weight, m is the mass of the body, and g is the acceleration due to gravity.

Mass will be constant.

Write the relation between weight and gravity at a location and on the moon.

$$\frac{W_{moon}}{W} = \frac{g_{moon}}{g}$$

Here, W_{moon} is the weight of man on the moon, W is the weight of the man at a location, g_{moon} is the acceleration due to gravity on the moon, and g is the acceleration due to gravity at a location.

Substitute $210\,\text{lbf}$ for W , $5.47\,\text{ft/s}^2$ for g_{moon} , and $32.1\,\text{ft/s}^2$ for g .

$$\frac{W_{moon}}{210} = \frac{5.47}{32.1}$$
$$W_{moon} = 35.78\,\text{lbf}$$

Therefore, the weight of the man on the moon is $\boxed{35.78\,\text{lbf}}$.

Question: 1.013P

Determine the mass and the weight of the air contained in a room whose dimensions are 6 m × 6 m × 8 m. Assume the density of the air is 1.16 kg/m3.

Step 1 of 2

Calculate the mass of the air in the room.

$$m = \rho V$$

Here, V is volume and ρ is the density,

Substitute 1.16 kg/m^3 for ρ and 6 m×6 m×8 m for V in the equation.

$$m = \rho V$$
$$= 1.16 \times 6 \times 6 \times 8$$
$$= 334.08 \text{ kg}$$

Therefore, the mass of air in the room is **334.08 kg**.

Step 2 of 2

Calculate the weight of air in the room.

$$W = m \times g$$

Here, g is acceleration due to gravity.

Substitute 9.81 m/s^2 for g and 334.08 kg for m.

$$W = m \times g$$
$$= 334.08 \times 9.81$$
$$= 3277.32 \text{N}$$

Therefore, the weight of air in the room is **3277.32N**.

Question:

1.014P

At 45° latitude, the gravitational acceleration as a function of elevation zabove sea level is given by g =a-bz,where a=9.807 m/s2 and b= 3.32 × 10-6 s-2. Determine the height above sea level where the weight of an object will decrease by 0.5 percent.

Step 1 of 3

Given:
gravitational acceleration a function for elevation Z above sea level as
$g = a - bz$
Where
$a = 9.807 \frac{\text{m}}{s^2}$,
$b = 3.32 \times 10^{-6} s^{-2}$
Percentage of weight of the object decreased 0.5%

Step 2 of 3

The weight of object at the elevation z can be expressed as
$W = mg$
$= m(a - bz)$
$= m(9.807 - 3.32 \times 10^{-6} \times z)$.....(1)
According to our problem new weight of object is 0.995 times the weight at sea level
$W = 0.995W$,
$= 0.995m \times 9.807$.....(2)

Step 3 of 3

Equating the equation 1 and 2 we get
Height above the sea level is
$\Rightarrow m \times (9.81 - 3.32 \times 10^{-6} \times z) = 0.995 \times m \times 9.807$
$\Rightarrow z = 14,770m$
$\boxed{z = 14,770\text{m}}$

Question: 1.015P

What is the weight, in N, of an object with a mass of 200 kg at a location where g =9.6 m/s2?

Step 1 of 2

Given:

Mass $m = 200\,\text{kg}$

Acceleration due to gravity $g = 9.6\,\frac{\text{m}}{\text{s}^2}$

Step 2 of 2

Using Newton's second law

Weight of object

$W = mg$

$$W = 200(\text{kg}) \times 9.6\left(\frac{\text{m}}{\text{s}^2}\right)\left(\frac{1\text{N}}{\frac{1\text{kg}\cdot\text{m}}{\text{s}^2}}\right)$$

$$= 1920\text{N}$$

$$\boxed{W = 1920\text{N}}$$

Question: 1.016P

The constant-pressure specific heat of air at 25°C is 1.005 kJ/kg°C. Express this value in kJ/kg-K, J/g.°C, kcal/kg.°C, and Btu/lbm °F.

Step 1 of 4

From the unit conversion table

$$1\text{kJ/kg}\cdot\text{K}=1\text{kJ/kg}\cdot{}^\circ\text{C}$$

Use above conversion factor to convert the specific heat kJ/kg·°C to kJ/kg·K

$$c_p = 1.005\text{kJ/kg}\cdot{}^\circ\text{C}\left(\frac{1\text{kJ/kg}\cdot\text{K}}{1\text{kJ/kg}\cdot{}^\circ\text{C}}\right)$$

$$=1.0051\text{kJ/kg}\cdot\text{K}$$

Thus, constant pressure specific heat in terms of kJ/kg·K is $\boxed{1.0051\text{kJ/kg}\cdot\text{K}}$

Step 2 of 4

From the unit conversion table

$$1\text{kg}=1000\text{g}$$

$$1\text{kJ}=1000\text{J}$$

Use above conversion factor to convert the specific heat kJ/kg·°C to J/g·°C

$$c_p = 1.005\text{kJ/kg}\cdot{}^\circ\text{C}\left(\frac{1\text{kg}}{1000\text{g}}\right)\left(\frac{1000\text{J}}{1\text{kJ}}\right)$$

$$=1.0051\text{J/g}\cdot{}^\circ\text{C}$$

Hence, constant pressure specific heat in terms of J/g·°C is $\boxed{1.0051\text{J/g}\cdot{}^\circ\text{C}}$

Step 3 of 4

From the unit conversion table

$$1\text{kcal}=4.1868\text{kJ}$$

Use above conversion factor to convert the specific heat kJ/kg·°C to kcal/kg·°C

$$c_p = 1.005\text{kJ/kg}\cdot{}^\circ\text{C}\left(\frac{1\text{kcal}}{4.1868\text{ kJ}}\right)$$

$$=0.2400\text{kcal/kg}\cdot{}^\circ\text{C}$$

Hence, constant pressure specific heat in terms of kcal/kg·°C is $\boxed{0.2400\text{kcal/kg}\cdot{}^\circ\text{C}}$

Step 4 of 4

From the unit conversion table

$$1\text{Btu/lbm}\cdot{}^\circ\text{F}=4.1868\text{kJ/kg}\cdot{}^\circ\text{C}$$

Use above conversion factor to convert the specific heat kJ/kg·°C to Btu/lbm·°F

$$c_p = 1.005\text{kJ/kg}\cdot{}^\circ\text{C}\left(\frac{\text{Btu/lbm}\cdot{}^\circ\text{F}}{4.1868\text{kJ/kg}\cdot{}^\circ\text{C}}\right)$$

$$=0.2400\text{Btu/lbm}\cdot{}^\circ\text{F}$$

Hence, constant pressure specific heat in terms of Btu/lbm·°F is $\boxed{0.2400\text{Btu/lbm}\cdot{}^\circ\text{F}}$

Question: 1.017P

. A 3-kg rock is thrown upward with a force of 200 N at a location where the local gravitational acceleration is 9.79 m/s2. Determine the acceleration of the rock, in m/s2.

Step 1 of 4

Given:

Mass of the rock $m = 3\text{kg}$

Upward force $F_{up} = 200\,\text{N}$

Acceleration due to gravity $g = 9.79\frac{\text{m}}{s^2}$

Step 2 of 4

Weigh of the rock

$W = m \times g$

$$= 3(\text{kg}) \times 9.79\left(\frac{\text{m}}{s^2}\right)\left(\frac{1\text{N}}{\frac{1\text{kg}\cdot\text{m}}{s^2}}\right)$$

$$= 29.37\text{N}$$

From the equilibrium of the rock weight of the block is equal to the force acting on the downward. So

$W = F_{Downward} = 29.37\,\text{N}$

Step 3 of 4

Net force that acts on the rock is

$F_{net} = F_{up} - F_{down}$

$= (200 - 29.37)\text{N}$

$= 170.63\text{N}$

Step 4 of 4

From the Newton's second law,

$\text{Force}(F) = \text{mass}(m) \times \text{acceleration}(a)$

The acceleration of the rock becomes

$$a = \frac{F}{m}$$

$$= \frac{170.63\text{N}}{3\text{kg}}\left(\frac{\frac{1\text{kg}\cdot\text{m}}{s^2}}{1\text{N}}\right)$$

$$= 56.88\frac{\text{m}}{s^2}$$

$$\boxed{a = 56.88\frac{\text{m}}{s^2}}$$

Question: 1.018P

Solve Prob. 1–17 using EES (or other) software. Print out the entire solution, including the numerical results with proper units.

Step 1 of 4

Free body diagram.

Step 2 of 4

Figure (1)

Step 3 of 4

Question:

Solve Prob. 1–17 using EES (or other) software. Print out the entire solution, including the numerical results with proper units.

Step 1 of 4

Free body diagram.

Step 2 of 4

Figure (1)

Step 3 of 4

Calculate the acceleration of the rock using EES software.

Write the following code in the software.

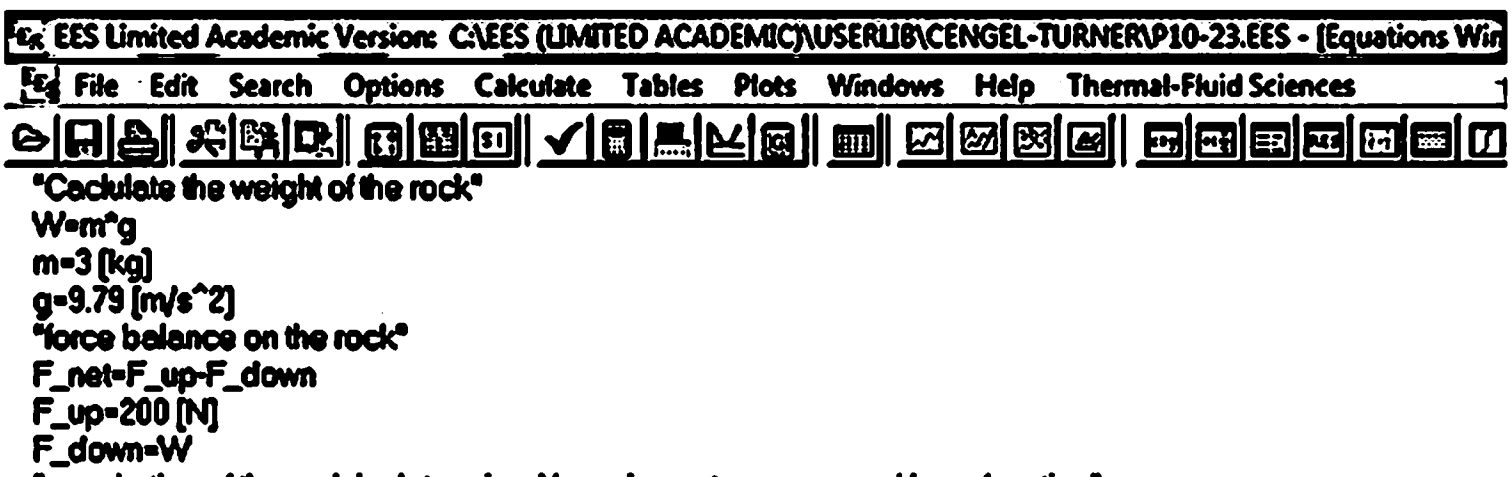
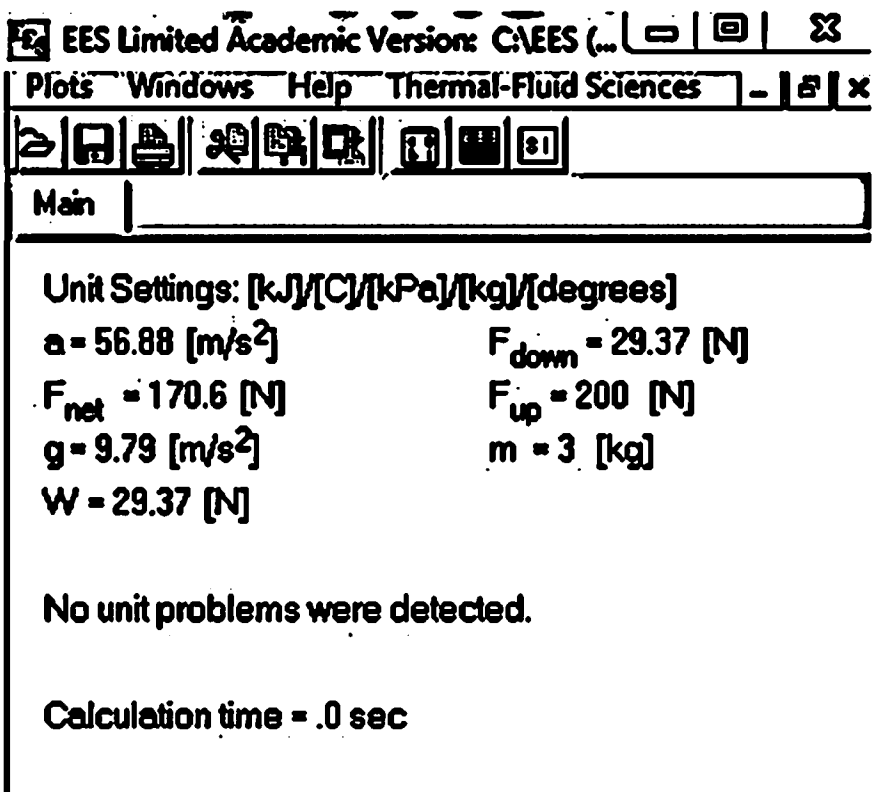


Figure (2).

Define the units of all variable by using "options" in the menu bar .

Step 4 of 4

Click on the "calculate" option in the menu bar and select solve and following output window will be displayed:



Question:

1.019P

While solving a problem, a person ends up with the equation E= 25 kJ + 7 kJ/kg at some stage. Here E is the total energy and has the unit of kilojoules. Determine how to correct the error and discuss what may have caused it.

Step 1 of 2

$E = 25\text{kJ} + 7\text{kJ/kg}$.

E has the unit of kilojoules. The cause of this problem is error in dimension of this formula. Since do not have the same units, therefore they can't be added to obtain the total energy.

To correct the answer we have to multiply the number 7 by mass of person. Then expression dimensionally homogenous.

Step 2 of 2

Then expression becomes,

$$E = 25\text{kJ} + 7\text{kJ/kg}$$

Question:

$$E = 25\text{kJ} + 7\text{kJ/kg}\,(\text{kg})$$

$$E(\text{kJ}) = 25\text{kJ} + 7\text{kJ}$$

Now above expression is dimensionally homogenous, therefore they can be added to obtain the total energy.

Question: 1.020P

A 4-kW resistance heater in a water heater runs for 2 hours to raise the water temperature to the desired level. Determine the amount of electric energy used in both kWh and kJ.

Step 1 of 3

Given:
Power rating of resistance heater $P = 4\text{kW}$
Time $t = 2\text{hr}$

Step 2 of 3

Amount of energy used in kJ is $= 28800\text{kJ}$

$$E = \left(4\frac{\text{kJ}}{\text{s}}\right) \times (2 \times 3600\text{s})$$

$$= 28800\text{kJ}$$

$$\boxed{E = 28800\text{kJ}}$$

Step 3 of 3

Electric energy used in kWh

$$E = 4\text{kW} \times 2\text{hour}$$

$$= 8\text{kWh}$$

$$\boxed{E = 8\text{kWh}}$$

Question: 1.021P

The gas tank of a car is filled with a nozzle that discharges gasoline at a constant flow rate. Based on unit considerations of quantities, obtain a relation for the filling time in terms of the volume u of the tank (in L) and the discharge rate of gasoline ú (in L/s).

Step 1 of 4

Given:
Volume of the tank 'V'
Rate of discharge $\dot{V}$

Step 2 of 4

Let filling time
$t \propto \left(V^a \times \dot{V}^b\right)$
Dimension of $t = T^1$
Dimension of $V = L^3$
Dimension of rate of Discharge $\dot{V} = L^3T^{-1}$
$t = kV_x^a\dot{V}^b$ .....(1)
Where
k = Constant

Step 3 of 4

$t = [T]^1$
$V^a \times \dot{V}^b = [L]^{3a} \times [L]^{3b}[T]^{-b}$
Equation dimension on both sides of equation (i)
$[L]^0[T]^1 = [L]^{3a+3b}[T]^{-b}$
Equating power of $[L]$
$3a + 3b = 0$
$a = -b$
Equating $[T]$
$-b = 1$
$b = -1$
So $a = 1$

Step 4 of 4

So the correct relation for filling tank is
$t \propto \frac{V}{\dot{V}}$
$t = \frac{kV}{\dot{V}}$
$\boxed{t = \frac{kV}{\dot{V}}}$

Question: 1.022P

A pool of volume *U*(in m3) is to be filled with water using a hose of diameter *D*(in m). If the average discharge velocity is *v*(in m/s) and the filling time is *t*(in s), obtain a relation for the volume of the pool based on until considerations of quantities involved.

Step 1 of 4

Given:

Volume of the pool $V\left(\mathrm{m}^3\right)$

Diameter of the hose $D(\mathrm{m})$

Average discharge velocity $V\left(\frac{\mathrm{m}}{\mathrm{s}}\right)$

Filling time $t(\mathrm{sec})$

Volume of the pool $U\left(\mathrm{m}^3\right)$

Step 2 of 4

Let volume of pool is related to D , V and t by

$U \propto (D^a V^b t^c)$

Dimension of $U = [L]^3$

Dimension of $D = [L]$

Dimension of $V = [LT^{-1}]$

Dimension of $t = [T]$

Step 3 of 4

So,

$U = k[D]^a [V]^b [t]^c$

Where k = proportionality constant

$[M]^0 [L]^3 [T]^0 = k[L]^a [LT^{-1}]^b [T]^c$

Equating power of T on both sides

$0 = -b + c \Rightarrow b = c$

Equation power of L on both sides

$3 = a + b$

Assuming value of b $= 1$

$a = 2$ And

$c = 1$

Step 4 of 4

Putting these values in our dimensional formula our dimensional relation satisfies so the correct relation is

$U = k(D^2 \times V \times t)$

$\boxed{U = k(D^2 \times V \times t)}$

Question: 1.023P

What is the value of the engineering software packages in (*a*) engineering education and (*b*) engineering practice?

Step 1 of 1

Engineering software packages are only tools though they are more convenient and capable. These packages cannot replace all time traditional engineering courses. But at the same time packages in practice are of great values. In these present days engineers mostly depend on software packages for evaluating big and complex problems within a short time for quick answer and to continue research studies effectively.

Question: 1.024P

Determine a positive real root of this equation using EES:

$2x3-10x0.5-3x=-3$

Step 1 of 4

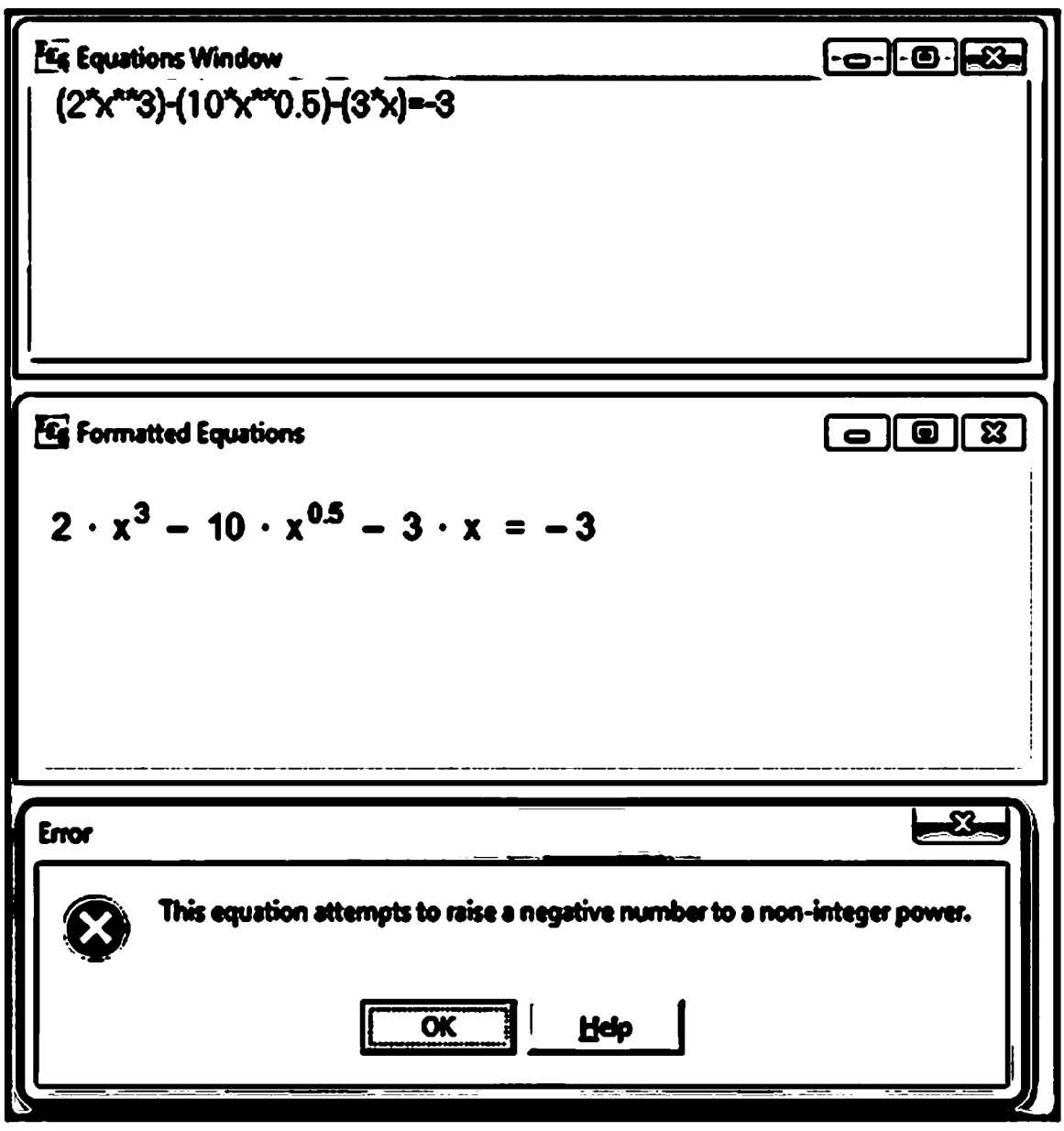
The given equation is $2x^3-10x^{0.5}-3x=-3$

For solving this equation using EES, the below steps are to be followed.

1. Enter the equation in new window of EES
2. Click on the 'windows tab' in the tool bar and then click on the 'formatted equations' (this will display the equations in mathematical form.)
3. Press F2 to solve the equation.

Step 2 of 4

Screenshot showing the EES output:



Step 3 of 4

The above error dialogue denotes that solving this equation requires a change of exponentials for variable.
In-order to find the solution to the given equation let us introduce a new variable y, so that the new equation can have a change of exponential.
Now the equations are

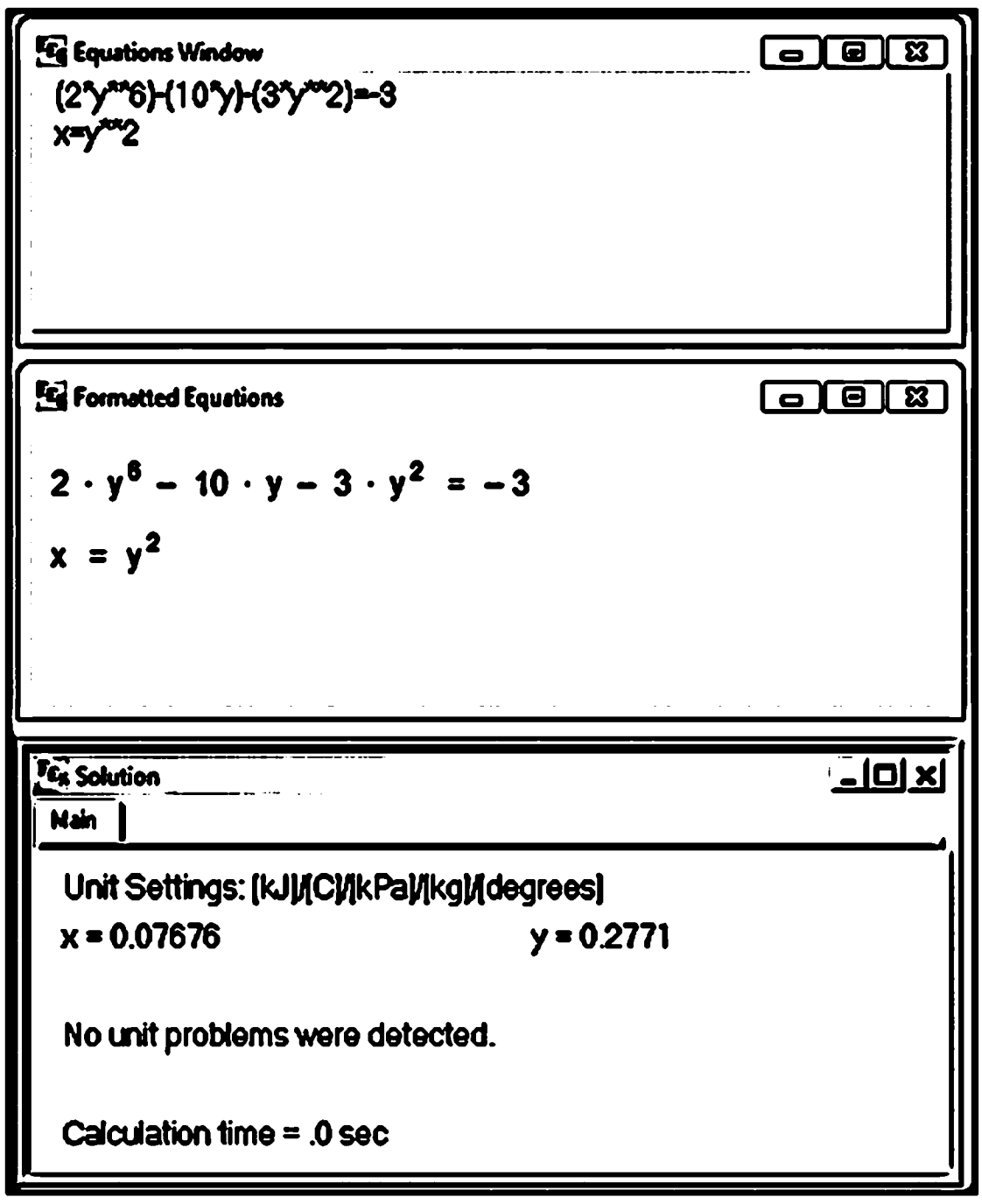
$2y^6-10y-3y^2=-3$

$x=y^2$

For solving this equation using EES; enter the equations, check the formatted equations and then press F2.

Step 4 of 4

Screenshot showing the EES output:



Hence the real positive root of the given equation is $\boxed{x=0.07676}$

Question: 1.025P

Solve this system of two equations with two unknowns using EES:

$x3-y2=7.75$

$3xy+y=3.5$

Step 1 of 2

Given data:

$x^3 - y^2 = 7.75$

$3xy + y = 3.5$

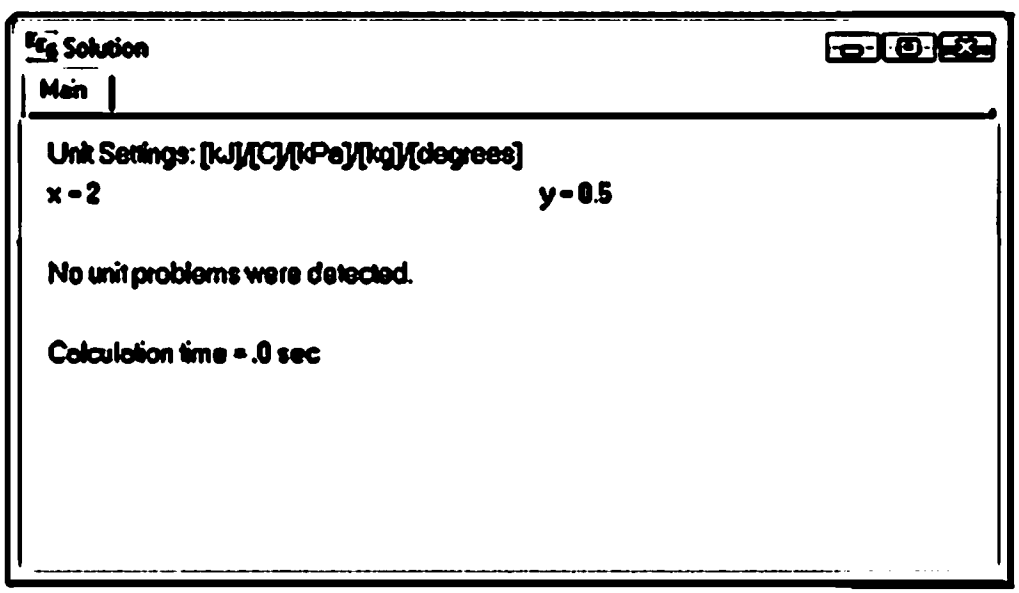
Step 2 of 2

By using EES software, writing the input equations as follows:

$(x\hat{}3)-(y\hat{}2)=7.75$

$(3*x*y)+y=3.5$

We get the final solution as;



Therefore, the two unknowns are $\boxed{x=2}$ and $\boxed{y=0.5}$

Question: 1.026P

Solve this system of three equations with three kiH unknowns using EES:

$2x-y+z=5$

$3x2+2y=z+2$

$xy+2z=8$

Step 1 of 4

The three given equations are

$$2x-y+z=5$$
$$3x^2+2y=z+2$$
$$xy+2z=8$$

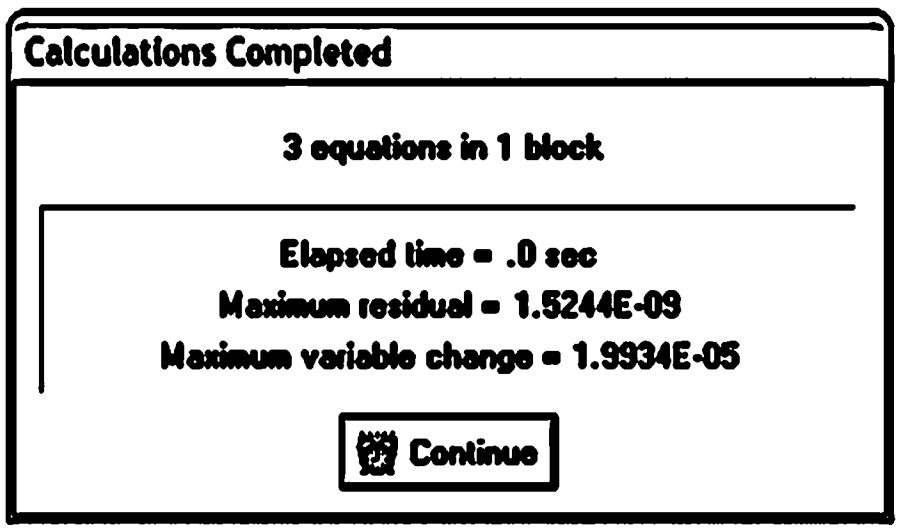
Step 2 of 4

Solve the above three equations by using EES software. Open EES window and write the three given equation in that window.

$$2*x-y+z=5$$
$$3*x\text{^}2+2*y=z+2$$
$$x*y+2*z=8$$

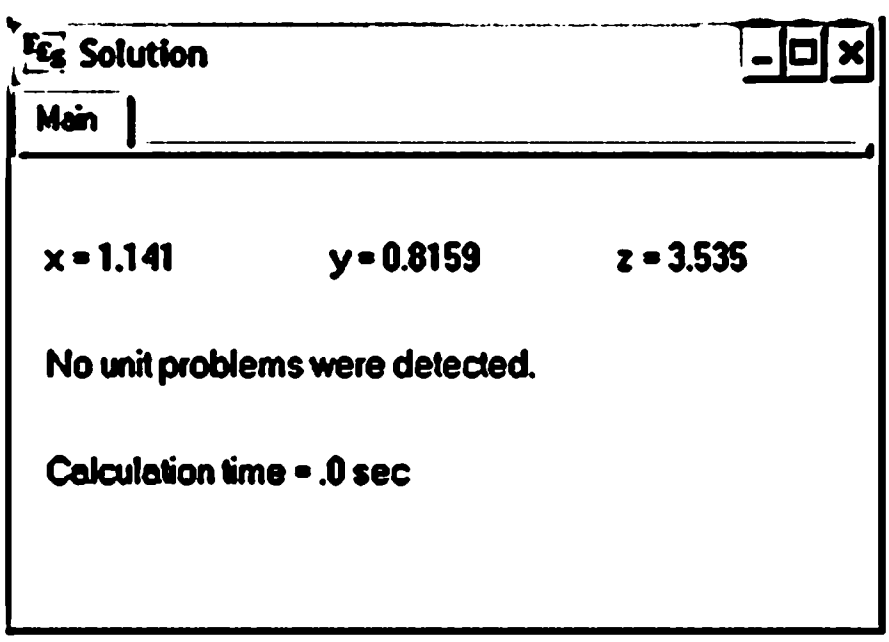
Step 3 of 4

Solve these equations by pressing F2. Then you will get one popup menu,



Step 4 of 4

Then press continue to get the final result.



Therefore, the solution for the given equations is

$\boxed{x=1.141}$

$\boxed{y=0.8159}$

$\boxed{z=3.535}$

Question: 1.027P

Solve this system of three equations with three unknowns using EES:

$x2y-z=1$

$x-3y0.5+xz=-2$

$x+y-z=2$

Step 1 of 2

Given data:

$$x^2y-z=1$$

$$x-3y^{0.5}+xz=-2$$

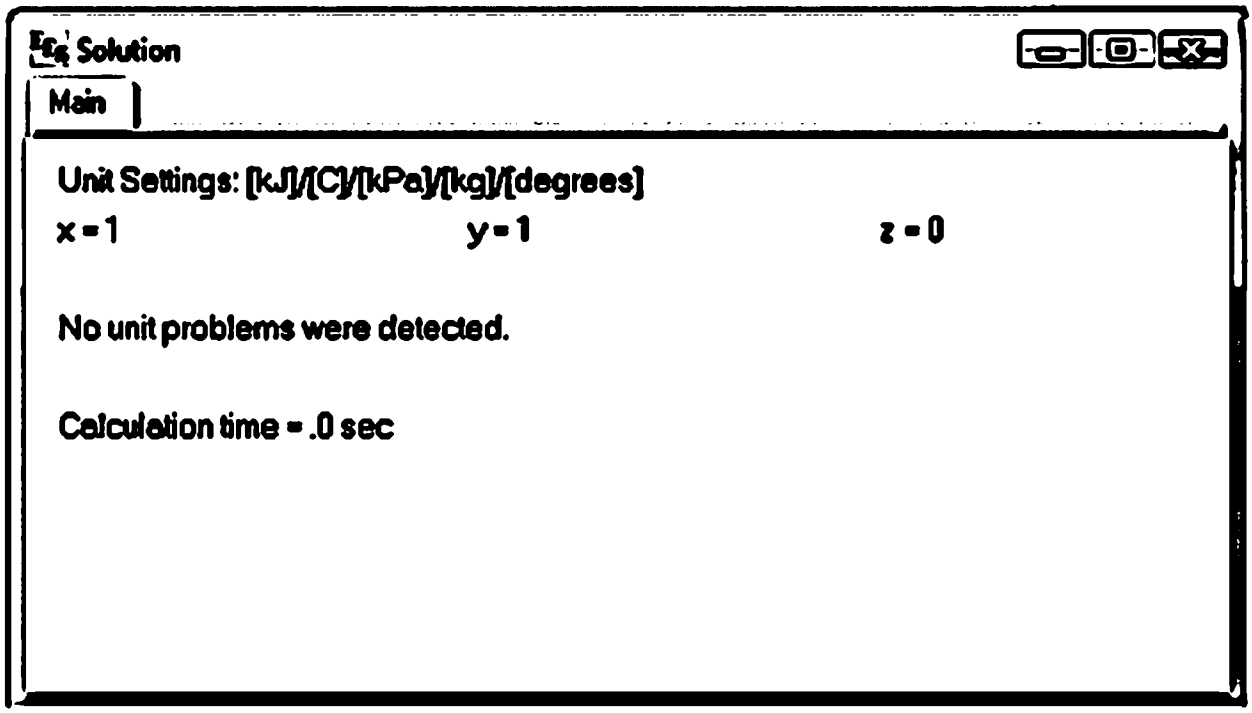
$$x+y-z=2$$

Step 2 of 2

By using EES software, writing the input equations as follows:

```
x^2*y-z=1
x-3*y^0.5+x*z=-2
x+y-z=2
```

We get the final solution as;



Therefore, the three unknowns are $\boxed{x=1}$, $\boxed{y=1}$ and $\boxed{z=0}$

Question: 1.028P

Specific heat is defined as the amount of energy needed to increase the temperature of a unit mass of a substance by one degree. The specific heat of water at room temperature is 4.18 kJ/kg·°C in SI unit system. Using the unit conversion function capability of EES, express the specific heat of water in (*a*) kJ/kg·K, (*b*) Btu/lbm-°F, (*c*) Btu/lbm-R, and (*d*) kcal/kg·°C units.

Step 1 of 4

Given data:

Specific heat of water at room temperature, $C_p = 4.18 \text{kJ/kg-°C}$

By using EES software, writing the input equations as follows:

Equations Window

```
"Given data:"
C_p=4.18 [kJ/kg-C]

"Using the below conversion formulae, we convert the units as:"
C_p_1=C_p*Convert(kJ/kg-C, kJ/kg-K)
C_p_2=C_p*Convert(kJ/kg-C, Btu/lbm-F)
C_p_3=C_p*Convert(kJ/kg-C, Btu/lbm-R)
C_p_4=C_p*Convert(kJ/kg-C, kCal/kg-C)
```

Step 2 of 4

We get the formatted equations as;

Formatted Equations

Given data:

$C_p = 4.18 \ [kJ/kg-C]$

Using the below conversion formulae, we convert the units as:

$$C_{p,1} = C_p \cdot \left| 1 \cdot \frac{\cdot kJ/kg-K}{kJ/kg-C} \right|$$

$$C_{p,2} = C_p \cdot \left| 0.238846 \cdot \frac{\cdot Btu/lbm-F}{kJ/kg-C} \right|$$

$$C_{p,3} = C_p \cdot \left| 0.238846 \cdot \frac{\cdot Btu/lbm-R}{kJ/kg-C} \right|$$

$$C_{p,4} = C_p \cdot \left| 0.238846 \cdot \frac{\cdot kCal/kg-C}{kJ/kg-C} \right|$$

Step 3 of 4

We get the final solution as;

Solution

Main

Unit Settings: [kJ]/[C]/[kPa]/[kg]/[degrees]

$C_p = 4.18$ [kJ/kg-C] $C_{p,1} = 4.18$ [kJ/kg-K] $C_{p,2} = 0.9984$ [Btu/lbm-F]

$C_{p,3} = 0.9984$ [Btu/lbm-R] $C_{p,4} = 0.9984$ [kCal/kg-C]

No unit problems were detected.

Purple units were automatically set. Right click on the variable to confirm or change the units.

Calculation time = .0 sec

Step 4 of 4

Therefore, the values after unit conversions are $\boxed{4.18 \text{kJ/kg-K}}$, $\boxed{0.9984 \text{Btu/lbm-°F}}$, $\boxed{0.9984 \text{Btu/lbm-R}}$ and $\boxed{0.9984 \text{kCal/kg-°C}}$

Question: 1.029P

A lunar exploration module weighs 4000 N at a location where g=9.8 m/s2. Determine the weight of this module in newtons when it is on the moon where g=1.64 m/s2.

Step 1 of 2

The weight of lunar module $W = 4000\ \text{N}$

The acceleration due to gravity on earth $g = 9.8\ \text{m/s}^2$

The acceleration due to gravity on moon $g = 1.64\ \text{m/s}^2$

Step 2 of 2

From Newton's second law

$$W = mg$$

$$m = \frac{W}{g}$$

$$= \frac{4000}{9.81}$$

$$= 407.7\ \text{kg}$$

As we all know mass is same everywhere in the universe

Weight of the lunar module in moon

$$W_{\text{moon}} = mg_{\text{moon}}$$

$$= 407.7 \times 1.64$$

$$\boxed{W_{\text{moon}} = 668.7\ \text{N}}$$

Question: 1.030P

The weight of bodies may change somewhat from one location to another as a result of the variation of the gravitational acceleration *g* with elevation. Accounting for this variation using the relation in Prob. 1–14, determine the weight of an 80-kg person at sea level (z= 0), in Denver (z = 1610 m), and on the top of Mount Everest (z=8848 m).

Step 1 of 5

The mass of a person $m = 80\,kg$

In problem 1-10, the gravitational acceleration as a function of elevation

$$g = a - bz$$

Where $a = 9.807\ m/s^2$

$b = 3.32\times10^{-6}\,s^{-2}$

$g = 9.807 - 3.32\times10^{-6}\times z$

z in m

Step 2 of 5

Weight $W = mg$

Here $W = m\left(9.807 - 3.32\times10^{-6}z\right)$

$W = 80\left(9.807 - 3.32\times10^{-6}z\right)$ .....(1)

Step 3 of 5

At sea level $z = 0$

Weight $W = 80\left(9.807 - 3.32\times10^{-6}\times0\right)$

$\boxed{W = 784.56\ N}$

Step 4 of 5

At Denver $z = 1610\ m$

$W = 80\left(9.807 - 3.32\times10^{-6}\times1610\right)$

$\boxed{W = 784.13\ N}$

Step 5 of 5

At mount Everest $z = 8848\ m$

$W = 80\left(9.807 - 3.32\times10^{-6}\times8848\right)$

$\boxed{= 782.2\ N}$

Question: 1.031P

A man goes to a traditional market to buy a steak for dinner. He finds a 12-oz steak (1 lbm = 16 oz) for $3.15. He then goes to the adjacent international market and finds a 300-g steak of identical quality for $2.95. Which steak is the better buy?

Step 1 of 2

The relation between pounds (lbm), ounces (oz) and kilograms (kg) is given as

$$1\,\text{lbm} = 16\,\text{oz} = 0.45359\,\text{kg}$$

Hence $1\,\text{oz} = \frac{0.45359}{16}\,\text{kg}$

$$12\,\text{oz} = 12 \times \frac{0.45359}{16}\,\text{kg}$$

$$12\,\text{oz} = 12 \times \frac{0.45359}{16}\,\text{kg}$$

$$12\,\text{oz} = 0.34\,\text{kg}$$

In traditional market cost of 0.34 kg of steak = $3.15

$$\text{Cost of 1kg steak} = \frac{(\$3.15)}{0.34\,\text{kg}}$$

$$= \$9.26/\text{kg}$$

Hence cost of 1kg of steak in traditional market is $ 9.26 per kg

Step 2 of 2

In international market, Cost of 300 gm of steak = $ 2.95

300 gram = 0.30 kg

$$\text{Cost of 1 kg steak in international market} = \frac{\$2.95}{0.3\,\text{kg}}$$

$$= \$9.83/\text{kg}$$

Hence cost of 1kg of steak in international market is $ 9.83 per kg

Therefore, the steak at the international market is the better buy.

Question: 1.032P

What is the weight of a 1-kg substance in N, kN, kg·m/s2, kgf, lbm·ft/s2, and lbf?

Step 1 of 2

Given

Mass of substance $m = 1\ \text{kg}$

Weight of substance $W = mg$

$$W = (1\ \text{kg}) \times (9.81\ \text{m/s}^2)$$

$$W = 9.81\ \text{kg} \cdot \text{m/s}^2$$

Hence weight of 1 kg substance in $\text{kg} \cdot \text{m/s}^2$ is $\boxed{W = 9.81\ \text{kg} \cdot \text{m/s}^2}$

But $9.81\ \text{kg} \cdot \text{m/s}^2 = 1\ \text{kgf}$

Hence weight of 1 kg substance in kgf is $\boxed{W = 1\ \text{kgf}}$

Also $1\ \text{kg} \cdot \text{m/s}^2 = 1\ \text{N}$,

Hence weight of 1 kg substance in N is $\boxed{W = 9.81\ \text{N}}$

$1\ \text{N} = 10^{-3}\ \text{kN}$

Hence weight of 1 kg substance in kN is $\boxed{W = 9.81 \times 10^{-3}\ \text{kN}}$

Step 2 of 2

From conversion data,

$1\ \text{kg} = 2.2\ \text{lbm}$

$1\ \text{m} = 3.28\ \text{ft}$

Units of weight can be converted into

$$W = 9.81\ \text{kg} \cdot \text{m/s}^2$$

$$W = 9.81 \times (2.2\ \text{lbm}) \cdot (3.28\ \text{ft})/\text{s}^2$$

$$W = 9.81 \times 2.2 \times 3.28\ \text{lbm} \cdot \text{ft/s}^2$$

$$W = 70.8\ \text{lbm} \cdot \text{ft/s}^2$$

Hence weight of 1 kg substance in $\text{lbm} \cdot \text{ft/s}^2$ is $\boxed{W = 70.8\ \text{lbm} \cdot \text{ft/s}^2}$

But $32.2\ \text{lbm} \cdot \text{ft/s}^2 = 1\ \text{lbf}$

Therefore, $W = 70.8\ \text{lbm} \cdot \text{ft/s}^2 \times \dfrac{1\ \text{lbf}}{32.2\ \text{lbm} \cdot \text{ft/s}^2}$

$$W = 2.2\ \text{lbf}$$

Hence weight of 1 kg substance in lbf is $W = 2.2\ \text{lbf}$

Question: 1.033P

Consider the flow of air through a wind turbine whose blades sweep an area of diameter D(in m). The average air velocity through the swept area is V(in m/s). On the bases of the units of the quantities involved, show that the mass flow rate of air (in kg/s) through the swept area is proportional to air density, the wind velocity, and the square of the diameter of the swept area.

Step 1 of 3

Mass flow rate of air depends on the following

Diameter, D in m

Velocity, V in m/s and

Air density, ρ in kg/m^3

Hence

$$\dot{m} \propto \left(D^a \times V^b \times \rho^c\right)$$

$$\dot{m} = kD^a V^b \rho^c$$

Where k = proportionality constant

Step 2 of 3

Dimension of $\dot{m} = [MT^{-1}]$

Dimension of $D = [L]$

Dimension of $V = [LT^{-1}]$

Dimension of $\rho = [ML^{-3}]$

Dimension of $k = [M^0L^0T^0]$

By replacing the terms in equation with dimensions, we get

$$[MT^{-1}] = [M^0L^0T^0][L]^a[LT^{-1}]^b[ML^{-3}]^c$$

$$[MT^{-1}] = [L]^{a+b-3c}[M]^c[T]^{-b}$$

Step 3 of 3

Comparing powers of M on both sides,

$$c = 1$$

Comparing powers of T on both sides,

$$b = 1$$

Comparing powers of L on both sides,

$$a + b - 3c = 0$$

$$a + 1 - 3 = 0$$

$$a = 2$$

Hence by substituting the values of a, b, c we get

$$\dot{m} \propto \left(D^2 \times V \times \rho\right)$$

So, the formula becomes $\boxed{\dot{m} \propto \left(D^2 \times V \times \rho\right)}$

Question: 1.034P

The drag force exerted on a car by air depends on a dimensionless drag coefficient, the density of air, the car velocity, and the frontal area of the car. That is, *F*D =function (CDrag *A*front *ρ*, *V*).Based on unit considerations alone, obtain a relation for the drag force.



FIGURE P1-34

Step 1 of 3

Drag force depends on

Drag coefficient, C_{Drag} (Dimensionless)

Frontal Area, A (m^2)

Air density, ρ $\left(kg/m^3\right)$

Velocity V (m/s)

Hence

$F_D \propto \left(C_{drag}, A_{front}, \rho, V\right)$

$F_D = C_{drag} A^a \rho^b V^c$

Step 2 of 3

Dimension of $F_D = [MLT^{-2}]$

Dimension of $A = [L^2]$

Dimension of $\rho = [ML^{-3}]$

Dimension of $V = [LT^{-1}]$

Dimension of $C_{Drag} = [M^0 L^0 T^0]$

By replacing the terms in equation with dimensions, we get

$[MLT^{-2}] = [M^0 L^0 T^0][L^2]^a [ML^{-3}]^b [LT^{-1}]^c$

$[MLT^{-2}] = M^b [L]^{2a-3b+c} [T]^{-c}$

Step 3 of 3

Comparing powers of M on both sides,

$b = 1$

Comparing powers of T on both sides,

$c = 2$

Comparing powers of L on both sides,

$2a - 3b + c = 1$

$\Rightarrow 2a = 2$

$a = 1$

Hence by substituting the values of a, b, c we get

$F_{Drag} = C_{Drag}(A_{Front})(\rho)(V)^2$

So, the formula becomes $\boxed{F_{Drag} = C_{Drag}(A_{Front})(\rho)(V)^2}$

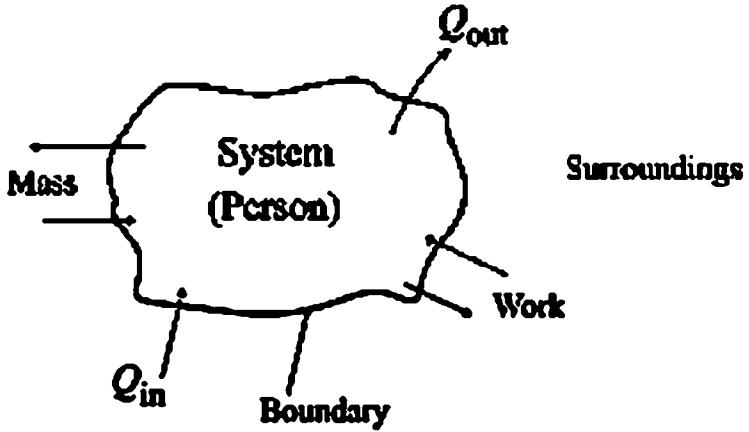
CHAPTER 02

SOLUTIONS

Question: 2.001P

You have been asked to do a metabolism (energy) analysis of a person. How would you define the system for this purpose? What type of system is this?

Step 1 of 2



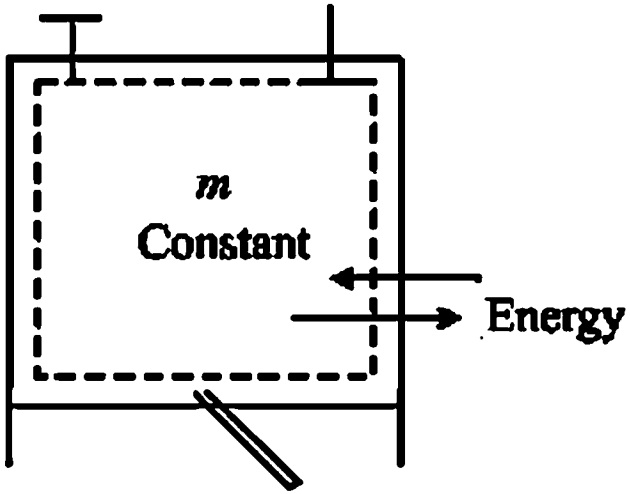
Step 2 of 2

If we doe the energy analysis for a person there will be heat (Q) in and out, work (w) in and out, mass (m) in and out. So this type of system is said to be open system
There will be both mass and energy transfer

Question: 2.002P

You are trying to understand how a reciprocating air compressor (a piston-cylinder device) works. What system would you use? What type of system is this?

Step 1 of 2



Step 2 of 2

In this type of system there will be energy transfer but no mass transfer, such type of system is called as closed system.
In this system air (refrigerant) is compressed by a pump which means work input is given so that it leads to increase in pressure/temperature.

Question: 2.003P

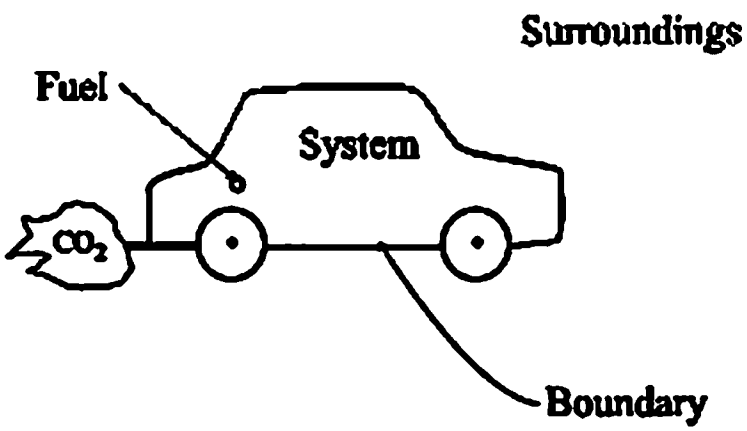
How might you define a system to study the depletion of ozone in the upper layers of the earth's atmosphere?

Step 1 of 3

Generation of CO_2, nitrogen oxide and hydrocarbons are the major cause of depletion of ozone layer.

Carbon dioxide, nitrogen oxide and hydrocarbons are generated by the combustion of fuel in the engine. Any system selected for this analysis must include the fuel and air while it is going combustion. The volume that contains the air-fuel mixture within piston-cylinder device can be used for this purpose.

Step 2 of 3



Step 3 of 3

One can also place the entire engine in a control boundary and trace the system surrounding interactions to determine the rate at which engine generates CO_2, nitrogen oxide and hydrocarbons by using the volumetric analysis. This gaseous produced due to unburned fuel in the combustion.

Nitrogen oxide and hydrocarbons react in the presence of sunlight on hot claim days to form smog which cause ground level ozone deflection. Since these gases have no boundary and it cross the entire global areas.

Question: 2.004P

What is the difference between intensive and extensive properties?

Step 1 of 1

Intensive properties	Extensive properties
It is independent of the mass of the system Example: Pressure, Temperature and density.	It is dependent on mass of the system Example:– total mass, Volume, and total momentum.

Question: 2.005P

Is the weight of a system an extensive or intensive property?

Step 1 of 1

Weight of a system = mass × acceleration due to gravity
So weight of system depends on mass and mass is an extensive property
So weight of a system is extensive property. Extensive properties are those whose values depend on the size or extent of the system. Because of its value changes with changes of the mass.

Question: 2.006P

The molar specific volume of a system $\bar{v}$ is defined as the ratio of the volume of the system to the number of moles of substance contained in the system. Is this an extensive or intensive property?

Step 1 of 2

$$\bar{v} = \frac{\text{volume of system}}{\text{number of mole}}$$

The number of moles of a substance in a system is directly proportional to the number of atomic particle contained in the system. If we divided the system into smaller portion, each portion contains fewer atomic particles than the original system. Therefore number moles are the extensive property.

And Volume also extensive property because of whose values depends on the size or extent of the system.

Step 2 of 2

An extensive property divided by extensive property is always intensive property.

Also we can state that all specific properties are intensive properties. Or extensive properties per unit mass are called specific properties.

So molar specific volume of a system $\bar{v}$ is an intensive property.

Question: 2.007P

For a system to be in thermodynamic equilibrium, do the temperature and the pressure have to be the same everywhere?

Step 1 of 1

If the temperature and pressure are same at every point, the system satisfies thermal and mechanical equilibrium respectively.

For a system to be in thermodynamic equilibrium, the temperature remains constant but the pressure may vary.

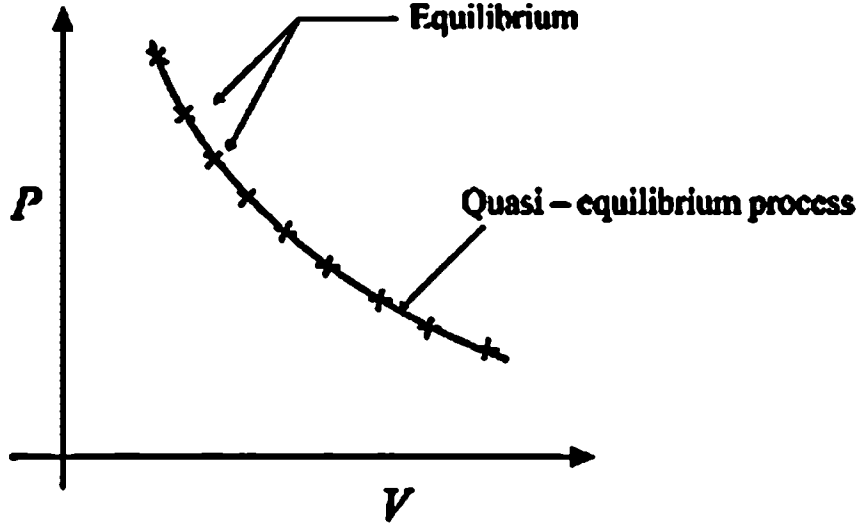
Question: 2.008P

What is a quasi-equilibrium process? What is its importance in engineering?

Step 1 of 3

Quasi – equilibrium process:–
The departure of the state of the system from the thermodynamic equilibrium state will be infinitesimally small. So every state passed through by the system will be an equilibrium state. Such a process, which is but a locus of all the equilibrium points passed through by the system is known as a quasi – equilibrium process. 'Quasi' meaning 'almost'

Step 2 of 3



Step 3 of 3

Quasi equilibrium process are easy to analyze and work – producing devices deliver the most work, so engineering applications can be approximated to quasi – equilibrium process.

Question: 2.009P

Define the isothermal, isobaric, and isochoric processes.

Step 1 of 1

Isothermal process:–

A process during which the temperature remains constant is called isothermal process

Isobaric process:–

A process during which the pressure remains constant is called isobaric process

Isochoric process:–

A process during which the volume remains constant is called isochoric process.

Question: 2.010P

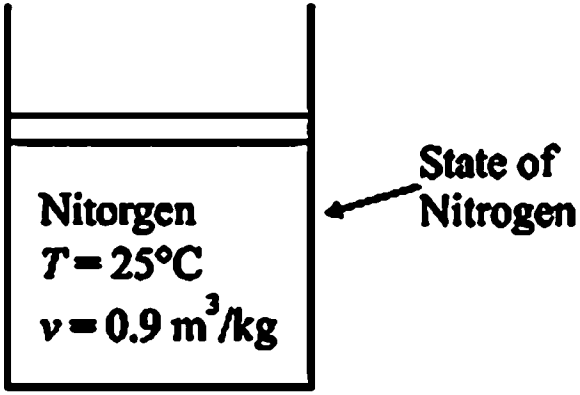
What is the state postulate?

Step 1 of 2

The number of properties to fix the state of a system is given by the state postulate. The state postulate requires that the two properties specified be independent, intensive properties.

For example:– Nitrogen in a container is defined by temperature $T = 25°C$, specific volume $v = 0.9 m^3/kg$

Step 2 of 2



Question: 2.011P

How would you describe the state of the water in a bathtub? How would you describe the process that this water experiences as it cools?

Step 1 of 3

The state of water in a bathtub is described by taking its temperature and pressure.

Let us take the temperature of the water as room temperature say $26^{\circ}C$, and atmospheric pressure acts on the water surface $(P_{atm} = 101.325 kPa)$.

From the properties of saturated water

At $P_{atm} = 101.325 kPa$

Saturation temperature is $T_{sat} = 99.99^{\circ}C$

Step 2 of 3

The temperature of water decreases as it cools. So that $T < T_{sat@P}$, we have considered as compressed liquid. So compressed liquid properties depends upon only on the temperature much more strongly than they do on the pressure. Since increasing the pressure 100 times often causes the properties to change less than 1 percent.

Step 3 of 3

As the water cools pressure remains fixed. The cooling process becomes as constant pressure process.

Question: 2.012P

When analyzing the acceleration of gases as they flow through a nozzle, what would you choose as your system? What type of system is this?

Step 1 of 1

When analyzing the acceleration of gases as they flow through a nozzle, we consider the Volume within nozzle as the control surface. Bounded by entire inlet surface and cross section of the nozzle.

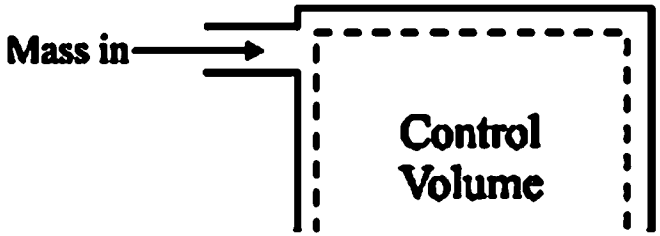
Question: 2.013P

When is a steady-flow process?

Step 1 of 3

The process which involves no change with time anywhere within the system is called steady process.
A large number of engineering devices operate for long periods of time under the same conditions, and they are classified as steady – flow devices. Process involving such devices can be some what idealized process, called the steady – flow process.

Step 2 of 3



Question: 2.014P

The density of atmospheric air varies with elevation, decreasing with increasing altitude. (*a*) Using the data given in the table, obtain a relation for the variation of density with elevation, and calculate the density at an elevation of 7000 m. (*b*) Calculate the mass of the atmosphere using the correlation you obtained. Assume the earth to be a perfect sphere with a radius of 6377 km, and take the thickness of the atmosphere to be 25 km.

z, km	ρ, kg/m3
6377	1.225
6378	1.112
6379	1.007
6380	0.9093
6381	0.8194
6382	0.7364
6383	0.6601
6385	0.5258
6387	0.4135
6392	0.1948
6397	0.08891
6402	0.04008

Step 1 of 4

Given data:

The earth is perfectly sphere with a radius of $r = 6377$ km

The thickness of the atmosphere is $z = 25$ km

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.

$rho = r + z$

After writing the equation in equation window, then select table on menu bar, then click new parameter table.

The dialog box will be appeared, then select the density, radius and elevation, then click ADD, then click OK button.

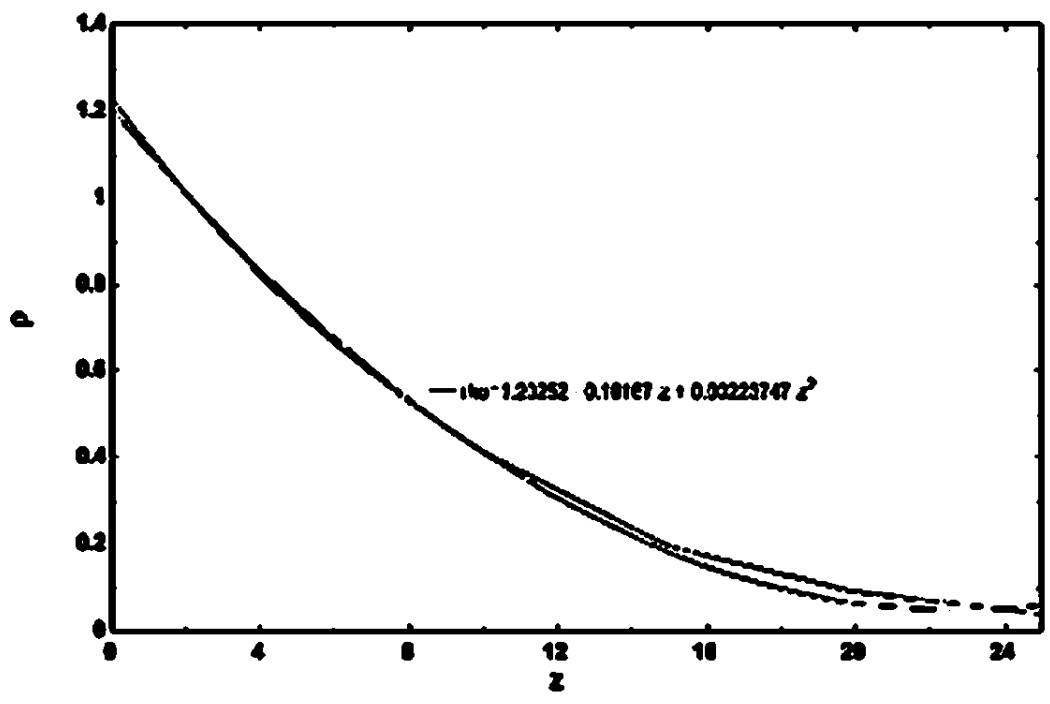
The following results are displayed

1..12	1 ρ	2 z	3 r
Run 1	1.225	0	6377
Run 2	1.112	1	6378
Run 3	1.007	2	6379
Run 4	0.9093	3	6380
Run 5	0.8194	4	6381
Run 6	0.7364	5	6382
Run 7	0.6601	6	6383
Run 8	0.5258	8	6385
Run 9	0.4135	10	6387
Run 10	0.1948	15	6392
Run 11	0.08891	20	6397
Run 12	0.04008	25	6402

Step 2 of 4

Select the plot on menu bar, then click x-y plot .Dialog box will appeared, then select variables to draw graph, then click ok button.

The graph will appear then click on curve fit and specify 2nd order polynomial and enter to get polynomial equation.



Step 3 of 4

a)

The second order polynomial equation is

$$\rho(z) = \left(1.20252 - 0.10167z + 0.00223747z^2\right) \text{ kg/m}^3$$

$$\rho(z) = \left(1.20252 - 0.10167z + 0.00223747z^2\right) \times 10^9 \text{ kg/km}^3 \quad \ldots\ldots (1)$$

At elevation of $z = 7$ m the density is given by

$$\rho = 1.20252 - 0.10167z + 0.00223747z^2$$

$$= 1.20252 - 0.10167(7) + 0.00223747(7)^2$$

$$= 1.20252 - 0.71169 + 0.10963$$

$$= 0.60046 \text{ kg/m}^3$$

The density at elevation $z = 7$ m is $\boxed{\rho = 0.60046 \text{ kg/m}^3}$

Step 4 of 4

b)

Volume of the sphere is given by

$$dV = 4\pi(r_o + z)^2 dz \quad \ldots\ldots (2)$$

The mass of atmosphere having the thickness $h = 0$ to 25 m and radius of the sphere $r = 6377$ km using the correlation is given by

$$m = \int_0^h \rho dV \quad \ldots\ldots (3)$$

Now by substituting the above equations (1) and (2) in (3) we get,

$$m = \int_0^{25} \left[\left(1.20252 - 0.10167z + 0.00223747z^2\right) 4\pi(6377 + z)^2 \times 10^3\right] dz$$

On solving the above equation we get the mass of the atmosphere,

$$m = 5.09234 \times 10^{18} \text{ kg}$$

Therefore, the mass of the atmosphere is $\boxed{m = 5.09234 \times 10^{18} \text{ kg}}$.

Question: 2.015P

What is the zeroth law of thermodynamics?

Step 1 of 1

When a body A is in thermal equilibrium with a body B, and also separately with abody C, then B and C will be in thermal equilibrium with each other. This is known as the Zeroth Law of Thermodynamics
It is used for temperature measurement

Question: 2.016P

What are the ordinary and absolute temperature scales in the SI and the English system?

Step 1 of 1

In SI system:–

Ordinary temperature – Celsius (°C)

Absolute temperature – Kelvin (K)

English system:–

Ordinary temperature – Fahrenheit (°F)

Absolute temperature – Rankine (R)

Question: 2.017P

Consider an alcohol and a mercury thermometer that read exactly 0°C at the ice point and 100°C at the steam point. The distance between the two points is divided into 100 equal parts in both thermometers. Do you think these thermometers will give exactly the same reading at a temperature of, say, 60°C? Explain.

Step 1 of 1

No, the alcohol and mercury are two different fluids, which will have different properties. Mainly the thermal expansions of fluids will be different for alcohol and mercury, so the thermometers will show different readings at 60°C.

Question: 2.018P

The deep body temperature of a healthy person is 37°C. What is it in kelvins?

Step 1 of 1

Given,

The temperature of a person $T(°C) = 37°C$

The Kelvin scale is related to the Celsius scale by

$$T(K) = T(°C) + 273.15$$

Here $T(K) = 37°C + 273.15$

$$\boxed{T(K) = 310.15K}$$

Question: 2.019P

What is the temperature of the heated air at 150°C in °F and R?

Step 1 of 3

Given:

Temperature of the heated air $T(°C) = 150°C$

Step 2 of 3

The temperature scale in the two unit system are related as

$$T(°F) = 1.8T(°C) + 32$$

$$T(°C) = 150°C$$

$$T(°F) = (1.8 \times 150) + 32$$

$$= 302°F$$

$$\boxed{T(°F) = 302°F}$$

Step 3 of 3

The Rankine scale is related to the Fahrenheit scale is given by

$$T(R) = T(°F) + 459.67$$

$$= 302 + 459.67$$

$$= 761.67R$$

$$\boxed{T(R) = 761.67R}$$

Question: 2.020P

The temperature of a system rises by 45°C during a heating process. Express this rise in temperature in kelvins.

Step 1 of 1

The magnitude of each division of 1K and 1°C are identical .Therefore, when we are dealing with temperature difference, the temperature interval on both scales is the same.

The temperature changes are always identical in Kelvin and Celsius scales.

Rise in temperature in Kelvin is,

$$\Delta T(\mathrm{K}) = \Delta T(^{\circ}\mathrm{C})$$
$$= 45\mathrm{K}$$

Therefore, the rise in temperature in Kelvins is $\boxed{45\,\mathrm{K}}$

Question: 2.021P

The flash point of an engine oil is 363°F. What is the absolute flash-point temperature in K and R?

Step 1 of 4

Given:

Flash point temperature of the engine oil $T(^\circ\text{F}) = 363^\circ\text{F}$

Step 2 of 4

Temperature scales in the two unit system are related as

$$T(^\circ\text{F}) = 1.8T(^\circ\text{C}) + 32$$

$$363 = 1.8T(^\circ\text{C}) + 32$$

$$\Rightarrow T(^\circ\text{C}) = \frac{363 - 32}{1.8}$$

$$= 183.89^\circ\text{C}$$

Step 3 of 4

Absolute Flash point temperature in Kelvin

$$T(\text{K}) = T(^\circ\text{C}) + 273.15$$

$$= 183.89 + 273.15$$

$$= 457.03\text{K}$$

$$\boxed{T(\text{K}) = 457.03\text{K}}$$

Step 4 of 4

Absolute Flash point temperature in Rankine

$$T(\text{R}) = T(^\circ\text{F}) + 459.67$$

$$= 363 + 459.67$$

$$= 822.67\text{R}$$

$$\boxed{T(\text{R}) = 822.67\text{R}}$$

Question: 2.022P

The temperature of ambient air in a certain location is measured to be −40°C. Express this temperature in Fahrenheit (°F), Kelvin (K), and Rankine (R) units.

Step 1 of 4

Given:

Temperature of ambient air $T(^\circ\text{C}) = -40^\circ\text{C}$

Step 2 of 4

The Kelvin scale is related to the Celsius scale is

$T(\text{K}) = T(^\circ\text{C}) + 273.15$

Temperature in Kelvin scale is

$T(\text{K}) = T(^\circ\text{C}) + 273.15$

$= -40 + 273.15$

$= 233.15\text{K}$

$\boxed{T(\text{K}) = 233.15\text{K}}$

Step 3 of 4

Temperature scales in the two unit system are related as

$T(^\circ\text{F}) = 1.8T(^\circ\text{C}) + 32$

Temperature in Fahrenheit scale is

$T(^\circ\text{F}) = 1.8T(^\circ\text{C}) + 32$

$= [1.8 \times (-40)] + 32$

$= -40^\circ\text{F}$

$\boxed{T(^\circ\text{F}) = -40^\circ\text{F}}$

Step 4 of 4

The Rankine scale related to the Fahrenheit scale is

$T(\text{R}) = T(^\circ\text{F}) + 459.67$

Temperature in Rankine scale is

$T(\text{R}) = T(^\circ\text{F}) + 459.67$

$= -40 + 459.67$

$= 419.67\text{R}$

$\boxed{T(\text{R}) = 419.67\text{R}}$

Question: 2.023P

The temperature of water changes by 10°F during a process. Express this temperature change in Celcius (°C), Kelvin (K), and Rankine (R) units.

Step 1 of 4

Given:

Temperature of water change $\Delta T = 10^\circ F$

Step 2 of 4

A temperature changes in Rankine scale

$$\Delta T(\mathrm{R}) = \Delta T(^\circ\mathrm{F})$$
$$= 10\mathrm{R}$$

Step 3 of 4

A temperature changes in Celsius scale

$$\Delta T(^\circ\mathrm{C}) = \Delta T(\mathrm{K})$$
$$= \frac{10}{1.8}$$
$$= 5.6^\circ\mathrm{C}$$

Step 4 of 4

Temperature changes in Kelvin is

$$\Delta T(\mathrm{K}) = \frac{T(\mathrm{R})}{1.8}$$
$$= \frac{10}{1.8}$$
$$= 5.6\mathrm{K}$$

Question: 2.024P

Humans are most comfortable when the temperature is between 65°F and 75°F. Express these temperature limits in °C. Convert the size of this temperature range (10°F) to K, °C, and R. Is there any difference in the size of this range as measured in relative or absolute units?

Step 1 of 9

Convert the temperature $65°F$ from degree Fahrenheit to degree Celsius.

$$T_1(°F) = 1.8 \times T_1(°C) + 32$$

Here, $T(°C)$ is the temperature in degree Celsius and $T(°F)$ is the temperature in degree Fahrenheit.

Substitute $65°F$ for $T_1(°F)$.

$$65 = 1.8 \times T_1(°C) + 32$$
$$T_1(°C) = 18.33°C$$

Therefore, the conversion of temperature from degree Fahrenheit to degree Celsius for $65°F$ is $\boxed{18.33°C}$.

Step 2 of 9

Convert the temperature $75°F$ from degree Fahrenheit to degree Celsius.

$$T_2(°F) = 1.8 \times T_2(°C) + 32$$

Substitute $75°F$ for $T_2(°F)$.

$$75 = 1.8 \times T_2(°C) + 32$$
$$T_2(°C) = 23.89°C$$

Therefore, the temperature limits from degree Fahrenheit to degree Celsius for $75°F$ is $\boxed{23.89°C}$.

Step 3 of 9

Convert the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Celsius.

$$(\Delta T) = T_2(°C) - T_1(°C)$$

Substitute $23.89°C$ for $T_2(°C)$. $18.33°C$ for $T_1(°C)$.

$$(\Delta T) = 23.89 - 18.33$$
$$= 5.56°C$$
$$\approx 5.6°C$$

Therefore, the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Celsius is $\boxed{5.6°C}$.

Step 4 of 9

Convert the temperature $65°F$ from degree Fahrenheit to degree Kelvin.

$$T_1(K) = (T_1(°F) - 32) \times \frac{5}{9} + 273.15$$

Substitute $65°F$ for $T_1(°F)$.

$$T_1(K) = (65 - 32) \times \frac{5}{9} + 273.15$$
$$= 291.483\ K$$

Therefore, the conversion of the temperature $65°F$ from degree Fahrenheit to degree Kelvin for $65°F$ is $\boxed{291.483K}$.

Step 5 of 9

Convert the temperature $75°F$ from degree Fahrenheit to degree Celsius.

$$T_2(K) = (T_2(°F) - 32) \times \frac{5}{9} + 273.15$$

Substitute $75°F$ for $T_2(°F)$.

$$T_2(K) = (75 - 32) \times \frac{5}{9} + 273.15$$
$$= 297.038\ K$$

Therefore, the conversion of the temperature $65°F$ from degree Fahrenheit to degree Kelvin for $75°F$ is $\boxed{297.038\ K}$.

Step 6 of 9

Convert the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Kelvin.

$$(\Delta T) = T_2(K) - T_1(K)$$

Substitute $297.038\ K$ for $T_2(K)$. $291.483\ K$ for $T_1(K)$.

$$(\Delta T) = 297.083 - 291.483$$
$$= 5.6K$$

Therefore, the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Kelvin is $\boxed{5.6\ K}$

Step 7 of 9

Convert the temperature $65°F$ from degree Fahrenheit to degree Rankine.

$$T_1(°F) = (T_1(°R) - 459.67)$$

Substitute $65°F$ for $T_1(°F)$.

$$65 = (T_1(°R) - 459.67)$$
$$T_1(°R) = 524.67°R$$

Therefore, the conversion of the temperature $65°F$ from degree Fahrenheit to degree Rankine for $65°F$ is $\boxed{524.67°R}$.

Step 8 of 9

Convert the temperature $75°F$ from degree Fahrenheit to degree Rankine.

$$T_2(°F) = (T_2(°R) - 459.67)$$

Substitute $75°F$ for $T_2(°F)$.

$$75 = (T_2(°R) - 459.67)$$
$$T_2(°R) = 534.67°R$$

Therefore, the conversion of the temperature $75°F$ from degree Fahrenheit to degree Rankine for $75°F$ is $\boxed{534.67°R}$.

Step 9 of 9

Convert the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Rankine.

$$(\Delta T) = T_2(°R) - T_1(°R)$$

Substitute $524.67°R$ for $T_2(°R)$. $534.67°R$ for $T_2(°R)$.

$$(\Delta T) = 534.67 - 524.67$$
$$= 10°R$$

Therefore, the size of this temperature range $(65°F \text{ to } 75°F)$ from degree Fahrenheit to degree Rankine is $\boxed{10°R}$.

No difference found between relative and absolute units (within same system).

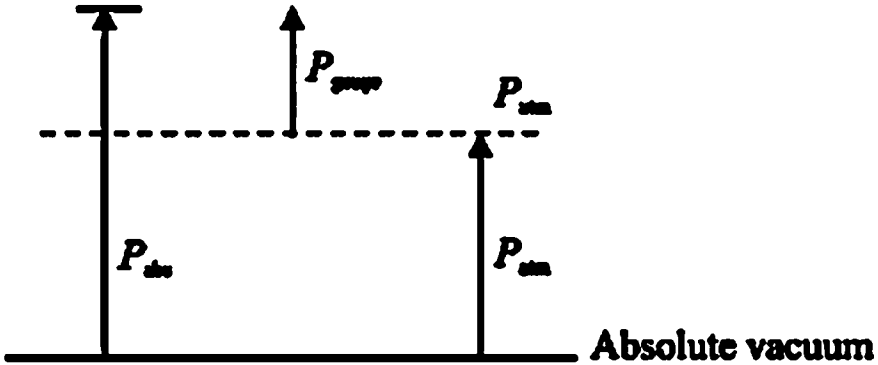
Question: 2.025P

What is the difference between gage pressure and absolute pressure?

Step 1 of 2

The pressure relative to the atmosphere is called gauge pressure
The pressure relative to a perfect Vacuum is called absolute pressure
Absolute pressure = Gauge pressure + Atmospheric pressure

Step 2 of 2



Question: 2.026P

A health magazine reported that physicians measured 100 adults' blood pressure using two different arm positions: parallel to the body (along the side) and perpendicular to the body (straight out). Readings in the parallel position were up to 10 percent higher than those in the perpendicular position, regardless of whether the patient was standing, sitting, or lying down. Explain the possible cause for the difference.

Step 1 of 1

When a person spreading his arms in t he parallel position will experience more strain i.e., pressure compared to the perpendicular position of arms.

The pressure exerted by the heart to pump the blood will be more in this case compared to the perpendicular position of arms. It is invariant of the position of a person whether sitting, lying or standing.

Since the parallel position the arms are in up position.

Question: 2.027P

Someone claims that the absolute pressure in a liquid of constant density doubles when the depth is doubled. Do you agree? Explain.

Step 1 of 3

The pressure at a depth 'h' from the free surface

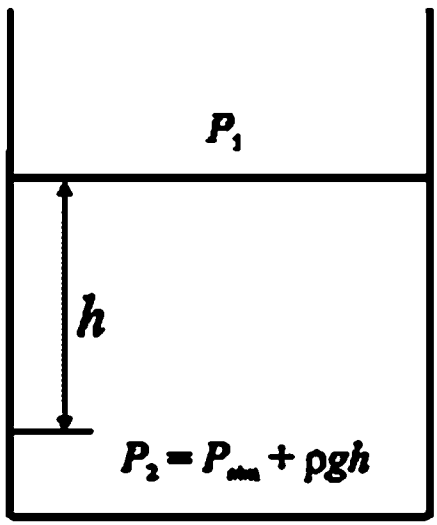
$P = P_{atm} + \rho g h$

$P_{gauge} = \rho g h$,

P_{atm} = atmospheric pressure

P = absolute pressure

Step 2 of 3



Step 3 of 3

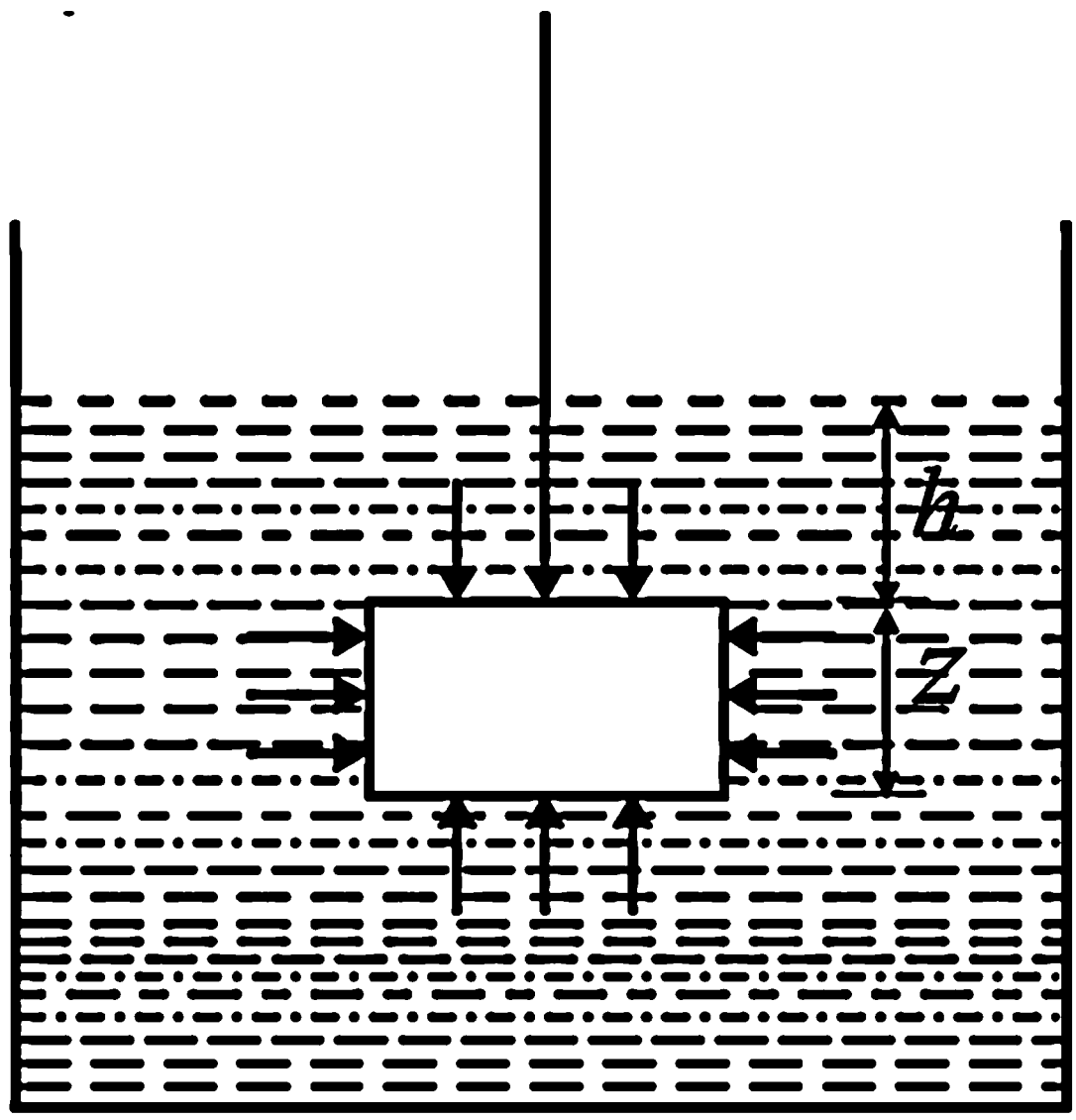
If the depth is double the gauge pressure gets double not the absolute pressure.
No, this statement is false.

Question: 2.028P

A tiny steel cube is suspended in water by a string. If the lengths of the sides of the cube are very small, how would you compare the magnitudes of the pressures on the top, bottom, and side surfaces of the cube?

Step 1 of 2

Draw the schematic diagram of a tiny steel cube in a beaker of water.



Step 2 of 2

Calculate the pressure at the top side of the cube.

$$P = P_{atm} + \rho g H$$

Here, atmospheric pressure is P_{atm}, density of the liquid is ρ, acceleration due to gravity is g and distance from the surface of the liquid is H.

Substitute h for H in the above equation.

$$P = P_{atm} + \rho g h \quad \text{..... (1)}$$

Calculate the pressure at the bottom side of the cube.

$$P = P_{atm} + \rho g H$$

Here, atmospheric pressure is P_{atm}, density of the liquid is ρ, acceleration due to gravity is g and distance from the surface of the liquid is H.

Substitute $(h+z)$ for H in the above equation.

$$P = P_{atm} + \rho g (h+z) \quad \text{..... (2)}$$

Neglect z in equation (2) by substituting 0 for z.

$$P = P_{atm} + \rho g h$$

The above equation is similar to equation (1).

Therefore, if the length of the cube is neglected, then the cube will experience same pressure on all sides of the cubes.

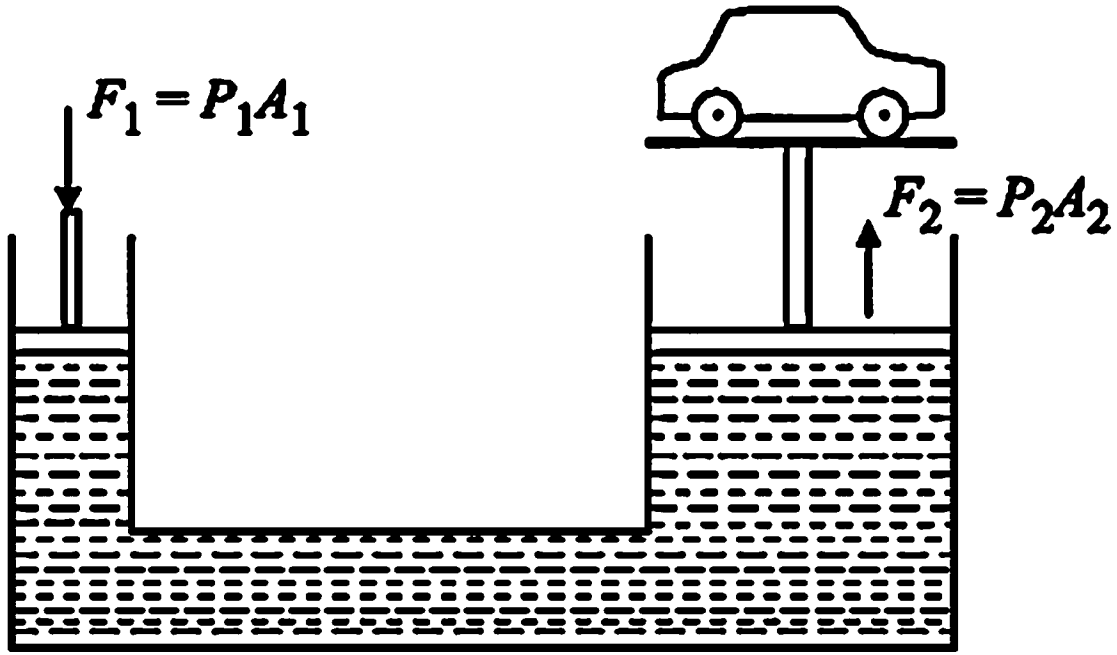
Question: 2.029P

Express Pascal's law, and give a real-world example of it.

Step 1 of 3

Pascal's law states that the pressure applied to a confined fluid increases the pressure throughout the same amount. The pressure in a fluid remaining constant in the horizontal direction.
The Pascal's law used in our daily lives such as hydraulic brakes and lifts

Step 2 of 3



Step 3 of 3

Lifting of a large weight by a small force by the application of Pascal's law

Question: 2.030P

The pressure at the exit of an air compressor is 150 psia. What is this pressure in kPa?

Step 1 of 2

Given:

Pressure exit of the compressor $P = 150\text{psia}$

Step 2 of 2

We know that

$14.7\text{psi} = 101.3\text{kPa}$

$1\text{psi} = 6.891\text{kPa}$

$150\text{psi} = 150 \times 6.891\text{kPa}$

$= 1033.67\text{kPa}$

$\boxed{P = 1033.67\text{kPa}}$

Question: 2.031P

The pressure in a compressed air storage tank is 1500 kPa. What is the tank's pressure in (*a*) kN and m units; (*b*) kg, m, and s units; and (*c*) kg, km, and s units?

Step 1 of 5

Given pressure in a compressed tank

$P = 1500\,\text{kPa}$

Step 2 of 5

a) In $\text{k}\cdot\text{N}$ and m units

$1\text{pascal} = 1\ \text{N/m}^2$

$\therefore\ \boxed{P = 1500\,\text{kN/m}^2}$

Step 3 of 5

b) In kg, m and s units

$1\,\text{Pa} = 1\text{N/m}^2$

$1\text{N} = 1\text{kg}\cdot\text{m/s}^2$

$$P = 1500 \times 1000 \times \frac{\text{kg}}{s^2} \times \frac{1}{m}$$

Pressure $\boxed{P = 1{,}500{,}000\ \text{kg/ms}^2}$

Step 4 of 5

c) In kg, m and s units

$$P = 1500{,}000 \frac{\text{kg}}{\frac{m \times s^2}{10^3}}$$

$1\text{km} = 1000\,\text{m}$

$$\boxed{P = 1500{,}000 \times 10^3 \frac{\text{kg}}{\text{km}\cdot s^2}}$$

Step 5 of 5

Question: 2.032P

The pressure in a water line is 1500 kPa. What is the line pressure in (*a*) lb/ft2 units and (*b*) lbf/in2 (psi) units?

Step 1 of 2

a) We know that

$$1\,\text{kPa} = 20.8854\,\text{lbf/ft}^2$$

$$1500\,\text{kPa} = 1500 \times 20.8854\,\text{lbf/ft}^2$$

$$= 31328.153\,\text{lbf/ft}^2$$

Step 2 of 2

(b) We know that

$$1\text{kPa} = 0.145041\,\text{lbf/in}^2$$

$$1500\text{kPa} = 1500 \times 0.145041\,\text{lbf/in}^2$$

$$= 217.56\,\text{lbf/in}^2$$

Question: 2.033P

If the pressure inside a rubber balloon is 1500 mm Hg, what is this pressure in pounds-force per square inch (psi)?

Step 1 of 3

Given:
Pressure inside rubber balloon $P = 1500\,\text{mm of Hg}$

Step 2 of 3

$$\text{Pressure} = 1500\text{mm of Hg} = 1.5\text{m of Hg}$$

$$\text{Taking, density of Hg} = 13570\ \frac{\text{kg}}{\text{m}^3}$$

$$\begin{aligned}\text{Pressure} &= \rho \times g \times h \\ &= 13570 \times 9.81 \times 1.5 \\ &= 199.682\,\text{kPa}\end{aligned}$$

Step 3 of 3

$$6.88\text{kPa} = 1\text{psi}$$

$$1\text{kPa} = \frac{1}{6.88}\text{psi}$$

$$\begin{aligned}199.682\text{kPa} &= \frac{199.682}{6.88}\text{psi} \\ &= 29\text{psi}\end{aligned}$$

$$\boxed{P = 29\text{psi}}$$

Question: 2.034P

The pressure of the helium inside a toy balloon is 1250 mm Hg. What is this pressure in kPa?

Step 1 of 2

Given:

Pressure inside balloon $P = 1250 \text{ mm of Hg}$

$= 1.25 \text{ m of Hg}$

Step 2 of 2

Density of mercury

$\rho = 13570 \frac{\text{kg}}{\text{m}^3}$

$P = \rho \times g \times h$

$= 13570 \times 9.81 \times 1.25$

$= 166.402 \text{kPa}$

Question: 2.035P

The water in a tank is pressurized by air, and the pressure is measured by a multifluid manometer as shown in Fig. P2-35. Determine the gage pressure of air in the tank if $h1$ = 0.2 m, $h2$ = 0.3 m, and $h3$=0.46 m. Take the densities of water, oil, and mercury to be 1000 kg/m3, 850 kg/ml and 13,600 kg/m3, respectively.

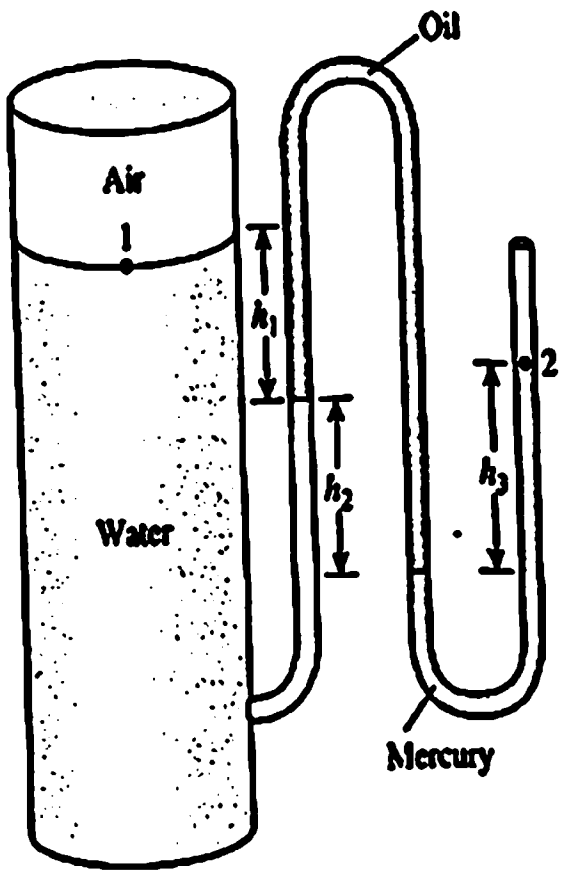
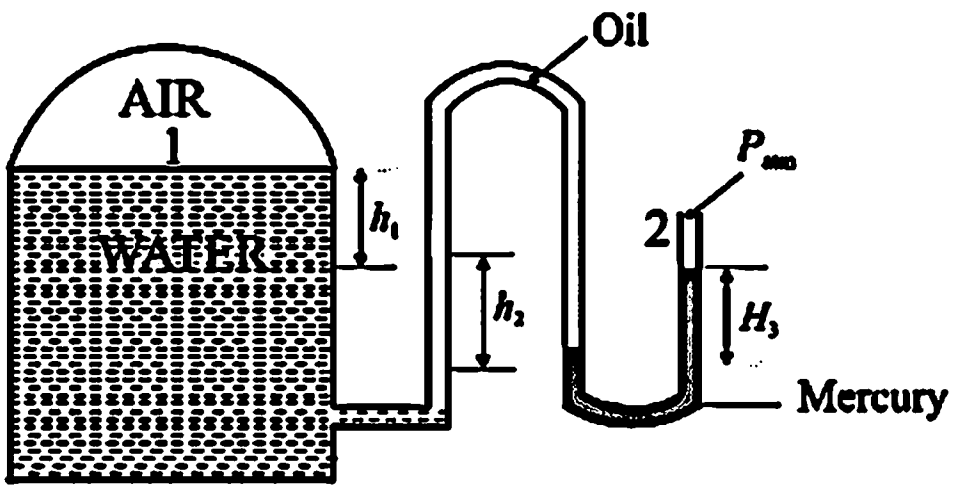


FIGURE P2-35

Step 1 of 4



Step 2 of 4

Let us consider the pressure at air – water enter phase is P_1

The pressure at point 2 is open to atmosphere i.e., $P_2 = P_{atm}$

The gauge pressure of air in tank is given by $P_{gauge} = P_1 - P_{atm}$ .....(1)

(Obtained from basic pressure relation $P_{abs} = P_{gauge} + P_{atm}$

The pressure inside tank can be determined by starting at air – water surface P_1, moving along the tube until we reach free surface, i.e., P_{atm}

Step 3 of 4

It gives

$$P_1 + \rho_{H_2O}\, g h_1 + \rho_{oil}\, g h_2 - \rho_{Hg}\, g h_3 = P_{atm}$$

$$\Rightarrow P_1 - P_{atm} = g\left(\rho_{Hg} h_3 - \rho_{oil} h_2 - \rho_{H_2O} h_1\right) \quad \text{.....(2)}$$

From equations (1) & (2)

$$P_{gauge} = g\left(\rho_{Hg} h_3 - \rho_{oil} h_2 - \rho_{H_2O} h_1\right) \quad \text{.....(3)}$$

Step 4 of 4

Given data

The height of water $h_1 = 0.2\,\text{m}$

The density of water $\rho_{water} = \rho_{H_2O} = 1000\,\text{kg/m}^3$

The height of water $h_2 = 0.3\,\text{m}$

The density of oil $\rho_{oil} = 850\ \text{kg/m}^3$

The height of mercury $h_3 = 0.46\,\text{m}$

The density of mercury $\rho_{mercury} = \rho_{Hg} = 13,600\,\text{kg/m}^3$

Substituting all these values in equation (3)

$$\therefore P_{gauge} = 9.81(13,600 \times 0.46 - 850 \times 0.3 - 1000 \times 0.2)$$

$$= 56907\,\text{Pa}$$

$$\boxed{P_{gauge} = 56.907\,\text{kPa}}$$

Substituting all these values in equation (3)

$$\therefore P_{gauge} = 9.81(13,600 \times 0.46 - 850 \times 0.3 - 1000 \times 0.2)$$

$$= 56907\,\text{Pa}$$

$$\boxed{P_{gauge} = 56.907\,\text{kPa}}$$

Question: 2.036P

Determine the atmospheric pressure at a location where the barometric reading is 750 mm Hg. Take the density of mercury to be 13,600 kg/m3.

Step 1 of 2

Given the barometric reading 750 mm Hg

The barometer measures the atmospheric pressure directly

So, the height of mercury column is $h = 750\,\text{mm}$

From the hydrostatic equation $P = \rho g h$ ……(1)

Here $P_{atm} = \rho_{mercury}\, g h$

$\rho_{mercury} = 13,600\ \text{kg/m}^3$

$g = 9.81\,\text{m/s}^2$

Step 2 of 2

Substitute in above equation (1)

$$P_{atm} = 13,600 \times 9.81 \times 750 \times 10^{-3}$$

$$1\text{m} = 1000\,\text{mm}$$

$$= 100,062\ \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{m}}{\text{s}^2} \cdot \text{m}$$

$$= 100.1 \times 10^3 \times \text{N/m}^2$$

$$\frac{1\text{kg} \cdot \text{m}}{\text{s}^2} = 1\text{N}$$

$$\boxed{P_{atm} = 100.1\text{kPa}}$$

Substitute in above equation (1)

$$P_{atm} = 13,600 \times 9.81 \times 750 \times 10^{-3}$$

$$1\text{m} = 1000\,\text{mm}$$

$$= 100,062\ \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{m}}{\text{s}^2} \cdot \text{m}$$

$$= 100.1 \times 10^3 \times \text{N/m}^2$$

$$\frac{1\text{kg} \cdot \text{m}}{\text{s}^2} = 1\text{N}$$

$$\boxed{P_{atm} = 100.1\text{kPa}}$$

Question: 2.037P

The gage pressure in a liquid at a depth of 3 m is read to be 42 kPa. Determi ne the gage pressure in the same liquid at a depth of 9 m.

Step 1 of 3

Given:

Depth of the liquid $h_1 = 3\text{m}$

Pressure of the liquid $P_1 = 42\text{kPa}$

Depth of the liquid $h_2 = 9\text{m}$

Step 2 of 3

Pressure at a certain depth 'h' can be expressed as

$P = \rho \times g \times h$

Where

$\rho =$ Density of the liquid

$g =$ Acceleration due to gravity

Since liquid is same so that density and acceleration due to gravity of the liquid does not change or its remains constant.

$P_1 = \rho \times g \times h_1 \ldots\ldots\ldots\ldots (1)$

$P_2 = \rho \times g \times h_2 \ldots\ldots\ldots\ldots (2)$

Step 3 of 3

Dividing equation (1) by (2)

$$\frac{P_1}{P_2} = \frac{\rho \times g \times h_1}{\rho \times g \times h_2}$$

$$= \frac{h_1}{h_2}$$

$$\Rightarrow P_2 = \frac{P_1 \times h_2}{h_1}$$

$$= \frac{42 \times 9}{3}$$

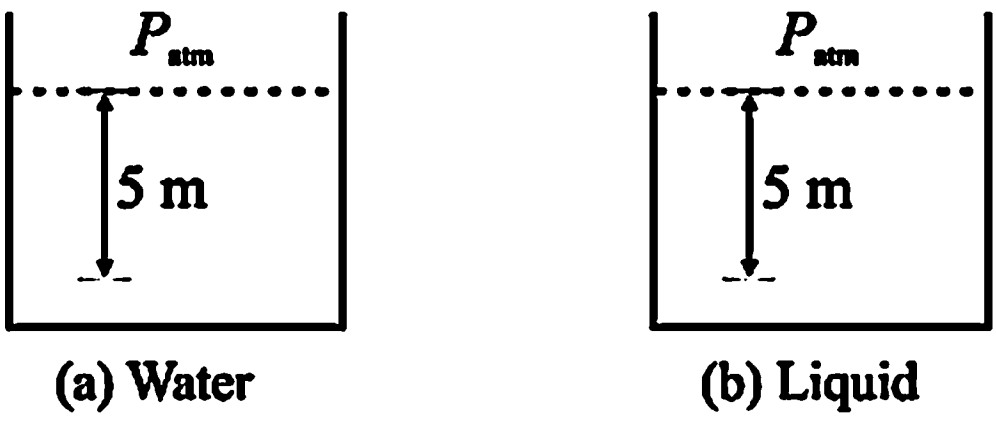
$$= 126\text{kPa}$$

$$\boxed{P_2 = 126\text{kPa}}$$

Question: 2.038P pg 1

The absolute pressure in water at a depth of 5 m is read to be 145 kPa. Determine (*a*) the local atmospheric pressure, and (*b*) the absolute pressure at a depth of 5 m in a liquid whose specific gravity is 0.85 at the same location.

Step 1 of 6

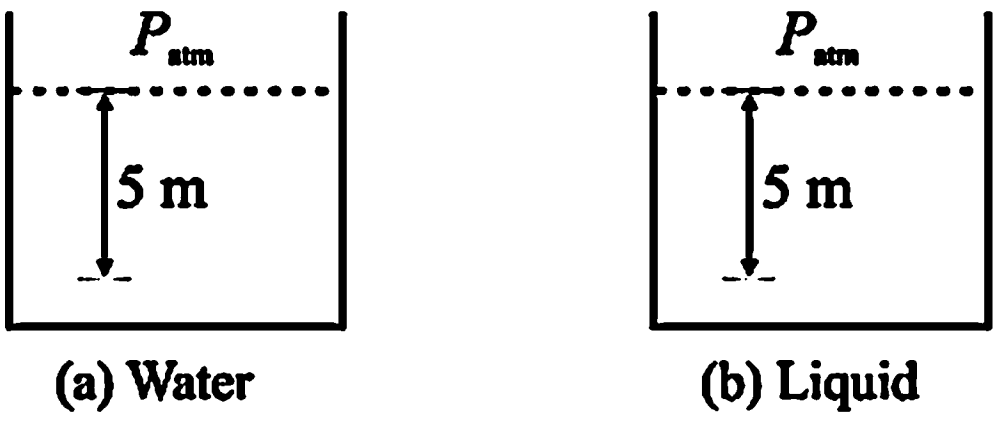


(a) Water (b) Liquid

Question:

The absolute pressure in water at a depth of 5 m is read to be 145 kPa. Determine (*a*) the local atmospheric pressure, and (*b*) the absolute pressure at a depth of 5 m in a liquid whose specific gravity is 0.85 at the same location.

Step 1 of 6



(a) Water (b) Liquid

Step 2 of 6

Specific gravity is defined as ratio of density of liquid to density of water

$$\text{i.e. } S.G = \frac{\rho_{liquid}}{\rho_{water}}$$

$$\therefore \rho_{liq} = 0.85 \times 1000$$

$$= 850 \text{ kg/m}^3$$

The absolute pressure in this liquid at a depth of 5m

$$P_{abs} = P_{atm} + \rho g h$$

$$= 95,950 + 850 \times 9.81 \times 5$$

$$= 137642 \text{ N/m}^2$$

$$\boxed{P_{abs} = 137.64 \text{ kPa}}$$

Step 5 of 6

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$$\boxed{P_{abs} = 137.64 \text{ kPa}}$$

Step 5 of 6

Step 6 of 6

Step 6 of 6

Question: 2.039P

Show that 1 kgf/cm2 = 14.223 psi.

Step 1 of 1

We have to prove that $1\text{kg f/cm}^2 = 14.223\text{psi}$

In English units

$\text{psi} = 1\,\text{pound} - \text{force per square inch}\ (\text{lbf/in}^2)$

$1\text{kg} = 2.2046\ \text{pounds}(\text{lbm})$

$1\text{kgf} = 2.2046\,\text{lbf}$

$1\text{inch} = 2.54\,\text{cm}$

$$1\text{kgf/cm}^2 = \frac{2.2046\,\text{lbf}}{\frac{1}{(2.54)^2}\text{in}^2}$$

$$1\text{kgf/cm}^2 = 2.2046\times(2.54)^2\,\text{lbf/in}^2$$

$$= 14.223\,\text{lbf/in}^2$$

$\therefore$ $\boxed{1\text{kgf/cm}^2 = 14.223\ \text{psi}}$

Hence proved

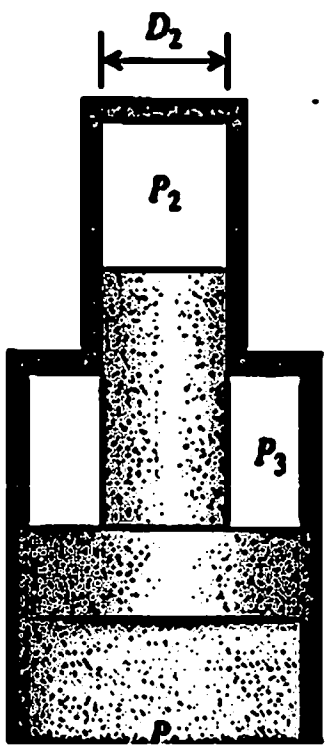
$= 14.223\,\text{lbf/in}^2$

$\therefore$ $\boxed{1\text{kgf/cm}^2 = 14.223\ \text{psi}}$

Hence proved

Question:

The diameters of the pistons shown in Fig. P2-40E are $D1$ = 3 in and $D2$=1.5 in. Determine the pressure in chamber 3, in psia, when the other pressures are $p1$=150 psia and $P2$ = 250 psia.



Question:

The diameters of the pistons shown in Fig. P2-40E are $D1$ = 3 in and $D2$=1.5 in. Determine the pressure in chamber 3, in psia, when the other pressures are $p1$=150 psia and $P2$ = 250 psia.

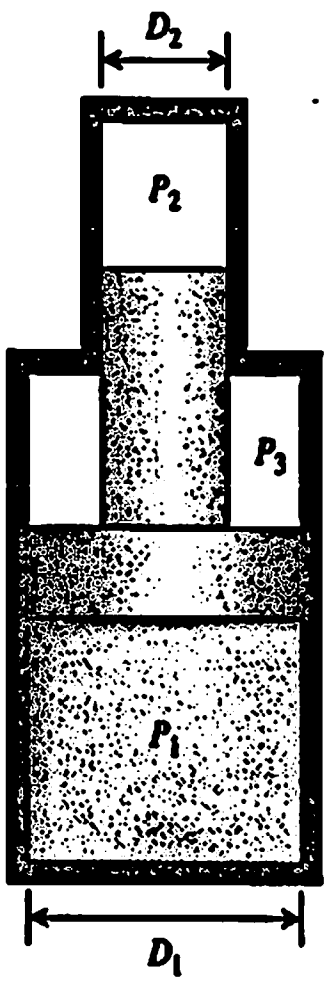


FIGURE P2-40E

Step 1 of 7

Given:

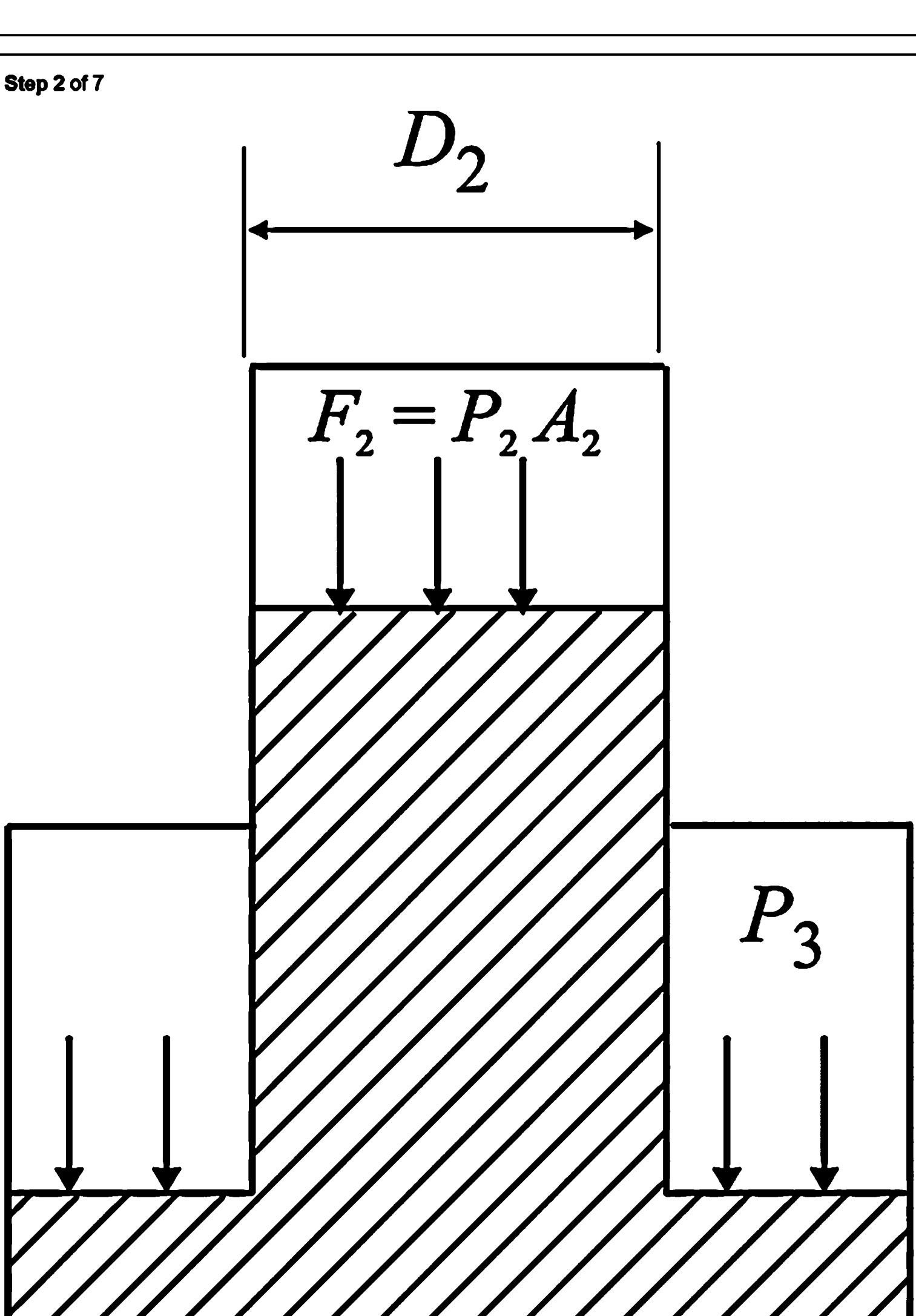
Diameter of the piston 1 $D_1 = 3\,\text{in}$

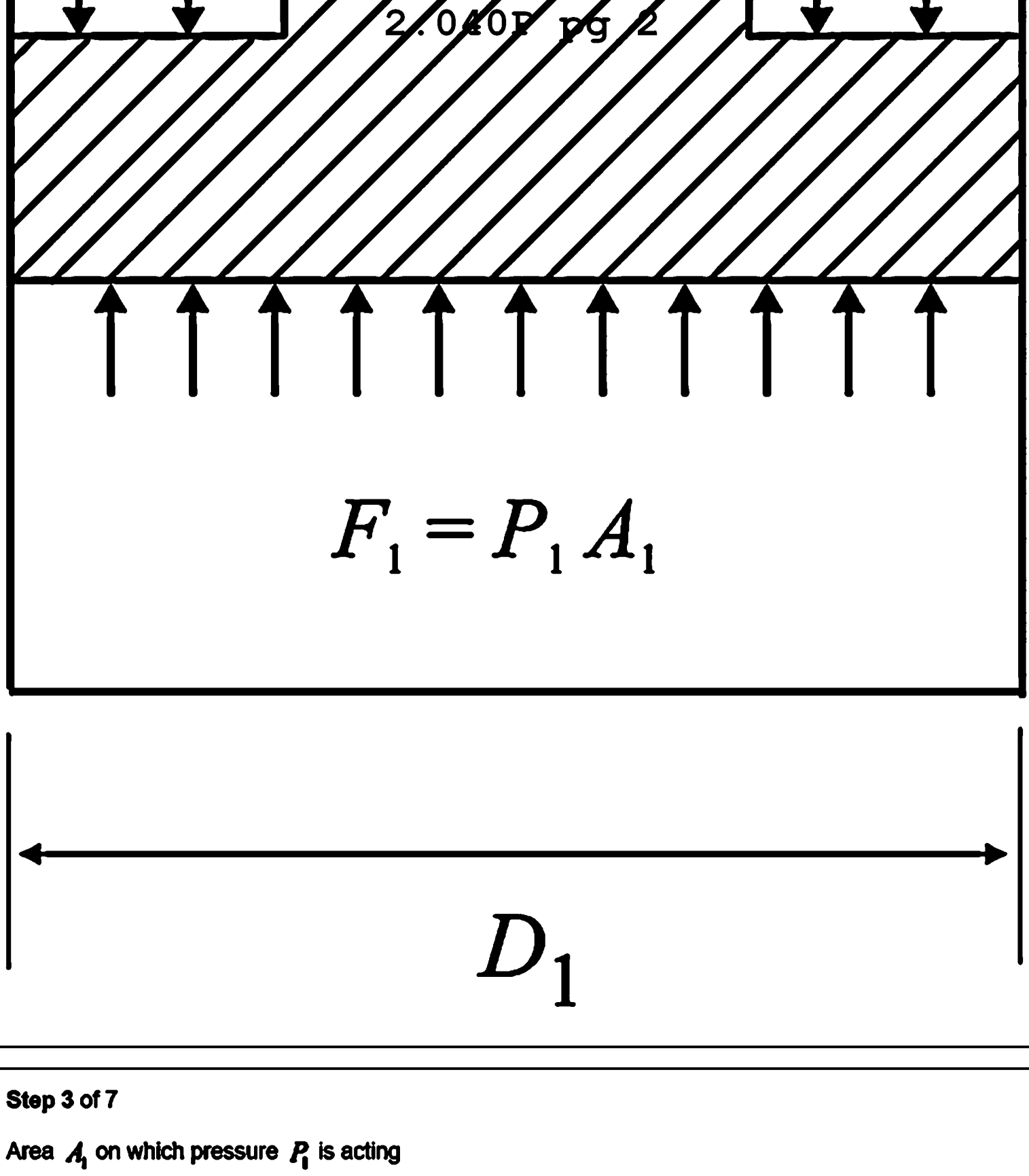
Diameter of the piston 2 $D_2 = 1.5\,\text{in}$

Pressure in chamber 1 $P_1 = 150\,\text{psia}$

Pressure in chamber 1 $P_2 = 250\,\text{psia}$

Step 2 of 7





Step 3 of 7

Area A_1 on which pressure P_1 is acting

$$A_1 = \frac{\pi(D_1^2)}{4}$$

$$= \frac{\pi(3\text{in})^2}{4}$$

$$= 7.069\text{in}^2$$

Step 4 of 7

Force F_1 acting on piston

$$F_1 = P_1 A_1$$

$$= (150\text{psia})(7.069\text{in}^2)\left(\frac{1\frac{\text{lbf}}{\text{in}^2}}{1\text{psia}}\right)$$

$$= 1060.35\text{lbf}$$

Step 5 of 7

Area A_2 on which pressure P_2 is acting

$$A_2 = \frac{\pi(D_2^2)}{4}$$

$$= \frac{\pi(1.5\text{in})^2}{4}$$

$$= 1.76\text{in}^2$$

Force F_2 acting on piston

$$F_2 = P_2 A_2$$

$$= (250\text{psia})(1.76\text{in}^2)\left(\frac{1\frac{\text{lbf}}{\text{in}^2}}{1\text{psia}}\right)$$

$$= 441.78\text{lbf}$$

Step 6 of 7

So, net vertical force acting on the piston in downward direction to balance the piston

$$F_3 = F_1 - F_2$$

$$= (1060.35 - 441.78)\text{lbf}$$

$$= 618.56\text{lbf}$$

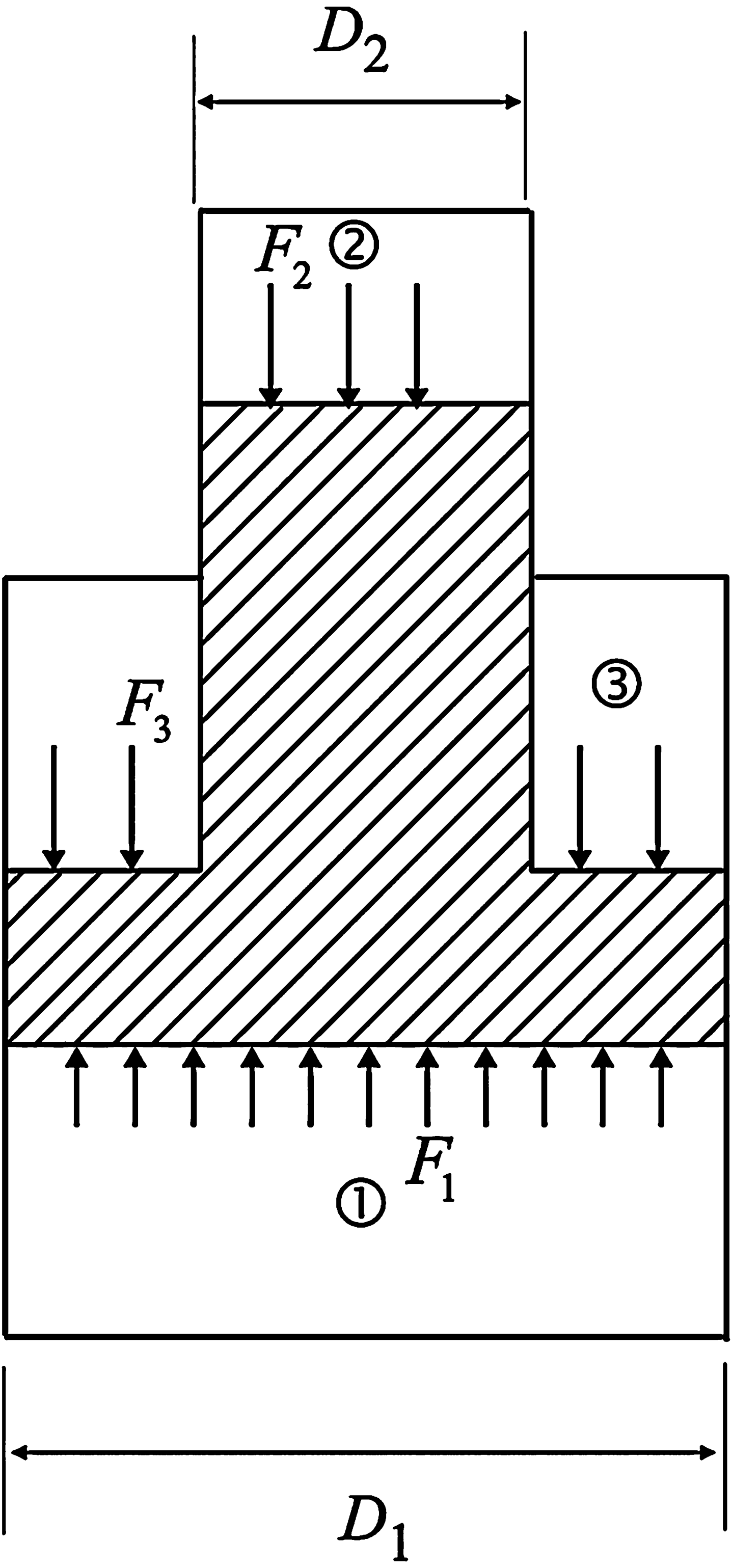
Area 'A_3' on which this force acts

Question: 2.041P pg 1

The piston diameters in Fig. P2-40E are $D1$ = 10 cm and $D2$ = 4 cm. When the pressure in chamber 2 is 2000 kPa and the pressure in chamber 3 is 700 kPa, what is the pressure in chamber 1, in kPa?

Step 1 of 5

Draw the schematic diagram of two-piston cylinder arrangement.



Step 2 of 5

Calculate the area of the piston on which force F_1 acts.

$$A_1 = \frac{\pi}{4} D_1^2$$

Here, diameter of the piston is D_1.

Substitute $0.1\,\text{m}$ for D_1.

$$A_1 = \frac{\pi}{4}(0.1)^2$$
$$= 7.854 \times 10^{-3}\ \text{m}$$

Step 3 of 5

Calculate the area of the piston on which force F_2 acts.

$$A_2 = \frac{\pi}{4} D_2^2$$

Here, diameter of the piston is D_2.

Substitute $0.04\,\text{m}$ for D_2.

$$A_2 = \frac{\pi}{4}(0.04)^2$$
$$= 1.257 \times 10^{-3}\ \text{m}$$

Step 4 of 5

Calculate the area of the piston on which force F_3 acts.

$$A_3 = A_1 - A_2$$

Step 4 of 5

Calculate the area of the piston on which force F_3 acts.

$$A_3 = A_1 - A_2$$

Here, the area of the piston on which force F_1 acts is A_1 and the area of the piston on which force F_2 acts is A_2.

Substitute $7.854\times10^{-3}\ \text{m}$ for A_1 and $1.257\times10^{-3}\ \text{m}$ for A_2.

$$\begin{aligned} A_3 &= \left(7.854\times10^{-3}\ \text{m}\right) - \left(1.257\times10^{-3}\ \text{m}\right) \\ &= 6.597\times10^{-3}\ \text{m} \end{aligned}$$

Step 5 of 5

According to Pascal's law, the force acting on the piston is given by the equation.

$$F = PA \quad \text{..... (1)}$$

Here, the pressure acting on the piston is P, and the area on which the pressure acting is A.

Calculate the sum of vertical forces acting on the piston.

$$\begin{aligned} \sum F_v &= 0 \\ F_2 + F_3 - F_1 &= 0 \end{aligned}$$

Substitute P_1A_1 for F_1, P_2A_2 for F_2 and P_3A_3 for F_3 in above equation according to equation (1).

$$\begin{aligned} P_2A_2 + P_3A_3 - P_1A_1 &= 0 \\ P_1A_1 &= P_2A_2 + P_3A_3 \\ P_1 &= \frac{P_2A_2 + P_3A_3}{A_1} \end{aligned} \quad \text{..... (2)}$$

Here, pressure in chamber 2 is P_2, pressure in chamber 3 is P_3, the area of the piston on which force F_1 acts is A_1, the area of the piston on which force F_2 acts is A_2 and the area of the piston on which force F_1 acts is A_3.

Substitute $2000\ \text{kPa}$ for P_2, $700\ \text{kPa}$ for P_3, $7.854\times10^{-3}\ \text{m}$ for A_1, $1.257\times10^{-3}\ \text{m}$ for A_2 and $6.597\times10^{-3}\ \text{m}$ for A_3 in equation (2).

$$\begin{aligned} P_1 &= \frac{\left(2000\times1.257\times10^{-3}\right) + \left(700\times6.597\times10^{-3}\right)}{7.854\times10^{-3}} \\ &= \frac{2.514 + 4.618}{7.854\times10^{-3}} \\ &= \frac{7.132}{7.854\times10^{-3}} \\ &= 908.07\ \text{kPa} \end{aligned}$$

Therefore, the pressure in chamber 1 of the two-piston cylinder is $\boxed{908\ \text{kPa}}$.

Question: 2.042P

Consider a 70-kg woman who has a total foot imprint area of 400 cm2. She wishes to walk on the snow, but the snow cannot withstand pressures greater than 0.5 kPa. Determine the minimum size of the snowshoes needed (imprint area per shoe) to enable her to walk on the snow without sinking.

Step 1 of 3

The maximum force exerted by a 70-kg women is her weight

Mass of a woman $m = 70\ \text{kg}$

Acceleration due to gravity of a earth $g = 9.81\ \text{m/s}^2$

The total foot imprints area $A = 400\ \text{cm}^2$

The maximum pressure the snow can withstand $p = 0.5\ \text{kPa}$

Step 2 of 3

$$\therefore \text{ Weight of the women} = mg$$
$$= 70 \times 9.81$$
$$= 686.7\ \text{kg} \cdot \text{m/s}^2$$
$$W = 686.7\ \text{N}$$

Step 3 of 3

The weight of the women will be acted on shown by her foot i.e., shoes

$$\therefore \text{ Weight of the women} = \text{pressure} \times \text{Area}$$
$$686.7 = 0.5 \times 10^3 \times A$$
$$\therefore \text{ Area of foot print } \boxed{A = 1.37\ \text{m}^2}$$

So, she can wear show-shoes which should not cover area more than $1.37\ \text{m}^2$

Question: 2.043P

A vacuum gage connected to a tank reads 30 kPa at a location where the barometric reading is 750 mm Hg. Determine the absolute pressure in the tank. Take ρHg = 13,590 kg/m3.

Step 1 of 3

Given:

Vacuum pressure $P_{vacuum} = 30\text{kPa}$

Density of $\rho_{Hg} = 13590\,\frac{\text{kg}}{\text{m}^3}$

Barometric reading $P = 750\,\text{mm of Hg}$

$= 0.75\text{m of Hg}$

Step 2 of 3

Atmospheric pressure in tank is

$$P_{atm} = \rho \times g \times h$$

$$= \left(13590\frac{\text{kg}}{\text{m}^3}\right)\left(9.81\frac{\text{m}}{\text{s}^2}\right)(0.75\text{m})\left(\frac{1\text{N}}{1\frac{\text{kg}\cdot\text{m}}{\text{s}^2}}\right)\left(\frac{1\text{kPa}}{1000\frac{\text{N}}{\text{m}^2}}\right)$$

$= 99.99\text{kPa}$

$= 100\text{kPa}$

Step 3 of 3

Absolute pressure in tank

$$P_{abs} = P_{atm} - P_{vac}$$

$= 100 - 30$

$= 70\text{kPa}$

$\boxed{P_{abs} = 70\text{kPa}}$

Question: 2.044P

The vacuum pressure of a condenser is given to be 80 kPa. If the atmospheric pressure is 98 kPa, what is the gage pressure and absolute pressure in kPa, kN/m2, lbf/in2, psi, and mm Hg.

Step 1 of 4

Calculate the Absolute pressure using the equation,

$$P_{abs} = P_{atm} - P_{vac}$$

Here, P_{atm} is the atmospheric pressure and P_{vac} is the vacuum pressure.

Substitute $98\,\text{kPa}$ for P_{atm} and $80\,\text{kPa}$ for P_{vac}.

$$\begin{aligned} P_{abs} &= 98\text{kPa} - 80\text{kPa} \\ &= 18\text{kPa} \end{aligned}$$

Therefore, the Absolute pressure is $\boxed{18\text{kPa}}$.

Calculate the Absolute pressure using the equation,

$$P_{gage} = P_{abs} - P_{atm}$$

Here, P_{atm} is the atmospheric pressure and P_{abs} is the absolute pressure.

Substitute $98\,\text{kPa}$ for P_{atm} and $18\,\text{kPa}$ for P_{abs}.

$$\begin{aligned} P_{gage} &= 18\text{kPa} - 98\text{kPa} \\ &= -80\text{kPa} \end{aligned}$$

Therefore, the gauge pressure is $\boxed{-80\text{kPa}}$.

Step 2 of 4

Convert the units of absolute pressure from kPa to kN/m^2

$$\begin{aligned} P_{abs} &= 18\text{kPa}\left|\frac{1\ \text{kN/m}^2}{1\,\text{kPa}}\right| \\ &= 18\,\text{kN/m}^2 \end{aligned}$$

Therefore, the absolute pressure in kN/m^2 is $\boxed{18\,\text{kN/m}^2}$.

Convert the units of gauge pressure from kPa to kN/m^2

$$\begin{aligned} P_{gage} &= -80\,\text{kPa}\left|\frac{1\ \text{kN/m}^2}{1\,\text{kPa}}\right| \\ &= -80\,\text{kN/m}^2 \end{aligned}$$

Therefore, the gauge pressure in kN/m^2 is $\boxed{-80\,\text{kN/m}^2}$.

Step 3 of 4

Convert the units of absolute pressure from kPa to lbf/in.^2

$$\begin{aligned} P_{abs} &= 18\text{kPa}\left|\frac{1.45\times10^{-1}\ \text{lbf/in.}^2}{1\,\text{kPa}}\right| \\ &= 2.61\,\text{lbf/in.}^2 \end{aligned}$$

Therefore, the absolute pressure in lbf/in.^2 is $\boxed{2.61\,\text{lbf/in.}^2}$.

Convert the units of gauge pressure from kPa to lbf/in.^2

$$\begin{aligned} P_{gage} &= -80\text{kPa}\left|\frac{1.45\times10^{-1}\ \text{lbf/in.}^2}{1\,\text{kPa}}\right| \\ &= 11.6\,\text{lbf/in.}^2 \end{aligned}$$

Therefore, the gauge pressure in lbf/in.^2 is $\boxed{11.6\,\text{lbf/in.}^2}$.

Step 4 of 4

Convert the units of absolute pressure from kPa to mm of Hg.

$$\begin{aligned} P_{abs} &= 18\text{kPa}\left|\frac{7.5\,\text{mm of Hg}}{1\,\text{kPa}}\right| \\ &= 135\,\text{mm of Hg.} \end{aligned}$$

Therefore, the absolute pressure in mm of Hg. is $\boxed{135\,\text{mm of Hg.}}$.

Convert the units of gauge pressure from kPa to mm of Hg.

$$\begin{aligned} P_{gage} &= -80\text{kPa}\left|\frac{7.5\,\text{mm of Hg}}{1\,\text{kPa}}\right| \\ &= -600\,\text{mm of Hg} \end{aligned}$$

Therefore, the gauge pressure in mm of Hg is $\boxed{-600\,\text{mm of Hg}}$.

The Negative sign indicates rise of mercury above sea level.

Question: 2.045P

The barometer of a mountain hiker reads 740 mbars at the beginning of a hiking trip and 630 mbars at the end. Neglecting the effect of altitude on local gravitational acceleration, determine the vertical distance climbed. Assume an average air density of 1.20 kg/m3.

Step 1 of 3

Given:-

Density of air $\rho_{air} = 1.20 \frac{kg}{m^3}$

Pressure at top of the mountain hiker $P_{Top} = 740 mbar$

Pressure at Bottom of the mountain hiker $P_{Bottom} = 630 mbar$

Step 2 of 3

We know that

Force balance equation we get

$(\rho g h)_{air} = P_{bottom} - P_{top} \ldots\ldots (1)$

$P_{bottom} - P_{top} = (740 - 630) mbar$

$= 110 m\,bar$

$= 0.11 bar$

Step 3 of 3

Substitute the values in equation we get

$\rho \times g \times h = 11 \times 10^3$

$$h = \frac{0.11 bar}{\left(1.20 \frac{kg}{m^3}\right) \times \left(9.81 \frac{m}{s^2}\right) \left(\frac{1N}{1 \frac{kg \cdot m}{s^2}}\right) \left(\frac{1 bar}{100000 \frac{N}{m^2}}\right)}$$

$= 934.42 m$

$\boxed{h \approx 934 m}$

Question: 2.046P

Consider a 1.75-m-tall man standing vertically in water and completely submerged in a pool. Determine the difference between the pressures acting at the head and at the toes of the man, in kPa.

Step 1 of 3

Given:

Height of man $h_2 - h_1 = 1.75\text{m}$

Taking density of water $\rho_{water} = 1000\ \frac{\text{kg}}{\text{m}^3}$

Step 2 of 3

Let h_1 and h_2 be the height of the man top and bottom respectively.

Pressure at depth

$h = \rho \times g \times h$

Where 'g' is the acceleration due to gravity $= 9.81\frac{\text{m}}{s^2}$

Pressure at the head of man

$P_{Head} = \rho \times g \times h_1$

Pressure at the toes of man

$P_{Bottom} = \rho \times g \times h_2$

$h_2 - h_1 =$ Height of man

Step 3 of 3

Pressure difference between toes and head of a man

$P_{Dif} = \rho \times g \times (h_2 - h_1)$

$$= \left(1000\frac{\text{kg}}{\text{m}^3}\right)\left(9.81\frac{\text{m}}{s^2}\right)(1.75\text{m})\left(\frac{1\text{N}}{1\frac{\text{kg}\cdot\text{m}}{s^2}}\right)\left(\frac{1\text{kPa}}{1000\frac{\text{N}}{\text{m}^2}}\right)$$

$$= 17.17\text{kPa}$$

$\boxed{P_{Dif} = 17.17\ \text{kPa}}$

Question: 2.048P

Reconsider Prob. 2–47. Using EES (or other) software, investigate the effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder. Plot the pressure against the spring force, and discuss the results.

Step 1 of 4

Given data:

The effect of the spring force initially on the cylinder, $F_i = 0$ N

The effect of the spring force finally on the cylinder, $F_f = 500$ N

Step 2 of 4

By Using the EES software, we have to write the above equation in new window.

File Edit Search Options Calculate Tables Plots Windows Help Therm

```
g=9.807 [m/s^2]
P_atm=95 [kPa]
m_piston=4 [kg]
"{F_spring=60 [N]}"
A=35*CONVERT('cm^2','m^2')"[m^2]"
W_piston=m_piston*g"[N]"
F_atm=P_atm*A*CONVERT('kPa','N/m^2')"[N]"
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas=F_atm+F_spring+W_piston"[N]"
"P_gas=F_gas/A*CONVERT('N/m^2','kPa')"[kPa]"
```

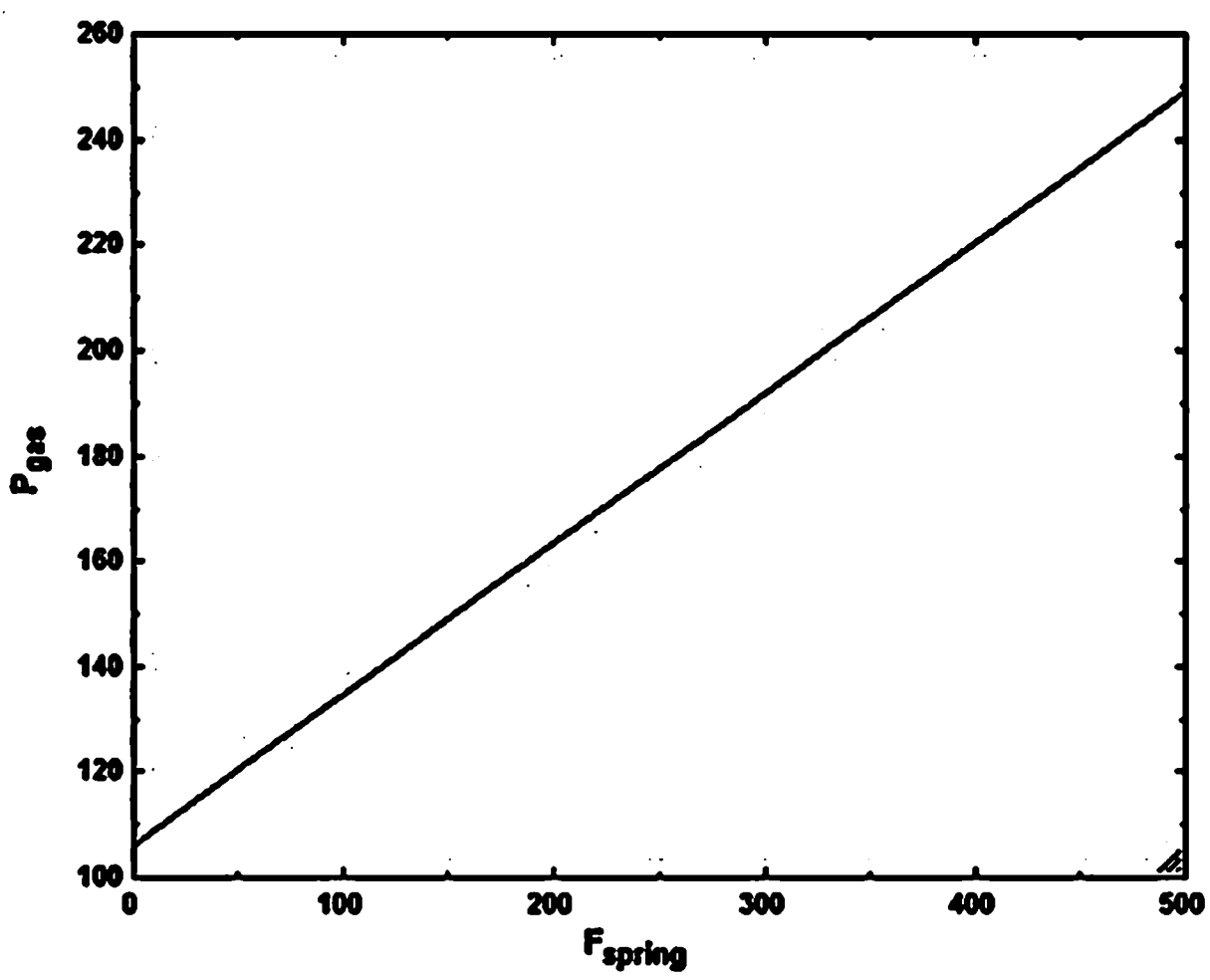
Step 3 of 4

After writing the equations in the Equation window then click on table in menu bar. Then select new parameter table. The spring force varying from the value $F = 0$ to 500 N we get

$\triangleright$ 1..10	1 F_{spring}	2 P_{gas}
Run 1	0	106.2
Run 2	55.56	122.1
Run 3	111.1	138
Run 4	166.7	153.8
Run 5	222.2	169.7
Run 6	277.8	185.6
Run 7	333.3	201.4
Run 8	388.9	217.3
Run 9	444.4	233.2
Run 10	500	249.1

Step 4 of 4

By plotting the graph between springs force versus Pressure of the gas:



Question: 2.049P

Both a gage and a manometer are attached to a gas tank to measure its pressure. If the reading on the pressure gage is 80 kPa, determine the distance between the two fluid levels of the manometer if the fluid is (*a*) mercury (ρ = 13,600 kg/m3) or (*b*) water (ρ = 1000 kg/m3).

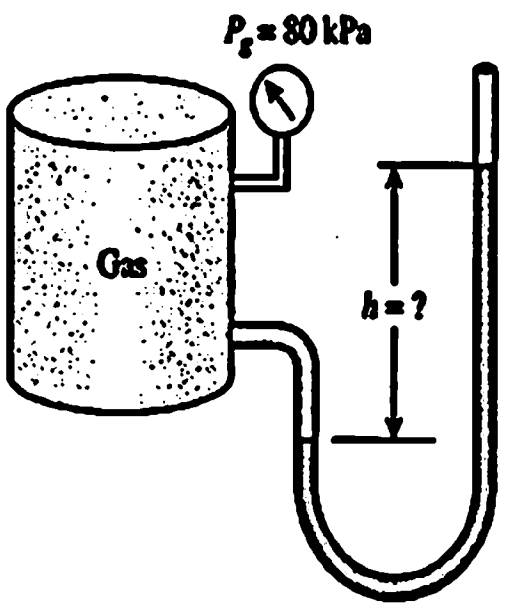
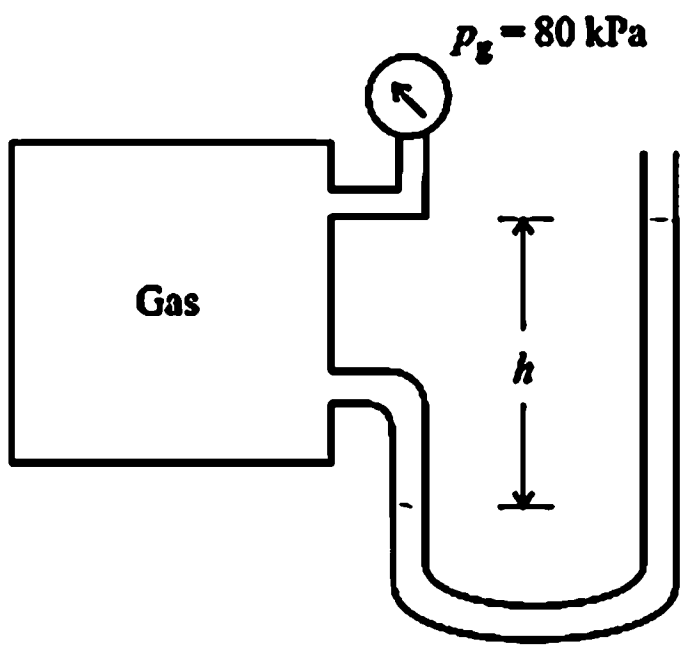


FIGURE P2–49

Step 1 of 4



Step 2 of 4

Given

The gage pressure $p_{gage} = 80$ kPa

The gage pressure related to the height of liquid in manometer is

$$p_{gage} = \rho g h$$

Step 3 of 4

(a) Fluid-Mercury (Hg)

The density of mercury $\rho_{Hg} = 13,600 \text{ kg/m}^3$,

$$g = 9.81 \text{ m/s}^2$$

$$p_{gage} = \rho_{Hg} g h$$

$$h = \frac{80 \times 10^3}{13,600 \times 9.81}$$

$$\boxed{h = 0.6 \text{ m}}$$

Step 4 of 4

(b) The density of Water $\rho = 1000 \text{ kg/m}^3$

$$\text{Here } h = \frac{p_{gage}}{\rho_{H_2O} \times g}$$

$$= \frac{80 \times 10^3}{10^3 \times 9.81}$$

$$\boxed{h = 8.155 \text{ m}}$$

Question: 2.050P

Reconsider Prob. 2–49. Using EES (or other) software, investigate the effect of the manometer fluid density in the range of 800 to 13,000 kg/m3 on the differential fluid height of the manometer. Plot the differential fluid height against the density, and discuss the results.

Step 1 of 1

4749-2-50P **AID: 1118 | 23/5/2012**

Initial fluid density in the manometer, $\rho_i = 800\ kg/m^3$

Final fluid density in the manometer, $\rho_f = 13000\ kg/m^3$

By Using the EES software, we have to write the above equation in new window.

```
Function fluid_density(Fluid$)
If fluid$='Mercury' then fluid_density=13600[kg/m^3] else fluid_density=1000[kg/m^3]
end

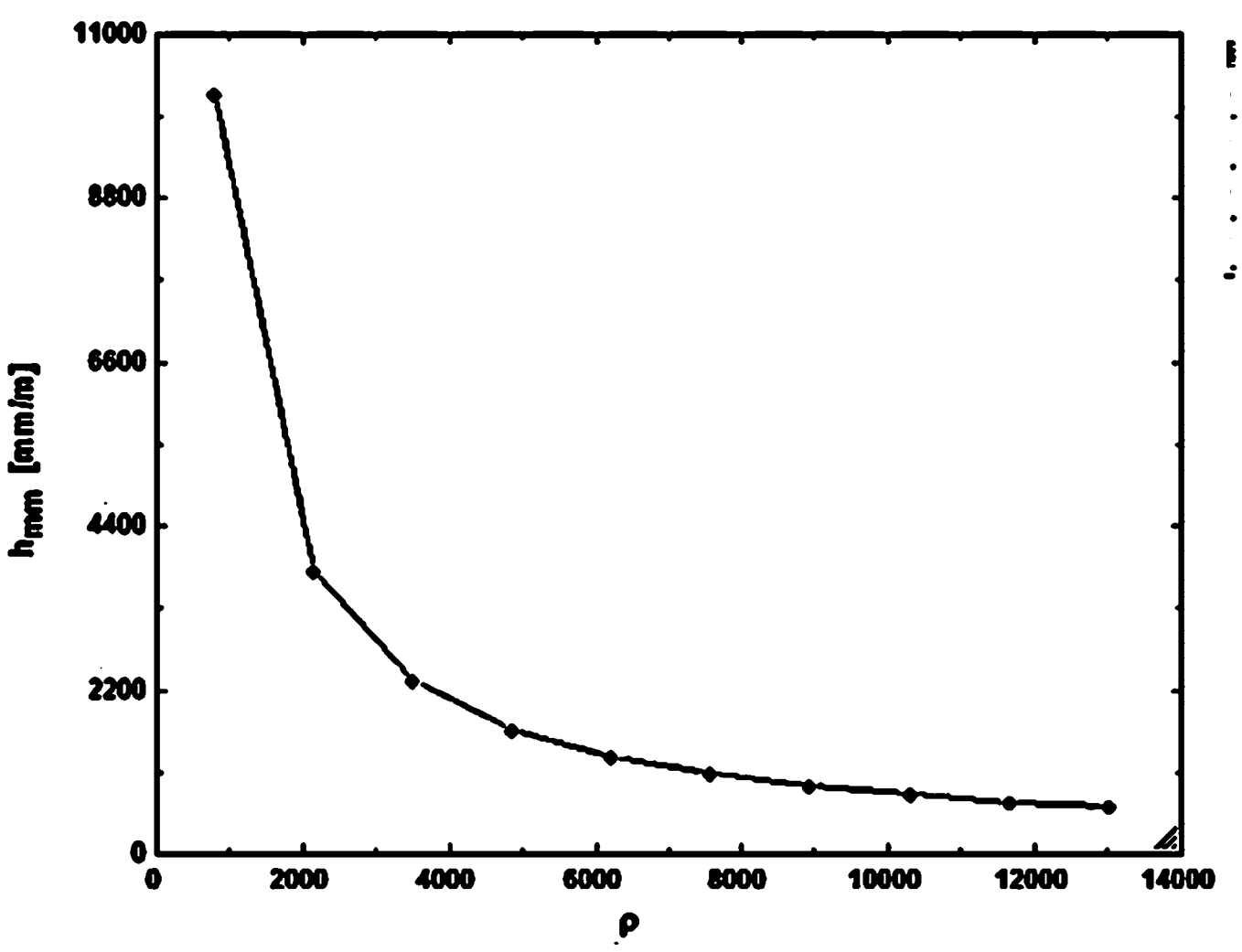
Fluid$='Mercury'
    P_atm = 101.325 [kPa]
        DELTAP=80 [kPa]
g=9.807 [m/s^2]
"rho=Fluid_density(Fluid$)"  "Get the fluid density, either Hg or H2O, from the function."


"To plot fluid height against density place {} around the above equation. Then set up the
parametric table and solve."
DELTAP = rho*g*h*CONVERT(Pa, kPa)
h_mm=h*convert(m, mm)    "The fluid height in mm is found using the built-in CONVERT function."
P_abs= P_atm + DELTAP    "Definition relating gage and absolute pressure."
```

After writing the equations in the Equation window then click on table in menu bar. Then select new parameter table. The density varying from the value $\rho = 800$ to $13000\ kg/m^3$ we get

1..10	h_{mm} [mm/m]	ρ
Run 1	10197	800
Run 2	3784	2156
Run 3	2323	3511
Run 4	1676	4867
Run 5	1311	6222
Run 6	1076	7578
Run 7	913.1	8933
Run 8	792.8	10289
Run 9	700.5	11644
Run 10	627.5	13000

By plotting the graph between density versus manometer height:



Question: 2.051P

A manometer containing oil (ρ = 850 kg/m3) is attached to a tank filled with air. If the oil-level difference between the two columns is 36 cm and the atmospheric pressure is 98 kPa, determine the absolute pressure of the air in the tank.

Step 1 of 3

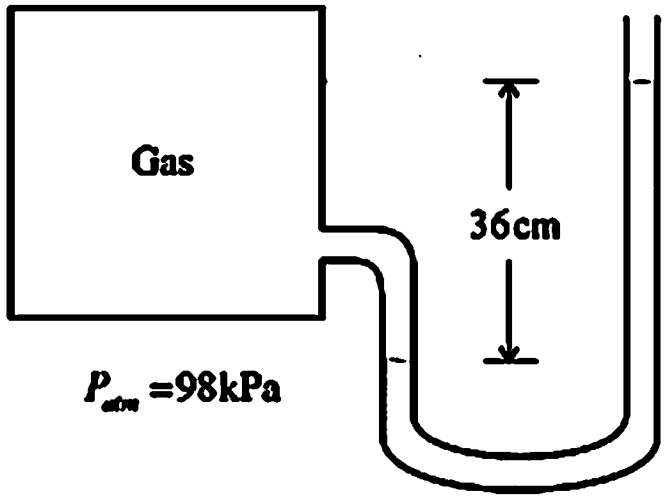
Density of oil $\rho_{oil} = 850 \frac{kg}{m^3}$

Oil level difference in the manometer $h = 36\,cm$

$= 0.36\,m$

Atmospheric pressure $P_{atm} = 98 kPa$

Step 2 of 3



Step 3 of 3

Absolute pressure of the air in the tank is

$P_{abs} = P_{atm} + P_{gauge}$

$P_{abs} = P_{atm} + (\rho \times g \times h)$

$$= 98 kPa + \left[\left(850 \frac{kg}{m^3} \right) \left(9.81 \frac{m}{s^2} \right) (0.36 m) \left(\frac{1 kPa}{1000 \frac{N}{m^2}} \right) \right]$$

$= 101 kPa$

$\boxed{P_{abs} = 101 kPa}$

Question: 2.052P

A mercury manometer (ρ = 13,600 kg/m3) is connected to an air duct to measure the pressure inside. The difference in the manometer levels is 15 mm, and the atmospheric pressure is 100 kPa. (*a*) Judging from Fig. P2-52, determine if the pressure in the duct is above or below the atmospheric pressure, (*b*) Determine the absolute pressure in the duct.

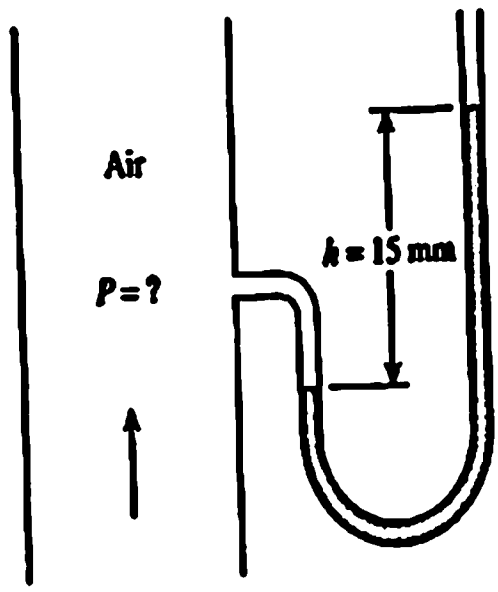
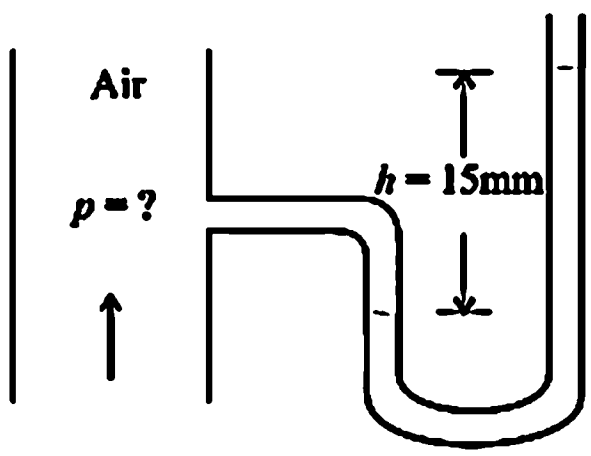


FIGURE P2-52

Step 1 of 4



Step 2 of 4

(a) From figure the mercury column is lower where our pressure is acting. So the pressure 'p' is more than atmospheric pressure, which pushes the mercury in the manometer.

Step 3 of 4

(b)
Given
The atmospheric pressure $p_{atm} = 100\ \text{kPa}$
The density of mercury $\rho_{Hg} = 13,600\ \text{kg/m}^3$
The difference in the manometer levels $h = 15\ \text{mm}$

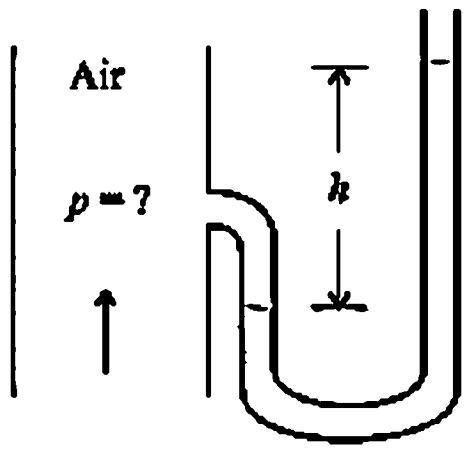
Step 4 of 4

$$p_{absolute} = p_{atm} + \rho g h$$
$$1\ \text{mm} = 10^{-3}\ \text{m}$$
$$g = 9.81\ \text{m/s}^2$$
$$\therefore\ p_{abs} = 100\times10^3 + 13,600\times9.81\times15\times10^{-3}$$
$$= 100\times10^3 + 2\times10^3$$
$$= 102\times10^3\ \text{Pa}$$
$$\boxed{p = 102\ \text{kPa}}$$

Question: 2.053P

Repeat Prob. 2–52 for a differential mercury height of 45 mm.

Step 1 of 3



Step 2 of 3

Given

Atmospheric pressure $p_{atm} = 100\,\text{kPa}$

Density of mercury $\rho_{Hg} = 13,600\,\text{kg/m}^3$

The difference of mercury column in manometer $\left.\begin{array}{l} h = 45\,\text{mm} \\ \quad = 45\times10^{-3}\,\text{m} \end{array}\right\}$

$g = 9.81\,\text{m/s}^2$

Step 3 of 3

Absolute pressure $'p'_{abs} = p_{atm} + \rho g h$

Here $p = 100\times10^3 + 13600\times9.81\times45\times10^{-3}$

$= 100\times10^3 + 6\times10^3$

$= 106\times10^3\,\text{N/m}^2$

$\boxed{p = 106\,\text{kPa}}$

Question: 2.054P

Blood pressure is usually measured by wrapping a closed air-filled jacket equipped with a pressure gage around the upper arm of a person at the level of the heart. Using a mercury manometer and a stethoscope, the systolic pressure (the maximum pressure when the heart is pumping) and the diastolic pressure (the minimum pressure when the heart is resting) are measured in mm Hg. The systolic and diastolic pressures of a healthy person are about 120 mm Hg and 80 mm Hg, respectively, and are indicated as 120/80. Express both of these gage pressures in kPa, psi, and meter water column.

Step 1 of 5

The minimum heart pressure of a healthy person is 80 mmHg

The maximum heart (blood) pressure of a healthy person is 120 mmHg

The gage pressure $\boxed{p = \rho g h}$

Step 2 of 5

(a) In kPa

Low pressure $p_{low} = \rho g h$

Here ρ is $\rho_{Hg} = 13,600$

$h = 80 \text{ mm}$

$= 80 \times 10^{-3} \text{m}$

$g = 9.81 \text{ m/s}^2$

$\therefore \quad p_{low} = 13,600 \times 80 \times 10^{-3} \times 9.81$

$\boxed{p_{low} = 10.67 \text{ kPa}}$

High pressure $p_{high} = \rho_{Hg} g h$

Here $h = 120 \text{ mm}$

$\therefore \quad p_{high} = 13,600 \times 9.81 \times 120 \times 10^{-3}$

$p_{high} = 16,000 \text{ N/m}^2$

$\boxed{p_{high} = 16 \text{ kPa}}$

Step 3 of 5

(b) In psi

$1 \text{ psi} = 6.896 \text{ kPa}$

$p_{low} = 10.67 \text{ kPa}$

$= 10.67 \dfrac{1 \text{ psi}}{6.896}$

$\boxed{p_{low} = 1.54 \text{ psi}}$

$p_{high} = 16 \text{ kPa}$

$= 16 \cdot \dfrac{1 \text{ psi}}{6.896}$

$\boxed{p_{high} = 2.32 \text{ psi}}$

Step 4 of 5

(c) In meter water column

Given in mm of Hg

We have to convert into m of H_2O

Density of water $\rho_{H_2O} = 1000 \text{ kg/m}^3$

From the relation $p = \rho g h$

Here $\rho_{H_2O} g h_{H_2O} = \rho_{Hg} g h_{Hg}$

$$h_{H_2O} = \frac{\rho_{Hg}}{\rho_{H_2O}} \times h_{Hg}$$

Step 5 of 5

$\therefore$ Low pressure water column

$$h_{H_2O} = \frac{13,600}{1000} \times 80 \times 10^{-3}$$

$\boxed{h_{H_2O} = 1.088 \text{ m}}$

High pressure water column

$$h_{H_2O} = \frac{13,600}{1000} \times 120 \times 10^{-3}$$

$\boxed{h_{H_2O} = 1.632 \text{ m}}$

Question: 2.055P

The maximum blood pressure in the upper arm of a healthy person is about 120 mm Hg. If a vertical tube open to the atmosphere is connected to the vein in the arm of the person, determine how high the blood will rise in the tube. Take the density of the blood to be 1050 kg/m3.

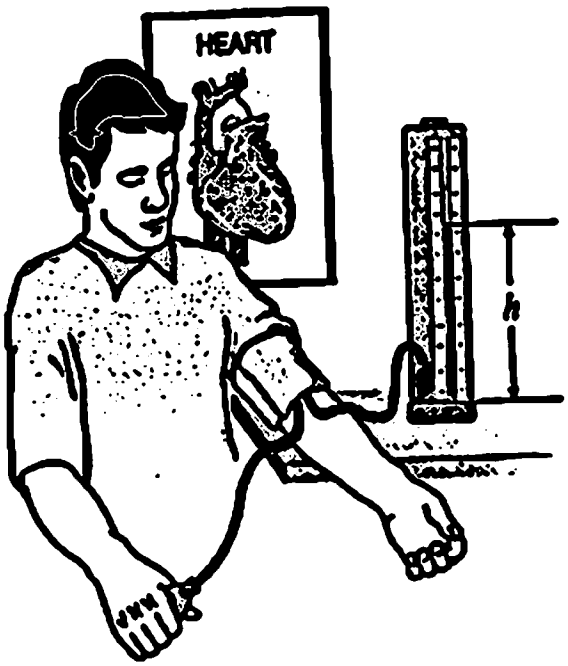


FIGURE P2-55

Step 1 of 1

Given

the maximum blood pressure of a healthy man = 120 mm of Hg

If a vertical cube open to the atmosphere is connected to the vain, the maximum height the blood rises will be equal to the 120 mmHg

i.e., $\rho_{blood} g h_{blood} = \rho_{Hg} g h_{Hg}$

$$\therefore \quad h_{blood} = \frac{\rho_{Hg}}{\rho_{blood}} \times h_{Hg}$$

Given $\rho_{blood} = 1050 \text{ kg/m}^3$

$\rho_{Hg} = 13,600$

$h_{Hg} = 120 \text{ mm}$

$$\therefore \quad h_{blood} = \frac{13,600}{1050} \times 120 \times 10^{-3} \text{m}$$

$$\boxed{h_{blood} = 1.55 \text{ m}}$$

The height raised by blood

Question: 2.056P

Determine the pressure exerted on a diver at 30 m below the free surface of the sea. Assume a barometric pressure of 101 kPa and a specific gravity of 1.03 for seawater.

Step 1 of 3

Given:

Specific gravity of sea water $S.G = 1.03$

Barometric pressure $P = 101\text{kPa}$

Height of the head From the free surface $h = 30\text{m}$

Step 2 of 3

Taking density of water $\rho_{Water} = 1000\ \frac{\text{kg}}{\text{m}^3}$

Density of sea water $\rho_{seawater} = \text{specific gravity} \times \rho_{water}$

$$= 1.03 \times 1000\ \frac{\text{kg}}{\text{m}^3}$$

$$= 1030\ \frac{\text{kg}}{\text{m}^3}$$

Step 3 of 3

The pressure exerted on diver at 30m below the fresh surface of the sea is the absolute pressure at that point

$P = P_{atm} + P_{gauge}$

$P = P_{atm} + (\rho \times g \times h)$

$$= 101\text{kPa} + \left(1030\ \frac{\text{kg}}{\text{m}^3}\right)\left(9.81\ \frac{\text{m}}{\text{s}^2}\right)(30\text{m})\left(\frac{1\text{N}}{1\ \frac{\text{kg}\cdot\text{m}}{\text{s}^2}}\right)\left(\frac{1\text{kPa}}{1000\ \frac{\text{N}}{\text{m}^2}}\right)$$

$$= 404\,\text{kPa}$$

$\boxed{P = 404\ \text{kPa}}$

Question: 2.057P

Consider a U-tube whose arms are open to the atmosphere. Now water is poured into the U-tube from one arm, and light oil (ρ = 790 kg/m3) from the other. One arm contains 70-cm-high water, while the other arm contains both fluids with an oil-to-water height ratio of 4. Determine the height of each fluid in that arm.

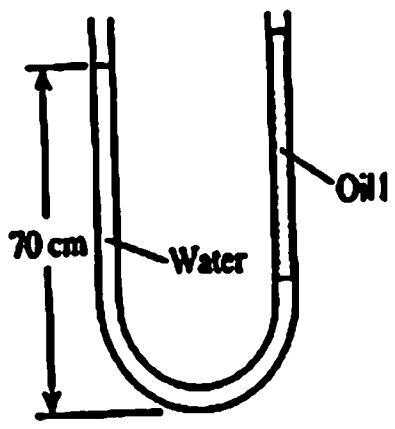
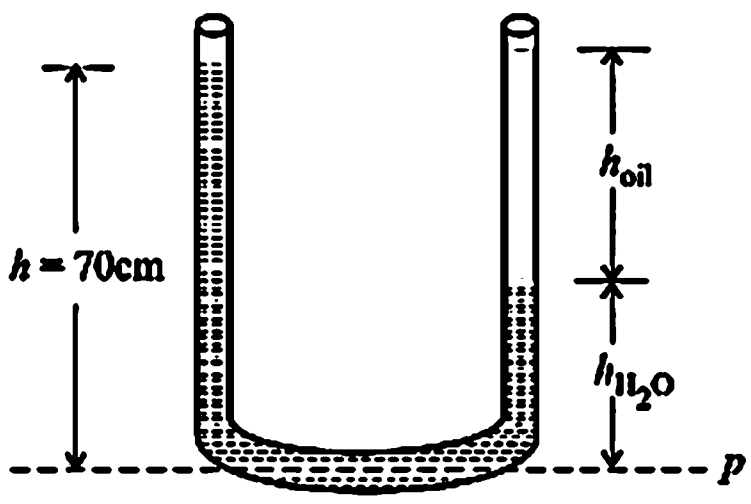


FIGURE P2–57

Step 1 of 5



Step 2 of 5

Let us take height of water column in left arm of U-tube manometer is $h = 70$ cm And on right arm as h_{oil}, h_{water}

The density of water $\rho_{H_2O} = 1000\ kg/m^3$

The density of oil $\rho_{oil} = 790\ kg/m^3$

Height of oil $h_{oil} = 4\ h_w$

$$h_w = \frac{1}{4} h_{oil}$$

Step 3 of 5

The pressure acting on the bottom of the U-tube manometer is same on considering the left and right arms.

$$p_{left} = p_{atm} + \rho_{H_2O} g h$$

$$p_{right} = p_{atm} + \rho_{oil} g\, h_{oil} + \rho_{H_2O} g\, h_{H_2O}$$

Step 4 of 5

On equating both

$$p_{atm} + \rho_{H_2O} g h = p_{atm} + \rho_{oil} g\, h_{oil} + \rho_{H_2O} g\, h_{H_2O}$$

$$\rho_{H_2O} = 1000\ kg/m^3$$

$$\rho_{oil} = 790\ kg/m^3$$

$$h_{oil} = 4\ hw$$

$$\therefore\ 1000 \times 70 \times 10^{-2} = 790 \times h_{oil} + 1000 \times \frac{1}{4} h_{oil}$$

$$1000 \times 70 \times 10^{-2} = (790 + 250) h_{oil}$$

$$\boxed{h_{oil} = 0.673\ m}$$

Step 5 of 5

Height of oil in U-tube manometer arm

$$h_{water} = \frac{1}{4} h_{oil}$$

$$h_{water} = \frac{1}{4} \times 0.673$$

$$= 0.168\ m$$

$$\boxed{h_{water} = 0.168\ m}$$

Height of water in U-tube manometer right arm

Question: 2.058P

Freshwater and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer, as shown in Fig. P2-58. Determine the pressure difference between the two pipelines. Take the density of seawater at that location to be ρ = 1035 kg/m3. Can the air column be ignored in the analysis?

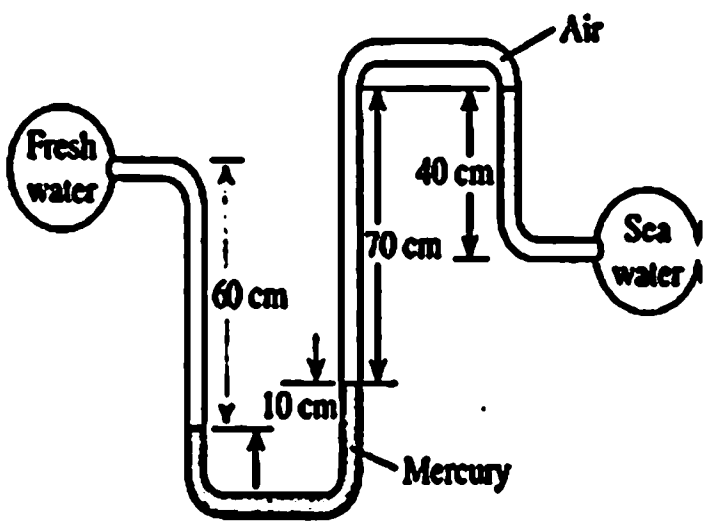
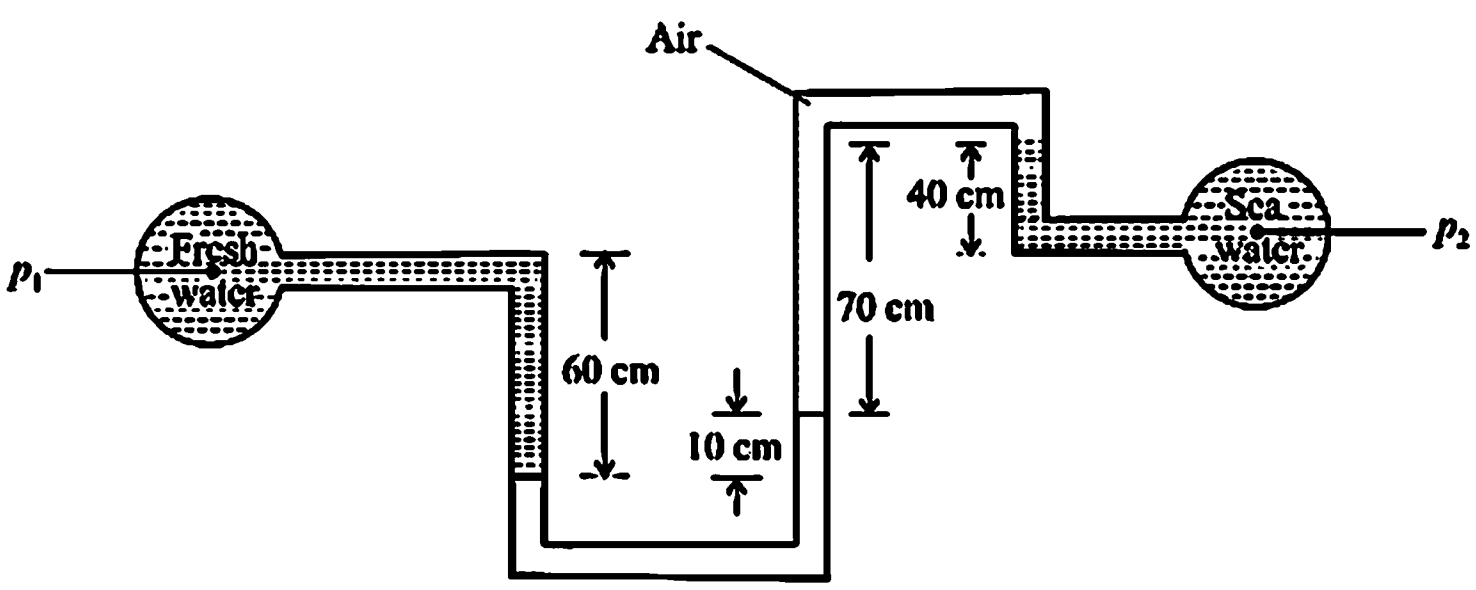


FIGURE P2–58

Step 1 of 4



Step 2 of 4

Given

The height of fresh water $h_w = 60$ cm (fresh water)

The height of mercury $h_{Hg} = 10$ cm (mercury)

The height of air $h_{air} = 70$ cm (air-column)

The height of sea water $h_{sea} = 40$ cm (sea-water)

The density of sea water $\rho_{sea} = 1035\ kg/m^3$

The density of air $\rho_{air} = 1.2256\ kg/m^3$ (density of air)

The density of fresh water $\rho_{water} = 1000\ kg/m^3$

The density of mercury $\rho_{Hg} = 13,600\ kg/m^3$

The acceleration due to gravity of earth $g = 9.81\ m/s^2$

Step 3 of 4

Let us consider the pressure in fresh water tube is p_1 and the pressure in sea water tube is p_2

As we go from p_1 to p_2 in the two U-tube manometers, various pressures are calculated according to the fluid levels in the manometer.

Taking positive as we go down and negative in upward direction

$$\therefore\ p_1 + \rho_{H_2O} g h_w - \rho_{Hg} g h_{Hg} - \rho_{air} g h_{air} + \rho_{sea} g h_{sea} = p_2 \quad \text{.....(1)}$$

The pressure difference between two pipe lines is $p_1 - p_2$

$$\therefore\ p_1 - p_2 = \rho_{Hg} g h_{Hg} + \rho_{air} g h_{air} - \rho_{sea} g h_{sea} - \rho_{H_2O} g h_w$$

Step 4 of 4

Substituting all these values in above equation

$$\Rightarrow p_1 - p_2 = 13,600 \times 9.8 \times 10 \times 10^{-2} + 1.2256 \times 9.81 \times 10 \times 10^{-2} - 1035 \times 9.81 \times 40 \times 10^{-2} - 1000 \times 9.81 \times 60 \times 10^{-2}$$

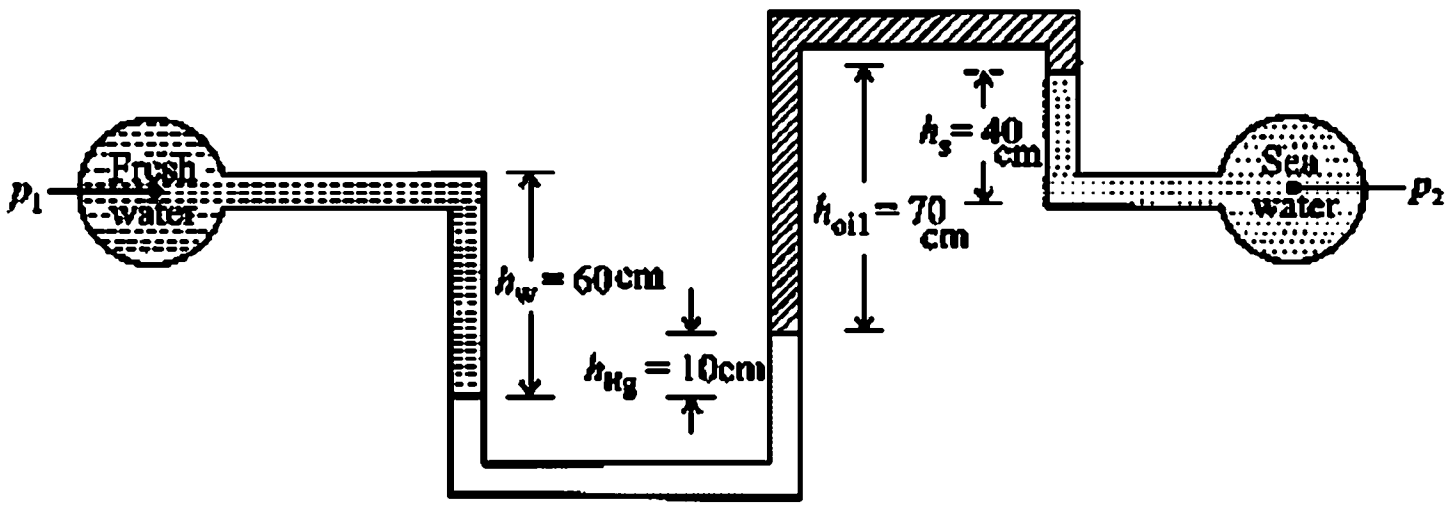
$$\boxed{p_1 - p_2 = 3.4\ kPa}$$

The air column can be ignored because the air columns produce a very low pressure of 3.4 Pa

Question: 2.059P

Repeat Prob. 2–58 by replacing the air with oil whose specific gravity is 0.72.

Step 1 of 4



Step 2 of 4

The height of fresh water $h_w = 60$ cm (fresh water)

The height of mercury $h_{Hg} = 10$ cm (mercury)

The height of oil $h_{oil} = 70$ cm (oil-column)

The height of sea water $h_{sea} = 40$ cm (sea-water)

The density of sea water $\rho_{sea} = 1035\ \text{kg/m}^3$

The density of fresh water $\rho_{water} = 1000\ \text{kg/m}^3$

The density of mercury $\rho_{Hg} = 13,600\ \text{kg/m}^3$

The acceleration due to gravity of earth $g = 9.81\ \text{m/s}^2$

The specific gravity of oil S.G= 0.72

$$\text{S.G} = \frac{\rho_{oil}}{\rho_{water}}$$

The density of oil $\rho_{oil} = 0.72 \times 1000$

$= 720\ \text{kg/m}^3$

Step 3 of 4

Let us consider the pressure in fresh water tube is p_1 and the pressure in sea water tube is p_2

As we move from p_1 to p_2 in the two U-tube manometers, various pressures is calculated according to the fluid levels in the manometer.

Taking positive as we go down and negative pressure in upward direction

$\therefore\ p_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} + \rho_s g h_s = p_2$ .....(1)

The pressure difference between two pipe lines is $p_1 - p_2$

$\Rightarrow\ p_1 - p_2 = \rho_{Hg} g h_{Hg} + \rho_{oil} g h_{oil} - \rho_w g h_w - \rho_s g h_s$ .....(2)

Step 4 of 4

Substituting all the values in equation (2)

$$\Rightarrow\ p_1 - p_2 = 13,600 \times 9.81 \times 10 \times 10^{-2} + 720 \times 9.81 \times 70 \times 10^{-2} - 1000 \times 9.81 \times 60 \times 10^{-2} - 1035 \times 9.81 \times 40 \times 10^{-2}$$

$$= 13.34 \times 10^3 + 4.94 \times 10^3 - 5.88 \times 10^3 - 4.06 \times 10^3$$

$$= 8.338 \times 10^3\ \text{N/m}^2$$

$10^3\ \text{N/m}^2 = 1\ \text{kPa}$

The pressure difference $\boxed{p_1 - p_2 = 8.338\ \text{kPa}}$

Question: 2.060P

Calculate the absolute pressure, $P1$ of the manometer shown in Fig. P2-60 in kPa. The local atmospheric pressure is 758 mm Hg.

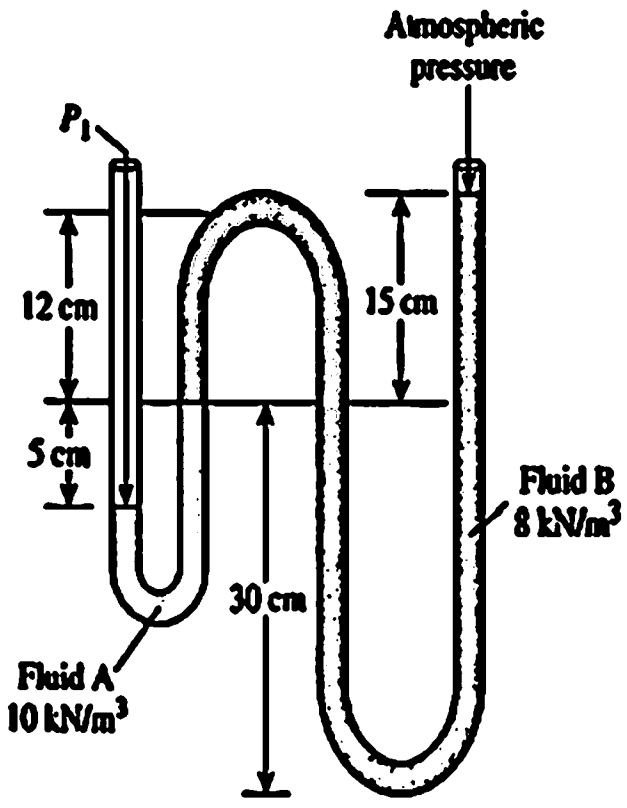
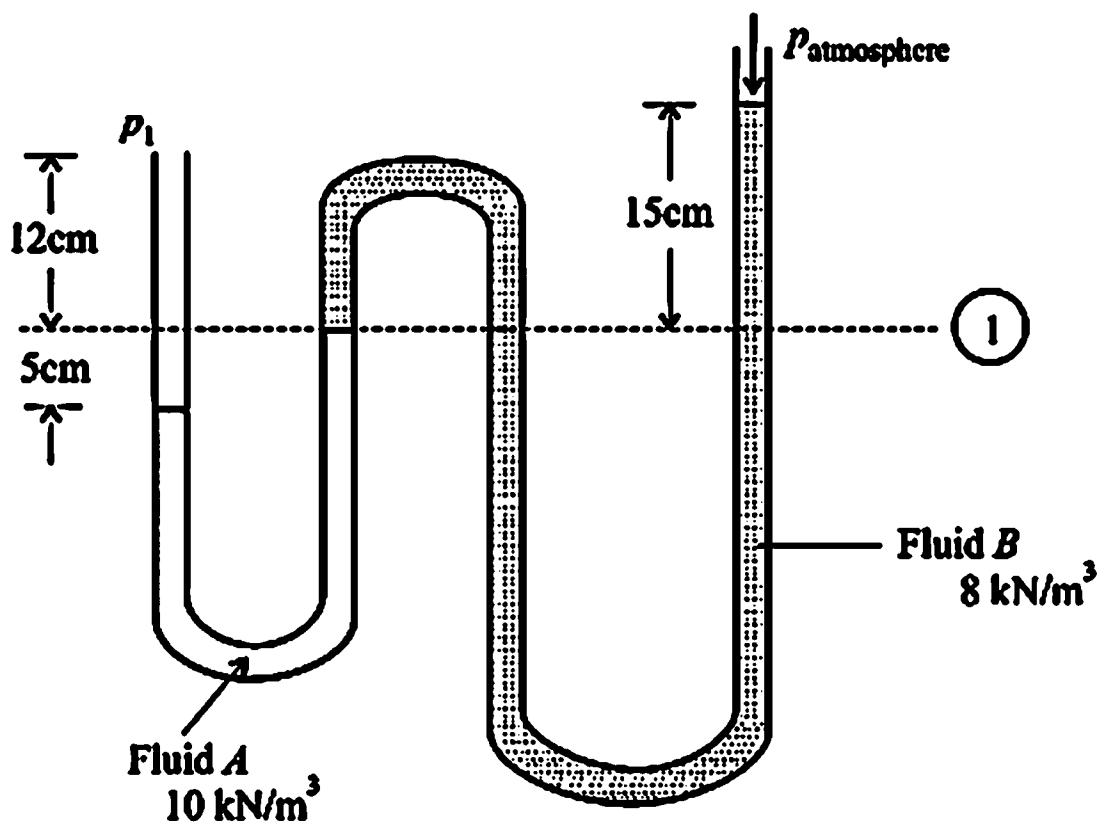


FIGURE P2–60

Step 1 of 4



Step 2 of 4

Given

Local atmospheric pressure = 758 mmHg

$\rho_{Hg} = 13,600 \text{ kg/m}^3$

$g = 9.81 \text{ m/s}^2$

$\therefore \quad p_{atm} = \rho_{Hg} g h$

$= 13,600 \times 9.81 \times 758 \times 10^{-3}$

$= 101.13 \text{ kPa}$

Step 3 of 4

The absolute pressure p_1 is calculated by

$p_1 = p_{atm} + \rho_a g h_a + \rho_B g h_B$ .....(1)

Given specific weights of fluids A and B

i.e., $\gamma_A = \rho_a g$

$= 10 \text{ kN/m}^3$

$\gamma_B = \rho_B g$

$= 8 \text{ kN/m}^3$

Step 4 of 4

Taking the line (1) in figure as reference the height $h_a = 5$ cm and $h_B = 15$ cm

Equation (1)

$\Rightarrow \quad p_1 = p_{atm} + \gamma_A h_a + \gamma_B h_B$

$= 101.13 \times 10^3 + 10 \times 10^3 \times 5 \times 10^{-2} + 8 \times 10^3 \times 15 \times 10^{-2}$

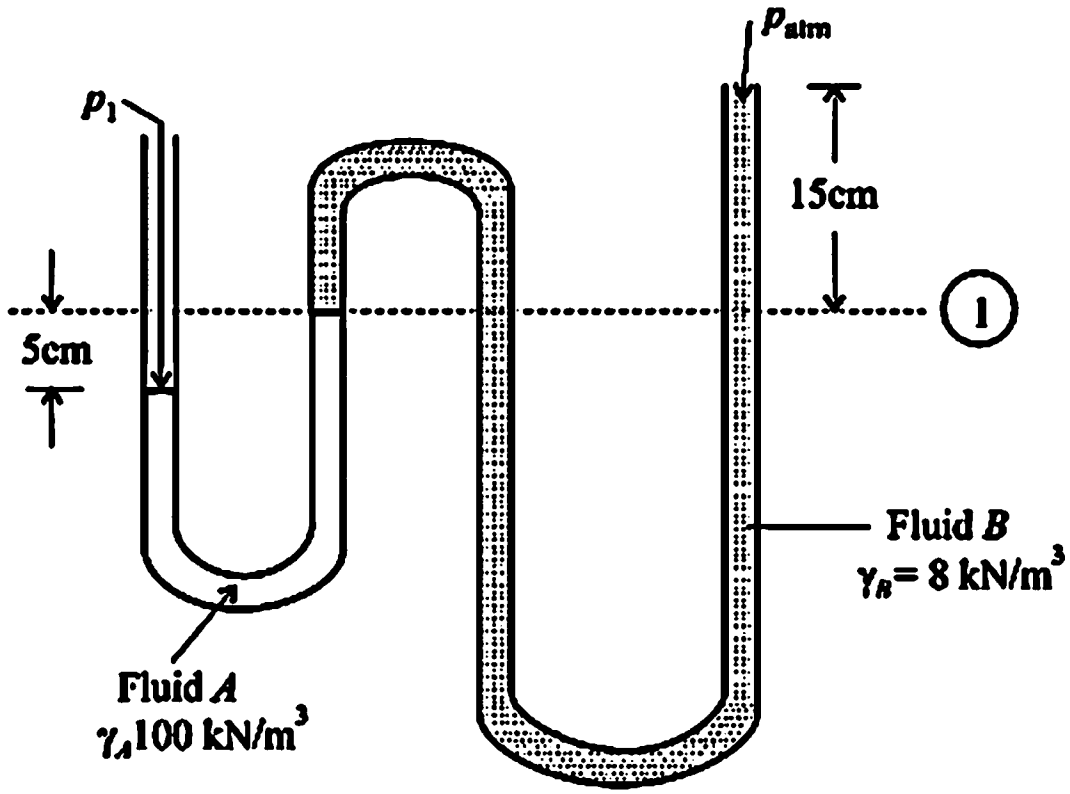
$= 102.83 \times 10^3 \text{ N/m}^2$

$\boxed{p_1 = 102.83 \text{ kPa}}$

Question: 2.061P

Consider the manometer in Fig. P2-60. If the specific weight of fluid A is 100 kN/m3, what is the absolute pressure, in kPa, indicated by the manometer when the local atmospheric pressure is 90 kPa?

Step 1 of 3



Step 2 of 3

Given

the local atmospheric pressure

i.e., $p_{atm} = 90$ kPa

Specific weight of

Fluid A $\gamma_A = 100$ kN/m^3

Fluid B $\gamma_B = 8$ kN/m^3

The height $h_A = 5$ cm

The height $h_B = 15$ cm

Step 3 of 3

The absolute pressure $p_1 = p_{atm} + \gamma_A h_A + \gamma_B h_B$

$$p_1 = 90\times10^3 + 100\times10^3 \times 5\times10^{-2} + 8\times10^3 \times 15\times10^{-2}$$

$$= 96.2\times10^3 \text{ N/m}^2$$

$$\boxed{p_1 = 96.2 \text{ kPa}}$$

Question: 2.062P

Consider the manometer in Fig. P2-60. If the specific weight of fluid B is 12 kN/m3, what is the absolute pressure, in kPa, indicated by the manometer when the local atmospheric pressure is 720 mm Hg?

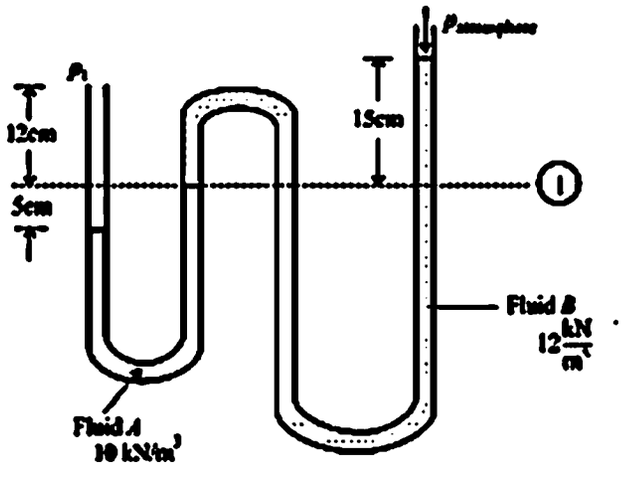
Step 1 of 4

Given:

Specific weight of the fluid $\gamma_B = 12\frac{\text{kN}}{\text{m}^3}$

Atmospheric pressure $P_{atm} = 720\,\text{mm of Hg}$

Step 2 of 4



Step 3 of 4

Atmospheric pressure in kPa is

$$P_{atm} = 720\,\text{mm of Hg}$$

$$= (720\,\text{mm of Hg})\left(0.1333\frac{\text{kPa}}{\text{mm of Hg}}\right)$$

$$= 95.98\,\text{kPa}$$

Step 4 of 4

Absolute pressure is

$$P_1 = P_{atm} + (\gamma_A h_1) + (\gamma_B h_2)$$

$$= 95.98\,\text{kPa} + \left(\left(10\frac{\text{kN}}{\text{m}^3}\right)(0.05\text{m}) + \left(12\frac{\text{kN}}{\text{m}^3}\right)(0.15\text{m})\right)$$

$$= 98.28\frac{\text{kN}}{\text{m}^2} \qquad \left(1\text{kPa} = \frac{\text{kN}}{\text{m}^2}\right)$$

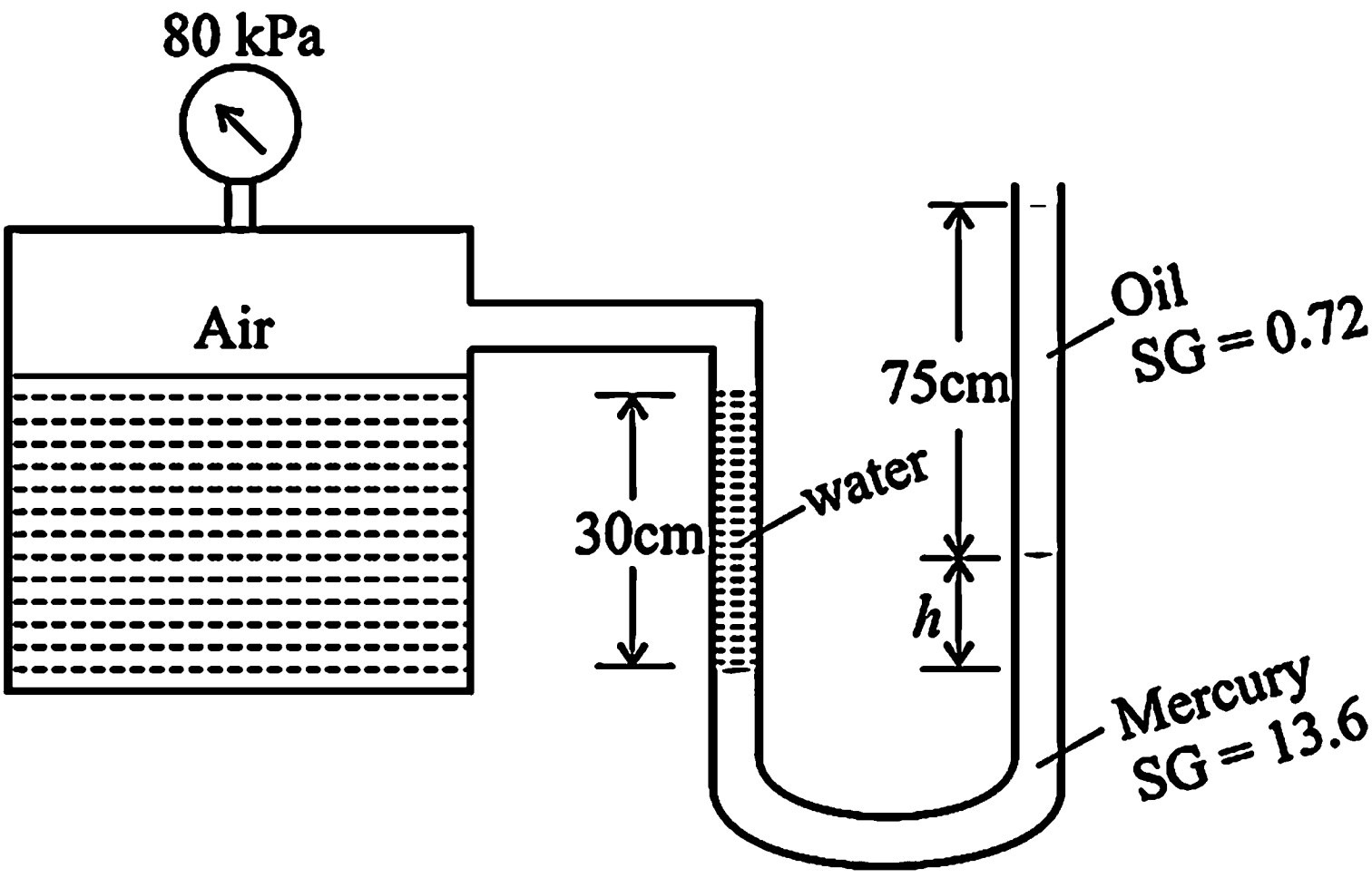
$$\boxed{P_1 = 98.28\,\text{kPa}}$$

Question: 2.063P

The gage pressure of the air in the tank shown in Fig. P2-63 is measured to be 80 kPa. Determine the differential height *h* of the mercury column.

Step 1 of 2

Draw the tank that consists of air, and pressure of air is measured by pressure gage and the differential manometer as shown in figure:



Step 2 of 2

Find the density of mercury by using specific gravity of mercury

$$SG = \frac{\rho_{Hg}}{\rho_{H_2O}}$$

Here, ρ_{Hg} is density of mercury and ρ_{H_2O} is density of water.

Substitute 13.6 for SG and $1000\,kg/m^3$ for ρ_{H_2O}

$$13.6 = \frac{\rho_{Hg}}{1000}$$

$$\rho_{Hg} = 13600\,kg/m^3$$

Similarly find the density of oil by using the equation,

$$SG = \frac{\rho_{oil}}{\rho_{H_2O}}$$

Substitute 0.72 for SG

$$0.72 = \frac{\rho_{oil}}{1000}$$

$$\rho_{oil} = 720\,kg/m^3$$

Write the pressure equation starting from gage pressure in the tank and adding or subtracting the pressure terms along the length of the tube and equate to the atmospheric pressure (since the surface of oil is exposed to atmosphere).

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = P_{atm}$$

$$\left(P_{gage} + P_{atm}\right) + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = P_{atm}$$

$$P_{gage} + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{oil} g h_{oil} = 0$$

Here, P_1 is absolute air pressure in the tank measured by gage pressure, P_{gage} is gage pressure in the tank, P_{atm} is atmospheric pressure, h_w is height of water column, h_{Hg} is height of mercury column and h_{oil} is height of oil column and *g* is acceleration due to gravity.

Substitute $80\times10^3\,Pa$ for P_{gage}, $30\,cm = 0.3\,m$ for h_w, $75\,cm = 0.75\,m$ for h_{oil} and $9.81\,m/s^2$ for g

$$80\times10^3 + 1000\times9.81\times0.3 - 13600\times9.81\times h_{Hg} - 720\times9.81\times0.75 = 0$$

$$80000 + 2943 - 133416h_{Hg} - 5297.4 = 0$$

$$77645.6 = 133416h_{Hg}$$

$$h_{Hg} = 0.5819\,m$$

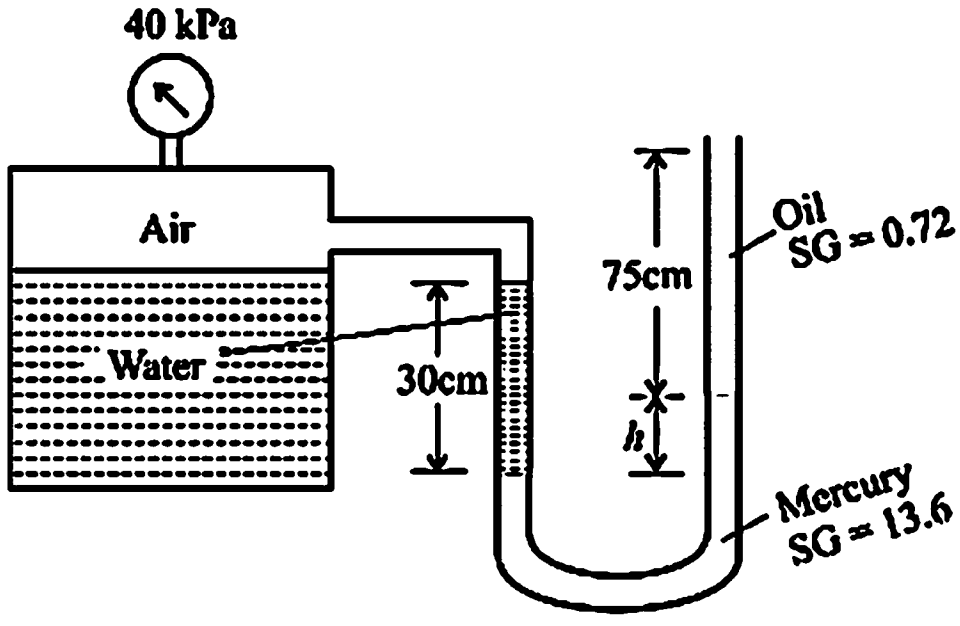
The differential height h of the mercury column is **0.5819 m**

Question:

2.064P

Repeat Prob. 2–63 for a gage pressure of 40 kPa.

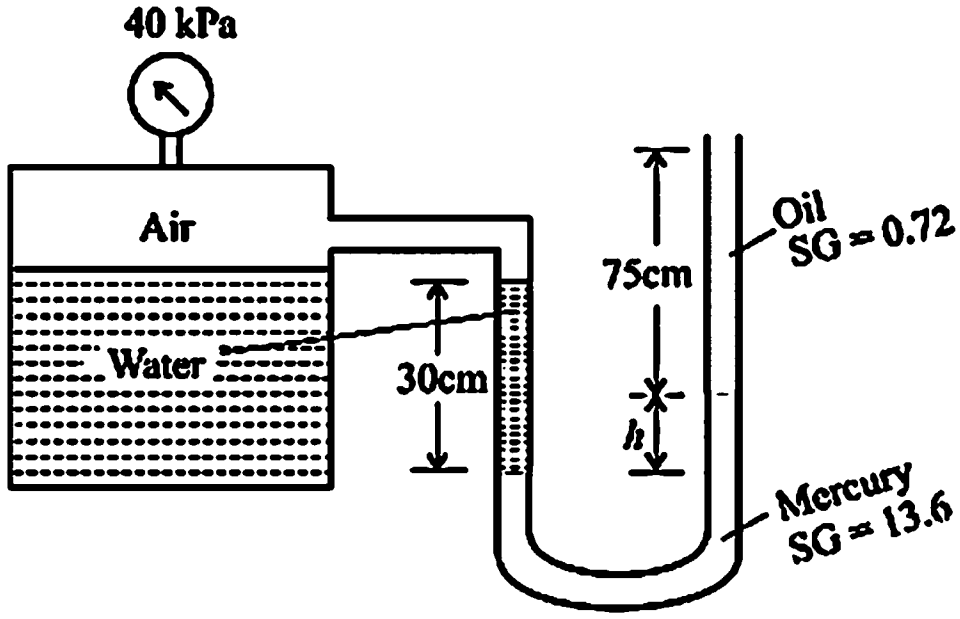
Step 1 of 4



Question:

Repeat Prob. 2–63 for a gage pressure of 40 kPa.

Step 1 of 4



Mercury $h_{Hg} = ?$

$1\text{ cm} = 10^{-2}\text{m}$

$g = 9.81\text{ m/s}^2$

Step 4 of 4

Substituting all these values in equation (1)

$$p_{gage} = \rho_{oil} g h_{oil} + \rho_{Hg} g h_{Hg} - \rho_w g h_w$$

$$\Rightarrow 40\times10^3 = 720\times9.81\times75\times10^{-2} + 13,600\times9.81\times h_{Hg} - 1000\times9.81\times30\times10^{-2}$$

$$h_{Hg} = \frac{\left(40-(5.29-2.94)\right)\times10^3}{13,600\times9.81}$$

$$= 0.2832\text{ m}$$

The differential height h of the mercury column is $\boxed{h_{Hg} = 28.32\text{ cm}}$

Question: 2.065P

The top part of a water tank is divided into two compartments, as shown in Fig. P2-65. Now a fluid with an unknown density is poured into one side, and the water level rises a certain amount on the other side to compensate for this effect. Based on the final fluid heights shown on the figure, determine the density of the fluid added. Assume the liquid does not mix with water.

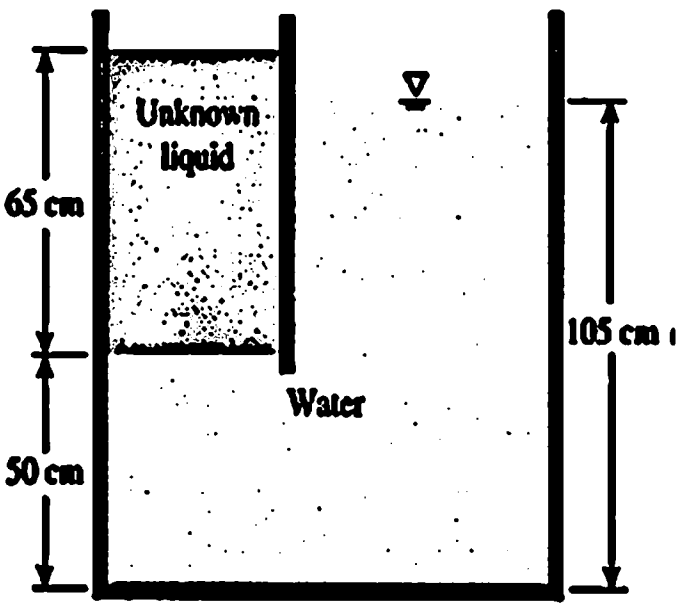


FIGURE P2-65

Question:

The top part of a water tank is divided into two compartments, as shown in Fig. P2-65. Now a fluid with an unknown density is poured into one side, and the water level rises a certain amount on the other side to compensate for this effect. Based on the final fluid heights shown on the figure, determine the density of the fluid added. Assume the liquid does not mix with water.

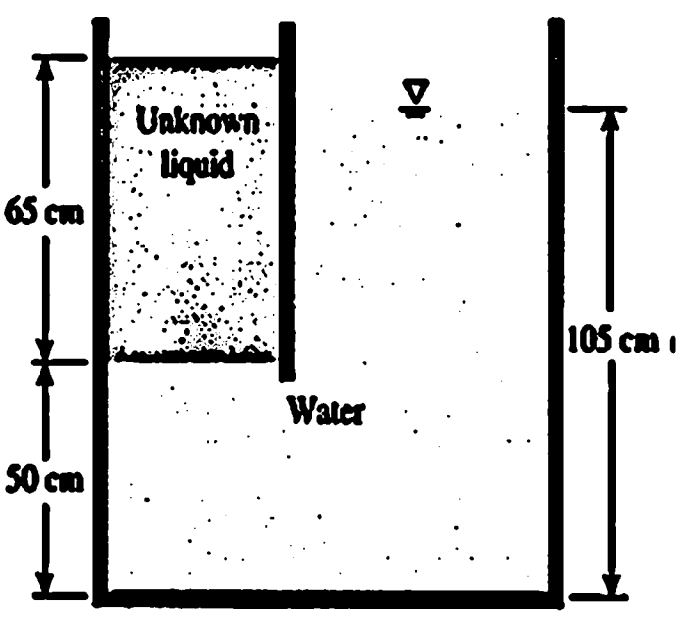


FIGURE P2-65

Step 3 of 3

Substituting the values we have,

$h_f = 65cm$

$h_w = 105\,\text{cm} - 50\,\text{cm}$

$= 55\,\text{cm}$

$$\rho_f = \frac{55}{65} \times 1000 \frac{\text{kg}}{\text{m}^3}$$

$$= 846.15 \frac{\text{kg}}{\text{m}^3}$$

$$\boxed{\rho_f = 846.15 \frac{\text{kg}}{\text{m}^3}}$$

Step 3 of 3

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$h_f = 65cm$

$h_w = 105\,\text{cm} - 50\,\text{cm}$

$= 55\,\text{cm}$

$$\rho_f = \frac{55}{65} \times 1000 \frac{\text{kg}}{\text{m}^3}$$

$$= 846.15 \frac{\text{kg}}{\text{m}^3}$$

$$\boxed{\rho_f = 846.15 \frac{\text{kg}}{\text{m}^3}}$$

Question:

2.066P

Consider the system shown in Fig. P2-66.If a change of 0.7 kPa in the pressure of air causes the brine-mercury interface in the right column to drop by 5 mm in the brine level in the right column while the pressure in the brine pipe remains constant, determine the ratio of *A2/A1*.

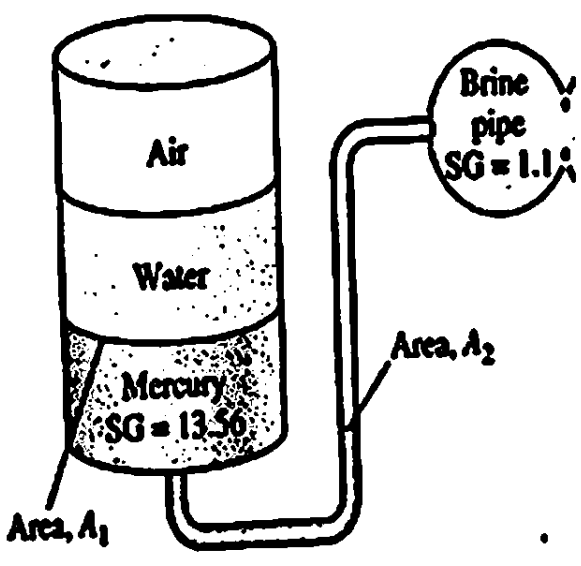


FIGURE P2–66

Question:

Consider the system shown in Fig. P2-66.If a change of 0.7 kPa in the pressure of air causes the brine-mercury interface in the right column to drop by 5 mm in the brine level in the right column while the pressure in the brine pipe remains constant, determine the ratio of *A2/A1*.

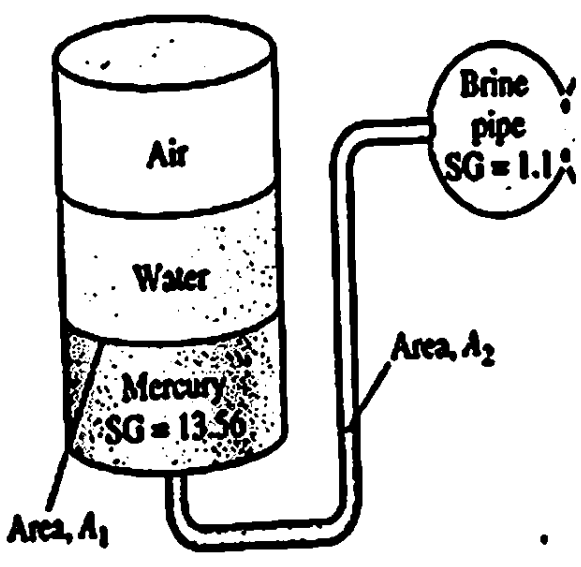


FIGURE P2–66

Consider the pressure drop in the left and right hand side of the area A_1 column.

Write the relation for pressure change in the column.

$$P_{A_1} + \rho_w g(h_w) + \rho_{Hg} g(h_{Hg})_{A_1} = \rho_{brine} g(h_{brine})_{A_1} + P_B$$

$$P_{A_1} + \rho_w g h_w + \rho_w (SG_{Hg}) g(h_{Hg})_{A_1} = \rho_w (SG_{brine}) g(h_{brine})_{A_1} + P_B \quad \text{..... (1)}$$

Consider the pressure drop in the left and right hand side of the area A_2 column.

Write the relation for pressure change in the column.

$$P_{A_2} + \rho_w g(h_w) + \rho_{Hg} g(h_{Hg})_{A_2} = \rho_{brine} g(h_{brine})_{A_2} + P_B$$

$$P_{A_2} + \rho_w g(h_w) + \rho_w (SG_{Hg}) g(h_{Hg})_{A_2} = \rho_w (SG_{brine}) g(h_{brine}) A_2 + P_B \quad \text{..... (2)}$$

Here, specific gravity of mercury is $SG_{(Hg)}$, specific gravity of brine solution is $SG_{(brine)}$, the density of water is ρ_w, change in pressure of the air from the area A_1 to A_2 is $P_{A_1} - P_{A_2}$, change in brine level in the right column is Δh_{brine}, the mercury level in the right column is $(h_{Hg})_{at A_2}$, and pressure in brine pipe is P_B .

Step 3 of 5

Subtract the equation (1) from (2).

$$\left\{\begin{array}{l}\left(P_{A_2} + \rho_w g(h_w) + \rho_w (SG_{Hg}) g(h_{Hg})_{A_2}\right) \\ \quad -\left(P_{A_1} + \rho_w g h_w + \rho_w (SG_{Hg}) g(h_{Hg})_{A_1}\right)\end{array}\right\} = \left\{\begin{array}{l}\left(\rho_w (SG_{brine}) g(h_{brine})_{A_2} + P_B\right) \\ \quad -\left(\rho_w (SG_{brine}) g(h_{brine})_{A_1} + P_B\right)\end{array}\right\}$$

$$P_{A_2} - P_{A_1} = \left\{\rho_w (SG_{brine}) g\left[(h_{brine})_{A_2} - (h_{brine})_{A_1}\right]\right\} - \left\{\rho_w (SG_{Hg}) g\left[(h_{Hg})_{A_2} - (h_{Hg})_{A_1}\right]\right\}$$

$$P_{A_2} - P_{A_1} = \rho_w (SG_{brine}) g \Delta h_{brine} - \rho_w (SG_{Hg}) g \Delta h_{Hg} \quad \text{..... (3)}$$

Step 4 of 5

As Mercury is incompressible, the volume of the mercury remains constant in the left and right limb.

$$V_{left} = V_{right}$$

$$A_1 (\Delta h_{Hg})_{left} = A_2 (\Delta h_{Hg})_{right}$$

Write the relation for the total change of the mercury height.

$$\Delta h_{Hg} = (\Delta h_{Hg})_{left} + (\Delta h_{Hg})_{right}$$

$$= (\Delta h_{Hg}) + \frac{A_2 (\Delta h_{Hg})}{A_1 (\Delta h_{Hg})}$$

$$\Delta h_{Hg} = \Delta h_{Hg} \left(1 + \frac{A_2}{\cdot}\right)$$

$$\left\{\begin{array}{l}\left(P_{A_2} + \rho_w g(h_w) + \rho_w (SG_{Hg}) g(h_{Hg})_{A_2}\right) \\ \quad -\left(P_{A_1} + \rho_w g h_w + \rho_w (SG_{Hg}) g(h_{Hg})_{A_1}\right)\end{array}\right\} = \left\{\begin{array}{l}\left(\rho_w (SG_{brine}) g(h_{brine})_{A_2} + P_B\right) \\ \quad -\left(\rho_w (SG_{brine}) g(h_{brine})_{A_1} + P_B\right)\end{array}\right\}$$

$$P_{A_2} - P_{A_1} = \left\{\rho_w (SG_{brine}) g\left[(h_{brine})_{A_2} - (h_{brine})_{A_1}\right]\right\} - \left\{\rho_w (SG_{Hg}) g\left[(h_{Hg})_{A_2} - (h_{Hg})_{A_1}\right]\right\}$$

$$P_{A_2} - P_{A_1} = \rho_w (SG_{brine}) g \Delta h_{brine} - \rho_w (SG_{Hg}) g \Delta h_{Hg} \quad \text{..... (3)}$$

Step 4 of 5

As Mercury is incompressible, the volume of the mercury remains constant in the left and right limb.

$$V_{left} = V_{right}$$

$$A_1 (\Delta h_{Hg})_{left} = A_2 (\Delta h_{Hg})_{right}$$

Write the relation for the total change of the mercury height.

$$\Delta h_{Hg} = (\Delta h_{Hg})_{left} + (\Delta h_{Hg})_{right}$$

$$= (\Delta h_{Hg}) + \frac{A_2 (\Delta h_{Hg})}{A_1 (\Delta h_{Hg})}$$

$$\Delta h_{Hg} = \Delta h_{Hg} \left(1 + \frac{A_2}{A_1}\right)$$

$$\left(1 + \frac{A_2}{A_1}\right) = \frac{0.0768}{0.0678}$$

$$\frac{A_2}{A_1} = 1.133 - 1$$

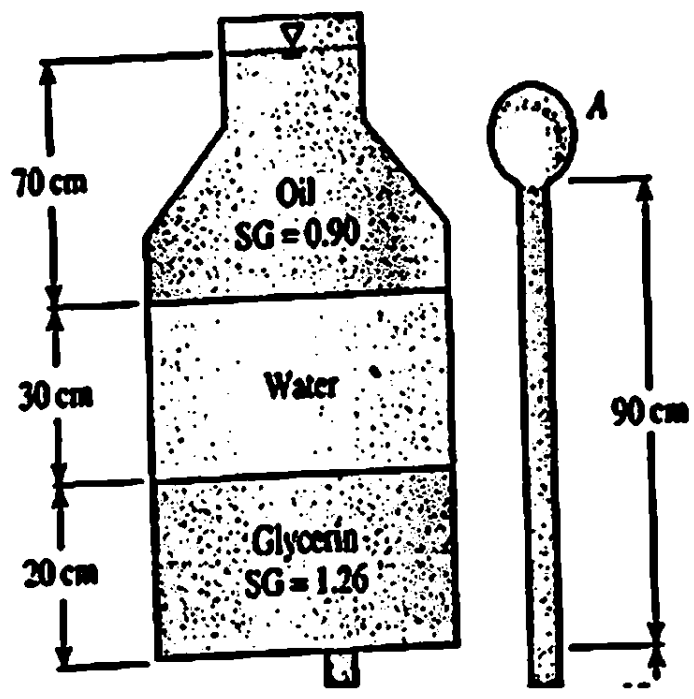
$$= 0.133$$

Therefore, the $\frac{A_2}{A_1}$ ratio is $\boxed{0.133}$.

Question:

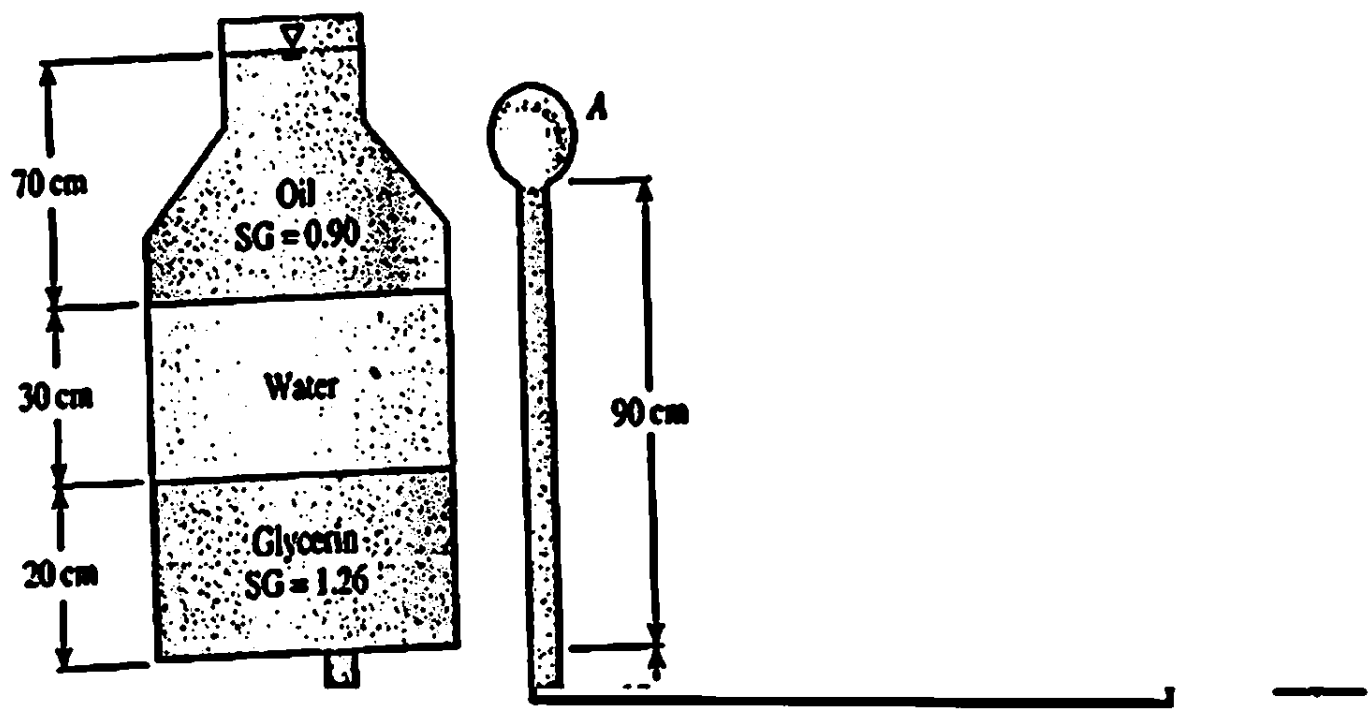
2.067P

A multifluid container is connected to a U-tube, as shown in Fig. P2-67. For the given specific gravities and fluid column heights, determine the gage pressure at *A*.Also determine the height of a mercury column that would create the same pressure at *A*.



Question:

A multifluid container is connected to a U-tube, as shown in Fig. P2-67. For the given specific gravities and fluid column heights, determine the gage pressure at *A*.Also determine the height of a mercury column that would create the same pressure at *A*.



Step 2 of 4

The gage pressure $p_{gage} = \rho gh$

Write the specific gravities of different fluids:

Oil $S.G = 0.9$

$S.G = \dfrac{\rho_{oil}}{\rho_w}$

$\rho_{oil} = 0.9 \times 1000$

$\rho_{oil} = 900 \text{ kg/m}^3$

$\rho_w = 1000 \text{ kg/m}^3$

Glycerin $S.G = 1.26$

$\rho_{gly} = 1.26 \times 1000$

$\rho_{gly} = 1260 \text{ kg/m}^3$

Step 3 of 4

Step 2 of 4

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Glycerin $S.G = 1.26$

$\rho_{gly} = 1.26 \times 1000$

$\rho_{gly} = 1260 \text{ kg/m}^3$

Step 3 of 4

$h_{Hg} = \dfrac{0.447 \times 10}{13,600 \times 9.81} \times 10^{-2}$

Therefore, the head produced due to mercury is $\boxed{h_{Hg} = 0.352 \text{ cm}}$.

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Therefore, the head produced due to mercury is $\boxed{h_{Hg} = 0.352 \text{ cm}}$.

Question: 2.068P

The pilot of an airplane reads the altitude 9000 m and the absolute pressure 25 kPa when flying over a city. Calculate the local atmospheric pressure in that city in kPa and in mm Hg. Take the densities of air and mercury to be 1.15 kg/m3 and 13,600 kg/m3, respectively.

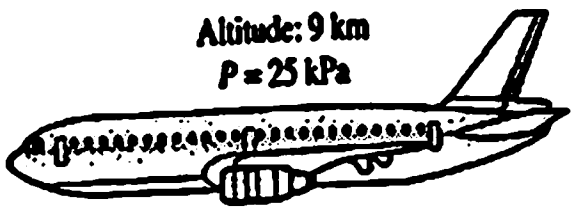


FIGURE P2–68

Step 1 of 2

Calculate the atmospheric pressure in kPa.

$$P_{abs} = P_{atm} + (-\rho_A g h)$$

Question:

The pilot of an airplane reads the altitude 9000 m and the absolute pressure 25 kPa when flying over a city. Calculate the local atmospheric pressure in that city in kPa and in mm Hg. Take the densities of air and mercury to be 1.15 kg/m3 and 13,600 kg/m3, respectively.

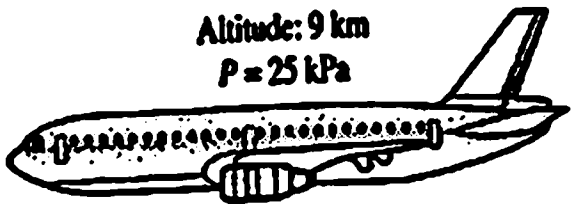


FIGURE P2–68

Step 1 of 2

Calculate the atmospheric pressure in kPa.

$$P_{abs} = P_{atm} + (-\rho_A g h)$$

$$h_{Hg} = \frac{\Delta P_{atm}}{\rho g}$$

$$= \frac{(126.4\,\text{kPa})}{\left(13600\,\frac{\text{kg}}{\text{m}^3}\right)\left(9.81\,\frac{\text{m}}{\text{s}^2}\right)}\left(\frac{1000\,\text{Pa}}{1\,\text{kPa}}\right)\left(\frac{1000\,\text{mm}}{1\,\text{m}}\right)$$

$$= 947.4\,\text{mm}$$

Therefore, the atmospheric pressure in mm of Hg is, $\boxed{h_{Hg} = 947.4\,\text{mm}}$.

$$h_{Hg} = \frac{\Delta P_{atm}}{\rho g}$$

$$= \frac{(126.4\,\text{kPa})}{\left(13600\,\frac{\text{kg}}{\text{m}^3}\right)\left(9.81\,\frac{\text{m}}{\text{s}^2}\right)}\left(\frac{1000\,\text{Pa}}{1\,\text{kPa}}\right)\left(\frac{1000\,\text{mm}}{1\,\text{m}}\right)$$

$$= 947.4\,\text{mm}$$

Therefore, the atmospheric pressure in mm of Hg is, $\boxed{h_{Hg} = 947.4\,\text{mm}}$.

Question: 2.069P

The efficiency of a refrigerator increases by 3 percent for each °C rise in the minimum temperature in the device. What is the increase in the efficiency for each (*a*) K, (*b*) °F, and (*c*) R rise in temperature?

Step 1 of 4

Given $\eta_{\text{in C}} = 3\%\ /°\text{C}$

Step 2 of 4

(a) $T(\text{K}) = 273.15 + °\text{C}$

So 3% for K rise in temperature

Step 3 of 4

(b) $\Delta T(°\text{F}) = \Delta T(\text{R}) = 1.8\Delta T(\text{K})$

Here the percentage increase in efficiency is $\frac{3}{1.8} = 1.67\%$ for each °F

Step 4 of 4

(c) 1.67% °R Rise in temperature

Question: 2.070P

The boiling temperature of water decreases by about 3°C for each 1000-m rise in altitude. What is the decrease in the boiling temperature in (*a*) K, (*b*) °F, and (*c*) R for each 1000-m rise in altitude?

Step 1 of 4

The decreases of boiling temperature of water for 1000 m rise in altitude is $\Delta T = 3°C$
The relation between Kelvins to degree-Centigrade is

$$T(\mathrm{K}) = 273.15 + T(°\mathrm{C})$$

$$\Delta T(\mathrm{K}) = \Delta T(°\mathrm{C})$$

Step 2 of 4

(a) The decrease in boiling temperature is same in K i.e, 3 K

Step 3 of 4

(b) The relation between Fahrenheit – Kelvin

$$T(°\mathrm{F}) = 1.8T(°\mathrm{C}) + 32$$

$$\Delta T(°\mathrm{F}) = 1.8\Delta T(°\mathrm{C})$$

$$= 1.8 \times 3$$

$$\boxed{\Delta T(°\mathrm{F}) = 5.4°\mathrm{F}}$$

Step 4 of 4

(c) The relation between Fahrenheit - Ranking

$$T(\mathrm{R}) = T(°\mathrm{F}) + 459.67$$

$$\Delta T(\mathrm{R}) = \Delta T(°\mathrm{F})$$

$$\therefore \boxed{\Delta T(\mathrm{R}) = 5.4\ \mathrm{R}}$$

Question: 2.071P

Hyperthermia of 5°C (i.e., 5°C rise above the normal body temperature) is considered fatal. Express this fatal level of hyperthermia in (*a*) K, (*b*) °F, and (*c*) R.

Step 1 of 4

Given $\Delta T(^\circ\text{C}) = 5^\circ\text{C}$

We have to express in K, °F and R

Step 2 of 4

(a) We know $\Delta T(\text{K}) = \Delta T(^\circ\text{C})$

$\therefore \boxed{\Delta T(\text{K}) = 5\,\text{K}}$

Step 3 of 4

(b) $\Delta T(^\circ\text{F}) = 1.8\Delta T(^\circ\text{C})$

$= 1.8 \times 5$

$\boxed{\Delta T(^\circ\text{F}) = 9^\circ\text{F}}$

Step 4 of 4

(c) $\Delta T(^\circ\text{F}) = \Delta T(\text{R})$

$\therefore \boxed{\Delta T(^\circ\text{F}) = 9\,\text{R}}$

Question: 2.072P

A house is losing heat at a rate of 2700 kJ/h per °C temperature difference between the indoor and the outdoor temperatures. Express the rate of heat loss from this house per (*a*) K, (*b*) °F, and (*c*) R difference between the indoor and the outdoor temperature.

Step 1 of 3

Given

Rate of loss of heat is $Q = 2700\ (\text{kJ/h})$ per ^0C

Change in magnitude of temperature is given by

$1\text{K} = 1^\circ\text{C} = 1.8\text{R} = 1.8^\circ\text{F}$

(a)

$1\text{K} = 1^\circ\text{C}$

Therefore the, rate of heat loss from the house is

$$Q = 2700\ (\text{kJ/h}^0\text{C}) \times \frac{1^0\text{C}}{1\text{K}}$$

$$Q = 2700\ (\text{kJ/h K})$$

$$Q = 2700\ (\text{kJ/h})\ \text{per K}$$

Hence rate of heat loss from the house per K is $\boxed{Q = 2700\ (\text{kJ/h})\ \text{per K}}$

Step 2 of 3

(b)

$1^\circ\text{C} = 1.8\text{R}$

Therefore the, rate of heat loss from the house is

$$Q = 2700\ (\text{kJ/h}^0\text{C}) \times \frac{1^0\text{C}}{1.8\text{R}}$$

$$Q = \frac{2700}{1.8}\ (\text{kJ/hR})$$

$$Q = 1500\ (\text{kJ/h})\text{per R}$$

Hence rate of heat loss from the house per R is $\boxed{Q = 1500\ (\text{kJ/h})\text{per R}}$

Step 3 of 3

(c)

$1^\circ\text{C} = 1.8^\circ\text{F}$

Therefore the, rate of heat loss from the house is

$$Q = 2700\ (\text{kJ/h}^0\text{C}) \times \frac{1^0\text{C}}{1.8^\circ\text{F}}$$

$$Q = \frac{2700}{1.8}\ (\text{kJ/h}^\circ\text{F})$$

$$Q = 1500\ (\text{kJ/h})\text{per }^\circ\text{F}$$

Hence rate of heat loss from the house per R is $\boxed{Q = 1500\ (\text{kJ/h})\text{per }^\circ\text{F}}$

Question: 2.073P

The average temperature of the atmosphere in the world is approximated as a function of altitude by the relation

$T_{atm} = 288.15 - 6.5z$

where T_{atm} is the temperature of the atmosphere in K and z is the altitude in km with $z = 0$ at sea level. Determine the average temperature of the atmosphere outside an airplane that is cruising at an altitude of 12,000 m.

Step 1 of 1

Given relation between average temperature to altitude

$T_{atm} = 288.15 - 6.5z$

Where T_{atm} = atmosphere temperature in K

z is altitude in km

The average temperature at $z = 12,000$ m

$z = 12$ km

$\therefore T_{atm} = 288.15 - 6.5 \times 12$

$\boxed{T_{atm} = 210.15 \text{ K}}$

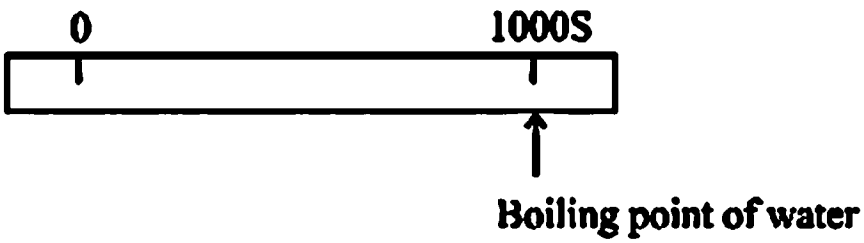
Question: 2.074P

Joe Smith, an old-fashioned engineering student, believes that the boiling point of water is best suited for use as the reference point on temperature scales. Unhappy that the boiling point corresponds to some odd number in the current absolute temperature scales, he has proposed a new absolute temperature scale that he calls the Smith scale. The temperature unit on this scale is *smith*, denoted by S, and the boiling point of water on this scale is assigned to be 1000 S. From a thermodynamic point of view, discuss if it is an acceptable temperature scale. Also, determine the ice point of water on the Smith scale and obtain a relation between the Smith and Celsius scales.

Step 1 of 4

The new absolute temperature scale is smith scale, unit is smith 's'

Step 2 of 4



Step 3 of 4

Boiling point of water in this scale is $(B.P)_S = 1000\ s$ .....(1)

We know that boiling point in our Kelvin scale is

$$(B.P)_K = 373.15\ K \quad .....(2)$$

i.e., point $(I.P)_K = 273.15\ K$

$\therefore$ From (1) and (2)

$$T_s = \frac{1000}{373.15} T_K$$

$$\boxed{T_s = 2.68\ T_K}$$

Step 4 of 4

$\therefore$ Ice point of water in Smith scale

$$(I.P)_S = 2.68(T_K)_{ice\ pt}$$

$$= 2.68 \times 273.15$$

$$\boxed{(I.P)_S = 732S}$$

Question: 2.075P

It is well-known that cold air feels much colder in windy weather than what the thermometer reading indicates because of the "chilling effect" of the wind. This effect is due to the increase in the convection heat transfer coefficient with increasing air velocities. The *equivalent wind chill temperature* in °F is given by [ASHRAE, *Handbook of Fundamentals* (Atlanta, GA, 1993), p. 8.15]

$$T_{equiv} = 91.4 - (91.4 - T_{ambient}) \times (0.475 - 0.0203V + 0.304\sqrt{V})$$

where V is the wind velocity in mi/h and Tambient is the ambient air temperature in °F in calm air, which is taken to be air with light winds at speeds up to 4 mi/h. The constant 91.4°F in the given equation is the mean skin temperature of a resting person in a comfortable environment. Windy air at temperature

TAmbient and velocity Vwill feel as cold as the calm air at temperature Tequiv. Using proper conversion factors, obtain an equivalent relation in SI units where $V$is the wind velocity in km/h and Tambient is the ambient air temperature in °C.

Step 1 of 2

Given the equivalent wind chill temperature in °F

$$T_{equiv} = 91.4 - (91.4 - T_{ambient}) \times \left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \quad \text{.....(1)}$$

Here V is in mi/hr (miles per hour)

1 mi/hr = 1.609 km/hr

The relation between Fahrenheit - °C is

$$T(°F) = 1.8T(°C) + 32$$

Step 2 of 2

$$(1) \Rightarrow 1.8T_{eq}(°C) + 32 = 91.4 - \left\{\left[91.4 - (1.8 \times T_{amb}(°C) + 32)\right] \times \left(0.475 - 0.0203\frac{V}{1.609} + 0.304\sqrt{\frac{V}{1.609}}\right)\right\}$$

$$T_{eq}(°C) = \frac{91.4 - 32}{1.8} - \frac{1}{1.8}\left\{\left[(91.4 - 32) - 1.8T_{amb}(°C)\right] \times \left[0.475 - 0.0126V + 0.240\sqrt{V}\right]\right\}$$

The equivalent relation in SI units

$$\boxed{T_{equiv} = 33 - (33 - T_{ambient}) \times \left(0.475 - 0.0126V + 0.24\sqrt{V}\right)}$$

Question: 2.076P

Reconsider Prob. 2-75E. Using EES (or other) software, plot the equivalent wind chill temperatures in °F as a function of wind velocity in the range of 4 to 40 mph for the ambient temperatures of 20, 40, and 60°F. Discuss the results.

Step 1 of 5

Plot the equivalent wind chill temperature as a function of wind velocity 4 to **40 mph** for ambient temperature of **20° F** , **40° F** and **60° F** using EES software .

Write the following code in the software.

" T_equiv=91.4-(91.4-Tambient)*90.475-0.0203*V+0.304*sqrt(V)) "

Step 2 of 5

Use the parametric table option and tabulate the $T_{equivalent}$ for $T_{ambient}$ at **20°F** and varying the velocity value, *V* between **4 mph to 40 mph** .

▷ 1..10	1 $T_{ambient}$	2 T_{equiv}	3 V
Run 1	20	19.87	4
Run 2	20	7.688	8
Run 3	20	-0.3124	12
Run 4	20	-6.147	16
Run 5	20	-10.6	20
Run 6	20	-14.06	24
Run 7	20	-16.79	28
Run 8	20	-18.92	32
Run 9	20	-20.57	36
Run 10	20	-21.82	40

Table (1)

Step 3 of 5

Use the parametric table option and tabulate the $T_{equivalent}$ for $T_{ambient}$ at **40°F** and varying the velocity values *V* between **4 mph to 40 mph**

▷ 1..10	1 $T_{ambient}$	2 T_{equiv}	3 V
Run 1	40	39.91	4
Run 2	40	31.14	8
Run 3	40	25.38	12
Run 4	40	21.18	16
Run 5	40	17.97	20
Run 6	40	15.48	24
Run 7	40	13.52	28
Run 8	40	11.98	32
Run 9	40	10.79	36
Run 10	40	9.897	40

Table (2)

Step 4 of 5

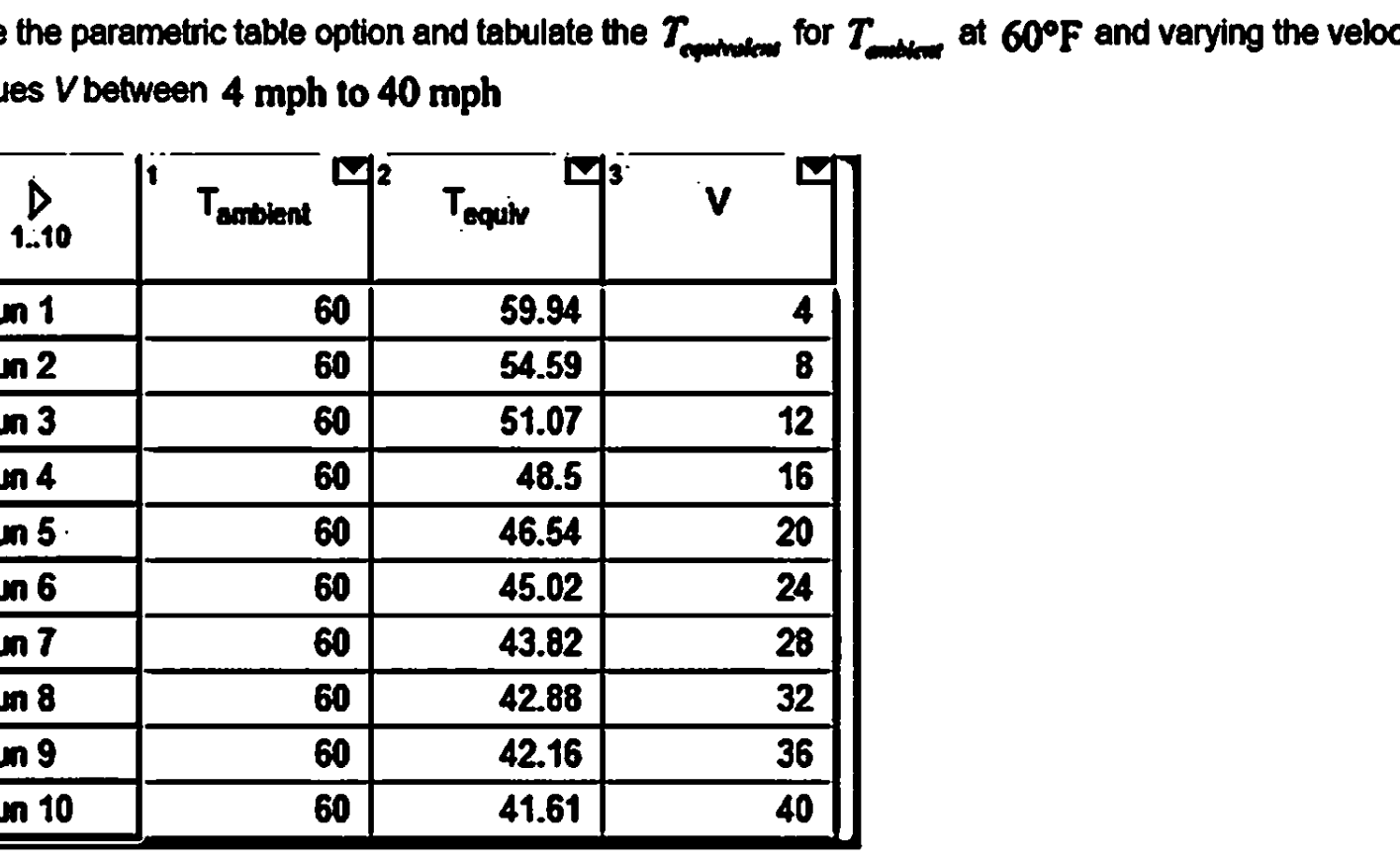
Use the parametric table option and tabulate the $T_{equivalent}$ for $T_{ambient}$ at **60°F** and varying the velocity values *V* between **4 mph to 40 mph**

▷ 1..10	1 $T_{ambient}$	2 T_{equiv}	3 V
Run 1	60	59.94	4
Run 2	60	54.59	8
Run 3	60	51.07	12
Run 4	60	48.5	16
Run 5	60	46.54	20
Run 6	60	45.02	24
Run 7	60	43.82	28
Run 8	60	42.88	32
Run 9	60	42.16	36
Run 10	60	41.61	40

Table (3)

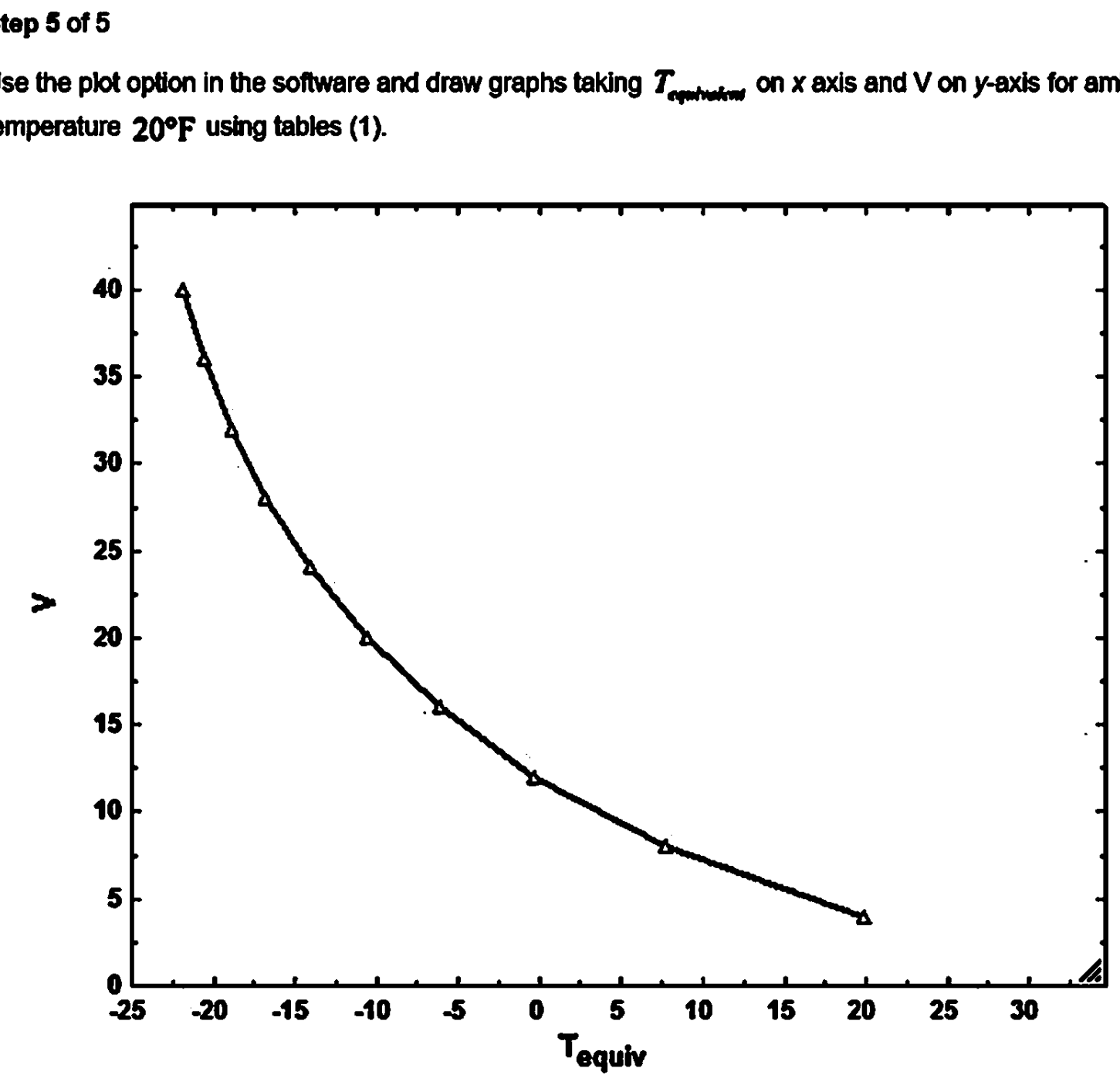
Step 5 of 5

Use the plot option in the software and draw graphs taking $T_{equivalent}$ on *x* axis and V on *y*-axis for ambient temperature **20°F** using tables (1).



Graph (1)

Use the overlay option and plot the curves at ambient temperature **40°F** using table (2) and at ambient temperature **60°F** using table (3) on the same graph.



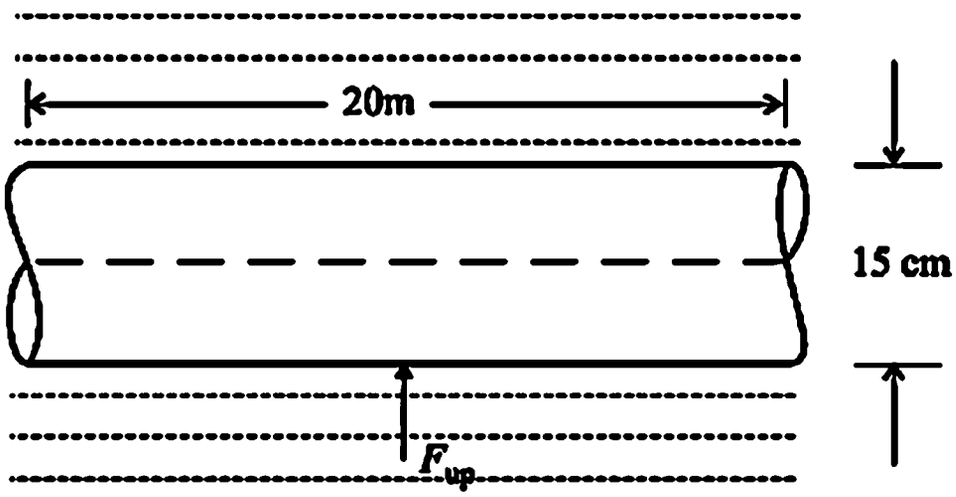
Graph (2)

With the increase in the ambient temperature, slope of the curve increases.

Question: 2.077P

An air-conditioning system requires a 20-m-long section of 15-cm diameter duct work to be laid underwater. Determine the upward force the water will exert on the duct. Take the densities of air and water to be 1.3 kg/m3 and 1000 kg/m3, respectively.

Step 1 of 3



Step 2 of 3

Given

Length of duct $h = 20\ \text{m}$

Diameter of duct $d = 15\ \text{cm}$

$= 0.15\ \text{m}$ $\quad 1\ \text{cm} = 10^{-2}\text{m}$

The density of air $\rho_{air} = 1.3\ \text{kg/m}^3$

The density of water $\rho_w = 1000\ \text{kg/m}^3$

g is accelerating due to gravity $\left(9.81\ \text{m/s}^2\right)$

V is the volume of immersed body

Step 3 of 3

Where ρ is density in which it is immersed

The upward force acting here is the buoyancy force

$F_{buoyancy} = \rho g V$

$$V = \frac{\pi}{4} D^2 L$$

$$\therefore \quad F_{up} = \rho_w g V$$

$$= 1000 \times 9.81 \times \frac{\pi}{4} \times (0.15)^2 \times 20$$

$$= 3467.14\ \text{N}$$

$$\boxed{F_{up} = 3.467\ \text{kN}}$$

Question: 2.078P

Balloons are often filled with helium gas because it weighs only about one-seventh of what air weighs under identical conditions. The buoyancy force, which can be expressed as $Fb=\rho$ airguballon will Pusn me balloon upward. If the balloon has a diameter of 12 m and carries two people, 85 kg each, determine the acceleration of the balloon when it is first released. Assume the density of air is ρ = 1.16 kg/m3, and neglect the cage

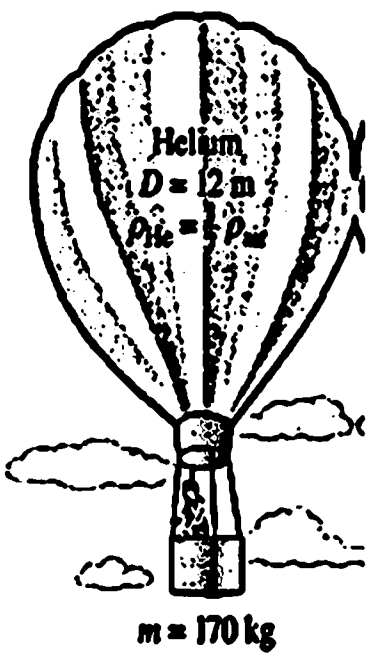


FIGURE P2–78

Step 1 of 5

Density of air $\rho_{air}=1.16\ \text{kg/m}^3$

Calculate the density of helium by using given relation between helium and air

$$\rho_{He}=\frac{1}{7}\rho_{air}$$

Substitute $1.16\ \text{kg/m}^3$ for ρ_{air}

$$\rho_{He}=\frac{1}{7}\rho_{air}$$
$$=\frac{1}{7}(1.16)$$
$$=0.166\text{kg/m}^3$$

Step 2 of 5

Calculate the volume of the spherical balloon

$$V_{balloon}=\frac{4}{3}\pi R^3$$

Here, R is the radius of the spherical balloon

Substitute 6m^3 for R

$$V_{balloon}=\frac{4}{3}\pi R^3$$
$$=\frac{4}{3}\pi(6)^3$$
$$=904.8\text{m}^3$$

Step 3 of 5

Calculate the buoyance force acting on balloon

$$F_B=\rho_{air}gV_{balloon}$$

Here, g is the acceleration due gravity

Substitute 9.81m/s^2 for g, 904.8m^3 for $V_{balloon}$

$$F_B=1.16\times9.81\times904.8$$
$$=10296.26\text{N}$$

Calculate the total mass of Helium,

$$m_{He}=\rho_{He}V$$

Substitute 0.166kg/m^3 for ρ_{He}, 9.81m/s^2 for g, 904.8m^3 for $V_{balloon}$

$$m_{He}=0.166\times904.8$$
$$=150.196\text{kg}$$

Calculate the total mass of the balloon by using the mass of the helium and mass of the people

$$m_{total}=m_{He}+m_{people}$$
$$=150.196+(2\times85)$$
$$=320.196\text{kg}$$

Now, total weight of the balloon

$$W=m_{total}\times g$$
$$=320.196\times9.81$$
$$=3141.130\text{N}$$

Step 4 of 5

Calculate the Net force acting on the balloon

$$F_{net}=W-F_B$$

Where F_{net} is the net force, F_B is the upward force due to buoyancy and W is the weight that pulls the balloon down

Substitute 10296.26N for F_B, 3141.130N for W

$$F_{net}=10296.26-3141.130$$
$$=7155.129\text{N}$$

Step 5 of 5

Calculate the net force

$$F_{net}=m_{total}\times a$$

Here, a is the acceleration of the balloon

Substitute 7155.129N for F_{net}

$$a=\frac{F_{net}}{m_{total}}$$
$$=\frac{7155.129}{320.196}$$
$$=22.346\text{m/s}^2$$

Thus, the acceleration of balloon is $\boxed{22.346\text{m/s}^2}$

Question: 2.079P

Reconsider Prob. 2–78. Using EES (or other) software, investigate the effect of the number of people carried in the balloon on acceleration. Plot the acceleration against the number of people, and discuss the results.

Step 1 of 4

"Given "
D=12 [m]
"N_person=2"
M_person=85[kg]
Rho_air=1.16[kg/m^3]
Rho_He=rho_air/7

Step 2 of 4

"Analysis"
g=9.81 [m/s^2]
V_ballon=pi*D^3/6
F_B=rho_air*g*V_ballon
M_he=rho_He*V_ballon
M_people=N_person*m_person
M_total=m_He+m_people
W=m_total*g
F_net=F_B-W
A+F_net/m_total

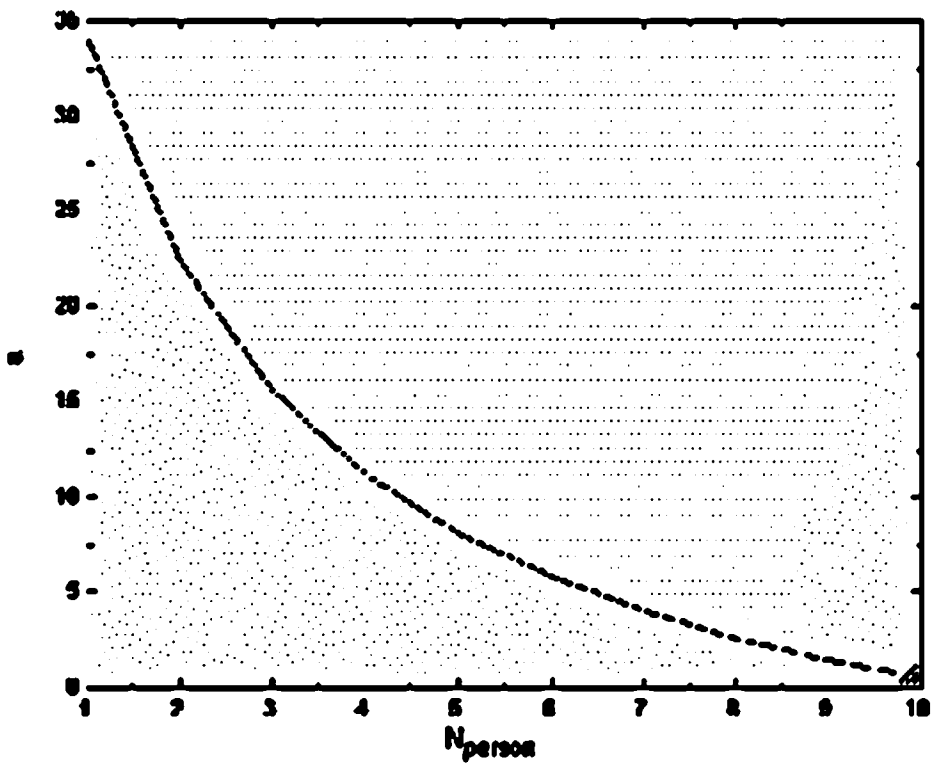
Step 3 of 4

From the menu bar, click the table, then select the new parameter table.
Then Run the table from 1 to 10.
Then click ok.

D 1..10	N_{person}	a
Run 1	1	34.02
Run 2	2	22.37
Run 3	3	15.62
Run 4	4	11.21
Run 5	5	8.093
Run 6	6	5.792
Run 7	7	4.011
Run 8	8	2.595
Run 9	9	1.443
Run 10	10	0.4867

Step 4 of 4

Graph:



Question: 2.080P

Determine the maximum amount of load, in kg, the balloon described in Prob. 2–78 can carry.

Step 1 of 5

Given

Density of air $\rho_{air} = 1.16 \text{ kg/m}^3$

Density of helium $\rho_{He} = \frac{1.16}{7} \text{ kg/m}^3$

$$\rho_{He} = 0.1657 \text{ kg/m}^3$$

Volume of balloon is $V_{balloon} = \frac{4}{3}\pi R^3$

$$V_{balloon} = \frac{4}{3} \times \pi \times 6^3$$

$$V_{balloon} = 904.32 \text{ m}^3$$

Step 2 of 5

Buoyancy force acting on the balloon is

$$F_B = \rho_{air} g V_{balloon}$$

$$F_B = 1.16 \times 9.81 \times 904.32$$

$$F_B = 10290.79 \text{ N}$$

Total mass of Helium,

$$m_{He} = \rho_{He} V$$

$$m_{He} = 0.1657 \times 904.32$$

$$m_{He} = 149.85 \text{ kg}$$

Step 3 of 5

Net force acting on the balloon is,

$F_{net} = W - F_B$

Where F_{net} is the net force

F_B is the upward force due to buoyancy

W is the weight that pulls the balloon down

Step 4 of 5

For maximum condition, the net force on balloon is zero. So,

$0 = W - F_B$

$W = F_B$

$m_{total} \times g = F_B$

$$m_{total} = \frac{F_B}{g}$$

$$m_{total} = \frac{10290.79}{9.81} \text{ kg}$$

$$m_{total} = 1049.01 \text{ kg}$$

Step 5 of 5

But total mass is due to mass of people and the mass of helium. Thus

$m_{total} = m_{people} + m_{He}$

$m_{people} = m_{total} - m_{He}$

$m_{people} = 1049.01 - 149.85$

$m_{people} = 899.16 \text{ kg}$

Hence mass of people is $\boxed{m_{people} = 899.16 \text{ kg}}$

Question:

2.081P

The lower half of a 10-m-high cylindrical container is filled with water (ρ = 1000 kg/m3) and the upper half with oil that has a specific gravity of 0.85. Determine the pressure difference between the top and bottom of the cylinder.

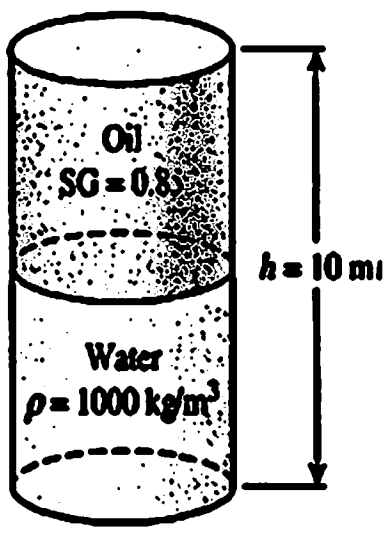


FIGURE P2–81

Question:

The lower half of a 10-m-high cylindrical container is filled with water (ρ = 1000 kg/m3) and the upper half with oil that has a specific gravity of 0.85. Determine the pressure difference between the top and bottom of the cylinder.

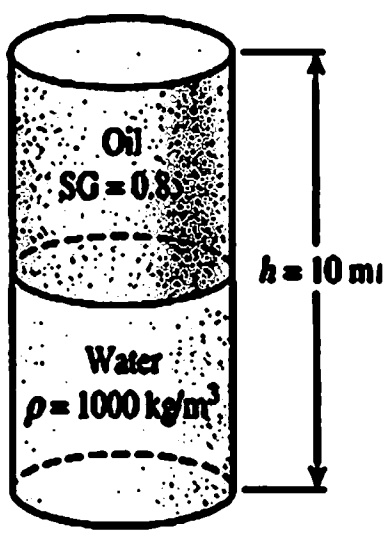


FIGURE P2–81

specific gravity of the oil $\times \rho_w = \rho_{oil}$

ρ_{oil} is the density of the oil and ρ_w is the density of water.

Substitute 0.85 for specific gravity of the oil and $1000\,^{kg}/_{m^3}$ for ρ_w.

$$\rho_{oil} = 0.85 \times 1000$$
$$= 850\,kg/m^3$$

Calculate the pressure exerted by the oil.

Write the formula for the pressure acting at a certain datum.

$$\Delta p = \rho g h$$

$$\Delta p = (\rho h)_{oil}\, g$$

Here, ρ_{oil} is the density of the oil, g is the gravity and h_{oil} is the height of the oil from the datum.

Substitute $850\,kg/m^3$ for ρ_{oil} and 9.81 for g and 5 m for h_{oil}.

$$\Delta p = (850 \times 5)_{oil} \times 9.81$$
$$= 41{,}692.5\ kPa$$

Step 3 of 3

Calculate the pressure exerted by the water.

Write the formula for the pressure acting at a certain datum.

$$\Delta p = \rho g h$$

$$\Delta p = (\rho h)_{water}\, g$$

Here, ρ_w is the density of the water, g is the gravity and h_w is the height of the water from the datum.

Substitute $1000\,kg/m^3$ for ρ_w and 9.81 for g and 5 m for h_w.

$$\Delta p = (1000 \times 5)_{oil} \times 9.81$$
$$= 49{,}050\ kPa$$

Calculate the pressure difference between the top and the bottom of the cylinder.

The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids.

$$\Delta p = (\rho g h)_{oil} + (\rho g h)_{water}$$

Substitute 41,692.5 kPa for the value $(\rho g h)_{oil}$ and 49,050 kPa for the value $(\rho g h)_{water}$.

$$\Delta P = (41{,}692.5 + 49{,}050)$$
$$= 90{,}742.5\,N/m^2$$
$$= 90.7\ kPa$$

Therefore, the pressure difference Δp between the top and the bottom of the cylinder is $\boxed{90.74\,kPa}$.

Question: 2.082P

A vertical, frictionless piston-cylinder device contains a gas at 180 kPa absolute pressure. The atmospheric pressure outside is 100 kPa, and the piston area is 25 cm2. Determine the mass of the piston.

Step 1 of 4

Given

Pressure on the piston $P = 180\ \text{kPa}$

Atmospheric pressure on piston $P_{atm} = 100\ \text{kPa}$

Area of the piston is $A = 25\ \text{cm}^2$

$A = 25 \times 10^{-4}\ \text{m}^2$

Acceleration due to gravity $g = 9.81\ \text{m/s}^2$

Let mass of the piston be m

Step 2 of 4

Free body diagram of the piston

Question:

A vertical, frictionless piston-cylinder device contains a gas at 180 kPa absolute pressure. The atmospheric pressure outside is 100 kPa, and the piston area is 25 cm2. Determine the mass of the piston.

Step 1 of 4

Given

Pressure on the piston $P = 180\ \text{kPa}$

Atmospheric pressure on piston $P_{atm} = 100\ \text{kPa}$

Area of the piston is $A = 25\ \text{cm}^2$

$A = 25 \times 10^{-4}\ \text{m}^2$

Acceleration due to gravity $g = 9.81\ \text{m/s}^2$

Let mass of the piston be m

Step 2 of 4

Free body diagram of the piston

$m \times 9.81 = 80 \times 2.5$

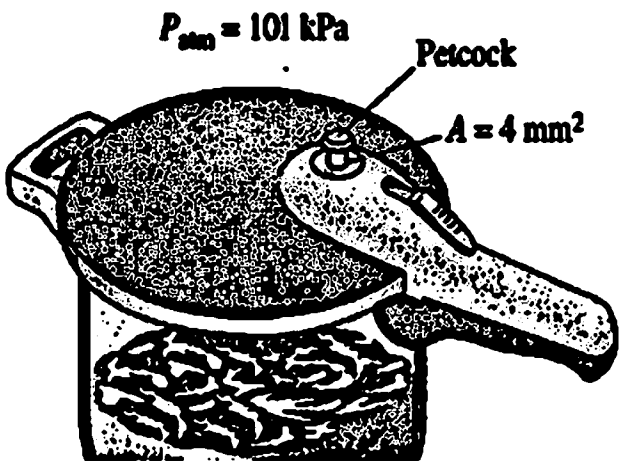
$$m = \frac{200}{9.81}$$

$m = 20.39\ \text{kg}$

Hence the mass of the piston is $\boxed{m = 20.39\ \text{kg}}$

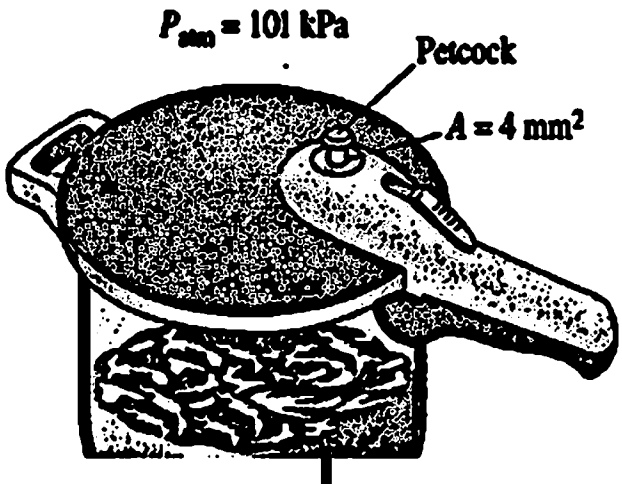
Question: 2.083P

A pressure cooker cooks a lot faster than an ordinary pan by maintaining a higher pressure and temperature inside. The lid of a pressure cooker is well sealed, and steam can escape only through an opening in the middle of the lid. A separate metal piece, the petcock, sits on top of this opening and prevents steam from escaping until the pressure force overcomes the weight of the petcock. The periodic escape of the steam m this manner prevents any potentially dangerous pressure buildup and keeps the pressure inside at a constant value. Determine the mass of the petcock of a pressure cooker whose operation pressure is 100 kPa gage and has an opening cross-sectional area of 4 mm2. Assume an atmospheric pressure of 101 kPa, and draw the free-body diagram of the petcock.



Question:

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Step 2 of 2

The weight of the petcock acts downwards and the pressure inside the cooker will act upwards. So the weight of petcock (W) is balanced by gage pressure inside the cooker.

$$W = P_{gage} \times A$$

$$mg = P_{gage} \times A$$

$$m = \frac{P_{gage} \times A}{g}$$

Here the mass of petcock is m

the gage pressure acting inside the cooker is P_{gage}

the cross sectional area of opening is A



Step 2 of 2

Question:

2.084P

A glass tube is attached to a water pipe, as shown in Fig. P2-84. If the water pressure at the bottom of the tube is 120 kPa and the local atmospheric pressure is 99 kPa, determine how high the water will rise in the tube, in m. Take the density of water to be 1000 kg/m3.

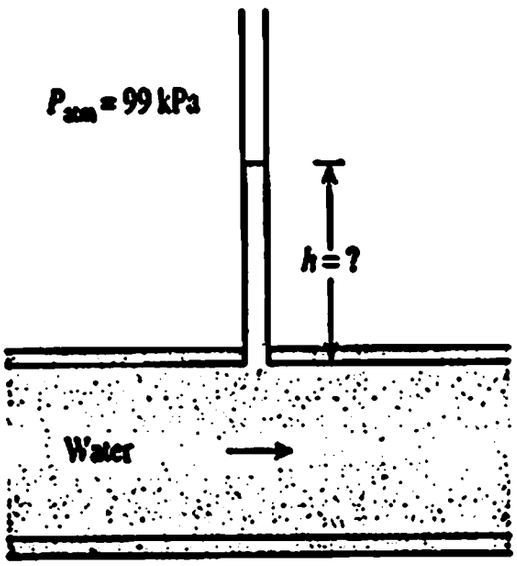


FIGURE P2–84

Question:

A glass tube is attached to a water pipe, as shown in Fig. P2-84. If the water pressure at the bottom of the tube is 120 kPa and the local atmospheric pressure is 99 kPa, determine how high the water will rise in the tube, in m. Take the density of water to be 1000 kg/m3.

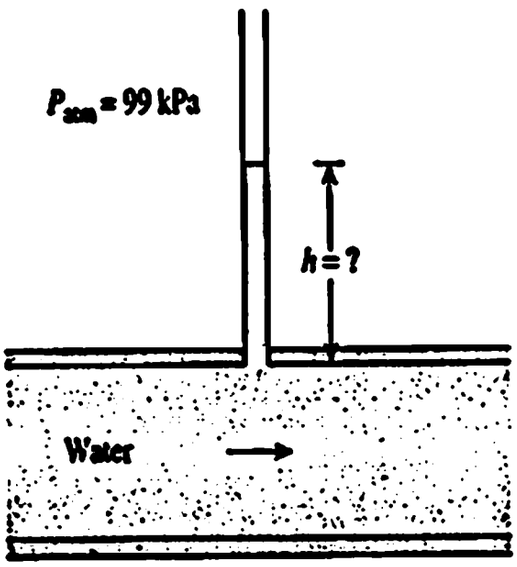


FIGURE P2–84

Step 1 of 1

Calculate the height the water will raise in the tube.

Therefore, the height the water will raise in the tube is $\boxed{2.140\,\text{m}}$.

Question: 2.085P

When measuring small pre ssure differences with a manometer, often one arm of the manometer is inclined to improve the accuracy of reading. (The pressure difference is still proportional to the *vertical* distance and not the actual length of the fluid along the tube.) The air pressure in a circular duct is to be measured using a manometer whose open arm is inclined 45° from the horizontal, as shown in Fig. P2-85 The density of the liquid in the manometer is 0.81 kg/L, and the vertical distance between the fluid levels in the two arms of the manometer is 12 cm. Determine the gage pressure of air in the duct and the length of the fluid column in the inclined arm above the fluid level in the vertical arm.

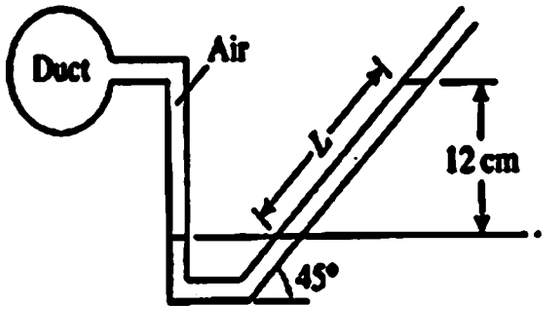


FIGURE P2–85

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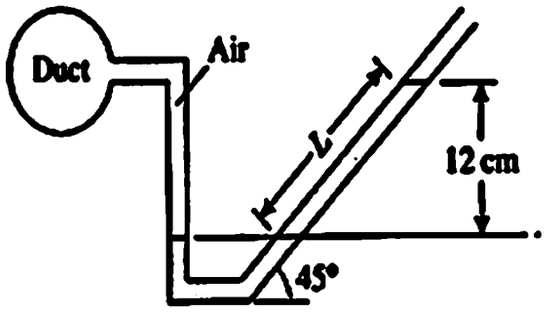


FIGURE P2–85

Step 1 of 2

Question: 2.086P

Consider a U-tube whose arms are open to the atmosphere. Now equal volumes of water and light oil (ρ = 49.3 lbm/ft3) are poured from different arms. A person blows from the oil side of the U-tube until the contact surface of the two fluids moves to the bottom of the U-tube, and thus the liquid levels in the two arms are the same. If the fluid height in each arm is 30 in, determine the gage pressure the person exerts on the oil by blowing.

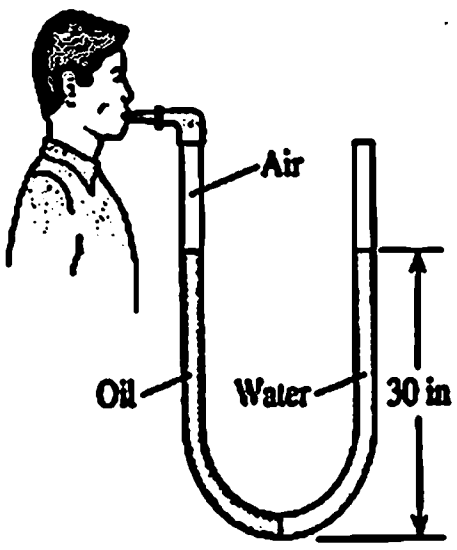
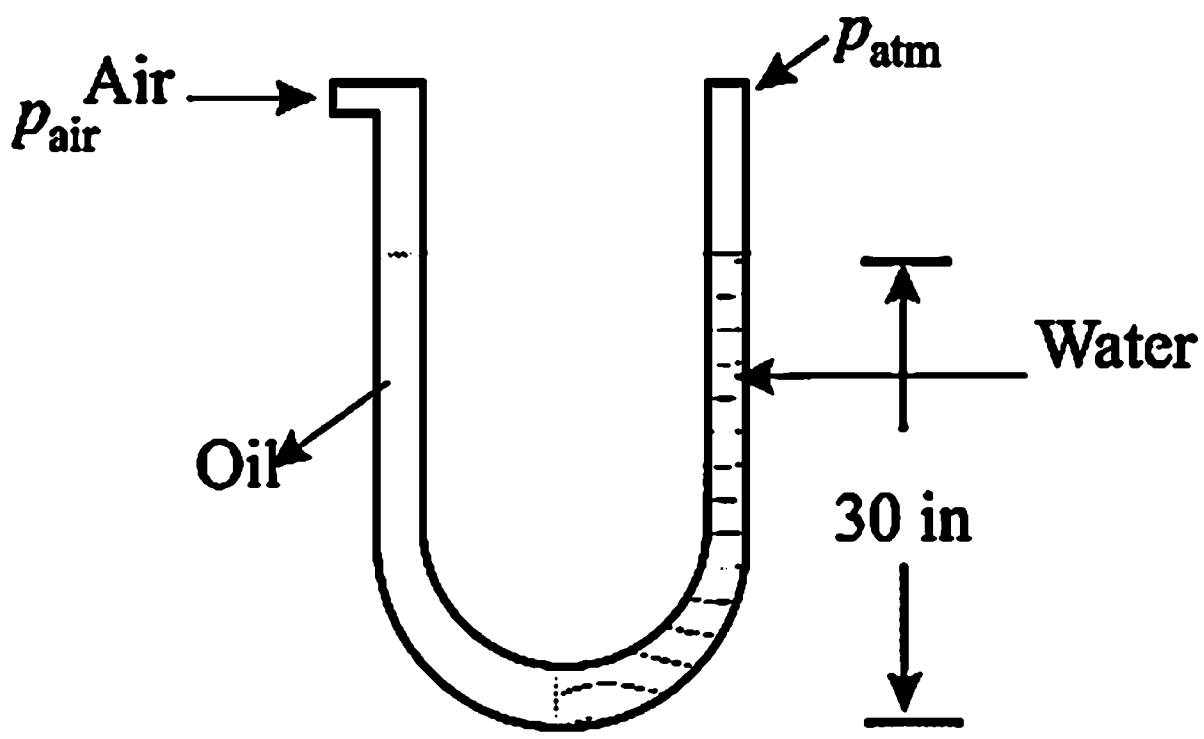


FIGURE P2–86E

Step 1 of 3



Step 2 of 3

The atmospheric pressure $p_{atm} = 14.3\text{Psi}$

The density of water $\rho_w = 62.4\ \text{lbm/ft}^3$

The density of oil $\rho_{oil} = 49.3\text{lbm/ft}^3$

The height of oil and water $h_{oil} = hw = 30\ \text{in}$

Step 3 of 3

In order to balance the level of two fluids in the manometer the pressure of air to atmospheric pressure is related as

$$p_{blow} + (\rho gh)_{oil} = p_{atm} + (\rho gh)_{water}$$

The acceleration due to gravity of earth $g = 32.174\text{ft/s}^2$

1ft=12in

$$1\text{lbf} = \frac{1}{32.174}\text{lbm ft/s}^2$$

$$p_{gage,blow} = p_{blow} - p_{atm}$$

$$p_{gage,blow} = (\rho_w - \rho_{oil})gh$$

$$p_{gage,blow} = (62.4 - 49.3)32.174 \times \frac{30}{12} \times \frac{1}{32.174} \times \frac{1}{(144)}$$

$$\boxed{p_{gage,blow} = 0.227\text{Psi}}$$

Question: 2.087P

Intravenous infusions are usually driven by gravity by hanging the fluid bottle at sufficient height to counteract the blood pressure in the vein and to force the fluid into the body. The higher the bottle is raised, the higher the flow rate of the fluid will be. (*a*) If it is observed that the fluid and the blood pressures balance each other when the bottle is 80 cm above the arm level, determine the gage pressure of the blood, (*b*) If the gage pressure of the fluid at the arm level needs to be 15 kPa for sufficient flow rate, determine how high the bottle must be placed. Take the density of the fluid to be 1020 kg/m3.

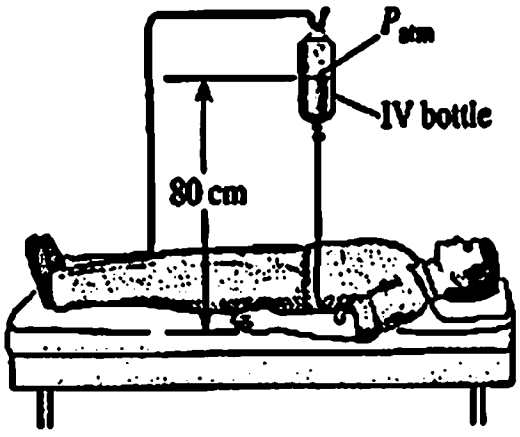


FIGURE P2–87

Step 1 of 3

Note the following data from the problem.

Density of the fluid is $\rho = 1020\ \text{kg/m}^3$

Height of bottle is $h = 80\ \text{cm}$

Step 2 of 3

(a)

Calculate the gage pressure of the blood using the relation,

$$P_{gauge,arm} = (\rho g h)_{air\ bottle}$$

Substitute 80 cm for h, $1020\ \text{kg/m}^3$ for ρ , and $9.81\ \text{m/s}^2$ for g.

$$\begin{aligned} P_{gauge,arm} &= \left(1020\ \frac{\text{Kg}}{\text{m}^3}\right)\left(9.81\frac{\text{m}}{\text{s}^2}\right)(0.8\ \text{m}) \\ &= 8004.96\ \text{Pa} \\ &= 8.00496\ \text{kPa} \end{aligned}$$

Gauge Pressure in the arm is $\boxed{P_{gauge,arm} = 8.00496\ \text{kPa}}$.

Step 3 of 3

(b)

If the Pressure in the arm level needs to be 15 kPa, calculate the height using the following relation.

$$P_{gauge,arm} = \rho g h_{arm-bottle}$$

$$h_{arm-bottle} = \frac{P_{gauge,arm}}{\rho g}$$

Substitute 15 kPa for $P_{gauge,arm}$, $1020\ \text{kg/m}^3$ for ρ , and $9.81\ \text{m/s}^2$ for g.

$$\begin{aligned} h_{arm-bottle} &= \frac{15\times 1000}{1020\times 9.81} \\ &= 1.5\ \text{m} \end{aligned}$$

Height of bottle from arm level is $\boxed{h_{arm-bottle} = 1.5\ \text{m}}$

Question: 2.088P

A water pipe is connected to a double-U manometer as shown in Fig. P2-88E at a location where the local atmospheric pressure is 14.2 psia. Determine the absolute pressure at the center of the pipe.

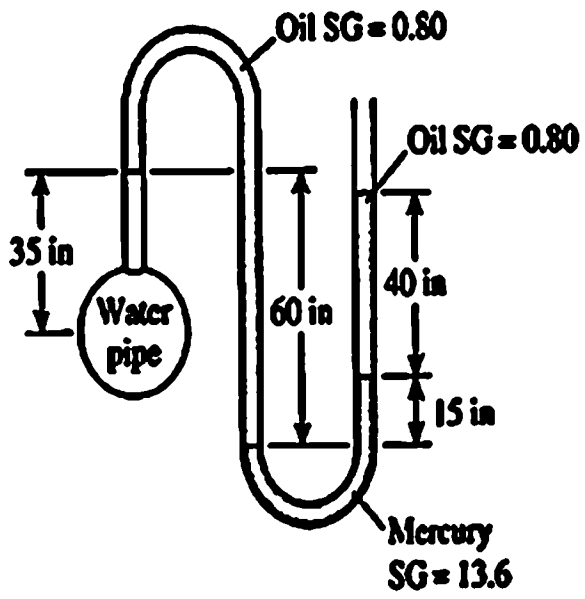
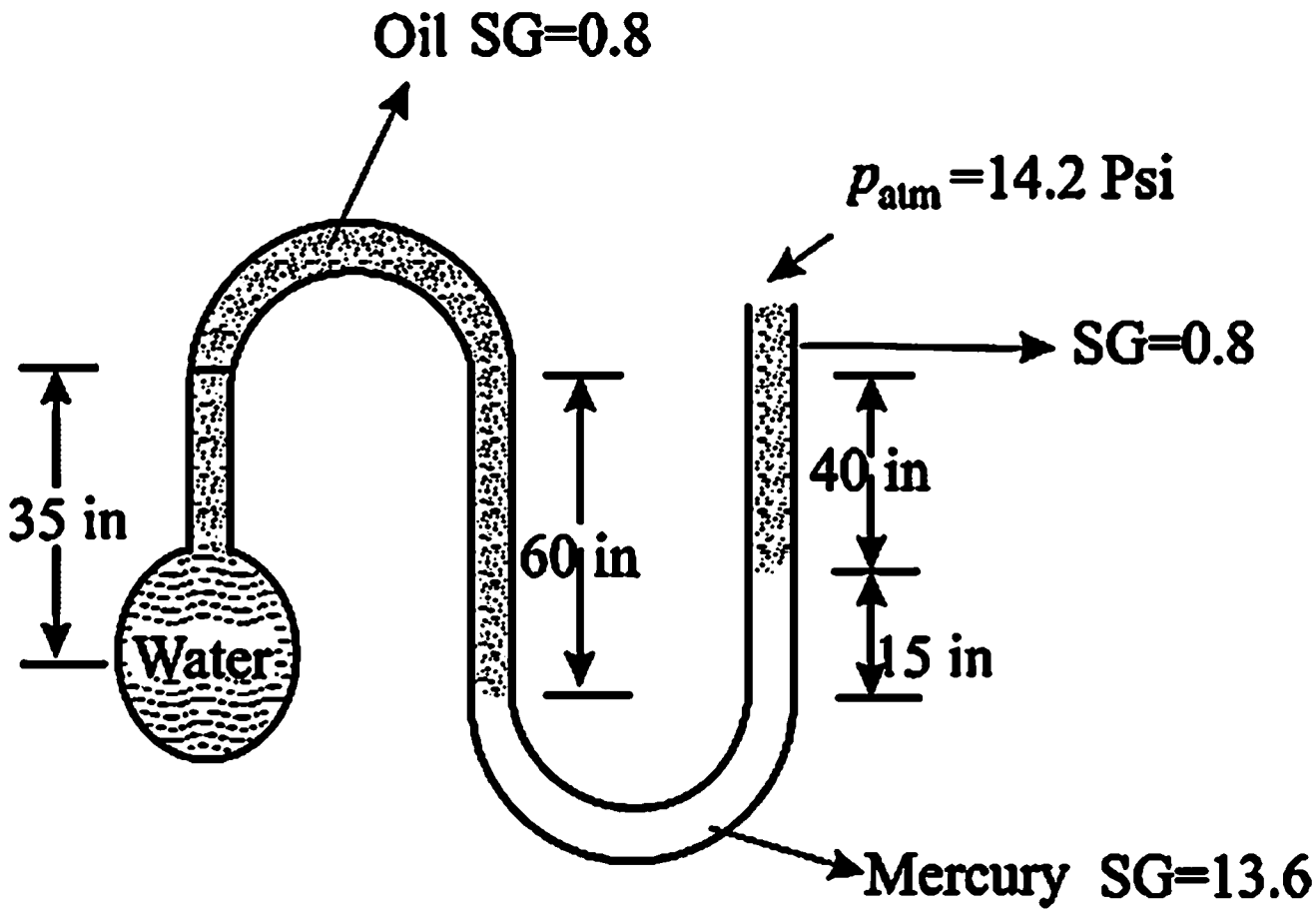


FIGURE P2–88E

Step 1 of 4



Step 2 of 4

Given

The local atmospheric pressure $p_{atm} = 14.2 \text{psia}$

Density of water $\rho_w = 62.4 \text{lbm/ft}^3$

Specific gravity of oil S.G=0.8

The height of oil $h_{1oil} = 40 \text{in}$

The height of mercury $h_{Hg} = 15 \text{in}$

The height of oil $h_{2oil} = 60 \text{in}$

The height of water $h_w = 35 \text{in}$

Step 3 of 4

The density of oil $\rho_{oil} = s.G \times \rho_w$

$$\rho_{oil} = 0.8 \times 62.4$$

$$\rho_{oil} = 49.92 \text{lbm/ft}^3$$

The acceleration due to gravity of earth $g = 32.174 \text{ft/s}^2$

Specific gravity of mercury $H_g = 13.6$ (mercury)

The density of oil $\rho_{H_g} = S.G \times \rho_w$

$$\rho_{Hg} = 13.6 \times 62.4$$

$$\rho_{Hg} = 848.64 \text{lbm/ft}^3$$

Step 4 of 4

The relation between absolute pressure and atmospheric pressure

$$p_{abs} = p_{atm} + \rho g h$$

The height column 'h'is taken positive while giving downwards and negative upwards.

In our problem, the absolute pressure at the center of pipe

$$p_{abs} = p_{atm} + (\rho_{oil} g h_{oil})_1 + \rho_{Hg} g h_{Hg} - (\rho_{oil} g h_{oil})_2 + \rho_w g h_w$$

$$\Rightarrow P_{abs} = 14.2 + (49.92 \times 40 + 848.64 \times 15 - 49.92 \times 60 + 624 \times 35) g$$

$$P_{abs} = 14.2 + (13915.2) g \cdot \frac{\text{lbm}}{\text{ft}^3} \cdot \text{in} \times \frac{1}{g} \frac{\text{ft}^3}{\text{in}^3 (12)^3}$$

$$P_{abs} = 14.2 + 8.054 \left(\frac{\text{lbf}}{\text{in}^2} \right)$$

$$\boxed{P_{abs} = 22.26 \text{Psi}}$$

Question: 2.089P

The average atmospheric pressure on earth is approximated as a function of altitude by the relation *p*atm=101.325 (1–0.02256*z*)5.256, where *P*atm is the atmospheric pressure in kPa and *z*is the altitude in km with $z = 0$ at sea level. Determine the approximate atmospheric pressures at Atlanta (z =306 m), Denver (z = 1610 m), Mexico City (z =2309 m), and the top of Mount Everest (z =8848 m).

Step 1 of 5

The approximate relation of atmospheric pressure is given by

$$P_{atm} = 101.325(1 - 0.02256z)^{5.256}$$

Where z is the altitude in km

P_{atm} is the atmospheric pressure in kPa

Step 2 of 5

Altitude of Atlanta is $z = 306$ m

$z = 0.306$ km

Therefore

$$P_{atm} = 101.325[1 - 0.02256(0.306)]^{5.256} \text{ kPa}$$

$$P_{atm} = 97.70 \text{ kPa}$$

Atmospheric Pressure at Atlanta is $\boxed{P_{atm} = 97.70 \text{ kPa}}$

Step 3 of 5

Altitude of Denver is $z = 1610$ m

$z = 1.61$ km

Therefore

$$P_{atm} = 101.325[1 - 0.02256(1.61)]^{5.256} \text{ kPa}$$

$$P_{atm} = 83.38 \text{ kPa}$$

Atmospheric Pressure at Denver is $\boxed{P_{atm} = 83.38 \text{kPa}}$

Step 4 of 5

Altitude of Mexico city is $z = 2309$ m

$z = 2.309$ km

Therefore

$$P_{atm} = 101.325[1 - 0.02256(2.309)]^{5.256} \text{ kPa}$$

$$P_{atm} = 76.53 \text{ kPa}$$

Atmospheric Pressure at Mexico city is $\boxed{P_{atm} = 76.53 \text{kPa}}$

Step 5 of 5

Altitude of Mount Everest is $z = 8848$ m

$z = 8.848$ km

Therefore

$$P_{atm} = 101.325[1 - 0.02256(8.848)]^{5.256} \text{ kPa}$$

$$P_{atm} = 31.36 \text{ kPa}$$

Atmospheric Pressure at Mount Everest is $\boxed{P_{atm} = 31.36 \text{kPa}}$

CHAPTER 03

SOLUTIONS

Question: 3.001P

What is the difference between the macroscopic and microscopic forms of energy?

Step 1 of 1

The macroscopic forms of energy are those a system possesses as a whole with respect to some outside reference frame, such as kinetic and potential energies.

The microscopic forms of energy are those related to the molecular structure of a system and the degree of the molecular activity, and they are independent of outside reference frames. The sum of all the microscopic forms of energy is called the internal energy of a system and is denoted by U.

Question: 3.002P

What is total energy? Identify the different forms of energy that constitute the total energy.

Step 1 of 5

Energy can exist in numerous forms such as thermal, mechanical, kinetic, potential, electric, magnetic, chemical, and nuclear, and their sum constitutes the total energy E of a system. The total energy of a system on a unit mass basis is denoted by '*e*' and is expressed as

$$e = \frac{E}{m} \quad (\text{kJ/kg})$$

Step 2 of 5

The magnet, electric, and surface tension effects are significant in some specialized cases only and are usually ignored. In the absence of such effects, the total energy of the system consists of the kinetic, potential, and internal energies and is expressed as

$$E = U + KE + PE \quad (\text{kJ})$$

Or, on a unit mass basics,

$$e = u + ke + pe \quad (\text{kJ/kg})$$

Step 3 of 5

Where KE is called as the energy that a system possesses as a result of its motion relative to some reference frame is called kinetic energy (KE).The kinetic energy is expressed as

$$KE = m\frac{V^2}{2} \quad (\text{kJ})$$

Step 4 of 5

PE is the called as potential energy, the energy that a system possesses as a result of its elevation in a gravitational field is called potential energy (PE) and is expressed as

$$PE = mgz \quad (\text{kJ})$$

Step 5 of 5

U is known as internal energy of a system.
Therefore the total energy of the system is expressed as

$$E = U + KE + PE \quad (\text{kJ})$$

$$\boxed{E = U + m\frac{V^2}{2} + mgz \quad (\text{kJ})}$$

Question: 3.003P

How are heat, internal energy, and thermal energy related to each other?

Step 1 of 1

Heat is defined as the form of energy that is transferred between two systems (or a system and its surroundings) by virtue of a temperature difference.
Heat is energy in transition. It is recognized only as it crosses the boundary of a system.

The sum of all the microscopic forms of energy is called the internal energy of a system and is denoted by U.

Thermal energy is generated and measured by heat of any kind. In daily life, we frequently refer to the sensible and latent forms of internal energy as heat, and we talk about heat content of bodies. In thermodynamics, however, we usually refer to those forms of energy as thermal energy to prevent any confusion with heat transfer.

Question: 3.004P

What is mechanical energy? How does it differ from thermal energy? What are the forms of mechanical energy of a fluid stream?

Step 1 of 3

The mechanical energy can be defined as the form of energy that can be converted to mechanical work completely and directly by an ideal mechanical device such as an ideal turbine. Kinetic and potential energies are the familiar forms of mechanical energy.

Step 2 of 3

Thermal energy is not mechanical energy, however, since it cannot be converted to work directly and completely, where mechanical energy converts to work directly.

Step 3 of 3

A pump transfer's mechanical energy to a fluid by raising its pressure, and a turbine extracts mechanical energy from a fluid by dropping its pressure. Therefore, the pressure of a flowing fluid is also associated with its mechanical energy. The pressure itself is not a form of energy. But a pressure force acting on a fluid through a distance produces work, called flow work.

Flow work is expressed in terms of fluid properties, and it is convenient to view it as part of the energy of a flowing fluid and call it flow energy. Therefore, the mechanical energy of a flowing fluid can be expressed on a unit mass basis as

$$e_{mech} = \frac{P}{\rho} + \frac{V^2}{2} + gz \quad (\text{kJ/kg})$$

Where $\frac{P}{\rho}$ is the flow energy, $\frac{V^2}{2}$ is the kinetic energy, and gz is the potential energy of the fluid, all per unit mass.

Question: 3.005P

Natural gas, which is mostly methane CH4, is a fuel and a major energy source. Can we say the same about hydrogen gas, H2?

Step 1 of 1

Since it can be burned, Hydrogen is also a fuel. It is not an energy source as there are no hydrogen reserves in the world. Water is the energy source which provides hydrogen. Hydrogen obtained from water can be used as fuel for cars and generators. Therefore hydrogen is more of an energy carrier than an energy source.

Question: 3.006P

Calculate the total kinetic energy, in Btu, of an object with a mass of 15 lbm when its velocity is 100 ft/s.

Step 1 of 2

Given

Mass of the object $m = 15$ lbm

Velocity of object, $V = 100$ ft/s

But 1 lbm $= 0.4536$ kg

And 1 ft $= 0.3048$ m

Hence $m = 15 \times 0.4536$ kg

$m = 6.804$ kg

$V = 100 \times 0.3048$ m/s

$V = 30.48$ m/s

Step 2 of 2

Kinetic energy of an object,

$$KE = \frac{1}{2}mv^2$$

$$KE = \frac{1}{2}(6.804\text{ kg}) \times (304.8\text{ m/s})^2$$

$$KE = 3160.56\text{ J}$$

$$KE = 3.16056\text{ kJ}$$

But 1 Btu $= 1.05435$ KJ

Hence $KE = 3.16056\text{ kJ} \times \frac{1\text{ Btu}}{1.05435\text{ KJ}}$

$KE = 2.9976$ Btu

$KE = 3$ Btu

Hence Kinetic energy of an object is $\boxed{KE = 3\text{ Btu}}$

Question: 3.007P

Calculate the total kinetic energy, in kJ, of an object whose mass is 100 kg and whose velocity is 20 m/s.

Step 1 of 1

Calculate the kinetic energy of the object using the equation,

$$KE = \frac{1}{2}mV^2$$

Here, m is mass of the object and V is velocity of the object.

Substitute 100 kg for m and $20\ \text{m/s}$ for V.

$$KE = \frac{1}{2} \times 100 \times 20^2$$

$$= 20 \times 10^3\ \text{J} \left| \frac{1\ \text{kJ}}{10^3\ \text{J}} \right|$$

$$= 20\ \text{kJ}$$

Therefore, the kinetic energy of the object is $\boxed{20\ \text{kJ}}$.

Question: 3.008P

The specific potential energy of an object with respect to some datum level is given by gz where g is the local gravitational acceleration and z is the elevation of the object above the datum. Determine the specific potential energy, in Btu/lbm, of an object elevated 100 ft above a datum at a location where g = 32.1 ft/s2.

Step 1 of 1

Find the specific potential energy of an object by using the following equation:

$$(pe) = gz$$

Here, g is acceleration due to gravity and z is the elevation of the object above the datum.

Substitute $32.2\ ft/s^2$ for g and 100 ft for z.

$$pe = 32.2 \times 100$$
$$= 3220\ ft^2/s^2$$

Convert potential energy from ft^2/s^2 to Btu/lbm by using conversion formula.

$$pe = 3220\ ft^2/s^2 \times \frac{1\ Btu/lbm}{25037\ ft^2/s^2}$$
$$= 0.128\ Btu/lbm$$

Therefore, the specific potential energy of an object is $\boxed{0.128\ Btu/lbm}$

Question: 3.009P

Calculate the total potential energy, in Btu, of an object with a mass of 200 lbm when it is 10 ft above a datum level at a location where standard gravitational acceleration exists.

Step 1 of 2

Mass of an object $m = 200$ lbm

Height of object $z = 10$ ft

Acceleration due to gravity is $g = 9.81\ \text{m/s}^2$

But $1\ \text{lbm} = 0.4536\ \text{kg}$

And $1\ \text{ft} = 0.3048\ \text{m}$

Hence $m = 200 \times 0.4536\ \text{kg}$

$m = 90.72\ \text{kg}$

$z = 10 \times 0.3048\ \text{m}$

$z = 3.048\ \text{m}$

Step 2 of 2

Potential energy possessed by an object

$PE = mgz$

$PE = 90.72 \times 9.81 \times 3.048$

$PE = 2712.61\ \text{J}$

$PE = 2.71261\ \text{kJ}$

But $1\ \text{Btu} = 1.05435\ \text{kJ}$

$$PE = 2.71261\ \text{kJ} \times \frac{1\ \text{Btu}}{1.05435\ \text{kJ}}$$

$PE = 2.573\ \text{Btu}$

Hence Potential energy is $\boxed{PE = 2.573\ \text{Btu}}$

Question: 3.010P

Calculate the total potential energy, in kJ, of an object whose mass is 20 kg when it is located 20 m below a datum level in a location where g =9.5 m/s2.

Step 1 of 1

The total potential energy possessed by a body can be calculated with the help of the following formula:

$$PE = mgz$$

Here, m is the mass of an object, z is the datum height of the object and g is the acceleration due to gravity

Since the object is located 20 m below the datum line, the value of z will be $-20\,\text{m}$.

Substitute 20 kg for m, -20 m for z and 9.5 m/s^2 for g.

$$\begin{aligned} PE &= 20(-20)(9.5) \\ &= -3800\ \text{J} \\ &= -3.8\ \text{kJ} \end{aligned}$$

Hence, the potential energy is $\boxed{-3.8\ \text{kJ}}$

Question: 3.011P

A person gets into an elevator at the lobby level of a hotel together with his 30-kg suitcase, and gets out at the 10th floor 35 m above. Determine the amount of energy consumed by the motor of the elevator that is now stored in the suitcase.

Step 1 of 2

Given data:

The mass of a suitcase, $m = 30$ kg

The height up to which he climbed in elevator, $h = 35$ m

Assume acceleration due to gravity, $g = 9.8\ m/s^2$

Step 2 of 2

The energy stored in the suitcase is in the form of potential energy due to the elevation which is given by

$$PE = mgh$$
$$= 30 \times 9.8 \times 35$$
$$= 10290\ J$$
$$\boxed{PE = 10.3\ kJ}$$

The suitcase posses 10.3 kJ of energy at 10^{th} floor compared to 1^{st}.

Question: 3.012P

Electric power is to be generated by installing a hydraulic turbine-generator at a site 160 m below the free surface of a large water reservoir that can supply water at a rate of 3500 kg/s steadily. Determine the power generation potential.

Step 1 of 1

Calculate the power generation potential of the hydraulic turbine using the following formula:

$$P = \dot{m} \times g \times z$$

Here, $\dot{m}$ is the mass flow rate of water, z is the height of the water reservoir, g is the and acceleration due to gravity.

Substitute $3500\,\text{kg/s}$ for $\dot{m}$, $9.81\,\text{m/s}^2$ for g, $160\,\text{m}$ for z

$$\begin{aligned} P &= 3500 \times 9.81 \times 160 \\ &= 5.493 \times 10^6\ \text{W} \\ &= 5.493\ \text{MW} \end{aligned}$$

Hence power generation potential is $\boxed{5.493\ \text{MW}}$

Question: 3.013P

At a certain location, wind is blowing steadily at 10 m/s. Determine the mechanical energy of air per unit mass and the power generation potential of a wind turbine with 60-m-diameter blades at that location. Take the air density to be 1.25 kg/m3.

Step 1 of 5

Given data:

The average velocity of wind, $(V) = 10 \text{ m/s}$

The diameter of the wind turbine, $(d) = 60 \text{ m}$

The density of air, $(\rho) = 1.25 \text{ kg/m}^3$

Step 2 of 5

The mechanical energy of a flowing fluid can be expressed on a unit mass basis as

$$e_{mech} = \frac{P}{\rho} + \frac{V^2}{2} + gz \quad \text{in (kJ/kg)}$$

Where $\frac{P}{\rho}$ is the flow energy,

$\frac{V^2}{2}$ is the kinetic energy,

and gz is the potential energy of the fluid, all per unit mass.

Step 3 of 5

Here flow energy and potential energy can be neglected as it not mentioned.

Therefore total mechanical energy of wind turbine per unit mass

$$e_{mech} = \frac{10^2}{2} \text{ J/kg}$$

$$= 50$$

$$= \frac{50}{1000} \text{ kJ/kg}$$

$$= 0.05 \text{ kJ/kg}$$

$\therefore$ Mechanical energy $\boxed{e_{mech} = 0.05 \text{ kJ/kg}}$

Step 4 of 5

The power generation potential of the wind turbine is obtained by expressing in rate form of total mechanical energy

$$E_{mech} = \dot{m} e_{mech}$$

Where $\dot{m}$ is the mass flow rate of the fluid, it can be expressed as

$$\dot{m} = \rho A V \text{ kg/s}$$

Where the density of air is $(\rho) = 1.25 \text{ kg/m}^3$

Area of wind turbine $A = \frac{\pi}{4} d^2 \text{ m}^2$

Mass flow rate $\dot{m} = \rho \left(\frac{\pi}{4} d^2 \right) V \text{ kg/s}$

$$\dot{m} = (1.25) \times \left(\frac{\pi}{4} \times 60^2 \right) \times (10) \text{ kg/s}$$

$$\dot{m} = 35{,}342.9 \text{ kg/s}$$

Step 5 of 5

Therefore power generation of the wind turbine is calculated by

$$= 35342.9 \times 0.05 \text{ kJ/s}$$

$$= 1767.145 \text{ kW}$$

$\therefore$ Power generation of the wind turbine, $\boxed{E_{mech} = 1767.145 \text{ kW}}$

3.014P

Question:

A water jet that leaves a nozzle at 60 m/s at a flow rate of 120 kg/s is to be used to generate power by striking the buckets located on the perimeter of a wheel. Determine the power generation potential of this water jet.

Step 1 of 3

Given data:

The velocity of water jet $(V) = 60\ m/s$

Mass flow rate of water $\dot{m} = 120\ kg/s$

Step 2 of 3

The mechanical energy of a flowing fluid can be expressed on a unit mass basis as

$$e_{mech} = \frac{P}{\rho} + \frac{V^2}{2} + gz \text{ in (kJ/kg)}$$

Where P is the flow energy

Question:

A water jet that leaves a nozzle at 60 m/s at a flow rate of 120 kg/s is to be used to generate power by striking the buckets located on the perimeter of a wheel. Determine the power generation potential of this water jet.

Step 1 of 3

Given data:

The velocity of water jet $(V) = 60\ m/s$

Mass flow rate of water $\dot{m} = 120\ kg/s$

Step 2 of 3

3.015P

Question:

Two sites are being considered for wind power generation. In the first site, the wind blows steadily at 7 m/s for 3000 hours per year, whereas in the second site the wind blows at 10 m/s for 2000 hours per year. Assuming the wind velocity is negligible at other times for simplicity, determine which is a better site for wind power generation. *Hint:*Note that the mass flow rate of air is proportional to wind velocity.

Step 1 of 4

Calculate the power generation by wind turbine at sight 1 as follows:

$$E_{mech1} = \dot{m}_1\left(\frac{V_1^2}{2}\right) \quad \text{..... (1)}$$

Here, the mass flow rate of wind at sight 1 is $\dot{m}_1$, and the velocity of wind at sight 1 is V_1.

Calculate the mass flow rate of the wind at sight 1 as follows:

$$\dot{m}_1 = \rho A V_1$$

Question:

Two sites are being considered for wind power generation. In the first site, the wind blows steadily at 7 m/s for 3000 hours per year, whereas in the second site the wind blows at 10 m/s for 2000 hours per year. Assuming the wind velocity is negligible at other times for simplicity, determine which is a better site for wind power generation. *Hint:*Note that the mass flow rate of air is proportional to wind velocity.

Step 1 of 4

Calculate the power generation by wind turbine at sight 1 as follows:

$$E_{mech1} = \dot{m}_1\left(\frac{V_1^2}{2}\right) \quad \text{..... (1)}$$

Here, the mass flow rate of wind at sight 1 is $\dot{m}_1$, and the velocity of wind at sight 1 is V_1.

Calculate the mass flow rate of the wind at sight 1 as follows:

$$\dot{m}_1 = \rho A V_1$$

$$E_{max1} = 0.2144 \times 3000$$
$$= 643.2\,\text{kWh/yr}$$

Therefore, the maximum power generation at sight 1 is $\boxed{643.2\,\text{kWh/yr}}$.

Step 3 of 4

Calculate the power generation by wind turbine at sight 2.

$$E_{mech2} = \dot{m}_2\left(\frac{V_2^2}{2}\right) \quad \text{..... (1)}$$

Here, the mass flow rate of wind at sight 2 is $\dot{m}_2$, and the velocity of wind at sight 2is V_2.

Calculate the mass flow rate of the wind at sight 2.

$$\dot{m}_2 = \rho A V_2$$

Here, the density of the air is ρ, and the flow area is A.

Substitute, $1.25\,\text{kg/m}^3$ for ρ, $1\,\text{m}^2$ for A and $10\,\text{m/s}$ for V_1 .

$$\dot{m}_2 = 1.25 \times 1 \times 10$$
$$= 12.50\,\text{kg/s}$$

Step 4 of 4

Substitute $10\,\text{m/s}$ for V_2 and $12.50\,\text{kg/s}$ for $\dot{m}_2$ in equation (1).

$$E_{mech2} = \dot{m}_2\left(\frac{V_2^2}{2}\right)$$
$$= 12.50\left(\frac{10^2}{2}\right)$$
$$= 625\,\text{W}$$
$$= 0.625\,\text{kW}$$

Calculate the maximum electric power generations per year as follows:

$$E_{max2} = E_{mech2} \times \Delta t_2$$

Here, the working hours per year at sight 2is Δt_2.

Substitute $0.625\,\text{kW}$ for E_{mech2} and $2000\,\text{h/yr}$ for Δt_2.

$$E_{max1} = 0.625 \times 2000$$
$$= 1250\,\text{kWh/yr}$$

Thus, the maximum power generation at sight 2 is $\boxed{1250\,\text{kWh/yr}}$.

Therefore, second site is better for power generation than first site.

Question: 3.016P

A river flowing steadily at a rate of 175 m3/s is considered for hydroelectric power generation. It is determined that a dam can be built to collect water and release it from an elevation difference of 80 m to generate power. Determine how much power can be generated from this river water after the dam is filled.

Step 1 of 2

Given

Volume flow rate $\dot{V} = 175\ \text{m}^3/s$

Density of the water $\rho = 1000\ \text{kg/m}^3$

Elevation difference $z = 80\ \text{m}$

Mass flow rate of water $\dot{m} = \rho \dot{V}$

$$\dot{m} = 1000 \times 175$$

$$\dot{m} = 1.75 \times 10^5\ \text{kg/s}$$

Step 2 of 2

Power generated, $P = \dot{m} \times g \times z$

$$P = \left(1.75 \times 10^5\right) \times (9.81) \times (80)$$

$$P = 137.34 \times 10^6\ \text{W}$$

$$P = 137.34\ \text{MW}$$

Hence power generated is $\boxed{P = 137.34\ \text{MW}}$

Question: 3.017P

Consider a river flowing toward a lake at an average velocity of 3 m/s at a rate of 500 m3/s at a location 90 m above the lake surface. Determine the total mechanical energy of the river water per unit mass and the power generation potential of the entire river at that location.

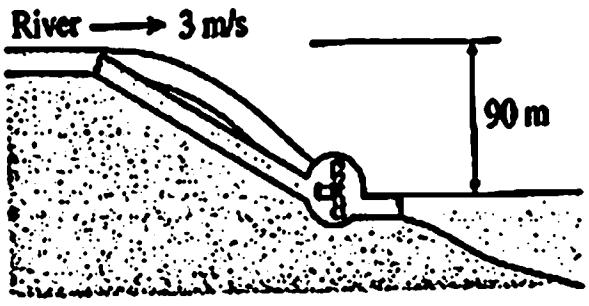


FIGURE P3–17

Step 1 of 3

Calculate the total mechanical energy of a flowing fluid using relation below.

$$e_{mech} = \frac{P}{\rho} + \frac{V^2}{2} + gz$$

Here, $\frac{P}{\rho}$ is the flow energy, $\frac{V^2}{2}$ is the kinetic energy and gz is the potential energy of the fluid per unit mass.

Neglect flow energy and substitute, $3\ \text{m/s}$ for V , $9.81\ \text{m/s}^2$ for g and $90\ \text{m}$ for z in relation above and calculate the total mechanical energy.

$$\begin{aligned} e_{mech} &= \frac{3^2}{2} + (9.81 \times 90) \\ &= 887.4 \times \left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2} \right) \\ &= 0.8874\ \text{kJ/kg} \end{aligned}$$

Therefore the total mechanical energy (e_{mech}) of the river water per unit mass

is $\boxed{0.887\ \text{kJ/kg}}$.

Step 2 of 3

Calculate the power generation potential of the river using relation. $E_{mech} = \dot{m} e_{mech}$ ..... (1)

Here, $\dot{m}$ is the mass flow rate of the fluid and is expressed as,

$$\dot{m} = \rho \dot{v}\ \text{kg/s}$$

Here, volumetric flow rate is $\dot{v}$ and density of water is ρ.

Step 3 of 3

Substitute, $1000\ \text{kg/m}^3$ for ρ and $500\ \text{m}^3/\text{s}$ for $\dot{v}$ in relation above and calculate the mass flow rate.

$$\begin{aligned} \dot{m} &= 1000 \times 500 \\ &= 500{,}000\ \text{kg/s} \end{aligned}$$

Substitute, $500{,}000\ \text{kg/s}$ for $\dot{m}$ and $0.887\ \text{kJ/kg}$ for e_{mech} in equation (1) and calculate the power generation.

$$\begin{aligned} E_{mech} &= 500000 \times 0.8874 \\ &= 443700\ \text{kW} \\ &= 443.7\ \text{MW} \\ &\approx 444\ \text{MW} \end{aligned}$$

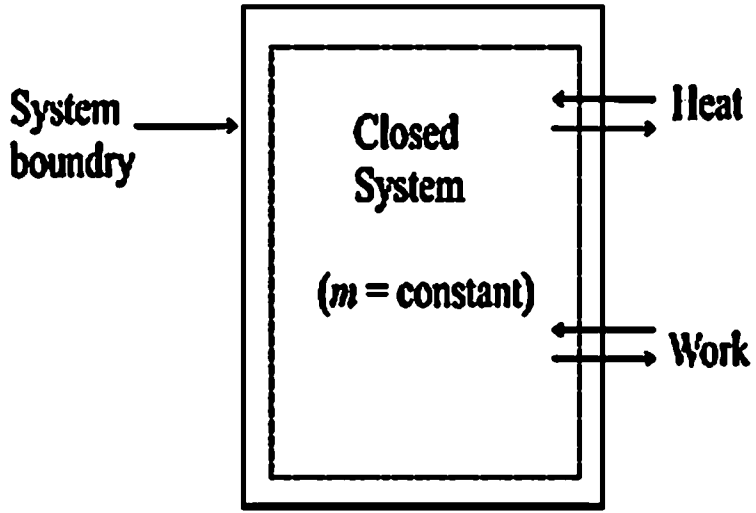
Therefore the power generation (E_{mech}) is $\boxed{444\ \text{MW}}$.

Question: 3.018P

In what forms can energy cross the boundaries of a closed system?

Step 1 of 1

Energy can cross the boundaries of a closed system in the form of heat and work.



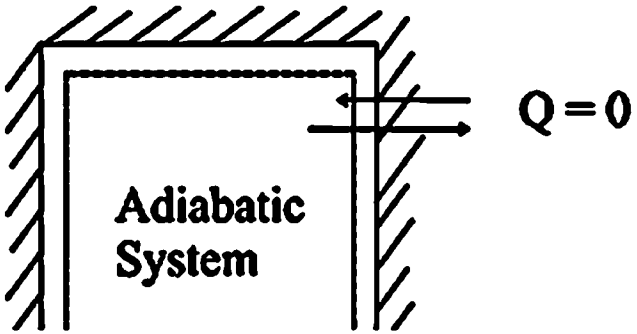
Question: 3.019P

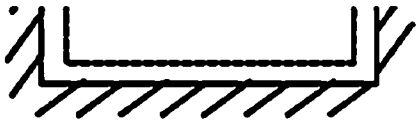
When is the energy crossing the boundaries of a closed system heat and when is it work?

Step 1 of 2

In a closed system there will be energy transfer but no mass transfer occurs. So across boundaries the heat is transferred due to the temperature difference and work is due to all other forms except temperature difference.

Step 2 of 2





Question: 3.020P

What is an adiabatic process? What is an adiabatic system?

Step 1 of 1

A process during which there is no heat transfer is called an adiabatic process i.e. $dQ = 0$

The system which undergoes adiabatic process is called an adiabatic system, and it is said to be no heat is added or removed from the system

Question: 3.021P

What are point and path functions? Give some examples.

Step 1 of 2

Thermodynamic properties are point functions, since for a given state, there is a definite value for each property. The change in thermodynamic property of a system in a change of state is independent of the path, and depends only on the initial and final states of the system. The cyclic integral of a point function is zero

Examples: Temperature, pressure, volume etc.

Step 2 of 2

The path function depends on the path followed by the process. Path functions have inexact differentials designated by the symbol ∂. The cyclic integral of a path function is non-zero

Examples: Heat, work etc.

Question: 3.022P

Consider an automobile traveling at a constant speed along a road. Determine the direction of the heat and work interactions, taking the following as the system: (*a*) the car radiator, (*b*) the car engine, (*c*) the car wheels, (*d*) the road, and (*e*) the air surrounding the car.

Step 1 of 5

(a) The car radiator:

Heat transfer is occurred from radiator to atmosphere due to temperature difference of the hot fluid and coolant.

No work transfer

Step 2 of 5

(b) The car engine:

Chemical energy of the fuel is converted into mechanical energy so both heat and work transfer will occur due to burning of fuel and piston movement respectively.

Step 3 of 5

(c) The car wheels:

Highly pressured wheels have cool air inside, which allows heat to transfer to occur to the surrounding atmosphere

No work is transferred at rest, while motion the wheels possess kinetic energy which can produce work.

Step 4 of 5

(d) The road:

Due to friction between the wheels and the road the temperature of the road increases because of the heat transfer.

No work transfer.

Step 5 of 5

(e) The air surrounding the car:

Due to exhausts from the engine and other hotter components of car heat is added to atmosphere.

Work is done on the air by pushing air back and moving our car forward.

Question: 3.023P

The length of a spring can be changed by (*a*) applying a force to it or (*b*) changing its temperature (i.e., thermal expansion). What type of energy interaction between the system (spring) and surroundings is required to change the length of the spring in these two ways?

Step 1 of 1

The application of force on the spring causes its elongation, this force with displacement is a work interaction.

Heat transfer will occur in the second case, the spring will elongation due to temperature changes.

Question: 3.024P

A personal computer is to be examined from a thermodynamic perspective. Determine the direction of the work and heat transfers (in or out) when the (*a*) keyboard, (*b*) monitor, (*c*) processing unit, and (*d*) all of these are taken as the system.

Step 1 of 7

Step 2 of 7

(b)

Step 3 of 7

The monitor has a direct power supply, so electrical work is supplied to it. Due to the electrical components in the monitor, heat is generated. Therefore, heat transfer will occur from monitor to surroundings.

Step 4 of 7

(c)

Step 5 of 7

The CPU has a direct power supply, so electrical work is supplied to it. Due to electrical components in the CPU, heat is generated. Therefore, heat transfer will occur from CPU to surroundings.

Step 6 of 7

Step 7 of 7

(d)

The complete unit has an electrical work done on it and also transfers heat from all its electrical parts to the surroundings.

Question: 3.025P

A small electrical motor produces 5 W of mechanical power. What is this power in (*a*) N, m, and s units; and (*b*) kg, m, and s units?

Step 1 of 2

(a) Use the following conversion factor to convert the power from W to N·m/s:

$$1\,\text{W} = 1\,\text{Nm/s}$$

Calculate the power required for the electrical motor.

$$P = 5\,\text{W}\left(\frac{1\,\text{N-m/s}}{1\,\text{W}}\right)$$

$$= 5\,\text{N-m/s}$$

Therefore, power required for electrical motor in terms of Nm/s is $\boxed{5\,\text{N-m/s}}$.

Step 2 of 2

(b) Use the following conversion factors to convert the power from W to 1 kg·m/s³:

$$1\,\text{W} = 1\,\text{Nm/s}$$

$$1\,\text{Nm/s} = 1\,\text{kgm/s}^3$$

$$1\,\text{N} = 1\,\text{kg·m/s}^2$$

$$1\,\text{J} = 1\,\text{N·m}$$

$$1\,\text{W} = 1\,\text{J/s}/1\,\text{W}$$

Calculate the power required for the electrical motor.

$$P = 5\,\text{W}\left(\frac{1\,\text{J/s}}{1\,\text{W}}\right)\left(\frac{1\,\text{N-m}}{1\,\text{J}}\right)\left(\frac{1\,\text{kg·m/s}^2}{1\,\text{N}}\right)$$

$$= 5\,\text{kg·m}^2/\text{s}^3$$

Hence, power required for electrical motor in terms of kg·m²/s³ is $\boxed{5\,\text{kg·m}^2/\text{s}^3}$

Question: 3.026P

A model aircraft internal-combustion engine produces 10 W of power. How much power is this in (*a*) lbf·ft/s and (*b*) hp?

Step 1 of 2

Given that,
The power produced by an aircraft engine $(P) = 10\ \text{W}$

(a) In lbf · ft/s units:

$$1\ \text{Btu/s} = 1055\ \text{W}$$
$$1\ \text{Btu/s} = 778.2\ \text{lbf} \cdot \text{ft/s}$$
$$\therefore 1\ \text{W} = \frac{778.2}{1055}\ \text{lbf} \cdot \text{ft/s}$$
$$1\ \text{W} = 0.737\ \text{lbf} \cdot \text{ft/s}$$
$$10\ \text{W} = 0.737 \times 10\ \text{lbf} \cdot \text{ft/s}$$
$$\therefore \text{Power } \boxed{P = 7.37\ \text{lbf} \cdot \text{ft/s}}$$

Step 2 of 2

(b) In hp:

$$1\ \text{horse power} = 746\ \text{Watts}$$
$$\therefore 10\ \text{W} = \frac{10}{746}\ \text{hp}$$
$$\therefore \text{Power } \boxed{P = 0.0134\ \text{hp}}$$

Question: 3.027P

A car is accelerated from rest to 85 km/h in 10 s. Would the energy transferred to the car be different if it were accelerated to the same speed in 5 s?

Step 1 of 1

The kinetic energy depends only on the velocity of the object, but not on the time on which it travels.

$$KE = \frac{1}{2}mV^2$$

Here, m is mass of the object and V is velocity of the object.

Therefore, if a car accelerates at the e speed at different time durations, the energy transferred is e but the power is different.

Question: 3.028P

Determine the energy required to accelerate an 800-kg car from rest to 100 km/h on a level road.

Step 1 of 2

Given data:

Mass of the car, $(m) = 800$ kg

Initial velocity, $V_1 = 0$ m/s

Final velocity, $V_2 = 100$ km/h

$$\left(\because 1 \text{ km/h} = \frac{1000}{3600} \text{ m/s}\right)$$

$$V_2 = 100 \times \frac{1000}{3600} \text{ m/s}$$

$$\therefore V_2 = 27.78 \text{ m/s}$$

Step 2 of 2

The change in kinetic energy of the body is the energy required to accelerate the car

$$KE = \frac{1}{2} m(V_2^2 - V_1^2)$$

$$= \frac{1}{2}(800)(27.78^2 - 0^2) \text{ J}$$

$$= 308.69 \text{ kJ}$$

$\therefore$ The energy required to accelerate the car, $\boxed{KE = 308.69 \text{ kJ}}$

Question: 3.029P

A construction crane lifts a prestressed concrete beam weighing 3 tons from the ground to the top of piers that are 24 ft above the ground. Determine the amount of work done considering (*a*) the beam and (*b*) the crane as the system. Express your answers in both lbf·ft and Btu.

Step 1 of 4

a)

Use the conversion factor to convert the weight from ton to lbm

$$1\,\text{ton}=2000\,\text{lbm}$$

Calculate the weight of the concrete beam in terms of lbm

$$m=3\,\text{ton}\frac{2000\,\text{lbm}}{1\,\text{ton}}$$

$$=6000\,\text{lbm}$$

Step 2 of 4

From the conversion table

$$1\,\text{lbf}=32.174\,\text{lbm}\cdot\text{ft/s}^2$$

Calculate the work done in lbf - ft

$$W=mgh$$

Here, m is the mass of the car, g is the acceleration due to gravity

Substitute $32.2\,\text{ft/s}^2$ for g and $3\,\text{ton}$ for m

$$W=mgh$$

$$=\left(3\,\text{ton}\times\frac{2000\,\text{lbm}}{1\,\text{ton}}\right)(32.2)\left(\frac{1\,\text{lbf}}{32.2\,\text{lbm}\cdot\text{ft/s}^2}\right)(24)$$

$$=144000\,\text{lbf}\cdot\text{ft}$$

Therefore, the work done in lbf - ft is $\boxed{144000\,\text{lbf}\cdot\text{ft}}$

Step 3 of 4

Calculate the work done in Btu

$$W=(144000\,\text{lbf}\cdot\text{ft})\left(\frac{1\,\text{Btu}}{778.169\,\text{lbf}\cdot\text{ft}}\right)$$

$$=185.049\,\text{Btu}$$

Therefore, the work done in Btu is $\boxed{185.049\,\text{Btu}}$

Step 4 of 4

b)

If crane as considered as system, it must do same amount of work as is required to lift beam as calculated above,

Thus, amount of work done becomes $W=185.049\,\text{Btu}$

Therefore, the work done in Btu is $\boxed{185.049\,\text{Btu}}$

Question: 3.030P

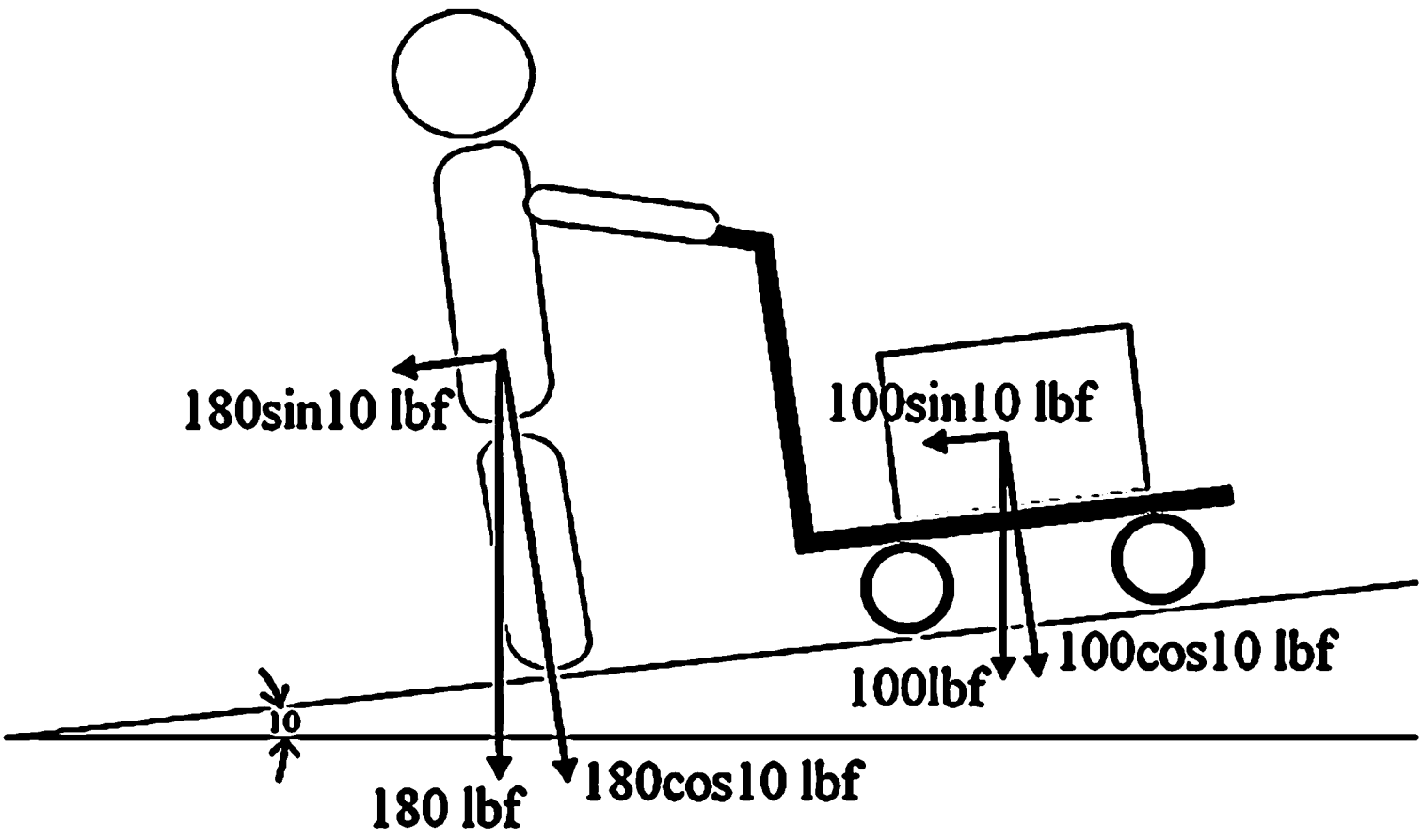
A man weighing 180 lbf is pushing a cart that weighs 100 lbf with its contents up a ramp that is inclined at an angle of 10° from the horizontal. Determine the work needed to move along this ramp a distance of 100 ft considering (*a*) the man and (*b*) the cart and its contents as the system. Express your answers in both lbf-ft and Btu.



FIGURE P3–30E

Step 1 of 4

Draw the free body diagram of the weight of the man and weight of the cart.



Step 2 of 4

Write the weight component of cart along the ramp.

$$W_{cart} = 100\sin 10^\circ \text{ lbf}$$
$$= 17.3648 \text{ lbf}$$

Write weight component of man along the ramp.

$$W_{man} = 180\sin 10^\circ \text{ lbf}$$
$$= 31.256 \text{ lbf}$$

Step 3 of 4

(a)

Consider man and its contents as the system and find the work done by the man on cart.

$$W = (W_{man} \times d) + (W_{cart} \times d)$$

Here, d is the distance moved along the ramp.

Substitute 100 ft for d, 31.256 lbf for W_{man}, and 17.3648 lbf for W_{cart}

$$W = 31.256 \times 100 + 17.3648 \times 100$$
$$= 4862.08 \text{ lbf-ft}$$
$$= 4862.08 \text{ lbf-ft} \times \frac{1 \text{ Btu}}{778 \text{ lbf-ft}}$$
$$= 6.249 \text{ Btu}$$

Therefore, work needed to move along the ramp when the man and its contents are considered as the system is $\boxed{4862.08 \text{ lbf-ft}}$ or $\boxed{6.249 \text{ Btu}}$.

Step 4 of 4

(b)

Consider cart and its contents as the system and find the work needed to move along the ramp.

$$W = W_{cart} \times d$$

Substitute 100 ft for d and 17.3648 lbf for W_{cart}

$$W = 17.3648 \times 100$$
$$= 1736.48 \text{ lbf-ft}$$
$$= 1736.48 \text{ lbf-ft} \times \frac{1 \text{ Btu}}{778 \text{ lbf-ft}}$$
$$= 2.2319 \text{ Btu}$$

Therefore, work needed to move along the ramp when the cart and its contents as the system is $\boxed{1736.48 \text{ lbf-ft}}$ or $\boxed{2.2319 \text{ Btu}}$.

Question: 3.031P

The force *F* required to compress a spring a distance *x* is given by $F - F0 = kx$ where *k* is the spring constant and *F*0 is the preload. Determine the work required to compress a spring whose spring constant is $k = 200$ lbf/in a distance of one inch starting from its free length where $F0 = 0$ lbf. Express your answer in both lbf·ft and Btu.

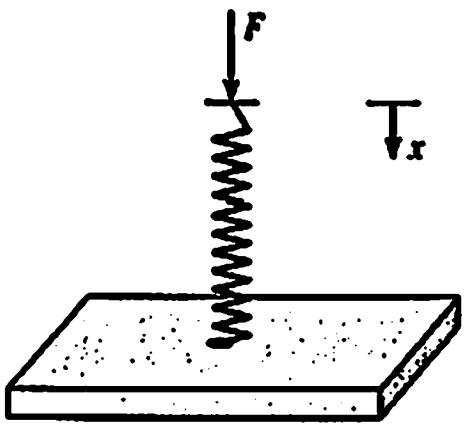


FIGURE P3–31E

Step 1 of 5

Given data:

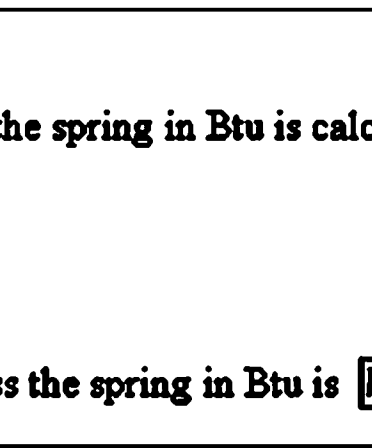
Preload, $F_o = 0$ lbf

Spring constant, $k = 200$ lbf/in

Initial length, $x_1 = 0$

Final length, $x_2 = 1$ in

Step 2 of 5



Step 3 of 5

The force at any point during the deflection of the spring is given by $F = F_o + kx$,

where F_o is the initial force

and x is the deflection as measured from the point where the initial force occurred.

The work required to compress the spring is

$$W = \int F ds$$

$$W = \int_1^2 (F_o + kx) dx$$

Step 4 of 5

Therefore work required to compress the spring is

$$W = \int_1^2 (kx) dx \qquad (\because F_o = 0 \text{ lbf})$$

$$= k\int_1^2 x dx$$

$$= k\frac{1}{2}(x^2_2 - x^2_1)$$

$$= 200 \times \frac{1}{2} \times (1^2 - 0) \text{ lbf} \cdot \text{in}$$

$$= \frac{100}{12} \text{ lbf} \cdot \text{ft} \qquad (\because 1\text{ft} = 12\text{in})$$

$$= 8.333 \text{ lbf} \cdot \text{f}$$

∴ The work required to compress the spring in lbf · ft is $\boxed{W = 8.333 \text{ lbf} \cdot \text{ft}}$

Step 5 of 5

The work required to compress the spring in Btu is calculated by

Where $1\text{Btu} = 778 \text{ lbf} \cdot \text{ft}$

$$= \frac{8.333}{778} \text{ Btu}$$

$$= 0.0107 \text{ Btu}$$

∴ The work required to compress the spring in Btu is $\boxed{W = 0.0107 \text{ Btu}}$

Question: 3.032P

A spherical soap bubble with a surface-tension of 0.005 lbf/ft is expanded from a diameter of 0.5 in to 2.0 in. How much work, in Btu, is required to expand this bubble?

Step 1 of 2

Convert the units of d_2 from in. to ft.

$$\begin{aligned} d_2 &= 2.0\,\text{in} \\ &= \frac{2\,\text{in.}}{12} \times \frac{\text{ft}}{\text{in.}} \\ &= 0.1667\,\text{ft} \end{aligned}$$

Here, d_2 is the final diameter of the bubble.

Convert the units of d_1 from in. to ft.

$$\begin{aligned} d_1 &= 0.5\,\text{in} \\ &= \frac{0.5\,\text{in.}}{12} \times \frac{\text{ft}}{\text{in.}} \\ &= 0.041667\,\text{ft} \end{aligned}$$

Here, d_1 is the initial diameter of the bubble.

Step 2 of 2

Calculate the work associated with the stretching of a film as follows:

$$\begin{aligned} W_{surface} &= \int_1^2 \sigma_s dA \\ &= \sigma(A_2 - A_1) \\ &= \sigma\left(4\pi d_2^2 - 4\pi d_1^2\right) \\ &= 4\pi\sigma \times \left(d_2^2 - d_1^2\right) \end{aligned}$$

Here, σ is the surface tension of the soap bubble.

Substitute $0.005\,\text{lb/ft}$ for σ, $0.1667\,\text{ft}$ for d_2, $0.041667\,\text{ft}$ for d_1.

$$\begin{aligned} W_{surface} &= 4\pi \times 0.005\left[(0.1667)^2 - (0.041667)^2\right] \\ &= 0.001637\,\text{lbf}\cdot\text{ft} \\ &= (0.001637\,\text{lbf}\cdot\text{ft})\left(\frac{1\,\text{Btu}}{778.169\,\text{lbf}\cdot\text{ft}}\right) \\ &= 2.104\times10^{-6}\,\text{Btu} \end{aligned}$$

Therefore, the work required to expand the bubble is $\boxed{2.104\times10^{-6}\,\text{Btu}}$.

Question: 3.033P

A 0.5-in diameter, 12-in-long steel rod with a Young's modulus of 30,000 lbf/in2 is stretched 0.125 in. How much work does this require, in Btu? The strain work is given by $W = \frac{V_0 E}{2}(\varepsilon_2^2 - \varepsilon_1^2)$ where is the original volume of the V_0 is the original volume of the solid, E is Young's modulus and ε is the strain at the beginning and ending of the process.

Step 1 of 3

Given data:

Diameter of the rod, $D = 0.5\text{in}$.

Length of the rod, $L = 12\text{in}$.

Young's modulus of the steel rod, $E = 30000\text{lbf/in}^2$.

Stretched to a length = 0.125 in.

Strain work is $W = \frac{V_0 E}{2}\left(\varepsilon_2^2 - \varepsilon_1^2\right)$.

Where V_0 is the original volume of the rod

E is Young's modulus

ε is the strain

Step 2 of 3

The original volume of the rod is calculated from

$$V_0 = \frac{\pi D^2}{4} L$$

$$= \frac{\pi(0.5)^2}{4} \times 12$$

$$= 2.356\text{in}^3$$

Step 3 of 3

The work required to stretch the rod 0.125 in is calculated from the given formula

$$W = \frac{V_0 E}{2}\left(\varepsilon_2^2 - \varepsilon_1^2\right)$$

$$= \frac{(2.356)(30,000)}{2}\left[(0.125/12)^2 - 0^2\right]$$

$$= 3.8346\text{lbf} \cdot \text{in}$$

$$= (3.8346\text{lbf} \cdot \text{in})\left(\frac{1\text{Btu}}{9331.78\text{lbf} \cdot \text{in}}\right)$$

$$= 4.109 \times 10^{-4}\text{Btu}$$

Therefore the work required in Btu is $\boxed{W = 4.109 \times 10^{-4}\text{Btu}}$

Question: 3.034P

A spring whose spring constant is 200 lbf/in has an initial force of 100 lbf acting on it. Determine the work, in Btu, required to compress it another 1 inch.

Step 1 of 5

Given data:

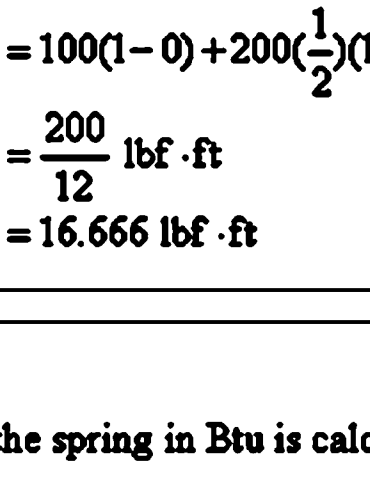
Initial force, $F_o = 100$ lbf

Spring constant, $k = 200$ lbf/in

Initial length, $x_1 = 0$

Final length, $x_2 = 1$ in

Step 2 of 5



Step 3 of 5

The force at any point during the deflection of the spring is given by $F = F_o + kx$,

where F_o is the initial force

and x is the deflection as measured from the point where the initial force occurred.

The work required to compress the spring is

$$W = \int F ds$$

$$W = \int_1^2 (F_o + kx)dx$$

Step 4 of 5

Therefore work required to compress the spring is

$$W = F_o(x_2 - x_1) + k\frac{1}{2}(x^2_2 - x^2_1)$$

$$= 100(1-0) + 200(\frac{1}{2})(1^2 - 0)\ \text{lbf} \cdot \text{in}$$

$$= \frac{200}{12}\ \text{lbf} \cdot \text{ft} \qquad (\because 1\text{ft} = 12\text{in})$$

$$= 16.666\ \text{lbf} \cdot \text{ft}$$

Step 5 of 5

The work required to compress the spring in Btu is calculated by

Where $1\text{Btu} = 778\ \text{lbf} \cdot \text{ft}$

$$= \frac{16.666}{778}\ \text{Btu}$$

$$= 0.0214\ \text{Btu}$$

$\therefore$ The work required to compress the spring, $\boxed{W = 0.0214\ \text{Btu}}$

3.035P

Question:

How much work, in kJ, can a spring whose spring constant is 3 kN/cm produce after it has been compressed 3 cm from its unloaded length?

Step 1 of 3

Given data:

Initial force, $F_o = 0$ N

Spring constant, $k = 3$ kN/cm

$k = 300$ kN/m

Initial length, $x_1 = 0$

Final length, $x_2 = 0.03$ m

Step 2 of 3

The force at any point during the deflection of the spring is given by $F = F_o + kx$, where F_o is the initial force and x is the deflection as measured from the point where the initial

Question:

How much work, in kJ, can a spring whose spring constant is 3 kN/cm produce after it has been compressed 3 cm from its unloaded length?

Step 1 of 3

Given data:

Initial force, $F_o = 0$ N

Spring constant, $k = 3$ kN/cm

$k = 300$ kN/m

Initial length, $x_1 = 0$

Question: 3.036P

A ski lift has a one-way length of 1 km and a vertical rise of 200 m. The chairs are spaced 20 m apart, and each chair can seat three people. The lift is operating at a steady speed of 10 km/h. Neglecting friction and air drag and assuming that the average mass of each loaded chair is 250 kg, determine the power required to operate this ski lift. Also estimate the power required to accelerate this ski lift in 5 s to its operating speed when it is first turned on.

Step 1 of 5

Find the total mass of the ski lift by using the equation,

$$M = m \times n$$

Here, m is average mass of each chair and n is number of chairs placed in 1 km length of ski lift.

Substitute 250 kg for m and $\frac{1000}{20}$ for n.

$$M = 250 \times \frac{1000}{20}$$

Question:

A ski lift has a one-way length of 1 km and a vertical rise of 200 m. The chairs are spaced 20 m apart, and each chair can seat three people. The lift is operating at a steady speed of 10 km/h. Neglecting friction and air drag and assuming that the average mass of each loaded chair is 250 kg, determine the power required to operate this ski lift. Also estimate the power required to accelerate this ski lift in 5 s to its operating speed when it is first turned on.

Step 1 of 5

Find the total mass of the ski lift by using the equation,

$M = m \times n$

Here, m is average mass of each chair and n is number of chairs placed in 1 km length of ski lift.

Substitute 250 kg for m and $\frac{1000}{20}$ for n.

$$\begin{aligned} M &= 250 \times \frac{1000}{20} \\ &= 0.1 \times 3600 \\ &= 360\,\text{s} \end{aligned}$$

Calculate the power required to operate the lift by using the equation,

$$\begin{aligned} P &= \frac{W}{t}\ \text{kJ/s} \\ &= \frac{24525}{360} \\ &= 68.125\,\text{kW} \end{aligned}$$

Therefore, the power required to raise the lift is $\boxed{68.125\ \text{kW}}$.

Step 3 of 5

Calculate the acceleration of the lift in 5 sec.

$$a = \frac{\Delta V}{t}$$

Here, ΔV is change in velocity.

$$\begin{aligned} a &= \frac{\left(10 \times \frac{1000}{3600}\right)\text{m/s} - 0}{5} \\ &= \frac{\left(10 \times \frac{5}{18}\right)\text{m/s} - 0}{5} \\ &= 0.556\,\text{m/s}^2 \end{aligned}$$

Step 4 of 5

Calculate the work needed to accelerate a body is simply the change in the kinetic energy of the body by using the equation.

$$\begin{aligned} W_a &= \frac{1}{2} M (V_2^2 - V_1^2)\ \text{kJ} \\ &= \frac{1}{2}(12500) \times \left(\left(10 \times \frac{5}{18}\right)^2 - 0^2\right) \\ &= 48225.308\,\text{J} \\ &= 48.225\,\text{kJ} \end{aligned}$$

Calculate the power required to accelerate the ski lift in 5s by using the expression.

$$\begin{aligned} \dot{W}_a &= \frac{W_a}{\Delta t}\ \text{kW} \\ &= \frac{48.225}{5} \\ &= 9.645\,\text{kW} \end{aligned}$$

Therefore, power required during the acceleration is $\boxed{9.645\ \text{kW}}$.

Therefore, the total power required to operate is $\boxed{43.734\ \text{kW}}$.

Question: 3.037P

Determine the power required for a 1150-kg car to climb a 100-m-long uphill road with a slope of 30° (from horizontal) in 12 s (*a*) at a constant velocity, (*b*) from rest to a final velocity of 30 m/s, and (c) from 35 m/s to a final velocity of 5 m/s. Disregard friction, air drag, and rolling resistance.

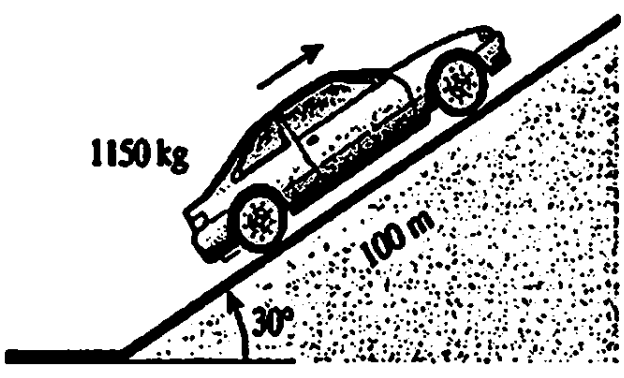


FIGURE P3–37

Question:

Determine the power required for a 1150-kg car to climb a 100-m-long uphill road with a slope of 30° (from horizontal) in 12 s (*a*) at a constant velocity, (*b*) from rest to a final velocity of 30 m/s, and (c) from 35 m/s to a final velocity of 5 m/s. Disregard friction, air drag, and rolling resistance.

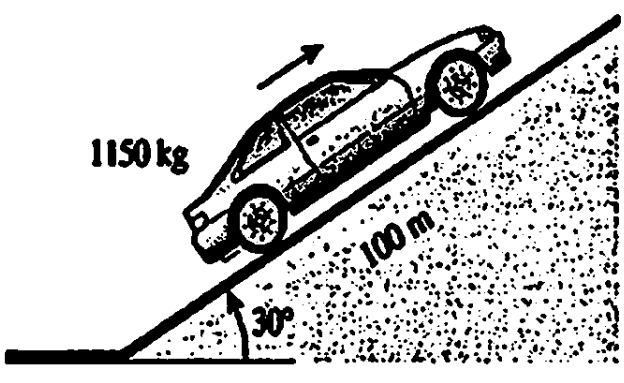


FIGURE P3–37

Step 1 of 8

Draw the diagram shows the slope of the uphill as shown below:

Step 1 of 8

Draw the diagram shows the slope of the uphill as shown below:

Calculate the work done in climbing the uphill road will be stored in the form of potential energy in the car

$$W = m \times g \times h$$

Here, m is the mass of the car, g is the acceleration due to gravity

Substitute 9.81m/s^2 for g and 1150kg for m

$$W = m \times g \times h$$
$$= 1150 \times 9.81 \times 50$$
$$= 564.075 \text{ kJ}$$

Step 4 of 8

Calculate the Power required for car

$$P = \frac{\text{Work}}{\text{time}}$$

Substitute 12s for time

$$P = \frac{\text{Work}}{\text{time}}$$
$$= \frac{564.075}{12}$$
$$= 47 \text{ kJ/s}$$
$$= 47 \text{ kW}$$

Power required for the car to climb uphill is $\boxed{47 \text{ kW}}$

Step 5 of 8

Calculate the work done in climbing the uphill road will be stored in the form of potential energy in the car

$$W = m \times g \times h$$

Here, m is the mass of the car, g is the acceleration due to gravity

Substitute 9.81m/s^2 for g and 1150kg for m

$$W = m \times g \times h$$
$$= 1150 \times 9.81 \times 50$$
$$= 564.075 \text{ kJ}$$

Step 4 of 8

Calculate the Power required for car

$$P = \frac{\text{Work}}{\text{time}}$$

Substitute 12s for time

$$P = \frac{\text{Work}}{\text{time}}$$
$$= \frac{564.075}{12}$$
$$= 47 \text{ kJ/s}$$
$$= 47 \text{ kW}$$

Power required for the car to climb uphill is $\boxed{47 \text{ kW}}$

Step 5 of 8

b)

Initial velocity, $V_1 = 0$

Final velocity, $V_2 = 30 \text{ m/s}$

On reaching 100 m uphill, the car has both potential and kinetic energy. This is due to the effect of work done on it.

Calculate the total work done

$$\text{Work} = PE + KE$$
$$= (m \times g \times h) + \left[\frac{1}{2} m \left(V_2^2 - V_1^2\right)\right]$$

Substitute 30m/s for V_2, 0m/s for V_1

$$Work = PE + KE$$
$$= (m \times g \times h) + \left[\frac{1}{2} m \left(V_2^2 - V_1^2\right)\right]$$
$$= (1150 \times 9.81 \times 50) + \left[\frac{1}{2} 1150 \left(30^2 - 0^2\right)\right]$$
$$= 564.07 + 517.5$$
$$= 1081.57 \text{ kJ}$$

Step 6 of 8

Calculate the Power required

$$P = \frac{\text{Work}}{\text{time}}$$
$$= \frac{1081.57}{12}$$
$$= 90.1 \text{ kW}$$

Therefore, Power required for the car to climb uphill is $\boxed{90.1 \text{kW}}$

Step 7 of 8

(c)

$$\text{time}$$
$$= \frac{-125.93}{12}$$
$$= -10.49 \text{ kW}$$

Power required for the car to climb uphill is $\boxed{-10.49 \text{ kW}}$

Question: 3.038P

A damaged 1200-kg car is being towed by a truck. Neglecting the friction, air drag, and rolling resistance, determine the extra power required (*a*) for constant velocity on a level road, (*b*) for constant velocity of 50 km/h on a 30° (from horizontal) uphill road, and (*c*) to accelerate on a level road from stop to 90 km/h in 12 s.

Step 1 of 7

Given data:

The mass of a the car, $(m) = 1200$ kg

The acceleration due to gravity, $(g) = 9.8$ m/s

Step 2 of 7

(a) For constant velocity on level road:

The work needed to accelerate a body is simply the change in the kinetic energy of the car on a level road

$$W_{ke} = W_a = \frac{1}{2}m(V_2^2 - V_1^2)\ \text{kJ}$$

Here at constant velocity the work done due to acceleration is zero.

∴ Power required, $\boxed{P = 0\ \text{kW}}$

Step 3 of 7

(b) On a uphill road with constant velocity $V = 50$ km/h

The slope of the uphill road $(\theta) = 30°$

The work needed to accelerate a body is simply the change in the kinetic energy of the car and the change in the potential energy of the car.

$$W_{total} = W_{ke} + W_{pe}$$

Where, $W_{ke} = W_a = 0$ since at constant velocity acceleration is zero.

$$W_{pe} = mgh$$

$$\therefore W_{total} = W_{pe}$$

Step 4 of 7

The power required is determined by work per unit time

$$P = \frac{W_{total}}{\Delta t}\ \text{kW}$$

$$= \frac{mgh}{\Delta t}$$

$$= \frac{mgd\sin\theta}{\Delta t}$$

We know that, $\frac{d}{\Delta t} = V$

The velocity of the car, given $V = 50$ km/h

Step 5 of 7

$$\therefore P = mgV\sin\theta$$

$$= 1200 \times 9.8 \times 50 \times \frac{5}{18} \times \sin 30° \quad \left(\because 1\ \text{km/h} = \frac{5}{18}\ \text{m/s}\right)$$

$$= \frac{81667}{1000}\ \text{kW}$$

$$= 81.7\ \text{kW}$$

∴ Power required, $\boxed{P = 81.7\ \text{kW}}$

Step 6 of 7

(c) Acceleration on a level road

Due to no elevation of the road the potential energy will be zero

i.e. $W_{pe} = 0$ kJ

The car accelerates from rest $(V_1 = 0\ \text{m/s})$ to $(V_2 = 90\ \text{km/h})$

Therefore kinetic energy $W_{ke} = W_a = \frac{1}{2}m(V_2^2 - V_1^2)\ \text{kJ}$

$$= \frac{1}{2}(1200)(25^2 - 0)\ \text{J}$$

$$= 375\ \text{kJ}$$

Step 7 of 7

Therefore the power required

$$P = \frac{W_{total}}{\Delta t}\ \text{kW}$$

$$= \frac{375}{12}\ \text{kJ}$$

$$= 31.25\ \text{kW}$$

∴ Power required, $\boxed{P = 31.25\ \text{kW}}$

Question: 3.039P

For a cycle, is the net work necessarily zero? For what kind of systems will this be the case?

Step 1 of 1

For a cyclic process, the network is not necessarily being zero for all systems except for adiabatic systems.

For adiabatic systems, the heat transfer, Q is zero. From first law of thermodynamics the work interaction is equal to heat interactions in a system undergoing cyclic process.

$$\Sigma Q = \Sigma W$$

For adiabatic systems, $\Sigma Q = 0$ then,

$$0 = \Sigma W$$

$$\Sigma W = 0$$

So, for adiabatic systems undergoing cyclic process the net work is also zero.

Question: 3.040P

What are the different mechanisms for transferring energy to or from a control volume?

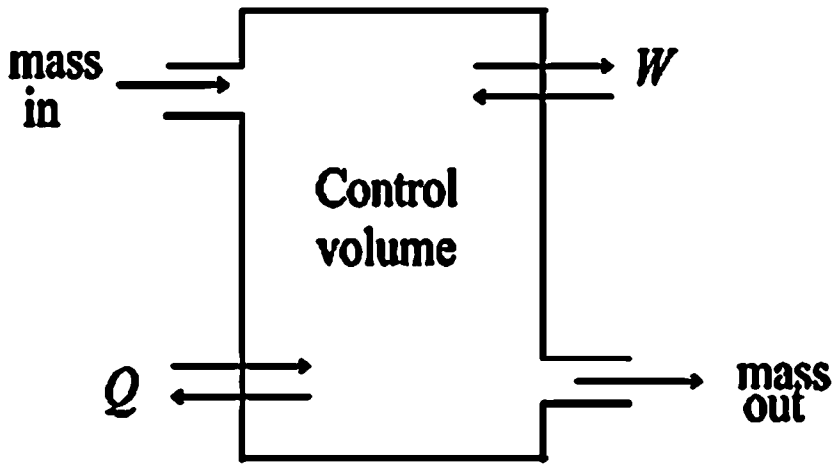
Step 1 of 2

For a control volume (open system) the energy can be transferred to or from a system in three forms:

1. Heat
2. work
3. mass flow

Heat is the process of energy transfer from one body or system due to thermal contact, Energy transfer by heat can occur between objects by radiation, conduction and convection.

Step 2 of 2



Question: 3.041P

On a hot summer day, a student turns his fan on when he leaves his room in the morning. When he returns in the evening, will the room be warmer or cooler than the neighboring rooms? Why? Assume all the doors and windows are kept closed.

Step 1 of 1

Yes the room will be warmer, due to continuously running of the fan heat gets added in the room because of the air friction and electrical power in the room.

Question: 3.042P

One way to improve the fuel efficiency of a car is to use tires that have a lower rolling resistance—tires that roll with less resistance, and highway tests at 65 mph showed that tires with the lowest rolling resistance can improve the fuel efficiency by nearly 2 mpg (miles per gallon). Consider a car that gets 35 mpg on high rolling resistance tires and is driven 15,000 miles per year. For a fuel cost of $2.20/gal, determine how much money will be saved per year by switching to low rolling resistance tires.

Step 1 of 2

Given

Distance travelled by Car per year $d = 15000$ miles

Lowest rolling resistance will improve efficiency by 2 mpg

Fuel efficiency of car with high rolling resistance tires $= 35$ mpg

So, total fuel required by a car with high rolling resistance tires $= \frac{15000}{35}$ gallons

$= 428.57$ gallons

Step 2 of 2

Fuel efficiency of car with high rolling resistance tires $= 35$ mpg $+ 2$ mpg

$= 37$ mpg

So, total fuel required by a car with low rolling resistance tires $= \frac{15000}{37}$ gallons

$= 405.405$ gallons

So, fuel saved by switching to low resistance tires $= (428.57 - 405.405)$ gallon

$= 23.165$ gallon

Cost of 1 gallon $= \$\ 2.20$

Cost of 23.165 gallon $= 23.165 \times \$2.20$

$= \$\ 50.963$

Hence total cost saved by switching to low resistance tires is $\boxed{\$\ 50.963}$

Question: 3.043P

An adiabatic closed system is accelerated from 0 m/s to 30 m/s. Determine the specific energy change of this system, in kJ/kg.

Step 1 of 2

Given data:

The initial velocity of the system, $(V_1) = 0$ m/s

The final velocity of the system, $(V_2) = 30$ m/s

Step 2 of 2

The energy needed to accelerate a body is simply occurred due to change in velocity i.e. the kinetic energy.

The specific kinetic energy is expressed as

$$ke = \frac{V_2^2 - V_1^2}{2} \text{ kJ/kg}$$

$$ke = \left(\frac{30^2 - 0}{2}\right) \text{ J/kg}$$

$$ke = \frac{450}{1000} \text{ kJ/kg}$$

$$ke = 0.45 \text{ kJ/kg}$$

∴ Specific kinetic energy, $\boxed{ke = 0.45 \text{ kJ/kg}}$

Question: 3.044P

An adiabatic closed system is raised 100 m at a location where the gravitational acceleration is 9.8 m/s2. Determine the energy change of this system, in kJ/kg.

Step 1 of 1

2760-2-47P SA Code: 2981

SR Code: 5070 RID1: 3211 | 19/08/2013
As the system undergoes change in elevation, the potential energy comes into action. Therefore, the energy change of the system is called the specific potential energy change.

Calculate the potential energy using the following relation:

$$\Delta PE = mg(z_2 - z_1)$$
$$= g\Delta z$$

Here, mass is m, acceleration due to gravity is g, and change in head is $(z_2 - z_1)$.

Substitute, $9.81\ m/s^2$ for g and $100\ m$ for Δz in the relation above, and calculate the energy change in the system.

$$\Delta PE = (9.8) \times (100 - 0) \times \left(\frac{1\ kJ/kg}{1000\ m^2/s^2} \right)$$
$$= 0.98\ kJ/kg$$

Therefore, the specific potential energy change (ΔPE) for the system is $\boxed{0.98\ kJ/kg}$.

Question: 3.045P

A water pump increases the water pressure from 10 psia to 50 psia. Determine the power input required, in hp, to pump 1.2 ft3/s of water. Does the water temperature at the inlet have any significant effect on the required flow power?

Step 1 of 2

Given data:

The initial pressure in the water pump, $P_1 = 10$ psia

The increased pressure in the water pump, $P_2 = 50$ psia

The volume flow rate of the pump, $\dot{v} = 1.2\ \text{ft}^3/\text{s}$

Step 2 of 2

The power input required by the pump is expressed as

$$\begin{aligned}
\dot{W} &= \dot{v}(P_2 - P_1) \\
&= 1.2 \times (50 - 10) \times 144\ \text{lbf} \cdot \text{ft/s} \qquad (\because 1\ \text{ft} = 12\ \text{in}) \\
&= 6912\ \text{lbf} \cdot \text{ft/s} \\
&= \frac{6912}{778.2}\ \text{lbf} \cdot \text{ft/s} \qquad (\because 1\ \text{Btu/s} = 778.2\ \text{lbf} \cdot \text{ft/s}) \\
&= 8.882\ \text{Btu/s} \qquad (\because 1\ \text{Btu/s} = 1.42\ \text{hp}) \\
&= 8.882 \times 1.42\ \text{hp} \\
&= 12.6\ \text{hp}
\end{aligned}$$

$\therefore$ The power input required, $\boxed{\dot{W} = 12.6\ \text{hp}}$

No the temperature has no effect the power outflow.

Question: 3.046P

A classroom that normally contains 40 people is to be air-conditioned with window air-conditioning units of 5-kW cooling capacity. A person at rest may be assumed to dissipate heat at a rate of about 360 kJ/h. There are 10 light-bulbs in the room, each with a rating of 100 W. The rate of heat transfer to the classroom through the walls and the windows is estimated to be 15,000 kJ/h. If the room air is to be maintained at a constant temperature of 21°C, determine the number of window air-conditioning units required.

Step 1 of 6

Given data:

The classroom contains 40 people and 10 bulbs which generates the heat in the room

Cooling capacity of air-conditioning, $Q_{capacity} = 5\ \text{kW}$

Person at rest dissipate heat at a rate of $\dot{Q} = 360\ \text{kJ/h}$

The rate of heat transfer through the walls to be 15000 kJ/h

Step 2 of 6

In order to maintain the classroom at constant temperature we have to estimate the total heat generate.

Heat generated by single person $\dot{Q} = 360\ \text{kJ/h}$

So total heat generated by 40 people $\dot{Q}_p = 40 \times 360\ \text{kJ/h}$

$$= 14400\ \text{kJ/h}$$

$$= \frac{14400}{3600}\ \text{kJ/s} \qquad (\because 1\ \text{hr} = 3600\text{s})$$

$$= 4\ \text{kW}$$

Step 3 of 6

No. of light bulbs in the room = 10

Heat generated due to bulbs $\dot{Q}_b = 10 \times 100\ \text{W}$

$$= 1\ \text{kW}$$

Step 4 of 6

Heat transferred trough the walls $\dot{Q}_{walls} = 15000\ \text{kJ/h}$

$$= \frac{15000}{3600}\ \text{kJ/s}$$

$$= 4.167\ \text{kW}$$

Step 5 of 6

Total heat generated in the classroom $\dot{Q}_{heat} = \dot{Q}_p + \dot{Q}_b + \dot{Q}_{walls}$

$$\dot{Q}_{heat} = 4 + 1 + 4.167$$

$$\dot{Q}_{heat} = 9.167\ \text{kW}$$

Step 6 of 6

The no. of air conditioning units required is $\frac{9.167}{5} = 1.83 \approx 2$

2 air-conditioning units will be required to cool the classroom.

Question: 3.047P

The lighting needs of a storage room are being met by 6 fluorescent light fixtures, each fixture containing four lamps rated at 60 W each. All the lamps are on during operating hours of the facility, which are 6 am to 6 pm 365 days a year. The storage room is actually used for an average of 3 h a day. If the price of electricity is $0.08/kWh, determine the amount of energy and money that will be saved as a result of installing motion sensors. Also, determine the simple payback period if the purchase price of the sensor is $32 and it takes 1 hour to install it at a cost of $40.

Step 1 of 4

Given data:
The total number of the lamps in the storage room is 24 lamps
The power consumed by each bulb is $P = 60\ \text{W}$
The price of electricity, $\$0.08/\text{kWh}$

Step 2 of 4

By using the motion sensor the usage of the lights from 12 hrs can be reduced to 3 hrs, i.e. 9 hrs the lights will be off.
So the number of hours the lights will not be operated in a year is
$9 \times 365 = 3285$ hrs in a year

Step 3 of 4

The amount of energy saved by the lights in a year is $= \dfrac{24 \times 60 \times 3285}{1000}\ \text{kWh/year}$

$= 4730.4\ \text{kWh/year}$

The amount of money saved is $= 4730.4 \times 0.08/\text{year}$

$= \$378.432/\text{year}$

The amount of money saved per month is $\dfrac{\$378.432}{12} = \$31.5$

Step 4 of 4

The total investment cost for the sensor to install is $\$32 + \$40 = \$72$
The pay back period is calculated by

$$PP = \frac{\text{The cost of investment}}{\text{Annual cash inflows}}$$

Therefore the time required to pay back is $PP = \dfrac{\$72}{\$31.5}$

$PP = 2.28$

The pay back period will be about 2 months and 9 days

Question: 3.048P

Consider a room that is initially at the outdoor temperature of 20°C. The room contains a 100-W lightbulb, a 110-W TV set, a 200-W refrigerator, and a 1000-W iron. Assuming no heat transfer through the walls, determine the rate of increase of the energy content of the room when all of these electric devices are on.

Step 1 of 3

The net change (increase or decrease) in the total energy of the system during a process is equal to the difference between the total energy entering and the total energy leaving the system during that process. That is,

$$\begin{pmatrix}\text{Total energy}\\ \text{entering the system}\end{pmatrix}-\begin{pmatrix}\text{Total energy}\\ \text{leaving the system}\end{pmatrix}=\begin{pmatrix}\text{Total energy change}\\ \text{in the system}\end{pmatrix}$$

Or $$E_{in}-E_{out}=\Delta E_{system} \quad \text{---------- (1)}$$

Step 2 of 3

The amount of heat increased in the room due to usage of the electrical equipment is

Light bulb = 100-W

TV set = 110-W

Refrigerator = 200-W

Iron = 1000 W

So total amount of heat input to the room is $E_{in}=100+110+200+1000$

$$E_{in}=1410\ \text{W}$$

Given there is no heat transfer across the walls of the room so $E_{out}=0\ \text{W}$

Step 3 of 3

Therefore from equation (1)

$E_{in}-E_{out}=\Delta E_{system}$

$1410-0=\Delta E_{system}$

$\Delta E_{system}=1410\ \text{W}$

The rate of increase in the energy content of the room is $\boxed{\Delta E_{system}=1410\ \text{W}}$

Question: 3.049P

A fan is to accelerate quiescent air to a velocity of 8 m/s at a rate of 9 m3/s. Determine the minimum power that must be supplied to the fan. Take the density of air to be 1.18 kg/m3.

Step 1 of 3

Given

Velocity of air flow $V = 8\ \text{m/s}$

Rate of flow of air $\dot{V} = 9\ \text{m}^3/\text{s}$

Density of the air $\rho = 1.18\ \text{kg/m}^3$

Step 2 of 3

For a control volume that encloses the fan, the energy balance can be written as

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{system}}{dt}$$

For steady conditions,

$$\frac{dE_{system}}{dt} = 0$$

Hence $\dot{E}_{in} - \dot{E}_{out} = 0$

$$\dot{E}_{in} = \dot{E}_{out}$$

Step 3 of 3

Here Energy input is kinetic energy from fan shaft and this energy is used by the air. So,

$$\dot{W}_{sh,in} = \dot{m}_{air}\frac{V_{out}^2}{2}$$

Where $\dot{m}_{air} = \rho\dot{V}$

$$\dot{m}_{air} = \left(1.18\ \text{kg/m}^3\right)\left(9\ \text{m}^3/\text{s}\right)$$

$$\dot{m}_{air} = 10.62\ \text{kg/s}$$

The minimum power input required is

$$\dot{W}_{sh,in} = \left(10.62\ \text{kg/s}\right)\frac{\left(8\ \text{m/s}^2\right)}{2}\left(\frac{1\ \text{J/kg}}{1\ \text{m}^2/\text{s}^2}\right)$$

$$\dot{W}_{sh,in} = 339.84\ \text{J/s}$$

$$\dot{W}_{sh,in} = 339.84\ \text{W}$$

Hence the minimum power input required is $\boxed{\dot{W}_{sh,in} = 339.84\ \text{W}}$

Question: 3.050P

Consider a fan located in a 3 ft × 3 ft square duct. Velocities at various points at the outlet are measured, and the average flow velocity is determined to be 22 ft/s. Taking the air density to 0.075 lbm/ft3, estimate the minimum electric power consumption of the fan motor.

Step 1 of 4

Given data:

The area of the square duct, $(A) = 9\ \text{ft}^2$

The average velocity of air caused by fan is $(V) = 22\ \text{ft/s}$

The density of air is $(\rho) = 0.075\ \text{lbm/ft}^3$

Step 2 of 4

Considering a control volume across the fan, the energy balance equation in rate form is expressed as

$$\dot{E}_{in} - \dot{E}_{out} = dE_{system}/dt\ \text{kW}$$

For a steady state the derivative term with time will be zero.

$$\dot{E}_{in} = \dot{E}_{out}$$

The minimum power supplied to run the fan is kinetic energy generated by fan

$$\dot{E}_{in} = \dot{W}_{ke} = \dot{m}\frac{V^2}{2}$$

Step 3 of 4

Where $\dot{m}$ is the mass flow rate of the air and is given by the equation

$$\dot{m} = \rho AV\ \text{lbm/s}$$

$$\dot{m} = 0.075 \times 9 \times 22\ \text{lbm/s}$$

$$\dot{m} = 14.85\ \text{lbm/s}$$

Step 4 of 4

Therefore minimum power supplied

$$\dot{E}_{in} = 14.85 \times \frac{22^2}{2}\ \text{lbm/s} \cdot \text{ft}^2/\text{s}^2$$

$$= \frac{3593.7}{25000}\ \text{Btu/s} \qquad \left(\because 1\ \text{Btu/lbm} = 25{,}000\ \text{ft}^2/\text{s}^2\right)$$

$$= 0.1437\ \text{Btu/s}$$

$\therefore$ The minimum power consumption, $\boxed{\dot{E}_{in} = 0.1437\ \text{Btu/s}}$

Question: 3.051P

The driving force for fluid flow is the pressure difference, and a pump operates by raising the pressure of a fluid (by converting the mechanical shaft work to flow energy). A gasoline pump is measured to consume 3.8 kW of electric power when operating. If the pressure differential between the outlet and inlet of the pump is measured to be 7 kPa and the changes in velocity and elevation are negligible, determine the maximum possible volume flow rate of gasoline.

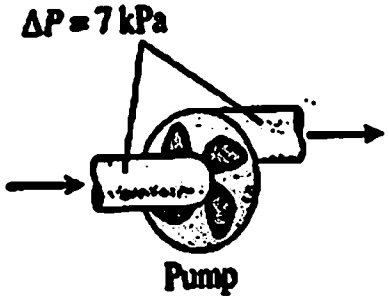


FIGURE P3–51

Step 1 of 1

Electric Power consumed by gasoline pump = 3.8 kW

Pressure difference between inlet and outlet of pump = 7 KPa

Power = pressure difference × volume flowrate

Volume flow rate $\dot{V} = \dfrac{3.8\ \text{kW}}{7\ \text{kPa}}$

$$\dot{V} = 0.5428\ \text{m}^3/\text{s}$$

Hence Volume flow rate of gasoline is $\boxed{\dot{V} = 0.5428\ \text{m}^3/\text{s}}$

Question: 3.052P

An escalator in a shopping center is designed to move 30 people, 75 kg each, at a constant speed of 0.8 m/s at 45° slope. Determine the minimum power input needed to drive this escalator. What would your answer be if the escalator velocity were to be doubled?

Step 1 of 3

Given data:
Mass of a person = 75 kg
Total number of persons = 30
Velocity of the escalator = 0.8 m/s
Angle of inclination = 45°

Step 2 of 3

Total mass on the escalators,

$$M = 30 \times 75$$
$$= 2250 \text{ kg}$$

Question:

An escalator in a shopping center is designed to move 30 people, 75 kg each, at a constant speed of 0.8 m/s at 45° slope. Determine the minimum power input needed to drive this escalator. What would your answer be if the escalator velocity were to be doubled?

Step 1 of 3

Given data:
Mass of a person = 75 kg
Total number of persons = 30
Velocity of the escalator = 0.8 m/s
Angle of inclination = 45°

Step 2 of 3

Total mass on the escalators,

$$M = 30 \times 75$$
$$= 2250 \text{ kg}$$

The actual velocity of the escalator used for lifting the mass is the vertical component of escalator velocity,

$$V_{\text{lift}} = V \sin 45$$
$$= 0.8 \times \sin 45$$
$$= 0.5657 \text{ m/s}$$

Power needed by escalators

$$P = \frac{\Delta PE}{\Delta t}$$
$$= mg\left(\frac{\Delta h}{\Delta t}\right)$$

Question: 3.053P

An automobile moving through the air causes the air velocity (measured with respect to the car) to decrease and fill a larger flow channel. An automobile has an effective flow channel area of 3 m2. The car is traveling at 90 km/h on a day when the barometric pressure is 70 cm of mercury and the temperature is 20°C. Behind the car, the air velocity (with respect to the car) is measured to be 82 km/h, and the temperature is 20°C. Determine the power required to move this car through the air and the area of the effective flow channel behind the car.

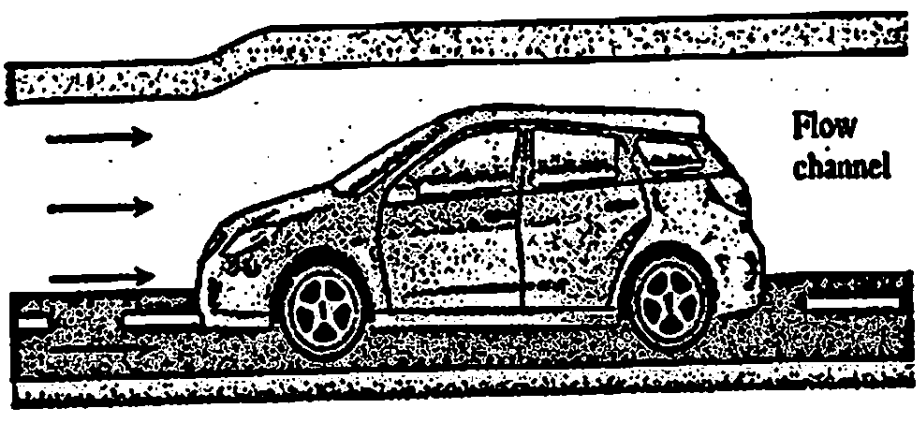


FIGURE P3-53

Step 1 of 5

Use the following conversion factor to convert the pressure from mm of Hg to kPa.

$$1\text{mmHg} = 0.1333\text{kPa}$$

Calculate the pressure in terms of kpa.

$$P = (700\text{mmHg})\left(\frac{0.1333\text{kPa}}{1\text{mmHg}}\right)$$

$$= 93.31\text{kPa}$$

Step 2 of 5

State the ideal gas equation.

$$Pv = RT$$

Here, P is the pressure , v is the specific volume , R is the gas constant and T is the temperature .

Substitute $0.287\text{kJ/kg}\cdot\text{K}$ for R, 293K for T, 93.31kPa for P

$$v = \frac{RT}{P}$$

$$= \frac{0.287 \times 293}{93.31}$$

$$= 0.9012\text{m}^3/\text{kg}$$

Step 1 of 5

Use the following conversion factor to convert the pressure from mm of Hg to kPa.

$$1\text{mmHg} = 0.1333\text{kPa}$$

Calculate the pressure in terms of kpa.

$$P = (700\text{mmHg})\left(\frac{0.1333\text{kPa}}{1\text{mmHg}}\right)$$

$$= 93.31\text{kPa}$$

Step 2 of 5

State the ideal gas equation.

$$Pv = RT$$

Here, P is the pressure , v is the specific volume , R is the gas constant and T is the temperature .

Substitute $0.287\text{kJ/kg}\cdot\text{K}$ for R, 293K for T, 93.31kPa for P

$$v = \frac{RT}{P}$$

$$= \frac{0.287 \times 293}{93.31}$$

$$= 0.9012\text{m}^3/\text{kg}$$

Step 3 of 5

$$= 83.22 \times \frac{(25)^2 - (22.78)^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)$$

$$= 4.413\text{kW}$$

Therefore the power required to move the car is $\boxed{4.413\text{kW}}$

Step 5 of 5

Calculate the outlet area by using the mass flow rate equation.

$$\dot{m} = \frac{A_2 V_2}{v}$$

$$A_2 = \frac{\dot{m} v}{V_2}$$

$$= \frac{83.22 \times 0.9012}{22.78}$$

$$= 3.29\text{m}^2$$

Therefore, the area of the effective flow channel behind the car is $\boxed{3.29\text{m}^2}$

$$= 83.22 \times \frac{(25)^2 - (22.78)^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)$$

$$= 4.413\text{kW}$$

Therefore the power required to move the car is $\boxed{4.413\text{kW}}$

Step 5 of 5

Calculate the outlet area by using the mass flow rate equation.

$$\dot{m} = \frac{A_2 V_2}{v}$$

$$A_2 = \frac{\dot{m} v}{V_2}$$

Question: 3.054P

What is mechanical efficiency? What does a mechanical efficiency of 100 percent mean for a hydraulic turbine?

Step 1 of 2

In the absence of any irreversibility's such as friction, mechanical energy can be converted entirely from one mechanical form to another, and the mechanical efficiency of a device or process can be defined as the ratio of the mechanical energy output to the mechanical energy input.

$$\eta_{mech} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy input}}$$

$$= \frac{E_{mech,out}}{E_{mech,in}}$$

Question: 3.055P

How is the combined pump-motor efficiency of a pump and motor system defined? Can the combined pump-motor efficiency be greater than either the pump or the motor efficiency?

Step 1 of 1

2760-2-59P SA: 2981

SR: 5070 RID1: 3211 | 19/08/2013
The combined pump-motor efficiency of a pump/motor system is defined as the ratio of the increase in mechanical energy of the fluid to the electrical power consumed by the motor.

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}} \times \eta_{\text{motor}}$$

$$= \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{electric,in}}}$$

$$= \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elec,in}}}$$

Question: 3.056P

Define turbine efficiency, generator efficiency, and combined turbine-generator efficiency.

Step 1 of 3

The turbine efficiency is the ratio of energy developed by the turbine to the energy available in the fluid that rotates turbine

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{energy in the fluid}}$$

$$= \frac{\dot{W}_{\text{shaft,out}}}{\left|\Delta \dot{E}_{\text{mech,fluid}}\right|}$$

Hence the turbine efficiency is $\boxed{\eta_{\text{turbine}} = \frac{\dot{W}_{\text{shaft,out}}}{\left|\Delta \dot{E}_{\text{mech,fluid}}\right|}}$

Step 2 of 3

The generator efficiency is the ratio of the electrical power output to the mechanical power input to generator

$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}}$$

$$= \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

Hence the generator efficiency is $\boxed{\eta_{\text{generator}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}}$

Step 3 of 3

The combined turbine generator efficiency is the product of the turbine efficiency and generator efficiency

$$\eta_{\text{turbine-generator}} = \eta_{\text{turbine}}\eta_{\text{generator}}$$

$$= \frac{\dot{W}_{\text{shaft}}}{\left|\Delta \dot{E}_{\text{mech,fluid}}\right|} \times \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft}}}$$

$$= \frac{\dot{W}_{\text{elect,out}}}{\left|\Delta \dot{E}_{\text{mech,fluid}}\right|}$$

Hence the combined turbine generator efficiency is $\boxed{\eta_{\text{turbine-generator}} = \frac{\dot{W}_{\text{elect,out}}}{\left|\Delta \dot{E}_{\text{mech,fluid}}\right|}}$

Question: 3.057P

Can the combined turbine-generator efficiency be greater than either the turbine efficiency or the generator efficiency? Explain.

Step 1 of 1

No, the combined turbine-generator efficiency cannot be grater than the either the turbine efficiency or the generator efficiency.

We know that $\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{gen}}$,

So, the efficiency of turbine and generator will always be less than one, the product will be smaller than the individual efficiencies.

Question: 3.058P

Consider a 24-kW hooded electric open burner in an area where the unit costs of electricity and natural gas are \$0.10/kWh and \$1.20/therm (1 therm = 105,500 kJ), respectively. The efficiency of open burners can be taken to be 73 percent for electric burners and 38 percent for gas burners. Determine the rate of energy consumption and the unit cost of utilized energy for both electric and gas burners.

Step 1 of 3

Given

Power Consumption of burner, $P = 24$ kW

Unit cost of electricity = \$ 0.10/kWh

Unit cost of natural gas = \$ 1.20/therm

Efficiency of electric burner $\eta_{electric} = 73\%$

Efficiency of gas burner $\eta_{gas} = 38\%$

Step 2 of 3

Utilized energy of an electric burner is

$$\dot{Q}_{utilized} = (\text{Energy input}) \times (\text{Efficiency})$$

$$\dot{Q}_{utilized} = (24\ \text{kW})(0.73)$$

$$\dot{Q}_{utilized} = 17.52\ \text{kW}$$

Energy consumption rate, $\boxed{\dot{Q}_{utilized} = 17.52\ \text{kW}}$

The unit cost of utilized energy is inversely proportional to the efficiency,

$$\text{Cost of utilized energy} = (\text{energy input}) \times \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$= 17.52 \times \frac{\$0.1/\text{kWh}}{0.73}$$

$$= \$2.4/\text{h}$$

Cost of utilized energy $\boxed{\$2.4/\text{h}}$

Step 3 of 3

Utilized energy of a gas burner is

$$\dot{Q}_{utilized} = (\text{Energy input}) \times (\text{Efficiency})$$

$$\dot{Q}_{utilized} = (24\ \text{kW})(0.38)$$

$$\dot{Q}_{utilized} = 9.12\ \text{kW}$$

Energy consumption rate $\boxed{\dot{Q}_{utilized} = 9.12\ \text{kW}}$

The unit cost of utilized energy is inversely proportional to the efficiency,

$$\text{Cost of utilized energy} = (\text{energy input}) \times \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$= 9.12 \times \frac{\$1.2/\text{therm}}{0.38}$$

1 therm = 105500 kJ Hence 1 therm = 29.307 kWh

$$= 9.12 \times \frac{\$1.2/\text{therm}}{0.38} \times \frac{1\ \text{therm}}{105500\ \text{kJ}}$$

$$= 9.12 \times \frac{\$1.2/\text{therm}}{0.38} \times \frac{1\ \text{therm}}{29.307\ \text{kWh}}$$

$$= \$0.983/\text{h}$$

Cost of utilized energy $\boxed{\$0.983/\text{h}}$

Question: 3.059P

A 75-hp (shaft output) motor that has an efficiency of 91.0 percent is worn out and is replaced by a high-efficiency 75-hp motor that has an efficiency of 95.4 percent. Determine the reduction in the heat gain of the room due to higher efficiency under full-load conditions.

Step 1 of 4

Given data:

The shaft output power of a motor $\dot{W}_{sh} = 75\text{ hp}$

The efficiency of the motor, $\eta_{motor} = 91\%$

The high-efficiency of the motor, $\eta_{motor-inc} = 95.4\%$

Step 2 of 4

The efficiency of the motor is defined as $\eta_{motor} = \dfrac{\dot{W}_{sh}}{\dot{W}_{elec}}$

The electric input given to motor is $\dot{W}_{elec} = \dfrac{75\times746}{0.91}$ $(\because 1\text{ hp} = 746\text{ W})$

$$= \frac{61483.5}{1000}\text{ kW}$$

$$= 61.48\text{ kW}$$

Step 3 of 4

The electric input given to motor due to increased efficiency

$$\dot{W}_{elec-inc} = \frac{75\times746}{0.954} \quad (\because 1\text{ hp} = 746\text{ W})$$

$$= \frac{58647.79}{1000}$$

$$= 58.647\text{ kW}$$

Step 4 of 4

The reduction of the heat gain of the room due to higher efficiency is

$$\dot{Q} = \dot{W}_{elec} - \dot{W}_{elec-inc}$$

$$= 61.48 - 58.647$$

$$= 2.833\text{ kW}$$

$\therefore$ The reduction in the heat gain, $\boxed{\dot{Q} = 2.833\text{ kW}}$

Question: 3.060P

A 90-hp (shaft output) electric car is powered by an electric motor mounted in the engine compartment. If the motor has an average efficiency of 91 percent, determine the rate of heat supply by the motor to the engine compartment at full load.

Step 1 of 2

Convert the units of input power.

$$\dot{W}_{shaft} = 90\ \text{hp}$$

$$= 90\ \text{hp} \times \left(\frac{746\ \text{W}}{1\ \text{hp}}\right)$$

$$= 67140\ \text{W}$$

$$\dot{W}_{shaft} = 67.14\ \text{kW}$$

Consider the efficiency of the electric motor.

$$\eta_{motor} = \frac{\dot{W}_{shaft}}{\dot{W}_{elec}}$$

$$\dot{W}_{elec} = \frac{\dot{W}_{shaft}}{\eta_{motor}}$$

$$\dot{W}_{elec} = \frac{67140}{0.91}$$

$$= \frac{73780.2}{1000}\ \text{kW}$$

$$= 73.78\ \text{kW}$$

Step 2 of 2

Calculate the heat generated in the electric car.

$$\dot{Q} = \dot{W}_{input} - \dot{W}_{output}$$

$$\dot{Q} = 73.78 - 67.14$$

$$= 6.64\ \text{kW}$$

Therefore, the heat generated is $\boxed{6.64\ \text{kW}}$.

Question: 3.061P

A 75-hp (shaft output) motor that has an efficiency of 91.0 percent is worn out and is to be replaced by a high-efficiency motor that has an efficiency of 95.4 percent. The motor operates 4368 hours a year at a load factor of 0.75. Taking the cost of electricity to be $0.08/kWh, determine the amount of energy and money saved as a result of installing the high-efficiency motor instead of the standard motor. Also, determine the simple payback period if the purchase prices of the standard and high-efficiency motors are $5449 and $5520, respectively.

Step 1 of 6

Given data:

Shaft power output $= 75\,\text{hp}$

Motor efficiency, $\eta_{low} = 91.0\%$

Motor with high efficiency, $\eta_{high} = 95.4\%$

Operating time of the motor $= 4368$ hours

Load factor $= 0.75$

Cost of electricity $= \$0.08/\text{kWh}$

Step 2 of 6

The electric power drawn by each motor and their difference can be expressed as

$$\dot{W}_{\text{electric in}} = \dot{W}_{\text{shaft}} / \eta_{low}$$
$$= (\text{Rated power} \times \text{Load factor}) / \eta_{low}$$

$$\dot{W}_{\text{electric high}} = \dot{W}_{\text{shaft}} / \eta_{high}$$
$$= (\text{Rated power} \times \text{Load factor}) / \eta_{high}$$

Step 3 of 6

$$\therefore \text{Power savings} = \dot{W}_{\text{elec low}} - \dot{W}_{\text{elec high}}$$
$$= (\text{Rated power} \times \text{Load factor}) \times (1/\eta_{low} - 1/\eta_{high})$$

Where η_{low} is the efficiency of the standard motor,

η_{high} is the efficiency of the comparable high-efficiency motor.

Step 4 of 6

Then the annual energy and cost savings associated with the installation of the high-efficiency motor become

$$\text{Energy savings} = \text{Power savings} \times \text{Operating hours}$$
$$= (\text{Rated power} \times \text{Load factor} \times \text{Operating hours}) \times (1/\eta_{low} - 1/\eta_{high})$$

Substituting values we get

$$\text{Therefore energy savings} = (75 \times 0.746 \times 4368 \times 0.75) \times \left[\frac{1}{0.91} - \frac{1}{0.954}\right] \text{kWh/year}$$
$$= 9289.8\ \text{kWh/year}$$

$$\boxed{\text{Energy savings} = 9289.8\ \text{kWh/year}}$$

Step 5 of 6

$$\text{Cost savings} = (\text{Energy savings})\ (\text{Unit cost of energy})$$
$$= 9290 \times \$0.08$$
$$= \$743.2/\text{year}$$

$$\boxed{\text{Cost savings} = \$743.2/\text{year}}$$

Step 6 of 6

$$\text{Also, Excess intial cost} = \text{Purchase price differential}$$
$$= \$5520 - \$5449$$
$$= \$71$$

This gives a simple pay back period

$$\text{Simple pay back period} = \frac{\text{Excess intial cost}}{\text{Annual cost savings}}$$
$$= \frac{\$71}{\$743.2}$$
$$= 0.0955\ \text{years}$$

The pay back period in months can be calculated as

$$0.0955 \times 12 = 1.15\ \text{Months} \qquad (\because 1\ \text{year} = 12\ \text{months})$$

$\therefore$ Simple pay back period, $\boxed{\approx 1\ \text{month 2 days}}$

Question: 3.062P

The steam requirements of a manufacturing facility are being met by a boiler whose rated heat input is 5.5 × 106 Btu/h. The combustion efficiency of the boiler is measured to be 0.7 by a hand-held flue gas analyzer. After tuning up the boiler, the combustion efficiency rises to 0.8. The boiler operates 4200 hours a year intermittently. Taking the unit cost of energy to be $4.35/106 Btu, determine the annual energy and cost savings as a result of tuning up the boiler.

Step 1 of 5

Given

The current rate of heat input to the boiler is given to be $\dot{Q}_{in,current} = 5.5\times10^6$ Btu/h

Operating hours $t = 4200$h

Unit cost of energy $= \$4.35/10^6$ Btu

Step 2 of 5

The heat output of boiler is related to the fuel energy input to the boiler by

Boiler output = (Boiler input)(Combustion efficiency)

$$\dot{Q}_{out} = \dot{Q}_{in}\eta_{furnace}$$

The rate of useful heat output of the boiler becomes

$$\dot{Q}_{out} = \left(\dot{Q}_{in}\eta_{furnace}\right)_{current}$$

$$= \left(5.5\times10^6 \text{ Btu/h}\right)(0.7)$$

$$\dot{Q}_{out} = 3.85\times10^6 \text{ Btu/h}$$

Step 3 of 5

The boiler must supply useful heat at the same rate after the tune up. Therefore, the rate of heat input to the boiler after the tune up and the rate of energy savings become

$$\dot{Q}_{in,new} = \dot{Q}_{out} / \eta_{furnace,new}$$

$$= \left(3.85\times10^6\right)/0.8$$

$$\dot{Q}_{in,new} = 4.81\times10^6 \text{ Btu/h}$$

Step 4 of 5

$$\dot{Q}_{in,saved} = \dot{Q}_{in,current} - \dot{Q}_{in,new}$$

$$= 5.5\times10^6 - 4.81\times10^6$$

$$\dot{Q}_{in,saved} = 0.69\times10^6 \text{ Btu/h}$$

Step 5 of 5

Then the annual energy and cost savings associated with tuning up the boiler become

Energy Savings $= \dot{Q}_{in,saved}$ (Operation hours)

$= (0.69\times10^6 \text{ Btu/h})(4200 \text{ h/year})$

Energy Savings $= 2898\times10^6$ Btu/yr

Cost Savings = (Energy Savings)(Unit cost of energy)

$= (2898\times10^6 \text{ Btu/yr})(\$4.35 \text{ per } 10^6 \text{ Btu})$

Cost Savings = $12606.3/year

Question: 3.063P

Reconsider Prob. 3-62E. Using EES (or other) software, study the effects of the unit cost of energy, the new combustion efficiency on the annual energy, and cost savings. Let the efficiency vary from 0.7 to 0.9, and the unit cost to vary from $4 to $6 per million Btu. Plot the annual energy and cost savings against the efficiency for unit costs of $4, $5, and $6 per million Btu, and discuss the results.

Step 1 of 5

Writing the program of the given conditions

"Given"
Q_dot_in_current=5.5E6 [Btu/h]
eta_furnace_current=0.7
"eta_furnace_new=0.8"
Hours=4200 [h/year]
UnitCost=4.35E-6 [$/Btu]

"Analysis"
Q_dot_out=Q_dot_in_current×η_furnace_current
Q_dot_in_new=Q_dot_out/η_furnace_new
Q_dot_in_saved=Q_dot_in_current-Q_dot_in_new
Energysavings=Q_dot_in_saved×Hours
CostSavings=EnergySavings×UnitCost

Step 2 of 5

Procedure to solve EES software:
File → Open the new file
Copy the given program in the file
Then go to tables → new parametric table
Select the input parameters and set the range of the input quantities
Then solve the program
To obtain the graphs go to plots and select the x and y axis parameters then click ok

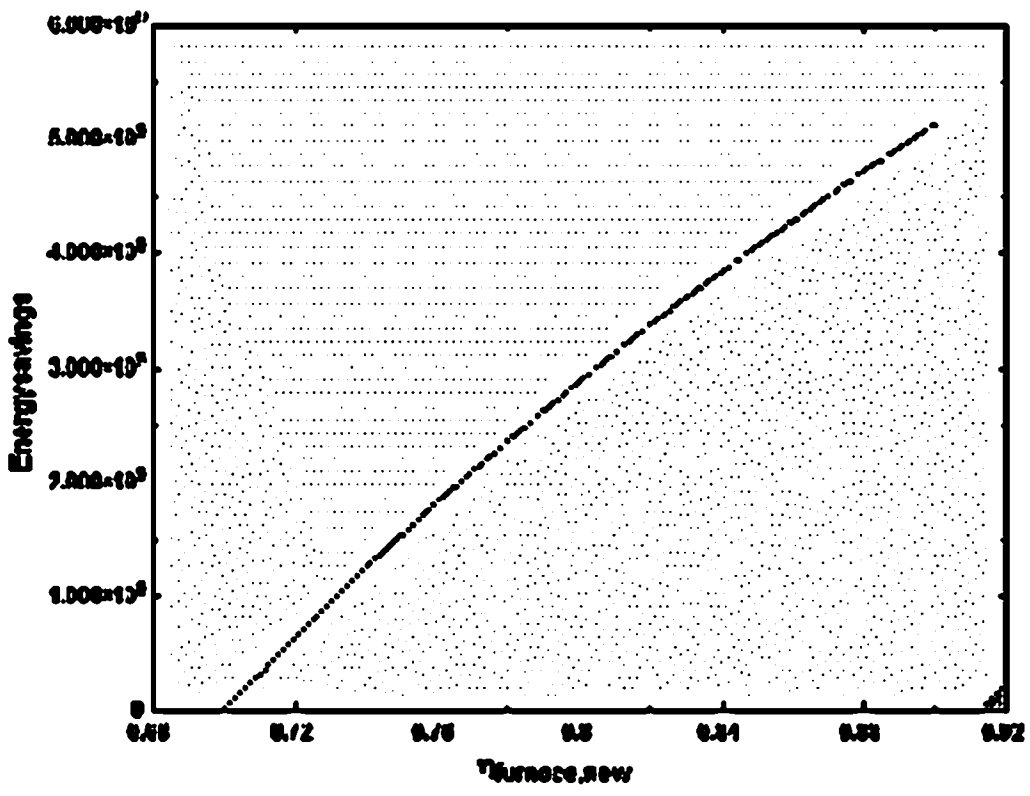
Step 3 of 5

Parametric table:

1..11	1 $\eta_{furnace,new}$	2 Energysavings	3 CostSavings
Run 1	0.7	0	0
Run 2	0.72	6.417E+08	2791
Run 3	0.74	1.249E+09	5432
Run 4	0.76	1.824E+09	7933
Run 5	0.78	2.369E+09	10306
Run 6	0.8	2.888E+09	12561
Run 7	0.82	3.380E+09	14705
Run 8	0.84	3.850E+09	16748
Run 9	0.86	4.298E+09	18695
Run 10	0.88	4.725E+09	20554
Run 11	0.9	5.133E+09	22330

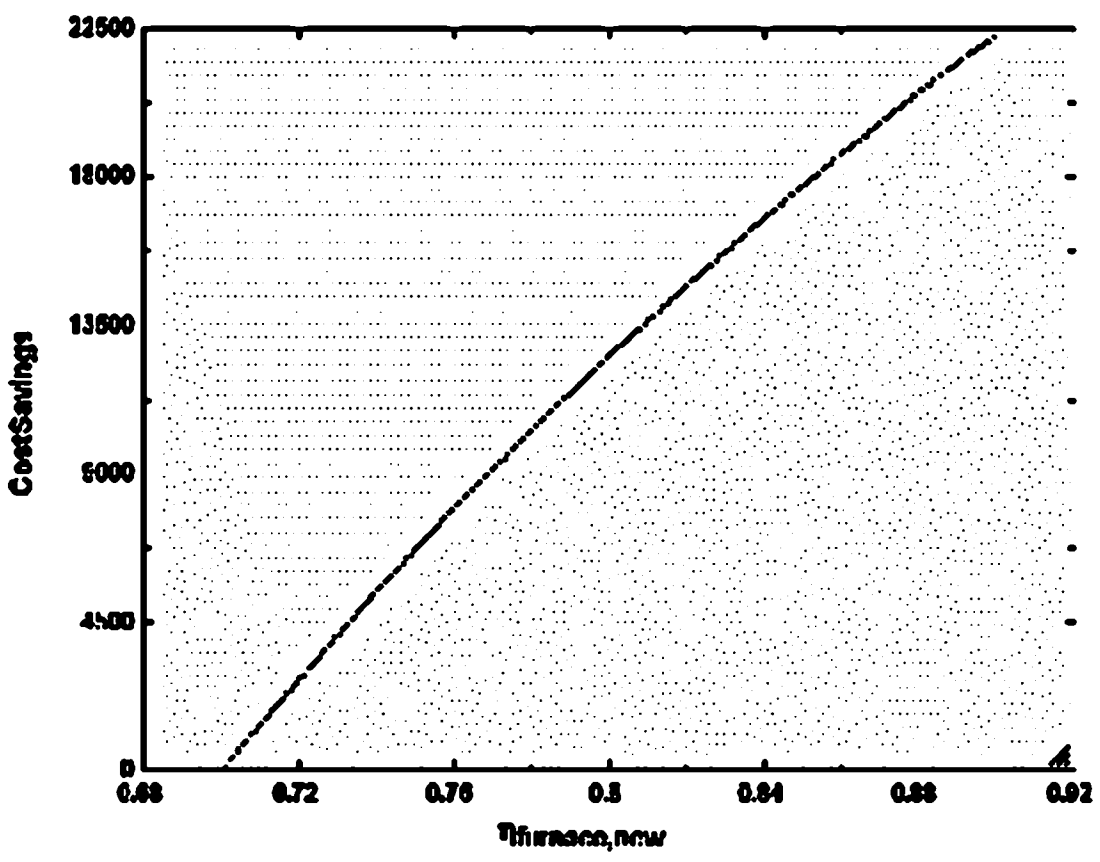
Step 4 of 5

Graph between furnace efficiency and energy savings:



Step 5 of 5

Graph between furnace efficiency and cost savings:



Question: 3.064P

An exercise room has eight weight-lifting machines that have no motors and four treadmills each equipped with a 2.5-hp (shaft output) motor. The motors operate at an average load factor of 0.7, at which their efficiency is 0.77. During peak evening hours, all 12 pieces of exercising equipment are used continuously, and there are also two people doing light exercises while waiting in line for one piece of the equipment. Assuming the average rate of heat dissipation from people in an exercise room is 525 W, determine the rate of heat gain of the exercise room from people and the equipment at peak load conditions.

Step 1 of 2

Calculate the total shaft power output from the motor as follows:

$$\begin{aligned}\dot{W}_{out} &= 4\times 2.5\times 746 \quad (\because 1\text{ hp} = 746\text{ W})\\ &= 7460\text{ W}\end{aligned}$$

Load factor, $= 0.7$

Efficiency, $\eta = 0.77$

We know that efficiency $\eta = \dfrac{\text{Work output}}{\text{Heat input}}$

Calculate the heat generated by the motors as follows:

$$\dot{Q}_{motor} = \frac{\dot{W}_{out}\times load\ factor}{\eta}$$

Here, η is the efficiency.

Substitute 7460 W for $\dot{Q}_{input}$, 0.7 for *load factor*, and 0.77 for η.

$$\begin{aligned}\dot{Q}_{motor} &= \frac{7460\times 0.7}{0.77}\\ &= 6781.8\text{ W}\end{aligned}$$

Step 2 of 2

There are 12 exercising equipment and 2 people doing light exercise, so at peak hours there will be total 14 people and each person dissipates a average heat of 525 W.

Therefore, calculate the heat generated from people as follows:

$$\begin{aligned}\dot{Q}_{people} &= 14\times 525\\ &= 7350\text{ W}\end{aligned}$$

Calculate the rate of heat gain from the people and the motor as follows:

$$\dot{Q}_{total} = \dot{Q}_{motor} + \dot{Q}_{people}$$

Substitute 6781.8 W for $\dot{Q}_{motor}$ and 7350 W for $\dot{Q}_{people}$.

$$\begin{aligned}\dot{Q}_{total} &= 6781.8 + 7350\\ &= 14131.8\text{ W}\end{aligned}$$

Therefore, the rate of heat gain is $\boxed{14131.8\text{ W}}$.

Question: 3.065P

A room is cooled by circulating chilled water through a heat exchanger located in a room. The air is circulated through the heat exchanger by a 0.25-hp (shaft output) fan. Typical efficiency of small electric motors driving 0.25-hp equipment is 54 percent. Determine the rate of heat supply by the fan-motor assembly to the room.

Step 1 of 2

Given data:

The shaft power output of the fan, $\dot{W}_{shaft} = 0.25\ \text{hp}$

The efficiency of the fan motor, $\eta = 54\%$

Step 2 of 2

The efficiency of the motor is defined as $\eta = \dfrac{\text{Energy output}}{\text{Energy input}}$

Here the energy input is the electrical energy, so the electrical energy consumed by the motor will be dissipated as heat.

$$\text{Therefore } \eta = \frac{\dot{W}_{shaft}}{\dot{Q}_{heat}} \quad (\because 1\ \text{hp} = 746\ \text{W})$$

$$\dot{Q}_{heat} = \frac{\dot{W}_{shaft}}{\eta}$$

$$\dot{Q}_{heat} = \frac{0.25 \times 746}{0.54}$$

$$\dot{Q}_{heat} = 345.37\ \text{W}$$

$\therefore$ The rate of heat supply by the fan-motor assembly, $\boxed{\dot{Q}_{heat} = 345.37\ \text{W}}$

Question: 3.066P

The water in a large lake is to be used to generate electricity by the installation of a hydraulic turbine-generator at a location where the depth of the water is 50 m (Fig. 3–62). Water is to be supplied at a rate of 5000 kg/s. If the electric power generated is measured to be 1862 kW and the generator efficiency is 95 percent, determine (*a*) the overall efficiency of the turbine—generator, (*b*) the mechanical efficiency of the turbine, and (c) the shaft power supplied by he turbine to the generator.

Step 1 of 6

Given

Depth of water $h = 50\text{m}$

Rate of water supply $\dot{m} = 5000\text{kg/s}$

Electric power generated $\dot{W} = 1862\text{kW}$

Generator efficiency $\eta_{gen} = 95\%$

Step 2 of 6

(a) We are taking the bottom of the lake as the reference level for convenience. Then kinetic and potential energies of water are zero, and the mechanical energy of water consists of pressure energy only which is

$$e_{mech,in} - e_{mech,out} = \frac{P}{\rho} = gh$$

$$e_{mech,in} - e_{mech,out} = (9.81\text{m/s}^2)(50\text{m})\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right)$$

$$e_{mech,in} - e_{mech,out} = 0.491\text{kJ/kg}$$

Step 3 of 6

The rate at which mechanical energy of fluid supplied to the turbine

$$\left|\Delta\dot{E}_{mech,fluid}\right| = \dot{m}\left(e_{mech,in} - e_{mech,in}\right)$$

$$= (5000)(0.491)$$

$$\left|\Delta\dot{E}_{mech,fluid}\right| = 2455\text{kW}$$

Step 4 of 6

The overall efficiency become

$$\eta_{overall} = \frac{\dot{W}_{elect,out}}{\left|\Delta\dot{E}_{mech,fluid}\right|}$$

$$= \frac{1862}{2455}$$

$$\boxed{\eta_{overall} = 75.8\%}$$

Step 5 of 6

(b)The mechanical efficiency of the turbine is

$$\eta_{turbine} = \frac{\eta_{overall}}{\eta_{generator}}$$

$$= \frac{0.758}{0.95}$$

$$\boxed{\eta_{turbine} = 79.78\%}$$

Step 6 of 6

(c) The shaft power supplied by the turbine to the generator

$$\dot{W}_{shaft,out} = \eta_{turbine}\left|\Delta\dot{E}_{mech,fluid}\right|$$

$$= (0.7978)(2455)$$

$$\boxed{\dot{W}_{shaft,out} = 1958.59\text{kW}}$$

Therefore, the lake supplies 2455 kW of mechanical energy to the turbine, which converts 1958.59 kW of it to shaft work that drives the generator, which generates 1862 kW of electric power.

Question: 3.067P

At a certain location, wind is blowing steadily at 7 m/s. Determine the mechanical energy of air per unit mass and the power generation potential of a wind turbine with 80-m-diameter blades at that location. Also determine the actual electric power generation assuming an overall efficiency of 30 percent. Take the air density to be 1.25 kg/m3.

Step 1 of 4

Given data:

Velocity of the wind blow $V = 7\,\text{m/s}$.

Diameter of the blades $d = 80\,\text{m}$.

Overall efficiency is $\eta_{overall} = 30\%$.

The density of air, $\rho = 1.25\,\text{kg/m}^3$.

Taking the density of water, $\rho = 1000\,\text{kg/m}^3$.

Step 2 of 4

The mechanical energy of air per unit mass is calculated from the kinetic energy of the wind blowing

$$\dot{e}_{mech} = \frac{\dot{m}V^2}{2}$$

Where $\dot{m}$ is the mass flow rate of air

$$\dot{m} = \rho AV$$

$$= 1.25 \times \frac{\pi}{4}(80)^2 \times 7$$

$$= 43982.29\,\text{kg/s}$$

Step 3 of 4

Therefore the mechanical energy and the power generation potential of the wind turbine is calculated as

$$\dot{W} = \dot{E}_{mech} = \frac{\dot{m}V^2}{2}$$

$$= \frac{4398.29 \times 7^2}{2}$$

$$= 1077566.1\,\text{W}$$

$$= 1077.566\,\text{kW}$$

Power generation is $\boxed{\dot{W} = \dot{E}_{mech} = 1077.566\,\text{kW}}$

Step 4 of 4

Then the actual electric power generation with the consideration of an overall efficiency of 30 percent is

$$\dot{W}_{electric} = \eta_{overall} \times \dot{W}$$

$$\dot{W}_{electric} = 0.3 \times 1077.566$$

$$\dot{W}_{electric} = 323.26\,\text{kW}$$

Therefore the electric power is $\boxed{\dot{W}_{electric} = 323.26\,\text{kW}}$

Question: 3.068P

Reconsider Prob. 3–67. Using EES (or other) software, investigate the effect of wind velocity and the blade span diameter on wind power generation. Let the velocity vary from 5 to 20 m/s in increments of 5 m/s, and the diameter vary from 20 to 120 m in increments of 20 m. Tabulate the results, and discuss their significance.

Step 1 of 4

Writing the program of the given conditions

```
"Given"
V=7 [m/s]
D=80 [m]
eta_overall=0.30
rho=1.25 [kg/m^3]
"Analysis"
g=9.81 [m/s^2]
A=pi*D^2/4
m_dot=rho*A*V
W_dot_max=m_dot*V^2/2*Convert(m^2/s^2, kJ/kg)
W_dot_elect=eta_overall*W_dot_max
```

Step 2 of 4

Procedure to solve EES software:
File → Open the new file
Copy the given program in the file
Then go to tables → new parametric table
Select the input parameters and set the range of the input quantities
And select the diameter as a range of 20 to 120m
And select the velocity as a range of 5 to 20m/s
Then solve the program
To obtain the graphs go to plots and select the *x* and *y* axis parameters then click ok

Step 3 of 4

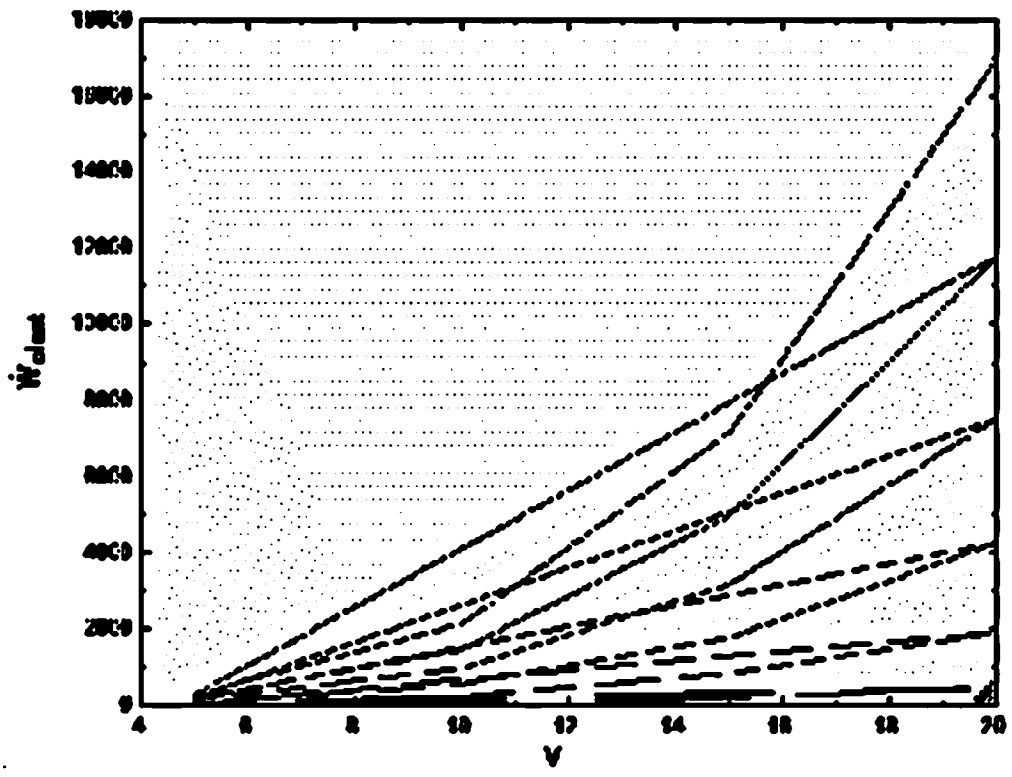
Parametric table:

Table 1

	D [1]	V [2]	m [3]	$\dot{W}_{elect}$ [4]
Run 1	20	5	1[illegible]03	[illegible]
Run 2	20	10	3927	58.[illegible]
Run 3	20	15	8[illegible]90	198.[illegible]
Run 4	20	20	7[illegible]64	471.[illegible]
Run 5	40	5	7[illegible]64	29.4[illegible]
Run 6	40	10	15708	235.[illegible]
Run 7	40	15	[illegible]	[illegible]
Run 8	40	20	31416	1[illegible]
Run 9	60	5	17671	66.27
Run 10	60	10	35343	530.1
Run 11	60	15	53014	1783
Run 12	60	20	70686	4241
Run 13	80	5	31416	117[illegible]
Run 14	80	10	62832	942.[illegible]
Run 15	80	15	94248	3181
Run 16	80	20	125664	7[illegible]
Run 17	100	5	49087	184.1
Run 18	100	10	98175	1473
Run 19	100	15	147262	4971
Run 20	100	20	196350	11781
Run 21	120	5	70686	265.1
Run 22	120	10	141372	2121
Run 23	120	15	212058	7157
Run 24	120	20	282743	1696[illegible]

Step 4 of 4

Graph between velocity and electric power generation with different diameters as follows:



Question: 3.069P

Water is pumped from a lake to a storage tank 20 m above at a rate of 70 L/s while consuming 20.4 kW of electric power. Disregarding any frictional losses in the pipes and any changes in kinetic energy, determine (*a*) the overall efficiency of the pump–motor unit and (*b*) the pressure difference between the inlet and the exit of the pump.

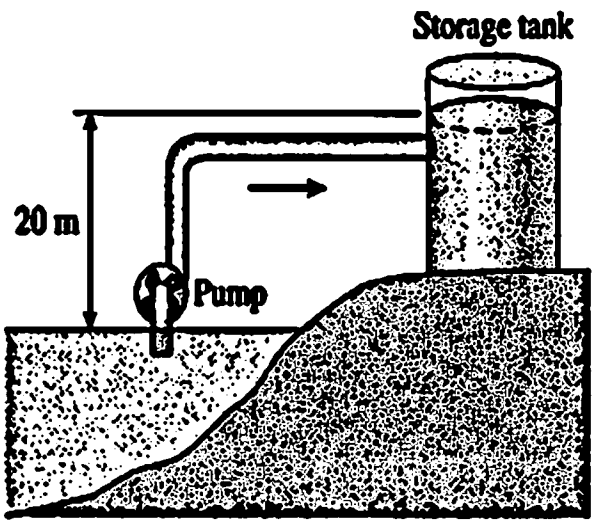


FIGURE P3–69

Step 1 of 5

Given data:

The height of the storage tank, $h = 20\ \text{m}$

The volume flow rate of the pump, $\dot{V} = 70\ \text{L/s}\ \left(\because 1\ \text{L/s} = 10^{-3}\ \text{m}^3/\text{s}\right)$

The electric power supplied the pump, $\dot{W}_{elec} = 20.4\ \text{kW}$

Taking density of water, $\rho = 1000\ \text{kg/m}^3$

Acceleration due to gravity $(g) = 9.8\ \text{m/s}^2$

Step 2 of 5

(a) The overall efficiency of the pump-motor unit:

Neglecting the frictional losses in the pipe and changes in kinetic energy, the power needed to deliver the water into the storage tank is expressed as

$$\dot{E}_{mech} = \dot{m}\,\Delta pe\ \text{kW}$$

Where $\dot{m}$ mass flow rate $\dot{m} = \rho A V$

$$= \rho \dot{V}\ \text{kg/s}\ \left(\because \dot{V} = AV\right)$$

Step 3 of 5

Δpe is the change in specific potential energy

$$\therefore \Delta pe = gh$$

Therefore power delivered by pump

$$\dot{E}_{mech} = \rho \dot{V} gh$$

$$= 1000 \times \frac{70}{1000} \times 9.8 \times 20\ \ (1000\ \text{J/s} = 1\ \text{kW})$$

$$= 13.72\ \text{kW}$$

Step 4 of 5

The efficiency of the fan-motor is expressed as

$$\eta = \frac{\dot{E}_{mech}}{\dot{W}_{elec}}$$

$$= \frac{13.72}{20.4}$$

$$= 0.6725$$

$\therefore$ The overall efficiency of the pump-motor unit, $\boxed{\eta = 67.25\%}$

Step 5 of 5

(b) The pressure difference ΔP between inlet and exit of the pump:

If considering pump alone as a system the pressure difference at the inlet and exit of the pump can be calculated as

$$\dot{E}_{mech} = \dot{V}\,\Delta P_{pump}$$

Therefore the pressure difference

$$\Delta P_{pump} = \frac{\dot{E}_{mech}}{\dot{V}}$$

$$= \frac{13.72}{0.07}$$

$$= 196\ \text{kPa}$$

$\therefore$ The pressure difference, $\boxed{\Delta P_{pump} = 196\ \text{kPa}}$

Question: 3.070P

Large wind turbines with blade span diameters of over 100 m are available for electric power generation. Consider a wind turbine with a blade span diameter of 100 m installed at a site subjected to steady winds at 8 m/s. Taking the overall efficiency of the wind turbine to be 32 percent and the air density to be 1.25 kg/m3, determine the electric power generated by this wind turbine. Also, assuming steady winds of 8 m/s during a 24-hour period, determine the amount of electric energy and the revenue generated per day for a unit price of $0.06/kWh for electricity.

Step 1 of 5

Find the amount of mechanical energy per unit mass generated by the turbine.

$$KE = \frac{1}{2}mv^2$$

$$\frac{KE}{m} = \frac{1}{2}v^2$$

$$\frac{KE}{m} = \frac{1}{2}v^2$$

Here, kinetic energy of the turbine is *KE*, mass of the wind due to which turbine rotates is *m*, and velocity of the turbine is *v*.

Substitute $8\ \text{m/s}$ for *v*.

$$\frac{KE}{m} = \frac{1}{2}(8)^2$$
$$= 32\ \text{J/kg}$$
$$= 0.032\ \text{kJ/kg}$$

Step 2 of 5

Find the power generated from wind turbine using the relation.

$$P_{mech} = \dot{m}\left(\frac{KE}{m}\right)$$
$$= \rho A v\left(\frac{KE}{m}\right)$$
$$= \rho\left(\frac{\pi}{4}d^2\right)v\left(\frac{KE}{m}\right)$$

Here, mass flow rate of the wind is $\dot{m}$, density of air is ρ, area covered by the turbine is *A*, diameter of the turbine wing area is *d*, and velocity of the flow is *V*.

Substitute, $1.25\ \text{kg/m}^3$ for ρ, 100 m for *d*, 8 m/s for *v*, 0.032 kJ/kg

$$P_{mech} = 1.25\left(\frac{\pi}{4}100^2\right)\times 8\times(0.032)$$
$$= 2513.27\ \text{kW}$$

Step 3 of 5

Find the electricity generated using the relation.

$$\eta = \frac{P_{elect}}{P_{mech}}$$

$$P_{elect} = \eta \times P_{mech}$$
$$= 0.32\times(2513.27)$$
$$= 804.2464\ \text{kW}$$

Therefore, the electric power generated is $\boxed{804.2464\ \text{kW}}$.

Step 4 of 5

Find the amount of energy electricity generated if the turbine operated for 24 hours.

$$P_{E\text{-}day} = P_{elect} \times t$$

Here, time of operation is *t*.

Substitute, 804.2464 kW for P_{elect} and 24 hr for *t*.

$$P_{E\text{-}day} = 804.2464\times 24$$
$$= 19301.9136\ \text{kW-h}$$

Therefore, the amount of electric energy generated, $\boxed{19{,}301.9136\ \text{kW-h}}$

Step 5 of 5

Find the amount of revenue generated per day using.

$$C = P_{E\text{-}day} \times u_p$$

Here, unit price of electricity is u_p.

Substitute, 19301.9136 kW-h for $P_{E\text{-}day}$, and $0.06\frac{\$}{\text{kW-h}}$ for u_p.

$$C = 19301.9136\times 0.06$$
$$= 1158.1148\frac{\$}{\text{day}}$$

Therefore, the revenue generated per day is $\boxed{\$1158.1148(\text{per day})}$.

Question: 3.071P

A water pump delivers 6 hp of shaft power when operating. If the pressure differential between the outlet and the inlet of the pump is measured to be 1.2 psi when the flow rate is 15 ft3/s and the changes in velocity and elevation are negligible, determine the mechanical efficiency of this pump.

Step 1 of 3

Given

Power of pump $\dot{W} = 6\,\text{hp}$

Pressure difference between outlet and inlet of the pump is $\Delta P = 1.2\,\text{psi}$

Flow rate of water $Q = 1.5\,\text{ft}^3/\text{s}$

Step 2 of 3

To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{mech,fluid} = \dot{m}\left(e_{mech,out} - e_{mech,in}\right)$$

$$\Delta \dot{E}_{mech,fluid} = \Delta P \times Q$$

$$= \left(15\,\text{ft}^3/\text{s}\right)\left(1.2\,\text{psi}\right)\left(\frac{1\,\text{Btu}}{5.404\,\text{psi}\times\text{ft}^3}\right)$$

$$= 3.33\,\text{Btu/s}$$

We know $1\,\text{hp} = 0.7068\,\text{Btu/s}$

$$\Delta \dot{E}_{mech,fluid} = 3.33 \times \frac{1}{0.7068}$$

$$\Delta \dot{E}_{mech,fluid} = 4.71\,\text{hp}$$

Step 3 of 3

The mechanical efficiency of the pump becomes

$$\eta_{pump} = \frac{\Delta \dot{E}_{mech,fluid}}{\dot{W}_{pump,shaft}}$$

$$= \frac{4.71}{6}$$

$$= 0.785$$

$$\boxed{\eta_{pump} = 78.5\%}$$

Question: 3.072P

Water is pumped from a lower reservoir to a higher reservoir by a pump that provides 20 kW of shaft power. The free surface of the upper reservoir is 45 m higher than that of the lower reservoir. If the flow rate of water is measured to be 0.03 m3/s, determine mechanical power that is converted to thermal energy during this process due to frictional effects.

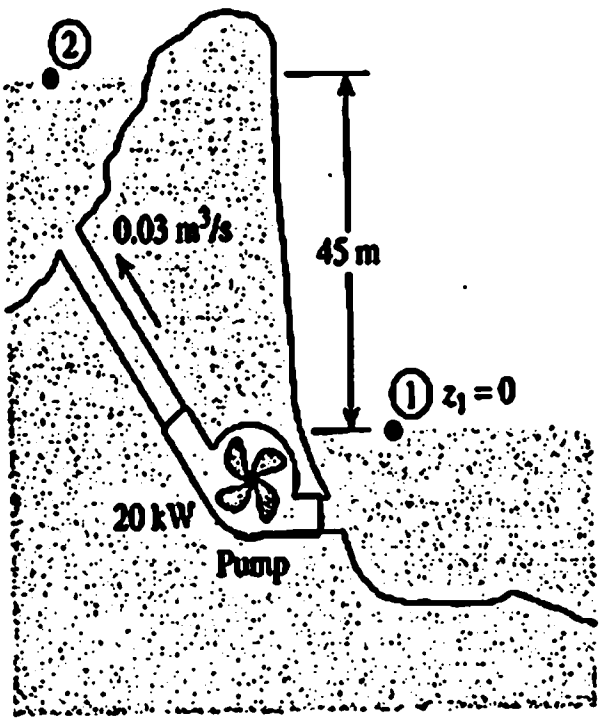


FIGURE P3–72

Step 1 of 4

Calculate the mass flow rate,

$$\dot{m} = \rho A V \text{ kg/s}$$
$$= \rho \dot{V} \text{ kg/s}$$

Here, ρ is the density of the water, and $\dot{V}$ is the volume flow rate of the water

Substitute 1000 kg/m^3 for ρ, and $0.03 \text{ m}^3/\text{s}$ for $\dot{V}$

$$\dot{m} = 1000 \times 0.03$$
$$= 30 \text{ kg/s}$$

Step 2 of 4

Calculate the change in specific potential energy,

$$\Delta pe = gh$$

Here, g is the acceleration due to gravity, and h is the height of the reservoir

Substitute 9.8 m/s^2 for g, and 45 m for h

$$\Delta pe = 9.8 \times 45$$
$$= 441 \text{ m}^2/\text{s}^2$$

Step 3 of 4

Calculate the power needed to deliver the water into the storage tank,

$$\dot{E}_{mech} = \dot{m} \Delta pe$$

Substitute the known values

$$\dot{E}_{mech} = 30 \times 441$$
$$= 13230 \text{ W}$$
$$= 13.23 \text{ kW}$$

Step 4 of 4

Calculate the mechanical power that is converted to thermal energy during this process due to friction effects,

$$\dot{W}_{fric} = \dot{W}_{shaft} - \dot{E}_{mech}$$

Here, $\dot{W}_{shaft}$ is the shaft power of the pump

Substitute 20 kW for $\dot{W}_{shaft}$, and 13.23 kW for $\dot{E}_{mech}$

$$\dot{W}_{fric} = (20 - 13.23)$$
$$= 6.77 \text{ kW}$$

Therefore, the mechanical power due to friction effect, $\dot{W}_{fric}$ is **6.77 kW**

Question: 3.073P

The water behind Hoover Dam in Nevada is 206 m higher than the Colorado River below it. At what rate must water pass through the hydraulic turbines of this dam to produce 100 MW of power if the turbines are 100 percent efficient?



FIGURE P3–73

Step 1 of 2

Given data:

The height of the dam, $h = 206$ m

The power produced by the turbines, $\dot{W} = 100$ MW

The efficiency of the turbine is considered as $\eta = 100\%$

Taking acceleration due to gravity, $g = 9.8\ m/s^2$

Step 2 of 2

The work done on the turbines is due to change in potential energy

$$\dot{W}_{mech} = \dot{m}\Delta pe$$

Δpe is the change in specific potential energy, i.e. $\Delta pe = gh$

$$= \dot{m}gh$$

The mass flow rate is determined from

$$\dot{m} = \frac{\dot{W}}{gh}$$

$$= \frac{100\times10^6}{9.8\times206} \qquad (1\ MW = 10^6\ W)$$

$$= 49{,}534\ kg/s$$

$\therefore$ The mass flow rate through the turbine, $\boxed{\dot{m} = 49{,}534\ kg/s}$

Question: 3.074P

An oil pump is drawing 35 kW of electric power while pumping oil with ρ = 860 kg/m3 at a rate of 0.1 m3/s. The inlet and outlet diameters of the pipe are 8 cm and 12 cm, respectively. If the pressure rise of oil in the pump is measured to be 400 kPa and the motor efficiency is 90 percent, determine the mechanical efficiency of the pump.

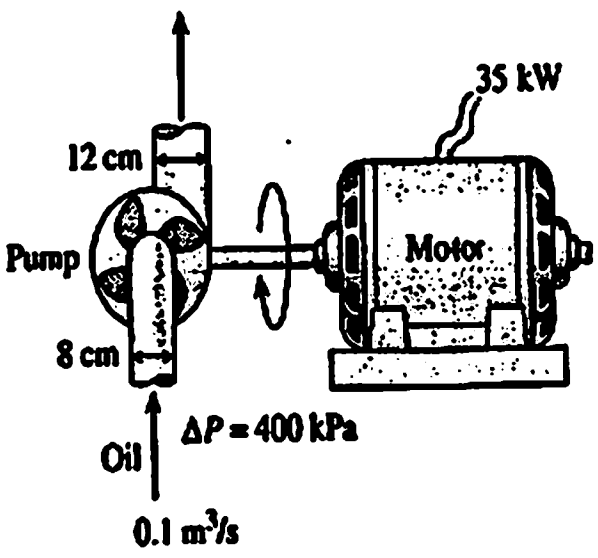


FIGURE P3–74

Step 1 of 6

Given data:

The power supplied to the motor, $\dot{W}_{elec-in} = 35$ kW

The density of oil, $\rho = 860$ kg/m^3

The volume flow rate of the pump, $\dot{V} = 0.1$ m^3/s

The inlet diameter of the pump, $d_{in} = 8$ cm

$= 0.08$ m

The outlet diameter of the pump, $d_{out} = 12$ cm

$= 0.12$ m

The pressure rise across the pump, $\Delta P = 400$ kPa

The efficiency of the motor, $\eta_{motor} = 90\%$

Step 2 of 6

The total mechanical energy by neglecting potential energy is given by

$$\dot{E}_{mech} = \dot{m}(P_2 v_2 - P_1 v_1) + \dot{m}\frac{V_2^2 - V_1^2}{2}$$

where $\Delta P = (P_2 - P_1)$,

and $\dot{m} = \frac{\dot{V}}{v}$

$\because$ volume flow rate remains constant $\dot{V} = \dot{m}v$ $\quad (\because v_1 = v_2 = v)$

$$\dot{E}_{mech} = \dot{m}v(P_2 - P_1) + \dot{m}\frac{V_2^2 - V_1^2}{2}$$

$$\dot{E}_{mech} = \dot{V}(P_2 - P_1) + (\dot{V} \times \rho)\frac{V_2^2 - V_1^2}{2} \quad \text{---------- (1)}$$

Step 3 of 6

The initial velocity $V_1 = \frac{\dot{V}}{A}$

$$= \frac{0.1}{\left(\frac{\pi \times 0.08^2}{4}\right)}$$

$= 19.89$ m/s

Similarly final velocity

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.1}{\left(\frac{\pi}{4} \times 0.12^2\right)}$$

$= 8.84$ m/s

Step 4 of 6

By substituting values in the equation (1) we get

Therefore the power delivered by pump is

$$\dot{E}_{mech} = (0.1) \times (400) + (0.1) \times 860\left(\frac{8.84^2 - 19.89^2}{2}\right)\left(\frac{1}{1000}\right) \text{kW}$$

$= 26.3$ kW

Step 5 of 6

The shaft power of the pump is

$$\dot{W}_{shaft} = \eta_{motor}\dot{W}_{elec-in}$$

$= 0.9(35)$

$= 31.5$ kW

Step 6 of 6

Therefore the mechanical efficiency of the pump is expressed as

$$\eta_{pump} = \frac{\dot{E}_{mech}}{\dot{W}_{shaft}}$$

$$= \frac{26.3}{31.5}$$

$= 0.8349$

$\therefore$ The mechanical efficiency of the pump, $\boxed{\eta_{pump} = 83.49\%}$

Question: 3.075P

An 80-percent efficient pump with a power input of 20 hp is pumping water from a lake to a nearby pool at a rate of 1.5 ft3/s through a constant-diameter pipe. The free surface of the pool is 80 ft above that of the lake. Determine the mechanical power used to overcome frictional effects in piping.

Step 1 of 4

Given

Power input of the pump $\dot{W}_{pump} = 20\,\text{hp}$

Efficiency of the pump $\eta = 80\%$

Flow rate of water $Q = 1.5\,\text{ft}^3/\text{s}$

Height of the pool from the lake $z = 80\,\text{ft}$

Step 2 of 4

Specific weight of water

$\rho = 62.4\,\text{lbm/ft}^3$

The useful mechanical pumping power delivered to water is

$$\dot{W}_{pump,u} = \eta_{pump}\dot{W}_{pump}$$
$$= (0.80)(20)$$
$$\dot{W}_{pump,u} = 16\,\text{hp}$$

Step 3 of 4

The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy

$$\Delta\dot{E}_{mech} = \dot{m}\Delta pe$$
$$= \dot{m}g\Delta z$$
$$\Delta\dot{E}_{mech} = \rho Qg\Delta z$$

Substituting the known values rate of change of mechanical energy of water becomes

$$\Delta\dot{E}_{mech} = \left(62.4\,\frac{\text{lbm}}{\text{ft}^3}\right)\left(1.5\,\frac{\text{ft}^3}{\text{s}}\right)\left(32.2\,\frac{\text{ft}}{\text{s}^2}\right)(80\,\text{ft})\left(\frac{1\,\text{lbf}}{32.2\,\text{lbm}\times\text{ft/s}^2}\right)\left(\frac{1\,\text{hp}}{550\,\text{lbf}\times\text{ft/s}}\right)$$

$$\Delta\dot{E}_{mech} = 13.614\,\text{hp}$$

Step 4 of 4

Then the mechanical power lost in piping because of frictional effects becomes

$$\dot{W}_{fric} = \dot{W}_{pump,u} - \Delta\dot{E}_{mech}$$
$$= 16 - 13.614$$
$$\boxed{\dot{W}_{fric} = 2.386\,\text{hp}}$$

Question: 3.076P

A wind turbine is rotating at 15 rpm under steady winds flowing through the turbine at a rate of 42,000 kg/s. The tip velocity of the turbine blade is measured to be 250 km/h. If 180 kW power is produced by the turbine, determine (*a*) the average velocity of the air and (*b*) the conversion efficiency of the turbine. Take the density of air to be 1.31 kg/m3.

Step 1 of 5

Given data:

The rpm of the wind turbine is $(N) = 15$ rpm

The mass flow rate trough the turbine $(\dot{m}) = 42,000$ kg/s

The tip velocity of the blade $(V_{tip}) = 250$ km/h

The power output from the turbine $(\dot{W}_{out}) = 180$ kW

Taking the density of air $(\rho) = 1.31\ \text{kg/m}^3$

Step 2 of 5

(a) The average velocity of the air

We know that mass flow rate $\dot{m} = \rho A V$ kg/s

The velocity is expressed as $V_{tip} = \dfrac{\pi dN}{60}$ m/s

The diameter of the turbine can be calculated as

$$d = \frac{V_{tip} \times 60}{\pi N}\ \text{m}$$

$$= \frac{(250 \times \frac{5}{18})(60)}{\pi(15)} \qquad \left(\because 1\ \text{km/h} = \frac{5}{18}\ \text{m/s}\right)$$

$$= 88.42\ \text{m}$$

Step 3 of 5

The average velocity of the wind turbine can be determined as

$$V = \frac{\dot{m}}{\rho A}\ \text{m/s}$$

Where $A = \dfrac{\pi}{4} \times d^2\ \text{m}^2$

$$= \frac{42,000}{(1.31)(\frac{\pi}{4} \times 88.42^2)}$$

$$= 5.22\ \text{m/s}$$

∴ The average velocity of the wind turbine, $\boxed{V = 5.22\ \text{m/s}}$

Step 4 of 5

(b) The conversion efficiency of turbine

Here efficiency is defined as $\eta_{turbine} = \dfrac{\dot{W}_{out}}{\dot{W}_{ke}}$

The kinetic energy of the turbine is expressed as

$$\dot{W}_{ke} = \frac{1}{2}\dot{m}V^2$$

$$= \frac{1}{2} \times 42,000 \times (5.22)^2$$

$$= \frac{572216}{1000}\ \text{kW}$$

$$= 572.2\ \text{kW}$$

Step 5 of 5

Therefore efficiency $\eta_{turbine} = \dfrac{\dot{W}_{out}}{\dot{W}_{ke}}$

$$= \frac{180}{572.2}$$

$$= 0.3145$$

∴ The efficiency of the turbine, $\boxed{\eta_{turbine} = 31.45\%}$

Question:

3.077P

Consider a classroom for 55 students and one instructor, each generating heat at a rate of 100 W. Lighting is provided by 18 fluorescent lightbulbs, 40 W each, and the ballasts consume an additional 10 percent. Determine the rate of internal heat generation in this classroom when it is fully occupied.

Step 1 of 5

Given data:
The total number of people in the classroom = 56
Heat generated by each person, $Q = 100$ W
There are 18 fluorescent light bulbs of 40 W each
The ballasts consume 10% of the heat generated by bulbs

Step 2 of 5

The rate of internal heat generation in the classroom when it is fully occupied is the sum of heat generated due to light bulbs, ballasts and persons is given by

$\dot{Q}$ $\dot{Q}$ $\dot{Q}$ W

Question:

Consider a classroom for 55 students and one instructor, each generating heat at a rate of 100 W. Lighting is provided by 18 fluorescent lightbulbs, 40 W each, and the ballasts consume an additional 10 percent. Determine the rate of internal heat generation in this classroom when it is fully occupied.

Step 1 of 5

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Step 2 of 5

The rate of internal heat generation in the classroom when it is fully occupied is the sum of heat generated due to light bulbs, ballasts and persons is given by

$\dot{Q}$ $\dot{Q}$ $\dot{Q}$ W

$$\dot{Q}_{total} = \dot{Q}_{people} + \dot{Q}_{lights}$$
$$= 5600 + 720 + 72$$
$$= 6392 \text{ W}$$

∴ Total heat generated, $\boxed{\dot{Q}_{total} = 6392 \text{ W}}$

Question: 3.078P

Consider a homeowner who is replacing his 25-year-old natural gas furnace that has an efficiency of 55 percent. The homeowner is considering a conventional furnace that has an efficiency of 82 percent and costs $1600 and a high-efficiency furnace that has an efficiency of 95 percent and costs $2700. The homeowner would like to buy the high-efficiency furnace if the savings from the natural gas pay for the additional cost in less than 8 years. If the homeowner presently pays $1200 a year for heating, determine if he should buy the conventional or high-efficiency model.

Step 1 of 4

Given data:

The old heater furnace efficiency, $\eta_{old\ furnace} = 55\%$

The efficiency of a conventional furnace, $\eta_c = 82\%$

The efficiency of a high efficient furnace, $\eta_h = 95\%$

The cost of a conventional furnace, $1600

The cost of a high efficient furnace, $2700

Step 2 of 4

The home owner pays $1200 a year for heating the furnace. The old heater has an efficiency of 55%

So the useful amount utilized for heating is $= 0.55 \times \$1200$

$= \$660/year$

The cost utilized by conventional furnace is $\dfrac{\$660/year}{0.82}$

$= \$805/year$

The cost utilized by high efficient furnace is $= \dfrac{\$660/year}{0.95}$

$= \$695/year$

Step 3 of 4

The excess initial cost of the efficient heater is calculated as

$= 2700 - 1600$

$= \$1100$

The annual saving cost with efficient heating is calculated as

$= 805 - 695$

$= \$110$

Step 4 of 4

Thus the simple pay back period is given by

$$\text{Simple pay back period} = \frac{\text{Excess intial cost}}{\text{Annual cost savings}}$$

$$= \frac{\$1100}{\$110}$$

$$= 10 \text{ years}$$

From the above we know that the high efficient furnace will cost high heating than the conventional type. So in this case purchase less efficient and cheaper heater.

Question: 3.079P

The energy contents, unit costs, and typical conversion efficiencies of various energy sources for use in water heaters are given as follows: 1025 Btu/ft3, $0.012/ft3 and 55 percent for natural gas; 138,700 Btu/gal, $ 1.15/gal, and 55 percent for heating oil; and 1 kWh/kWh, $0.084/kWh, and 90 percent for electric heaters, respectively. Determine the lowest-cost energy source for water heaters.

Step 1 of 5

Given data:

Natural gas heaters:

Energy content, $E = 1025\,\text{Btu/ft}^3$

Unit cost $= \$0.012/\text{ft}^3$

Efficiency, $\eta = 55\%$

Oil heating:

Energy content, $E = 138700\,\text{Btu/gal}$

Unit cost $= \$1.15/\text{gal}$

Efficiency, $\eta = 55\%$

Electric heaters:

Energy content, $E = 1\,\text{kWh/kWh}$

Unit cost $= \$0.084/\text{kWh}$

Efficiency, $\eta = 90\%$

Step 2 of 5

The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

Calculating the various energy sources used in water heaters.

Step 3 of 5

Natural gas heaters:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$\text{Cost of utilized energy} = \frac{\$0.012}{0.55} \times \frac{1}{1025} \qquad (\because 1\,\text{kWh} = 3410\,\text{Btu})$$

$$\text{Cost of utilized energy} = \$2.12 \times 10^{-5}\ /\text{Btu}$$

Step 4 of 5

Oil heating heaters:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$\text{Cost of utilized energy} = \frac{\$1.25}{0.55} \times \frac{1}{138,700}$$

$$\text{Cost of utilized energy} = \$163 \times 10^{-5}\ /\text{Btu}$$

Step 5 of 5

Electrical heaters:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$\text{Cost of utilized energy} = \frac{\$0.084}{0.90} \times \frac{1}{3410}$$

$$\text{Cost of utilized energy} = \$2.73 \times 10^{-5}\ /\text{Btu}$$

So from the above 3 cases the lowest cost energy source for water heating is by using heating oil. With a lowest cost of $\boxed{\$1.63 \times 10^{-5}\ /\text{Btu}}$

Question: 3.080P

A homeowner is considering these heating systems for heating his house: Electric resistance heating with \$0.09/kWh and 1 kWh = 3600 kJ, gas heating with \$1.24/therm and 1 therm = 105,500 kJ, and oil heating with \$1.25/gal and 1 gal of oil = 138,500 kJ. Assuming efficiencies of 100 percent for the electric furnace and 87 percent for the gas and oil furnaces, determine the heating system with the lowest energy cost.

Step 1 of 5

Given data:
Electric heaters:
Unit cost = \$0.09/kWh
Where 1kWh = 3600kJ
Efficiency, $\eta = 100\%$
Gas heaters:
Unit cost = \$1.24/therm
Where 1therm = 105,500kJ
Efficiency, $\eta = 87\%$
Oil heating:
Unit cost = \$1.25/gal
Where 1gal = 138,500kJ
Efficiency, $\eta = 87\%$

Step 2 of 5

The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

Calculating the various energy sources used for heating a house

Step 3 of 5

Electrical heating:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$\text{Cost of utilized energy} = \frac{\$0.09}{1} \times \frac{1}{3600}$$

$$\text{Cost of utilized energy} = \$2.5 \times 10^{-5}\ /\text{kJ}$$

Step 4 of 5

Gas heating:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{}$$

Unit cost = \$1.25/gal
Where 1gal = 138,500kJ
Efficiency, $\eta = 87\%$

Step 2 of 5

The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

Calculating the various energy sources used for heating a house

Step 3 of 5

Electrical heating:

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}}$$

$$\text{Cost of utilized energy} = \frac{\$0.09}{1} \times \frac{1}{3600}$$

Question: 3.081P

A typical household pays about $1200 a year on energy bills, and the U.S. Department of Energy estimates that 46 percent of this energy is used for heating and cooling, 15 percent for heating water, 15 percent for refrigerating and freezing, and the remaining 24 percent for lighting, cooking, and running other appliances. The heating and cooling costs of a poorly insulated house can be reduced by up to 30 percent by adding adequate insulation. If the cost of insulation is $200, determine how long it will take for the insulation to pay for itself from the energy it saves.

Step 1 of 3

Given data:

Nitrogen oxides released by the car per year, $(NO_x)_{car} = 11\ \text{kg}$

Nitrogen oxides released from natural gas, $(NO_x)_{gas} = 4.3\ \text{g/therm}$

Nitrogen oxides released for power generation, $(NO_x)_{power} = 7.1\ \text{g/kWh}$

Power consumption of the house per year, $E_{power} = 9000\ \text{kWh}$

Natural gas consumption of the house per year, $E_{gas} = 1200\ \text{therms}$

Step 2 of 3

The Nitrogen oxides released by the two cars per year,

$$\begin{aligned} 2(NO_x)_{car} &= 2\times 11 \\ &= 22\ \text{kg} \end{aligned}$$

The Nitrogen oxides released due to power consumption per year,

$$\begin{aligned} E_{gas}\times(NO_x)_{gas} &= (1200\ \text{therms})\times(4.3\ \text{g/therm}) \\ &= 5160\ \text{g} \\ &= 5.16\ \text{kg} \end{aligned}$$

The Nitrogen oxides released due to natural gas consumption per year,

$$\begin{aligned} E_{power}\times(NO_x)_{power} &= (9000\ \text{kWh})\times(7.1\ \text{g/kWh}) \\ &= 63900\ \text{g} \\ &= 63.9\ \text{kg} \end{aligned}$$

Step 3 of 3

The total amount of nitrogen oxides released per year by the house is,

$$\begin{aligned} (NO_x)_{house} &= \left[2(NO_x)_{car}\right]+\left[E_{gas}\times(NO_x)_{gas}\right]+\left[E_{power}\times(NO_x)_{power}\right] \\ &= 22+5.16+63.9 \\ &= 91.06\ \text{kg} \end{aligned}$$

Hence the amount of nitrogen oxides released per year, $\boxed{(NO_x)_{house} = 91.06\ \text{kg}}$

Question: 3.082P

The U.S. Department of Energy estimates that up to 10 percent of the energy use of a house can be saved by caulking and weatherstripping doors and windows to reduce air leaks at a cost of about $60 for materials for an average home with 12 windows and 2 doors. Caulking and weather-stripping every gas-heated home properly would save enough energy to heat about 4 million homes. The savings can be increased by installing storm windows. Determine how long it will take for the caulking and weatherstripping to pay for itself from the energy they save for a house whose annual energy use is $1500.

Step 1 of 2

Given
Percentage of energy saved by caulking and weather stripping doors and windows is 10%
Cost of annual energy used is $1500
Money saved by air leaks at a cost of $60

Step 2 of 2

The amount of money that would be saved per year is determined directly from
Money saved = ($1500/year)(0.10)
Money saved= $150/yr
Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}}$$

$$\text{Payback period} = \frac{\$60}{\$150/\text{yr}}$$

$$\boxed{\text{Payback period} = 0.4\,\text{yr}}$$

Therefore, the proposed measure will pay for it less than half a year.

Question: 3.083P

The energy stored in the spring of a railroad car is 5000 lbf·ft. What is this energy in (*a*) lbm, ft, and s units; (*b*) lbf and yd units; and (*c*) lbm, mile, and hour units?

Step 1 of 3

Given

Energy stored in the spring is $W = 5000\,\text{lbf-ft}$

Using appropriate conversion factors, we obtain

(a) $$W = (5000\,\text{lbf.ft})\left(\frac{32.174\,\text{lbm.ft/s}^2}{1\,\text{lbf}}\right)$$

$$\boxed{W = 160{,}870\,\text{lbm.ft}^2/\text{s}^2}$$

Step 2 of 3

(b) $$W = (5000\,\text{lbf}\cdot\text{ft})\left(\frac{0.33303\,\text{yd}}{1\,\text{ft}}\right)$$

$$\boxed{W = 1665.15\,\text{lbm}\cdot\text{yd}}$$

Step 3 of 3

(c) $$W = (5000\,\text{lbf.ft})\left(\frac{32.174\,\text{lbm.ft/s}^2}{1\,\text{lbf}}\right)\left(\frac{1\,\text{mile}}{5280\,\text{ft}}\right)^2\left(\frac{3600\,\text{s}}{1\,\text{h}}\right)^2$$

$$\boxed{W = 74784.607\,\text{mile}^2/\text{h}^2}$$

Question: 3.084P

The force required to expand the gas in a gas spring a distance x is given by Constant

$$F = \frac{\text{Constant}}{x^k}$$

where the constant is determined by the geometry of this device and k is determined by the gas used in the device. One such device has a constant of 200 lbf·in1.4 and k =1.4. Determine the work, in Btu, required to compress this device from 1 in to 4 in.

Step 1 of 2

Given

Force required in expanding the gas in a gas spring, by a distance x is

$$F = \frac{\text{constant}}{x^k}$$

$\text{Constant} = 200 \text{lbf-in}^{1.4}$

Isentropic index $k = 1.4$

Compress the device from 1in to 4in

Step 2 of 2

The work integral relation

$$W = \int_1^2 F ds$$

$$= \int_1^2 \frac{\text{Constant}}{x^k} dx$$

$$= \frac{\text{Constant}}{1-k}\left(x_2^{1-k} - x_1^{1-k}\right)$$

Where $x_2 = 4\text{in}$ and $x_1 = 1\text{in}$

Then

$$W = \frac{200}{1-1.4}\left[(4)^{-0.4} - (1)^{-0.4}\right]$$

$$W = 212.825 \text{lbf-in}$$

$$W = 212.825 \times \frac{1}{12}$$

$$W = 17.735 \text{lbf-ft}$$

$$W = 17.735 \times \frac{1}{778}$$

$$\boxed{W = 0.02279 \text{Btu}}$$

Question: 3.085P

A man weighing 180 lbf pushes a block weighing 100 lbf along a horizontal plane. The dynamic coefficient of friction between the block and plane is 0.2. Assuming that the block is moving at constant speed, calculate the work required to move the block a distance of 100 ft considering (*a*) the man and (*b*) the block as the system. Express your answers in both lbf·ft and Btu.

Step 1 of 2

a)

Consider man as the system.

Find the work required to move the block by using the formula.

$$W = \mu F d$$
$$= \mu (W_B + W_m) d$$

Here, μ is the coefficient of the dynamic friction, W_B is weight of the block, d is the distance travelled by the block, and W_m is the weight of the man.

Substitute 0.2 for μ, 100 lbf for W_B, 180 lbf for W_m, and 100 ft for d.

$$W = 0.2(180+100)(100)$$
$$= 5600\,\text{lbf}\cdot\text{ft}$$
$$= (5600\,\text{lbf}\cdot\text{ft})\left(\frac{1}{778}\frac{\text{Btu}}{\text{lbf}\cdot\text{ft}}\right)$$
$$= 7.198\,\text{Btu}$$

Therefore, the work required in $\text{lbf}\cdot\text{ft}$ is $\boxed{5600\,\text{lbf}\cdot\text{ft}}$.

Therefore, the work required in Btu is $\boxed{7.198\,\text{Btu}}$.

Step 2 of 2

b)

Consider block as the system.

Find the work required to move the block by using the formula.

$$W = \mu F d$$
$$= \mu (W_B) d$$

Substitute 0.2 for μ, 100 lbf for W_B and 100 ft for d.

$$W = 0.2(100)(100)$$
$$= 2000\,\text{lbf}\cdot\text{ft}$$
$$= (2000\,\text{lbf}\cdot\text{ft})\left(\frac{1}{778}\frac{\text{Btu}}{\text{lbf}\cdot\text{ft}}\right)$$
$$= 2.57\,\text{Btu}$$

Therefore, the work required in $\text{lbf}\cdot\text{ft}$ is $\boxed{2000\,\text{lbf}\cdot\text{ft}}$.

Therefore, the work required in Btu is $\boxed{2.57\,\text{Btu}}$.

Question: 3.086P

Water is pumped from a 200-ft-deep well into a 100-ft-high storage tank. Determine the power, in kW, that would be required to pump 200 gallons per minute.

Step 1 of 2

Given data:

The difference between the heights trough which water should be pumped is

$$\Delta h = 200 - (-100)$$
$$= 300\,\text{ft}$$

Take density of water, $\rho = 62.4\,\text{lbm/ft}^3$

Acceleration due to gravity, $g = 32.174\,\text{ft/s}^2$

Discharge through the pump, $\dot{V} = 200\,\text{gal/min}$
$$= 0.4456\,\text{ft}^3/\text{s}$$

Step 2 of 2

The power required to pump the water trough a given height is expressed as

$$\dot{P} = \dot{m}g\Delta h$$
$$= \rho \dot{V} g \Delta h$$
$$= (62.4)(0.4456)(32.174)(300)\,\frac{\text{lbm-ft}^2}{s^3}$$
$$= \left(268392.78\,\frac{\text{lbm}\cdot\text{ft}^2}{s^3}\right)\left(\frac{1}{32.174}\,\frac{\text{lbf}}{\text{lbm}\cdot\text{ft/s}^2}\right)$$
$$= 8341.9\,\frac{\text{lbf-ft}}{s}$$
$$= \frac{8341.9}{737.56}\,\text{kW}$$
$$= 11.3\,\text{kW}$$

∴ The power required, $\boxed{\dot{P} = 11.3\,\text{kW}}$

Question: 3.087P

A grist mill of the 1800s employed a water wheel that was 14 m high; 320 liters per minute of water flowed on to the wheel near the top. How much power, in kW, could this water wheel have produced?

Step 1 of 3

Given

Height of water wheel $z = 14\,\text{m}$

$$\text{Flow rate of water} = Q = \frac{320\,\text{litres}}{\text{min}}$$

$$Q = \frac{320\times10^{-3}}{60}\,\text{m}^3/\text{s}$$

$$Q = 0.00533\,\text{m}^3/\text{s}$$

Step 2 of 3

Mass flow rate

$$\dot{m} = \rho \times Q$$

Where ρ is density of water

$$\dot{m} = 1000\times0.00533$$

$$\dot{m} = 5.333\,\text{kg/s}$$

Step 3 of 3

Power produced by water wheel

$$\dot{W} = \dot{m}gz$$

$$= 5.333\times9.81\times14$$

$$\dot{W} = 732.48\,\text{W}$$

$$\boxed{\dot{W} = 0.732\,\text{kW}}$$

Question: 3.088P

Windmills slow the air and cause it to fill a larger channel as it passes through the blades. Consider a circular windmill with a 7-m-diameter rotor in a 10 m/s wind on a day when the atmospheric pressure is 100 kPa and the temperature is 20°C. The wind speed behind the windmill is measured at 9 m/s. Determine the diameter of the wind channel downstream from the rotor and the power produced by this windmill, presuming that the air is incompressible.

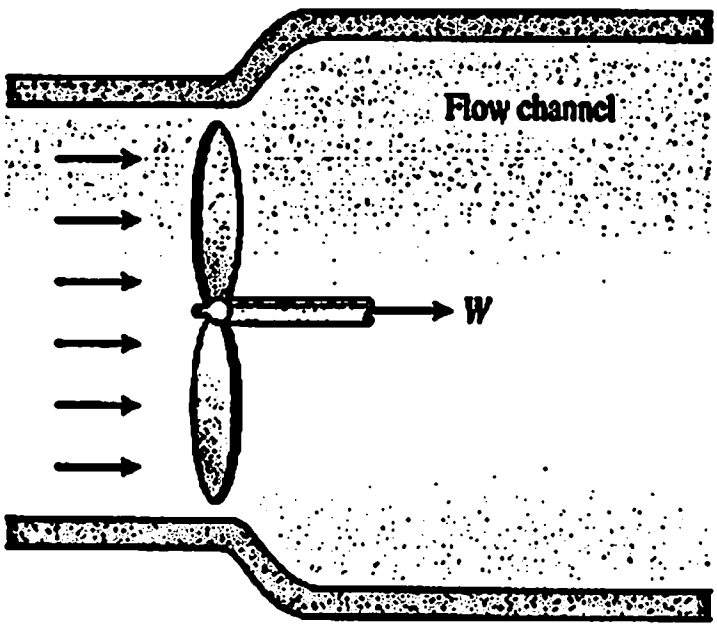


FIGURE P3–88

Step 1 of 4

Given data:

The atmospheric pressure is $P = 100$ kPa

The velocity of the wind at this pressure is $V_1 = 10$ m/s

The temperature is $T = 20°C$

The velocity behind the wind mill, $V_2 = 9$ m/s

The diameter of the rotor, $(d_1) = 7$ m

Step 2 of 4

The power required to drive the car is given by

$$\dot{W} = \dot{m}\left(\frac{V_1^2 - V_2^2}{2}\right)$$

Where $\dot{m}$ the mass flow rate of the air flow in the channel is expressed as

From the continuity equation, $\dot{m} = \rho AV$ kg/s

From the ideal gas equation $Pv = RT$

Density, $\rho = \dfrac{P}{RT} \quad \left(\because \rho = \dfrac{1}{v}\right)$

where $R = 0.287$ kJ/kg·K

$P = 100$ kPa

$T = 20°C$

$= 20 + 273$

$= 293$ K

$$\therefore \rho = \frac{100}{(0.287)(293)}$$

$\rho = 1.189$ kg/m^3

Step 3 of 4

Mass flow rate

$$\dot{m} = 1.189 \times \left(\frac{\pi}{4} \times 7^2\right) \times 10$$

$\dot{m} = 457.6$ kg/s

Therefore the power required is

$$\dot{W} = 457.6 \times \left(\frac{10^2 - 9^2}{2}\right)$$

$$= \frac{4347.2}{1000} \text{ kW}$$

$= 4.347$ kW

∴ The power required, $\boxed{\dot{W} = 4.347 \text{kW}}$

Step 4 of 4

From the continuity equation $\dot{m} = \rho A_2 V_2$

Since $\dot{m}$ remains constant in the channel

$$A_2 = \frac{\dot{m}}{\rho V_2}$$

$$\frac{\pi}{4} d_2^2 = \frac{457.6}{1.189 \times 9}$$

$d_2^2 = 54.44$

$d_2 = 7.37$m

∴ The diameter of the wind channel downstream, $\boxed{d_2 = 7.37\text{m}}$

Question: 3.089P

In a hydroelectric power plant, 65 m3/s of water flows from an elevation of 90 m to a turbine, where electric power is generated. The overall efficiency of the turbine– generator is 84 percent. Disregarding frictional losses in piping, estimate the electric power output of this plant.

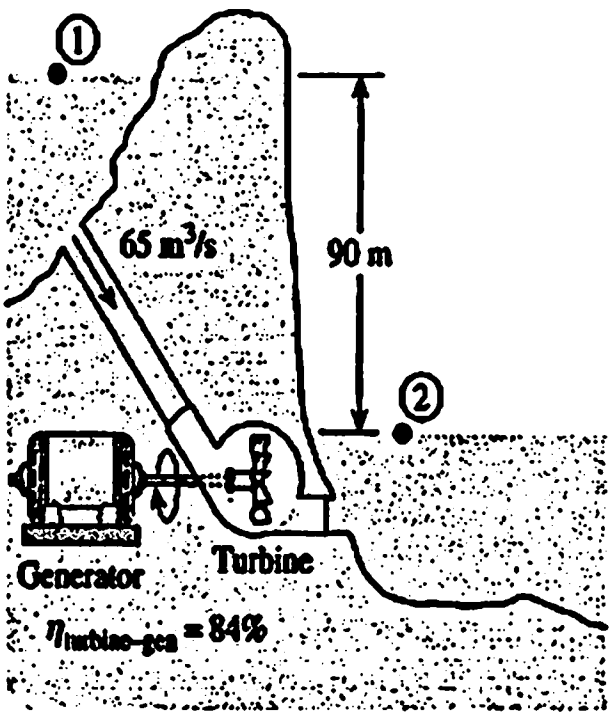


FIGURE P3–89

Step 1 of 6

Given

Flow rate of water $Q = 65\,\mathrm{m^3/s}$

Elevation $z = 90\mathrm{m}$

Overall efficiency of the turbine generator is $\eta = 84\%$

Step 2 of 6



Step 3 of 6

Density of water, $\rho = 1000\ \mathrm{kg/m^3}$

Mechanical energy is nothing but potential energy

$$e_{mech} = pe = gz$$

$$= (9.81\mathrm{m/s^2})(90\mathrm{m})\left(\frac{1\mathrm{kJ/kg}}{1000\mathrm{m^2/s^2}}\right)$$

$$e_{mech} = 0.8829\mathrm{kJ/kg}$$

Step 4 of 6

The mass flow rate is

$$\dot{m} = \rho Q$$

$$= (1000)\times(65)$$

$$\dot{m} = 65000\mathrm{kg/s}$$

Step 5 of 6

The maximum electric power generation becomes

$$\dot{W}_{max} = \dot{E}_{mech} = \dot{m} e_{mech}$$

$$= (65000\mathrm{kg/s})(0.8829\mathrm{kJ/kg})\left(\frac{1\mathrm{MW}}{1000\mathrm{kJ/s}}\right)$$

$$\dot{W}_{max} = 57.39\mathrm{MW}$$

Step 6 of 6

Actual electric power generation

$$\dot{W}_{electric} = \eta_{overall}\dot{W}_{max}$$

$$= 0.84(57.39)$$

$$\boxed{\dot{W}_{electric} = 48.2\mathrm{MW}}$$

Question: 3.090P

The demand for electric power is usually much higher during the day than it is at night, and utility companies often sell power at night at much lower prices to encourage consumers to use the available power generation capacity and to avoid building new expensive power plants that will be used only a short time during peak periods. Utilities are also willing to purchase power produced during the day from private parties at a high price.

Suppose a utility company is selling electric power for $0.03/kWh at night and is willing to pay $0.08/kWh for power produced during the day. To take advantage of this opportunity, an entrepreneur is considering building a large reservoir 40 m above the lake level, pumping water from the lake to the reservoir at night using cheap power, and letting the water flow from the reservoir back to the lake during the day, producing power as the pump-motor operates as a turbine-generator during reverse flow. Preliminary analysis shows that a water flow rate of 2 m3/s can be used in either direction. The combined pump-motor and turbine-generator efficiencies are expected to be 75 percent each. Disregarding the frictional losses in piping and assuming the system operates for 10 h each in the pump and turbine modes during a typical day, determine the potential revenue this pump-turbine system can generate per year.

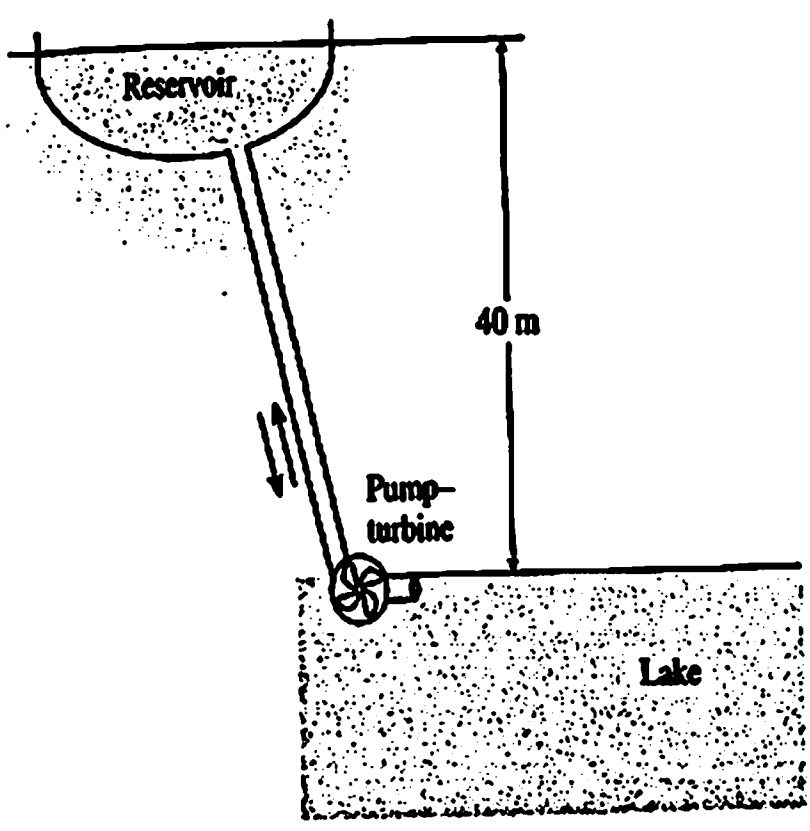


FIGURE P3–90

Step 1 of 6

Given data:

The height of the reservoir, $h = 40$ m

The volume flow rate of the water, $\dot{V} = 2\ m^3/s$

The overall turbine-generator efficiency and pump- motor efficiency is $\eta = 75\%$

Acceleration due to gravity, $(g) = 9.81\ m/s^2$

Step 2 of 6

The power generated by the operation of turbine or a pump is expressed as

$$\dot{E}_{mech} = \dot{m}\Delta pe$$

Where $\dot{m}$ mass flow rate

$$\dot{m} = \rho A V$$

$$\dot{m} = \rho \dot{V}$$

Δpe is the change in specific potential energy

$\Delta pe = gh$ J/kg

Step 3 of 6

Therefore power generated

$$\dot{E}_{mech} = \rho \dot{V} g h$$
$$= (1000)(2)(9.81)(40)\ J/s$$
$$= 784800\ J/s\ (or)\ W$$
$$= 784.8\ kW \qquad (10^3\ J/s = 1\ kW)$$

Step 4 of 6

The actual turbine work will be

$$\dot{W}_{turbine} = \eta_{gr-gn}\dot{E}_{mech}$$
$$= (0.75)(784.8)$$
$$= 588.6\ kW$$

Step 5 of 6

The minimum pump work is

$$\dot{W}_{pump} = \frac{\dot{E}_{mech}}{\eta_{pump-motor}}$$
$$= \frac{784.8}{0.75}$$
$$= 1046.4\ kW$$

Step 6 of 6

The cost to run a pump for 10hrs in a year is

Cost = (1046.4)(365×10)($0.03)/year

Cost = $114,580.8/year

The revenue developed by the turbine is

Revenue = (588.6)(365×10)($0.08)/year

Revenue = $171,871.2/year

Net profit generated by running pump-turbine is

Net profit = $171,871.2 − $114,580.8

Profit = $57290.4/year

CHAPTER 04

SOLUTIONS

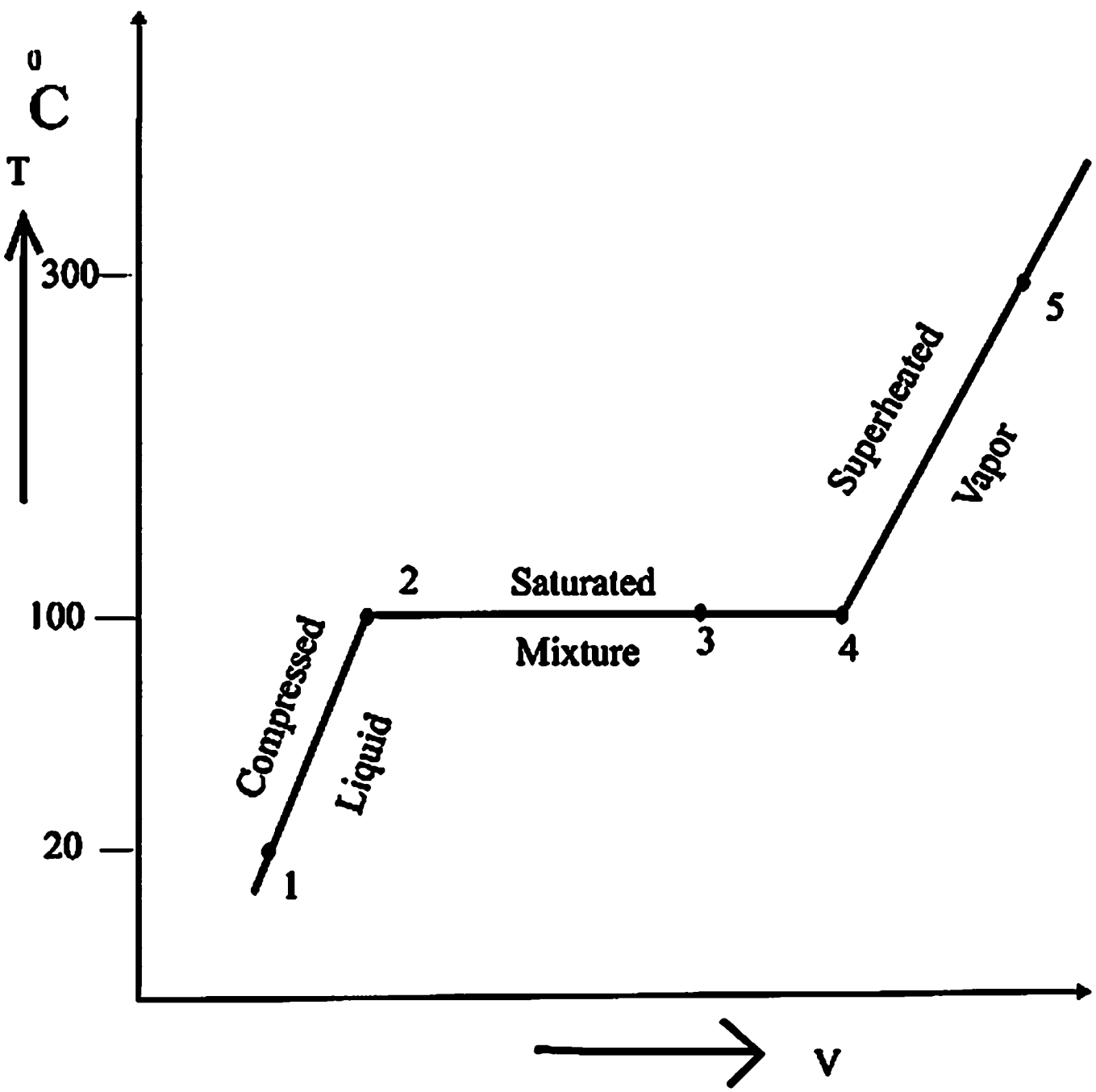
Question: 4.001P

What is the difference between saturated liquid and compressed liquid?

Step 1 of 1

A liquid that is about to vaporize is called saturated liquid. And a liquid that is not about to vaporize is called compressed liquid or subcooled liquid. A small amount of heat addition at the saturated condition will cause the liquid to vaporise i.e., phase change occurs from liquid to vapor but whereas in supercooled condition a small amount of heat addition or small increase in temperature will not cause any phase change.

The saturated liquid condition and compressed liquid can also be explained from the T-v diagram shown in figure. Point 2 indicates the saturated liquid condition and below point 2 it is supercooled condition.



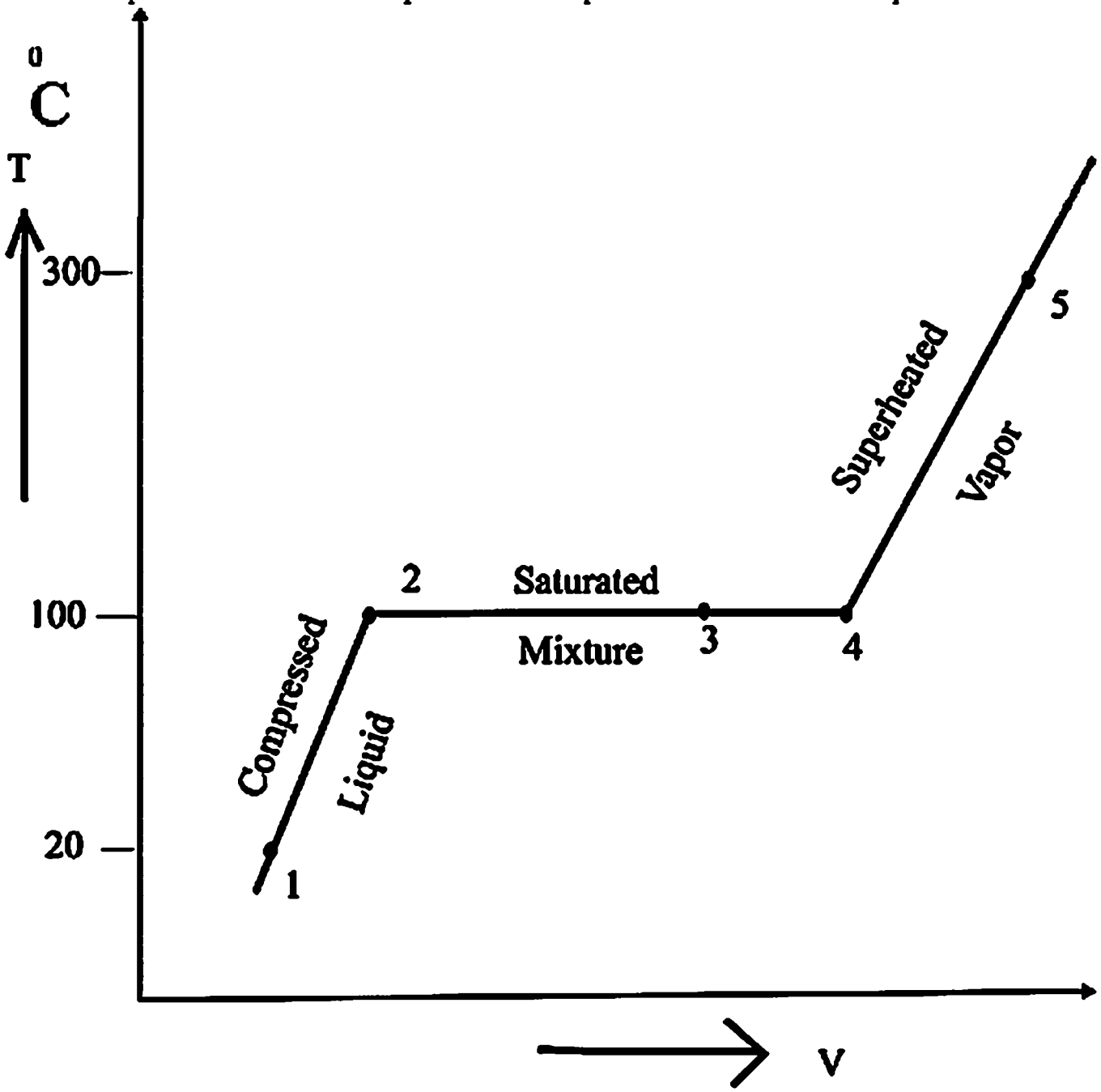
Question: 4.002P

What is the difference between saturated vapor and superheated vapor?

Step 1 of 1

A vapour that is about to condense is called saturated vapor. And a vapor that is not about to condense is called superheated vapor. A small amount of heat loss from the vapor at saturated condition will cause the vapor to condense i.e., phase changes from vapor to liquid but whereas in superheated condition a small amount of heat loss or small decrease in temperature will not cause any phase change.

The saturated vapor state and superheated vapor states can also be explained from the T-v diagram shown in figure. Point 4 in the figure indicates the saturated vapor state and after point 4 and between point 5 the vapor can be treated as superheated.



Question: 4.003P

Is there any difference between the intensive properties of saturated vapor at a given temperature and the vapor of a saturated mixture at the same temperature?

Step 1 of 3

No.

Consider the tank contains a saturated liquid vapour mixture. Now volume occupied by saturated liquid is v_f and saturated vapour is v_g :

Now find the total volume of the mixture in tank.

$$V = V_f + V_g$$

But, $V = mv$

Here, specific volume is *v*.

Substitute mv for V in above expression:

$$V = V_f + V_g$$
$$mv = m_f v_f + m_g v_g \quad \left(\because m = m_f + m_g\right)$$
$$mv = \left(m - m_g\right)v_f + m_g v_g$$

Step 2 of 3

Divide the above equation by *m* above expression:

$$v = \left(1 - \frac{m_g}{m}\right)v_f + \frac{m_g}{m}v_g$$
$$= \left(1 - x\right)v_f + xv_g \quad \left(\because x = \frac{m_g}{m}\right)$$

Here, dryness fraction is *x*.

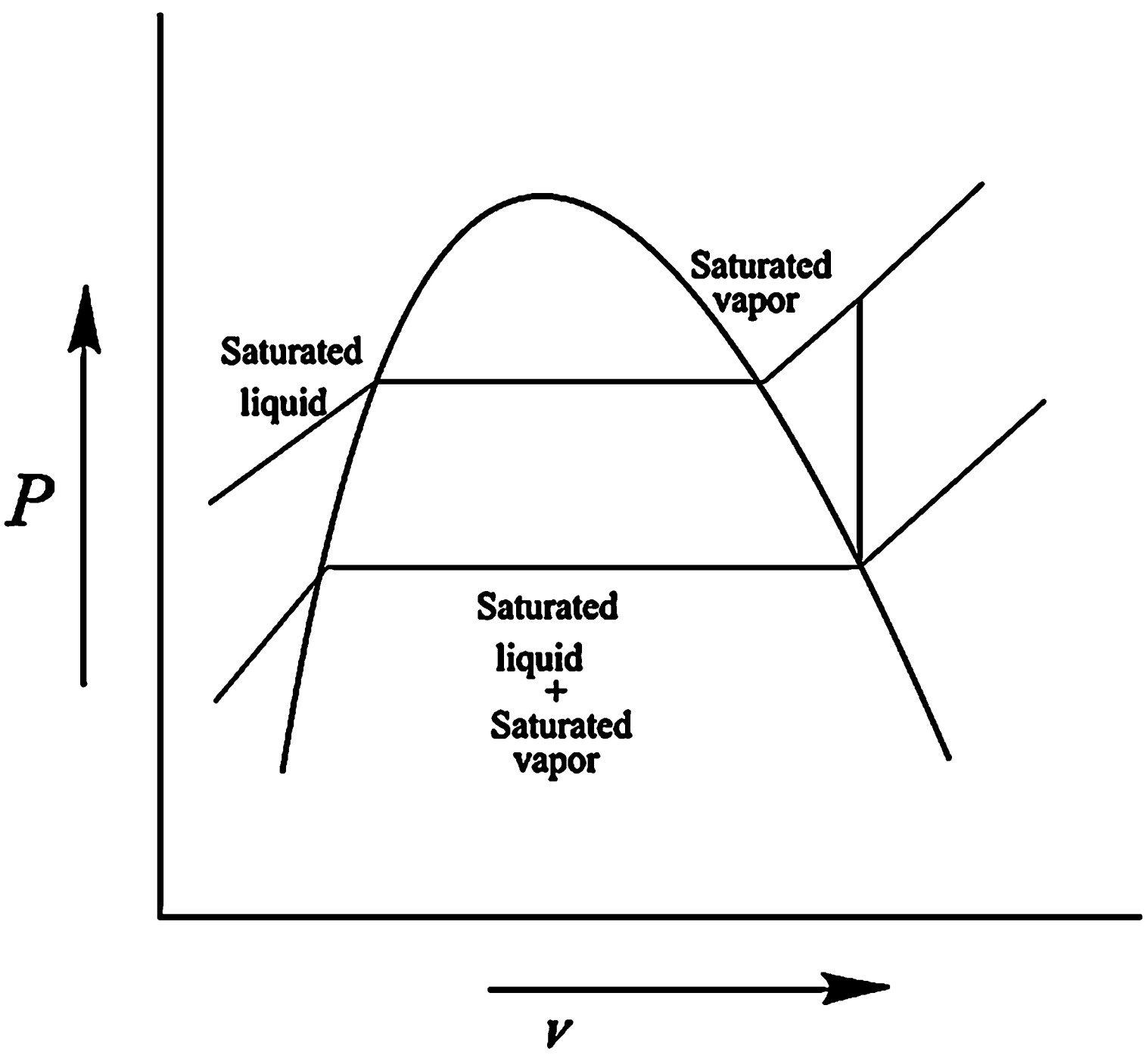
Re-write the above equation for the value of *x*:

$$x = \frac{v - v_f}{v_g}$$

From the above expression, the properties of saturated vapour mixture lie between the saturated liquid line and saturated vapour line. Therefore, there is no change in the intensive properties of saturated vapour at a given temperature and the vapour of saturated mixture at the same temperature. Because, both resemble the same horizontal axis on *P-v* and *T-s* diagram.

Step 3 of 3

Draw the *P-V* diagram:



Question: 4.004P

If the pressure of a substance is increased during a boiling process, will the temperature also increase or will it remain constant? Why?

Step 1 of 1

The temperature at which a liquid starts boiling depends on the pressure, and hence if the pressure is increased the boiling temperature also increases. For example water starts boiling at 100°C if the pressure is fixed at 1 atm. If the pressure during the boiling process is increased to 500 kPa, water would start boiling at 151.8°C.

Question: 4.005P

Why are the temperature and pressure dependent properties in the saturated mixture region?

Step 1 of 1

The pressure and temperature are dependant properties in the saturated mixture region because the phase change process i.e., either boiling or condensation occur at certain temperature only if the pressure is held constant. As the pressure changes the temperature also changes and hence the saturated mixture line also changes as shown in the T-v diagram.

Question: 4.006P

What is the difference between the critical point and the triple point?

Step 1 of 1

The critical point is the point at which the saturated liquid and saturated vapor states are identical. We can see from figure 3-16 that as the pressure increases the saturation line continues to shrink and finally it reaches to a point (i.e., at 22.06 MPa pressure for the case of water) and that point is called the critical point.

The point at which all the three phases of pure substance coexist in equilibrium is called triple point. The triple can also be explained from the P-T diagram of pure substances shown in figure 3-25.

Question: 4.007P

Is it possible to have water vapor at −10°C?

Step 1 of 1

Yes, it is possible to have water vapor at -10°C. We can see from table 3-1 that when the saturation pressure is maintained at 0.26 kPa, the saturation temperature is -10°C. Thus if the pressure is maintained at 0.26 kPa, water starts boiling at -10°C.

Question: 4.008P

A househusband is cooking beef stew for his family in a pan that is (*a*) uncovered, (*b*) covered with a light lid, and (*c*) covered with a heavy lid. For which case will the cooking time be the shortest? Why?

Step 1 of 1

The cooking time will be the shortest in the third case i.e., when the pan is covered with a heavy lid. Because heavier the lid, greater the pressure in the pan. And we can see from table 3-1 as the pressure is high the saturation temperature at which water starts boiling will be high. And higher boiling temperatures mean shorter cooking times.

Question: 4.009P

How does the boiling process at supercritical pressures differ from the boiling process at subcritical pressures?

Step 1 of 1

At supercritical pressures there is no distinct phase change process. The liquid uniformly and gradually expands into vapor. At subcritical pressures, there is always distinct surface between the phases.

Question: 4.010P

A perfectly fitting pot and its lid often stick after cooking, and it becomes very difficult to open the lid when the pot cools down. Explain why this happens and what you would do to open the lid.

Step 1 of 1

As the pot cools down, the temperature inside the pot drops and the corresponding saturation pressure also drops creating a vacuum inside the pot, and hence it will become difficult to open the lid after cooking. An easy way of removing the lid is to reheat the pot, so that the temperature rises to boiling level and the corresponding saturation pressure rises to atmospheric level and thus the lid will easily come off the pot.

Question: 4.011P

It is well known that warm air in a cooler environment rises. Now consider a warm mixture of air and gasoline on top of an open gasoline can. Do you think this gas mixture will rise in a cooler environment?

Step 1 of 1

The molar mass of air is 29 kg/kmol and the molar mass of gasoline is 114 kg/kmol which is much larger than that of air. Therefore, the gasoline vapor will settle down instead of rising even if it is at a much higher temperature than the surrounding air. Hence, the warm mixture of air and gasoline on top of an open gasoline can will most likely settle down instead of rising in a cooler environment.

Question: 4.012P

Does the amount of heat absorbed as 1 kg of saturated liquid water boils at 100°C have to be equal to the amount of heat released as 1 kg of saturated water vapor condenses at 100°C?

Step 1 of 1

Yes, the amount of heat absorbed as 1 kg of saturated liquid water boils at 100°C is equal to the amount of heat released as 1 kg of saturated water vapor condenses at 100°C. If they are not equal, energy can be created by alternately vaporizing and condensing the substance.

Question: 4.013P

Does the reference point selected for the properties of a substance have any effect on thermodynamic analysis? Why?

Step 1 of 1

No, the reference point selected for properties of substance doesn't have any effect on thermodynamic analysis. This is because, in thermodynamic analysis, we are concerned with the changes in the properties, and the reference state chosen is of no consequence in calculations as long as we use values from single consistent set of tables and charts.

Question: 4.014P

What is the physical significance of *hfg*? Can it be obtained from a knowledge of *hf* and *hg*? How?

Step 1 of 1

The quantity of h_{fg} is called the enthalpy of vaporization. It represents the amount of energy needed to vaporize a unit mass of saturated liquid at a given temperature or pressure. It can be determined from

$$h_{fg} = h_g - h_f$$

where h_f is the enthalpy of saturated liquid

h_g is the enthalpy of saturated vapor.

Question: 4.015P

Does *hfg* change with pressure? How?

Step 1 of 1

Yes, h_{fg} changes with pressure. It decreases with increase in pressure and becomes zero at the critical point.

Question: 4.016P

Is it true that it takes more energy to vaporize 1 kg of saturated liquid water at 100°C than it would at 120°C?

Step 1 of 1

Yes, it takes more energy to vaporize 1 kg of saturated liquid water at 100°C than it would at 120°C. This is because, as the temperature goes on increasing the amount of energy needed to vaporize a unit mass of saturated liquid i.e., h_{fg} goes on decreasing and it becomes zero at the critical point. Hence higher the temperatures lower the h_{fg} value.

Question: 4.017P

What is quality? Does it have any meaning in the superheated vapor region?

Step 1 of 1

Quality(x) is the ratio of mass of vapor to the total mass of the mixture. It has significance for saturated mixtures only. It has no meaning in the superheated vapor region.

$$x = \frac{m_{vapor}}{m_{total}}$$

where $m_{total} = m_{liquid} + m_{vapor}$

Question: 4.018P

Which process requires more energy: completely vaporizing 1 kg of saturated liquid water at 1 atm pressure or completely vaporizing 1 kg of saturated liquid water at 8 atm pressure?

Step 1 of 1

It requires more energy for completely vaporizing 1 kg of saturated liquid water at 1 atm pressure. Because higher the pressure the lower the h_{fg} value.

Question: 4.019P

Can quality be expressed as the ratio of the volume occupied by the vapor phase to the total volume? Explain.

Step 1 of 1

No, quality cannot be expressed as the ratio of volume occupied by the vapor phase to the total volume. Quality is the mass ratio and it is not identical to the volume ratio

$$\text{Quality, } x = \frac{m_{vapor}}{m_{total}}$$

we can see that the quality can also be rewritten as

$$x = \frac{v_{avg} - v_f}{v_{fg}}$$

where v_f is the specific volume of saturated liquid
v_g is the specific volume of saturated vapor
v_{avg} is the average specific volume.

Question: 4.020P

In the absence of compressed liquid tables, how is the specific volume of a compressed liquid at a given P and T determined?

Step 1 of 3

Treat the compressed liquid at a given state as a saturated liquid, when the compressible liquid tables are not available.

Step 2 of 3

Obtain the value of specific volume of the saturated liquid, v_f from, "Properties of saturated water-Temperature table", at the given temperature.

Step 3 of 3

Thus, the specific volume of compressible liquid at given P and T ($v_{P,T}$) is equal to the specific volume of saturated liquid at that temperature (v_f @ T).

i.e., $v_{P,T} = v_f @ T$.

Question: 4.021P

In 1775, Dr. William Cullen made ice in Scotland by evacuating the air in a water tank. Explain how that device works, and discuss how the process can be made more efficient.

Step 1 of 1

Ice can be made by evacuating the air in a water tank. This process is called vacuum freezing. The evacuation process can be seen in figure 3-15. As seen from the figure 3-15, during evacuation of air, vapor is also thrown out and thus the vapor pressure in the tank drops, causing a difference between the vapor pressures of water surface and in the tank. Whatever the heat is required for vaporization of liquid is taken from the liquid and the air in the neighbourhood, causing the temperature in the tank to drop. The process continues until water starts freezing. The process can be made more efficient by insulating the tank well so that the entire heat of vaporization comes from the water only.

Question: 4.022P

Complete this table for H20:

T,°C	P, kPa	v,m3/kg	Phase description
50		7.72	
	400		Saturated vapor
250	500		
110	350		

Step 1 of 4

Obtain the properties of water from "Table A-4: saturated water-Temperature table" in the textbook at $T = 50°C$.

$p_{sat} = 12.352\,kPa$

Question:

Complete this table for H20:

T,°C	P, kPa	v,m3/kg	Phase description
50		7.72	
	400		Saturated vapor
250	500		
110	350		

Step 1 of 4

Obtain the properties of water from "Table A-4: saturated water-Temperature table" in the textbook at $T = 50°C$.

$p_{sat} = 12.352\,kPa$

Obtain the properties of refrigerant R-134a from "Table A-6: superheated water" in the textbook at $p = 500\,kPa$ and $T = 250°C$.

$v = 0.47443\,m^3/kg$

Therefore, the specific volume at $p = 500\,kPa$ and $T = 250°C$ is $\boxed{0.47443\,m^3/kg}$.

Step 4 of 4

Obtain the properties of water from "Table A-4: saturated water-Temperature table" in the textbook at $T = 110°C$.

$p_{sat} = 143.38\,kPa$

Since the pressure $350\,kPa$ is greater than saturation pressure the water is in compressed liquid region.

Obtain the properties of water from "Table A-7: compressed liquid water" in the textbook at $p = 350\,kPa$ and $T = 110°C$.

$v = 0.0011\,m^3/kg$

Therefore, the specific volume at $p = 350\,kPa$ and $T = 110°C$ is $\boxed{0.0011\,m^3/kg}$.

Write the completed table.

T (°C)	P (kPa)	v (m^3/kg)	Phase description
50	12.352	7.72	Saturated mixture
143.61	400	0.46242	Saturated vapor
250	500	0.47443	Superheated vapor
110	350	0.0011	Compressed liquid

Obtain the properties of refrigerant R-134a from "Table A-6: superheated water" in the textbook at $p = 500\,kPa$ and $T = 250°C$.

$v = 0.47443\,m^3/kg$

Therefore, the specific volume at $p = 500\,kPa$ and $T = 250°C$ is $\boxed{0.47443\,m^3/kg}$.

Step 4 of 4

Obtain the properties of water from "Table A-4: saturated water-Temperature table" in the textbook at

Question: 4.023P

Reconsider Prob. 4–22. Using EES (or other) software, determine the missing properties of water. Repeat the solution for refrigerant-134a, refrigerant 22, and ammonia.

Step 1 of 8

The problem is solved using EES, and the solution is given below.
For steam,

```
"Given"
T[1]=50 [C]
v[1]=7.72 [m^3/kg]
P[2]=400 [kPa]
x[2]=1
T[3]=250 [C]
P[3]=500 [kPa]
T[4]=110 [C]
P[4]=350 [kPa]
"Analysis"
Fluid$='steam_iapws'
P[1]=pressure(Fluid$, T=T[1], v=v[1])
x[1]=quality(Fluid$, T=T[1], v=v[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
v[2]=volume(Fluid$, P=P[2], x=x[2])
v[3]=volume(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
v[4]=volume(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Step 2 of 8

Now from menu bar, select 'calculate' and then click on 'solve'. The array table is displayed as

	P kPa	T_i ^{0}C	v_i m^3/kg	x_i
1	12.35	50	7.72	0.6419
2	400	143.6	0.4624	1
3	500	250	0.4744	100
4	350	110	0.001051	-100

Step 3 of 8

For R134a,

```
"Given"
T[1]=50 [C]
v[1]=7.72 [m^3/kg]
P[2]=400 [kPa]
x[2]=1
T[3]=250 [C]
P[3]=500 [kPa]
T[4]=110 [C]
P[4]=350 [kPa]
"Analysis"
Fluid$='R134a'
P[1]=pressure(Fluid$, T=T[1], v=v[1])
x[1]=quality(Fluid$, T=T[1], v=v[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
v[2]=volume(Fluid$, P=P[2], x=x[2])
v[4]=volume(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Step 4 of 8

Now from menu bar, select 'calculate' and then click on 'solve'. The array table is displayed as

	P kPa	T_i ^{0}C	v_i m^3/kg	x_i
1	3.409	50	7.72	100
2	400	8.91	0.05127	1
3	500	250		
4	350	110	0.08666	100

Step 5 of 8

For R22,

```
"Given"
T[1]=50 [C]
v[1]=7.72 [m^3/kg]
P[2]=400 [kPa]
x[2]=1
T[3]=250 [C]
P[3]=500 [kPa]
T[4]=110 [C]
P[4]=350 [kPa]
"Analysis"
Fluid$='R22'
P[1]=pressure(Fluid$, T=T[1], v=v[1])
x[1]=quality(Fluid$, T=T[1], v=v[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
v[2]=volume(Fluid$, P=P[2], x=x[2])
v[3]=volume(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
v[4]=volume(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Step 6 of 8

Now from menu bar, select 'calculate' and then click on 'solve'. The array table is displayed as

	P kPa	T_i ^{0}C	v_i m^3/kg	x_i
1	4.023	50	7.72	100
2	400	-6.564	0.0586	1
3	500	250	0.09959	100
4	350	110	0.103	100

Step 7 of 8

For Ammonia,

```
"Given"
T[1]=50 [C]
v[1]=7.72 [m^3/kg]
P[2]=400 [kPa]
x[2]=1
T[3]=250 [C]
P[3]=500 [kPa]
T[4]=110 [C]
P[4]=350 [kPa]
"Analysis"
Fluid$='Ammonia'
P[1]=pressure(Fluid$, T=T[1], v=v[1])
x[1]=quality(Fluid$, T=T[1], v=v[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
v[2]=volume(Fluid$, P=P[2], x=x[2])
v[3]=volume(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
v[4]=volume(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Step 8 of 8

Now from menu bar, select 'calculate' and then click on 'solve'. The array table is displayed as

	P kPa	T_i ^{0}C	v_i m^3/kg	x_i
1	20.4	50	7.72	100
2	400	-1.891	0.3083	1
3	500	250	0.5076	100
4	350	110	0.5269	100

Question: 4.024P

Complete this table for H20:

T, °F	P,psia	u,Btu/lbm	Phase description
300		782	
	40		Saturated liquid
500	120		
400	400		

Step 1 of 4

(a)

Only temperature and internal energy are given. Obtain the internal energy from the table A-4E, "Saturated Water-Temperature table" in the textbook at $300°F$.

$u_f = 269.51$ Btu/lb

$u_g = 1099.8$ Btu/lb

Since the given value, $u = 782$ Btu/lbm is in-between the values of u_f and u_g. Therefore, the mixture will be in saturated stated and obtain the saturation pressure.

Hence,

$P_{sat@300°F} = 67.028$ psia

Therefore the phase is **saturated mixture** and the value for saturated pressure is **67.028 psia**.

Step 2 of 4

(b)

Obtain the internal energy from the table A-5E, "Saturated Water-Pressure table" in the textbook at a pressure of 40 psia .

At, $P_{sat} = 40$ psia and since dryness fraction $x = 0$,

$T_{sat} = 267.22°F$

$u_{f@40\,psia} = 236.02$ Btu/lb

Therefore, the value for saturation temperature is **267.22°F** and the value for internal energy is **236.02 Btu/lb** .

Step 3 of 4

(c)

Obtain the value of saturated temperature at pressure $P = 120$ psia energy from the table A-5E, "Saturated Water-Pressure table" in the textbook at a pressure of 120 psia .

$T_{sat} = 341.25°F$

Since the given temperature is greater than saturation temperature, the phase is in superheated state.

From the relations,

If $T < T_{sat\ @\ given\ P}$ we have compressed liquid

If $T = T_{sat\ @\ given\ P}$ we have saturated mixture

If $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

Obtain the internal energy from the table-6E, "Superheated Water" at a pressure of 120 psia and at a temperature of $500°F$ in the textbook.

$u = u_g$ @ $500°F$

$= 1174.4$ Btu/lbm

Therefore the phase is **superheated state** and the value for internal energy is **1174.4 Btu/lbm**.

Step 4 of 4

(d)

Obtain the value of saturated temperature at pressure $P = 120$ psia energy from the table A-5E, "Saturated Water-Pressure table" in the textbook at a pressure of 400 psia . $T_{sat} = 444.62°F$

Since the given temperature is less than saturation temperature, the phase is in saturated state.

From the relations,

If $T < T_{sat\ @\ given\ P}$ we have compressed liquid

If $T = T_{sat\ @\ given\ P}$ we have saturated mixture

If $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

Obtain the internal energy from the table A-4E, "Saturated Water-Temperature table" in the textbook at $400°F$.

$u = u_f$ @ $400°F$

$= 374.19$ Btu/lbm

Therefore, the phase is **compressed liquid state** and the value for internal energy is **374.19 Btu/lbm**.

Question: 4.025P

Reconsider Prob. 4–24E. Using EES (or other) software, determine the missing properties of water. Repeat the solution for refrigerant-134a, refrigerant-22, and ammonia.

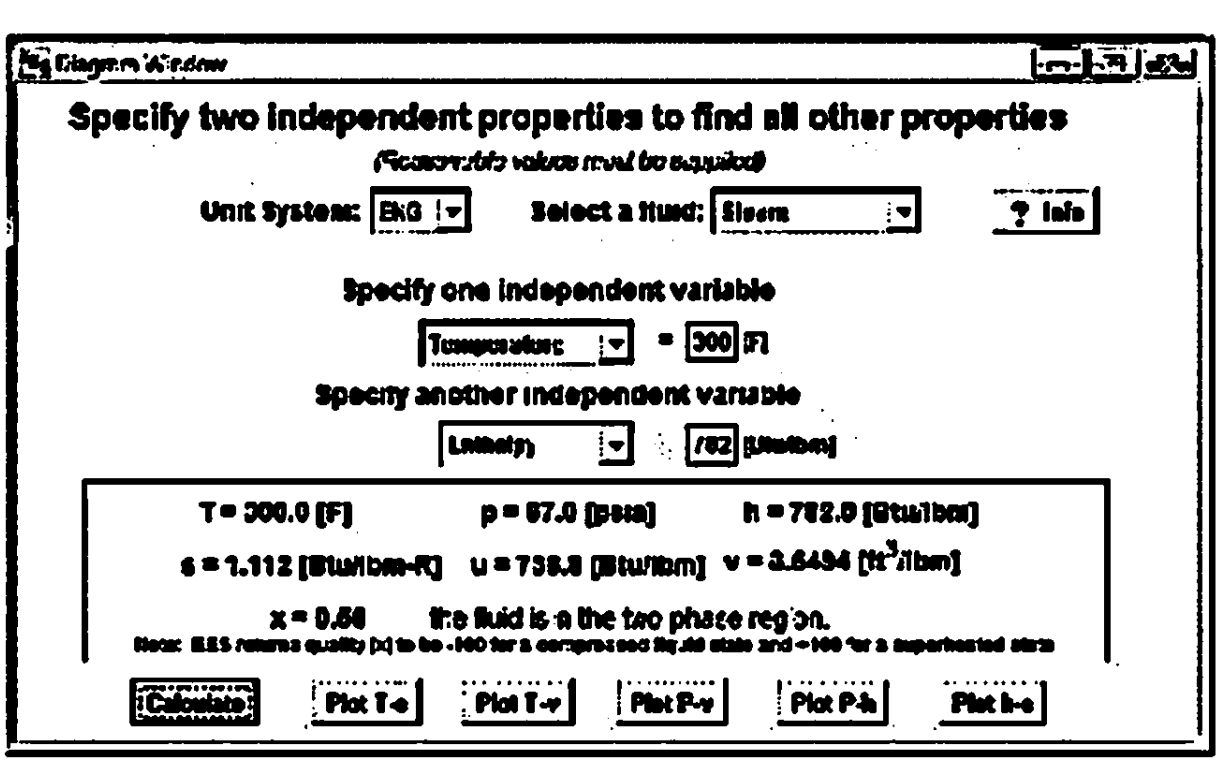
Step 1 of 8

Given table:

T	P	u Btu/lbm	phase description
300		782	
	40		saturated liquid
500	120		
400	400		

Step 2 of 8

For the properties of water
Temperature, $T = 300°F$
Internal energy, $u = 785 Btu/lbm$
By using the Engineering Equation Solver (EES) software

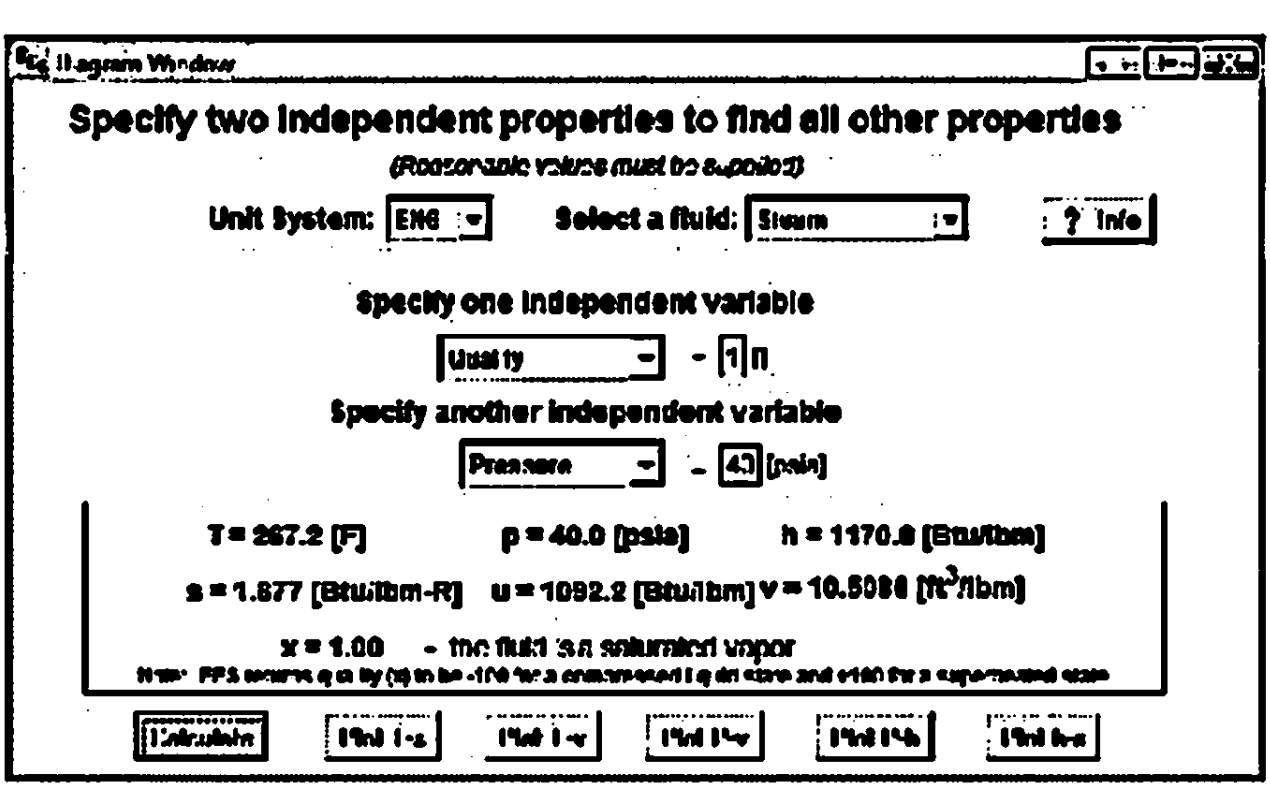


Step 3 of 8

So, the missing properties,
Pressure, $P = 67$ psia
Dryness fraction , $x = 0.56$
So, the fluid is in two phase condition.

Step 4 of 8

For the second row,
Pressure of the water, $P = 40$psia
water is in saturated vapor.
Means that $x = 1$
By using the Engineering Equation Solver (EES) software



Step 5 of 8

The missing properties of the water,
Temperature, $T = 267.2°F$
Internal energy, $u = 1092.2 Btu/lbm$

Step 6 of 8

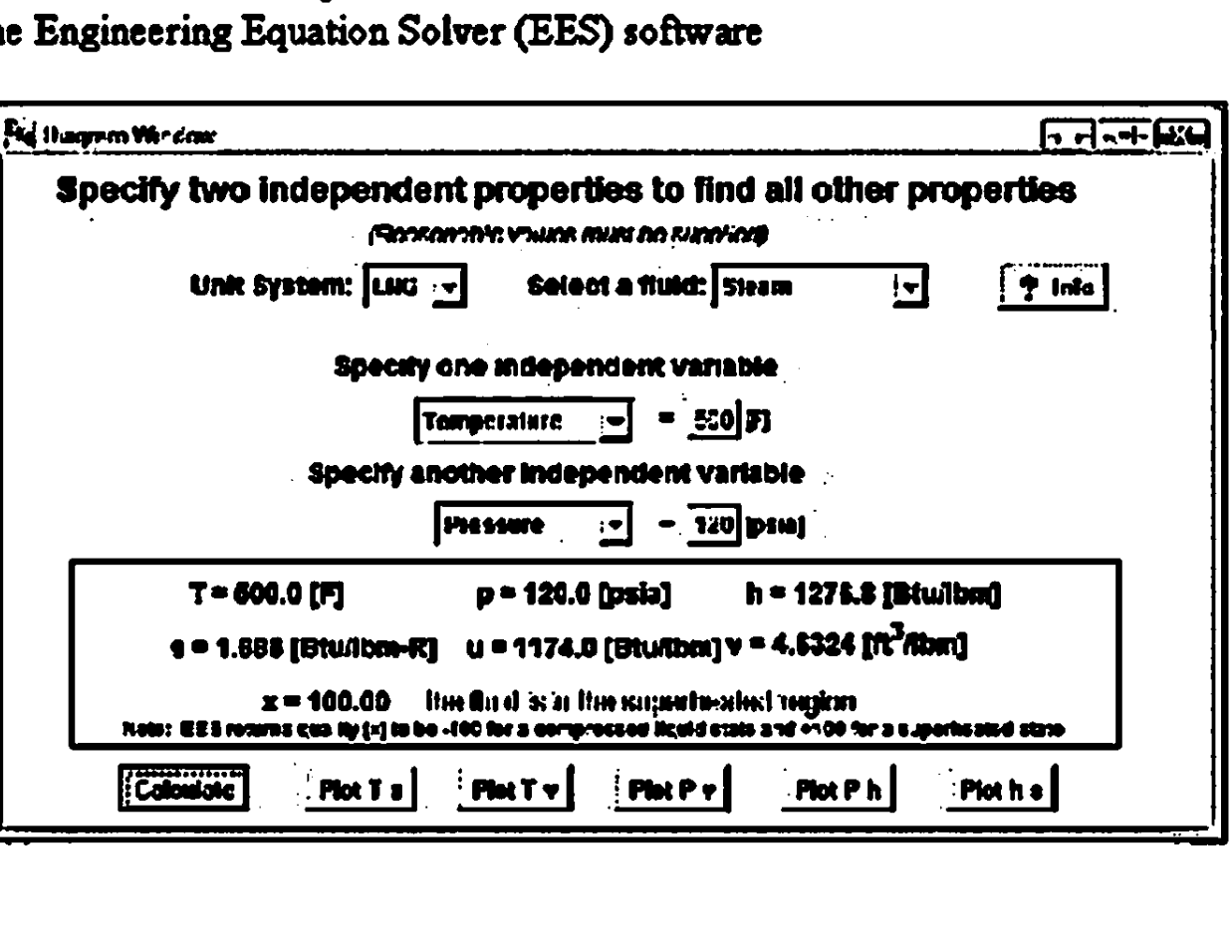
For the third row in the given table
Temperature of the steam, $T = 500°F$
Pressure of the steam, $P = 120$psia
By using the Engineering Equation Solver (EES) software

The missing properties of the water

Step 7 of 8

Internal energy, $u = 1174 Btu/lbm$
Dryness fraction, $x = 100$
So, the water is in superheated region.

For the last row in the given table
Temperature of the steam, $T = 400°F$
Pressure of the steam, $P = 400$psia
By using the Engineering Equation Solver (EES) software



Step 8 of 8

The missing properties of the water,
Internal energy, $u = 373.9 Btu/lbm$
Dryness fraction, $x = 100$
So, the water is in superheated region.

Question: 4.026P

Complete this table for H2O:

T,°C	*P*, kPa	*h*, kJ/kg	*x*	Phase description
	200		0.7	
140		1800		
	950		0.0	
80	500			
	800	3162.2		

Step 1 of 5

a) Given Pressure, P = 200 kPa

quality, x = 0.7

The quality, x = 0.7 implies that 70 percent of the mass is in the vapor state and the remaining 30 percent is in the liquid phase. Therefore, we have saturated liquid-vapor mixture at pressure of 200 kPa. From the saturated water – pressure table (Table A-4)

$T = T_{sat}$ @ 200 kPa = 120.21 °C.

and also h_f = 504.71 kJ/kg

h_{fg} = 2201.6 kJ/kg

Thus, the average enthalpy of the mixture is

$h = h_f + x h_{fg}$

= 504.71 + (0.7)*(2201.6)

= 2045.83 kJ/kg.

Step 2 of 5

b) Given T = 140 °C

h = 1800 kJ/kg

Only temperature and enthalpy are given. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the enthalpy of saturated liquid (h_f) and enthalpy of saturated vapor (h_g) from the saturated water – temperature table (Table A-4).

At T= 140 °C

h_f = 589.16 kJ/kg

h_{fg} = 2144.3 kJ/kg

h_g = 2733.5 kJ/kg

Thus, the given value h = 1800 kJ/kg is in between the values of h_f and h_g. Therefore, we have saturated liquid-vapor mixture and the pressure is the saturation pressure at given temperature (Table A-4)

Hence, $P = P_{sat}$ @ 140 °C = 361.53 kPa.

The quality is determined from

$$x = \frac{h - h_f}{h_{fg}}$$

$$= \frac{1800 - 589.16}{2144.3}$$

$$= 0.565$$

Step 3 of 5

c) Given Pressure, P = 950 kPa

quality, x = 0.0

The quality, x = 0.0 implies that 100 percent of the mass is in the liquid state. Therefore, we have saturated liquid at pressure of 950 kPa. From the saturated water – pressure table (Table A-5)

$T = T_{sat}$ @ 950 kPa = 177.66 °C.

and $h = h_f$ = 752.74 kJ/kg

Step 4 of 5

d) Given T = 80 °C

P = 500 kPa

In this case, the temperature and pressure are given, but again we do not know whether we have saturated mixture, compressed liquid or superheated vapor. To determine the region we are in, we go the saturation water – pressure table (Table A-5) and determine the saturation temperature at the given pressure. Thus, for P = 500 kPa, we have T_{sat} = 151.83 °C. We then compare the given T value to this T_{sat} value, keeping in mind that

if $T < T_{sat\ @\ given\ P}$ we have compressed liquid

if $T = T_{sat\ @\ given\ P}$ we have saturated mixture

if $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

In our case, the given T value is 80 °C, which is less than the T_{sat} value at the given pressure. Therefore, we have compressed liquid and normally we would determine the enthalpy value from the compressed liquid table (Table A-7). But in this case the given pressure is lower than the lowest pressure value in the compressed liquid table (which is 5 MPa), and therefore we are justified to treat the compressed liquid as saturated liquid at the given temperature (not pressure) and obtain the value of enthalpy from Table A-4.

$h = h_f$ @ 80 °C = 335.02 kJ/kg.

As we have compressed liquid, there is no meaning for quality in the compressed liquid region.

Step 5 of 5

e) Given P = 800 kPa

h = 3162.2 kJ/kg

Only pressure and enthalpy are given. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the enthalpy of saturated liquid (h_f) and enthalpy of saturated vapor (h_g) from the saturated water – pressure table (Table A-5).

At P= 800 kPa

h_f = 720.87 kJ/kg

h_g = 2768.3 kJ/kg

Thus, the given value h = 3162.2 kJ/kg is greater than the value of h_g. Therefore, we have superheated vapor and the temperature is determined from the superheated vapor table (Table A-6) at pressure, P=0.8 MPa and h= 3162.2 kJ/kg

Hence, T = 350 °C.

The quality has no meaning as we have superheated vapor.

Question: 4.027P

Complete this table for refrigerant-134a:

T, °C	P, kPa	v, m3/kg	Phase description
−12	320		
30		0.0065	
	550		Saturated vapor
60	600		

Step 1 of 4

Obtain the properties of refrigerant R-134a from "Table A-11: saturated refrigerant-134a-Temperature table" in the textbook at $T = -12°C$.

$p_{sat} = 185.37\,\text{kPa}$

$v_f = 0.00074\,\text{m}^3/\text{kg}$

Since the pressure $320\,\text{kPa}$ is greater than the saturation pressure the liquid is in compressed liquid state.

Therefore, the specific volume at $T = -12^0C$ and $P = 320\text{kPa}$ is $\boxed{0.0007\,\text{m}^3/\text{kg}}$

Step 2 of 4

Obtain the properties of refrigerant R-134a from "Table A-11: saturated refrigerant-134a-Temperature table" in the textbook at $T = 30°C$.

$p_{sat} = 770.64\,\text{kPa}$

$v_f = 0.0008421\,\text{m}^3/\text{kg}$

$v_g = 0.026622\,\text{m}^3/\text{kg}$

Since the specific volume $0.0065\,\text{m}^3/\text{kg}$ is in-between v_f and v_g the refrigerant is in two phase region.

Therefore, the pressure at $T = 30°C$ and $v = 0.0065\,\text{m}^3/\text{kg}$ is $\boxed{770.6\,\text{kPa}}$.

Step 3 of 4

Obtain the properties of refrigerant R-134a from "Table A-12: saturated refrigerant-134a-Pressure table" in the textbook at $p = 550\,\text{kPa}$ and saturated vapor condition.

$T_{sat} = 18.73°C$

$v_g = 0.0374\,\text{m}^3/\text{kg}$

Therefore, the specific volume and temperature at $p = 550\,\text{kPa}$ and saturated vapor condition is $\boxed{0.0374\,\text{m}^3/\text{kg}, 18.73°C}$.

Step 4 of 4

Obtain the properties of refrigerant R-134a from "Table A-12: saturated refrigerant-134a-Pressure table" in the textbook at $p = 600\,\text{kPa}$.

$T_{sat} = 21.55°C$

Since the temperature $60°C$ is greater than saturation temperature the refrigerant is in superheated region.

Obtain the properties of refrigerant R-134a from "Table A-13: superheated refrigerant-134a" in the textbook at $p = 600\,\text{kPa}$ and $T = 60°C$.

$v = 0.04138\,\text{m}^3/\text{kg}$

Therefore, the specific volume at $p = 600\,\text{kPa}$ and $T = 60°C$ is $\boxed{0.04138\,\text{m}^3/\text{kg}}$.

Write the completed table.

T, °C	P, kPa	v, m ^{3} /kg	Phase description
-12	320	0.0007	Compressed liquid
30	770.6	0.0065	Saturated mixture
18.73	550	0.0374	Saturated vapor
60	600	0.4138	Superheated vapor

Question: 4.028P

If sufficient data are provided, complete the blank cells in the following table of properties of water. In the last column describe the condition of water as compressed liquid, saturated mixture, superheated vapor, or insufficient information; and, if applicable, give the quality.

P, kPa	*T*,°C	v,m3/kg	*h*,kJ/kg	Condition description and quality (if applicable)
200			2706.3	
	130			0.650
	400		3277.0	
800	30			
450	147.90			

Step 1 of 6

Given

Pressure $P = 200\,\text{kPa}$

Enthalpy $h = 2706.3\,\text{kJ/kg}$

At $P = 200\,\text{kPa}$ and $h = 2706.3\,\text{kJ/kg}$ from saturated water tables

$h = h_g = 2706.3\,\text{kJ/kg}$ so the state is in saturated vapor condition

Then specific volume at this state is

$v = v_g = 0.88578\,\text{m}^3/\text{kg}$

$\boxed{v = 0.88578\,\text{m}^3/\text{kg}}$

Condition at this state is

$\boxed{x = 1}$

Temperature

$\boxed{T = 120.21^0\,\text{C}}$

Step 2 of 6

Given

Temperature $T = 130^0\,\text{C}$

Dryness fraction $x = 0.650$

At $T = 130^0\,\text{C}$ from saturated water tables

$P = P_{sat} = 270.28\,\text{kPa}$

$\boxed{P = 270.28\,\text{kPa}}$

Specific volume

$v = v_{f@130C} + x v_{fg@130C}$

$v = 0.001070 + 0.65 \times 0.66701$

$\boxed{v = 0.4346\,\text{m}^3/\text{kg}}$

Specific enthalpy

$h = h_{f@130C} + x h_{fg@130C}$

$h = 546.38 + 0.65 \times 2173.7$

$\boxed{h = 1959.285\,\text{kJ/kg}}$

Step 3 of 6

Given

Temperature $T = 400^0\,\text{C}$

Enthalpy $h = 3277.0\,\text{kJ/kg}$

At $T = 400^0\,\text{C}$ and $h = 3277.0\,\text{kJ/kg}$ from superheated water tables

$\boxed{P = 200\,\text{kPa}}$

Specific volume

$\boxed{v = 1.54934\,\text{m}^3/\text{kg}}$

Step 4 of 6

Given

Pressure $P = 800\,\text{kPa}$

Temperature $T = 30^0\,\text{C}$

At these properties the state is in compressed liquid region

Specific volume $\boxed{v = 0.0010\,\text{m}^3/\text{kg}}$

Specific enthalpy $\boxed{h = 126.4\,\text{kJ/kg}}$

Step 5 of 6

Given

Pressure $P = 450\,\text{kPa}$

Temperature $T = 147.90^0\,\text{C}$

At these properties the state is in saturated condition but it is not mentioned that whether the state is in saturated liquid or saturated vapor state.

Step 6 of 6

P(kPa)	T(°C)	v (m^3/kg)	h (kJ/kg)	Quality (x)	Condition
200	120.21	0.88578	2706.3	1	Saturated vapor
270.28	130	0.4346	1959.285	0.650	Saturated mixture
200	400	1.54934	3277.0		Superheated state
800	30	0.0010	126.4	-	Compressed liquid
450	147.9				Insufficient information

Question: 4.029P

Complete this table for refrigerant -134a:

T, °F	*P*, psia	*h*, Btu/lbm	*x*	Phase description
	80	78		
15			0.6	
10	70			
	180	129.46		
110		1.0	1.0	

Step 1 of 5

a) Given P = 80 psia

h = 78 Btu/lbm

Only pressure and enthalpy are given. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the enthalpy of saturated liquid (h_f) and enthalpy of saturated vapor (h_g) from the saturated refrigerant-134a-pressure table .

At P= 80 psia

h_f = 33.394 Btu/lbm

h_{fg} = 78.804 Btu/lbm

h_g = 112.20 Btu/lbm

Thus, the given value h = 78 Btu/lbm is in between the values of h_f and h_g. Therefore, we have saturated liquid-vapor mixture at pressure of 80 psia. The temperature is the saturation temperature at given pressure.

Hence, $T = T_{sat}$ @ 80 psia = 65.89 °F.

The quality is determined from

$$x = \frac{h - h_f}{h_{fg}}$$

$$= \frac{78 - 33.394}{78.804}$$

$$= 0.566$$

Step 2 of 5

b) Given Temperature, T = 15 °F

Quality, x = 0.6

The quality, x = 0.6 implies that 60 percent of the mass is in the vapor state and the remaining 40 percent is in the liquid phase. Therefore, we have saturated liquid-vapor mixture at temperature of 15 °F. From the saturated refrigerant-134a-temperature table

$P = P_{sat}$ @ 15 °F = 29.759 psia.

and also h_f = 16.889 Btu/lbm

h_{fg} = 88.377 Btu/lbm

Thus, the average enthalpy of the mixture is

$h = h_f + x h_{fg}$

= 16.889 + (0.6)*(88.377)

= 69.915 Btu/lbm

Step 3 of 5

c) Given T = 10 °F

P = 70 psia

In this case, the temperature and pressure are given, but again we do not know whether we have saturated mixture, compressed liquid or superheated vapor. To determine the region we are in, we go the saturation refrigerant-134a–pressure table (Table A-12E) and determine the saturation temperature at the given pressure. Thus, for P = 70 psia, we have T_{sat} = 58.30 °F. We then compare the given T value to this T_{sat} value, keeping in mind that

if $T < T_{sat\ @\ given\ P}$ we have compressed liquid

if $T = T_{sat\ @\ given\ P}$ we have saturated mixture

if $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

In our case, the given T value is 10 °F, which is less than the T_{sat} value at the given pressure. Therefore, we have compressed liquid and normally we would determine the enthalpy value from the compressed liquid tables of refrigerant 134a. But in this case, we do not have compressed liquid tables for the refrigerant and therefore we are justified to treat the compressed liquid as saturated liquid at the given temperature (not pressure) and obtain the value of enthalpy from Table.

$h = h_f$ @ 10 °F = 15.318 Btu/lbm

As we have compressed liquid, there is no meaning for quality in the compressed liquid region.

Step 4 of 5

d) Given P = 180 psia

h = 129.46 Btu/lbm

Only pressure and enthalpy are given. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the enthalpy of saturated liquid (h_f) and enthalpy of saturated vapor (h_g) from the saturated refrigerant-134a-pressure table (Table A-12E).

At P= 180 psia

h_f = 51.497 Btu/lbm

h_g = 117.96 Btu/lbm

Thus, the given value h = 124.96 Btu/lbm is greater than the value of h_g. Therefore, we have superheated vapor and the temperature is determined from the superheated vapor table at pressure, P=180 psia and h= 124.96 Btu/lbm

Hence, T = 160 °F.

The quality has no meaning as we have superheated vapor.

Step 5 of 5

e) Given Temperature, T = 110 °C

Quality, x = 1.0

The quality, x = 1.0 implies that 100 percent of the mass is in the vapor state. Therefore, we have saturated vapor at temperature of 110 °C. From the saturated refrigerant-134a-temperature table.

$P = P_{sat}$ @ 110 °F = 161.16 psia

and $h = h_g$ = 117.23 Btu/lbm

Question: 4.030P

A piston–cylinder device contains 0.85 kg of refriger-ant-134a at −10°C. The piston that is free to move has a mass of 12 kg and a diameter of 25 cm. The local atmospheric pressure is 88 kPa. Now, heat is transferred to refrigerant-134a until the temperature is 15°C. Determine (*a*) the final pressure, (*b*) the change in the volume of the cylinder, and (*c*) the change in the enthalpy of the refrigerant-134a.

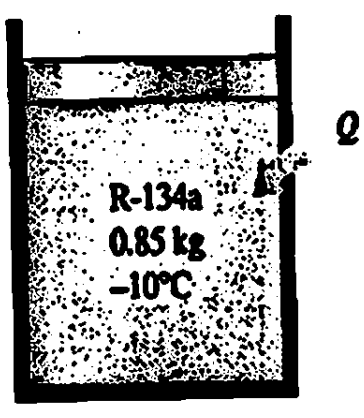


FIGURE P4–30

Step 1 of 5

(a)

Calculate the final pressure in the cylinder by using the equation.

$$P_1 = P_2$$
$$= P_{atm} + \frac{m_p g}{\pi D^2/4}$$

Here, P_{atm} is atmospheric pressure, m_p is mass of the piston, and D is diameter of the cylinder.

Substitute $88\,\text{kPa}$ for P_{atm}, $12\,\text{kg}$ for m_p, $9.81\,\text{m/s}^2$ for g and 0.25 m for D.

$$P_1 = 88\ \text{kPa} + \frac{(12\ \text{kg})\times(9.81\ \text{m/s}^2)}{\frac{\pi\times(0.25\ \text{m})^2}{4}} \times \frac{1\ \text{kN}}{10^3\ \text{kg}\cdot\text{m/s}^2}$$
$$= 88 + 2.4$$
$$= 90.4\ \text{kPa}$$

Therefore, the final pressure of the given system is $\boxed{90.4\ \text{kPa}}$.

Step 2 of 5

(b)

Obtain the properties of R-134a at pressure $P_1 = 90.4\ \text{kPa}$ and $T_1 = -10°\text{C}$ from the superheated tables of A-13 "Properties of the refrigerant R-134a" using linear interpolation.

Specific volume,

$$\frac{0.0904-0.06}{0.1-0.06} = \frac{v_1 - 0.35048}{0.20743-0.35048}$$
$$v_1 = 0.24176\ \text{m}^3/\text{kg}$$

Specific Enthalpy,

$$\frac{0.0904-0.06}{0.1-0.06} = \frac{h_1 - 248.58}{247.49-248.58}$$
$$h_1 = 247.752\ \text{kJ/kg}$$

Obtain the properties of R-134a at temperature $T_2 = 15°\text{C}$ at pressure 0.06 MPa and 0. 1 MPa from the superheated tables of A-13 "Properties of the refrigerant R-134a".

Specific volume,

$$v_{0.06\,\text{MPa}} = \frac{0.37893+0.39302}{2}$$
$$= 0.385975\ \text{m}^3/\text{kg}$$
$$v_{0.1\,\text{MPa}} = \frac{0.22506+0.23373}{2}$$
$$= 0.229395\ \text{m}^3/\text{kg}$$

Specific Enthalpy,

$$h_{0.06\,\text{MPa}} = \frac{264.66+272.94}{2}$$
$$= 268.8\ \text{kJ/kg}$$
$$h_{0.1\,\text{MPa}} = \frac{263.81+272.17}{2}$$
$$= 267.99\ \text{kJ/kg}$$

Step 3 of 5

Obtain the properties of R-134a at pressure $P_2 = 90.4\ \text{kPa}$ and $T_2 = 15°\text{C}$ from the superheated tables of "Properties of the refrigerant R-134a" using linear interpolation.

Specific volume,

$$\frac{0.0904-0.06}{0.1-0.06} = \frac{v_2 - 0.385975}{0.229395-0.385975}$$
$$v_2 = 0.26697\ \text{m}^3/\text{kg}$$

Specific Enthalpy,

$$\frac{0.0904-0.06}{0.1-0.06} = \frac{h_2 - 268.8}{267.99-268.8}$$
$$h_2 = 268.18\ \text{kJ/kg}$$

Step 4 of 5

Find the initial volume by using the expression.

$$V_1 = mv_1$$

Here, m is mass of the refrigerant.

Substitute 0.85 kg for m and $0.24176\ \text{m}^3/\text{kg}$ for v_1.

$$V_1 = 0.85\times 0.24176$$
$$= 0.205496\ \text{m}^3$$

Calculate the final volume by using the expression.

$$V_2 = mv_2$$
$$= 0.85\times 0.26697$$
$$= 0.2269245\ \text{m}^3$$

Now find the change in volume of the entire system.

$$\Delta V = V_2 - V_1$$

Substitute $0.205496\ \text{m}^3$ for V_1, and $0.2269245\ \text{m}^3$ for V_2.

$$\Delta V = (0.2269245 - 0.205496)$$
$$= 0.0214\ \text{m}^3$$

Therefore, the change in volume of the system is $\boxed{0.0214\ \text{m}^3}$.

Step 5 of 5

(c)

Find the total enthalpy change in the given system by using the expression.

$$\Delta H = m(h_2 - h_1)$$

Substitute 0.85 kg for m, $268.18\ \text{kJ/kg}$ for h_2, and $247.752\ \text{kJ/kg}$ for h_1.

$$\Delta H = 0.85(268.18 - 247.752)$$
$$= 17.36\ \text{kJ/kg}$$

Therefore, total change in enthalpy of the given system is $\boxed{17.36\,\text{kJ/kg}}$.

Question:

4.031P

R-134a. whose specific volume is 0.4618 ft3/lbm, flows through a tube at 120 psia. What is the temperature in the tube?

Step 1 of 2

Given

Specific volume of R-134a is $v = 0.4618\,\text{ft}^3/\text{lbm}$

Pressure $P = 120\text{psia}$

Step 2 of 2

At $P = 120\text{psia}$ from saturated R-134a tables

$v_g = 0.39644\,\text{ft}^3/\text{lbm}$

And given $v > v_g$ so the state is in superheated region

Therefore at $P = 120\,\text{psia}$ and $v = 0.4618\,\text{ft}^3/\text{lbm}$ from superheated R-134a tables

$$T = 120 + (140 - 120)\left(\frac{0.4618 - 0.43692}{0.46190 - 0.43692}\right)$$

$$\boxed{T = 139.919^0\text{F}}$$

Question: 4.032P

One kilogram of water fills a 150-L rigid container at an initial pressure of 2 MPa. The container is then cooled to 40°C. Determine the initial temperature and the final pressure of the water.

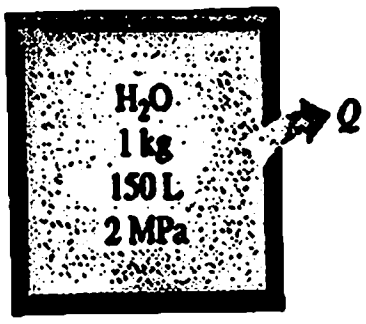


FIGURE P4–32

Step 1 of 4

Given

Mass of water $m = 1\text{kg}$

Volume of container $V = 150\text{L} = 150\times10^{-3}\text{m}^3$

Initial pressure $P_1 = 2\text{MPa}$

Final temperature $T_2 = 40^0\text{C}$

Step 2 of 4

Specific volume

$$v_1 = \frac{V}{m}$$

$$v_1 = \frac{150\times10^{-3}}{1}$$

$$v_1 = 0.15\text{m}^3/\text{kg}$$

Step 3 of 4

At $P_1 = 2\text{MPa}$ and $v_1 = 0.15\,\text{m}^3/\text{kg}$ from superheated water tables

$$T_1 = 350+(400-350)\left(\frac{0.15-0.13860}{0.15122-0.13860}\right)$$

$$\boxed{T_1 = 395.17^{\circ}\text{C}}$$

Step 4 of 4

Since it is rigid container, so the volume remains constant

$v_2 = v_1 = 0.15\text{m}^3/\text{kg}$

At $T_2 = 40^0\text{C}$ from saturated water tables

$v_f = 0.001008\text{m}^3/\text{kg}$

$v_g = 19.515\text{m}^3/\text{kg}$

And $v_f < v_2 < v_g$ so the final state is in two phase region

Therefore at $T_2 = 40^0\text{C}$ from saturated water table, we have

$P_2 = P_{sat} = 7.3851\text{kPa}$

$$\boxed{P_2 = 7.3851\text{kPa}}$$

Question: 4.033P

10-kg of R-134a fill a 1.348-m3 rigid container at an initial temperature of −40°C. The container is then heated until the pressure is 200 kPa. Determine the final temperature and the initial pressure.

Step 1 of 4

Calculate the specific volume at state 1.

$$v_1 = \frac{V}{m}$$

Here, the volume of rigid container is V and the mass of refrigerant is m.

Substitute 1.348m^3 for V and 10 kg for m.

$$v_1 = \frac{1.348}{10}$$
$$= 0.1348\ \text{m}^3/\text{kg}$$

Step 2 of 4

Obtain the properties of R-134a from saturated R-134a temperature table.

At $T_1 = -40^0\text{C}$, volumes are as follows:

The specific volume of saturated liquid, $v_f = 0.0007054\ \text{m}^3/\text{kg}$

The specific volume of saturated vapor, $v_g = 0.36081\ \text{m}^3/\text{kg}$

Saturation pressure, $P_{sat} = 51.25\text{kPa}$

Step 3 of 4

From the above calculated values, $v_f < v_1 < v_g$, so the initial state is in wet region.

Therefore initial pressure is saturation pressure at temperature $T_1 = -40^0\text{C}$.

$$P_1 = P_{sat} = 51.25\text{kPa}$$

Therefore, the initial pressure is $\boxed{51.25\ \text{kPa}}$.

Step 4 of 4

The specific volume remains constant, as it is a rigid tank.

$$v_2 = v_1$$
$$= 0.1348\ \text{m}^3/\text{kg}$$

Here, the specific volume at state 2 is v_2.

Calculate the final temperature of the refrigerant-134a as follows:

Obtain the properties of R-134a from saturated R-134a temperature table.

At $P_2 = 200\text{kPa}$ and $v_2 = v_1 = 0.1348\ \text{m}^3/\text{kg}$ from superheated R-134a tables.

$$\frac{T_2 - 60}{(70-60)} = \left(\frac{0.1348 - 0.13206}{0.13641 - 0.13206}\right)$$

$$T_2 = 60 + (70-60)\left(\frac{0.1348 - 0.13206}{0.13641 - 0.13206}\right)$$

$$T_2 = 66.3^0\text{C}$$

Therefore, the final temperature is $\boxed{66.3^\circ\text{C}}$.

Question: 4.034P

2-lbm of water at 500 psia intially fill the 1,5-ft3 left chamber of a partitioned system. The left chamber's volume is also 1.5 ft3, and it is initially evacuated. The partition is now ruptured, and heat is transferred to the water until its temperature is 300°F. Determine the final pressure of water, in psia, and the total internal energy, in Btu, at the final state.

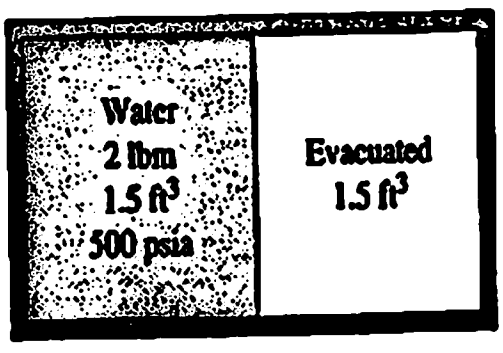


FIGURE P4–34E

Step 1 of 7

Given

Mass of water $m = 2\text{lbm}$

Initial pressure $P_1 = 500\text{psia}$

Volume of left chamber $V_{1L} = 1.5\text{ft}^3$

Volume of right chamber $V_{1R} = 1.5\text{ft}^3$

Final temperature $T_2 = 300\text{F}$

Step 2 of 7

The initial specific volume is

$$v_1 = \frac{V}{m}$$

$$= \frac{1.5}{2}$$

$$v_1 = 0.75\text{ft}^3/\text{lbm}$$

Step 3 of 7

At the final state, the water occupies two times the initial volume, then,

$$v_2 = 2v_1$$

$$= 2\times 0.75$$

$$v_2 = 1.5\,\text{ft}^3/\text{lbm}$$

Step 4 of 7

At $T_2 = 300\text{F}$ from saturated water tables

$v_f = 0.01745\text{ft}^3/\text{lbm}$

$v_g = 6.4663\text{ft}^3/\text{lbm}$

And $v_f < v_2 < v_g$ the state is in wet region

So the final pressure

$P_2 = P_{sat} = 67.028\text{psia}$

$\boxed{P_2 = 67.028\,\text{psia}}$

Step 5 of 7

$$v_2 = v_f + x_2(v_g - v_f)$$

$$1.5 = 0.01745 + x_2(6.4663 - 0.01745)$$

$$x_2 = 0.2298$$

Step 6 of 7

Specific Internal energy at final state

$$u_2 = u_{f@300F} + x_2 u_{fg@300F}$$

$$u_2 = 269.51 + 0.2298\times 830.25$$

$$u_2 = 460.301\,\text{btu/lbm}$$

Step 7 of 7

Internal energy at final state

$$U_2 = mu_2$$

$$U_2 = 2\times 460.301$$

$$\boxed{U_2 = 920.602\,\text{btu}}$$

Question: 4.035P

A 9-m3 container is filled with 300 kg of R-134a at 10°C. What is the specific enthalpy of the R-134a in the container?

Step 1 of 4

Given

Mass of R-134a $m = 300\text{kg}$

Volume of the container $V = 9\text{m}^3$

Temperature of R-134a $T = 10°\text{C}$

Step 2 of 4

Specific volume

$$v = \frac{V}{m}$$

$$v = \frac{9}{300}$$

$$v = 0.03\text{m}^3/\text{kg}$$

Step 3 of 4

At $T = 10°\text{C}$ from saturated R-134a tables

$v_f = 0.0007930\text{m}^3/\text{kg}$

$v_g = 0.049403\text{m}^3/\text{kg}$

We know

$$v = v_f + x(v_g - v_f)$$

$$0.03 = 0.0007930 + x \times (0.049403 - 0.0007930)$$

$$x = 0.601$$

Step 4 of 4

Specific enthalpy of R-134a in the container

$$h = h_{f@10C} + xh_{fg@10C}$$

$$= 65.43 + (0.601 \times 190.73)$$

$$\boxed{h = 180.06\text{ kJ/kg}}$$

Question: 4.036P

Refrigerant-134a at 200 kPa and 25°C flows through a refrigeration line. Determine its specific volume.

Step 1 of 1

At $P_1 = 200\text{kPa}$ from saturated R-134a tables

$$T_{sat} = -10.09^{\circ}\text{C}$$

And the given temperature 25°C is greater than the saturation temperature so the fluid is in superheated state.

Obtain specific value at given state using interpolation between two values.

$$v = 0.11418 + (0.11874 - 0.11418)\left(\frac{25-20}{30-20}\right)$$

$$= 0.11646\,\text{m}^3/\text{kg}$$

Therefore, the specific volume of the refrigerant at $200\,\text{kPa}, 25^{\circ}\text{C}$ is $\boxed{0.11646\,\text{m}^3/\text{kg}}$.

Question: 4.037P pg 1

A spring-loaded piston-cylinder device is initially filled with 0.2 lbm of an R-134a liquid-vapor mixture whose temperature is −30°F and whose quality is 80 percent. The spring constant in the spring force relation $F = kx$ is 37 lbf/in, and the piston diameter is 12 in. The R-134a undergoes a process that increases its volume by 40 percent. Calculate the final temperature and enthalpy of the R-134a.

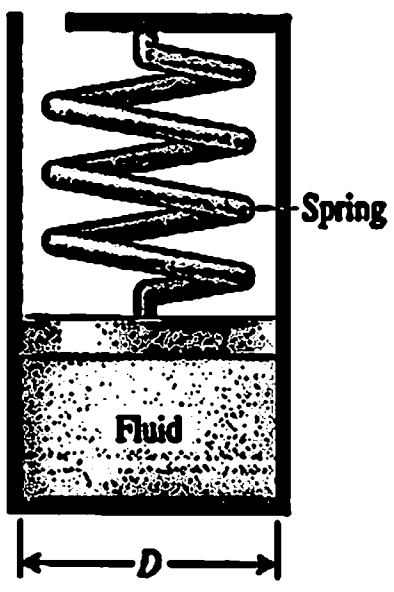
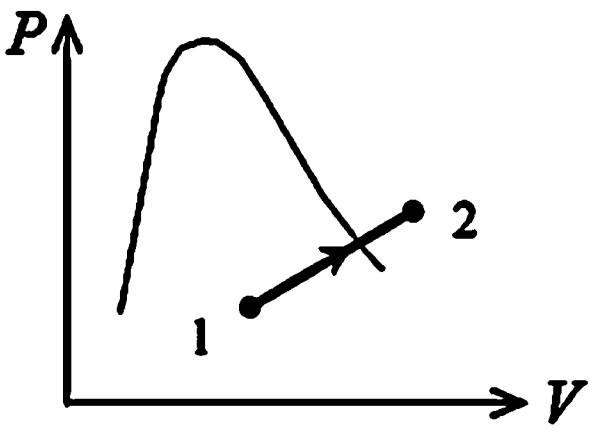


FIGURE P4–37E

Step 1 of 7

Draw the *p-v* diagram for the process.



Step 2 of 7

Calculate specific volume at state1 using the following relation.

$$v_1 = v_f + x_1\left(v_g - v_f\right) \quad \text{..... (1)}$$

Here, specific volume at state 1 is v_1 and quality of steam at state 1 is x_1 .

Obtain the values of specific volume of fluid and gas for R-134a from Table

A-16E, "Properties of saturated refrigerant-134a" in the text book at $\left(T_1 = -30^\circ\text{F}\right)$.

Specific volume of fluid, $v_f = 0.01143\ \text{ft}^3/\text{lbm}$.

Specific volume of gas, $v_g = 4.4300\ \text{ft}^3/\text{lbm}$

Substitute $0.01143\ \text{ft}^3/\text{lbm}$ for v_f , $4.4300\ \text{ft}^3/\text{lbm}$ for v_g , and 0.80 for x_1 in equation (1) .

$$\begin{aligned} v_1 &= 0.01143 + (0.80)(4.4300 - 0.01143) \\ &= 3.5463\ \text{ft}^3/\text{lbm} \end{aligned}$$

Step 3 of 7

Obtain the value of saturated pressure at temperature $\left(T_1 = -30^\circ\text{C}\right)$ from Table

A-16E, "Properties of saturated refrigerant-134a" in the text book.

$$\begin{aligned} P_1 &= P_{sat} \\ &= 9.9\ \text{psia} \end{aligned}$$

Calculate the volume of refrigerant at state1 using the following relation.

$$V_1 = mv_1$$

Substitute $0.2\ \text{lbm}$ for m and $3.5463\ \text{ft}^3/\text{lbm}$ for v_1 .

$$\begin{aligned} V_1 &= (0.2)(3.5463) \\ &= 0.7093\ \text{ft}^3 \end{aligned}$$

Since the volume is increased by 40% , calculate the volume at state 2 from the following relation.

$$V_2 = 1.4V_1$$

Substitute $0.7093\ \text{ft}^3$ for V_1 .

$$\begin{aligned} V_2 &= 1.4(0.7093) \\ &= 0.9930\ \text{ft}^3 \end{aligned}$$

Step 4 of 7

Calculate the distance that the piston moves between the initial and final conditions using the following relation.

$$\Delta x = \frac{\Delta V}{A_p}$$

Here, change in volume is ΔV , area of the piston is A_p , and displacement of the piston is Δx .

$$\begin{aligned} \Delta x &= \frac{\Delta V}{A_p} \\ &= \frac{(V_2 - V_1)}{\pi D^2/4} \end{aligned}$$

Substitute $0.9930\ \text{ft}^3$ for V_2 , $0.7093\ \text{ft}^3$ for V_1 and $1\ \text{ft}$ for D in the equation and calculate the displacement of the piston.

$$\begin{aligned} \Delta x &= \frac{(0.9930 - 0.7093)}{\frac{\pi(1)^2}{4}} \\ &= 0.3612\ \text{ft} \end{aligned}$$

$$\Delta x = \frac{(0.9930-0.7093)}{\frac{\pi(1)^2}{4}}$$
$$= 0.3612 \text{ ft}$$

Step 5 of 7

Calculate the pressure difference between the initial and final states from the following relation.

$$\Delta P = \frac{\Delta F}{A_p}$$
$$= \frac{k\Delta x}{A_p}$$
$$= \frac{k\Delta x}{\pi D^2/4}$$

Substitute 37 lbf/in for k, (0.3612×12) ft for Δx and 12 in. for D in the equation.

$$\Delta P = \frac{(37 \text{ lbf/in})(0.3612\times12 \text{ in})}{\left(\frac{\pi(12 \text{ in})^2}{4}\right)}$$
$$= 1.418 \text{ lbf/in}^2$$
$$= 1.418 \text{ psia}$$

Step 6 of 7

Calculate the final pressure using the following relation.

$$P_2 = P_1 + \Delta P$$

Substitute 9.9 psia for P_1 and 1.418 psia for ΔP.

$$P_2 = 9.9 + 1.418$$
$$= 11.318 \text{ psia}$$

Calculate the specific volume at state 2 using the following relation.

$$v_2 = \frac{V_2}{m}$$

Here, specific volume at state 2 is v_2.

Substitute 0.9930 ft^3 for V_2 and 0.2 lbm for m.

$$v_2 = \frac{0.9930}{0.2}$$
$$= 4.965 \text{ ft}^3/\text{lbm}$$

Step 7 of 7

Obtain the following properties of R-134a from Table A-13E, "Superheated refrigerant-134a" in the text book at a pressure of $(p_2 = 11.318 \text{ psia})$ and specific volume of $(v_2 = 4.965 \text{ ft}^3/\text{lbm})$.

Temperature, $T_2 = 81.5°\text{F}$.

Specific enthalpy, $h_2 = 120 \text{ Btu/lbm}$

Therefore, the final temperature is $\boxed{81.5°\text{F}}$ and final enthalpy is $\boxed{120 \text{ Btu/lbm}}$.

Question: 4.038P

One pound-mass of water fills a 2.4264.ft3 weighted piston-cylinder device at a temperature of 600°F. The piston-cylinder device is now cooled until its temperature is 200°F. Determine the final pressure of water, in psia, and the volume, in ft3.

Step 1 of 4

Find the specific volume using the following relation:

$$v_1 = \frac{V_1}{m}$$

Here, volume of the piston is V_1 and mass of the water is m.

Substitute $2.4264\,\text{ft}^3$ for V_1 and 1 lbm for m in the equation.

$$\begin{aligned} v_1 &= \frac{V_1}{m} \\ &= \frac{2.4264}{1} \\ &= 2.4264\ \text{ft}^3/\text{lbm} \end{aligned}$$

Step 2 of 4

Since the specific volume obtained is greater than gaseous and liquid state, this initial stage is under superheated conditions. So, form the superheated water table, at a temperature of $600°\text{F}$ and specific volume of $2.4264\text{ft}^3/\text{lbm}$, pressure is found to be $250\ \text{psia}$.

Step 3 of 4

Since this is a constant pressure process, $P_1 = P_2 = 250\text{psia}$

Step 4 of 4

From the saturated water table, the saturation temperature at pressure 250 psia is $400.97°\text{F}$.

Since the saturation temperature is greater than the final temperature $(T_2 = 200°\text{F})$, this stage is in compressed region.

The final volume will approximately be equal to the following value at $T_2 = 200°\text{F}$ from saturated water table.,

$$v_2 = v_f = 0.01663\ \text{ft}^3/\text{lbm}$$

Therefore, the final pressure is obtained as $\boxed{250\ \text{psia}}$.

The final volume is $\boxed{0.01663\ \text{ft}^3/\text{lbm}}$.

Question: 4.039P

Ten kilograms of R-134a fill a 1.595-m3 weighted piston-cylinder device at a temperature of −26.4°C. The container is now heated until the temperature is 100°C. Determine the final volume of the R-134a.

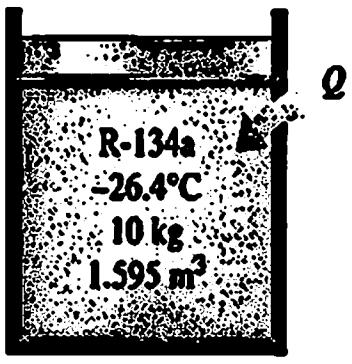


FIGURE P4–39

Step 1 of 2

The values of specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) at T = -26.4 °C lie in the range of values of specific volume of above two temperatures. We can see that the given value, v = 0.1595 m^3/kg lies in between the values of v_f and v_g. Hence we have saturated liquid-vapor mixture and the mixture pressure is the saturation pressure at that temperature which is determined by interpolation from Table-11

We have,

At T_A = -28 °C

P_{Asat} = 92.76 kPa

At T_B = -26 °C

P_{Bsat} = 101.73 kPa

By interpolation, we have

$$\frac{T-T_{Asat}}{T_{Bsat}-T_{Asat}} = \frac{P_{sat}-P_{Asat}}{P_{Bsat}-P_{Asat}}$$

$$\frac{-26.4+28}{-26+28} = \frac{P_{sat}-92.76}{101.73-92.76}$$

Therefore, $P_1 = P_{sat} = 100$ kPa

Step 2 of 2

As this is a constant-pressure process, $P_2 = P_1 = 100$ kPa.

In the final case, the temperature and pressure are given, but again we do not know whether we have saturated mixture, compressed liquid or superheated vapor. To determine the region we are in, we go the saturation refrigerant-134a-pressure table (Table A-12) and determine the saturation temperature at the given pressure.

We have,

For P =100 kPa, T_{sat} = -26.37 °C

if $T < T_{sat\ @\ given\ P}$ we have compressed liquid

if $T = T_{sat\ @\ given\ P}$ we have saturated mixture

if $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

In our case, the given T value is 100 °C, which is greater than the T_{sat} value at the given pressure. Therefore, we have superheated vapor. From the superheated refrigerant-134a table (Table A-13)

At P_2 = 0.100 MPa and temperature T_2= 100 °C

$v_A = 0.30138\ m^3/kg$

Final specific volume, $v_2 = 0.30138\ m^3/kg$ (approximately)

Therefore, final volume,. $V_2 = 10 \times 0.30138$

$$V_2 = m \times v_2$$

$$\boxed{V_2 = 3.0138\ m^3}$$

Question: 4.040P

One pound-mass of water fills a container whose volume is 2 ft3. The pressure in the container is 100 psia. Calculate the total internal energy and enthalpy in the container.

Step 1 of 4

Given

Mass of water inside the container, $m = 1$ lbm

Volume of the container, $V = 2\ ft^3$

Pressure inside the container, $P = 100$ psia

Step 2 of 4

Specific volume, $v = \frac{V}{m}$

Therefore, $v = 2\ ft^3/lbm$

Hers, the state is defined by pressure and specific volume. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) from the saturated water – pressure table (Table A-5E).

Therefore, at $P = 100$ psia

$v_f = 0.01774\ ft^3/lbm$

$v_g = 4.4327\ ft^3/lbm$

Thus, the given value $v = 2\ ft^3$ lies in between the values of v_f and v_g. Hence we have saturated liquid-vapor mixture. The quality of the mixture is given by

$$x = \frac{v - v_f}{v_g - v_f}$$

$$x = \frac{2 - 0.01774}{4.4327 - 0.01774}$$

$$x = 0.4489$$

Step 3 of 4

At $P = 100$ psia from Table A-5E

we have,

$u_f = 298.19$ Btu/lbm

$u_{fg} = 807.29$ Btu/lbm

Hence, the internal energy of this quality mixture is

$u = u_f + (x * u_{fg})$

$u = 298.19 + (0.4489 * 807.29)$

$u = 660.58$ Btu/lbm

Therefore, total internal energy in the container is

$U = m \times u = 1 \times 660.58$

$\boxed{U = 660.58\ \text{Btu.}}$

Step 4 of 4

At $P = 100$ psia from Table A-5E

we have,

$h_f = 298.51$ Btu/lbm

$h_{fg} = 888.99$ Btu/lbm

Hence, the enthalpy of this quality mixture is

$h = h_f + (x \times h_{fg})$

$h = 298.51 + (0.4489 \times 888.99)$

$h = 697.58$ Btu/lbm

Therefore, total enthalpy in the container is

$H = m \times u = 1 \times 697.58$

$\boxed{H = 697.58\ \text{Btu.}}$

4.041P

Question:

Three kilograms of water in a container have a pressure of 100 kPa and temperature of 250°C. What is the volume of this container?

Step 1 of 4

Given

Mass of water $m = 3\text{kg}$

Pressure $P = 100\text{kPa}$

Temperature $T = 250°\text{C}$

Step 2 of 4

At $P = 100\text{kPa}$ from saturated water tables

$T_{sat} = 99.61^0\text{C}$

And given temperature $T > T_{sat}$ and the given state is in superheated state.

Question:

Three kilograms of water in a container have a pressure of 100 kPa and temperature of 250°C. What is the volume of this container?

Question: 4.042P pg 1

10-kg of R-134a at 300 kPa fills a rigid container whose volume is 14 L. Determine the temperature and total enthalpy in the container. The container is now heated until the pressure is 600 kPa. Determine the temperature and total enthalpy when the heating is completed.

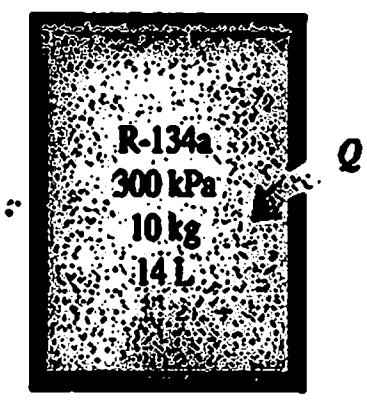


FIGURE P4–42

Question:

10-kg of R-134a at 300 kPa fills a rigid container whose volume is 14 L. Determine the temperature and total enthalpy in the container. The container is now heated until the pressure is 600 kPa. Determine the temperature and total enthalpy when the heating is completed.

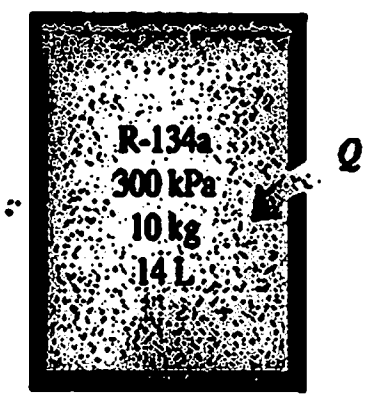


FIGURE P4–42

Step 1 of 12

At $P_B = 320$ kPa

$v_{fB} = 0.0007772\ m^3/kg$

$v_{gB} = 0.063604\ m^3/kg$

By interpolation,

$$\frac{v_f - v_{fA}}{v_{fB} - v_{fA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{v_f - 0.0007699}{0.0007772 - 0.0007699} = \frac{300 - 280}{320 - 280}$$

Step 4 of 12

Therefore, specific volume of saturated liquid at $P = 300$ kPa is $v_f = 0.0007735\ m^3/kg$

Similarly, by interpolation

$$\frac{v_g - v_{gA}}{v_{gB} - v_{gA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{v_g - 0.072352}{0.063604 - 0.072352} = \frac{300 - 280}{320 - 280}$$

Therefore, specific volume of saturated vapor at $P = 300$ kPa, $v_g = 0.067978\ m^3/kg$

Step 5 of 12

Hence, we can see that the given value, $v = 0.0014\ m^3/kg$ lies in between the values of v_f and v_g. Thus, we have saturated liquid-vapor mixture and the temperature is the saturation temperature at the given pressure which can be obtained from interpolation.

At $P_A = 280$ kPa

At $P_B = 320$ kPa

$v_{fB} = 0.0007772\ m^3/kg$

$v_{gB} = 0.063604\ m^3/kg$

By interpolation,

$$\frac{v_f - v_{fA}}{v_{fB} - v_{fA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{v_f - 0.0007699}{0.0007772 - 0.0007699} = \frac{300 - 280}{320 - 280}$$

Step 4 of 12

Therefore, specific volume of saturated liquid at $P = 300$ kPa is $v_f = 0.0007735\ m^3/kg$

Similarly, by interpolation

$$\frac{v_g - v_{gA}}{v_{gB} - v_{gA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{v_g - 0.072352}{0.063604 - 0.072352} = \frac{300 - 280}{320 - 280}$$

Therefore, specific volume of saturated vapor at $P = 300$ kPa, $v_g = 0.067978\ m^3/kg$

Step 5 of 12

Hence, we can see that the given value, $v = 0.0014\ m^3/kg$ lies in between the values of v_f and v_g. Thus, we have saturated liquid-vapor mixture and the temperature is the saturation temperature at the given pressure which can be obtained from interpolation.

At $P_A = 280$ kPa

The values of h_f and h_g at given pressure can be determined by interpolation, from the table.

At $P_A = 280$ kPa

$h_{fA} = 50.18$ kJ/kg

$h_{fgA} = 199.54$ kJ/kg

At $P_B = 320$ kPa

$h_{fB} = 55.16$ kJ/kg

$h_{fgB} = 196.71$ kJ/kg

By interpolation

$$\frac{h_f - h_{fA}}{h_{fB} - h_{fA}} = \frac{P - P_A}{P_B - P_A}$$

$h_{fgA} = 199.54$ kJ/kg

At $P_B = 320$ kPa

$h_{fB} = 55.16$ kJ/kg

$h_{fgB} = 196.71$ kJ/kg

By interpolation

$$\frac{h_f - h_{fA}}{h_{fB} - h_{fA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{h_f - 50.18}{55.16 - 50.18} = \frac{300 - 280}{320 - 280}$$

Therefore, enthalpy of saturated liquid, $h_f = 52.67$ kJ/kg

Step 8 of 12

Similarly,

$$\frac{h_{fg} - h_{fgA}}{h_{fgB} - h_{fgA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{h_{fg} - 199.54}{196.71 - 199.54} = \frac{300 - 280}{320 - 280}$$

Therefore, $h_{fg} = 198.125$ kJ/kg

Step 9 of 12

The values of h_f and h_{fg} at given pressure can be determined by interpolation, from the table.

At $P_A = 280$ kPa

$h_{fA} = 50.18$ kJ/kg

$h_{fgA} = 199.54$ kJ/kg

At $P_B = 320$ kPa

$h_{fB} = 55.16$ kJ/kg

$h_{fgB} = 196.71$ kJ/kg

By interpolation

$$\frac{h_f - h_{fA}}{h_{fB} - h_{fA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{h_f - 50.18}{55.16 - 50.18} = \frac{300 - 280}{320 - 280}$$

Therefore, enthalpy of saturated liquid, $h_f = 52.67$ kJ/kg

Step 8 of 12

Similarly,

$$\frac{h_{fg} - h_{fgA}}{h_{fgB} - h_{fgA}} = \frac{P - P_A}{P_B - P_A}$$

$$\frac{h_{fg} - 199.54}{196.71 - 199.54} = \frac{300 - 280}{320 - 280}$$

Therefore, $h_{fg} = 198.125$ kJ/kg

Step 9 of 12

Substituting, the values of h_f and h_{fg} in equation (1), we have

$$h = 52.67 + (0.009322 \times 198.125)$$

Therefore, $h = 54.517$ kJ/kg

Total enthalpy, $H = m \times h$

$H = 10 \times 54.517$

$\boxed{H = 545.17 \text{ kJ}}$

Step 10 of 12

As the container is rigid, the specific volume remains the same throughout the process.

Hence the final specific volume, $v = 0.0014$ m^3/kg

The final state is defined by pressure and specific volume. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) from the saturated refrigerant-134a-pressure table

At $P = 600$ kPa

$v_f = 0.0008199$ m^3/kg

$v_g = 0.034295$ m^3/kg

Hence, we can see that the given value, $v = 0.0014$ m^3/kg lies in between the values of v_f and v_g. Thus, we have saturated liquid-vapor mixture and the temperature is the saturation temperature at the given pressure.

Therefore, $T_2 = T_{sat\,@\,600\,kPa} = 21.55\,^{\circ}C$

Final temperature, $\boxed{T_2 = 21.55^{0}C}$

Question: 4.043P

100-kg of R-134a at 200 kPa are contained in a piston-cylinder device whose volume is 12.322 m3. The piston is now moved until the volume is one-half its original size. This is done such that the pressure of the R-134a does not change. Determine the final temperature and the change in the total internal energy of the R-134a.

Step 1 of 5

Given

Mass of R-134a in the piston cylinder device, $m = 100\ kg$

This is constant pressure process

Pressure inside the device, $P = 200\ kPa$

Initial volume, $V_1 = 12.322\ m^3$

Final volume, $V_2 = 0.5 \times V_1$

$$V_2 = 6.161 m^3$$

Question:

100-kg of R-134a at 200 kPa are contained in a piston-cylinder device whose volume is 12.322 m3. The piston is now moved until the volume is one-half its original size. This is done such that the pressure of the R-134a does not change. Determine the final temperature and the change in the total internal energy of the R-134a.

Step 1 of 5

Given

Mass of R-134a in the piston cylinder device, $m = 100\ kg$

This is constant pressure process

Pressure inside the device, $P = 200\ kPa$

Initial volume, $V_1 = 12.322\ m^3$

Final volume, $V_2 = 0.5 \times V_1$

$$V_2 = 6.161 m^3$$

Step 2 of 5

At $P = 200\ kPa$

$v_f = 0.0007533\ m^3/kg$

$v_g = 0.099867\ m^3/kg$

We can see that, the final specific volume $v_2 = 0.06161\ m^3/kg$ lies between the values of v_f and v_g at given pressure. Thus, we have saturated liquid-vapor mixture and the temperature of the mixture is the saturation temperature at that pressure, which can be obtained from saturated refrigerant-134a-pressure table.

Therefore, final temperature, $\boxed{T_2 = T_{sat}@200\ kPa = -10.09°C}$

Step 4 of 5

The quality (x) of the mixture can be obtained from

$$v_2 = v_f + x \times (v_g - v_f)$$

Therefore, $x = \dfrac{v_2 - v_f}{v_g - v_f}$

$$x = \frac{0.06161 - 0.0007533}{0.099867 - 0.0007533}$$

$$x = 0.614$$

Step 5 of 5

The internal energy of the final mixture can be obtained from

$$u_2 = u_f + (x \times u_{fg}) \text{ ---- (1)}$$

The values of u_f and u_{fg} at given pressure can be obtained from the table

At $P = 200\ kPa$

At $P = 200\ kPa$

$v_f = 0.0007533\ m^3/kg$

$v_g = 0.099867\ m^3/kg$

We can see that, the final specific volume $v_2 = 0.06161\ m^3/kg$ lies between the values of v_f and v_g at given pressure. Thus, we have saturated liquid-vapor mixture and the temperature of the mixture is the saturation temperature at that pressure, which can be obtained from saturated refrigerant-134a-pressure table.

Therefore, final temperature, $\boxed{T_2 = T_{sat}@200\ kPa = -10.09°C}$

Step 4 of 5

The quality (x) of the mixture can be obtained from

Question: 4.044P

Water initially at 300 kPa and 250°C is contained in a piston-cylinder device fitted with stops. The water is allowed to cool at constant pressure until it exists as a saturated vapor and the piston rests on the stops. Then the water continues to cool until the pressure is 100 kPa. On the *T*-v diagrams sketch, with respect to the saturation lines, the process curves passing through both the initial, intermediate, and final states of the water. Label the *T*, *P*, and v values for end states on the process curves. Find the overall change in internal energy between the initial and final states per unit mass of water.

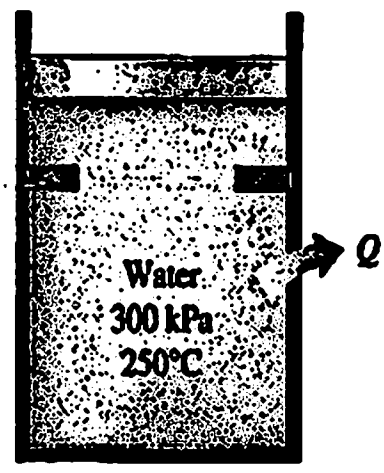


FIGURE P4–44

Question:

Water initially at 300 kPa and 250°C is contained in a piston-cylinder device fitted with stops. The water is allowed to cool at constant pressure until it exists as a saturated vapor and the piston rests on the stops. Then the water continues to cool until the pressure is 100 kPa. On the *T*-v diagrams sketch, with respect to the saturation lines, the process curves passing through both the initial, intermediate, and final states of the water. Label the *T*, *P*, and v values for end states on the process curves. Find the overall change in internal energy between the initial and final states per unit mass of water.

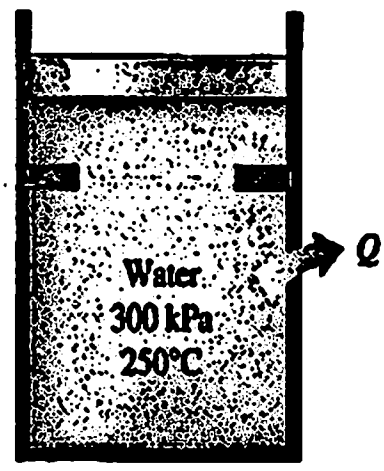


FIGURE P4–44

$0.0030 = 0.001043 + x_3 \times 1.03303$

$x_3 = 0.35719$

Then internal energy at state 3 is

$u_3 = u_f + x_3 u_{fg}$

$u_3 = 417.40 + 0.35719 \times 2088.2$

$u_3 = 1163.3\ \text{kJ/kg}$

Step 4 of 5

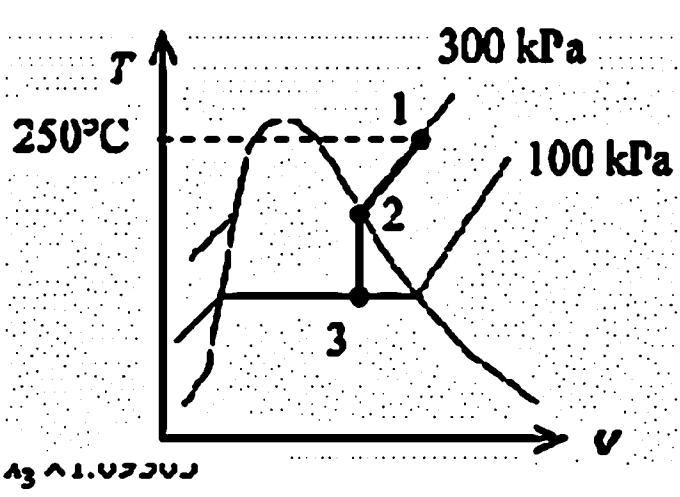
The overall change in internal energy is

$\Delta u = u_1 - u_3$

$= 2728.9 - 1163.3$

$\boxed{\Delta u = 1566\ \text{kJ/kg}}$

Step 5 of 5



$0.0030 = 0.001043 + x_3 \times 1.03303$

$x_3 = 0.35719$

Then internal energy at state 3 is

Question:

4.045P

The atmospheric pressure at a location is usually specified at standard conditions, but it changes with the weather conditions. As the weather forecasters frequently state, the atmospheric pressure drops during stormy weather and it rises during clear and sunny days. If the pressure difference between the two extreme conditions is given to be 0.2 in of mercury, determine how much the boiling temperatures of water will vary as the weather changes from one extreme to the other.

Step 1 of 2

The saturation pressure of water at 200°F is 11.538 psia
The saturation pressure of water at 212°F is 14.709 psia
One inch of mercury is equivalent to 0.491 psia
Hence the change of 0.2 in of mercury in atmospheric pressure corresponds to

$$\Delta P = (0.2)(0.491)$$
$$= 0.0982 \text{ psia}$$

At about boiling temperature the change in boiling temperature per 1 psia change in pressure is determined using data at 200°F and 212°F to be

$$\frac{\Delta T}{} = \frac{(212-200)}{}$$

Question:

The atmospheric pressure at a location is usually specified at standard conditions, but it changes with the weather conditions. As the weather forecasters frequently state, the atmospheric pressure drops during stormy weather and it rises during clear and sunny days. If the pressure difference between the two extreme conditions is given to be 0.2 in of mercury, determine how much the boiling temperatures of water will vary as the weather changes from one extreme to the other.

Step 1 of 2

$$\Delta P \quad (14.709-11.538)$$
$$= 3.783°\text{F/psia}$$

Step 2 of 2

Then the change in saturation (boiling) temperature corresponding to a change of 0.0982 psia becomes

$$\Delta T_{boiling} = (3.783)\Delta P$$
$$= (3.783)(0.0982)$$
$$= 0.03715°\text{F}$$

The calculated temperature difference value is very small; therefore the effect of variation of atmospheric pressure on the boiling pressure on the boiling temperature is negligible.

Question:

4.046P

A person cooks a meal in a 30-cm-diameter pot that is covered with a well-fitting lid and lets the food cool to the room temperature of 20°C. The total mass of the food and the pot is 8 kg. Now the person tries to open the pan by lifting the lid up. Assuming no air has leaked into the pan during cooling, determine if the lid will open or the pan will move up together with the lid.

Step 1 of 3

Given data:

Diameter of the pot, $D = 30\text{cm}$

Room temperature of the food, $T_1 = 20°\text{C}$

Total mass of the food and the pot, $m = 8\text{kg}$

Step 2 of 3

Let the reaction force on the lid is F due to the cooling.

Pressure in the pot $P_p = 1\text{atm} = 1.0325\text{bar}$

Area of the pot is

Question:

A person cooks a meal in a 30-cm-diameter pot that is covered with a well-fitting lid and lets the food cool to the room temperature of 20°C. The total mass of the food and the pot is 8 kg. Now the person tries to open the pan by lifting the lid up. Assuming no air has leaked into the pan during cooling, determine if the lid will open or the pan will move up together with the lid.

Step 1 of 3

Given data:

Diameter of the pot, $D = 30\text{cm}$

Room temperature of the food, $T_1 = 20°\text{C}$

Total mass of the food and the pot, $m = 8\text{kg}$

Step 2 of 3

Let the reaction force on the lid is F due to the cooling.

Pressure in the pot $P_p = 1\text{atm} = 1.0325\text{bar}$

Area of the pot is

$$A_p = \frac{\pi D^2}{4}$$

$$= \frac{\pi(0.3)^2}{4}$$

$$= 0.071\text{m}^2$$

By the force balance equation

Force in the pot due to the pressure is equal to the force on the pot due to the atmosphere.

So,

$$P_p A_p + F = P_{atm} A_p$$

$$F = A_p (P_{atm} - P_p)$$

$$W = mg$$

$$= 8 \times 9.81$$

$$= 78.48\text{N}$$

Therefore, weight of the food and pot, $\boxed{W = 78.48\text{N}}$

Hence,

The reaction force on the lid is far greater than the weight of the food and pot.

So, the pan will move up together with the lid when the person attempts to open the pan by lifting the lid up.

Question: 4.047P

Water is boiled at 1 atm pressure in a 25-cm-internal-diameter stainless steel pan on an electric range. If it is observed that the water level in the pan drops by 10 cm in 45 min, determine the rate of heat transfer to the pan.

Step 1 of 5

Given

Pressure at which water is boiled, $P = 1\ atm = 101.325\ kPa$

Diameter of pan, $d = 25\ cm = 0.25\ m$

The drop in water level due to evaporation, $h = 10\ cm = 0.1\ m$

Time taken, $t = 45\ min$

Step 2 of 5

Water in the pan is in the state of saturated liquid-vapor mixture at 1 atm pressure and thus at saturation temperature of 99.97 °C (from Table A-5)

The rate of heat transfer ($\dot{Q}$) to the pan for water to evaporate is given by

$$\dot{Q} = \dot{m} h_{fg} \qquad \text{----- (1)}$$

Where $\dot{m}$ = the rate of evaporation of water, kg/s

h_{fg} = enthalpy of evaporation at 1 atm pressure

= 2256.5 kJ/kg (from Table)

Step 3 of 5

Volume of water evaporated in 45 min is

V_{evap} = Area of the pan × drop in water level

$$V_{evap} = A \times h$$

$$V_{evap} = \frac{\pi}{4} d^2 \times h$$

$$V_{evap} = \frac{\pi}{4} \times 0.25^2 \times 0.1$$

$$V_{evap} = 0.00491\ m^3$$

Step 4 of 5

The mass of water evaporated can be obtained from

$$m = \frac{V_{evap}}{v_f}$$

where v_f = specific volume of saturated liquid at 1 atm pressure from table A-5, at $P = 1\ atm = 101.325\ kPa$

specific volume of saturated liquid, $v_f = 0.001043\ m^3/kg$

therefore,

$$m = \frac{0.00491}{0.001043}$$

$$= 4.7075\ kg$$

Rate of evaporation is given by

$$\dot{m} = \frac{\text{mass of water evaporated}}{\text{time taken for evaporation}}$$

$$\dot{m} = \frac{4.7075}{45 \times 60}$$

$$\dot{m} = 0.001743\ kg/s$$

Step 5 of 5

Therefore, from equation (1), rate of heat transfer is

$$\dot{Q} = 0.001743 \times 2256.5$$

$$\dot{Q} = 3.933\ kJ/s$$

$$\boxed{\dot{Q} = 3.933\ kW}$$

Question: 4.048P

Repeat Prob. 4–47 for a location at 2000-m elevation where the standard atmospheric pressure is 79.5 kPa.

Step 1 of 5

Given

Atmospheric pressure at 2000 m elevation, $P = 79.5$ kPa

Diameter of pan, $d = 25$ cm $= 0.25$ m

Drop in water level, $h = 10$ cm $= 0.1$ m

Time taken, $t = 45$ min

Step 2 of 5

The properties of water at pressure $P = 79.5$ kPa are obtained from saturated water-pressure table (Table A-5) by interpolation.

We have

At $P_1 = 75$ kPa

$v_{f1} = 0.001037\ m^3/kg$

$h_{fg1} = 2278.0$ kJ/kg

At $P_2 = 100$ kPa

$v_{f2} = 0.001043\ m^3/kg$

$h_{fg2} = 2257.5$ kJ/kg

By interpolation

$$\frac{v_f - v_{f1}}{v_{f2} - v_{f1}} = \frac{P - P_1}{P_2 - P_1}$$

$$\frac{v_f - 0.001037}{0.001043 - 0.001037} = \frac{79.5 - 75}{100 - 75}$$

Specific volume of saturated liquid at 79.5 kPa, $v_f = 0.001038\ m^3/kg$

Similarly

$$\frac{h_{fg} - h_{fg1}}{h_{fg2} - h_{fg1}} = \frac{P - P_1}{P_2 - P_1}$$

$$\frac{h_{fg} - 2278.0}{2257.5 - 2278.0} = \frac{79.5 - 75}{100 - 75}$$

Enthalpy of evaporation at 79.5 kPa, $h_{fg} = 2274.31$ kJ/kg

Step 3 of 5

Volume of water evaporated in 45 min is

V_{evap} = Area of the pan × drop in water level

$V_{evap} = A \times h$

$$V_{evap} = \frac{\pi}{4} d^2 \times h$$

$$V_{evap} = \frac{\pi}{4} \times 0.25^2 \times 0.1$$

$$V_{evap} = 0.00491\ m^3$$

Step 4 of 5

The mass of water evaporated can be obtained from

$$m = \frac{V_{evap}}{v_f}$$

$$m = \frac{0.00491}{0.001038}$$

$$= 4.73025\ kg$$

Rate of evaporation is given by

$$\dot{m} = \frac{\text{mass of water evaporated}}{\text{time taken for evaporation}}$$

$$\dot{m} = \frac{4.73025}{45 \times 60}$$

$$\dot{m} = 0.001752\ kg/s$$

Step 5 of 5

The rate of heat transfer ($\dot{Q}$) to the pan for water to evaporate is given by

$$\dot{Q} = \dot{m} h_{fg}$$

$$\dot{Q} = 0.001752 \times 2274.31$$

$$\dot{Q} = 3.984\ kJ/s$$

$$\boxed{\dot{Q} = 3.984\ kW}$$

Question:

4.049P

Saturated steam coming off the turbine of a steam power plant at 40°C condenses on the outside of a 3-cm-outer-diameter, 35-m-long tube at a rate of 130 kg/h. Determine the rate of heat transfer from the steam to the cooling water flowing through the pipe.

Step 1 of 1

Take the values of enthalpy of evaporation at $T = 40°C$ from saturated water tables.

$$h_{fg} = 2406\ \text{kJ/kg}$$

Calculate the rate of heat transfer from the steam to the cooling water flowing through the pipe using the below given formula.

$$Q = \dot{m} \times h_{fg}$$

Here, the mass flow rate is $\dot{m}$.

Substitute 130 kg/hr in the equation.

$$\begin{aligned} Q &= 130 \times 2406 \\ &= 312,780\ \text{kJ/h} \\ &= 312,780 \times \frac{1}{3600}\ \text{kJ/s} \\ Q &= 86.88\ \text{kW} \end{aligned}$$

Thus, the rate of heat transfer Q is $\boxed{86.88\ \text{kW}}$.

Question: 4.050P

Water in a 5-cm-deep pan is observed to boil at 98°C. At what temperature will the water in a 40-cm-deep pan boil? Assume both pans are full of water.

Step 1 of 4

Given

Boiling temperature of water in 5 cm deep pan, $T_1 = 98\,^\circ C$

Step 2 of 4

In the first case, as the water starts boiling at 98 °C, the pressure at the bottom of the 5 cm deep pan will be the saturation pressure at that temperature (i.e., 98 °C) which can be obtained from the saturated water-temperature table.

From table we have

At $T = 95\,^\circ C$

$P_{sat} = 84.609$ kPa

At $T = 100\,^\circ C$

$P_{sat} = 101.42$ kPa

The saturation pressure at the given temperature 98 °C can be obtained by interpolating between the above two values as

$$\frac{P_{sat} - 84.609}{101.42 - 84.609} = \frac{98 - 95}{100 - 95}$$

Therefore, $P_1 = P_{sat\,@\,98^\circ C} = 94.695$ kPa

Step 3 of 4

Now, consider 40 cm deep pan.

The pressure at the bottom of 40 cm deep pan = pressure at the bottom of 5cm deep pan + pressure due to the extra height (35 cm) of water

Therefore, $P_2 = P_1 + \rho gh$

Where ρ = density of water = $1000\ kg/m^3$

h = difference in heights = 35 cm = 0.35 m

Thus,

$$P_2 = 94.695 + \frac{1000 \times 9.81 \times 0.35}{1000}\ kPa$$

$$P_2 = 98.13\ kPa$$

Step 4 of 4

Thus the boiling temperature (T_2) of water in 40 cm deep pan will be the saturation temperature at the pressure 98.13 kPa which can be obtained from the saturation water-pressure table.

From table A-5, we have

At $P = 75$ kPa

$T_{sat} = 91.76\,^\circ C$

At $P = 100$ kPa

$T_{sat} = 99.61\,^\circ C$

The saturation temperature at the given pressure 104.85 kPa can be obtained by interpolating between the above two values as

$$\frac{T_{sat} - 91.76}{99.61 - 91.76} = \frac{98.13 - 75}{100 - 75}$$

Therefore, $T_2 = T_{sat\,@}\ 104.85$ kPa

$$\boxed{T_2 = 99.022^0 C}$$

Question: 4.051P

Water is being heated in a vertical piston-cylinder device. The piston has a mass of 20 kg and a cross-sectional area of 100 cm2. If the local atmospheric pressure is 100 kPa, determine the temperature at which the water starts boiling.

Step 1 of 3

Given

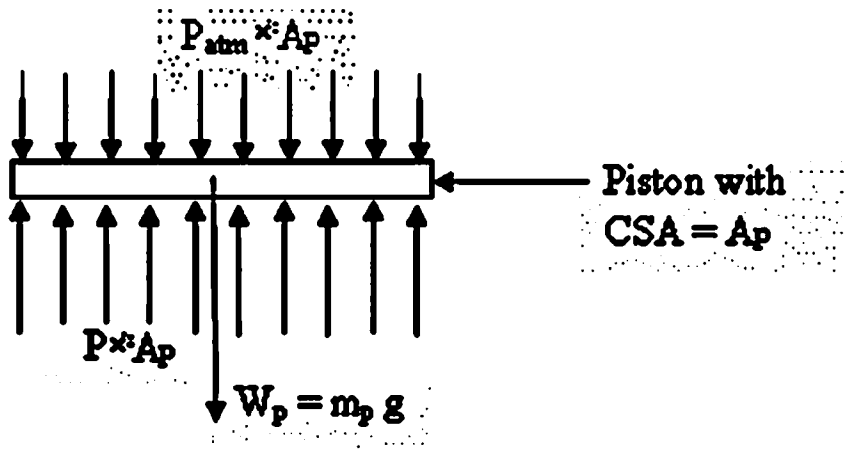
Mass of piston, $m_p = 20\ kg$

Cross-sectional area of piston, $A_p = 100\ cm^2 = 0.01\ m^2$

Local atmospheric pressure, $P_{atm} = 100\ kPa$

Step 2 of 3

The pressure inside the cylinder can be obtained from a force balance on the piston as shown in the figure below



Therefore, from the force balance, we have

$$P \times A_p = (P_{atm} \times A_p) + W_p$$

where P = pressure inside the cylinder, Pa

W_P = weight of the piston = $m_p g$, N

Therefore,

$$P = P_{atm} + \frac{m_p g}{A_p}$$

$$P = \left(100 \times 10^3\right) + \frac{20 \times 9.81}{0.01}$$

$$P = 119620\ Pa$$

Pressure inside the cylinder, P = 119.62 kPa

Step 3 of 3

The temperature at which water starts boiling will be the saturation temperature at this pressure, P = 119.62 kPa which can be obtained from saturated water-pressure table (Table A-5).

From Table A-5, we have

At $P_1 = 101.325\ kPa$

$T_{1sat} = 99.97\ °C$

At $P_2 = 125\ kPa$

$T_{2sat} = 105.97\ °C$

The value of saturation temperature at the pressure 119.62 kPa can be obtained by interpolating between the above specified pressures.

By interpolation

$$\frac{T_{sat} - T_{1sat}}{T_{2sat} - T_{1sat}} = \frac{P - P_1}{P_2 - P_1}$$

$$\frac{T_{sat} - 99.97}{105.97 - 99.97} = \frac{119.62 - 101.325}{125 - 101.325}$$

Therefore, $T_{sat} = 104.60\ °C$

Hence the boiling temperature, $\boxed{T = 104.60^0 C}$.

Question: 4.052P

A rigid tank with a volume of 1.8 m3 contains 15 kg of saturated liquid–vapor mixture of water at 90°C. Now the water is slowly heated. Determine the temperature at which the liquid in the tank is completely vaporized. Also, show the process on a *T*-v diagram with respect to saturation lines.

Step 1 of 4

Given

Volume of the container $V = 1.8\,\text{m}^3$

Mass of saturated liquid vapor mixture $m = 15\,\text{kg}$

Temperature $T = 90^0\,\text{C}$

Step 2 of 4

This is a constant volume process and the specific volume is determined to be

$$v = \frac{V}{m}$$

$$= \frac{1.8}{15}$$

$$v = 0.12\,\text{m}^3/\text{kg}$$

Step 3 of 4

When the liquid is completely vaporized, the tank will contain saturated vapor only

Thus,

$v_2 = v_g = 0.12\,\text{m}^3/\text{kg}$

The temperature at this point is the temperature that corresponds to this

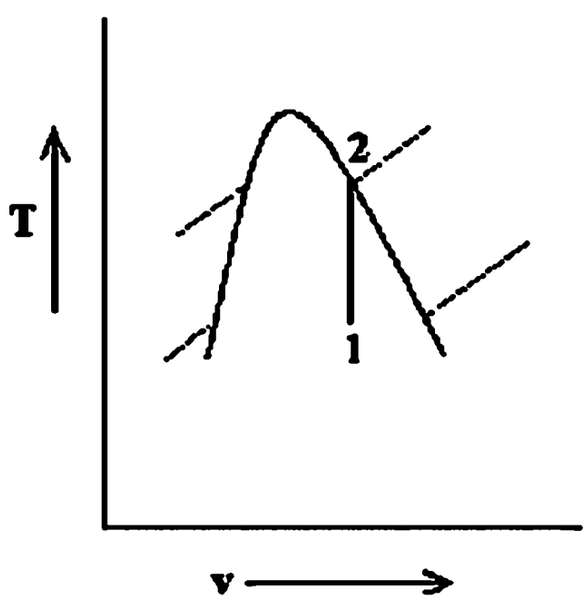
$v_2 = v_g = 0.12\,\text{m}^3/\text{kg}$ value from saturated water tables using method of interpolation

$$T = 200 + (205 - 200)\left(\frac{0.12 - 0.12721}{0.11508 - 0.12721}\right)$$

$$\boxed{T = 202.97^\circ\text{C}}$$

Step 4 of 4

T-v diagram will be



Question: 4.053P

A piston-cylinder device contains 0.005 m3 of liquid water and 0.9 m3 of water vapor in equilibrium at 600 kPa. Heat is transferred at constant pressure until the temperature reaches 200°C.

(*a*) What is the initial temperature of the water?

(*b*) Determine the total mass of the water.

(*c*) Calculate the final volume.

(*d*) Show the process on a *P-V* diagram with respect to saturation lines.

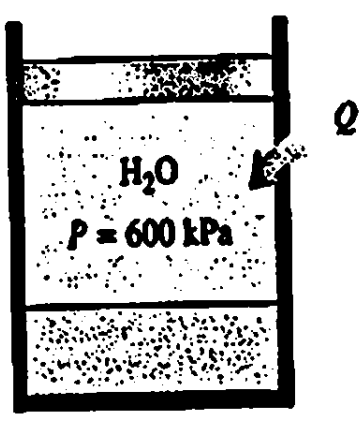


FIGURE P4–53

Step 1 of 6

(a)

Initially two phases co-exist in equilibrium, thus we have a saturated liquid vapor mixture. Then the temperature in the tank must be saturation temperature at the specified pressure.

Obtain the temperature from EES software, at 0 for quality (x) , and pressure $P = 600\ \text{kPa}$ as follows:

The saturation temperature, $T_1 = 158.8°\text{C}$

Therefore, the initial temperature of the water is, $\boxed{158.8°\text{C}}$.

Step 2 of 6

(b)

Obtain the specific volume of liquid water from EES software at pressure $600\ \text{kPa}$, and quality, 0 as follows:

The specific volume of the liquid, $v_f = 0.0011\ \text{m}^3/\text{kg}$

Calculate the mass of the liquid water.

$$m_f = \frac{V_f}{v_f}$$

Here, V_f is the volume of the saturated liquid.

Substitute $0.005\ \text{m}^3$ for V_f , and $0.0011\ \text{m}^3/\text{kg}$ for v_f .

$$m_f = \frac{0.005}{0.0011}$$
$$m_f = 4.545\ \text{kg}$$

Step 3 of 6

Obtain the specific volume of water vapor from EES software at pressure $600\ \text{kPa}$, and quality, 1 as follows:

The specific volume of the liquid, $v_g = 0.3158\ \text{m}^3/\text{kg}$

Calculate the mass of the water vapor.

$$m_g = \frac{V_g}{v_g}$$

Here, V_g is the volume of the saturated vapor.

Substitute $0.9\ \text{m}^3$ for V_g , and $0.3158\ \text{m}^3/\text{kg}$ for v_g .

$$m_g = \frac{0.9}{0.3158}$$
$$m_g = 2.849\text{kg}$$

Calculate the total mass.

$$m_t = m_f + m_g$$

Substitute $4.545\ \text{kg}$ for m_f and 2.849kg for m_g .

$$m_t = 4.545 + 2.849$$
$$= 7.394\ \text{kg}$$

Therefore, the total mass is, $\boxed{7.394\ \text{kg}}$.

Step 4 of 6

(c)

Heat is transferred at constant pressure. Hence, the pressure at state 2 is same as pressure at state 1.

$$P_2 = P_1$$
$$= 600\ \text{kPa}$$

Obtain the specific volume of the final mixture at pressure, $600\ \text{kPa}$, and temperature $200°\text{C}$ from the EES software.

The specific volume is, $v_2 = 0.3520\ \text{m}^3/\text{kg}$.

Calculate the final volume.

$$V_2 = m_t \times v_2$$

Substitute $7.394\ \text{kg}$ for m_t , and $0.3520\ \text{m}^3/\text{kg}$ for v_2 .

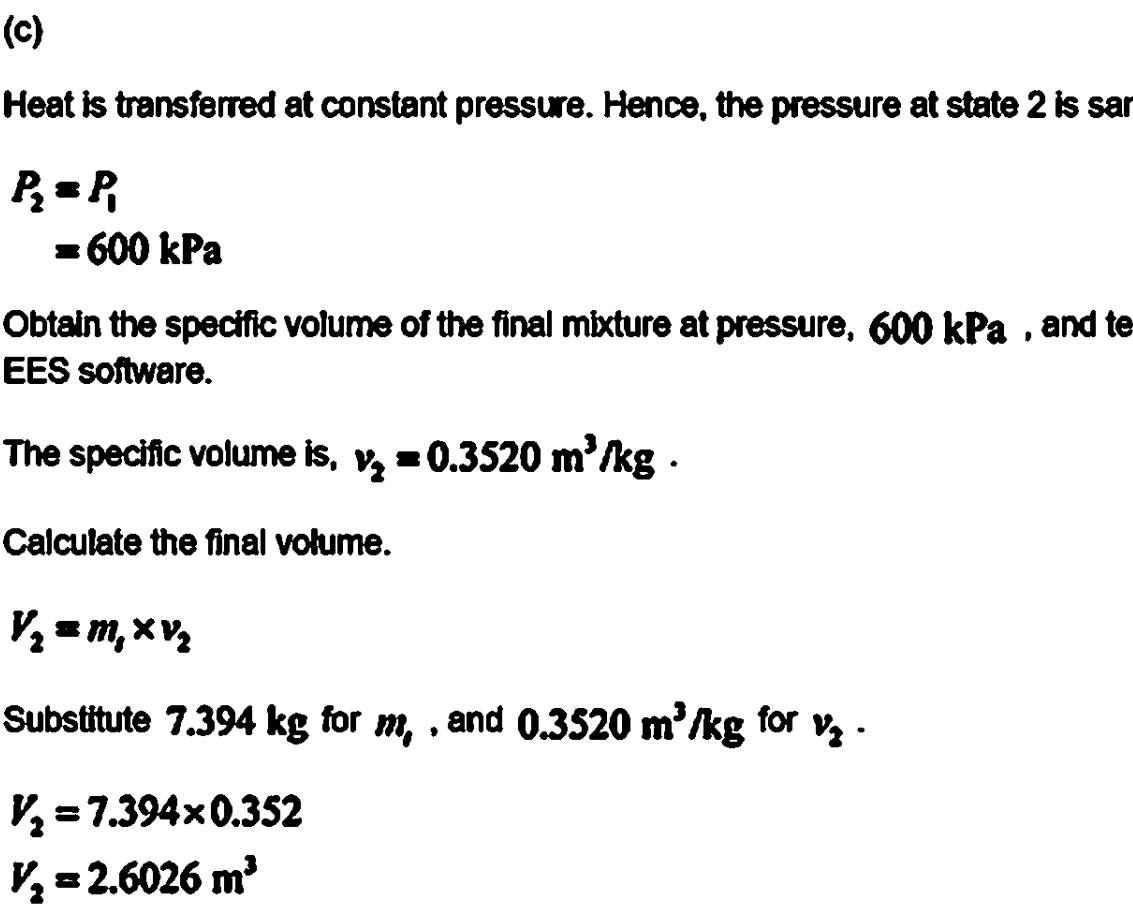
$$V_2 = 7.394 \times 0.352$$
$$V_2 = 2.6026\ \text{m}^3$$

Therefore, the final volume is, $\boxed{2.6026\ \text{m}^3}$.

Step 5 of 6

(d)

Draw the schematic representation of process on *P-v* diagram.



Step 6 of 6

Therefore, the required process is represented on saturation lines.

Question: 4.054P

Reconsider Prob. 4–53. Using EES (or other) software, investigate the effect of pressure on the total mass of water in the tank. Let the pressure vary from 0.1 MPa to 1 MPa. Plot the total mass of water against pressure, and discuss the results. Also, show the process in Prob. 4–53 on a *P*-v diagram using the property plot feature of EES.

Step 1 of 1

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Analysis: The problem is solved using EES, and the solution is given below.

```
P[1]=600[kPa]

P[2]=P[1]

T[2]=200[C]

V_f1 = 0.005[m^3]

V_g1 = 0.9[m^3]

spvsat_f1=volume (Steam_iapws, P = P[1], x=0) "sat. liq. Specific volume, m^3/kg"

spvsat_g1=volume (Steam_iapws, P = P[1], x=1) "sat. vap. Specific volume, m^3/kg"

m_f1=V_f1/spvsat_f1 "sat. liq. mass, kg"

m_g1=V_g1/spvsat_g1 "sat. vap. mass, kg"

m_tot=m_f1+m_g1

v[1]=V_f1+V_g1

spvol[1]=V[1]/m_tot "specific volume1, m^3"

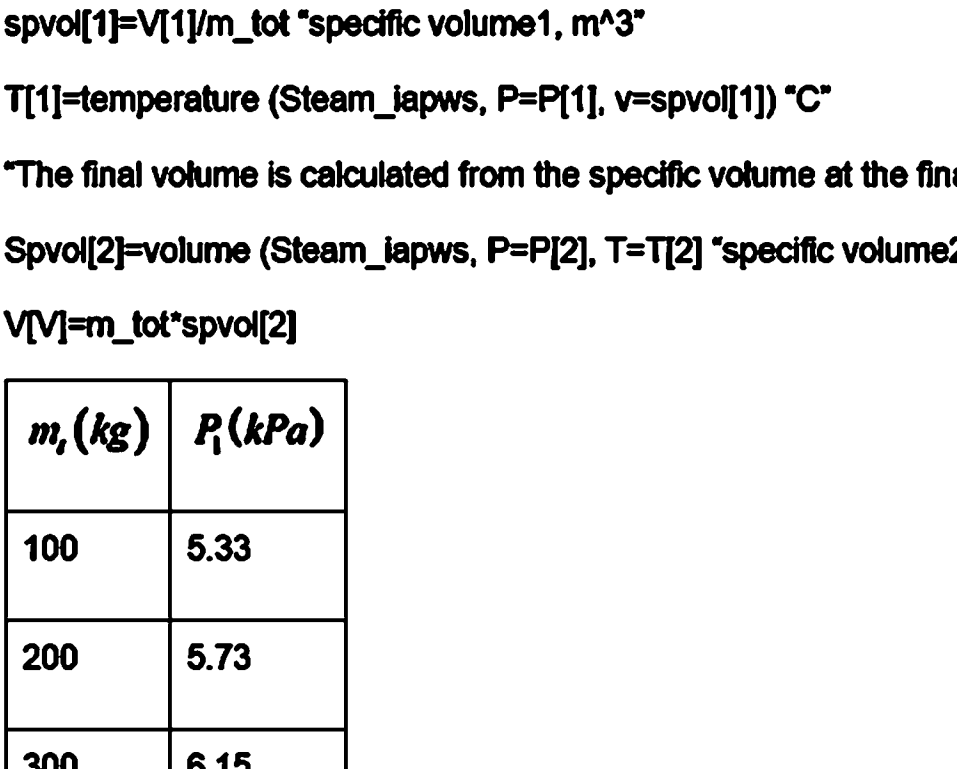
T[1]=temperature (Steam_iapws, P=P[1], v=spvol[1]) "C"

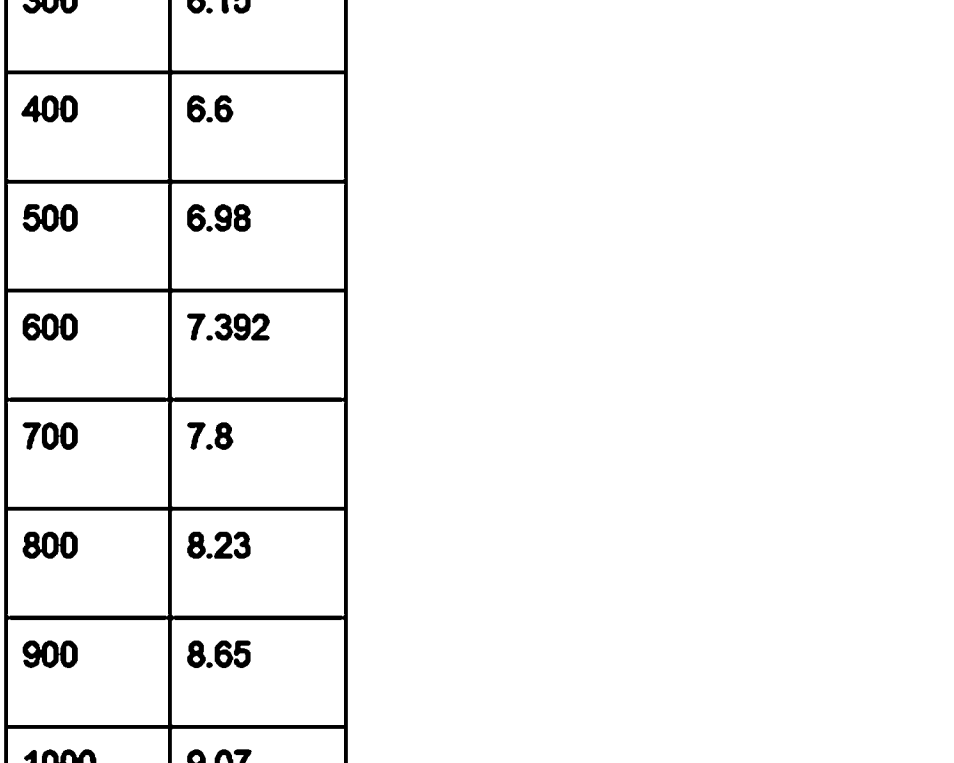
"The final volume is calculated from the specific volume at the final T and P"

Spvol[2]=volume (Steam_iapws, P=P[2], T=T[2] "specific volume2, m^3/kg"

V[V]=m_tot*spvol[2]
```

$m_t\,(kg)$	$P_1\,(kPa)$
100	5.33
200	5.73
300	6.15
400	6.6
500	6.98
600	7.392
700	7.8
800	8.23
900	8.65
1000	9.07





Question: 4.055P

Superheated water vapor at 180 psia and 500°F is allowed to cool at constant volume until the temperature drops to 250°F. At the final state, determine (*a*) the pressure, (*b*) the quality, and (*c*) the enthalpy. Also, show the process on a *T-V* diagram with respect to saturation lines.

Step 1 of 5

Given

Pressure of superheated water vapor, $P_1 = 180$ psia

Initial temperature, $T_1 = 500\ ^\circ F$

Final temperature, $T_2 = 250\ ^\circ F$

Constant volume process

Step 2 of 5

a) As the initial state is superheated vapor, the specific volume at the initial pressure and temperature can be obtained from superheated water table

At $P_1 = 180$ psia and $T_1 = 500\ ^\circ F$

Specific volume, $v = 3.0433\ ft^3/lbm$

As the volume is constant throughout the process,

Final specific volume $= v = 3.0433\ ft^3/lbm$

Therefore, the final state is defined by temperature and specific volume.). We do not know which table to use to determine the other properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) from the saturated water-temperature table .

At $T_2 = 250\ ^\circ C$

$v_f = 0.01700\ ft^3/lbm$

$v_g = 13.816\ ft^3/lbm$

In our case, the given specific volume $v = 3.0433\ ft^3/lbm$ lies in between the values of v_f and v_g at the given temperature. Thus, we have saturated liquid-vapor mixture. Thus the final pressure of the state will be the saturation pressure at the final temperature which can be obtained from saturated water-temperature table.

Therefore, at $T = 250\ ^\circ F$

Final pressure, $P_2 = P_{sat}$ @ 250°F

$$\boxed{P_2 = 29.844 \text{ psia}}$$

Step 3 of 5

b) The quality (x) of mixture can be determined from

$$v = v_f + (x \times (v_g - v_f))$$

Therefore, quality $x = \dfrac{v - v_f}{v_g - v_f}$

$$x = \frac{3.0433 - 0.01700}{13.816 - 0.01700}$$

$$\boxed{x = 0.219}$$

Step 4 of 5

c) From saturated water-temperature table (Table A-4E)

at $T_2 = 250\ ^\circ F$

$h_f = 218.63$ Btu/lbm

$h_{fg} = 945.41$ Btu/lbm

Therefore, enthalpy of mixture is

$h = h_f + (x \times h_{fg})$

$h = 218.63 + (0.219 \times 945.41)$

$$\boxed{h = 425.67 \text{ Btu/lbm}}$$

Step 5 of 5

d) The total process on a T-v diagram can be shown as

Question: 4.056P

Reconsider Prob. 4–55E. Using EES (or other) software, investigate the effect of initial pressure on the quality of water at the final state. Let the pressure vary from 100 psi to 300 psi. Plot the quality against initial pressure, and discuss the results. Also, show the process in Prob. 4–55E on a *T-V* diagram using the property plot feature of EES.

Step 1 of 5

Initial pressure of the super heated water vapor, $P_1 = 180\,\text{psia}$

Initial temperature of the super heated water vapor, $T_1 = 500°\text{F}$

Final temperature, $T_2 = 250°\text{F}$

Step 2 of 5

By the EES (Engineering Equation software),
The effect of initial pressure on the quality of the water at the final state,

Equations Window

```
T[1]=500 [F]
"P[1]=180 [psia]"
T[2]=250 [F]
v[ 1]=volume(steam_iapws,T=T[1],P=P[1])
v[2]=v[1]
P[2]=pressure(steam_iapws,T=T[2],v=v[2])
h[2]=enthalpy(steam_iapws,T=T[2],v=v[2])
x[2]=quality(steam_iapws,T=T[2],v=v[2])
```

Step 3 of 5

Result:

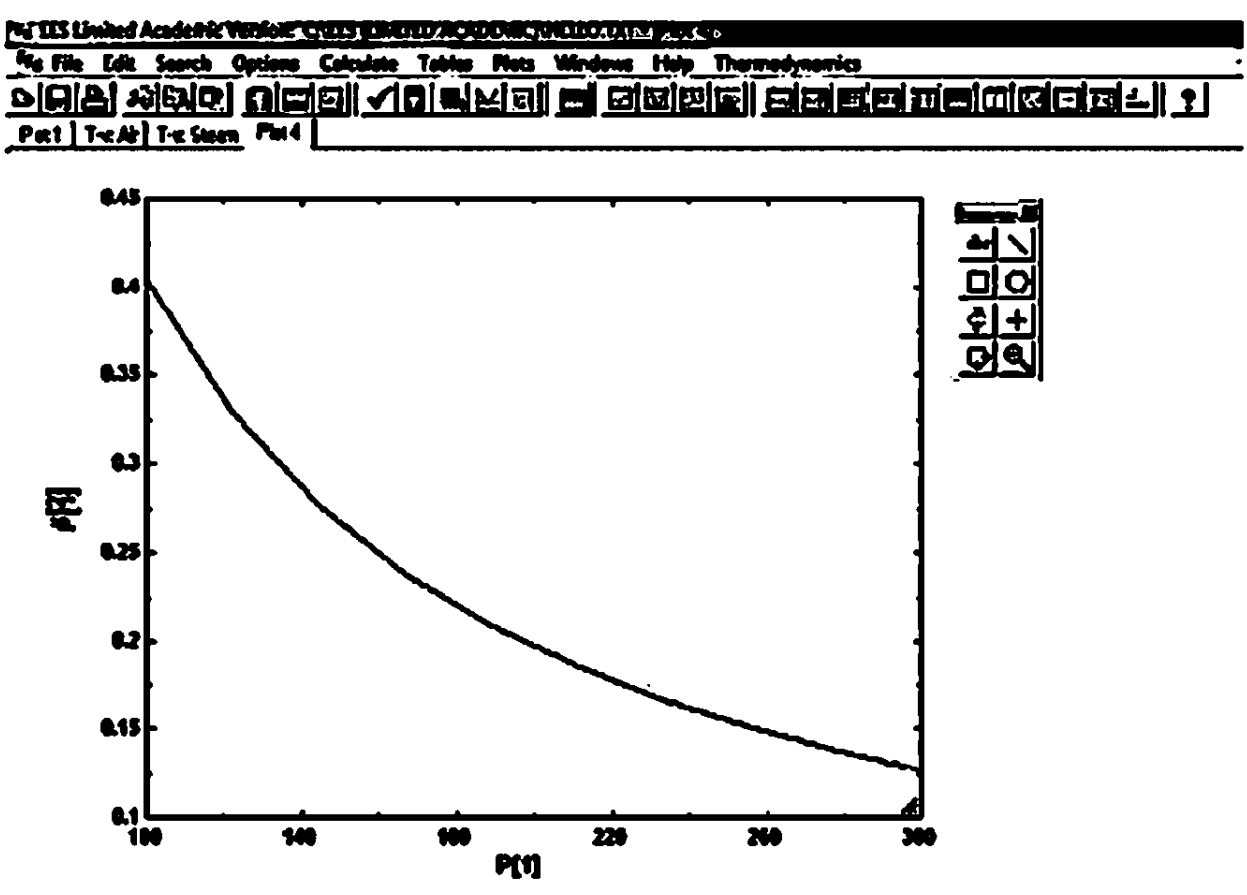
Parametric Table

Table 1

1..10	P_1	x_2
Run 1	100	0.4037
Run 2	122.2	0.3283
Run 3	144.4	0.2761
Run 4	166.7	0.2378
Run 5	188.9	0.2084
Run 6	211.1	0.1853
Run 7	233.3	0.1665
Run 8	255.6	0.151
Run 9	277.8	0.1379
Run 10	300	0.1268

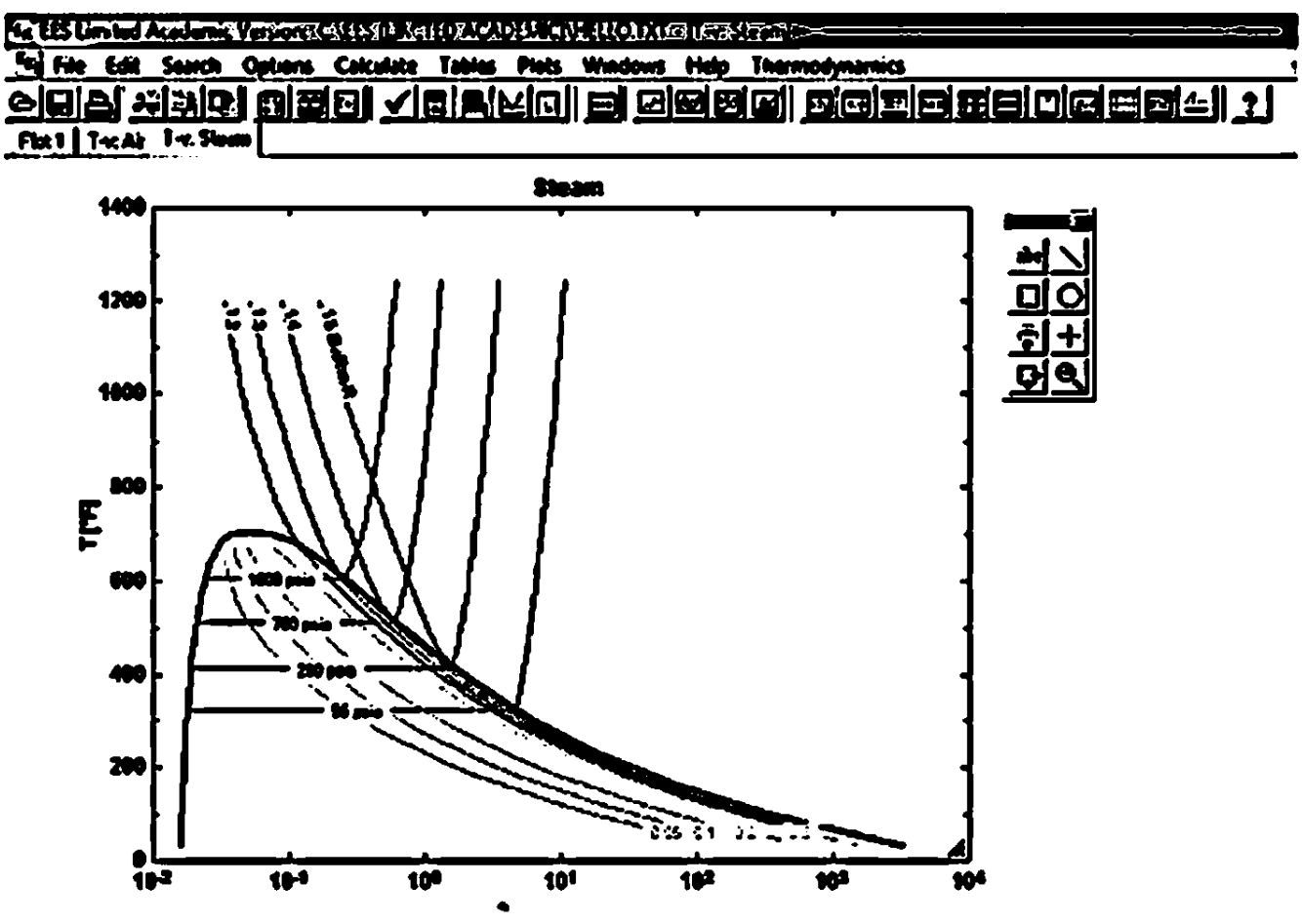
Step 4 of 5

Graph for the quality Vs initial pressure:



T-V diagram:

Step 5 of 5



Question: 4.057P

Determine the specific volume, internal energy, and enthalpy of compressed liquid water at 80°C and 20 MPa using the saturated liquid approximation. Compare these values to the ones obtained from the compressed liquid tables.

Step 1 of 6

Given
Pressure $P = 20\text{MPa}$
Temperature $T = 80^0\text{C}$

Step 2 of 6

Assuming saturated liquid approximation
T = 80°C from saturated water tables
$v \cong v_f = 0.001029\text{m}^3/\text{kg}$
$u \cong u_f = 334.97\text{kJ/kg}$
$h \cong h_f = 335.02\text{kJ/kg}$

Step 3 of 6

Using Compressed liquid approximation
At $P = 20\text{MPa}$ and $T = 80°\text{C}$
$v = 0.0010199\text{m}^3/\text{kg}$
$u = 330.50\text{kJ/kg}$
$h = 350.90\text{kJ/kg}$

Step 4 of 6

The percentage error involved in specific volume is

$$v_{y.error} = \frac{(0.001029 - 0.0010199)}{0.001029} \times 100$$

$$\boxed{v_{y.error} = 0.884\%}$$

Step 5 of 6

The percentage error involved in enthalpy is

$$h_{y.error} = \frac{(350.90 - 335.02)}{350.90} \times 100$$

$$\boxed{h_{y.error} = 4.525\%}$$

Step 6 of 6

The percentage error involved in internal energy is

$$u_{y.error} = \frac{(334.97 - 330.50)}{334.97} \times 100$$

$$\boxed{u_{y.error} = 1.334\%}$$

Question: 4.058P

Reconsider Prob. 4–57. Using EES (or other) software, determine the indicated properties of compressed liquid, and compare them to those obtained using the saturated liquid approximation.

Step 1 of 2

Analysis: The problem is solved using EES, and the solution is given below.

```
"Given"
T=80 [C]
P=20000 [kPa]
"Analysis"
Fluid$='steam_iapws'
"Saturated liquid assumption"
v_f=volume(Fluid$, T=T, x=0)
u_f=intenergy(Fluid$, T=T, x=0)
h_f=enthalpy(Fluid$, T=T, x=0)
"Compressed liquid treatment"
v=volume(Fluid$, T=T, P=P)
u=intenergy(Fluid$, T=T, P=P)
h=enthalpy(Fluid$, T=T, P=P)
"Percentage Errors"
error_v=100*(v_f-v)/v
error_u=100*(u_f-u)/u
error_h=100*(h-h_f)/h
```

Step 2 of 2

Solution to the above EES program is as below

$h = 350.90[\text{kJ/kg}]$

$h_{app} = 335.02$

$P = 20000[\text{kPa}]$

$T = 80[\text{C}]$

$u = 330.50[\text{kJ/kg}]$

$u_{app} = 334.97[\text{kJ/kg}]$

$v = 0.0010199[\text{m}^3/\text{kg}]$

$v_{app} = 0.001029[\text{m}^3/\text{kg}]$

Question: 4.059P

A piston–cylinder device contains 0.8 kg of steam at 300°C and 1 MPa. Steam is cooled at constant pressure until one-half of the mass condenses.

(*a*) Show the process on a *T-V* diagram.

(*b*) Find the final temperature.

(*c*) Determine the volume change.

Step 1 of 6

Given

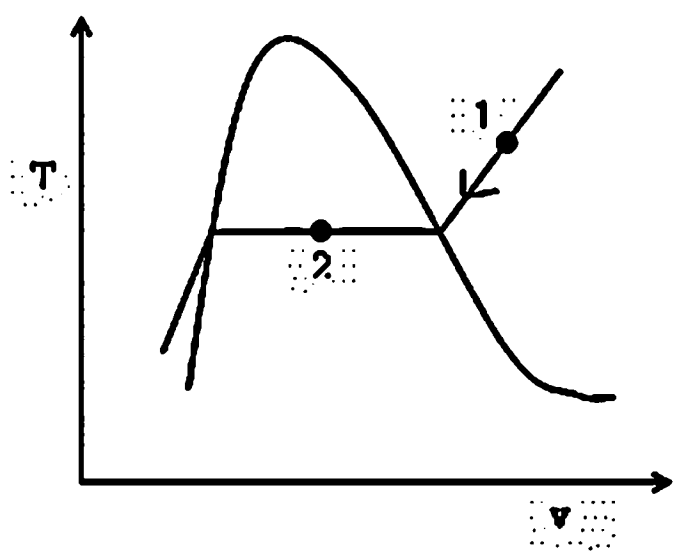
Mass of steam inside the cylinder, $m = 0.8\ kg$

Initial temperature, $T_1 = 300\ °C$

Final pressure = Initial pressure = $P = 1\ MPa$

Step 2 of 6

a) Given the initial state is superheated steam. And then the superheated steam is cooled at constant pressure until one half of the mass condenses. Thus the final state is a saturated water-vapor mixture. Thus the constant pressure process can be shown on a T-v diagram as



Step 3 of 6

b) The temperature of the saturated liquid-vapor mixture is the saturation temperature at the given pressure, P = 1 1MPa which can be obtained from saturated water-pressure table (Table A-5)

Therefore,

Final temperature, $T_2 = T_{sat}$ @ 1 MPa

$$\boxed{T_2 = 179.88°C}$$

Step 4 of 6

c) The initial state i.e., superheated steam is defined by temperature and pressure. The specific volume of superheated vapor at given temperature and pressure can be obtained from superheated water table (Table A-6)

Therefore, at $P = 1MPa$ and $T_1 = 300\ °C$

Initial specific volume, $v_1 = 0.25799\ m^3/kg$

Step 5 of 6

We know that steam is cooled till half of the mass gets condensed. Thus, finally the quality (x) of the mixture will be 0.5.

From the saturated water-pressure table (Table A-5)

At P = 1MPa

Specific volume of saturated liquid, $v_f = 0.001127\ m^3/kg$

Specific volume of saturated vapor, $v_g = 0.19436\ m^3/kg$

Step 6 of 6

Thus, the final specific volume (v_2) of the mixture with quality 0.5 can be determined as

$$v = v_f + (x \times (v_g - v_f))$$

$$v_2 = 0.001127 + (0.5 \times (0.19436 - 0.001127))$$

$$v_2 = 0.097743\ m^3/kg$$

Therefore, change in volume

$$\Delta V = m \times (v_2 - v_1)$$

$$\Delta V = 0.8 \times (0.097743 - 0.25799)$$

$$\boxed{\Delta V = -0.12819\ m^3}$$

Question: 4.060P

A rigid tank contains water vapor at 250°C and an unknown pressure. When the tank is cooled to 124°C, the vapor starts condensing. Estimate the initial pressure in the tank.

Step 1 of 3

At exit temperature $T_2 = 124°C$ the vapor starts condensing, and then the specific volume at this state is equal to the specific volume of saturated vapor. Obtain the specific volume from the saturated water table at temperature $T_2 = 124°C$. Now tabulate the values of specific volume at respective temperatures.

Temperature $(°C)$	Specific Volume (m^3/kg)
$120°\ (y_1)$	$0.89133\ m^3/kg\ (x_1)$
$125°\ (y_2)$	$0.77012\ m^3/kg\ (x_2)$
$124°\ (y)$	$v_2 = ?\ (x)$

Step 2 of 3

Relation for linear interpolation is expressed as,

$$\frac{x - x_1}{x_2 - x_1} = \frac{y - y_1}{y_2 - y_1}$$

Using linear interpolation calculate the specific volume.

$$v_2 = v_x$$
$$= 0.89133 + (0.77012 - 0.89133)\left(\frac{124 - 120}{125 - 120}\right)$$
$$= 0.794\,m^3/kg$$

Since it is a constant volume process, the initial specific volume is equal to the final specific volume. This can be expressed mathematically as follows:

$$v_1 = v_2 = 0.794\,m^3/kg$$

Step 3 of 3

Obtain the initial pressure at the initial tank temperature $T_1 = 250°C$ and specific volume $v_1 = 0.794\,m^3/kg$ from superheated water tables. Now tabulate the values of specific volume at respective temperatures.

Pressure (MPa) @ $250°C$ Superheated Water	Specific Volume
$0.2\ MPa$	$1.19890\ m^3/kg$
$0.3\ MPa$	$0.79645\ m^3/kg$
$P_1 = ?$	$0.794\,m^3/kg$

Using linear interpolation calculate the initial pressure.

$$P_1 = 0.2 + (0.3 - 0.2)\left(\frac{0.794 - 1.1989}{0.79645 - 1.1989}\right)$$
$$= 0.3006\,MPa$$
$$\simeq 0.3\,MPa$$

Therefore, initial pressure, P_1 in the tank is $\boxed{0.3\,MPa}$.

Question: 4.061P

A piston-cylinder device initially contains 1.4-kg saturated liquid water at 200°C. Now heat is transferred to the water until the volume quadruples and the cylinder contains saturated vapor only. Determine (*a*) the volume of the tank, (*b*) the final temperature and pressure, and (*c*) the internal energy change of the water.

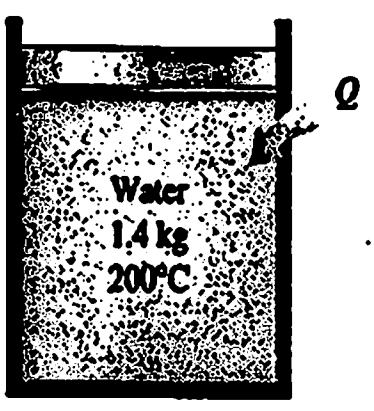


FIGURE P4–61

Step 1 of 6

Given

Initial state = saturated liquid

Initial temperature, $T_1 = 200\ °C$

Final state = saturated vapor

Final volume, $V_2 = 4 \times$ Initial volume $= 4V_1$

Mass inside the cylinder, $m = 1.4\ kg$

Step 2 of 6

a) We know that initial state is saturated liquid. The specific volume of the saturated liquid at the temperature T_1 can be obtained from the saturated water-temperature table

Therefore, at $T_1 = 200\ °C$

Initial specific volume, $v_1 = v_{f@200\,°C}$

$= 0.001157\ m^3/kg$

Therefore,

Initial volume of the tank, $V_1 = m \times v_1$

$V_1 = 1.4 \times 0.001157$

$V_1 = 0.0016198\ m^3$

Final volume of the tank, $V_2 = 4 \times V_1$

$$\boxed{V_2 = 0.0064792\ m^3}$$

Step 3 of 6

b) We know the final state is saturated vapor. The specific volume of saturated vapor is

$V_2 = 4 \times V_1$

$v_2 = 4 \times 0.001157$

$v_2 = 0.004628\ m^3/kg$

Thus we have a saturated vapor with specific volume 0.004628 m^3/kg. The temperature and pressure of this saturated vapor can be obtained from either saturated water- temperature table (Table A-4) or saturated water-pressure table (Table A-5).

From table A-4, we can see that we don't have the direct value of given specific volume of saturated vapor i.e., 0.004628 m^3/kg. Therefore, the temperature and pressure of this saturated vapor are obtained by interpolation

We have,

For $v_{gA} = 0.004953\ m^3/kg$

$T_A = 370\ °C$

$P_A = 21044\ kPa$

For $v_{gB} = 0.003106\ m^3/kg$

$T_B = 373.95\ °C$

$P_B = 22064\ kPa$

Step 4 of 6

By interpolation

$$\frac{T_2 - T_A}{T_B - T_A} = \frac{v_g - v_{gA}}{v_{gB} - v_{gA}}$$

$$\frac{T_2 - 370}{373.95 - 370} = \frac{0.004628 - 0.004953}{0.003106 - 0.004953}$$

Therefore, final temperature, $T_2 = 370.695\ °C$

Similarly

$$\frac{P_2 - P_A}{P_B - P_A} = \frac{v_g - v_{gA}}{v_{gB} - v_{gA}}$$

$$\frac{P_2 - 21044}{22064 - 21044} = \frac{0.004628 - 0.004953}{0.003106 - 0.004953}$$

Therefore, final pressure, $\boxed{P_2 = 21223.48\ kPa}$

Step 5 of 6

c) The initial internal energy i.e., internal energy of saturated liquid at 200 °C can be obtained from saturated water-temperature table (Table A-4)

The initial internal energy, $u_1 = u_{f@200\,°C} = 850.46\ kJ/kg$

The final internal energy i.e., internal energy of saturated liquid at 370.695 °C can be obtained from saturated water-temperature table (Table A-4) by interpolation.

For $T_A = 370\ °C$

$u_{gA} = 2230.1\ kJ/kg$

For $T_B = 373.95\ °C$

$u_{gB} = 2015.7\ kJ/kg$

Step 6 of 6

By interpolation

$$\frac{u_2 - u_{gA}}{u_{gB} - u_{gA}} = \frac{T_2 - T_A}{T_B - T_A}$$

$$\frac{u_2 - 2230.1}{2015.7 - 2230.1} = \frac{370.695 - 370}{373.95 - 370}$$

Therefore, final internal energy, $u_2 = 2192.376\ kJ/kg$

Therefore, total change in internal energy

$\Delta U = m \times (u_2 - u_1)$

$\Delta U = 1.4 \times (2192.376 - 850.46)$

$$\boxed{\Delta U = 1878.68\ kJ}$$

Question: 4.062P

How much error would one expect in determining the specific enthalpy by applying the incompressible-liquid approximation to water at 1500 psia and 400°F?

Step 1 of 4

Given

Pressure of compressed liquid water, $P = 1500$ psia

Temperature of compressed liquid, $T = 400\ ^\circ F$

Step 2 of 4

If we apply the incompressible liquid approximation i.e., the compressed liquid water is treated as saturated liquid at given temperature. The enthalpy of saturated liquid at 400 °F can obtained from saturated water-temperature table (Table A-4E)

Therefore at $T = 400\ ^\circ F$

Approximated enthalpy of compressed liquid, $h_{approx} = h_{f@400\ ^\circ F} = 375.04$ Btu/lbm

Step 3 of 4

Actual enthalpy of compressed liquid at given temperature and pressure can be obtained from compressed liquid water table (Table A-7E).

Therefore, at $P = 1500$ psia and $T = 400\ ^\circ F$

Actual enthalpy of compressed liquid, $h_{actual} = 376.51$ Btu/lbm

Step 4 of 4

Therefore, error involved in approximation is

$$\text{Percent error} = \frac{h_{actual} - h_{approx}}{h_{actual}} \times 100$$

$$\text{Percent error} = \frac{376.51 - 375.04}{376.51} \times 100$$

$$= 0.3904\ \%$$

Question: 4.063P

How much error would result in calculating the specific volume and enthalpy of water at 20 MPa and 140°C by using the incompressible-liquid approximation?

Step 1 of 5

Given
Water at pressure $P = 20\text{MPa}$
Water at temperature $T = 140°\text{C}$

Step 2 of 5

The state of water is in compressed liquid region
At $P = 20\text{MPa}$ and $T = 140°\text{C}$ from compressed liquid water tables
$v = 0.0010679\text{m}^3/\text{kg}$
$h = 602.07\text{kJ/kg}$

Step 3 of 5

Based upon the incompressible liquid approximation
$v = v_{f@140\text{kPa}} = 0.001080\text{m}^3/\text{kg}$
$h = h_{f140°C} = 589.16\text{kJ/kg}$

Step 4 of 5

The percent error involved in specific volume is

$$v_{\text{error}} = \frac{(0.001080 - 0.0010679)}{0.0010679} \times 100$$

$$\boxed{v_{\text{error}} = 1.1331\%}$$

Step 5 of 5

The percent error involved in enthalpy is

$$h_{\text{error}} = \frac{(602.07 - 589.16)}{602.07} \times 100$$

$$\boxed{h_{\text{error}} = 2.144\%}$$

Step 3 of 5

Based upon the incompressible liquid approximation
$v = v_{f@140\text{kPa}} = 0.001080\text{m}^3/\text{kg}$
$h = h_{f140°C} = 589.16\text{kJ/kg}$

Step 4 of 5

The percent error involved in specific volume is

$$v_{\text{error}} = \frac{(0.001080 - 0.0010679)}{0.0010679} \times 100$$

$$\boxed{v_{\text{error}} = 1.1331\%}$$

Question: 4.064P

100 grams of R-134a initially fill a weighted piston-cylinder device at 60 kPa and −20°C. The device is then heated until the temperature is 100°C. Determine the change in the device's volume as a result of the heating.

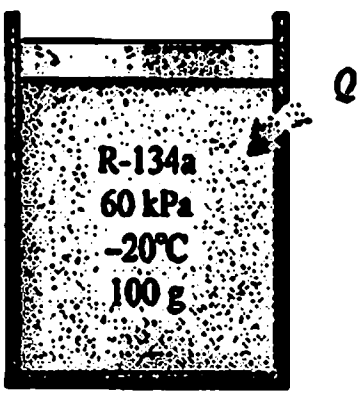


FIGURE P4–64

Step 1 of 4

Question:

100 grams of R-134a initially fill a weighted piston-cylinder device at 60 kPa and −20°C. The device is then heated until the temperature is 100°C. Determine the change in the device's volume as a result of the heating.

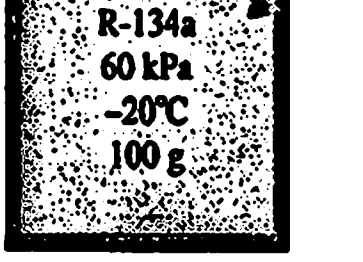


FIGURE P4–64

Step 1 of 4

Given

Mass of refrigerant-134a inside the cylinder, $m = 100\ g = 0.1\ kg$
Initial temperature, $T_1 = -20\ °C$.
Initial pressure, $P_1 = 60\ kPa$
Final temperature, $T_2 = 100\ °C$

Step 2 of 4

The initial state of refrigerant-134a is defined by temperature and pressure only. We do not know whether we have saturated mixture, compressed liquid or superheated vapor. To determine the region we are in, we go the saturated refrigerant-134a–pressure table (Table A-12) and determine the saturation temperature at the given pressure. Thus, for P = 60 kPa, we have $T_{sat} = -36.95\ °C$. We then compare the given T value to this T_{sat} value, keeping in mind that

if $T < T_{sat\ @\ given\ P}$ we have compressed liquid
if $T = T_{sat\ @\ given\ P}$ we have saturated mixture
if $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

In our case, the given T value is -20 °C, which is greater than the T_{sat} value at the given pressure. Therefore, we have superheated refrigerant initially. The specific volume of superheated refrigerant-134a can be obtained from superheated refrigerant-134a table (Table A-13)

Therefore, at $P_1 = 0.06\ MPa$ and $T_1 = -20\ °C$
Initial specific volume, $v_1 = 0.33608\ m^3/kg$

Step 3 of 4

As this is a constant pressure process

Question: 4.065P

A rigid vessel contains 2 kg of refrigerant-134a at 800 kPa and 120°C. Determine the volume of the vessel and the total internal energy.

Step 1 of 3

Given

Mass of refrigerant-134a inside the rigid vessel, $m = 2\ kg$

Pressure, $P = 800\ kPa$

Temperature, $T = 120°C$

Step 2 of 3

In this case, the temperature and pressure are given, but again we do not know whether we have saturated mixture, compressed liquid or superheated vapor. To determine the region we are in, we go the saturation refrigerant-134a-pressure table and determine the saturation temperature at the given pressure. Thus, for $P = 800$ kPa, we have $T_{sat} = 31.31$ °C. We then compare the given T value to this T_{sat} value, keeping in mind that

if $T < T_{sat\ @\ given\ P}$ we have compressed liquid

if $T = T_{sat\ @\ given\ P}$ we have saturated mixture

if $T > T_{sat\ @\ given\ P}$ we have superheated vapor.

In our case, the given T value is 120 °C, which is greater than the T_{sat} value at the given pressure. Therefore, we have superheated vapor. From the superheated refrigerant-134a table at pressure = 800 kPa = 0.8 MPa

Specific volume, $v = v\ @\ 120\ °C = 0.037625\ m^3/kg$

Hence, volume of the vessel, $V = m \times v$

$V = 2 \times 0.037625$

$\boxed{V = 0.07525\ m^3}$

Step 3 of 3

From the superheated refrigerant-134a table at pressure = 800 kPa = 0.8 MPa

Internal energy, $u = 327.87\ kJ/kg$

Therefore, total internal energy,

$U = m \times u$

$U = 2 \times 327.87$

$\boxed{U = 655.74\ kJ}$

Question: 4.066P

A 0.5-m3 vessel contains 10 kg of refrigerant-134a at −20°C. Determine (*a*) the pressure, (*b*) the total internal energy, and (*c*) the volume occupied by the liquid phase.

Step 1 of 5

Given

Volume of vessel, $V = 0.5\ m^3$

Mass of refrigerant-134a, $m = 10\ kg$

Temperature, $T = -20\ °C$

Step 2 of 5

Specific volume, $v = \frac{V}{m}$

$v = 0.05\ m^3/kg$

In this case the state is defined by temperature (T) and specific volume (v). We do not know which table to use to determine the other properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) from the saturated refrigerant-134a-temperature table (Table A-11).

At $T = -20\ °C$

$v_f = 0.0007362\ m^3/kg$

$v_g = 0.14729\ m^3/kg$

In our case, the given specific volume $v = 0.05\ m^3/kg$ lies in between the values of v_f and v_g at the given temperature. Thus, we have saturated liquid-vapor mixture. Thus the other properties of the state can be obtained from saturated refrigerant-134a table (Table A-11)

Step 3 of 5

a) Therefore, at $T = -20\ °C$

Pressure, $P = P_{sat\ @\ -20\ °C}$

$\boxed{P = 132.82\ kPa}$

Step 4 of 5

b) To determine the internal energy and volume occupied by the liquid phase, we need the quality of mixture. The quality (x) of mixture can be determined from

$$v = v_f + x \times (v_g - v_f)$$

Therefore, quality $x = \frac{v - v_f}{v_g - v_f}$

$$x = \frac{0.05 - 0.0007362}{0.14729 - 0.0007362}$$

$$x = 0.3361$$

From Table A-11, at $T = -20\ °C$

We have, $u_f = 25.39\ kJ/kg$

$u_{fg} = 193.45\ kJ/kg$

Therefore, the internal energy (u) of the mixture is

$u_2 = u_f + (x \times u_{fg})$

$u = 25.39 + (0.3361 * 193.45)$

$u = 90.41\ kJ/kg$

Thus, total internal energy,

$U = m \times u$

$U = 10 \times 90.41$

$\boxed{U = 904.1\ kJ}$

Step 5 of 5

c) The quality, x= 0.3361 implies that 33.61 percent of the mixture is vapor and the remaining 66.39 percent is liquid.

Hence, the mass of liquid phase,

$m_f = (1 - x) \times$ total mass of refrigerant

$m_f = (1 - 0.3361) \times 10$

$m_f = 6.639\ kg$

Volume occupied by the liquid phase,

$V_f =$ mass of liquid phase × specific volume of saturated liquid

$V_f = m_f \times v_f$

$V_f = 6.639 \times 0.0007362$

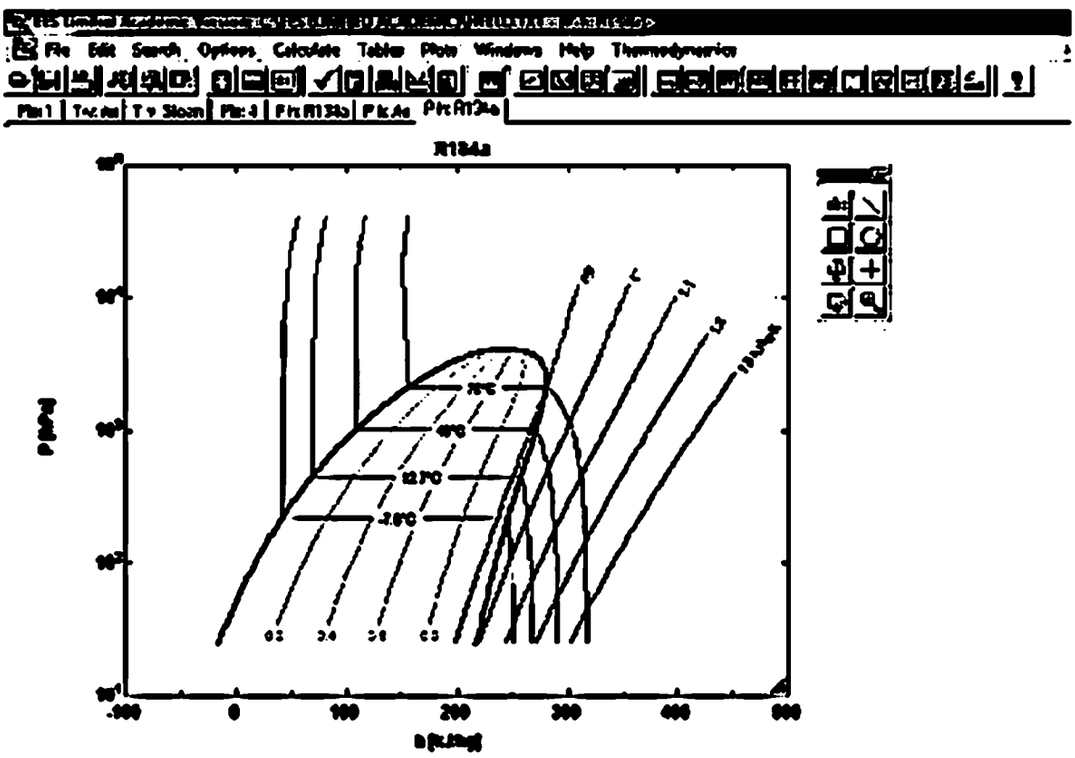
$\boxed{V_f = 0.00489\ m^3}$

Question: 4.067P

Pressure-enthalpy state diagrams are useful for studying refrigeration and like systems. Using EES (or other) software and actual property data, plot a pressure-enthalpy diagram for R-134a that includes the saturation lines. Also, sketch isothermal and constant-entropy processes on this diagram.

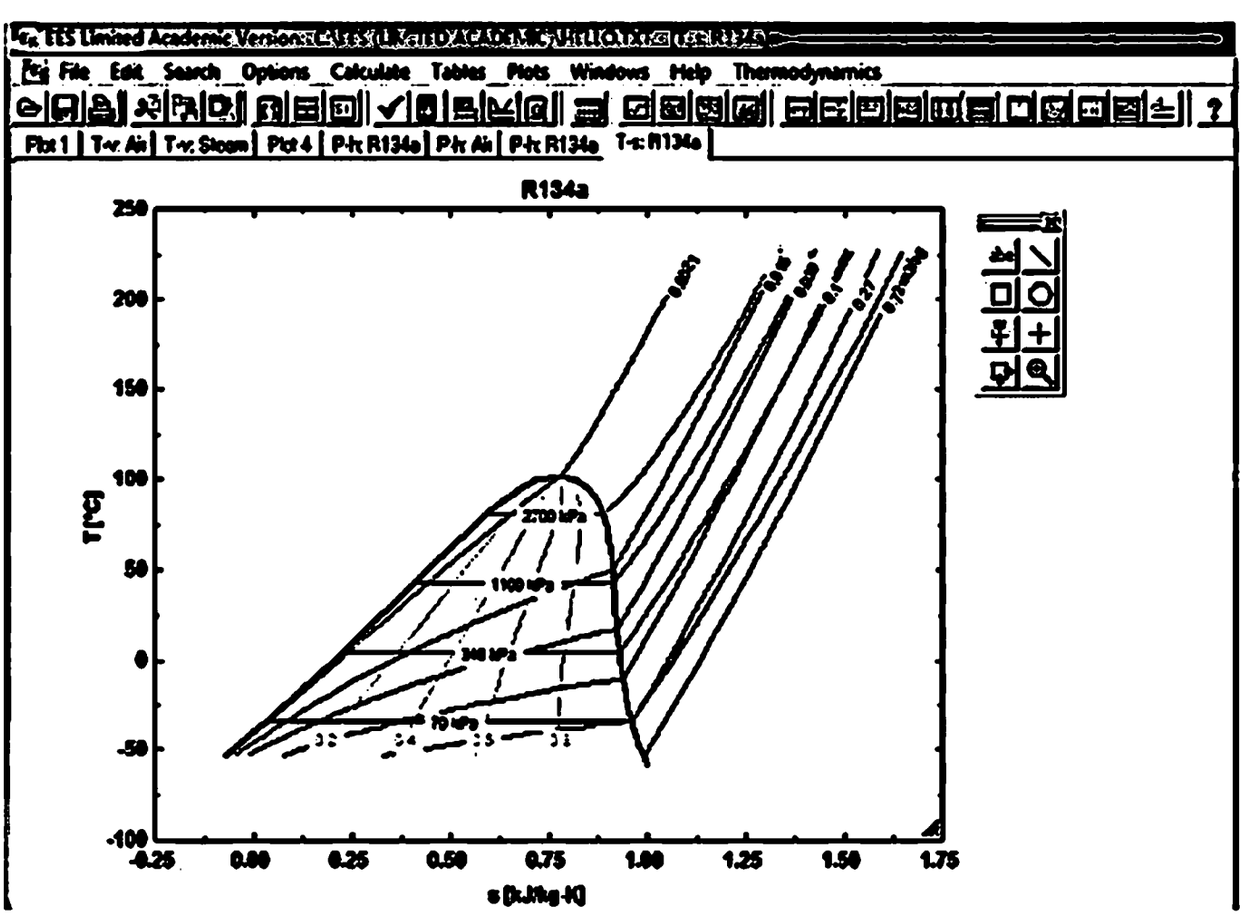
Step 1 of 2

Pressure-enthalpy diagram for refrigerator R-134a by using EES software:



Step 2 of 2

Temperature and entropy diagram for R-134a:



Question: 4.068P

A 0.3-m3 rigid vessel initially contains saturated liquid-vapor mixture of water at 150°C. The water is now heated until it reaches the critical state. Determine the mass of the liquid water and the volume occupied by the liquid at the initial state.

Step 1 of 6

Given

Volume of the rigid vessel, $V = 0.3\ m^3$

Temperature of saturated liquid-vapor mixture of water, $T_1 = 150\ °C$

Step 2 of 6

As the vessel is rigid, the process is a constant volume process. Hence the water is heated at constant volume till critical point. Thus the initial specific volume of the saturated mixture will be equal to the final specific volume which is the specific volume at critical state.

We know, the critical temperature of water is 373.95 °C. Thus the final specific volume can be obtained from saturated water-temperature table (Table A-4)

At $T = 373.95\ °C$

$v_{cr} = 0.003106\ m^3/kg$

Thus, initial specific volume, $v_1 = v_{cr} = 0.003106\ m^3/kg$.

Step 3 of 6

Total mass (m) of the saturated liquid-vapor mixture is

$$m = \frac{V}{v_1}$$

$$m = \frac{0.3}{0.003106}$$

$$m = 96.587\ kg$$

Step 4 of 6

The mass of liquid water can be obtained, once we know the quality of the saturated liquid-vapor mixture. The quality (x) of the mixture can be obtained from

$$v = v_f + (x \times (v_g - v_f))$$

From saturated water – temperature table (Table A-4)

At $T_1 = 150\ °C$

Specific volume of saturated liquid, $v_f = 0.001091\ m^3/kg$

Specific volume of saturated vapor, $v_g = 0.39248\ m^3/kg$

Therefore, quality $x = \frac{v_1 - v_f}{v_g - v_f}$

$$x = \frac{0.003106 - 0.001091}{0.39248 - 0.001091}$$

$$x = 0.00515$$

Step 5 of 6

Thus mass of liquid water, $m_L = (1-x) \times$ total mass

$$m_L = (1 - 0.00515) \times 96.587$$

$$\boxed{m_L = 96.089\ kg}$$

Step 6 of 6

Volume occupied by liquid at the initial state is

V_L = mass of liquid water (m_L) × specific volume of saturated liquid (v_f)

$$V_L = 96.089 \times 0.001091$$

$$\boxed{V_L = 0.1048\ m^3}$$

Question: 4.069P

What is the difference between mass and molar mass? How are these two related?

Step 1 of 1

Mass (m) is the total amount of matter or substance. Molar mass (M) is the mass of one mole of a substance in grams, or the mass of one kmol in kilograms. Molar mass is also called as molecular weight. Mass and molar mass can be related as

$$m = NM$$

where N= number of moles.

Question: 4.070P

Under what conditions is the ideal-gas assumption suitable for real gases?

Step 1 of 1

A gas can be treated as ideal gas when it is at higher temperatures or low pressures relative to its critical temperature and pressure. Because, at high temperatures and low pressures, the density of gas decreases and at low densities real gases behave as ideal gases.

Question: 4.071P

What is the difference between *R* and *R*u? How are these two related?

Step 1 of 1

R_u is the universal gas constant and is same for all the gases whereas R is the specific gas constant that is different for different gases. These two can be related as

$$R = \frac{R_u}{M}$$

Where M = molar mass of the gas or the molecular weight of the gas.

Question: 4.072P

Propane and methane are commonly used for heating in winter, and the leakage of these fuels, even for short periods, poses a fire danger for homes. Which gas leakage do you think poses a greater risk for fire? Explain.

Step 1 of 2

From table
Molar mass of propane, $M_P = 44.097$ kg/kmol
Molar mass of methane, $M_M = 16.043$ kg/kmol
Molar mass of air, $M_A = 28.97$ kg/kmol

Step 2 of 2

We can see that, the molar mass of propane is greater than the molar mass of air i.e, propane is heavier than air whereas methane is lighter than air.
Hence, when these gases leak out, propane will settle near the floor and methane will rise.
Thus leakage of propane poses a greater risk for fire.

Question: 4.073P

What is the specific volume of nitrogen at 300 kPa and 227°C?

Step 1 of 2

Given

Pressure of nitrogen $P = 300\text{kPa}$

Temperature of nitrogen $T = 227^0\text{C}$

Step 2 of 2

From properties of gas constant table, we get

Gas constant of nitrogen $R = 0.2968\text{kJ/kg.K}$

Assuming nitrogen behaves as an ideal gas, then by using ideal gas equation

$PV = mRT$

Specific volume of N_2

$$\frac{V}{m} = \frac{RT}{P}$$

$$v = \frac{0.2968 \times (227 + 273)}{300}$$

$$\boxed{v = 0.4946\,\text{m}^3/\text{kg}}$$

Question:

4.074P

A 3-ft3 container is filled with 2-lbm of oxygen at a pressure of 80 psia. What is the temperature of the oxygen?

Step 1 of 2

Given

Volume of the container $V = 3\text{ft}^3$

Mass of oxygen $m = 2\text{lbm}$

Pressure $P = 80\text{ psia}$

Step 2 of 2

From properties of gas constants gas constant of oxygen is

$R_{O_2} = 0.3353\text{ psia-ft}^3/\text{lbmR}$

Applying ideal gas equation of state

$$PV = mRT$$

$$T = \frac{PV}{mR}$$

$$= \frac{80 \times 3}{2 \times 0.3353}$$

$$T = 357.9\text{R}$$

Temperature of the oxygen in the container

$$\boxed{T = 357.9\text{R}}$$

Question:

4.075P

A 2-kg mass of helium is maintained at 300 kPa and 27°C in a rigid container. How large is the container, in m3?

Step 1 of 2

Given

Mass of Helium $m = 2\text{kg}$

Pressure $P = 300\text{kPa}$

Temperature $T = 27^0\text{C}$

Step 2 of 2

Assuming helium behaves as an ideal gas

From properties of gas constants

The gas constant of helium is

$R = 2.0769\text{kJ/kgK}$

According to the ideal gas equation of state,

$$PV = mRT$$

$$V = \frac{mRT}{P}$$

$$= \frac{2 \times 2.0769 \times (27+273)}{300}$$

$$V = 4.1538\text{m}^3$$

Volume of the container

$$\boxed{V = 4.1538\text{m}^3}$$

Question: 4.076P

A spherical balloon with a diameter of 9 m is filled with helium at 27°C and 200 kPa. Determine the mole number and the mass of the helium in the balloon.

Step 1 of 3

Calculate the volume of the sphere by using the following relation:

$$V = \frac{4\pi R^3}{3}$$

Here, R is the radius of the sphere.

Substitute $D/2$ for R.

$$V = \frac{4\pi (D/2)^3}{3}$$

Here, D is the diameter of the sphere.

Substitute $9\ \text{m}$ for D.

$$V = \frac{4\pi (9/2)^3}{3}$$
$$= 381.703\ \text{m}^3$$

Step 2 of 3

(a)

Calculate the number of moles of helium by using the following ideal gas equation:

$$n = \frac{PV}{R_u T}$$

Here, P is the pressure of the helium, T is the temperature of helium, and R_u is the universal gas constant.

Substitute $200\ \text{kPa}$ for P, $381.703\ \text{m}^3$ for V, $8.314\ \text{kJ/kg·K}$ for R_u, and $27°\text{C}$ for T.

$$n = \frac{200 \times 381.703}{8.314 \times ([27°\text{C} + 273]\ \text{K})}$$
$$= 30.6\ \text{kmol}$$

Therefore, the mole number of helium is $\boxed{30.6\ \text{kmol}}$.

Step 3 of 3

(b)

Obtain the molar mass of Helium from Table A-1, "Molar mass, gas constant, and critical-point properties," from the text book.

Molar mass of helium, $M = 4.003\ \text{kg/kmol}$

Calculate the mass of the helium in the balloon by using the following relation:

$$m = nM$$

Substitute $30.6\ \text{kmol}$ for n and $4.003\ \text{kg/kmol}$ for M.

$$m = 30.6 \times 4.003$$
$$= 122.5\ \text{kg}$$

Therefore, the mass of the Helium in the balloon is $\boxed{122.5\ \text{kg}}$.

Question: 4.077P

 Reconsider Prob. 4–76. Using EES (or other) software, investigate the effect of the balloon diameter on the mass of helium contained in the balloon for the pressures of (*a*) 100 kPa and (*b*) 200 kPa. Let the diameter vary from 5 m to 15 m. Plot the mass of helium against the diameter for both cases.

Step 1 of 2

Analysis: The problem is solved using EES, and the solution is given below.

```
"Given Data"
{D=9 [m]}
T=27 [C]
P=200 [kPa]
R_u=8.314 [kJ/kmol-K]
"Solution"
P*V=N*R_u*(T+273)
V=4*pi*(D/2)^3/3
m=N*MOLARMASS(Helium)
```

Step 2 of 2

Solution to the above EES program is as below

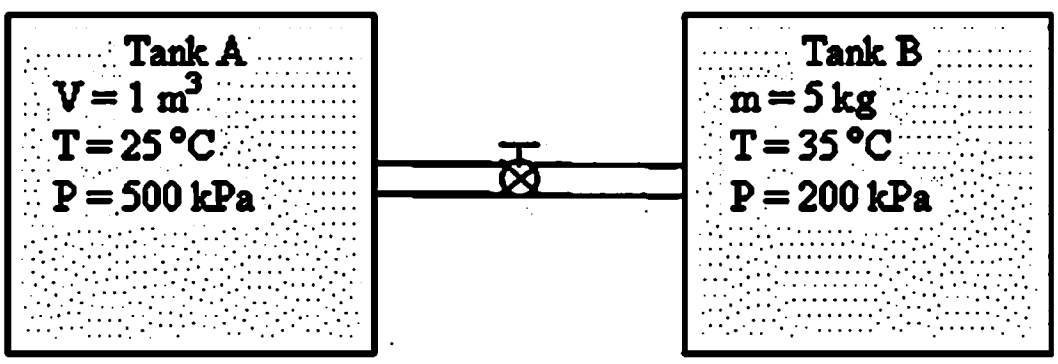
D[m]	m[kg]
5	21.02
6.111	38.37
7.222	63.33
8.333	97.28
9.444	141.7
10.56	197.745
11.67	266.92
12.78	350.743
13.89	450.397
15	567.344

Question: 4.078P

A 1-m3 tank containing air at 25 °C and 500 kPa is connected through a valve to another tank containing 5 kg of air at 35°C and 200 kPa. Now the valve is opened, and the entire system is allowed to reach thermal equilibrium with the surroundings, which are at 20°C. Determine the volume of the second tank and the final equilibrium pressure of air.

Step 1 of 4

Let the two tanks be represented by A and B as shown in the figure below.



Step 2 of 4

Given

Volume of the tank A, $V_A = 1\ m^3$

Temperature of air in tank A, $T_A = 25\ °C = 298\ K$

Pressure of air in tank A, $P_A = 500\ kPa$

Mass of air in tank B, $m_B = 5\ kg$

Temperature of air in tank B, $T_B = 35\ °C = 308\ K$

Pressure of air in tank B, $P_B = 200\ kPa$

Surrounding temperature, $T_{surr} = 20\ °C = 293\ K$

Step 3 of 4

Assuming, at given conditions air behaves as an ideal gas.

For air, gas constant $R = 0.287\ kJ/kmol\ K$

From ideal gas equation, mass of air in tank A is determined by

$$m_A = \frac{P_A V_A}{RT_A}$$

$$= \frac{500 \times 1}{0.287 \times 298}$$

$$m_A = 5.846\ kg$$

Volume of the tank B can be determined from

$$V_B = \frac{m_B R T_B}{P_B}$$

$$= \frac{5 \times 0.287 \times 308}{200}$$

$$V_B = 2.21\ m^3$$

Step 4 of 4

Now, when the valve is opened

Total volume, $V = V_A + V_B = 1 + 2.21 = 3.21\ m^3$

Total mass of air, $m = m_A + m_B = 5 + 5.846 = 10.846\ kg$

The final equilibrium pressure (P) can be obtained from the ideal gas equation applied to total volume

$$\text{Therefore, } P = \frac{mRT_{surr}}{V}$$

$$P = \frac{10.846 \times 0.287 \times 293}{3.21}$$

$$\boxed{P = 284.13\ kPa}.$$

Question:

4.079P

2.3 lbmol of air at 65°F and 32 psia is contained in an elastic tank. What is the volume of the tank? If the volume is doubled at the same pressure, determine the final temperature.

Step 1 of 3

Given

Number of moles of air $n = 2.3\text{lbmol}$

Initial temperature of air $T_1 = 65°\text{F}$

$$T_1 = 65+459.67$$

$$T_1 = 524.67\text{R}$$

Initial pressure of air $P_1 = 32\,\text{psia}$

Step 2 of 3

Universal gas constant

$R_u = 10.7316\text{psia-ft}^3/\text{lbmol-R}$

Question:

2.3 lbmol of air at 65°F and 32 psia is contained in an elastic tank. What is the volume of the tank? If the volume is doubled at the same pressure, determine the final temperature.

Step 1 of 3

Given

Number of moles of air $n = 2.3\text{lbmol}$

Initial temperature of air $T_1 = 65°\text{F}$

$$T_1 = 65+459.67$$

$$T_1 = 524.67\text{R}$$

Initial pressure of air $P_1 = 32\,\text{psia}$

Step 2 of 3

Question:

4.080P

A rigid tank contains an ideal gas at 1227°C and 200 kPa gage. The gas is cooled until the gage pressure reads 50 kPa. If the atmospheric pressure is 100 kPa, determine the final temperature of the gas.

Step 1 of 3

Given

Temperature of ideal gas $T_1 = 1227°C$

$$T_1 = 1227 + 273$$

$$T_1 = 1500K$$

Initial gage pressure of the ideal gas $P_1 = 200kPa$

Final gage pressure $P_2 = 50kPa$

Atmospheric pressure $P_{atm} = 100kPa$

Step 2 of 3

Initial absolute pressure of the gas

Question:

A rigid tank contains an ideal gas at 1227°C and 200 kPa gage. The gas is cooled until the gage pressure reads 50 kPa. If the atmospheric pressure is 100 kPa, determine the final temperature of the gas.

Step 1 of 3

Given

Temperature of ideal gas $T_1 = 1227°C$

$$T_1 = 1227 + 273$$

$$T_1 = 1500K$$

Initial gage pressure of the ideal gas $P_1 = 200kPa$

Final gage pressure $P_2 = 50kPa$

Atmospheric pressure $P_{atm} = 100kPa$

Question: 4.081P

A rigid tank whose volume is unknown is divided into two parts by a partition. One side of the tank contains an ideal gas at 927°C. The other side is evacuated and has a volume twice the size of the part containing the gas. The partition is now removed and the gas expands to fill the entire tank. Heat is now applied to the gas until the pressure equals the initial pressure. Determine the final temperature of the gas.

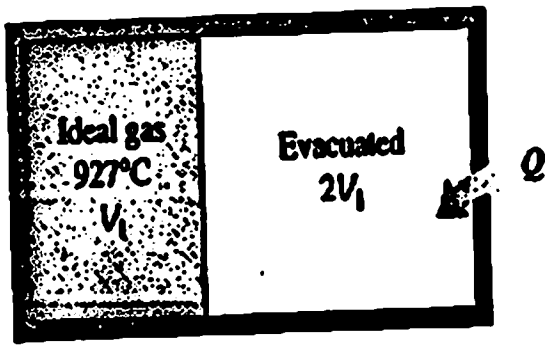


FIGURE P4–81

Step 1 of 3

Question:

A rigid tank whose volume is unknown is divided into two parts by a partition. One side of the tank contains an ideal gas at 927°C. The other side is evacuated and has a volume twice the size of the part containing the gas. The partition is now removed and the gas expands to fill the entire tank. Heat is now applied to the gas until the pressure equals the initial pressure. Determine the final temperature of the gas.

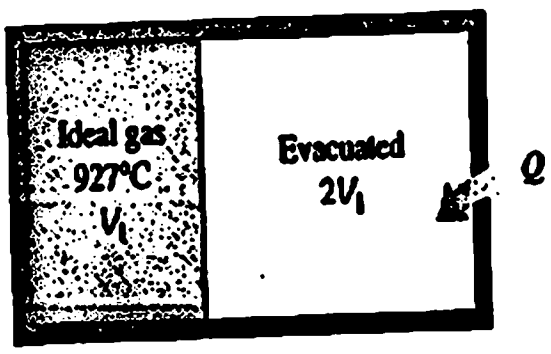


FIGURE P4–81

Step 1 of 3

Initial temperature $T_1 = 927°C$

$T_1 = 927 + 273$

$T_1 = 1200K$

Initial volume V_1

Evacuated volume $2V_1$

Step 2 of 3

$= 3 \times 1200$

$T_2 = 3600K$

$T_2 = 3327^0C$

Final temperature of the gas

$\boxed{T_2 = 3327^0C}$

Question: 4.082P

Argon in the amount of 0.6 kg fills a 0.05-m3 piston-cylinder device at 550 kPa. The piston is now moved by changing the weights until the volume is twice its original size. During this process, argon's temperature is maintained constant. Determine the final pressure in the device.

Step 1 of 2

Given

Mass of Argon $m = 0.6\text{kg}$

Initial volume $V_1 = 0.05\text{m}^3$

Initial pressure $P_1 = 550\text{kPa}$

Final volume $V_2 = 2V_1$

Step 2 of 2

Since temperature remains constant

Applying ideal gas equation

$$P_1V_1 = P_2V_2$$

$$P_2 = \frac{P_1V_1}{V_2}$$

$$= \frac{550 \times V_1}{2V_1}$$

$$P_2 = 275\text{kPa}$$

So, final pressure in the device

$$\boxed{P_2 = 275\text{kPa}}$$

$$P_1V_1 = P_2V_2$$

$$P_2 = \frac{P_1V_1}{V_2}$$

$$= \frac{550 \times V_1}{2V_1}$$

$$P_2 = 275\text{kPa}$$

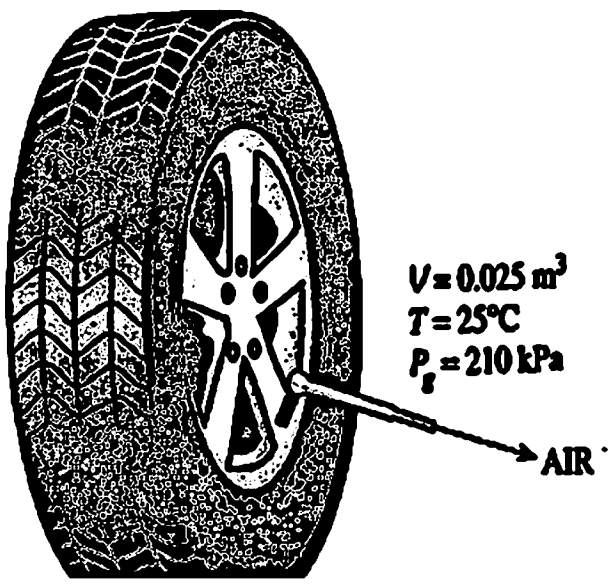
So, final pressure in the device

$$\boxed{P_2 = 275\text{kPa}}$$

Question:

4.083P

The pressure in an automobile tire depends on the temperature of the air in the tire. When the air temperature is 25 °C, the pressure gage reads 210 kPa. If the volume of the tire is 0.025 m3, determine the pressure rise in the tire when the air temperature in the tire rises to 50°C. Also, determine the amount of air that must be bled off to restore pressure to its original value at this temperature. Assume the atmospheric pressure is 100 kPa.



Question:

The pressure in an automobile tire depends on the temperature of the air in the tire. When the air temperature is 25 °C, the pressure gage reads 210 kPa. If the volume of the tire is 0.025 m3, determine the pressure rise in the tire when the air temperature in the tire rises to 50°C. Also, determine the amount of air that must be bled off to restore pressure to its original value at this temperature. Assume the atmospheric pressure is 100 kPa.

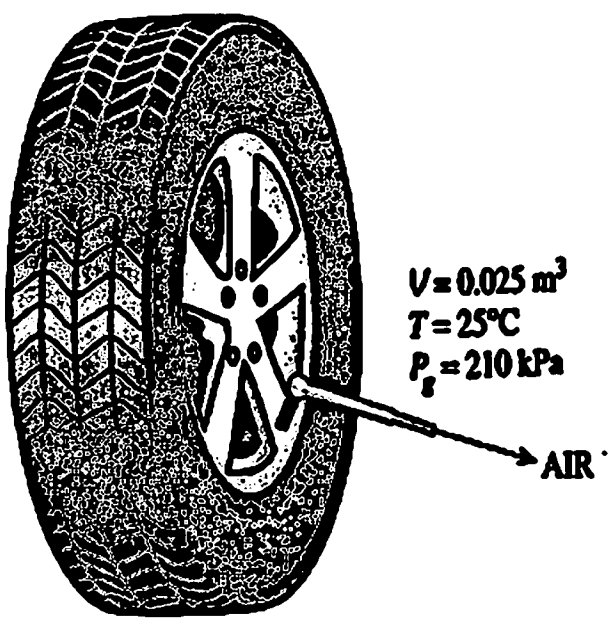


FIGURE P4–83

Step 1 of 4

Given

Initial air temperature, $T_1 = 25\,°C = 298\ K$
Initial gage pressure, $P_g = 210\ kPa$
Volume of the tire, $V = 0.025\ m^3$
Final Temperature, $T_2 = 50\,°C = 323\ K$
Atmospheric pressure, $P_{atm} = 100\ kPa$

Step 2 of 4

Assuming, at given conditions air behaves as an ideal gas.
Initially the absolute pressure in the tire is, $P_1 = P_g + P_{atm} = 210 + 100 = 310\ kPa$
From the ideal gas equation, we have

$$P_1V_1 = mRT_1 \quad \text{------ (1)}$$
$$P_2V_2 = mRT_2 \quad \text{------ (2)}$$

Assuming the volume of the tire remains constant
i.e., $V_1 = V_2$
From equations (1) and (2), eliminating m and R, we can write

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

Therefore, $P_2 = \dfrac{P_1T_2}{T_1}$

$$= \frac{310 \times 323}{298}$$

$$P_2 = 336\ kPa$$

Therefore pressure rise $\Delta p = P_2 - P_1$

$$= 336 - 310$$

Pressure rise $\Delta p = 26\ kPa$

Step 3 of 4

Therefore, $m_2 = \dfrac{310 \times 0.025}{0.287 \times 323}$

$$m_2 = 0.0836\ kg$$

The amount of air that must be bled off to restore pressure to its original value is,

$$\Delta m = m_1 - m_2$$
$$= 0.0906 - 0.0836$$
$$\boxed{\Delta m = 0.0070\ kg}$$

Question: 4.084P

What is the physical significance of the compressibility factor Z?

Step 1 of 1

Generally, gases deviate from the ideal-gas behaviour significantly at states near to the saturation region and the critical point (can be seen in fig. 3-49). This deviation from ideal-gas behaviour at a given temperature and pressure is accurately accounted by a correction factor called compressibility factor Z.

Question: 4.085P

What is the principle of corresponding states?

Step 1 of 1

The principle of corresponding states says that the compressibility factor for all gases is approximately the same at the same reduced pressure and temperature.

Question: 4.086P

How are the reduced pressure and reduced temperature defined?

Step 1 of 1

Reduced pressure and reduced temperature are the normalized pressure and temperature with respect to their critical pressure and temperature respectively.

Reduced pressure, P_R is

$$P_R = \frac{P}{P_{cr}}$$

where P_{cr} = critical pressure

Reduced temperature, T_R is

$$T_R = \frac{T}{T_{cr}}$$

where Tcr = critical temperature.

Question: 4.087P

Determine the specific volume of superheated water vapor at 15 MPa and 350°C, using (*a*) the ideal-gas equation, (*b*) the generalized compressibility chart, and (*c*) the steam tables. Also determine the error involved in the first two cases.

Step 1 of 6

Given
Superheated water at pressure $P = 15\text{MPa}$
Temperature $T = 350^0\,\text{C}$

Step 2 of 6

(a) Using ideal gas equation of state

$$v = \frac{RT}{P}$$

Where gas constant for superheated steam is

$$R = 0.4615\text{kJ/kg.K}$$

$$v = \frac{(0.4615)\times(350+273)}{(15000)}$$

$$\boxed{v = 0.01917\text{m}^3/\text{kg}}$$

Step 3 of 6

(b) Using compressibility chart
Critical state of water is

$$P_{cr} = 22.06\text{MPa}$$

$$T_{cr} = 647.1\text{K}$$

Reduced pressure is

$$P_R = \frac{P}{P_{cr}}$$

$$= \frac{15}{22.06}$$

$$P_R = 0.68$$

Reduced temperature is

$$T_R = \frac{T}{T_{cr}}$$

$$= \frac{350+273}{647.1}$$

$$T_R = 0.963$$

Using compressibility chart at $P_R = 0.68$ and $T_R = 0.963$

Compressibility factor $Z = 0.65$

Thus,

$$v_1 = Z\, v_{ideal}$$

$$= 0.65\times 0.01917$$

$$\boxed{v_1 = 0.01246\,\text{m}^3/\text{kg}}$$

Step 4 of 6

(c) From the superheated steam table
At $P = 15\text{MPa}, T = 350°\text{C}$

$$\boxed{v = 0.011481\text{m}^3/\text{kg}}$$

Step 5 of 6

Error involved by using ideal gas equation and by using superheated steam tables

$$\text{percentage error} = \frac{0.01917 - 0.011481}{0.011481}$$

$$\boxed{\text{Percentage error}=66.97\%}$$

Step 6 of 6

Error involved by using generalized compressibility chart and by using superheated steam tables

$$\text{percentage error} = \frac{0.01246 - 0.011481}{0.011481}$$

$$\boxed{\text{percentage error} = 8.52\%}$$

Question: 4.088P

Reconsider Prob. 4–87. Solve the problem using the generalized compressibility factor feature of the EES software. Again using EES, compare the specific volume of water for the three cases at 15 MPa over the temperature range of 350 to 600°C in 25°C intervals. Plot the percent error involved in the ideal-gas approximation against temperature, and discuss the results.

Step 1 of 1

4672-3-88E SA 0261

Analysis: The problem is solved using EES, and the solution is given below.

$$P = 15[MPa] \times Convert(MPa, kPa)$$

$$\{T_{Celsius} = 350[C]\}$$

$$T = T_{Celsius} + 273[K]$$

$$T_{critical} = T_{CRIT}(Steam_{IAPWS})$$

$$P_{critical} = P_{CRIT}(Steam_{IAPWS})$$

$$\{v = Vol / m\}$$

$$P_{table} = P; P_{comp} = P; P_{ideal\ gas} = P$$

$$T_{table} = T; T_{comp} = T; T_{ideal\ gas} = T$$

$$v_{table} = volume(Steam_{IAPWS}, P = P_{table}, T = T_{table})$$ "EES data for steam as a real gas"

$$\{P_{table} = pressure(Steam_{IAPWS}, T = T_{table}, v = v)\}$$

$$\{T_{sat} = temperature(Steam_{IAPWS}, P = P_{table}, v = v)\}$$

MM = MOLARMASS (water)

$$R_u = 8.314[kJ / kmol - K]$$ "Universal gas constant"

$$R = R_u / MM$$ "[kJ/kg-K], Particular gas constant"

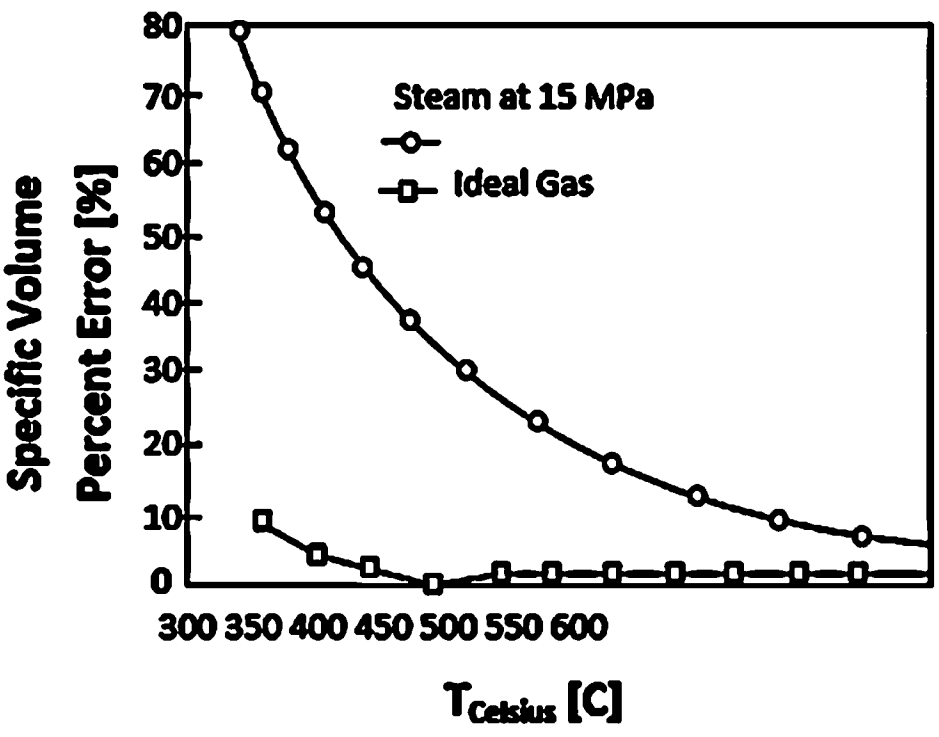
$$P_{ideal\ gas} \times v_{ideal\ gas} = R \times T_{ideal\ gas}$$ "Ideal gas equation"

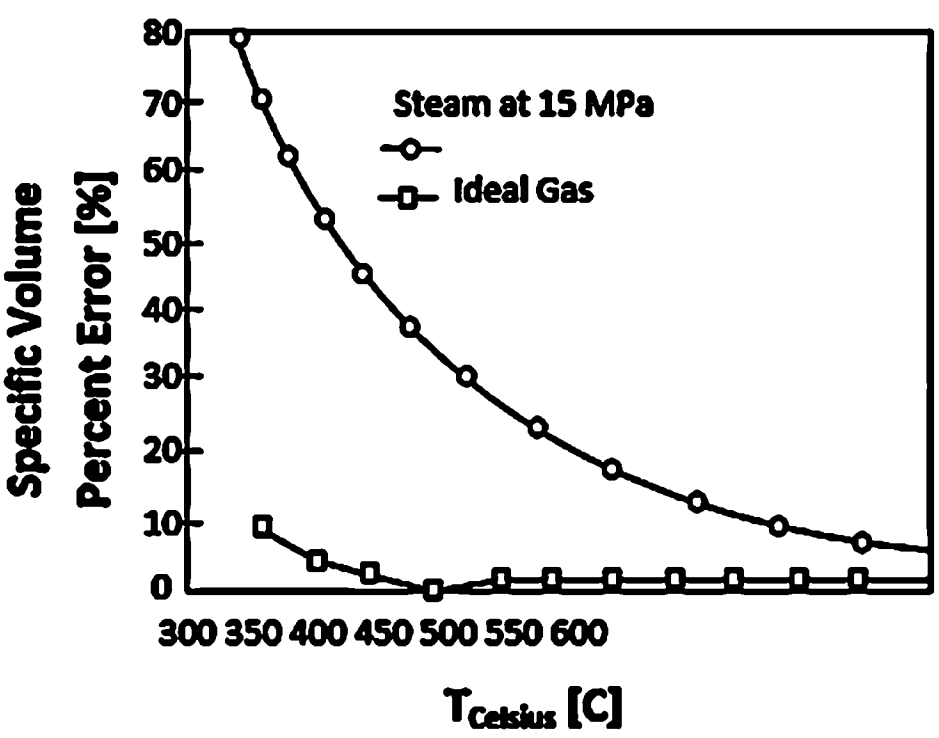
Z = COMPRESS $(T_{comp} / T_{critical}, P_{comp} / P_{critical})$

$$P_{comp} \times v_{comp} = z \times R \times T_{comp}$$ "generalized Compressibility factor"

$$Error_{ideal\ gas} = Abs(v_{table} - v_{ideal\ gas}) / v_{table} \times Convert(,\%)$$

$$Error_{comp} = Abs(v_{table} - v_{comp}) / v_{table} \times Convert(,\%)$$





Question: 4.089P

Determine the specific volume of refrigerant-134a vapor at 0.9 MPa and 70°C based on (*a*) the ideal-gas equation, (*b*) the generalized compressibility chart, and (*c*) data from tables. Also, determine the error involved in the first two cases.

Step 1 of 6

Given

Pressure, $P = 0.9$ MPa

Temperature, $T = 70\,^{\circ}C = 343$ K

Step 2 of 6

a) Assuming, the superheated refrigerant-134a behaves as an ideal gas at specified conditions.

For refrigerant-134a, gas constant $R = 0.08149$ kJ/kg K

From the ideal gas equation, specific volume is given by

$$v = \frac{RT}{P}$$

$$= \frac{0.08149 \times 343}{0.9 \times 1000}$$

$v = 0.03105$ m^3/kg.

Step 3 of 6

b) The critical pressure and temperature of refrigerant-134a are

$P_{cr} = 4.059$ MPa

$T_{cr} = 374.2$ K

Therefore,

$$\text{Reduced pressure, } P_r = \frac{P}{P_{cr}}$$

$$= \frac{0.9}{4.059}$$

$$P_r = 0.22$$

$$\text{Reduced temperature, } T_r = \frac{T}{T_{cr}}$$

$$= \frac{343}{374.2}$$

$$T_r = 0.916$$

Step 4 of 6

From compressibility chart (A-15), corresponding to the obtained P_r and T_r values, we have

$Z = 0.895$

Therefore, actual specific volume is

$$v_{actual} = Z \times v_{ideal}$$

$$= 0.895 \times 0.03105$$

$v_{actual} = 0.02778$ m^3/kg

Step 5 of 6

c) For superheated refrigerant-134a

for pressure, $P = 0.9$ MPa

and $T = 70\,^{\circ}C$

We have, $v = 0.027413$ m^3/kg.

Step 6 of 6

The error involved in the first case is,

$$\text{Error} = \frac{0.03105 - 0.027413}{0.027413} \times 100 = 13.26\%$$

The error involved in the second case is,

$$\text{Error} = \frac{0.02778 - 0.027413}{0.027413} \times 100 = 1.34\%$$

Question: 4.090P

Determine the specific volume of superheated water vapor at 3.5 MPa and 450°C based on (*a*) the ideal-gas equation, (*b*) the generalized compressibility chart, and (*c*) the steam tables. Determine the error involved in the first two cases.

Step 1 of 1

2760-3-92P

SA: 3801
SR: 4584

Given

Pressure, $P = 3.5$ MPa

Temperature, $T = 450\,^{\circ}C = 723$ K

---

a) Assuming, the superheated water behaves as an ideal gas at specified conditions.

For water, gas constant $R = 0.4615$ kJ/kg K (from Table-1)

From the ideal gas equation, specific volume is given by

$$v = \frac{RT}{P}$$

$$= \frac{0.4615 \times 723}{3.5 \times 1000}$$

$$\boxed{v = 0.09533\ m^3/kg}.$$

---

b) The critical pressure and temperature of water are

$P_{cr} = 22.06$ MPa

$T_{cr} = 647.1$ K

Therefore,

$$\text{Reduced pressure, } P_r = \frac{P}{P_{cr}}$$

$$= \frac{3.5}{22.06}$$

$$P_r = 0.158$$

$$\text{Reduced temperature, } T_r = \frac{T}{T_{cr}}$$

$$= \frac{723}{647.1}$$

$$T_r = 1.12$$

From compressibility chart (A-15), corresponding to the obtained P_r and T_r values, we have

$Z = 0.96$

Therefore, actual specific volume is

$$v_{actual} = Z \times v_{ideal}$$

$$= 0.96 \times 0.09533$$

$$v_{actual} = 0.09152\ m^3/kg$$

---

c) For superheated water (from Table A-6)

For pressure, $P = 3.5$ MPa

and $T = 450\,^{\circ}C$

We have, $\boxed{v = 0.09198\ m^3/kg}$

The error involved in the first case is,

$$\text{Error} = \frac{0.09533 - 0.09198}{0.09198} \times 100 = 3.64\%$$

The error involved in the second case is,

$$\text{Error} = \frac{0.09198 - 0.09152}{0.09198} \times 100 = 0.5\%$$

Question: 4.091P

Ethane in a rigid vessel is to be heated from 80 psia and 100°F until its temperature is 540°F. What is the final pressure of the ethane as predicted by the compressibility chart?

Step 1 of 6

Given

Initial pressure of Ethane $P_1 = 80\,\text{psia}$

Initial temperature $T_1 = 100^0\text{F}$

Final temperature $T_2 = 540^0\text{F}$

Step 2 of 6

The gas constant, the critical pressure, and the critical temperature of ethane are from table Molar mass, gas constant, and critical point properties

$R = 0.3574\,\text{psia.ft}^3\text{/lbm.R}$

$T_{cr} = 549.8\,\text{R}$

$P_{cr} = 708\,\text{psia}$

Step 3 of 6

Reduced parameters at state 1

$$T_{R_1} = \frac{T_1}{T_{cr}}$$

$$= \frac{(100+459.67)}{549.8}$$

$$T_{R1} = 1.018$$

$$P_{R_1} = \frac{P_1}{P_{cr}}$$

$$= \frac{80}{708}$$

$$P_{R_1} = 0.113$$

From compressibility chart at $T_{R1} = 1.018$ and $P_{R_1} = 0.113$

$Z_1 = 0.961$

Step 4 of 6

The specific volume does not change during the process. Then,

$$v_1 = v_2 = \frac{Z_1 R T_1}{P_1}$$

$$= \frac{0.961\times(0.3574)\times(559.67)}{80}$$

$$v_1 = v_2 = 2.403\,\text{ft}^3\text{/lbm}$$

Step 5 of 6

At the final state the reduced parameters are

$$T_{R_2} = \frac{T_2}{T_{cr}}$$

$$= \frac{(540+459.67)}{549.8}$$

$$T_{R_2} = 1.82$$

$$v_{R_2} = \frac{v_{2actual}}{RT_{cr}/P_{cr}}$$

$$= \frac{(2.403)}{(0.3574\times549.8)/708}$$

$$v_{R_2} = 8.66\text{ft}^3\text{/lbm}$$

Step 6 of 6

From compressibility chart at $T_{R_2} = 1.82$ and $v_{R_2} = 8.66\text{ft}^3\text{/lbm}$

$Z_2 = 0.98$

Thus,

$$P_2 = \frac{Z_2 R T_2}{v_2}$$

$$= \frac{(0.98\times0.3574\times999.67)}{2.403}$$

$$\boxed{P_2 = 145.71\,\text{psia}}$$

Question: 4.092P

Ethylene is heated at constant pressure from 5 MPa and 20°C to 200°C. Using the compressibility chart, determine the change in the ethylene's specific volume as a result of this heating.

Step 1 of 6

Given

Initial temperature of ethylene, $T_1 = 20\,^\circ C = 293\,K$

Final temperature, $T_2 = 200\,^\circ C = 473\,K$

Initial pressure = final pressure = $P = 5\,MPa$

Step 2 of 6

The gas constant, critical pressure and temperature of ethylene are

$R = 0.2964\ kJ/kg\,K$

$P_{cr} = 5.12\ MPa$

$T_{cr} = 282.4\ K$

The reduced pressure and temperature of initial state are

$$P_{R1} = \frac{P}{P_{cr}} = \frac{5}{5.12} = 0.9765$$

$$T_{R1} = \frac{T_1}{T_{cr}} = \frac{293}{282.4} = 1.0375$$

From the compressibility chart (Figure A-15), corresponding to P_{R1} and T_{R1},

The compressibility factor, $Z_1 = 0.58$ (approximately)

Step 3 of 6

Thus, initial specific volume (v_1) can be obtained from

$$v_1 = \frac{Z_1 R T_1}{P} = \frac{0.58 \times 0.2964 \times 293}{5000} = 0.01007\ m^3/kg$$

Step 4 of 6

The reduced pressure and temperature of final state are

$$P_{R2} = \frac{P}{P_{cr}} = \frac{5}{5.12} = 0.9765$$

$$T_{R2} = \frac{T_2}{T_{cr}} = \frac{473}{282.4} = 1.675$$

From the compressibility chart (Figure A-15), corresponding to P_{R2} and T_{R2},

The compressibility factor, $Z_2 = 0.95$ (approximately)

Step 5 of 6

Thus, final specific volume (v_2) can be obtained from

$$v_2 = \frac{Z_2 R T_2}{P} = \frac{0.95 \times 0.2964 \times 473}{5000} = 0.02664\ m^3/kg$$

Step 6 of 6

Therefore, change in specific volume of ethylene due to heating is

$$\Delta v = v_2 - v_1 = 0.02664 - 0.01007$$

$$\boxed{\Delta v = 0.01637\ \frac{m^3}{kg}}$$

Question: 4.093P

Saturated water vapor at 350°C is heated at constant pressure until its volume has doubled. Determine the final temperature using the ideal gas equation of state, the compressibility charts, and the steam tables.

Step 1 of 7

(a)

Calculate the temperature of steam from ideal gas equation.

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

Here, pressure of steam at state 1 is P_1, volume of steam at state 1 is v_1, temperature of steam at state 1 is T_1, pressure of steam at state 2 is P_2, volume of steam at state 2 is v_2 and temperature of steam at state 2 is T_2.

Substitute $P_1 = P_2$ because the process is isobaric pressure, $v_2 = 2v_1$ and $623\ \text{K}$ for T_1.

$$\frac{v_1}{623} = \frac{2v_1}{T_2}$$
$$T_2 = 1246\ \text{K}$$

Therefore, final temperature of steam is $\boxed{1246\ \text{K}}$.

Step 2 of 7

(b)

Write the gas constant, critical pressure and critical temperature of water from Table A-1.

$$R = 0.4615\ \text{kJ/kg}\cdot\text{K}$$
$$P_{cr} = 22.06\ \text{MPa}$$
$$T_{cr} = 647.1\ \text{K}$$

Calculate the initial pressure of saturated water vapour from Table A-4: saturated water-temperature table.

$$P_1 = P_{sat@350°C}$$
$$= 16.529\ \text{MPa}$$

Step 3 of 7

Calculate the reduced pressure at state 1.

$$P_{R1} = \frac{P_1}{P_{cr}}$$

Here, pressure at state 1 is P_1 and critical pressure of water is P_{cr}.

Substitute $16.529\ \text{MPa}$ for P_1 and $22.06\ \text{MPa}$ for P_{cr}.

$$P_{R1} = \frac{16.529}{22.06}$$
$$= 0.7493$$

Step 4 of 7

Calculate the reduced temperature at state 1.

$$T_{R1} = \frac{T_1}{T_{cr}}$$

Here, temperature of water at state 1 is T_1 and critical temperature of water is T_{cr}.

Substitute $623\ \text{K}$ for T_1 and $647.1\ \text{K}$ for T_{cr}.

$$T_{R1} = \frac{623}{647.1}$$
$$= 0.962$$

Step 5 of 7

Write the compressibility factor by using the compressibility chart from figure A-15 corresponding to $P_{R1} = 0.7493$ and $T_{R1} = 0.962$.

$$Z_1 = 0.593$$

Write the pseudo-reduced specific volume by using the compressibility chart from figure A-15 corresponding to $P_{R1} = 0.7493$ and $T_{R1} = 0.962$.

$$v_{R1} = 0.75$$

Step 6 of 7

Calculate the reduced pressure at state 2.

$$P_{R2} = P_{R1}$$

Here, reduced pressure at state 1 is P_{R1} and reduced pressure at state 2 is P_{R2}.

Substitute 0.7493 for P_{R1}.

$$P_{R2} = 0.7493$$

Calculate the reduced specific volume at state 3.

$$v_{R2} = 2v_{R1}$$

Here, reduced specific volume at state 1 is v_{R1} and reduced specific volume at state 2 is v_{R2}.

Substitute 0.75 for v_{R1}.

$$v_{R2} = 2 \times 0.75$$
$$= 1.5$$

Write the compressibility factor by using the compressibility chart from figure A-15 corresponding to $P_{R2} = 0.7493$ and $v_{R2} = 1.5$.

$$Z_2 = 0.88$$

Calculate the temperature at state 2.

$$T_2 = \frac{P_2 v_2}{Z_2 R}$$
$$= \frac{P_2 v_{R2} T_{cr}}{Z_2 P_{cr}}$$

Substitute $16529\ \text{kPa}$ for P_2, 1.5 for v_{R2}, $647.1\ \text{K}$ for T_{cr}, 0.88 for Z_2 and $22060\ \text{kPa}$ for P_{cr}.

$$T_2 = \frac{16529 \times 1.5 \times 647.1}{0.88 \times 22060}$$
$$= 826\ \text{K}$$

Therefore, the temperature of steam at exit is $\boxed{826\ \text{K}}$.

Step 7 of 7

(c)

Write the specific volume at state 1 corresponding to saturated temperature of $350°\text{C}$ from Table A-4: superheated steam table.

$$v_g = v_{g@350°C}$$
$$= 0.008806\ \text{m}^3/\text{kg}$$

Calculate the final specific volume.

$$v_2 = 2v_1$$

Substitute $0.008806\ \text{m}^3/\text{kg}$ for v_1.

$$v_2 = 2 \times 0.008806$$
$$= 0.017612\ \text{m}^3/\text{kg}$$

Write the temperature of the steam at exit from Table: A-6 corresponding to pressure $16529\ \text{kPa}$ and specific volume of $0.017612\ \text{m}^3/\text{kg}$.

$$T_{2@16529\ \text{kPa}, 0.017612\ \text{m}^3/\text{kg}} = 477°\text{C}$$
$$= 750\ \text{K}$$

Therefore, the temperature of steam at exit is $\boxed{750\ \text{K}}$.

Question: 4.094P pg 1

Saturated water vapor at 400°F is heated at constant pressure until its volume has doubled. Determine the final temperature using the ideal gas equation of state, the compressibility charts, and the steam tables.

Step 1 of 1

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Given

Initial temperature of saturated water vapor, $T_1 = 400\ ^oF = 859.67\ R$

Final volume, $V_2 = 2 * \text{initial volume} = 2V_1$

---

a) From the ideal gas equation, we have

$P_1v_1 = RT_1$ ---- (1)

and $P_2v_2 = RT_2$ ---- (2)

This is constant pressure process.

Therefore, $P_2 = P_1$

From equations (1) and (2), eliminating pressure, mass and gas constant terms, we can write

$$\frac{v_2}{v_1} = \frac{T_2}{T_1}$$

Therefore, $T_2 = 2 * T_1$ (since $v_2 = 2v_1$)

$T_2 = 2 * 859.67$
Final temperature, $T_2 = 1719.34\ R$

---

b) The gas constant, critical pressure and temperature of water are (from Table A-1E)

$R = 0.5956$ psia ft^3/lbm R

$P_{cr} = 3200$ psia

$T_{cr} = 1164.8\ R$

Initial pressure of the saturated vapor is the saturation pressure at the given temperature which can be obtained from saturated water-temperature table (Table A-4E).

Therefore, $P_1 = P_{sat\ @\ 4000\ ^oF} = 247.26$ psia.

The reduced pressure and temperature of initial state are

$$P_{R1} = \frac{P_1}{P_{cr}}$$

$$= \frac{247.26}{3200}$$

$= 0.0773$

$$T_{R1} = \frac{T_1}{T_{cr}}$$

$$= \frac{859.67}{1164.8}$$

$= 0.738$

From the compressibility chart (Figure A-15), corresponding to P_{R1} and T_{R1},

The compressibility factor, $Z_1 = 0.93$ (approximately)

Thus, initial specific volume (v_1) can be obtained from

$$v_1 = \frac{Z_1RT_1}{P_1}$$

$$= \frac{0.93 \times 0.5956 \times 859.67}{247.26}$$

$v_1 = 1.9258\ ft^3$/lbm

Thus, final specific volume, $v_2 = 2v_1 = 3.8516\ ft^3$/lbm.

Therefore pseudo-reduced specific volume of final state is

$$v_{R2} = \frac{v_{actual}}{RT_{cr}/P_{cr}}$$

$$= \frac{3.8516}{0.5956 \times 1164.8 / 3200}$$

$= 17.766$

At the final state, we have

Reduced pressure, $P_{R2} = P_{R1} = 0.0773$

Pseudo- reduced specific volume, $v_{R2} = 17.766$

From the compressibility chart (Figure A-15), corresponding to P_{R2} and v_{R2},

The compressibility factor, $Z_2 = 0.98$ (approximately)

Therefore, final temperature is

$$T_2 = \frac{P_2v}{Z_2R}$$

$$= \frac{247.26 \times 3.8516}{0.98 \times 0.5956}$$

$T_2 = 1631.60\ R$

$$= \frac{247.26 \times 3.8516}{0.98 \times 0.5956}$$

$T_2 = 1631.60$ R

---

c) Given that the initial state is saturated water vapor at 400 °F. Thus the pressure of the saturated water vapor will be the saturation pressure at the given temperature and specific volume will be the specific volume of saturated vapor at the given temperature which can be determined from saturated water-temperature table (Table A-4E)

Therefore,

Initial pressure, $P_1 = P_{sat\ @\ 400\ °C} = 247.26$ psia

Initial specific volume = $v_1 = v_g = 1.8639$ ft^3/lbm.

Thus final pressure, $P_2 = P_1 = 247.26$ psia.

Final specific volume, $v_2 = 2 * v_1 = 2 * 1.8639$

$v_2 = 3.7278$ ft^3/lbm

Thus, the final state is defined by pressure and specific volume. We do not know which table to use to determine the missing properties because we have no clue as to whether we have saturated mixture, compressed liquid or superheated vapor, which can be determined from the specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) from the saturated water – pressure table (Table A-5E).

Actually from table A-5E, we have

At $P_A = 200$ psia

$v_{fA} = 0.01839$ ft^3/lbm

$v_{gA} = 2.2882$ ft^3/lbm

At $P_B = 250$ psia

$v_{fB} = 0.01865$ ft^3/lbm

$v_{gB} = 1.8440$ ft^3/lbm

It is known that the values of specific volume of saturated liquid (v_f) and specific volume of saturated vapor (v_g) at P = 247.26 psia lie in the range of values of specific volume of above two pressures. We can see that the given value, v = 3.7278 ft^3/lbm is greater than the value v_g. Thus we have superheated vapor and the temperature of the superheated vapor at given pressure can be obtained from superheated water table (Table A-6E)

At P = 247.26 psia and v = 3.7278 ft^3/lbm
We get T = 1100 °F = 155.67 R (approximately)

Question: 4.095P

Methane at 8 MPa and 300 K is heated at constant pressure until its volume has increased by 50 percent. Determine the final temperature using the ideal gas equation of state and the compressibility factor. Which of these two results is more accurate?

Step 1 of 5

Let V_1 be the initial volume.

Given

Initial pressure of methane, $P_1 = 8\ MPa$

Initial temperature, $T_1 = 300\ K$

Final volume, $V_2 = 1.5\ V_1$ (50 % more than initial volume)

Step 2 of 5

a) From the ideal gas equation, we have

$P_1 v_1 = RT_1$ ------ (1)

and $P_2 v_2 = RT_2$ -------- (2)

This is constant pressure process.

Therefore, $P_2 = P_1$

From equations (1) and (2), eliminating pressure, mass and gas constant terms, we can write

$$\frac{v_2}{v_1} = \frac{T_2}{T_1}$$

Therefore, $T_2 = 1.5T_1$ (since $v_2 = 1.5v_1$)

$T_2 = 1.5 \times 300$

Final temperature, $\boxed{T_2 = 450\ K}$

Step 3 of 5

b) The gas constant, critical pressure and temperature of methane are (from Table A-1)

$R = 0.5182\ kJ/kg\ K$

$P_{cr} = 4.64\ MPa$

$T_{cr} = 191.1\ K$

The reduced pressure and temperature of initial state are

$$P_{R1} = \frac{P_1}{P_{cr}} = \frac{8}{4.64} = 1.724$$

$$T_{R1} = \frac{T_1}{T_{cr}} = \frac{300}{191.1} = 1.57$$

From the compressibility chart (Figure A-15), corresponding to P_{R1} and T_{R1},

The compressibility factor, $Z_1 = 0.88$ (approximately)

Thus, initial specific volume (v_1) can be obtained from

$$v_1 = \frac{Z_1 RT_1}{P_1} = \frac{0.88 \times 0.5182 \times 300}{8000}$$

$v_1 = 0.0171006\ m^3/kg$

Step 4 of 5

Thus, final specific volume, $v_2 = 1.5v_1 = 0.02565\ m^3/kg$.

Therefore pseudo-reduced specific volume of final state is

$$v_{R2} = \frac{v_{actual}}{R\frac{T_{cr}}{P_{cr}}} = \frac{0.02565}{0.5182 \times \frac{191.1}{4640}} = 1.2018$$

At the final state, we have

Reduced pressure, $P_{R2} = P_{R1} = 1.724$

Pseudo- reduced specific volume, $v_{R2} = 1.2018$

From the compressibility chart (Figure A-15), corresponding to P_{R2} and v_{R2},

The compressibility factor, $Z_2 = 0.97$ (approximately)

Therefore, final temperature is

$$T_2 = \frac{P_2 v}{Z_2 R} = \frac{8000 \times 0.02565}{0.97 \times 0.5182}$$

$\boxed{T_2 = 408.23\ K}$

Step 5 of 5

Of these two results, the second result will be more accurate as it takes into account the real behaviour of the gases. The accuracy of the second result is only affected by the error in reading the charts.

Question: 4.096P

What is the percentage of error involved in treating carbon dioxide at 5 MPa and 25°C as an ideal gas?

Step 1 of 4

Obtain the properties of CO_2 from table "gas constant and critical point properties".

Critical temperature $T_{cr} = 304.2\text{K}$

Critical pressure $P_{cr} = 7.39\text{MPa}$

Gas constant $R = 0.1889\text{kJ/kg}\cdot\text{K}$

State the ideal gas equation.

$$Pv_{ideal} = RT$$

Here, P is pressure of the gas, v is specific volume and T is temperature of the gas.

Substitute $5\text{MPa} = 5\times10^3\text{ kPa}$ for P and $25°\text{C} = (25+273)\text{K}$ for T to find the specific volume.

$$5\times10^3\times v_{ideal} = 0.1889\times(25+273)$$

$$v_{ideal} = 0.01125\text{m}^3/\text{kg}$$

Step 2 of 4

Find the reduced pressure by using the following equation.

$$P_R = \frac{P}{P_{cr}}$$

$$= \frac{5}{7.39}$$

$$= 0.68$$

Find the reduced temperature by using the following equation.

$$T_R = \frac{T}{T_{cr}}$$

$$= \frac{25+273}{304.2}$$

$$= 0.98$$

Step 3 of 4

Note down the compressibility factor value from the "compressibility chart" at $P_R = 0.68$ and $T_R = 0.98$.

$$Z = 0.7$$

Now find the specific volume by using the following equation:

$$Pv = ZRT$$

$$5\times10^3\times v = 0.7\times0.1889\times(25+273)$$

$$v = 0.0078\text{m}^3/\text{kg}$$

Step 4 of 4

Then find the error involved in treating CO_2 as an ideal gas by using the following expression:

$$\%\text{ error} = \frac{v_{ideal} - v}{v}\times100$$

$$= \frac{0.01125 - 0.0078}{0.0078}\times100$$

$$= 44.23\%$$

Therefore, the percentage of error involved in treating Carbon dioxide as an ideal gas is $\boxed{44.23\%}$.

Question: 4.097P

Carbon dioxide gas enters a pipe at 3 MPa and 500 K at a rate of 2 kg/s. CO2 is cooled at constant pressure as it flows in the pipe and the temperature of CO2 drops to 450 K at the exit. Determine the volume flow rate and the density of carbon dioxide at the inlet and the volume flow rate at the exit of the pipe using (*a*) the ideal-gas equation and (*b*) the generalized compressibility chart. Also, determine (*c*) the error involved in the first case.

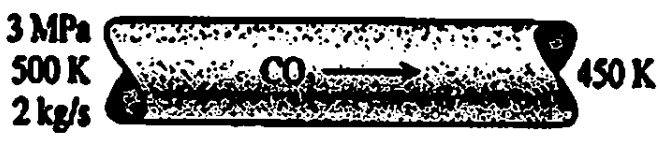


FIGURE P4–97

Step 1 of 3

(a)

From the properties table of carbon dioxide, the gas constant (*R*) is 0.1889 kJ/kg-K. Now find the density at inlet substituting 500 K for T_1 and 3000 kPa for P_1

$$P_1 = \rho_1 R T_1$$

$$\rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{3000}{0.1889 \times 500}$$

$$= 31.763 \text{ kg/m}^3$$

Thus the density of carbon dioxide at inlet is $\boxed{31.763 \text{ kg/m}^3}$

Determine the ideal volume flow rate by using the density of carbon dioxide at inlet and mass flow rate $(\dot{m})$ of 2 kg/s.

$$\dot{V}_1 = \frac{\dot{m}}{\rho_1}$$

$$= \frac{2}{31.763}$$

$$= 0.0629 \text{ m}^3/\text{s}$$

Thus the ideal volume flow rate of carbon dioxide at inlet is $\boxed{0.0629 \text{ m}^3/\text{s}}$

Determine the ideal volume flow rate at exit by applying the ideal gas relation to carbon dioxide whose exit temperature (T_2) is 450 K and exit pressure (P_2) is 3000 kPa.

$$\dot{V}_2 = \frac{\dot{m} R T_2}{P_2}$$

$$= \frac{2 \times 0.1889 \times 450}{3000}$$

$$= 0.05667 \text{ m}^3/\text{s}$$

Thus the ideal volume flow rate of carbon dioxide at exit is $\boxed{0.05667 \text{ m}^3/\text{s}}$

Step 2 of 3

(b)

From the critical properties table, for carbon dioxide the critical pressure (P_{cr}) value is 7390 kPa and critical temperature (T_{cr}) is 304.2 K

The reduced properties (reduced temperature and reduced pressure) at inlet are

$$T_{R1} = \frac{T_1}{T_{cr}}$$

$$= \frac{500}{304.2}$$

$$= 1.644$$

$$P_{R1} = \frac{P_1}{P_{cr}}$$

$$= \frac{3000}{7390}$$

$$= 0.406$$

From the compressibility chart, the factor (Z_1) corresponding to the reduced temperature and pressure is 0.97

So determine the actual volume flow rate using the compressibility factor.

$$\dot{V}_1 = Z_1 \left(\frac{\dot{m} R T_1}{P_1} \right)$$

$$= 0.97 \left(\frac{2 \times 0.1889 \times 500}{3000} \right)$$

$$= 0.061077 \text{ m}^3/\text{s}$$

Thus the actual volume flow rate of carbon dioxide at inlet is $\boxed{0.061077 \text{ m}^3/\text{s}}$

Step 3 of 3

Find the actual density at inlet using the mass flow rate and actual inlet volume flow rate

$$\rho_1 = \frac{\dot{m}}{\dot{V}_1}$$

$$= \frac{2}{0.061077}$$

$$= 32.745 \text{ kg/m}^3$$

Thus the actual density of carbon dioxide at inlet is $\boxed{32.745 \text{ kg/m}^3}$

The reduced properties (reduced temperature and reduced pressure) at exit are

$$T_{R2} = \frac{T_2}{T_{cr}}$$

$$= \frac{450}{304.2}$$

$$= 1.48$$

$$P_{R2} = \frac{P_2}{P_{cr}}$$

$$= \frac{3000}{7390}$$

$$= 0.406$$

From the compressibility chart, the factor (Z_1) corresponding to the reduced temperature and pressure is 0.9656

So determine the actual volume flow rate using the compressibility factor.

$$\dot{V}_2 = Z_2 \left(\frac{\dot{m} R T_2}{P_2} \right)$$

$$= 0.9656 \left(\frac{2 \times 0.1889 \times 450}{3000} \right)$$

$$= 0.05472 \text{ m}^3/\text{s}$$

Thus the actual volume flow rate of carbon dioxide at exit is $\boxed{0.05472 \text{ m}^3/\text{s}}$

Question: 4.098P

Determine the specific volume of nitrogen gas at 10 MPa and 150 K based on (*a*) the idea-gas equation and (*b*) the generalized compressibility chart. Compare these results with the experimental value of 0.002388 m3/kg, and determine the error involved in each case.

Step 1 of 5

Given

Pressure, $P = 10\ MPa$

Temperature, $T = 150\ K = -123\ °C$

Step 2 of 5

a) Assuming, the nitrogen behaves as an ideal gas at specified conditions.

For nitrogen, gas constant $R = 0.2968\ kJ/kg\ K$

From the ideal gas equation, specific volume is given by

$$v = \frac{RT}{P}$$

$$= \frac{0.2968 \times 150}{10 \times 1000}$$

$$\boxed{v = 0.004452\ m^3/kg}$$

Step 3 of 5

b) The critical pressure and temperature of nitrogen are (from Table-1)

$P_{cr} = 3.39\ MPa$

$T_{cr} = 126.2\ K$

Therefore,

$$\text{Reduced pressure, } P_r = \frac{P}{P_{cr}}$$

$$= \frac{10}{3.39}$$

$$P_r = 2.95$$

$$\text{Reduced temperature, } T_r = \frac{T}{T_{cr}}$$

$$= \frac{150}{126.2}$$

$$T_r = 1.188$$

Step 4 of 5

From compressibility chart (A-15), corresponding to the obtained P_r and T_r values, we have

$Z = 0.54$

Therefore, actual specific volume is

$$v_{actual} = Z \times v_{ideal}$$

$$= 0.54 \times 0.004452$$

$$v_{actual} = 0.002403\ m^3/kg$$

Step 5 of 5

c) Given the experimental value is, $v = 0.002388\ m^3/kg$

The error involved in the first case is,

$$\text{Error} = \frac{0.004452 - 0.002388}{0.002388} \times 100 = 86.43\%$$

The error involved in the second case is,

$$\text{Error} = \frac{0.002403 - 0.002388}{0.002388} \times 100 = 0.628\%$$

Question: 4.099P

A rigid tank contains nitrogen gas at 227°C and 100 kPa gage. The gas is heated until the gage pressure reads 250 kPa. If the atmospheric pressure is 100 kPa, determine the final temperature of the gas, in °C.

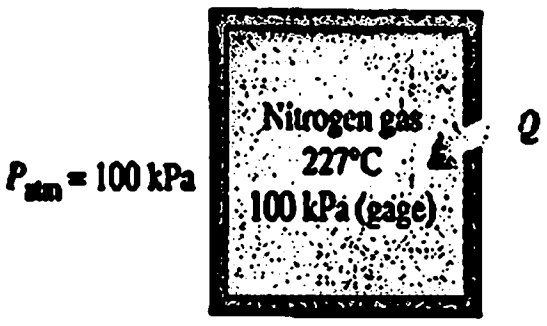


FIGURE P4–99

Step 1 of 3

Given

Initial temperature of nitrogen $T_1 = 227^\circ C$

Question:

A rigid tank contains nitrogen gas at 227°C and 100 kPa gage. The gas is heated until the gage pressure reads 250 kPa. If the atmospheric pressure is 100 kPa, determine the final temperature of the gas, in °C.

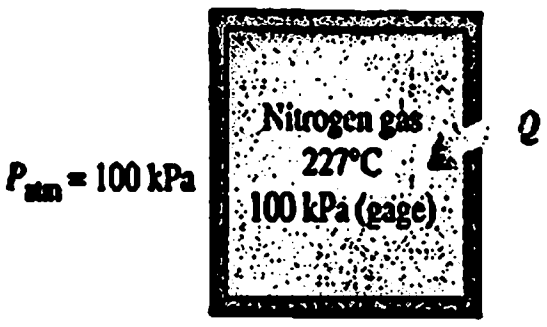


FIGURE P4–99

Step 1 of 3

Given

Initial temperature of nitrogen $T_1 = 227^\circ C$

$t_1 \quad t_2$

$$T_2 = \frac{P_2 T_1}{P_1}$$

$$= \frac{350 \times 500}{200}$$

$$T_2 = 875K$$

$$T_2 = 602^\circ C$$

Therefore final temperature of gas

$$\boxed{T_2 = 602^\circ C}$$

$t_1 \quad t_2$

$$T_2 = \frac{P_2 T_1}{P_1}$$

$$= \frac{350 \times 500}{200}$$

$$T_2 = 875K$$

$$T_2 = 602^\circ C$$

Therefore final temperature of gas

$$\boxed{T_2 = 602^\circ C}$$

Question: 4.100P

Carbon-dioxide gas at 3 MPa and 500 K flows steadily in a pipe at a rate of 0.4 kmol/s. Determine (*a*) the volume and mass flow rates and the density of carbon dioxide at this state. If CO2 is cooled at constant pressure as it flows in the pipe so that the temperature of CO2 drops to 450 K at the exit of the pipe, determine (*b*) the volume flow rate at the exit of the pipe.



FIGURE P4–100

Step 1 of 4

Find the volume flow rate of carbon dioxide at the entrance of the pipe by using the following equation:

$$p_1\dot{v}_1 = \dot{n}R_uT_1$$

$$\dot{v}_1 = \frac{\dot{n}R_uT_1}{p_1}$$

Question:

Carbon-dioxide gas at 3 MPa and 500 K flows steadily in a pipe at a rate of 0.4 kmol/s. Determine (*a*) the volume and mass flow rates and the density of carbon dioxide at this state. If CO2 is cooled at constant pressure as it flows in the pipe so that the temperature of CO2 drops to 450 K at the exit of the pipe, determine (*b*) the volume flow rate at the exit of the pipe.

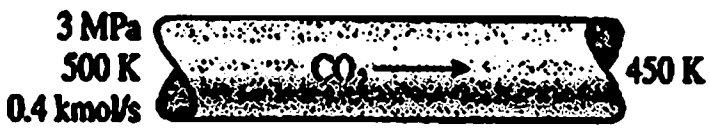


FIGURE P4–100

Step 1 of 4

Find the volume flow rate of carbon dioxide at the entrance of the pipe by using the following equation:

$$p_1\dot{v}_1 = \dot{n}R_uT_1$$

$$\dot{v}_1 = \frac{\dot{n}R_uT_1}{p_1}$$

$$= \frac{0.1889\times500}{3}$$

$$= 17.596\,\text{kg/s}$$

Therefore, mass flow rate of carbon dioxide at the entrance of the pipe is $\boxed{17.596\,\text{kg/s}}$

Step 3 of 4

Find the density of carbon dioxide at the entrance of the pipe by using the following equation:

$$p_1 = \rho_1 RT_1$$

$$\rho_1 = \frac{p_1}{RT_1}$$

Substitute 3 MPa for p_1, 500 K for T_1, and $0.1889\,\text{kJ/kg.K}$ for R.

$$\rho_1 = \frac{3\times1000}{0.1889\times500}$$

$$= 31.76\,\text{kg/m}^3$$

Therefore, the density of carbon dioxide at the entrance of the pipe is $\boxed{31.76\,\text{kg/m}^3}$.

Step 4 of 4

Find the volume flow rate of carbon dioxide at the exit of the pipe by using the following equation:

$$\dot{v}_1 = \frac{0.1889\times500}{3}$$

$$= 17.596\,\text{kg/s}$$

Therefore, mass flow rate of carbon dioxide at the entrance of the pipe is $\boxed{17.596\,\text{kg/s}}$

Step 3 of 4

Find the density of carbon dioxide at the entrance of the pipe by using the following equation:

$$p_1 = \rho_1 RT_1$$

$$\rho_1 = \frac{p_1}{RT_1}$$

Substitute 3 MPa for p_1, 500 K for T_1, and $0.1889\,\text{kJ/kg.K}$ for R.

$$\rho_1 = \frac{3\times1000}{0.1889\times500}$$

$$= 31.76\,\text{kg/m}^3$$

Therefore, the density of carbon dioxide at the entrance of the pipe is $\boxed{31.76\,\text{kg/m}^3}$.

Question:

4.101P

The combustion in a gasoline engine may be approximated by a constant volume heat addition process. There exists the air–fuel mixture in the cylinder before the combustion and the combustion gases after it, and both may be approximated as air, an ideal gas. In a gasoline engine, the cylinder conditions are 1.4 MPa and 450°C before the combustion and 1600°C after it. Determine the pressure at the end of the combustion process.

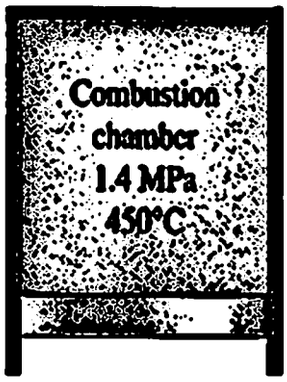


FIGURE P4–101

Step 1 of 2

Given

Pressure before combustion $P_1 = 1.4\,\text{MPa}$

Temperature before combustion $T_1 = 450^0\text{C}$

$$T_1 = 450 + 273$$

$$T_1 = 723\,\text{K}$$

Temperature after combustion $T_2 = 1600^0\text{C}$

$$T_2 = 1600 + 273$$

$$T_2 = 1873\,\text{K}$$

Step 2 of 2

Assume air fuel mixture is an ideal gas

Since the Process is constant volume

Applying the ideal gas equation

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

$$P_1 T_2 = P_2 T_1$$

$$P_2 = \frac{P_1 T_2}{T_1}$$

$$= \frac{1.4 \times 1873}{723}$$

$$\boxed{P_2 = 3.63\,\text{MPa}}$$

Question: 4.102P

Combustion in a diesel engine may be modeled as a constant-pressure heat addition process with air in the cylinder before and after combustion. Consider a diesel engine with cylinder conditions of 950 K and 75 cm3 before combustion, and 150 cm3 after it. The engine operates with an air-fuel ratio of 22 kg air/kg fuel (the mass of the air divided by the mass of the fuel). Determine the temperature after the combustion process.

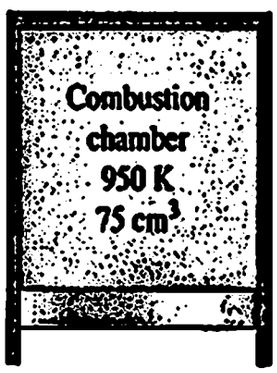


FIGURE P4–102

Step 1 of 2

Given data:

Temperature of the air before combustion, $T_1 = 950\text{K}$

Volume of the air before combustion, $V_1 = 75\,\text{cm}^3$

Volume of the air after the combustion, $V_2 = 150\,\text{cm}^3$

Air fuel ratio, $\frac{m_a}{m_f} = 22\,\text{kg air/kg fuel}$

Step 2 of 2

The ratio of initial and final mass of the air,

$$\frac{m_1}{m_2} = \frac{AF}{AF+1}$$
$$= \frac{22}{22+1}$$
$$= 0.956$$

From the ideal gas equation, $PV = mRT$

$$\frac{T_2}{T_1} = \left(\frac{m_1}{m_2}\right)\left(\frac{V_2}{V_1}\right)$$
$$T_2 = \left(\frac{m_1}{m_2}\right)\left(\frac{V_2}{V_1}\right)T_1$$
$$= 0.956\left(\frac{150}{75}\right)(950)$$
$$= 1816.4\text{K}$$

Therefore, the temperature of the air after the combustion process is $\boxed{T_2 = 1816.4\text{K}}$

Question: 4.103P

One kilogram of R-134a fills a 0.090 m3 rigid container at an initial temperature of −40°C. The container is then heated until the pressure is 280 kPa. Determine the initial pressure and final temperature.

Step 1 of 3

Given

Mass of R-134a $m = 1\text{kg}$

Volume of the rigid container $V = 0.090\text{m}^3$

Initial temperature $T_1 = -40^0\text{C}$

Final pressure $P_2 = 280\text{kPa}$

Step 2 of 3

Specific volume

$v_1 = V / m$

$v_1 = 0.090 / 1$

$v_1 = 0.090\text{m}^3/\text{kg}$

At $T_1 = -40^0\text{C}$ from saturated R-134a tables

$v_f < v_1 < v_g$

Therefore the state is in liquid and vapor mixture

Then initial pressure will be the saturation pressure at temperature $T_1 = -40^0\text{C}$ from saturated R-134a tables

$P_1 = P_{sat} = 51.25\text{kPa}$

$\boxed{P_1 = 51.25\text{kPa}}$

Step 3 of 3

Since the process is constant volume

At $P_2 = 280\text{kPa}$ and $v_2 = v_1 = 0.090\text{m}^3/\text{kg}$ from superheated R-134a tables

$T_2 = 50°\text{C}$

Therefore the final temperature is $\boxed{T_2 = 50°\text{C}}$

Question: 4.104P

One pound-mass of water fills a 2.649 ft3 weighted piston-cylinder device at a temperature of 400°F. The piston-cylinder device is now cooled until its temperature is 100°F. Determine the final pressure and volume of the water.

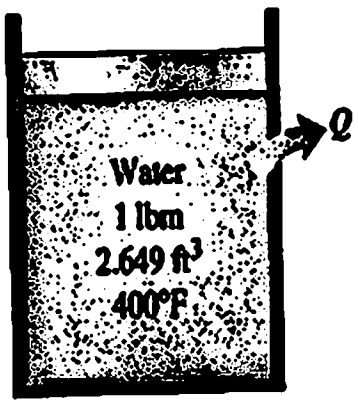


FIGURE P4–104E

Step 1 of 2

Calculate the Initial specific volume of the water,

$$v_1 = \frac{V}{m}$$

Here,

Volume of the water the device, V

Mass of the water, m

Substitute $2.649\,\text{ft}^3$ for V , $1\,\text{lb}$ for m

$$\begin{aligned} v_1 &= \frac{V}{m} \\ &= \frac{2.649}{1} \\ &= 2.649\,\text{ft}^3/\text{lbm} \end{aligned}$$

From the properties of the superheated water tables,

At $T_1 = 400°\text{F}$ and specific volume $v_1 = 2.649\,\text{ft}^3/\text{lbm}$

Pressure $P_1 = 180\,\text{psia}$

Since, the cooling process in piston cylinder device is constant pressure process,

The final pressure of the water in the piston cylinder device, $\boxed{180\,\text{psia}}$

Step 2 of 2

From the saturated water –temperature table

Specific volume of the water at $T_2 = 100°\text{F}$,

$$v_2 = 0.01613\,\text{ft}^3/\text{lbm}$$

Here,

Specific volume of the water is v_2

Calculate the Final volume of the water in the device,

$$V_2 = mv_2$$

Substitute $0.01613\,\text{ft}^3/\text{lbm}$ for v_2

$$\begin{aligned} V_2 &= mv_2 \\ &= (1)(0.01613) \\ &= 0.01613\,\text{ft}^3 \end{aligned}$$

Therefore, volume of the water after the cooling process, $\boxed{0.01613\,\text{ft}^3}$

Question: 4.105P

Ethane at 10 MPa and 100°C is heated at constant pressure until its volume has increased by 60 percent. Determine the final temperature using (*a*) the ideal gas equation of state and (*b*) the compressibility factor. Which of these two results is the more accurate?

Step 1 of 5

Given data:

Initial temperature of ethane $T_1 = 100°C$

Initial pressure of the nitrogen $P_1 = 10MPa$

Step 2 of 5

Let the initial volume of the gas V_1

Increased volume after the heating process

$$V_2 = V_1 + 0.6V_1$$
$$= 1.6V_1$$

(a) By the ideal gas equation $PV = mRT$

For the constant pressure process

$$\frac{RT_1}{V_1} = \frac{RT_2}{V_2}$$
$$T_2 = \frac{T_1V_2}{V_1}$$
$$= \frac{(100+273)(1.6V_1)}{V_1}$$
$$= 596.8K$$

Therefore, final temperature of the ethane gas, $\boxed{T_2 = 596.8K}$

Step 3 of 5

(b) Using the compressibility factor:

From the properties of the ethane,

Critical temperature $T_{cr} = 305.5K$

Critical pressure $P_{cr} = 4.48MPa$

critical volume $V_{cr} = 0.148 m^3/kmol$

Reduced pressure

$$P_R = \frac{P_1}{P_{cr}}$$
$$= \frac{10}{4.48}$$
$$= 2.23$$

Reduced temperature

$$T_R = \frac{T_1}{T_{cr}}$$
$$= \frac{100}{305.5}$$
$$= 0.327$$

Step 4 of 5

From the Nelson- Obert compressibility chart at $P_{R1} = 2.23$ and $T_{R1} = 0.327$

Compressibility factor $Z_1 = 0.6$

At $P_{R1} = 2.23$ and $Z_1 = 0.6$

Reduced volume $V_{R1} = 0.351$

For the final conditions,

Reduced pressure $P_{R1} = P_{R2} = 2.23$

Reduced volume

$$V_{R2} = 1.6V_{R1}$$
$$= 1.6(0.351)$$
$$= 0.562$$

At $P_{R2} = 2.23$ and $V_{R2} = 0.562$

Compressibility factor $Z_2 = 0.83$

Step 5 of 5

Final temperature of the ethane,

$$T_2 = \frac{P_2V_2}{Z_2R}$$

Pseudo- reduced specific volume

$$V_R = \frac{V_{actual}}{(RT_{cr}/P_{cr})}$$

Therefore,

$$T_2 = \frac{P_2V_2}{Z_2R}$$
$$= \left(\frac{P_2}{Z_2R}\right)\left(\frac{V_{R2}RT_{cr}}{P_{cr}}\right)$$
$$= \left(\frac{P_2}{Z_2}\right)\left(\frac{V_{R2}T_{cr}}{P_{cr}}\right)$$
$$= \left(\frac{P_1}{Z_2}\right)\left(\frac{V_{R2}T_{cr}}{P_{cr}}\right) \qquad (\text{since } P_2 = P_1)$$
$$= \left(\frac{10MPa}{0.83}\right)\left(\frac{0.562\times305.5K}{4.48MPa}\right)$$
$$= 461.733K$$

Therefore, the final temperature of the ethane, $\boxed{T_2 = 461.733K}$

Extracting the error associated with the chart reading, temperature found by the compressibility chart is more accurate.

Question:

4.106P

A 20-m3 tank contains nitrogen at 23°C and 600 kPa. Some nitrogen is allowed to escape until the pressure in the tank drops to 400 kPa. If the temperature at this point is 20°C, determine the amount of nitrogen that has escaped.

Step 1 of 3

Given data:

Volume of the tank $V = 20\,\text{m}^3$

Initial temperature of nitrogen $T_1 = 23°\text{C}$

Initial pressure of the nitrogen $P_1 = 600\,\text{kPa}$

Final pressure of the nitrogen $P_2 = 400\,\text{kPa}$

Final temperature of the nitrogen $T_2 = 20°\text{C}$

Step 2 of 3

From the ideal gas equation

$PV = mRT$

For the initial conditions $T_1 = 23°\text{C}$ and $P_1 = 600\,\text{kPa}$

$$P_1V_1 = m_1RT_1$$

$$m_1 = \frac{P_1V_1}{RT_1}$$

$$= \frac{600\,\text{kPa} \times 20\,\text{m}^3}{\left(\frac{8.314}{28}\right)\frac{\text{kPa}\cdot\text{m}^3}{\text{kg}\cdot\text{K}} \times (23+273)\,\text{K}}$$

$$= 136.533\,\text{kg}$$

Step 3 of 3

For the final condition $T_2 = 20°\text{C}$ and $P_2 = 400\,\text{kPa}$

$$P_2V_2 = m_2RT_2$$

$$m_2 = \frac{P_2V_2}{RT_2}$$

$$= \frac{400\,\text{kPa} \times 20\,\text{m}^3}{\left(\frac{8.314}{28}\right)\frac{\text{kPa}\cdot\text{m}^3}{\text{kg}\cdot\text{K}} \times (20+273)\,\text{K}}$$

$$= 91.954\,\text{kg}$$

By the mass balance equation

Amount of nitrogen escaped from the tank,

$$\Delta m = m_1 - m_2$$

$$= 136.533 - 91.954$$

$$= 44.58\,\text{kg}$$

Therefore, the amount of nitrogen escaped from the tank, $\boxed{\Delta m = 44.58\,\text{kg}}$

Question: 4.107P

A 4-L rigid tank contains 2 kg of saturated liquid–vapor mixture of water at 50°C. The water is now slowly heated until it exists in a single phase. At the final state, will the water be in the liquid phase or the vapor phase? What would your answer be if the volume of the tank were 400 L instead of 4 L?

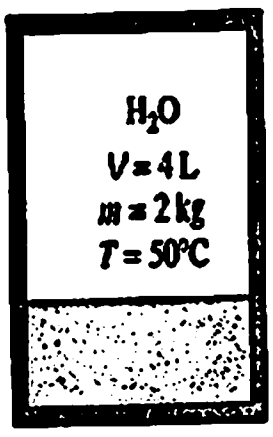


FIGURE P4–107

Step 1 of 3

The water is heated in rigid tank and the volume is constant after the heating process.

Therefore, the heating process is a constant volume process ($v = V/m = \text{constant}$).

Thus the final specific volume of mixture, v_f , is equal to initial specific volume v_i .

$$v_f = v_i = v$$

Obtain the critical specific volume of water from table A-1, "The critical-point properties table".

$$\begin{aligned} v_{cr} &= 0.0560\,\text{m}^3/\text{kmol} \\ &= 0.0560\,\text{m}^3/\text{kmol} \times \frac{1}{18.015\,\text{kg/kmol}} \\ &= 0.0031\,\text{m}^3/\text{kg} \end{aligned}$$

Step 2 of 3

If the final specific volume of water is less than the critical specific volume (v_{cr}), then the water is in liquid state, otherwise the water is in vapor state.

Find the specific volume of the $4\,\text{L}$ tank volume.

$$v = \frac{V}{m}$$

Here, m is mass of liquid-vapor mixture of water, V is the total volume.

Substitute 2 kg for m and $4\,\text{L}$ for V.

$$\begin{aligned} v &= \frac{4\,\text{L} \times \dfrac{10^{-3}\,\text{m}^3}{\text{L}}}{2\,\text{kg}} \\ &= \frac{0.004}{2} \\ &= 0.002\,\text{m}^3/\text{kg} \\ &< v_{cr} \end{aligned}$$

Therefore, the water will exist in liquid state.

Step 3 of 3

If the rigid tank volume is considered as $400\,\text{L}$, then calculate the specific volume of tank by using the following equation:

$$v = \frac{V}{m}$$

Substitute 2 kg for m and $400\,\text{L}$ for V.

$$\begin{aligned} v &= \frac{400\,\text{L} \times \dfrac{10^{-3}\,\text{m}^3}{\text{L}}}{2\,\text{kg}} \\ &= \frac{0.4}{2} \\ &= 0.2\,\text{m}^3/\text{kg} > v_{cr} \end{aligned}$$

Therefore, the water will exist in vapor state.

Question:

4.108P

The gage pressure of an automobile tire is measured to be 200 kPa before a trip and 220 kPa after the trip at a location where the atmospheric pressure is 90 kPa. Assuming the volume of the tire remains constant at 0.035 m3, determine the percent increase in the absolute temperature of the air in the tire.

Step 1 of 3

Given data:

The gage pressure of an automobile tire before trip, $P_{gauge,1} = 200\,\text{kPa}$

The gage pressure of an automobile tire after trip, $P_{gauge,2} = 220\,\text{kPa}$

The atmospheric pressure, $P_{atm} = 90\,\text{kPa}$

The constant volume of tire, $V_1 = V_2 = 0.035\,\text{m}^3$

Step 2 of 3

The absolute pressure of an automobile tire before trip,

$$P_1 = P_{gauge,1} + P_{atm}$$
$$= 200 + 90$$
$$= 290\,\text{kPa}$$

The absolute pressure of an automobile tire after trip,

$$P_2 = P_{gauge,2} + P_{atm}$$
$$= 220 + 90$$
$$= 310\,\text{kPa}$$

Step 3 of 3

By using ideal gas law, we get $\frac{PV}{T} = \text{constant}$

Therefore we can write,

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\frac{T_2}{T_1} = \frac{P_2V_2}{P_1V_1}$$

$$\frac{T_2}{T_1} = \frac{P_2}{P_1} \quad (\text{since}, V_1 = V_2)$$

$$= \frac{310}{290}$$
$$= 1.069$$

$$T_2 = 1.069T_1$$

Therefore, the percentage increase in absolute temperature of the air in the tire is 6.9.

Question: 4.109P

Steam at 400°C has a specific volume of 0.02 m3/kg. Determine the pressure of the steam based on (*a*) the ideal-gas equation, (*b*) the generalized compressibility chart, and (*c*) the steam tables.

Step 1 of 6

Given data:

The temperature of steam,

$T = 400\,^{\circ}\mathrm{C}$

$= 400+273$

$= 673\mathrm{K}$

The specific volume of steam, $v = 0.02\,\mathrm{m}^3/\mathrm{kg}$

Step 2 of 6

From the property tables, the properties of water:

Gas constant, $R = 0.4615\,\mathrm{kJ/kg\cdot K}$

Critical temperature, $T_c = 647.1\mathrm{K}$

Critical pressure, $P_c = 22.06\,\mathrm{MPa}=22060\,\mathrm{kPa}$

Step 3 of 6

(a)

By using ideal gas equation, we get

$$P = \frac{RT}{v}$$

$$= \frac{0.4615\times 673.1}{0.02}$$

$$= 15531.782\,\mathrm{kPa}$$

The pressure of steam, $\boxed{P = 15531.782\,\mathrm{kPa}}$.

Step 4 of 6

(b)

From the compressibility chart, we have

Reduced temperature,

$$T_R = \frac{T}{T_c}$$

$$= \frac{673.1}{647.1}$$

$$= 1.0401$$

Pseudo reduced volume,

$$v_R = \frac{v_{actual}}{RT_c/P_c}$$

$$= \frac{0.02}{(0.4615\times 647.1)/22060}$$

$$= 1.477$$

Step 5 of 6

From compressibility chart at reduced pressure and pseudo reduced volume, the value of reduced pressure, $P_R = 0.57$

By using formula for pressure of steam, we get

$$P = P_R P_c$$

$$= 0.57\times 22060$$

$$= 12574.2\,\mathrm{kPa}$$

Therefore, the pressure of steam by using compressibility chart, $\boxed{P = 12574.2\,\mathrm{kPa}}$.

Step 6 of 6

(c)

From super heated water tables,

At temperature $400\,^{\circ}\mathrm{C}$ and volume $0.020030\,\mathrm{m}^3/\mathrm{kg}$, the value of pressure is $12.5\mathrm{MPa}$

At temperature $400\,^{\circ}\mathrm{C}$ and volume $0.015671\,\mathrm{m}^3/\mathrm{kg}$, the value of pressure is $15.0\mathrm{MPa}$

By using interpolation, at temperature $400\,^{\circ}\mathrm{C}$ and volume $0.02\,\mathrm{m}^3/\mathrm{kg}$, the value of pressure P is calculated as follows

$$\frac{0.02-0.020030}{0.015671-0.020030} = \frac{P-12.5}{15.0-12.5}$$

$$6.882\times 10^{-3} = \frac{P-12.5}{2.5}$$

$$0.017205 = P-12.5$$

$$P = 0.017205+12.5$$

$$= 12.517\mathrm{MPa}$$

Therefore, the pressure of steam by using super heated water tables, $\boxed{P = 12.517\mathrm{MPa}}$.

Question: 4.110P

A tank whose volume is unknown is divided into two parts by a partition. One side of the tank contains 0.03 m3 of refrigerant-134a that is a saturated liquid at 1.2 MPa, while the other side is evacuated. The partition is now removed, and the refrigerant fills the entire tank. If the final state of the refrigerant is 30°C and 400 kPa, determine the volume of the tank.

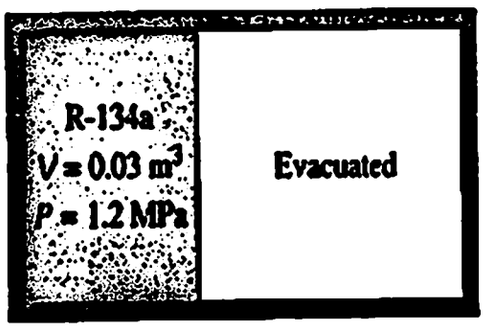


FIGURE P4–110

Step 1 of 3

Given

Volume of the refrigerant occupied in the tank $V_{ref} = 0.03\text{m}^3$

Refrigerant occupied as saturated liquid at pressure $P_1 = 1.2\text{MPa}$

Final state of the refrigerant at temperature $T_2 = 30^0\text{C}$

Final state of the refrigerant at pressure $P_2 = 400\text{kPa}$

Step 2 of 3

At $P_1 = 1.2\text{MPa}$ from R-134a saturated tables

$v_1 = v_f = 0.0008934\,\text{m}^3/\text{kg}$

The mass of the refrigerant contained in the tank is

$$m = \frac{V_1}{v_1}$$

$$= \frac{0.03}{0.0008934}$$

$$m = 33.58\text{kg}$$

Step 3 of 3

At the final state $P_2 = 400\text{kPa}$ and $T_2 = 30^0\text{C}$ from superheated R-134a tables

$v_2 = 0.056796\text{m}^3/\text{kg}$

Since mass of R-134a remains constant

$V_{tank} = V_2 = mV_2$

$V_{tank} = (33.58 \times 0.056796)$

$$\boxed{V_{tank} = 1.91\text{m}^3}$$

Question: 4.111P

 Reconsider Prob. 4–110. Using EES (or other) software, investigate the effect of the initial pressure of refrigerant-134a on the volume of the tank. Let the initial pressure vary from 0.5 to 1.5 MPa. Plot the volume of the tank versus the initial pressure, and discuss the results.

Step 1 of 1

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The problem is solved using EES, and the solution is given below.

"Given Data"

$x_1 = 0.0$

$Vol_1 = 0.03[m^3]$

$P_1 = 1200[kPa]$

$T_2 = 30[C]$

$P_2 = 400[kPa]$

"Solution"

$v_1 = volume(R134a, P = P_1, x = x_1)$

$Vol_1 = m \times v_1$

$v_2 = volume(R134a, P = P_2, T_2)$

$Vol_2 = m \times v_2$

P_1	V_2	m
500	2.1132	37.21
600	2.08	36.6
700	2.043	36
800	2.016	35.49
900	1.985	34.95
1000	1.96	34.47
1100	1.931	34
1200	1.91	33.58
1300	1.884	33.17
1400	1.86	32.73
1500	1.835	32.3

Question: 4.112P

Liquid propane is commonly used as a fuel for heating homes, powering vehicles such as forklifts, and filling portable picnic tanks. Consider a propane tank that initially contains 5 L of liquid propane at the environment temperature of 20°C. If a hole develops in the connecting tube of a propane tank and the propane starts to leak out, determine the temperature of propane when the pressure in the tank drops to 1 atm. Also, determine the total amount of heat transfer from the environment to the tank to vaporize the entire propane in the tank.

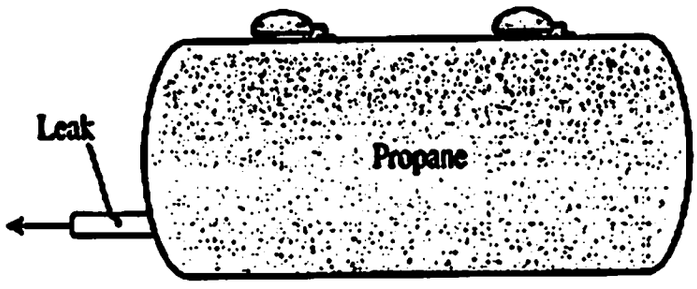


FIGURE P4–112

Step 1 of 3

Given data:

The volume of the propane tank, $V = 5\text{L}$

The atmospheric temperature, $T_{atm} = 20\,^{\circ}\text{C}$

The pressure after leak out, $P = 1\text{atm}$

Step 2 of 3

From the properties of common liquids, solids, and foods table,

The properties of propane at 1atm pressure:

Density, $\rho = 581\text{kg}/\text{m}^3$

Latent heat of vaporization, $h_{fg} = 427.8\text{kJ/kg}$

Saturation temperature, $T_{sat} = -42.1\,^{\circ}\text{C}$

Therefore, the temperature of propane when the pressure drops to 1atm, $\boxed{T = T_{sat} = -42.1\,^{\circ}\text{C}}$.

Step 3 of 3

The initial mass of liquid propane,

$$m = \rho V$$
$$= 581 \times 0.005$$
$$= 2.905\text{kg}$$

By using formula for heat transfer, we get

$$Q_{absorbed} = mh_{fg}$$
$$= 2.905 \times 427.8$$
$$= 1242.759\text{kJ}$$

Therefore, the heat absorbed by the propane to vaporize completely,

$\boxed{Q_{absorbed} = 1242.759\text{kJ}}$.

Question: 4.113P

Repeat Prob. 4–112 for isobutane.

Step 1 of 3

From the table of Isobutene

At pressure $P = 1\text{atm}$

Saturation temperature is $T = -11.70°\text{C}$

Since temperature of isobutene at pressure drops 1 atm is simply equals to saturation temperature corresponding to pressure of 1 atm.

Hence, temperature of the isobutene when pressure drops to 1 atm is $\boxed{-11.70°\text{C}}$

Step 2 of 3

From the table of Isobutene

At pressure $P = 1\text{atm}$

Density is $\rho = 593.80\text{kg/m}^3$

Latent heat of vaporization is $h_{fg} = 367.10\text{kJ/kg}$

Find the initially of the isobutene is

$$m = \rho V$$

Here,

Mass of the isobutene is m

Density of the isobutene is ρ

Volume of the tank is V

Substitute 593.80kg/m^3 for ρ, 0.005m^3 for V

$$\begin{aligned} m &= \rho V \\ &= 593.80 \times 0.005 \\ &= 2.969\text{kg} \end{aligned}$$

Step 3 of 3

Calculate the amount of heat transfer is

$$Q = mh_{fg}$$

Substitute 367.10kJ/kg for h_{fg}, 2.969kg for m

$$\begin{aligned} Q &= mh_{fg} \\ &= 2.969 \times 367.01 \\ &= 1089.65\text{kJ} \end{aligned}$$

Hence, the amount of heat transfer is $\boxed{1089.65\text{kJ}}$

Question:

4.114P

A tank contains helium at 77°C and 110 kPa gage. The helium is heated in a process by heat transfer from the surroundings such that the helium reaches a final equilibrium state at 300°C. Determine the final gage pressure of the helium. Assume atmospheric pressure is 100 kPa.

Step 1 of 4

Given

Initial temperature $T_1 = 77^0\,C = 350K$

Initial gage pressure $P_1 = 110kPa$

Final temperature $T_2 = 300^0\,C = 573K$

Atmospheric pressure $P_{atm} = 100kPa$

Step 2 of 4

Initial absolute pressure

$$P_1 = P_1(gage) + P_{atm}$$

$$P_1 = 110 + 100$$

$$P_1 = 210kPa$$

Step 3 of 4

Since volume remains constant by using ideal gas equation

$$P_2 = \frac{P_1 T_2}{T_1}$$

$$= \frac{210 \times 573}{350}$$

$$P_2 = 343.8kPa$$

Step 4 of 4

Then the final gauge pressure becomes

$$P_{gauge,2} = P_2 - P_{atm}$$

$$= 343.8 - 100$$

$$\boxed{P_{gauge,2} = 243.8kPa}$$

Question: 4.115P

If sufficient data are provided, complete the blank cells in the following table of properties of water. In the last column describe the condition of water as compressed liquid, saturated mixture, superheated vapor, or insufficient information; and, if applicable, give the quality.

P, kPa	*T*,°C	v,m3/kg	u,kJ/kg	Condition description and quality (if applicable)
	250		2728.9	
300			1560.0	
101.42	100			
3000	180			

Step 1 of 5

Given

Temperature $T = 250^0 \text{C}$

Internal energy $u = 2728.9 \text{kJ/kg}$

At $T = 250^0 \text{C}$ and $u = 2728.9 \text{kJ/kg}$ from superheated water tables

Pressure $P = 300 \text{kPa}$

Specific volume $v = 0.79645 \text{m}^3/\text{kg}$

Step 2 of 5

Pressure $P = 300 \text{kPa}$

Specific volume $u = 1560.0 \text{kJ/kg}$

At $P = 300 \text{kPa}$ from saturated water tables

$u = u_{f@300\text{kPa}} + x u_{fg@300\text{kPa}}$

$1560.0 = 561.11 + x \times 1982.1$

$x = 0.5039$

$T = T_{sat@300\text{kPa}} = 133.52^0 \text{C}$

$v = v_{f@300\text{kPa}} + x v_{fg@300\text{kPa}}$

$v = 0.001073 + 0.5039 \times 0.6047$

$v = 0.30578 \text{m}^3/\text{kg}$

Step 3 of 5

Pressure $P = 101.42$ kPa

Temperature $T = 100^0 \text{C}$

At $T = 100^0 \text{C}$ and $P = 101.42 \text{kPa}$ it is in saturated condition but it is not mentioned that whether it is in saturated liquid or saturated vapor state

Step 4 of 5

Pressure $P = 3000 \text{kPa}$

Temperature $T = 180^0 \text{C}$

At $P = 3000 \text{kPa}$ and $T = 180^0 \text{C}$ from compressed water tables

$v = 0.0011 \text{m}^3/\text{kg}$

$u = 760.9 \text{kJ/kg}$

Step 5 of 5

P (kPa)	T(°C)	v (m^3/kg)	u(kJ/kg)	Phase
300	250	0.79645	2728.9	Superheated steam
300	133.52	0.30578	1560	Saturated mixture $x = 0.5039$
101.42	100	-	-	Insufficient data
3000	180	0.0011	760.9	Compressed liquid

Question: 4.116P

If sufficient data are provided, complete the blank cells in the following table of properties of water. In the last column describe the condition of water as compressed liquid, saturated mixture, superheated vapor, or insufficient information; and, if applicable, give the quality.

P, kPa	T,°C	v,m3/kg	u,kJ/kg	Condition description and quality (if applicable)
200			2529.1	
	125			$x = 0.650$
	400		2967.2	
1000	30			
120.90	150			

Step 1 of 6

Given

Pressure $P = 200\text{kPa}$

Internal energy $u = 2529.1\text{kJ/kg}$

From saturated water tables at $P = 200\text{kPa}$ and $u = 2529.1\text{kJ/kg}$ the state is in saturated vapor state

Temperature $\boxed{T = 120.21^0\text{C}}$

Specific volume

$v = v_g = 0.88578\text{m}^3\text{/kg}$

$\boxed{v = 0.88578\text{m}^3\text{/kg}}$

Step 2 of 6

Given

Temperature $T = 125^0\text{C}$

Dryness fraction $x = 0.65$

From saturated water tables

$P = P_{sat} = 232.23\text{kPa}$

$\boxed{P = 232.23\text{kPa}}$

Specific volume

$v = v_{f@125C} + xv_{fg@125C}$

$v = 0.001065 + 0.65 \times 0.769$

$\boxed{v = 0.5009\text{m}^3\text{/kg}}$

Internal energy

$u = u_{f@125C} + xu_{fg@125C}$

$u = 524.83 + 0.65 \times 2009.5$

$\boxed{u = 1831.005\text{kJ/kg}}$

Step 3 of 6

Given

Temperature $T = 400^0\text{C}$

Internal energy $u = 2967.2\text{kJ/kg}$

The state is in superheated state

So from superheated water tables at $T = 400^0\text{C}$ and $u = 2967.2\text{kJ/kg}$

$\boxed{P = 200\text{kPa}}$

$\boxed{v = 1.54934\text{m}^3\text{/kg}}$

Step 4 of 6

Given

Pressure $P = 1000\text{kPa}$

Temperature $T = 30^0\text{C}$

The state is in compressed liquid state

So at $P = 1000\text{kPa}$ and $T = 30^0\text{C}$ from compressed liquid water tables

$\boxed{v = 0.0010\text{m}^3\text{/kg}}$

$\boxed{u = 125.6\text{kJ/kg}}$

Step 5 of 6

Given

Pressure $P = 120.90\text{kPa}$

Temperature $T = 150^0\text{C}$

From superheated water tables

Specific volume $\boxed{v = 1.5088\text{m}^3\text{/kg}}$

Internal energy $\boxed{u = 2581.2\text{kJ/kg}}$

Step 6 of 6

P(kPa)	T(°C)	v (m³/kg)	u(kJ/kg)	Phase
200	120.21	0.88578	2529.1	Saturated vapor
232.23	125	0.5009	1831.005	$x = 0.650$
200	400	1.54394	2967.2	Superheated steam
1000	30	0.0010	125.6	Compressed state
120.9	150	1.5988	2581.2	Superheated steam

Question: 4.117P

Water initially at 300 kPa and 250°C is contained in a constant volume tank. The water is allowed to cool until its pressure is 150 kPa. On the *P-V*and *T-V* diagrams sketch, with respect to the saturation lines, the process curve passing through both the initial and final states of the water. Label the end states on the process curve. Also, on both the *P-V* and *T-V* diagrams, sketch the isotherms passing through both states and show their values, in °C, on the isotherms.

Step 1 of 3

Given

Initial pressure $P_1 = 300\text{kPa}$

Initial temperature $T_1 = 250^0\text{C}$

Final pressure $P_2 = 150\text{kPa}$

Step 2 of 3

At $P_1 = 300$ kPa and $T_1 = 250^0\text{C}$ from superheated water tables

$v_1 = 0.79645\text{m}^3/\text{kg}$

Since volume remains constant

At $P_2 = 150\text{kPa}$ and $v_2 = v_1 = 0.79645\text{m}^3/\text{kg}$ from saturated water tables

$v_2 = v_{f@150kPa} + x_2 v_{fg@150kPa}$

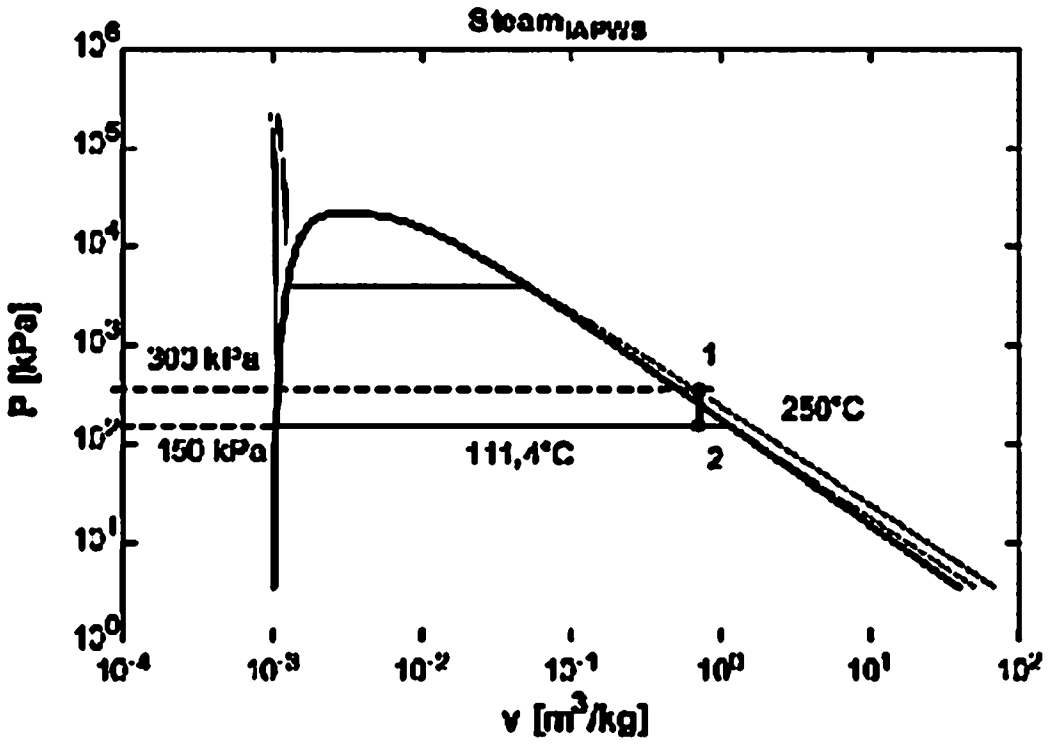
$0.79645 = 0.001053 + x_2 \times 1.15834$

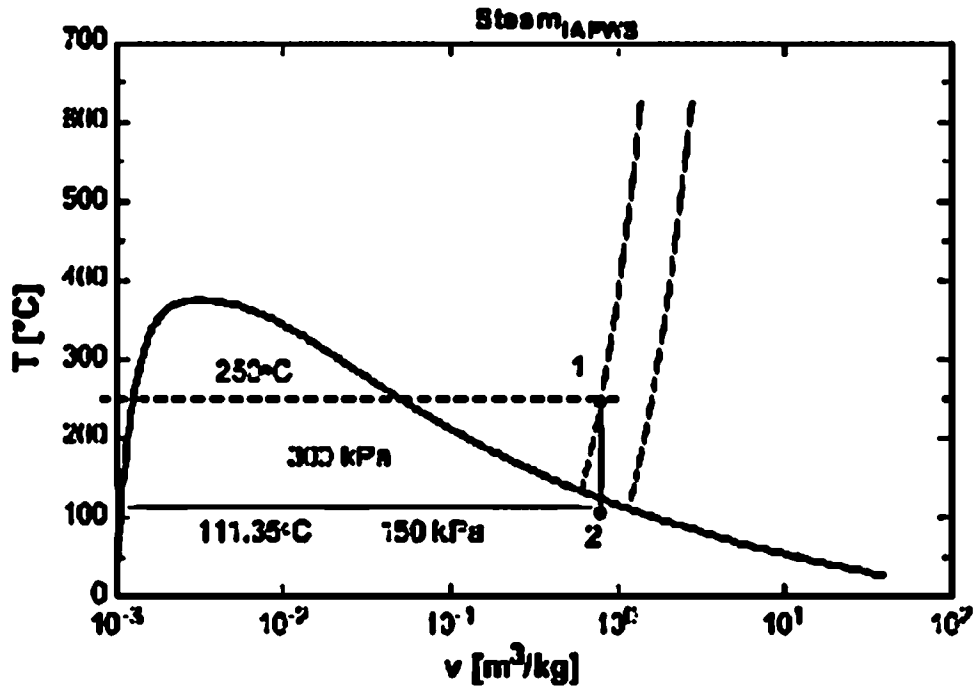
$x_2 = 0.686$

$T_2 = T_{sat@150kPa} = 111.35^0\text{C}$

Step 3 of 3

Using Property Plot feature of EES, and by adding state points we obtain following diagrams.





CHAPTER 05
SOLUTIONS

Question: 5.001P

Is the boundary work associated with constant-volume systems always zero?

Step 1 of 1

The boundary work during a quasi-equilibrium process is expressed as

$$Area = A = \int_1^2 dA = \int_1^2 PdV$$

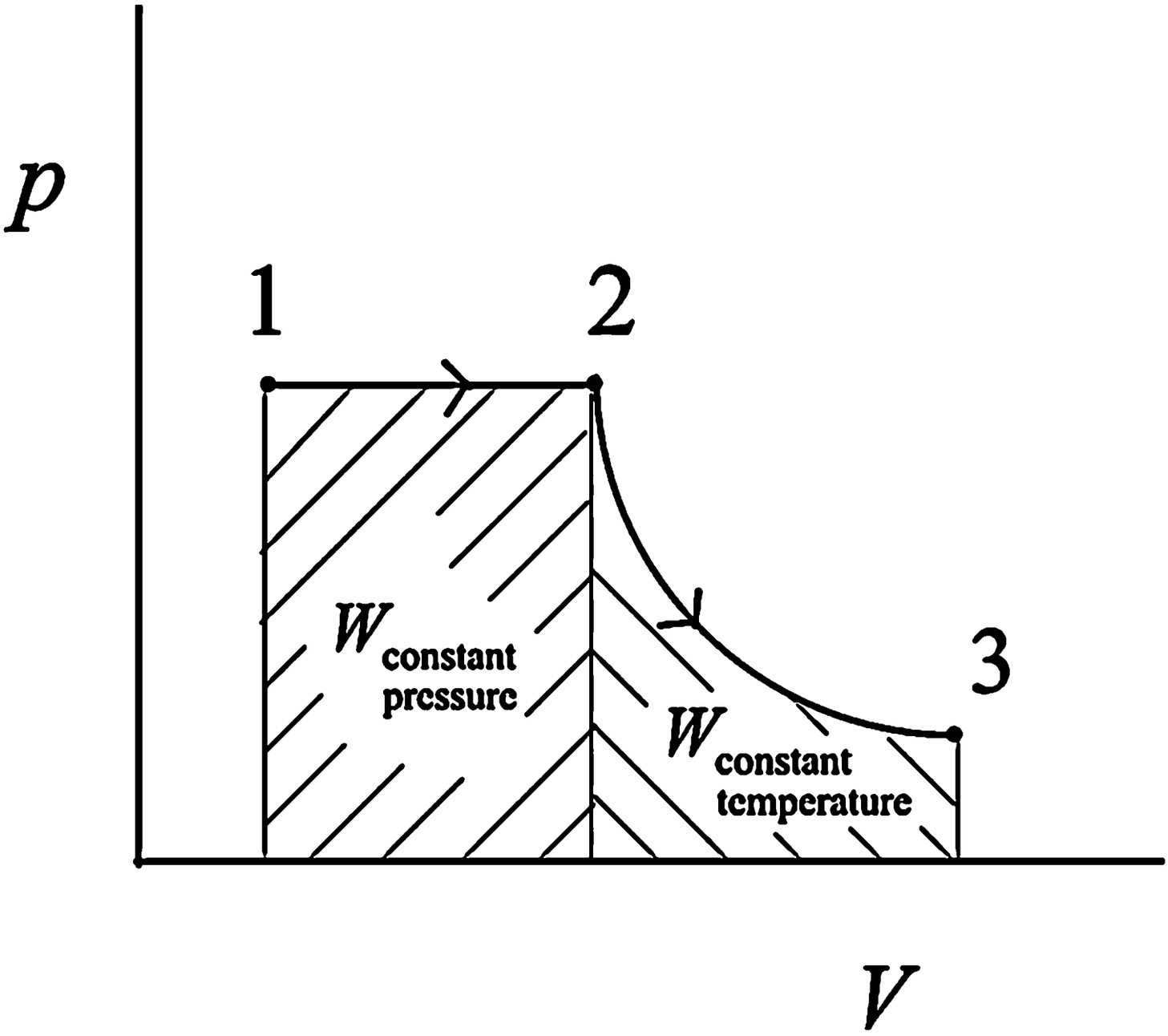
For a constant volume process $dV = 0$.
Therefore the boundary work is zero.

Question: 5.002P

An ideal gas at a given state expands to a fixed final volume first at constant pressure and then at constant temperature For which case is the work done greater?

Step 1 of 2

Draw the two expansion processes on the $p-V$ diagram.



Step 2 of 2

The process $1-2$ is a constant pressure expansion process and the process $2-3$ is an isothermal expansion process.

The area under the curve in a $p-V$ diagram represents the work done during the process.

From figure, the area under the curve in a constant pressure process is more as compared to the area in a constant temperature process.

Therefore, the work done is greater for constant pressure process.

Question: 5.003P

Show that 1 kPa-m3 = 1 kJ.

Step 1 of 1

We know that $1\,Pa = 1\,N/m^2$

$$1\,kPa.m^3 = 1\,kN/m^2.m^3$$

$$= 1\,kNm \qquad (1Nm = 1J)$$

Therefore $1\,kPa.m^3 = 1\,kJ$

Question: 5.004P

The volume of 1 kg of helium in a piston-cylinder device is initially 7 m3. Now helium is compressed to 3 m3 while its pressure is maintained constant at 150 kPa. Determine the initial and final temperatures of helium as well as the work required to compress it, in kJ.

Step 1 of 3

Obtain the characteristic gas constant of helium from "Table A-1: Molar mass, gas constant, and critical-point properties" in the textbook.

$$R = 2.0769\,\text{kJ/kg}\cdot\text{K}$$

Write the ideal gas equation to find the initial temperature of helium.

$$P_1V_1 = mRT_1$$

Here, P_1 is the initial pressure of the gas , V_1 is the initial volume of the gas, m is the mass of the helium gas, R is the characteristic gas constant, and T_1 is the initial temperature of the helium gas.

Substitute $150\,\text{kPa}$ for P_1 , $7\,\text{m}^3$ for V_1 , $1\,\text{kg}$ for m , and $2.0769\,\text{kJ/kg}\cdot\text{K}$ for R .

$$150\times 7 = 1\times 2.0769\times T_1$$
$$T_1 = 505.5\,\text{K}$$

Therefore, the initial temperature of the helium gas is $\boxed{505.5\,\text{K}}$.

Step 2 of 3

Write the ideal gas equation to find the final temperature of helium.

$$P_2V_2 = mRT_2$$

Here, P_2 is the final pressure of the gas , V_2 is the final volume of the gas, and T_2 is the final temperature of the helium gas.

Substitute $150\,\text{kPa}$ for P_2 , $3\,\text{m}^3$ for V_2 , $1\,\text{kg}$ for m , and $2.0769\,\text{kJ/kg}\cdot\text{K}$ for R .

$$150\times 3 = 1\times 2.0769\times T_2$$
$$T_2 = 216.66\,\text{K}$$

Therefore, the final temperature of the helium gas is $\boxed{216.66\,\text{K}}$.

Step 3 of 3

Find the work required to compress the helium.

$$W = \int_1^2 P\,dV$$
$$= P(V_2 - V_1)$$

Substitute $150\,\text{kPa}$ for P , $7\,\text{m}^3$ for V_1 , and $3\,\text{m}^3$ for V_2 .

$$W = 150(3-7)$$
$$= -600\,\text{kJ}$$

Since work is done by the system work is negative.

Therefore, the work required to compress the helium is $\boxed{600\,\text{kJ}}$.

Question: 5.005P

A piston-cylinder device initially contains 0.07 m3 of nitrogen gas at 130 kPa and 120°C. The nitrogen is now expanded polytropically to a state of 100 kPa and 100°C. Determine the boundary work done during this process.

Step 1 of 3

Calculate the mass of the nitrogen gas inside the piston cylinder

$$P_1V_1 = mRT_1$$

$$m = \frac{P_1V_1}{RT_1}$$

Here, P_1 is the pressure at initial state, V_1 is the Initial volume, T_1 is the Initial Temperature and R is the gas constant

Use the conversion factor to convert the temperature from $1°C$ to $1K$

$$1K = 1°C + 273$$

Substitute $130\ kPa$ for P_1, $0.07\ m^3$ for V_1, $0.2968\ kJ/kg \cdot K$ for R and $120°C$ for T_1

$$m = \frac{130(0.07)}{(0.2968)(120+273)}$$
$$= 0.0780\ kg$$

Step 2 of 3

Calculate the volume after expansion

$$V_2 = \frac{mRT_2}{P_2}$$

Here, T_2 is the final temperature and P_2 is the final pressure

Substitute $100°C$ for T_2, $100\ kPa$ for P_2 and $0.0780\ kg$ for m

$$V_2 = \frac{mRT_2}{P_2}$$
$$= \frac{(0.078)(0.2968)(100+273)}{100}$$
$$= 0.08635\ m^3$$

Step 3 of 3

Calculate the polytropic index by using the expression,

$$P_1V_1^n = P_2V_2^n$$

$$P_1V_1^n = P_2V_2^n$$
$$130(0.07)^n = 100(0.08635)^n$$
$$n = 1.24986$$

Calculate the work done during the process by using the equation,

$$W = \frac{P_2V_2 - P_1V_1}{1-n}$$
$$= \frac{100(0.08635) - 130(0.07)}{1-1.24986}$$
$$= 1.86\ kJ$$

Therefore, work done is $\boxed{1.86\ kJ}$

Question: 5.006P

A piston-cylinder device with a set of stops initially contains 0.3 kg of steam at 1.0 MPa and 400°C. The location of the stops corresponds to 60 percent of the initial volume. Now the steam is cooled. Determine the compression work if the final state is (*a*) 1.0 MPa and 250°C and (*b*) 500 kPa. (*c*) Also determine the temperature at the final state in part (*b*).

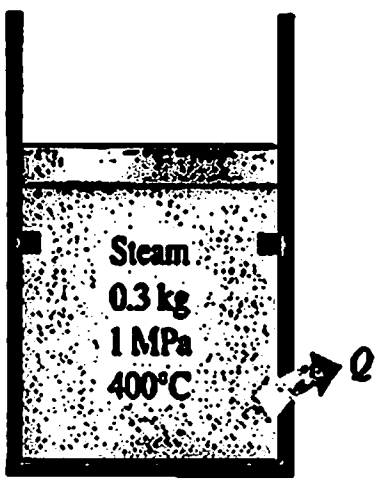
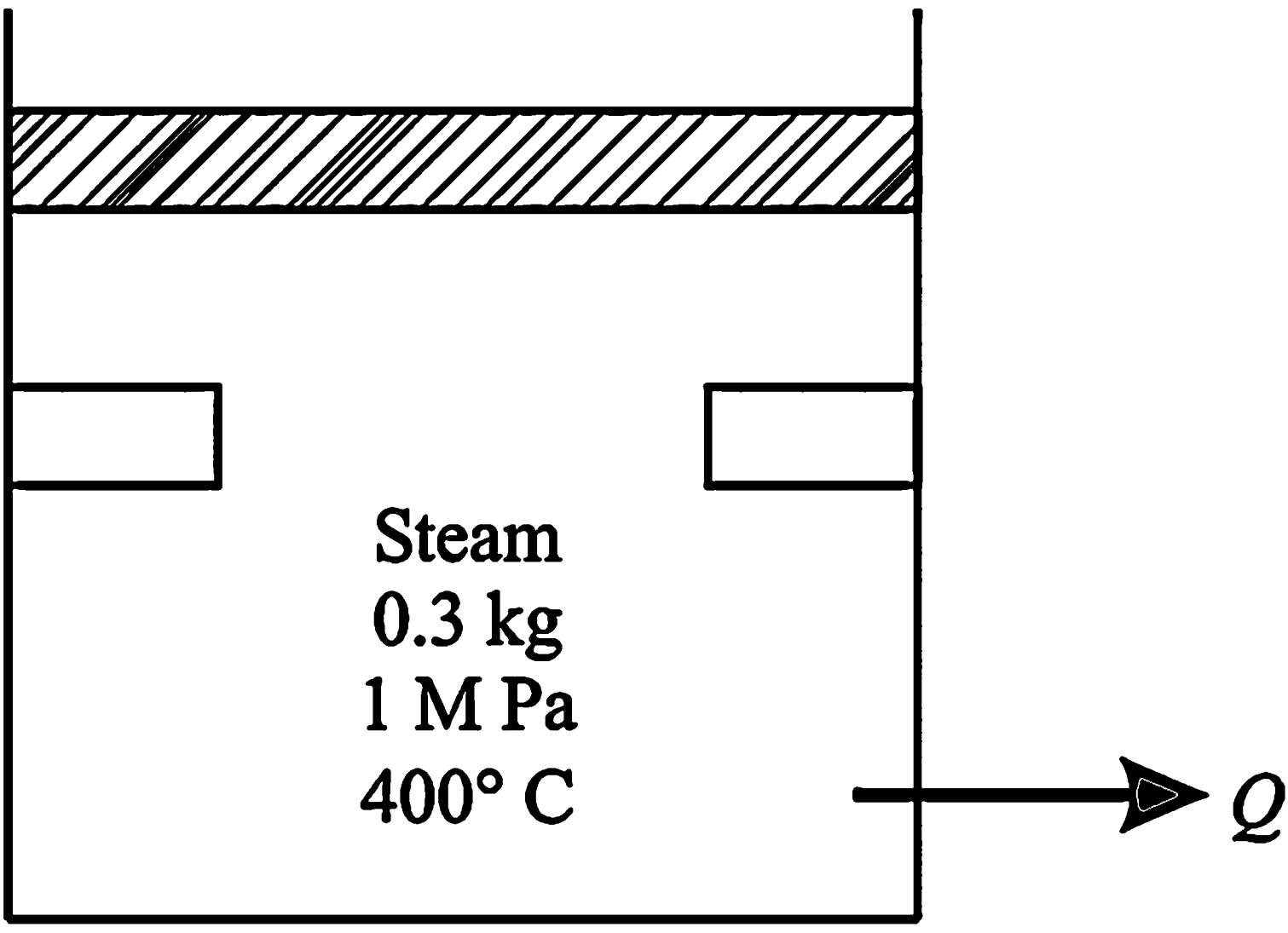


FIGURE P5–6

Step 1 of 4

Draw the piston-cylinder device with set of stops as follows:



Step 2 of 4

(a)

Obtain the specific volume of the steam at a pressure of 1 MPa and at a temperature of $400°C$ from the table A-6, "Superheated water" in the textbook.

$$v_1 = 0.30661\ \text{m}^3/\text{kg}$$

Obtain the specific volume of the steam at a pressure of 1 MPa and at a temperature of $250°C$ from the table A-6, "Superheated water" in the textbook.

$$v_2 = 0.23275\ \text{m}^3/\text{kg}$$

Calculate the compression work as follows:

$$W = mP_1(v_1 - v_2)$$

Here, m is the mass of the steam.

Substitute 0.3 kg for m, 1 MPa for P_1, $0.30661\ \text{m}^3/\text{kg}$ for v_1, and $0.23275\ \text{m}^3/\text{kg}$ for v_2.

$$\begin{aligned} W &= 0.3(1000)(0.30661 - 0.23275) \\ &= 22.158\ \text{kJ} \end{aligned}$$

Therefore, the compression work is $\boxed{22.158\ \text{kJ}}$.

Step 3 of 4

(b)

Here pressure $P_2 = 500\ \text{kPa}$

Calculate the compression work as follows:

$$W = mP_1(v_1 - v_2)\ \text{kJ}$$

Here the $v_2 = 0.6v_1$, re-write the equation as follows:

$$W = mP_1(v_1 - 0.6v_1)$$

Substitute 0.3 kg for m, 1 MPa for P_1, and $0.30661\ \text{m}^3/\text{kg}$ for v_1.

$$\begin{aligned} W &= 0.3(1000)(0.30661 - 0.6 \times 0.30661) \\ &= 36.8\ \text{kJ} \end{aligned}$$

Therefore, the compression work is $\boxed{W = 36.8\ \text{kJ}}$.

Step 4 of 4

(c)

Obtain the specific volume of the steam at a pressure of 500 kPa and from the table A-5, "Saturated water-Pressure table" in the textbook.

$$T_2 = 151.83°C$$

Therefore, the temperature is $\boxed{152°C}$.

Question: 5.007P

A piston-cylinder device initially contains 0.07 m3 of nitrogen gas at 130 kPa and 180°C. The nitrogen is now expanded to a pressure of 80 kPa polytropically with a polytropic exponent whose value is equal to the specific heat ratio (called *isentropic expansion*) . Determine the final temperature and the boundary work done during this process.

Step 1 of 6

Given:
Process: polytropic process
Initially volume $V_1 = 0.07\,\mathrm{m}^3$
Initially pressure $P_1 = 130\,\mathrm{kPa}$
Initially temperature $T_1 = 180^\circ\mathrm{C}$
Final pressure $P_2 = 80\,\mathrm{kPa}$

Step 2 of 6

From the ideal gas specific heat of various gases
For nitrogen

$$R = 0.2968\frac{\mathrm{kJ}}{\mathrm{kg}\cdot\mathrm{K}}$$

$k = 1.4$

Step 3 of 6

Now applying the idea gas equation

$$P_1V_1 = mRT_1$$

$$\Rightarrow m = \frac{P_1V_1}{RT_1}$$

$$= \frac{(130)(0.07)}{(0.2968\mathrm{K})(180+273\mathrm{K})}$$

$$= 0.0676\,\mathrm{kg}$$

Step 4 of 6

We know that
For polytropic process

$$P_1V_1^k = P_2V_2^k$$

$$V_2^k = \frac{(130)\times(0.07)^{1.4}}{80}$$

$$V_2 = (0.0393)^{1/1.4}$$

$$= 0.0991\mathrm{m}^3$$

Step 5 of 6

The final temperature is
Applying the ideal gas equation

$$P_2V_2 = mRT_2$$

$$\Rightarrow T_2 = \frac{P_2V_2}{mR}$$

$$= \frac{80\times0.0991}{0.0677\times0.2968}$$

$$= 394.55\mathrm{K}$$

$$\boxed{T_2 = 394.55\mathrm{K}}$$

Step 6 of 6

The boundary work for polytropic process is

$$W_b = \frac{P_2V_2 - P_1V_1}{1-k}$$

$$= \frac{(80\times0.0991)-(130\times0.07)}{1-1.4}$$

$$= 2.93\mathrm{kJ}$$

$$\boxed{W_b = 2.93\mathrm{kJ}}$$

Question: 5.008P

A mass of 5 kg of saturated water vapor at 300 kPa is heated at constant pressure until the temperature reaches 200°C. Calculate the work done by the steam during this process.

Step 1 of 3

Given

Mass of water vapor $m = 5\ \text{kg}$

Initial Pressure $P_1 = 300\ \text{kPa}$

Initial Temperature $T_2 = 200°\text{C}$

Step 2 of 3

From superheated steam tables

The properties of steam at $P_1 = 300\ \text{kPa}$

Specific volume is $v_1 = 0.60582\ \text{m}^3/\text{kg}$

From superheated steam tables

The properties of steam at $P_2 = 300\ \text{kPa}$ and $T_2 = 200°\text{C}$

Specific volume is $v_2 = 0.71643\ \text{m}^3/\text{kg}$

Step 3 of 3

The work done during the process to

$$W = mP_1(v_2 - v_1)\ \text{kJ}$$

$$W = 5(300)(0.71643 - 0.60582)\ \text{kJ}$$

$$\boxed{W = 165.915\ \text{kJ}}$$

Question: 5.009P

A frictionless piston-cylinder device initially contains 50 L of saturated liquid refrigerant-134a. The piston is free to move, and its mass is such that it maintains a pressure of 500 kPa on the refrigerant. The refrigerant is now heated until its temperature rises to 70°C. Calculate the work done during this process.

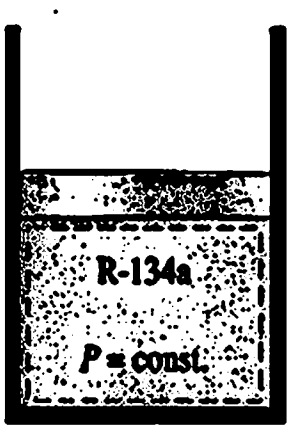


FIGURE P5–9

Step 1 of 5

Given:
Working fluid: R-134a
Initially volume $V_1 = 50\text{L}$
Pressure of the refrigerant $P_1 = P_2 = 500\ \text{kPa}$
Final temperature $T_2 = 70°\text{C}$

Step 2 of 5

State1:
From the Properties of saturated refrigerant -134a
At $P_1 = 500\text{kPa}$

$$v_1 = v_f == 0.0008059\ \frac{\text{m}^3}{\text{kg}}$$

Step 3 of 5

State2:
$T_{sat} < T_2$
$15.71°\text{C} < 70°\text{C}$
So state 2 is superheated state.
From the superheated refrigerant-134a
At $P_2 = 500\text{kPa}$ and $T_2 = 70°\text{C}$

$$v_2 = 0.052427\ \frac{\text{m}^3}{\text{kg}}$$

Step 4 of 5

Initially volume
$V_1 = 50\text{L}$

$$= (50\text{L})\left(\frac{1\text{m}^3}{1000\text{L}}\right)$$

$$= 50\times10^{-3}\ \text{m}^3$$

Then mass of the system is

$$m = \frac{V_1}{v_1}$$

$$= \frac{50\times10^{-3}}{0.0008059}$$

$$= 62.042\text{kg}$$

Step 5 of 5

The boundary work is

$$W = mP(v_2 - v_1)$$

$$= 62.042\times500(0.052427 - 0.0008059)$$

$$= 1601.33\text{kJ}$$

$$\boxed{W = 1601.33\ \text{kJ}}$$

Question: 5.010P



Reconsider Prob. 5–9. Using EES (or other) software, investigate the effect of pressure on the work done. Let the pressure vary from 200 to 1200 kPa. Plot the work done versus the pressure, and discuss the results. Explain why the plot is not linear. Also plot the process described in Prob. 5–9 on the *P- V* diagram.

We don't have the solution to this problem yet.

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Question: 5.011P

1-m3 of saturated liquid water at 200°C is expanded isothermally in a closed system until its quality is 80 percent. Determine the total work produced by this expansion, in kJ.

Step 1 of 3

Obtain the properties from the saturated temperature table at temperature $T_1 = 200°C$

Saturated pressure is $P = 1554.9\,\text{kPa}$

Specific volume at state point 1 is $v_1 = v_f = 0.001157\,\text{m}^3/\text{kg}$

Specific volume of saturated mixture is $v_g = 0.12721\,\text{m}^3/\text{kg}$

Calculate the Specific volume at state2 .

$$\begin{aligned} v_2 &= v_f + xv_{fg} \\ &= v_f + x\left(v_g - v_f\right) \\ &= 0.001157 + [0.8\times(0.12721 - 0.001157)] \\ &= 0.102\,\text{m}^3/\text{kg} \end{aligned}$$

Step 2 of 3

Calculate the volume of the liquid at state point 2

$$\frac{V_2}{V_1} = \frac{v_2}{v_1}$$

Here, V_1 and V_2 is the volume of the state point 1 and 2 respectively.

Substitute $1\,\text{m}^3$ for V_1, $0.001157\,\text{m}^3/\text{kg}$ for v_1 and $0.102\,\text{m}^3/\text{kg}$ for v_2

$$\frac{V_2}{V_1} = \frac{v_2}{v_1}$$
$$\frac{V_2}{1} = \frac{0.102}{0.001157}$$
$$V_2 = 88.16\,\text{m}^3$$

Step 3 of 3

Calculate the work done during the isothermal process, since there is only phase change we can consider pressure constant.

$${}_1W_2 = P\left(V_2 - V_1\right)$$

Substitute $88.16\,\text{m}^3$ for V_2, $1\,\text{m}^3$ for V_1, $1554.9\,\text{kPa}$ for P

$$\begin{aligned} {}_1W_2 &= P\left(V_2 - V_1\right) \\ &= 1554.9(88.16 - 1) \\ &= 1.355\times10^5\,\text{kJ} \end{aligned}$$

Hence, the work done is $\boxed{1.355\times10^5\,\text{kJ}}$

Question: 5.012P

A mass of 2.4 kg of air at 150 kPa and 12°C is contained in a gas-tight, frictionless piston-cylinder device. The air is now compressed to a final pressure of 600 kPa. During the process, heat is transferred from the air such that the temperature inside the cylinder remains constant. Calculate the work input during this process.

Step 1 of 2

Obtain the properties from the tables of ideal-gas specific heats of various common gases.

The gas constant of air is $R = 0.287\ \text{kJ/kg}\cdot\text{K}$.

Calculate the work done during the process by using the equation as shown below.

$$W = \int_1^2 PdV\ \text{kJ}$$

$$= P_1V_1 \ln\frac{V_2}{V_1}\ \text{kJ}$$

$$= mRT_1 \ln\frac{P_1}{P_2}\ \text{kJ}$$

Here, the mass of air is m , the gas constant of air is R , the temperature at state 1 and state 2 is T_1 , the pressure at state 1 is P_1 , and the pressure at state 2 is P_2 .

Step 2 of 2

Substitute $2.4\ \text{kg}$ for m , $150\ \text{kPa}$ for P_1 , $600\ \text{kPa}$ for P_2 and $12°\text{C}$ for T_1 .

$$W = 2.4(0.287)(12+273)\ln\left(\frac{150}{600}\right)$$

$$= -272.14\ \text{kJ}$$

Therefore, the work done during the process is $\boxed{-272.14\ \text{kJ}}$.

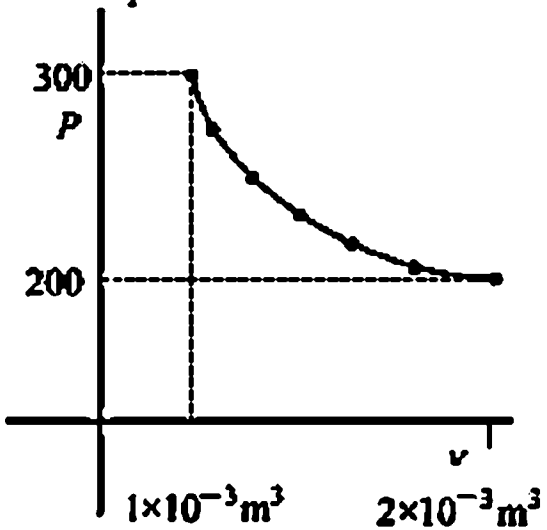
The negative sign indicates that the work is done on the system to compress the air.

Question: 5.013P

Determine the boundary work done by a gas during an expansion process if the pressure and volume values at various states are measured to be 300 kPa, 1 L;290 kPa, 1.1 L; 270 kPa, 1.2 L; 250 kPa, 1.4 L; 220 kPa, 1.7 L; and 200 kPa, 2 L.

Step 1 of 2

The work done in the process 1-2 equal to area under the curve



Step 2 of 2

The area under the curve is expressed as

$$W=\frac{P_1+P_2}{2}(V_2-V_1)$$

$$W=\frac{300+200}{2}(2-1)\times10^{-3}$$

$$\boxed{W=0.25\,\text{kJ}}$$

Question: 5.014P

During some actual expansion and compression processes in piston-cylinder devices, the gases have been observed to satisfy the relationship *PVn* = *C*, where *n* and *C* are constants. Calculate the work done when a gas expands from 350 kPa and 0.03 m3 to a final volume of 0.2 m3 for the case of *n* = 1.5.

Step 1 of 4

Given:

Process: $PV^n = C$

Initially pressure $P_1 = 350\,\text{kPa}$

Initially volume $v_1 = 0.03\text{m}^3$

Final volume $V_2 = 0.2\,\text{m}^3$

Process constant $n = 1.5$

Step 2 of 4

We know that

$$PV^n = C$$

$$C = P_1V_1^n$$

$$= 350\times(0.03)^{1.5}$$

$$= 1.81$$

Step 3 of 4

And also

Final pressure process is

$$P_2V_2^n = C$$

$$\Rightarrow P_2 = \frac{C}{V_2^n}$$

$$= \frac{1.81}{(0.2)^{1.5}}$$

$$= 20.23\text{kPa}$$

Step 4 of 4

Work done

$$W = \frac{P_1V_1 - P_2V_2}{n-1}$$

$$= \frac{(350\times0.03)-(20.23\times0.2)}{1.5-1}$$

$$= 12.908\text{kJ}$$

$$\boxed{W = 12.908\ \text{kJ}}$$

Question: 5.015P



Reconsider Prob. 5–14. Using the EES (or other) software, plot the process described in the problem on a *P*- *V* diagram, and investigate the effect of the polytropic exponent *n* on the boundary work. Let the polytropic exponent vary from 1.1 to 1.6. Plot the boundary work versus the polytropic exponent, and discuss the results.

We don't have the solution to this problem yet.

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Question: 5.016P

A frictionless piston–cylinder device contains 2 kg of nitrogen at 100 kPa and 300 K. Nitrogen is now compressed slowly according to the relation $PV1.4$ = constant until it reaches a final temperature of 360 K. Calculate the work input during this process.

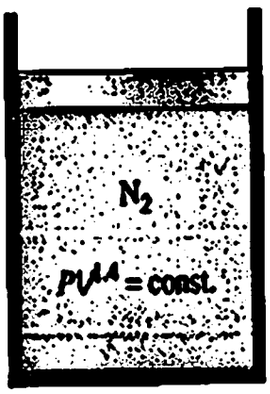


FIGURE P5–16

Step 1 of 3

Given

The mass of nitrogen $m = 2\,\text{kg}$

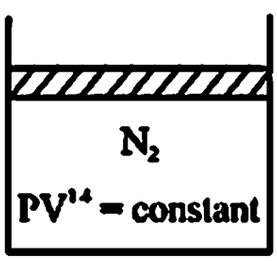
Initial Pressure $P_1 = 100\,\text{kPa}$

Initial Temperature $T_1 = 300\text{K}$

Final temperature $T_2 = 360\text{K}$

The polytrophic process index $n = 1.4$

Step 2 of 3



Step 3 of 3

From tables of ideal gas specific heats of various common gases

R = 0.2968 kJ/kg.K

The boundary work during the process is

$$W = \frac{P_2V_2 - P_1V_1}{1-n}\ \text{kJ} \qquad (PV = mRT)$$

$$W = \frac{mR(T_2 - T_1)}{1-1.4}$$

$$W = \frac{2(0.2968)(360-300)}{1-1.4}$$

$$\boxed{W = -89\ \text{kJ}}$$

The negative sign indicates that work is done on the system

Question: 5.017P

The equation of state of a gas is given as $\bar{v}(P + 10/\bar{v}^2) = Ru\,T$, where the units of $\bar{v}$ and P are m3/kmol and kPa, respectively. Now 0.2 kmol of this gas is expanded in a quasi-equilibrium manner from 2 to 4 m3 at a constant temperature of 350 K. Determine (*a*) the unit of the quantity 10 in the equation and (*b*) the work done during this isothermal expansion process.

Step 1 of 2

Write the equation of state of gas

$$\bar{v}\left(P+\frac{10}{\bar{v}^2}\right)=R_u T$$

Here, $\bar{v}$ is specific molar volume in $\mathrm{m^3/kmol}$, P is pressure in kPa, R_u is universal gas constant, and T is temperature.

a)

Since the term $\frac{10}{\bar{v}^2}$ is added to pressure, so it must have pressure units.

$$\frac{10}{\bar{v}^2}=\mathrm{kPa}$$

Substitute $\frac{\mathrm{m^3}}{\mathrm{kmol}}$ for $\bar{v}$

$$\frac{10}{\left(\frac{\mathrm{m^3}}{\mathrm{kmol}}\right)^2}=\mathrm{kPa}$$

$$10=\mathrm{kPa}\times\frac{\mathrm{m^6}}{\mathrm{kmol^2}}$$

Therefore, the unit of quantity 10 in the equation is $\boxed{\mathrm{kPa}\times\frac{\mathrm{m^6}}{\mathrm{kmol^2}}}$

Step 2 of 2

(b)

Find the boundary work done during the isothermal expansion process by using the following equation:

$$W_{b,out}=\int_1^2 P\,dV$$

Write the equation of state of a gas

$$\bar{v}\left(P+\frac{10}{\bar{v}^2}\right)=R_u T$$

$$P=\frac{R_u T}{\bar{v}^2}-\frac{10}{\bar{v}^2}$$

Substitute $\frac{R_u T}{\bar{v}^2}-\frac{10}{\bar{v}^2}$ for P in work done equation.

$$\begin{aligned}
W_{b,out} &= \int_1^2\left(\frac{R_u T}{\bar{v}}-\frac{10}{\bar{v}^2}\right)dV \\
&= \int_1^2\left(\frac{R_u T}{V/N}-\frac{10}{(V/N)^2}\right)dV \\
&= NR_u T\int_1^2\left(\frac{1}{V}\right)dV-10N^2\int_1^2\left(\frac{1}{V^2}\right)dV \\
&= NR_u T(\ln V)_1^2-10N^2\left(\frac{-1}{V}\right)_1^2 \\
&= NR_u T\ln\left(\frac{V_2}{V_1}\right)+10N^2\left(\frac{1}{V_2}-\frac{1}{V_1}\right)
\end{aligned}$$

Substitute 0.2 kmol for N, $8.314\,\mathrm{kJ/kmol.K}$ for R, 350 K for T, $2\,\mathrm{m^3}$ for V_1, and $4\,\mathrm{m^3}$ for V_2.

$$\begin{aligned}
W_{b,out} &= (0.2)(8.314)(350)\ln\left(\frac{4}{2}\right)+(10)(0.2)^2\left[\frac{1}{4}-\frac{1}{2}\right] \\
&= 403.397+(-0.1) \\
&= 403.297\,\mathrm{kJ}
\end{aligned}$$

Therefore, the work done during the isothermal expansion process is $\boxed{403.297\,\mathrm{kJ}}$

Question: 5.018P



Reconsider Prob. 5–17. Using the integration feature of the EES software, calculate the work done, and compare your result with the "hand-calculated" result obtained in Prob. 5–17. Plot the process described in the problem on a *P- v* diagram.

We don't have the solution to this problem yet.

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Question: 5.019P

Carbon dioxide contained in a piston-cylinder device is compressed from 0.3 to 0.1 m3. During the process, the pressure and volume are related by $P = a\,V^{-2}$, where $a = 8$ kPam6. Calculate the work done on the carbon dioxide during this process.

Step 1 of 2

Given

Initial volume $V_1 = 0.3\ \text{m}^3$

Final volume $V_2 = 0.1\ \text{m}^3$

The relation between pressure and volume is

$P = aV^{-2}$ (Where $a = 8\ \text{kPa.m}^6$)

Step 2 of 2

The work done during any process involving boundary work is expressed as

$$W = \int_1^2 P\,dV\ \text{kJ}$$

$$W = \int_1^2 \frac{a}{V^2}\,dV\ \text{kJ}$$

$$W = -a\left(\frac{1}{V_2} - \frac{1}{V_1}\right)$$

$$W = -8\left(\frac{1}{0.1} - \frac{1}{0.3}\right)$$

$$\boxed{W = -53.3\ \text{kJ}}$$

The negative sign indicates work is done on the system

Question: 5.020P

During an expansion process, the pressure of a gas changes from 15 to 100 psia according to the relation $P = a\,V + b$, where $a = 5$ psia/ft3 and b is a constant. If the initial volume of the gas is 7 ft3, calculate the work done during the process.

Step 1 of 4

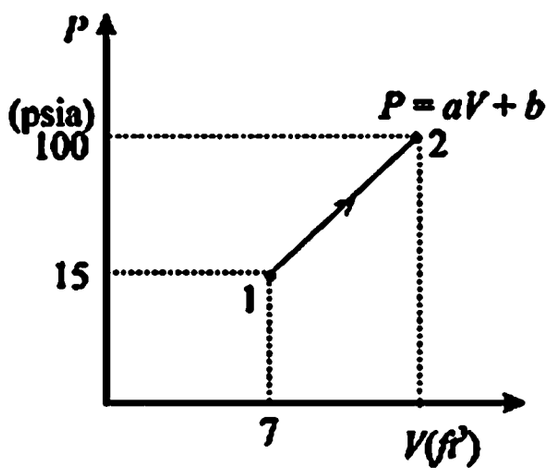
Given

Initial Pressure $P_1 = 15$ psia

Final Pressure $P_2 = 100$ psia

Initial volume $V_1 = 7\ \text{ft}^3$

Step 2 of 4



Step 3 of 4

The relation $P = aV + b$, where a = 5 psia/ft^3, b is constant

Initial state

$$P_1 = aV_1 + b$$
$$15 = (5)(7) + b$$
$$b = -20$$

Final state

$$P_2 = aV_2 + b$$
$$100 = (5)V_2 - 20$$
$$V_2 = 24\ \text{ft}^3$$

Step 4 of 4

The work done during the process is

Area under the curve $W = \dfrac{P_1 + P_2}{2}(V_2 - V_1)$

$$W = \frac{15+100}{2}(24-7) \qquad (1\ \text{Btu} = 5.404\ \text{psia.ft}^3)$$

$$W = 977.5 \times \frac{1\ \text{Btu}}{5.404}$$

$$\boxed{W = 181\ \text{Btu}}$$

Question: 5.021P

A piston-cylinder device initially contains 0.25 kg of nitrogen gas at 130 kPa and 180°C. The nitrogen is now expanded isothermally to a pressure of 80 kPa. Determine the boundary work done during this process.

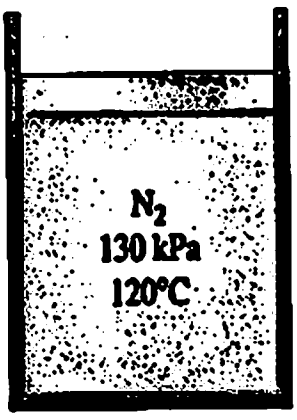


FIGURE P5–21

Step 1 of 4

From the ideal gas specific heat of various gases
For nitrogen
Gas constant is $R = 0.2968 \text{kJ/kg·K}$
Specific heat ratio is $k = 1.4$

Step 2 of 4

Use the conversion factor to convert the temperature from 1°C to 1K

$$1\text{K} = 1^\circ\text{C} + 273$$

$$T_1 = 180 + 273$$

$$= 453\text{K}$$

Calculate the initially volume by using the ideal gas equation

$$P_1V_1 = mRT_1$$

$$V_1 = \frac{mRT}{P_1}$$

Here, P_1 is the pressure at state point 1, V_1 is the volume at state point 1, m is the mass flow rate and T_1 is the initially temperature

Substitute 0.25kg for m, 0.2968kJ/kg·K for R, 453K for T_1 and 130kPa for P_1

$$V_1 = \frac{mRT}{P_1}$$

$$= \frac{(0.25)(0.2968)(453)}{(130)}$$

$$= 0.2586\text{m}^3$$

Step 3 of 4

Calculate the final volume

$$P_2V_2 = mRT_2$$

$$V_2 = \frac{mRT}{P_2}$$

$$= \frac{(0.25)(0.2968)(180+273)}{(80)}$$

$$= 0.420\text{m}^3$$

Step 4 of 4

Calculate the Work done in isothermal process

$$W = P_1V_1 \ln\left(\frac{V_2}{V_1}\right)$$

Substitute 130kPa for P_1, 0.2586m³ for V_1 and 0.420m³ for V_2

$$W = P_1V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$= (130)(0.2586)\ln\left(\frac{0.420}{0.2586}\right)$$

$$= 16.30\text{kJ}$$

Hence, the Work done in isothermal process is $\boxed{16.30\text{kJ}}$

Question: 5.022P

A piston-cylinder device contains 0.15 kg of air initially at 2 MPa and 350°C. The air is first expanded isothermally to 500 kPa, then compressed polytropically with a polytropic exponent of 1.2 to the initial pressure, and finally compressed at the constant pressure to the initial state. Determine the boundary work for each process and the net work of the cycle.

Step 1 of 6

Given

The mass of air $m = 0.15\ \text{kg}$

Initial Pressure $P_1 = 2\ \text{MPa}$

Temperature $T_1 = 350°\text{C}$

State 2 pressure $P_2 = 500\ \text{kPa}$

Final pressure $P_3 = P_1$

The polytrophic process index $n = 1.2$

Question:

A piston-cylinder device contains 0.15 kg of air initially at 2 MPa and 350°C. The air is first expanded isothermally to 500 kPa, then compressed polytropically with a polytropic exponent of 1.2 to the initial pressure, and finally compressed at the constant pressure to the initial state. Determine the boundary work for each process and the net work of the cycle.

Step 1 of 6

Given

The mass of air $m = 0.15\ \text{kg}$

Initial Pressure $P_1 = 2\ \text{MPa}$

Temperature $T_1 = 350°\text{C}$

State 2 pressure $P_2 = 500\ \text{kPa}$

Final pressure $P_3 = P_1$

The polytrophic process index $n = 1.2$

$$W_{1-2} = \frac{(2000)(0.0134)\ln\dfrac{}{0.0134}}{}$$

$$W_{1-2} = 37.18\ \text{kJ}$$

Step 4 of 6

In process 2-3

The boundary work during the process is

$$W_{2-3} = \frac{P_3V_3 - P_2V_2}{1-n}\ \text{kJ}$$

The polytrophic process $PV^n = C$

$$P_2V_2^n = P_3V_3^n$$

$$500(0.0536)^{12} = 2000(V_3)^{12}$$

Volume $V_3 = 0.0169\ \text{m}^3$

$$W_{2-3} = \frac{2000(0.0169) - 500(0.0536)}{1-1.2}\ \text{kJ}$$

$$W_{2-3} = -35\ \text{kJ}$$

Step 5 of 6

For the process3-1

The boundary work in constant pressure process is

$$W_{3-1} = P_1(V_1 - V_3)\ \text{kJ}$$

$$W_{3-1} = 2000(0.0134 - 0.0169)\ \text{kJ}$$

$$W_{1-2} = \frac{(2000)(0.0134)\ln\dfrac{}{0.0134}}{}$$

$$W_{1-2} = 37.18\ \text{kJ}$$

Step 4 of 6

In process 2-3

The boundary work during the process is

$$W_{2-3} = \frac{P_3V_3 - P_2V_2}{1-n}\ \text{kJ}$$

Question: 5.023P

1-kg of water that is initially at 90°C with a quality of 10 percent occupies a spring-loaded piston-cylinder device, such as that in Fig. P5-23. This device is now heated until the pressure rises to 800 kPa and the temperature is 250°C. Determine the total work produced during this process, in kJ.

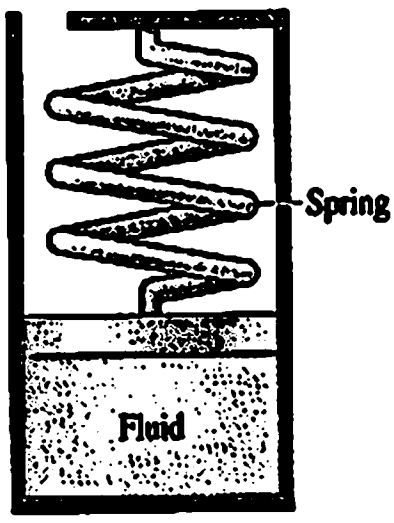


FIGURE P5–23

Step 1 of 4

Given

Mass of water $m = 1\text{kg}$

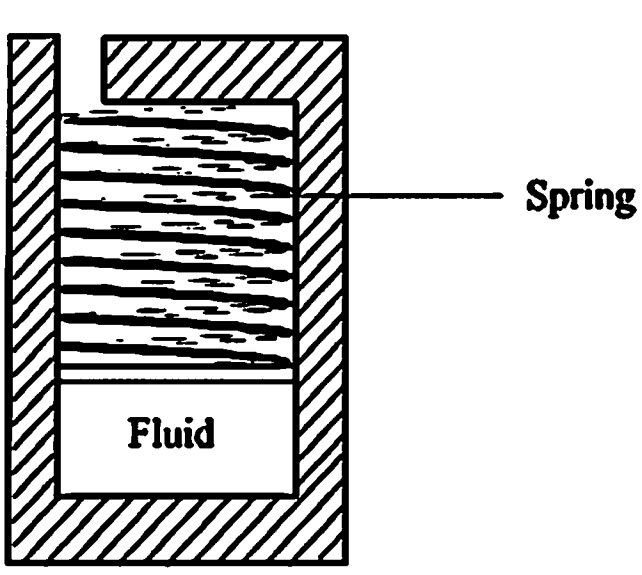
Initial temperature $T_1 = 90°\text{C}$

Dryness fraction $x = 0.1$

The final pressure $P_2 = 800\text{ kPa}$

Final Temperature $T_2 = 250°\text{C}$

Step 2 of 4



Step 3 of 4

From Saturated water Temperature table at $T = 90°\text{C}$

Step 1 of 4

Given

Mass of water $m = 1\text{kg}$

Initial temperature $T_1 = 90°\text{C}$

Dryness fraction $x = 0.1$

The final pressure $P_2 = 800\text{ kPa}$

Final Temperature $T_2 = 250°\text{C}$

Step 2 of 4

Question: 5.024P

1.5-kg water that is initially at 1 MPa and 30 percent quality occupies a spring-loaded piston-cylinder device. This device is now cooled until the water is a saturated liquid at 100°C. Calculate the total work produced during this process, in kJ.

Step 1 of 5

Given:

Mass of the water $m = 1.5\text{kg}$

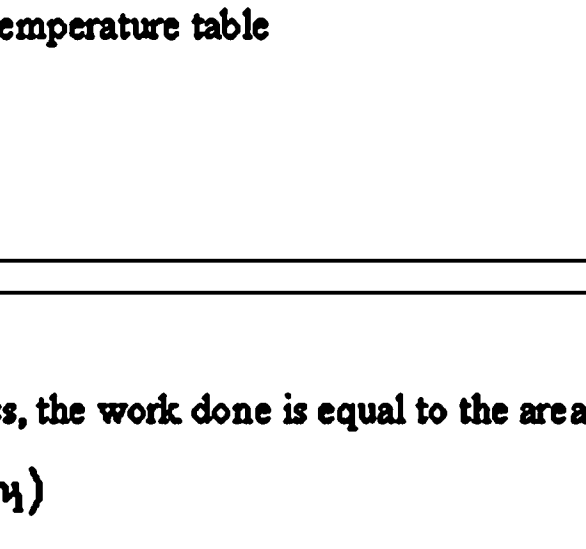
Initially pressure $P_1 = 1\text{MPa}$

Quality of the steam $x = 30\%$

Final temperature $T_2 = 100^\circ\text{C}$

Step 2 of 5

P-V Diagram:



Step 3 of 5

The initial state is saturated mixture

From the saturated water pressure table

At 1MPa

$$v_f = 0.001127\frac{\text{m}^3}{\text{kg}}$$

$$v_g = 0.19436\frac{\text{m}^3}{\text{kg}}$$

$$v_1 = v_f + xv_{fg}$$

$$v_1 = v_f + x(v_g - v_f)$$

$$= 0.001127 + (0.3)(0.19436 - 0.001127)$$

$$= 0.0590\frac{\text{m}^3}{\text{kg}}$$

Step 4 of 5

The final state is saturated liquid

$P_2 = 101.42kPa$

From the saturated water Temperature table

At 100°C

$$v_2 = v_f = 0.001043\frac{\text{m}^3}{\text{kg}}$$

Step 5 of 5

Since this is a linear process, the work done is equal to the area under the process line 1-2

$$W_{b,out} = \left(\frac{P_1 + P_2}{2}\right)(m)(v_2 - v_1)$$

$$= \left(\frac{1000 + 101.42}{2}\right)(1.5)(0.001043 - 0.0590)$$

$$= -47.82\text{kJ}$$

$$\boxed{W_{b,out} = -47.82\text{kJ}}$$

The negative sign shows that the work is done on the system

Question: 5.025P

An ideal gas undergoes two processes in a piston-cylinder device as follows:

1-2 Polytropic compression from T 1 and P 1 with a polytropic exponent n and a compression ratio of r = V1 /V2.

2-3 Constant pressure expansion at P 3 = P 2 until V3 = V1.

(*a*) Sketch the processes on a single P- V diagram.

(*b*) Obtain an expression for the ratio of the compression-to-expansion work as a function of n and r.

(*c*) Find the value of this ratio for values of n = 1.4 and

$r = 6$. *Answers:* (*b*) $\frac{1}{n-1}\left(\frac{1-r^{1-n}}{r-1}\right)$, (*c*) 0.256

We don't have the solution to this problem yet.

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Question: 5.026P

Complete the table below on the basis of the conservation of energy principle for a closed system.

Q in Btu	W out Btu	E 1 Btu	E 2 Btu	m lbm	e 2 – e 1 Btu/lbm
350	-	1020	860	3	-
350	130	550	-	5	-
-	260	600	-	2	150
-500	-	1400	900	7	-
-	-50	1000	-	3	-200

Step 1 of 8

Given:

Q_{in} Btu	W_{out} Btu	E_1 Btu	E_2 Btu	m lbm	$e_2 - e_1$ $\frac{\text{Btu}}{\text{lbm}}$
350	-				
350	130	550	-	5	-
-	260	600	-	2	150
-500	-	1400	900	7	-
-	-50	1000	-	3	-200

Step 2 of 8

The energy balance for a closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{in} - W_{out} = E_2 - E_1 = m(e_2 - e_1)$$

Step 3 of 8

Row 1:

$$Q_{in} - W_{out} = E_2 - E_1$$
$$350 - W_{out} = 860 - 1020$$
$$W_{out} = 510\,\text{Btu}$$
$$Q_{in} - W_{out} = m(e_2 - e_1)$$
$$350 - 510 = 3(e_2 - e_1)$$
$$(e_2 - e_1) = -53.333$$

Step 4 of 8

Row 2:

$$Q_{in} - W_{out} = E_2 - E_1$$
$$350 - 130 = E_2 - 550$$
$$E_2 = 770\,\text{Btu}$$
$$Q_{in} - W_{out} = m(e_2 - e_1)$$
$$350 - 130 = 5(e_2 - e_1)$$
$$(e_2 - e_1) = 44.0\,\frac{\text{Btu}}{\text{lbm}}$$

Step 5 of 8

Row 3:

$$Q_{in} - W_{out} = E_2 - E_1 = m(e_2 - e_1)$$
$$Q_{in} - W_{out} = m(e_2 - e_1)$$
$$Q_{in} - 260 = 2(150)$$
$$Q_{in} = 560\,\text{Btu}$$
$$Q_{in} - W_{out} = E_2 - E_1$$
$$560 - 260 = E_2 - 600$$
$$E_2 = 900\,\text{Btu}$$

Step 6 of 8

Row 4:

$$Q_{in} - W_{out} = E_2 - E_1$$
$$-500 - W_{out} = 900 - 1400$$
$$W_{out} = 0$$
$$Q_{in} - W_{out} = m(e_2 - e_1)$$
$$-500 - 0 = 7(e_2 - e_1)$$
$$(e_2 - e_1) = -71.4\,\frac{\text{Btu}}{\text{lbm}}$$

Step 7 of 8

Row 5:

$$E_2 - E_1 = m(e_2 - e_1)$$
$$E_2 - 1000 = 3(-200)$$
$$E_2 = 400\,\text{Btu}$$
$$Q_{in} - W_{out} = E_2 - E_1$$
$$Q_{in} - (-50) = 400 - 1000$$
$$Q_{in} = -650\,\text{Btu}$$

Step 8 of 8

Application of this equation gives the following completed table:

Q_{in} Btu	W_{out} Btu	E_1 Btu	E_2 Btu	m lbm	$e_2 - e_1$ $\frac{\text{Btu}}{\text{lbm}}$
350	510	1020	860	3	-53.3
350	130	550	770	5	44.0
560	260	600	900	2	150
-500	0	1400	900	7	-71.4
-650	-50	1000	400	3	-200

Question: 5.027P

A closed system like that shown in Fig. P5-27E is operated in an adiabatic manner. First, 15,000 lbf·ft of work are done by this system. Then, work is applied to the stirring device to raise the internal energy of the fluid by 10.28 Btu. Calculate the net increase in the internal energy of this system.

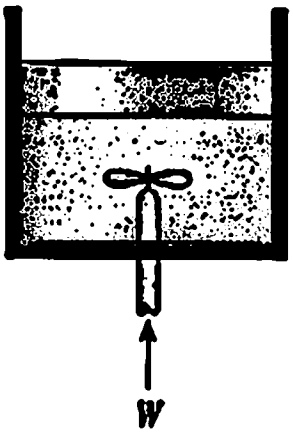


FIGURE P5–27E

Step 1 of 2

Use the conversion factor to convert the work output from $\text{lbf}\cdot\text{ft}$ to Btu

$$1\text{Btu} = 778.169\ \text{lbf}\cdot\text{ft}$$

Find the work output in Btu

$$W_{out} = (15000\ \text{lbf}\cdot\text{ft})\left(\frac{1\text{Btu}}{778.169\ \text{lbf}\cdot\text{ft}}\right)$$

$$= 19.276\text{Btu}$$

Step 2 of 2

Write the energy balance equation for a closed system

$$\dot{E}_{in} - \dot{E}_{out} = \Delta E_{system}$$

$$\dot{W}_{in} - \dot{W}_{out} = \Delta U$$

Here, ΔU is the change in internal energy and $\dot{W}_{in}$ is the work done by the system

Since process is adiabatic, therefore, rate of heat transfer is zero

$$\dot{W}_{in} - \dot{W}_{out} = \Delta U$$

Substitute 19.276Btu for W_{out} and 10.28Btu for $\dot{W}_{in}$

$$\dot{W}_{in} - \dot{W}_{out} = \Delta U$$

$$10.28 - 19.276 = \Delta U$$

$$\Delta U = -8.996\text{Btu}$$

Hence, change in internal energy of the system is $\boxed{-8.996\text{Btu}}$

Question: 5.028P

A closed system undergoes a process in which there is no internal energy change. During this process, the system produces 1.1 ×106 lbf·ft of work. Calculate the heat transfer for this process, in Btu.

Step 1 of 2

Given:

$W = 1.1\times10^{6}\,\text{lbf}\cdot\text{ft}$

Change in internal energy $\Delta U = 0$

Step 2 of 2

Applying the energy balance equation

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} - W_{out} = \Delta U = 0$$

$$(Since\ K\cdot E = P\cdot E = 0)$$

$$Q_{in} = W_{out}$$

$$Q_{in} = 1.1\times10^{6}\,\text{lbf}\cdot\text{ft}\left(\frac{1\text{Btu}}{778.171\text{bf}\cdot\text{ft}}\right)$$

$$= 1413.573\text{Btu}$$

$$\boxed{Q_{in} = 1413.573\text{Btu}}$$

Question: 5.029P

A rigid container equipped with a stirring device contains 1.5 kg of motor oil. Determine the rate of specific energy increase when heat is transferred to the oil at a rate of 1 W, and 1.5 W of power is applied to the stirring device.

Step 1 of 2

Consider the energy balance equation for a system undergoing a process.

$$E_{in} - E_{out} = \Delta E_{system}$$

Here, ΔE_{system} is the change in energy of the system, which include kinetic and potential energies.

Neglect the kinetic and potential energies for the system.

Apply energy balance for the system

$$\begin{aligned}\dot{Q}_{in} + \dot{W}_{in} &= \dot{E}_{in} - \dot{E}_{out} \\ &= \Delta \dot{E}_{system}\end{aligned}$$

Here, $\dot{Q}_{in}$ is the rate of heat transfer and $\dot{W}_{in}$ is the rate of work transfer.

Substitute 1 W for $\dot{Q}_{in}$ and 1.5 W for $\dot{W}_{in}$

$$\begin{aligned}\Delta \dot{E} &= 1 + 1.5 \\ &= 2.5 \text{ W}\end{aligned}$$

Step 2 of 2

Calculate the rate of specific energy increase using the relation

$$\Delta \dot{e} = \frac{\Delta \dot{E}}{m}$$

Here, m is the mass of the motor oil.

Substitute 1.5 kg for m

$$\begin{aligned}\Delta \dot{e} &= \frac{2.5}{1.5} \\ &= 1.67 \text{ J/kg} \cdot \text{s}\end{aligned}$$

Therefore, the rate of specific energy increase is $\boxed{1.667 \text{ J/kg} \cdot \text{s}}$

Question: 5.030P

A well-insulated rigid tank contains 2 kg of a saturated liquid–vapor mixture of water at 150 kPa. Initially, three-quarters of the mass is in the liquid phase. An electric resistor placed in the tank is connected to a 110-V source, and a current of 8 A flows through the resistor when the switch is turned on. Determine how long it will take to vaporize all the liquid in the tank. Also, show the process on a *T*- v diagram with respect to saturation lines.

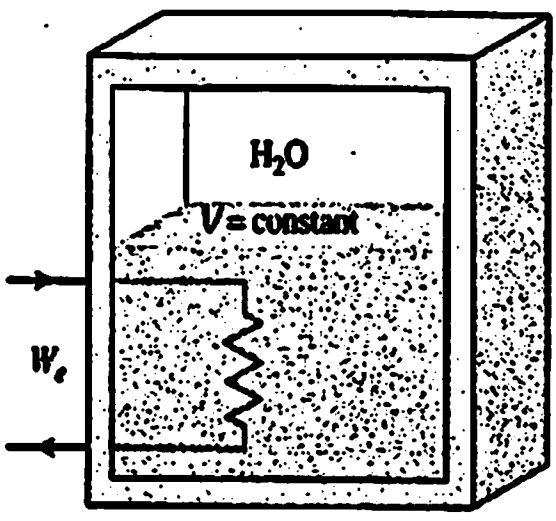


FIGURE P5–30

Step 1 of 6

From the energy balance equation

$$E_{in} - E_{out} = \Delta E_{system}$$
$$W_{e,in} = \Delta U$$
$$= m(u_2 - u_1)$$
$$VI \times \Delta t = m(u_2 - u_1)$$

Here, Voltage supplied by the electrical heater is V ,Current flow through a system is I and mass flow rate of mixture is m

Step 2 of 6

From the saturated water-pressure table

At $P_1 = 150\text{kPa}$

Specific volume of liquid vapor mixture is $v_f = 0.001053\,\text{m}^3/\text{kg}$

Specific volume of vapor is $v_g = 1.1594\,\text{m}^3/\text{kg}$

Internal energy of liquid state is $u_f = 446.97\,\text{kJ/kg}$

Internal energy of liquid vapor mixture is $u_{fg} = 2052.3\,\text{kJ/kg}$

Step 3 of 6

Calculate the Specific volume at state point 1

$$v_1 = v_f + x_1 v_{fg}$$
$$= 0.001053 + [0.25(1.1594 - 0.001053)]$$
$$= 0.290\,\text{m}^3/\text{kg}$$

Step 4 of 6

Calculate the Internal energy at state point 1

$$u_1 = u_f + x_1 u_{fg}$$
$$= 466.97 + [0.25 \times 2052.3]$$
$$= 980.045\,\text{kJ/kg}$$

Since tank is rigid so volume remains constant.

$$v_1 = v_2 = 0.290\,\text{kJ/kg}$$

Step 5 of 6

From the saturated water-pressure table

At $v_1 = v_2 = 0.290\,\frac{\text{kJ}}{\text{kg}}$ and saturated vapor

Which is lies between the $650\,\text{kPa}$ and 700kPa

Use the method of interpolation

$$\frac{0.290 - 0.27278}{0.29260 - 0.27278} = \frac{u_2 - 2571.8}{2569.4 - 2571.8}$$
$$u_2 = 2569.71\,\text{kJ/kg}$$

Step 6 of 6

Calculate the Time required

$$VI\Delta t = m(u_2 - u_1)$$

Substitute $110\,\text{V}$ for V, $8\,\text{A}$ for I , 2kg for m, $2569.71\,\text{kJ/kg}$ for u_2

$$(110 \times 8)\Delta t = (2)(2569.71 - 980.045)(1000)$$
$$\Delta t = 3612.87\,\text{s}$$
$$= \frac{3612.87\,\text{s} \times 1\,\text{min}}{60\,\text{s}}$$
$$= 60.21\,\text{min}$$

Therefore, time required to vaporize all liquid in tank is $\boxed{60.21\,\text{min}}$

Question: 5.031P

Reconsider Prob. 5–30. Using EES (or other) software, investigate the effect of the initial mass of water on the length of time required to completely vaporize the liquid. Let the initial mass vary from 1 to 10 kg. Plot the vaporization time against the initial mass, and discuss the results.

Step 1 of 9

Given:

Mass of the liquid vapor $m = 2\text{kg}$

Pressure of mixture $P = 150\text{kPa}$

Amount of liquid phase $x = 0.25$

Source voltage $V = 110\text{V}$

Source current $I = 8\text{A}$

Mass of the liquid varies from the 1kg to10kg

Step 2 of 9

```
PROCEDURE P2X2(v[1]:P[2],x[2])
Fluid$='Steam_IAPWS'
If v[1] > V_CRIT(Fluid$) then
P[2]=pressure(Fluid$,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Fluid$,v=v[1],x=0)
x[2]=0
End If
End
```

Step 3 of 9

```
"Knowns"
"{m=2 [kg]}"
P[1]=150 [kPa]
y=0.75 "moisture"
Volts=110 [V]
I=8 [amp]
```

Step 4 of 9

```
"Solution"
"Conservation of Energy for the closed tank:"
E_dot_in-E_dot_out=DELTAE_dot
E_dot_in=W_dot_ele  "[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW) "[kW]"
E_dot_out=0 "[kW]"
DELTAE_dot=m*(u[2]-u[1])/DELTAt_s "[kW]"
DELTAt_min=DELTAt_s*convert(s,min) "[min]"
"The quality at state 1 is:"
Fluid$='Steam_IAPWS'
```

Step 5 of 9

```
x[1]=1-y
u[1]=INTENERGY(Fluid$,P=P[1], x=x[1]) "[kJ/kg]"
v[1]=volume(Fluid$,P=P[1], x=x[1]) "[m^3/kg]"
T[1]=temperature(Fluid$,P=P[1], x=x[1]) "[C]"
"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Fluid$,P=P[2], x=x[2]) "[kJ/kg]"
v[2]=volume(Fluid$,P=P[2], x=x[2]) "[m^3/kg]"
T[2]=temperature(Fluid$,P=P[2], x=x[2]) "[C]"
```

Step 6 of 9

Run the new parameter table on manu bar.

Then select the m_kg and Δt(min) then click ok

Vary the mass 1kg to 10kg then solve .

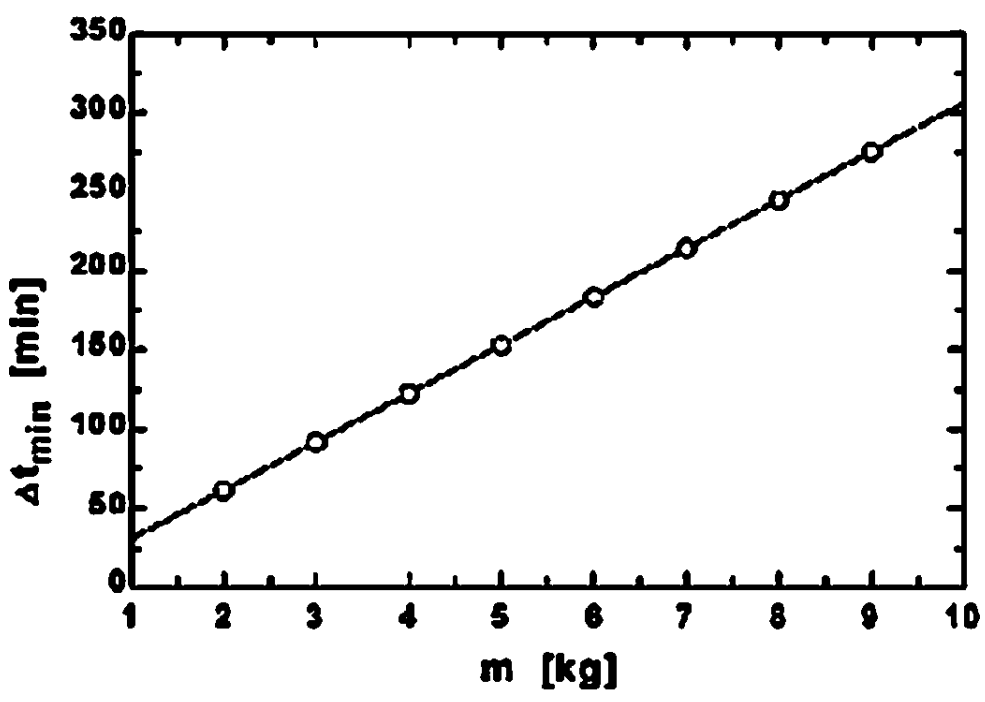
Step 7 of 9

Solution table:

Δt_{min} (min)	m_kg
30.11	1
60.22	2
90.31	3
120.42	4
150.525	5
180.63	6
210.735	7
240.84	8
270.945	9
301.05	10

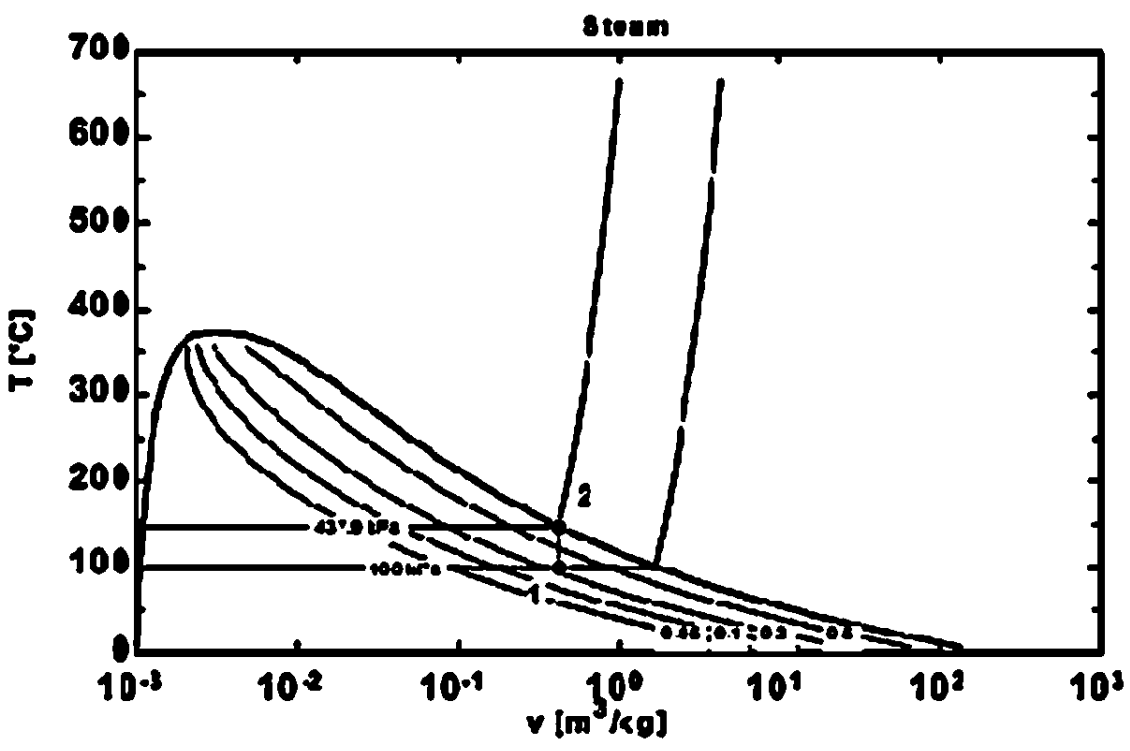
Step 8 of 9

Graph between m_kg and Δt(min)



Step 9 of 9

P-v Daigram:



Question: 5.032P

Saturated water vapor at 200°C is isothermally condensed to a saturated liquid in a piston-cylinder device. Calculate the heat transfer and the work done during this process, in kJ/kg.

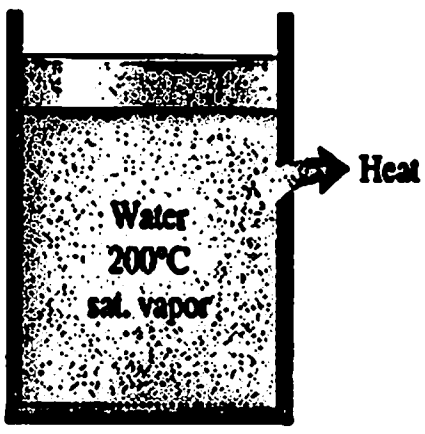


FIGURE P5–32

Step 1 of 5

Apply energy balance to the closed system by neglecting the change in kinetic and potential energies.

$$E_{in} - E_{out} = \Delta E_{system}$$

$$W_{in} - Q_{out} = \Delta U$$
$$W_{in} - Q_{out} = U_2 - U_1$$

Here, W_{in} is the amount of energy transferred as work input, Q_{out} is the amount of energy transferred as heat output and ΔU is change in internal energy.

Step 2 of 5

Represent the equation for a unit mass: $w_{in} - q_{out} = u_2 - u_1$ ..... (1)

Step 3 of 5

At the initial state that is at state 1 the state of steam in the piston cylinder device is in the form of saturated vapor; at the final state that is at state 2 the state of steam is in the form of saturated liquid.

Quote the following values from the saturated water temperature table at temperature $T_1 = 200°C$:

Saturated vapor $P_{sat} = P_1 = P_2 = 1554.9\ \text{kPa}$, Saturated vapor of specific volume $v_1 = v_g = 0.12721\ \text{m}^3/\text{kg}$ and Saturated vapor of specific internal energy $u_1 = u_g = 2594.2\ \text{kJ/kg}$.

Since the condensation process is isothermal process, the temperature remains constant. Hence at temperature $T_1 = T_2 = 200°C$, the following can be stated from the saturated water temperature table:

Saturated liquid of specific volume $v_2 = v_f = 0.001157\ \text{m}^3/\text{kg}$, saturated liquid of specific internal energy $u_2 = u_f = 850.46\ \text{kJ/kg}$

Step 4 of 5

Represent the boundary work done during this condensation process by using the following equation:

$$\begin{aligned} w_{out} &= \int_1^2 P dv \\ &= P(v_2 - v_1) \\ &= 1554.9 \times (0.001157 - 0.12721) \\ &= -195.99\ \text{kJ/kg} \end{aligned}$$

The negative sign indicates work input to the system. Therefore, the amount of work input w_{in} is $\boxed{195.99\ \text{kJ/kg}}$.

Step 5 of 5

Calculate the heat transfer by substituting the known values in equation (1).

$$\begin{aligned} w_{in} - q_{out} &= u_2 - u_1 \\ 195.99 - q_{out} &= 850.46 - 2594.2 \\ 195.99 - q_{out} &= -1743.74 \\ q_{out} &= 1939.73\ \text{kJ/kg} \end{aligned}$$

Therefore, heat transfer from the system is $\boxed{1939.73\ \text{kJ/kg}}$.

Question: 5.033P

A rigid 10-L vessel initially contains a mixture of liquid water and vapor at 100°C with 12.3 percent quality. The mixture is then heated until its temperature is 150°C. Calculate the heat transfer required for this process.

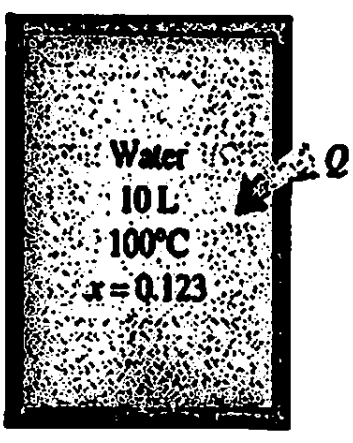


FIGURE P5–33

Step 1 of 8

Write the energy balance for closed rigid tank system

$$E_{in} - E_{out} = \Delta E_{system}$$

Since kinetic and potential energy are negligible, therefore, KE = PE = 0

$$Q_{in} = \Delta U$$
$$= m(u_2 - u_1)$$

Here, m is the mass flow rate of the mixture, u_2 and u_1 are the internal energy at state points 2 and 1 respectively.

Step 2 of 8

From the saturated water temperature table

At Inlet temperature $T_1 = 100^\circ C$ and quality of the mixture is $x_1 = 0.123$

Specific volume of the liquid is $v_f = 0.001043\ m^3/kg$

Specific volume of the liquid vapor mixture is $v_g = 1.6720\ m^3/kg$

Calculate the Specific volume at state 1

$$v_1 = v_f + x_1 v_{fg}$$
$$= v_f + x_1 (v_g - v_f)$$

Substitute $0.001043\ m^3/kg$ for v_f, $1.6720\ m^3/kg$ for v_g

$$v_1 = 0.001043 + [0.123 \times (1.6720 - 0.001043)]$$
$$= 0.2065 \frac{m^3}{kg}$$

Step 3 of 8

From the saturated water temperature table

At $T_1 = 100^\circ C$ and $x_1 = 0.123$

$$u_f = 419.06 \frac{kJ}{kg}$$

$$u_{fg} = 2087.0 \frac{kJ}{kg}$$

Calculate the Internal energy at state 1

$$u_1 = u_f + x_1 u_{fg}$$
$$= 419.06 + [0.123 \times 2087]$$
$$= 675.761 kJ/kg$$

Step 4 of 8

Since tank is rigid so volume remains constant.

$$v_2 = v_1 = 0.2065 m^3/kg$$

From the saturated water temperature table

At $T_2 = 150^\circ C$

$$v_f = 0.001091 \frac{m^3}{kg}$$

$$v_g = 0.39248 \frac{m^3}{kg}$$

Step 5 of 8

Find the Quality of the mixture at final state

$$v_2 = v_f + x_2 v_{fg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}}$$
$$= \frac{0.2065 - 0.001091}{0.39248 - 0.001091}$$
$$= 0.524$$

Step 6 of 8

From the saturated water temperature table

At $T_2 = 150^\circ C$

$$u_f = 631.66 \frac{kJ}{kg}$$

$$u_{fg} = 1927.4 \frac{kJ}{kg}$$

Calculate the internal energy at final state point 2

$$u_2 = u_f + x_2 u_{fg}$$
$$= 631.66 + [0.524 \times 1927.4]$$
$$= 1641.61 kJ/kg$$

Step 7 of 8

Calculate the mass in the system

$$m = \frac{V_1}{v_1}$$
$$= \frac{10 \times 10^{-3}}{0.2065}$$
$$= 0.04842\ kg$$

Step 8 of 8

Calculate the heat transfer

$$Q_{in} = m(u_2 - u_1)$$

Substitute $1641.61 kJ/kg$ for u_2, $675.761 kJ/kg$ for u_1 and $0.04842\ kg$ for m

$$Q_{in} = m(u_2 - u_1)$$
$$= (0.04842)(1641.61 - 675.761)$$
$$= 46.76 kJ$$

Therefore, the heat transfer is $\boxed{46.76 kJ}$

Question: 5.034P

A fixed mass of saturated water vapor at 300 kPa is isothermally cooled until it is a saturated liquid. Calculate the amount of heat rejected during this process, in kJ/kg.

Step 1 of 6

Given:
Process: isothermal
Saturated water vapor pressure $P_1 = P_2 = 300\text{kPa}$

Step 2 of 6

The energy balance equation for the closed system can be expressed as:-

$$W_{b,in} - Q_{out} = \Delta U$$

$$Q_{out} = W_{b,in} - m(u_2 - u_1)$$

Step 3 of 6

From the saturated water pressure table
At $P_1 = P_2 = 300\text{kPa}$ and $x_1 = 1$
Since water initially saturated state.

$$T_1 = T_2 = 133.52^\circ\text{C}$$

$$v_1 = v_g = 0.60582\frac{\text{m}^3}{\text{kg}}$$

$$u_1 = u_g = 2543.2\frac{\text{kJ}}{\text{kg}}$$

Step 4 of 6

We know that
Final state is saturated liquid, So that

$$x_2 = 0$$

From the saturated water pressure table
At $P_1 = P_2 = 300\text{kPa}$ and $x_2 = 1$

$$v_2 = v_f = 0.001073\frac{\text{m}^3}{\text{kg}}$$

$$u_2 = u_f = 561.11\frac{\text{kJ}}{\text{kg}}$$

Step 5 of 6

The work done during this process is

$$W_{b,out} = \int_1^2 PdV = P(v_2 - v_1)$$
$$= 300(0.001073 - 0.60582)$$
$$= -181.4241\frac{\text{kJ}}{\text{kg}}$$

Negative sign indicated that on the system.
Now take the entire process as the system, so work done by the system so take the positive sign

$$W_{b,in} = 181.4241\frac{\text{kJ}}{\text{kg}}$$

Step 6 of 6

Substituting the energy balance equation, we get

$$q_{out} = W_{b,in} - (u_2 - u_1)$$
$$= W_{b,in} + u_{fg}$$
$$= 181.424 + 1982.1$$
$$= 2163.524\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{q_{out} = 2163.524\frac{\text{kJ}}{\text{kg}}}$$

Question: 5.035P

Saturated water vapor in a closed system is condensed by cooling it at constant pressure to a saturated liquid at 40 kPa. Determine the heat transferred and the work done during this process, in kJ/kg.

Step 1 of 2

Obtain the properties from saturated water pressure table at $P_1 = P_2 = 40\text{kPa}$

Specific volume of saturated vapor at state point 1 is $v_1 = v_g = 3.9933\text{m}^3/\text{kg}$

Specific volume of saturated liquid $v_2 = v_f = 0.001026\text{m}^3/\text{kg}$

Internal energy of vaporization $u_{fg} = 2158.8\text{kJ/kg}$

Calculate the Work done for closed system

$$W_{b,out} = \int_1^2 P dV$$
$$= P(v_2 - v_1)$$

Substitute 40kPa for P, $0.001026\text{m}^3/\text{kg}$ for v_2 and $3.9933\text{m}^3/\text{kg}$ for v_1

$$W_{b,out} = 40(0.001026 - 3.9933)$$
$$= -159.7\text{kJ/kg}$$

Negative sign indicates work done on the system

Thus, Work done on closed system is $\boxed{159.7\text{kJ/kg}}$

Step 2 of 2

Calculate the heat transfer for closed system

$$Q_{out} = W_{b,in} - m(u_2 - u_1)$$
$$q_{out} = W_{b,in} - (u_2 - u_1)$$
$$= W_{b,in} - (u_2 - u_1)$$

Substitute 159.7kJ/kg for $W_{b,in}$ and 2158.8kJ/kg for u_{fg}

$$q_{out} = W_{b,in} + u_{fg}$$
$$= 159.7 + 2158.8$$
$$= 2318.5\text{kJ/kg}$$

Therefore, heat transfer for closed system is $\boxed{2318.5\text{kJ/kg}}$

Question: 5.036P pg 1

An insulated piston-cylinder device contains 5 Lof saturated liquid water at a constant pressure of 175 kPa. Water is stirred by a paddle wheel while a current of 8 A flows for 45 min through a resistor placed in the water. If one-half of the liquid is evaporated during this constant-pressure process and the paddle-wheel work amounts to 400 kJ, determine the voltage of the source. Also, show the process on a *P*-v diagram with respect to saturation lines.

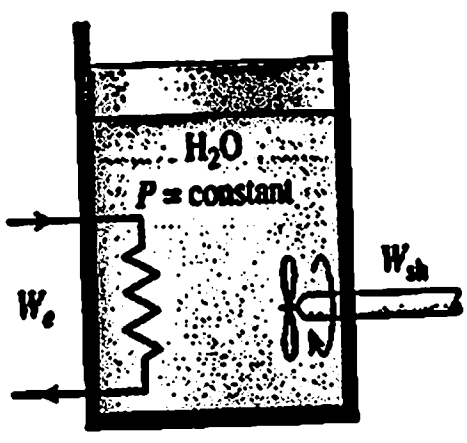
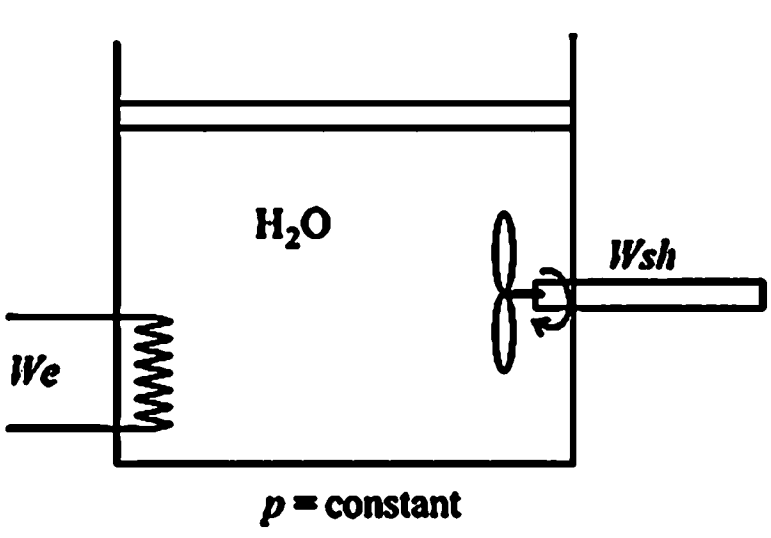


FIGURE P5–36

Step 1 of 7

Draw the diagram for the system of the piston cylinder arrangement.



Step 2 of 7

Obtain the properties of saturated liquid water from, "TABLE A-5 Saturated Water-Pressure table" in the textbook.

At pressure 175 kPa,

Specific enthalpy of liquid, $h_f = h_1 = 487.01\,\text{kJ/kg}$.

Specific enthalpy of evaporation, $h_{fg} = 2213.1\,\text{kJ/kg}$.

Specific volume of liquid, $v_f = v_1 = 0.001057\,\text{m}^3/\text{kg}$

Step 3 of 7

Calculate the total mass of water.

$$m = \frac{V}{v_1}$$

Here, total volume of the water is V .

Substitute $0.001057\,\text{m}^3/\text{kg}$ for v_1 , and $\left(5\times10^{-3}\right)\text{m}^3$ for *V*.

$$m = \frac{5\times10^{-3}}{0.001057}$$
$$= 4.730\,\text{kg}$$

Determine the quality of steam at final state.

$$x_2 = \frac{V_v}{V}$$

Here, volume of the evaporated liquid is V_v ,

◊

Substitute 2.5 L for V_v , and 5L for V ◊

$$x_2 = \frac{2.5}{5}$$
$$= 0.5\,\text{dry}$$

Step 4 of 7

Determine the enthalpy at final state using the relation:

$$h_2 = h_f + x_2\left(h_{fg}\right)$$

Substitute $487.01\,\text{kJ/kg}$ for h_f , $2213.1\,\text{kJ/kg}$ for h_{fg} , and 0.5 for x_2 .

$$h_2 = 487.01 + 0.5(2213.1)$$
$$= 1593.56\,\text{kJ/kg}$$

Write the expression to calculate the enthalpy change within the system.

$$\Delta H = m\left(h_2 - h_1\right)$$

Here, the specific enthalpy at the initial state is h_1 , specific enthalpy at final state is h_2 , mass of the water is m .

Substitute $487.01\,\text{kJ/kg}$ for h_1 , $1593.56\,\text{kJ/kg}$ for h_2 , and $4.730\,\text{kg}$ for m .

$$\Delta H = 4.730(1593.56 - 487.01)$$
$$= 5234.389\,\text{kJ}$$

Step 5 of 7

Express the energy balance for any system undergoing any kind of process.

$$E_{in} - E_{out} = \Delta E_{system}$$

Neglect the changes in potential and kinetic energies:

Express the energy balance for any system undergoing any kind of process.

$$E_{in} - E_{out} = \Delta E_{system}$$

Neglect the changes in potential and kinetic energies:

$$\Delta PE = \Delta KE = Q = 0$$

Rewrite the equation as follows:

$$W_{in} = W_{out} + \Delta U$$

Here, the input work is W_{in} , output work is W_{out} and change in internal energy is ΔU .

Substitute ΔH for $W_{out} + \Delta U$.

$$W_{in} = \Delta H \qquad \left[\because \Delta U + W = \Delta H = m(h_2 - h_1)\right]$$

Rewrite the expression introducing the paddle and electrical works in the place of input work respectively.

$$W_{elec} + W_{sh} = \Delta H \quad \text{..... (1)}$$

Step 6 of 7

Calculate the work supplied in the form of electrical energy

$$W_{elec} = VI\Delta t$$

Here, voltage is , current is *I* and time is *t*.

Substitute $VI\Delta t$ for W_{elec} in the equation (1).

$$VI\Delta t + W_{sh} = \Delta H$$

Substitute 8 A for I, (45×60)sec for Δt , 400 kJ for W_{sh} , $5234.389\,\text{kJ}$ for ΔH and calculate the voltage.

$$(V\times 8\times 45\times 60) + 400 = 5234.389$$

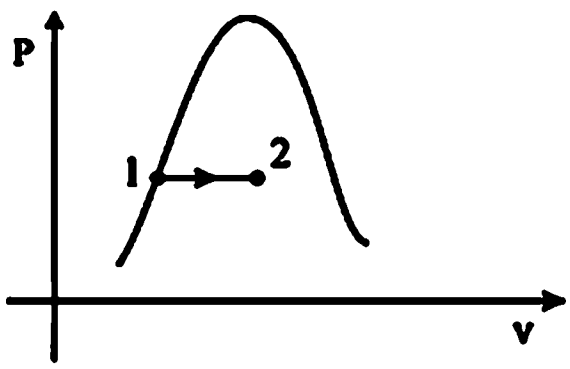
$$V = \frac{5234.389 - 400}{8\times 45\times 60}\times 1000$$

$$= 223.8\,\text{V}$$

Therefore, the voltage of the source is $\boxed{223.8\,\text{V}}$.

Step 7 of 7

Sketch the process on the p-V diagram.



Question: 5.037P

A piston-cylinder device initially contains steam at 200 kPa, 200°C, and 0.4 m3. At this state, a linear spring ($F \propto x$) is touching the piston but exerts no force on it. Heat is now slowly transferred to the steam, causing the pressure and the volume to rise to 250 kPa and 0.6 m3, respectively. Show the process on a *P*-*v* diagram with respect to saturation lines and determine (*a*) the final temperature, (*b*) the work done by the steam, and (*c*) the total heat transferred.

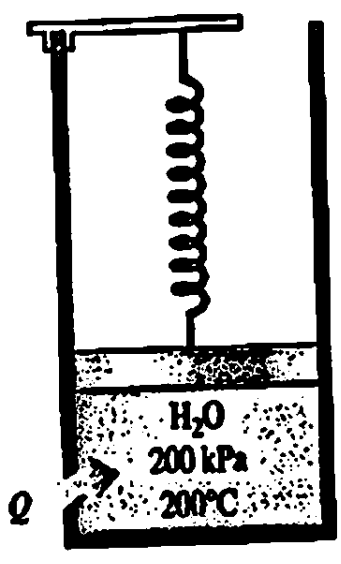


FIGURE P5–37

Step 1 of 8

Given:

Initially pressure $P_1 = 200\text{kPa}$

Initially temperature $T_1 = 200^\circ\text{C}$

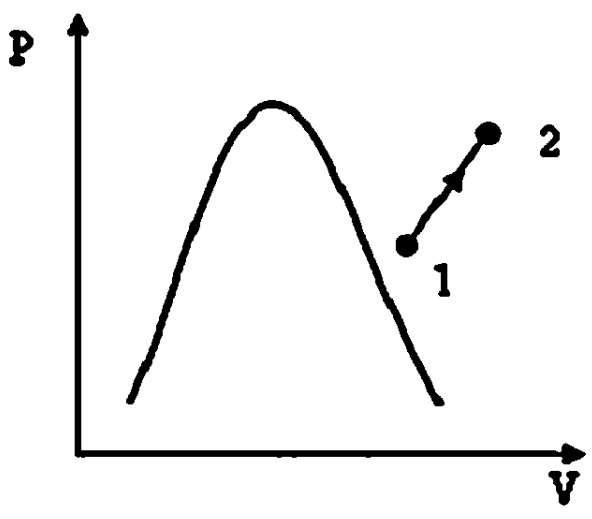
Initially Volume $V_1 = 0.4\text{m}^3$

Final pressure $P_2 = 250\text{kPa}$

Final volume $V_2 = 0.6\text{m}^3$

Step 2 of 8

P-V Diagram:



Step 3 of 8

a) The energy balance for this stationary closed system can be expressed as
From the superheated water

$P_1 = 200\text{kPa}$, and $T_1 = 200^\circ\text{C}$

$$v_1 = 1.08049\frac{\text{m}^3}{\text{kg}}$$

$$u_1 = 2654.6\frac{\text{kJ}}{\text{kg}}$$

Step 4 of 8

Mass of the system is

$$m = \frac{V_1}{v_1}$$
$$= \frac{0.4}{1.08049}$$
$$= 0.3702\text{kg}$$

Step 5 of 8

Final specific volume is

$$v_2 = \frac{V_2}{m}$$
$$= \frac{0.6}{0.3702}$$
$$= 1.621\frac{\text{m}^3}{\text{kg}}$$

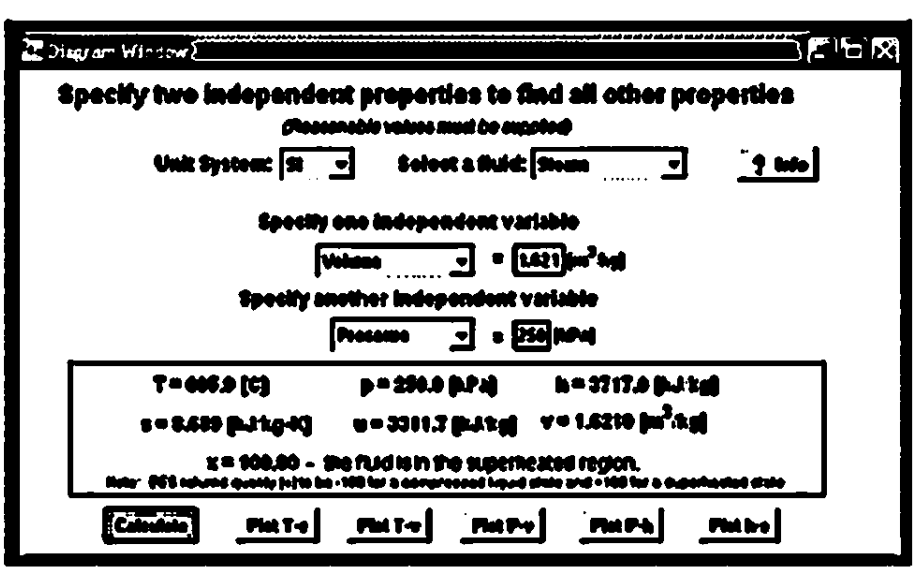
Step 6 of 8

We have,

$P_2 = 250\text{kPa}$

$$v_2 = 1.621\frac{\text{m}^3}{\text{kg}}$$

Using method of interpolation or EES



We get

$T_2 = 606.9^\circ\text{C}$

$\boxed{T_2 = 606^\circ\text{C}}$

$$u_2 = 3311.7\frac{\text{kJ}}{\text{kg}}$$

Step 7 of 8

(b) The pressure of the gas changes linearly with volume, and thus the prcess curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus

$$W_b = \frac{(P_1 + P_2)}{2}(V_2 - V_1)$$
$$= \frac{(200 + 250)}{2}(0.6 - 0.4)$$
$$= 45\text{kJ}$$
$$\boxed{W_b = 45\text{kJ}}$$

Step 8 of 8

(c) From the energy balance, we have

$$Q_{in} = m(u_2 - u_1) + W_b$$
$$Q_{in} = [(0.3702)(3311.002 - 2654.6)] + 45$$
$$= 288\text{kJ}$$
$$\boxed{Q_{in} = 288\text{kJ}}$$

Question: 5.038P

Reconsider Prob. 5–37. Using EES (or other) software, investigate the effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer. Let the initial temperature vary from 150 to 250°C. Plot the final results against the initial temperature, and discuss the results.

Step 1 of 9

Given:

Initially pressure $P_1 = 200\text{kPa}$

Initially temperature varies from $T_1 = 150^\circ\text{C to } 250^\circ\text{C}$

Initially Volume $V_1 = 0.4\,\text{m}^3$

Final pressure $P_2 = 250\text{kPa}$

Final volume $V_2 = 0.6\,\text{m}^3$

Step 2 of 9

"The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"
m[2]=m[1]

Step 3 of 9

"The conservation of energy for the closed system is"
"E_in - E_out = DeltaE, neglect DeltaKE and DeltaPE for the system"
Q_in - W_out = m[1]*(u[2]-u[1])
DELTAU=m[1]*(u[2]-u[1])

"Input Data"
P[1]=200 [kPa]
V[1]=0.4 [m^3]
"T[1]=200 [C]"
P[2]=250 [kPa]
V[2]=0.6 [m^3]

Step 4 of 9

Fluid$='Steam_IAPWS'

m[1]=V[1]/spvol[1]
spvol[1]=volume(Fluid$,T=T[1], P=P[1])
u[1]=intenergy(Fluid$, T=T[1], P=P[1])
spvol[2]=V[2]/m[2]

Step 5 of 9

"The final temperature is:"
T[2]=temperature(Fluid$,P=P[2],v=spvol[2])

u[2]=intenergy(Fluid$, P=P[2], T=T[2])
Wnet_other = 0
W_out=Wnet_other + W_b
"W_b = integral of P[2]*dV[2] for 0.4<V[2]<0.6 and is given by:"
W_b=P[1]*(V[2]-V[1])+Spring_const/2*(V[2]-V[1])^2

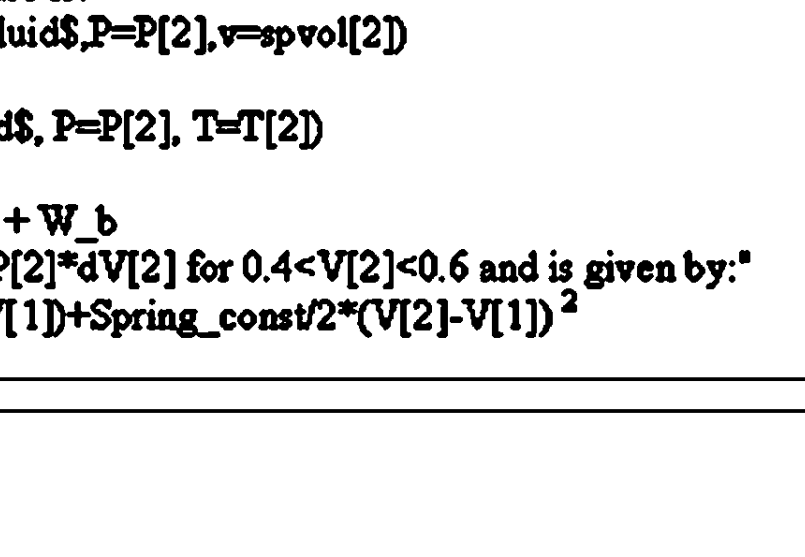
Step 6 of 9

Solution table:

1..11	Q_{in}	T_1 [C]	T_2 [C]	W_{out}
Run 1	281.5	150	508.2	45
Run 2	282.8	160	527.9	45
Run 3	284.2	170	547.5	45
Run 4	285.5	180	567	45
Run 5	286.9	190	586.4	45
Run 6	288.3	200	605.8	45
Run 7	289.8	210	625	45
Run 8	291.2	220	644.3	45
Run 9	292.7	230	663.4	45
Run 10	294.2	240	682.6	45
Run 11	295.7	250	701.7	45

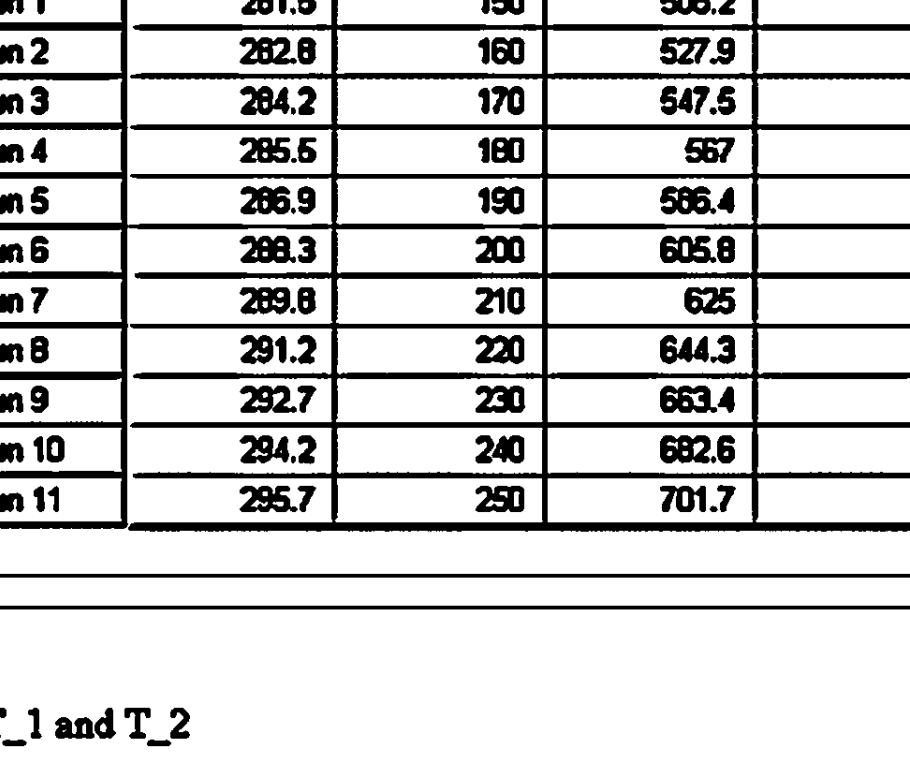
Step 7 of 9

Graph between T_1 and T_2



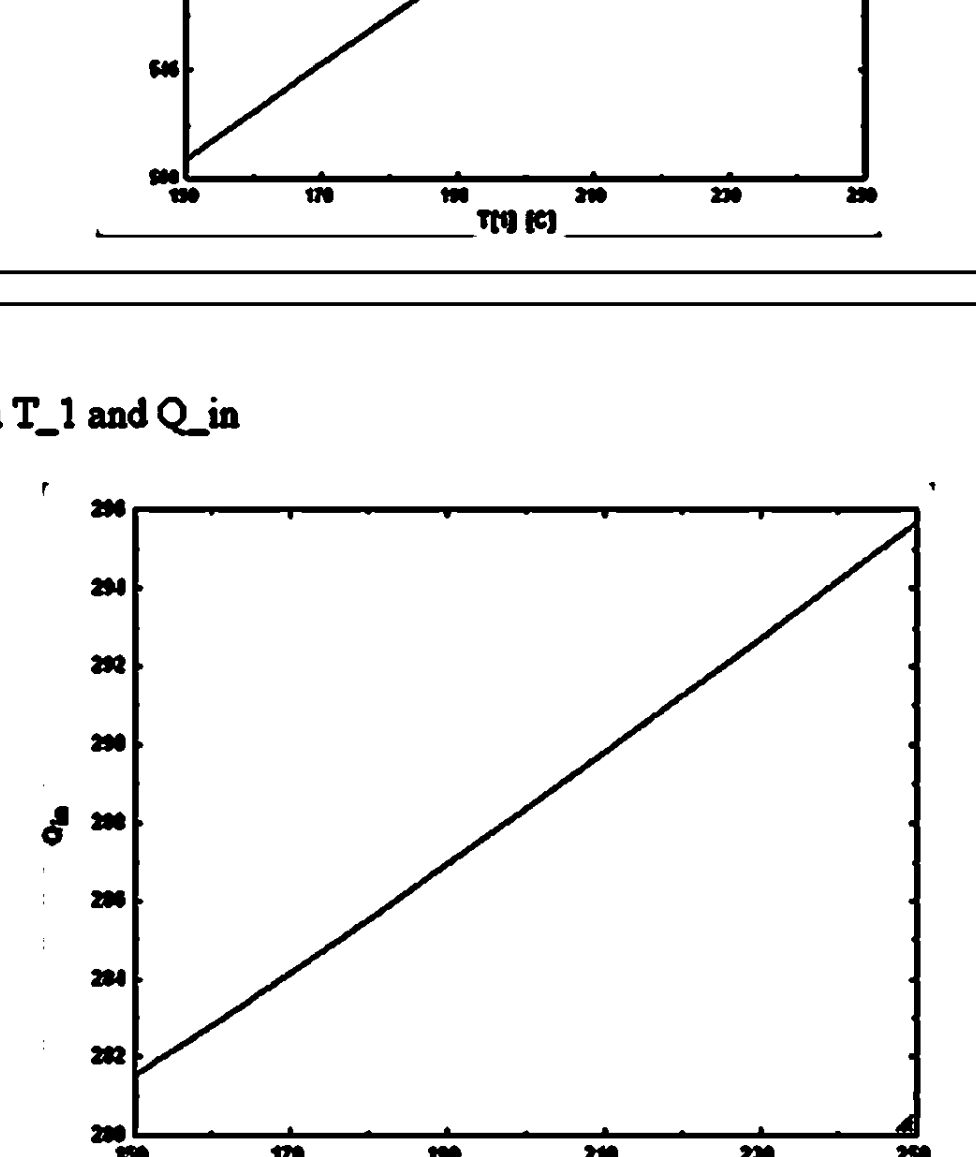
Step 8 of 9

Graph between T_1 and Q_in



Step 9 of 9

Graph between pressure and volume



Question: 5.039P

A 30-L electrical radiator containing heating oil is placed in a 50-m3 room. Both the room and the oil in the radiator are initially at 10°C. The radiator with a rating of 1.8 kW is now turned on. At the same time, heat is lost from the room at an average rate of 0.35 kJ/s. After some time, the average temperature is measured to be 20°C for the air in the room, and 50°C for the oil in the radiator. Taking the density and the specific heat of the oil to be 950 kg/m3 and 2.2 kJ/kg·°C, respectively, determine how long the heater is kept on. Assume the room is well-sealed so that there are no air leaks.

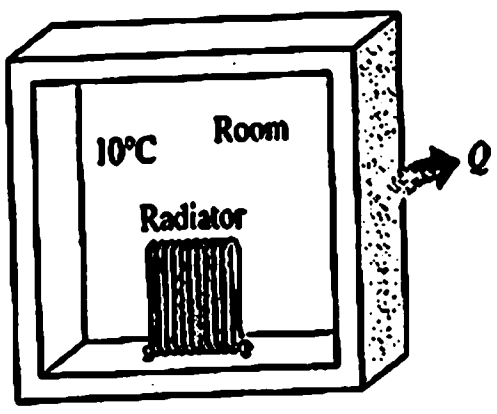


FIGURE P5–39

Step 1 of 5

Given

Volume of room $V = 50\ \text{m}^3$

Initial temperature of radiator $T_{1air} = 10°\text{C}$

Capacity of radiator 30L

Power input of radiator $W_{in} = 1.8\ \text{kW}$

Heat lost $\dot{Q}_{out} = 0.35\ \text{kJ/s}$

Final temperature of radiator $T_{2air} = 20°\text{C}$

Temperature of oil $T_{2oil} = 50°\text{C}$

Density of oil $\rho = 950\ \text{kg/m}^3$

Specific heat of oil $c = 2.2\ \text{kJ/kg.}^0\text{C}$

Step 2 of 5

From table ideal gases table properties of air are

$R = 0.287\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$Q_{in} - W_{out} = \Delta U$

$Q_{in} - W_{out} = m(u_2 - u_1)$

$\left(\dot{Q}_{in} - \dot{W}_{out}\right)\Delta t = mc_v(T_2 - T_1)_{air} + mc(T_2 - T_1)_{oil}$

Step 3 of 5

The mass of oil is determined by

$m_{oil} = \rho_{oil} V_{oil}$

$= 950 \times \left(30 \times 10^{-3}\right)$

$m_{oil} = 28.5\ \text{kg}$

Step 4 of 5

Mass of air can be obtained by using ideal gas equation

$$m_{air} = \frac{P_1 V}{R T_1}$$

$$= \frac{100 \times (50)}{0.287 \times (10 + 273)}$$

$m_{air} = 61.56\ \text{kg}$

Step 5 of 5

Therefore substituting all the values in the equation

$\left(\dot{Q}_{in} - \dot{W}_{out}\right)\Delta t = (mc_v)_{air}(T_2 - T_1)_{air} + (mc)_{oil}(T_2 - T_1)_{oil}$

$(1.8 - 0.35)\Delta t = (61.56)(0.718)(20 - 10) + (28.5)(2.2)(50 - 10)$

$1.45 \times \Delta t = 442.0008 + 2508$

$\Delta t = 2034.48\ \text{sec}$

$$\Delta t = 2034.48 \times \frac{1}{60}$$

$\boxed{\Delta t = 33.9\ \text{min}}$

Question: 5.040P

Steam at 75 kPa and 8 percent quality is contained in a spring-loaded piston-cylinder device, as shown in Fig. P5-40, with an initial volume of 2 m3. Steam is now heated until its volume is 5 m3 and its pressure is 225 kPa. Determine the heat transferred to and the work produced by the steam during this process.

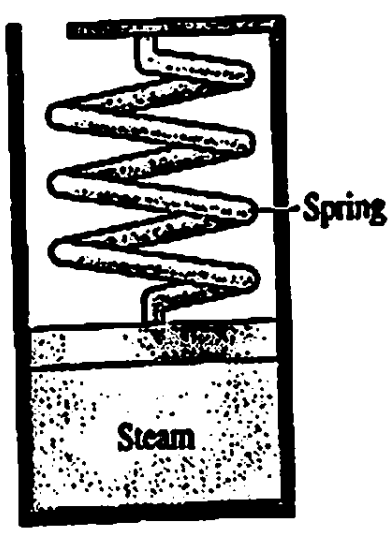
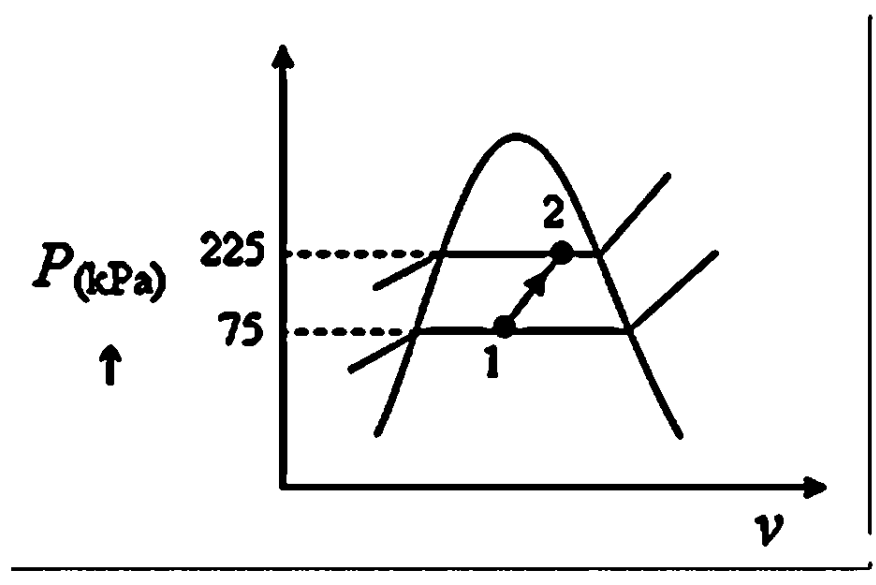


FIGURE P5–40

Step 1 of 11

Draw 1-2 process on *P-V* diagram is as shown below:



Step 2 of 11

Write the energy balance for this stationary closed system

$$Q_{in} - W_{b,out} = \Delta U$$
$$= m(u_2 - u_1)$$
$$Q_{in} = W_{b,out} + m(u_2 - u_1)$$

Here, Heat input to the system is Q_{in}, work produced by the system is $W_{b,out}$, mass flow rate of the system is m and u_1 and u_2 are internal energy of the system at state point 1 and 2 respectively.

Step 3 of 11

Obtain the properties from "saturated water pressure table" at pressure P_1=75 kPa

Specific volume of saturated liquid is $v_f = 0.001037\,m^3/kg$

Specific volume of saturated vapor is $v_g = 2.2173\,m^3/kg$

Internal energy of saturated liquid is $u_f = 384.36 kJ/kg$

Internal energy of the liquid vapor mixture is $u_{fg} = 2111.8 kJ/kg$

Step 4 of 11

Calculate the Specific volume at state 1

$$v_1 = v_f + x_1 v_{fg}$$
$$= 0.001037 + [0.08 \times (2.2172 - 0.001037)]$$
$$= 0.17833\,m^3/kg$$

Step 5 of 11

Calculate Internal energy at state 1

$$u_1 = u_f + x_1 u_{fg}$$
$$= 384.36 + [0.08 \times (2111.8)]$$
$$= 553.304 kJ/kg$$

Step 6 of 11

Calculate the mass of water

$$m = \frac{V_1}{v_1}$$
$$= \frac{2}{0.17833}$$
$$= 11.22 kg$$

Step 7 of 11

Calculate the final specific volume

$$v_2 = \frac{V_2}{m}$$
$$= \frac{5}{11.22}$$
$$= 0.445633\,m^3/kg$$

Obtain the properties from saturated water pressure table at P_2=225kPa

Specific volume of saturated liquid is $v_f = 0.001064\,m^3/kg$

Specific volume of saturated vapor is $v_g = 0.79329\,m^3/kg$

Internal energy of saturated liquid is $u_f = 520.47 kJ/kg$

Internal energy of liquid vapor mixture is $u_{fg} = 2012.7 kJ/kg$

Step 8 of 11

Calculate the dryness fraction at state 2 by using specific volume at state 2

$$v_2 = v_f + x_2 v_{fg}$$
$$x_2 = \frac{v_2 - v_f}{v_{fg}}$$
$$= \frac{0.445633 - 0.001064}{(0.79329 - 0.001064)}$$
$$= 0.5612$$

Step 9 of 11

Calculate the internal energy

$$u_2 = u_f + x_2 u_{fg}$$
$$= 520.47 + [0.5612 \times 2012.7]$$
$$= 1649.997 kJ/kg$$

Step 10 of 11

Calculate the work done is equal to the area under the process line 1-2

$$W_{b,out} = \frac{(P_1 + P_2)(V_2 - V_1)}{2}$$
$$= \frac{(75 + 225)(5 - 2)}{2}$$
$$= 450 kJ$$

Therefore, work done processes 1-2 is $\boxed{450 kJ}$

Step 11 of 11

Calculate the heat transfer

$$Q_{in} = W_{b,out} + m(u_2 - u_1)$$
$$= 450 + [11.22(1649.997 - 553.304)]$$
$$= 12,754.9 kJ$$

Therefore, the heat transfer is $\boxed{12,754.9 kJ}$

Question: 5.041P

Refrigerant-134a at 600 kPa and 150°C is contained in a spring-loaded piston-cylinder device with an initial volume of 0.3 m3. The refrigerant is now cooled until its temperature is -30°C and its volume is 0.1 m3. Determine the heat transferred to and the work produced by the refrigerant during this process.

Step 1 of 6

Given

Initial pressure $P_1 = 600$ kPa

Initial Temperature $T_1 = 150°C$

Initial Volume $V_1 = 0.3\ m^3$

Final temperature $T_2 = -30°C$

Final Volume $V_2 = 0.1\ m^3$

Step 2 of 6

Energy balance for any system undergoing any kind of process was expressed as

$$E_{in} - E_{out} = \Delta E_{system} \quad \text{(kJ)} \quad \text{----------(1)}$$

In the process $\Delta PE = \Delta KE = 0$

Therefore equation (1) $-Q_{out} + W_{in} = \Delta U$

$$-Q_{out} = -W_{in} + \Delta U \qquad (\text{since } \Delta U = m(u_2 - u_1))$$

$$Q_{out} = W_{in} - m(u_2 - u_1)$$

Step 3 of 6

From Superheated Refrigerant R-134a table

At $P_1 = 600$ kPa , $T_1 = 150°C$

The specific volume $v_1 = 0.055522\ m^3/kg$

Internal energy $u_1 = 357.96$ kJ/kg

The mass of water is determined by

$$m = \frac{V}{v_1} = \frac{0.3}{0.055522} = 5.4\ kg$$

Therefore the specific volume is expressed as $v_2 = \frac{V_2}{m}\ m^3/kg$

$$v_2 = \frac{0.1}{5.4} = 0.0185\ m^3/kg$$

Step 4 of 6

From Saturated Refrigerant-134a temperature table

At temperature $T_2 = -30°C$

$$v_2 = v_f + x_2 v_{fg}$$

$$0.0185 = 0.0007203 + x_2(0.225)$$

$$x_2 = 0.079$$

Enthalpy $u_2 = u_f + x_2 u_{fg}$

$$u_2 = 12.59 + 0.079(200.52)$$

$$u_2 = 28.43\ kJ/kg$$

Final pressure $P_2 = 84.4$ kPa

Step 5 of 6

The work done is expressed as the area under the curve

The area under the curve is expressed as

$$W = \frac{P_1 + P_2}{2}(V_2 - V_1)$$

$$W = \frac{600 + 84.4}{2}(0.1 - 0.3)$$

$\boxed{W_{in} = -68.44\ kJ}$ (Negative sign for work done on the system)

Step 6 of 6

Therefore heat supplied to the system

$$Q_{out} = 68.44 - 5.4(28.43 - 357.96)$$

$$Q_{out} = 68.44 - 5.4(28.43 - 357.96)$$

$$\boxed{Q_{out} = 1847.9\ kJ}$$

Question: 5.042P

Saturated R-134a vapor at 100°F is condensed at constant pressure to a saturated liquid in a closed piston-cylinder system. Calculate the heat transfer and work done during this process, in Btu/lbm.

Step 1 of 5

Given

Temperature $T_1 = 100°F$

Step 2 of 5

Saturated vapor condensed to saturated liquid at constant pressure

Energy balance for any system undergoing any kind of process was expressed as

$$E_{in} - E_{out} = \Delta E_{system} \quad \text{(kJ)} \text{----------(1)}$$

In the process $\Delta PE = \Delta KE = 0$

Therefore equation (1) $\quad -Q_{out} + W_{in} = \Delta U$

$$-Q_{out} = -W_{in} + \Delta U \qquad (\text{since } \Delta U = m(u_2 - u_1))$$

$$Q_{out} = W_{in} - m(u_2 - u_1)$$

Step 3 of 5

From Saturated Refrigerant R-134a table

At $T_1 = 100°F$, $x_1 = 1$

$v_1 = v_g = 0.34045 \text{ ft}^3/\text{lbm}$

$u_1 = u_g = 107.45 \text{ Btu/lbm}$

At $T_2 = 100°F$, $x_2 = 0$

$v_2 = v_f = 0.01386 \text{ ft}^3/\text{lbm}$

$u_2 = u_f = 44.768 \text{ Btu/lbm}$

Step 4 of 5

From Saturated Refrigerant R-134a table

At $T_1 = 100°F$

Pressure $P_1 = P_2 = 138.93 \text{ psia}$

$h_{fg} = 71.08 \text{ Btu/lbm}$

$u_{fg} = 62.68 \text{ Btu/lbm}$

The work done during the process to

$$w_{out} = P_1(v_1 - v_2) \text{ psia.ft}^3/\text{lbm}$$

$$w_{out} = 138.93(0.01386 - 0.34045)\ \frac{1 \text{ Btu}}{5.41} \qquad (1 \text{ Btu} = 5.41 \text{ psia.ft}^3)$$

$$\boxed{w_{out} = -8.386 \text{ Btu/lbm}} \text{ (Negative since work done on the system)}$$

Step 5 of 5

Substituting into equation (1) gives

$$Q_{out} = W_{in} - m(u_2 - u_1)$$

$$q_{out} = w_{in} - (u_2 - u_1)$$

$$q_{out} = +8.386 - (44.768 - 107.45)$$

$$q_{out} = 8.386 + 62.68$$

$$\boxed{q_{out} = 71.068 \text{ Btu/lbm}}$$

Question: 5.043P

0.8-kg of saturated liquid R-134a with an initial temperature of −5°C is contained in a well-insulated, weighted piston-cylinder device. This device contains an electrical resistor, as shown in Fig.P5-43, to which 10 volts are applied causing a current of 2 amperes to flow through the resistor. Determine the time required for the refrigerant to be converted to a saturated vapor, and the final temperature.

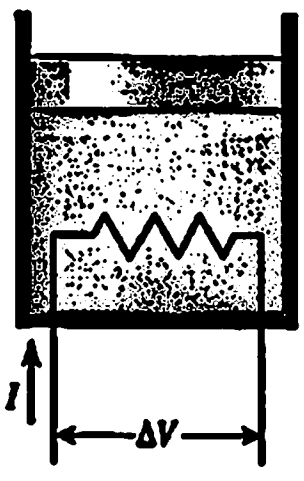


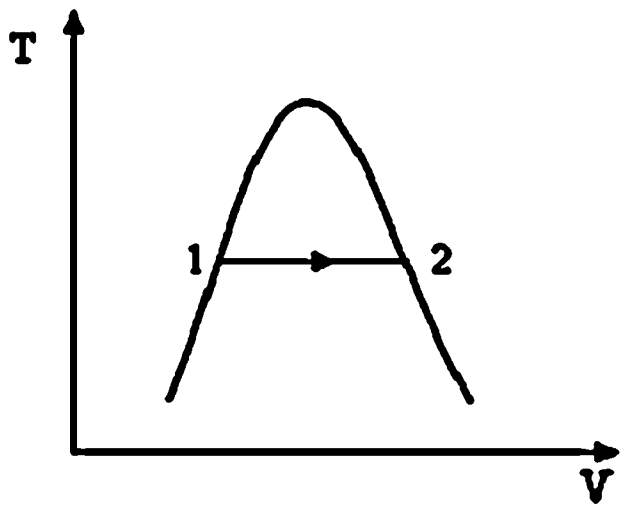
FIGURE P5–43

Step 1 of 6

Given:
Working fluid: Refrigerant-134a
Mass $m = 0.8\text{kg}$
Initially temperature $T = -5^{\circ}\text{C}$
Supply voltage $V = 10\text{V}$
Supply current $I = 2\text{amp}$

Step 2 of 6

T-V Diagram:



Step 3 of 6

From the first law of thermodynamics

$$W_{e,in} - W_{b,out} = \Delta U$$
$$= m(u_2 - u_1)$$
$$W_{e,in} = W_{b,out} + m(u_2 - u_1)$$
$$W_{e,in} = H_2 - H_1 = m(h_2 - h_1)$$
$$= mh_{fg}$$
$$W_{e,in}\Delta t = mh_{fg}$$

Step 4 of 6

Since $\Delta U + W_b = \Delta H$
During a constant pressure quasi-equilibrium process
The electrical power is
$$W_{e,in} = VI$$
$$= 10 \times 2$$
$$= 20\text{W}$$

Step 5 of 6

The enthalpy of vaporization of $R-134a$
From the saturated refrigerant-134a
At $T = -5^{\circ}\text{C}$

$$h_{fg} = 203.07 + \frac{(-5-(-6))(201.60-203.07)}{(-4-(-6))}$$

$$h_{fg} = 202.335\frac{\text{kJ}}{\text{kg}}$$

Step 6 of 6

Substituting values in energy balance equation
$$W_{e,in}\Delta t = mh_{fg}$$
$$(0.020)\Delta t = (0.8)(202.335)$$
$$\Rightarrow \Delta t = 8093.4\text{s}$$
$$= 2.25\text{hour}$$
$$\boxed{\Delta t = 2.25\text{hour}}$$
The temperature remains constant during this phase change process, so we have
$$T_2 = T_1 = -5^{\circ}\text{C}$$
$$\boxed{T_2 = T_1 = -5^{\circ}\text{C}}$$

Question: 5.044P

Two tanks (Tank A and Tank B) are separated by a partition. Initially Tank A contains 2-kg steam at 1 MPa and 300°C while Tank B contains 3-kg saturated liquid–vapor mixture at 150°C with a vapor mass fraction of 50 percent. The partition is now removed and the two sides are allowed to mix until the mechanical and thermal equilibrium are established. If the pressure at the final state is 300 kPa, determine (*a*) the temperature and quality of the steam (if mixture) at the final state and (*b*) the amount of heat lost from the tanks.

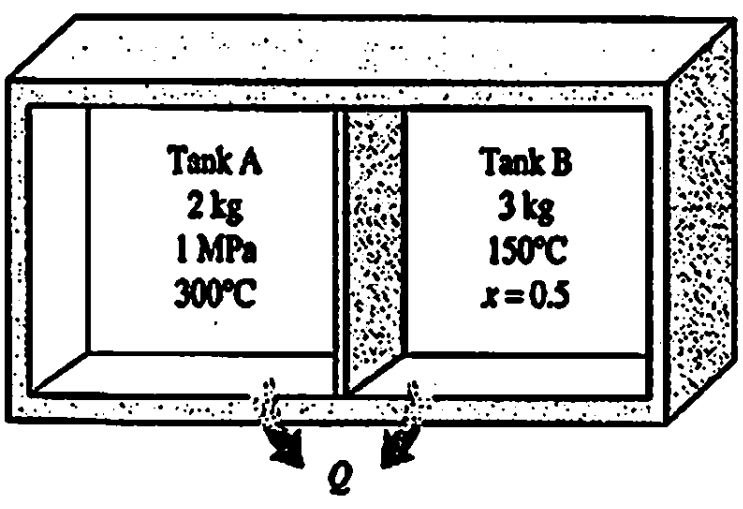


FIGURE P5–44

Step 1 of 8

Given

Mass of steam in tank A $m_A = 2\text{kg}$

Pressure in tank A $P_{1A} = 1\text{MPa}$

Temperature in tank A $T_{1A} = 300°\text{C}$

Mass of saturated liquid vapor mixture in tank B $m_B = 3\text{kg}$

Temperature of saturated liquid vapor mixture in tank B at $T_{1B} = 150°\text{C}$

Dryness fraction in tank B $x = 0.5$

Final Pressure $P_2 = 300\,\text{kPa}$

Step 2 of 8

At $P_{1A} = 1000\ \text{kPa}$ and $T_{1A} = 300°\text{C}$ from superheated steam tables

$u_{1A} = 2793.7\ \text{kJ/kg}$

$v_{1A} = 0.25799\ \text{m}^3/\text{kg}$

Step 3 of 8

At $T_B = 150°\text{C}$ from saturated water tables

$u_f = 631.66\ \text{kJ/kg}$

$u_{fg} = 1927.4\ \text{kJ/kg}$

$v_f = 0.001091\ \text{m}^3/\text{kg}$

$v_g = 0.39248\ \text{m}^3/\text{kg}$

Therefore specific volume of saturated liquid vapor mixture in tank B is

$$v_{1B} = v_f + x v_{fg}$$
$$= 0.001091 + 0.5(0.001091 - 0.39248)$$
$$v_{1B} = 0.19679\ \text{m}^3/\text{kg}$$

Internal energy of saturated liquid vapor mixture in tank B is

$$u_{1B} = u_f + x u_{fg}$$
$$= 631.66 + 0.5(1927.4)$$
$$u_{1B} = 1595.36\ \text{kJ/kg}$$

Step 4 of 8

The total volume of the system

$$V_2 = m_{1A} v_{1A} + m_{1B} v_{1B}$$
$$V = 2 \times 0.25799 + 3 \times 0.19679$$
$$V = 1.10635\ \text{m}^3$$

The total mass of the system

$$m_2 = m_A + m_B$$
$$= 2 + 3$$
$$m_2 = 5\ \text{kg}$$

Step 5 of 8

The final specific volume of the system

$$v_2 = \frac{V_2}{m_2}$$
$$= \frac{1.10635}{5}$$
$$v_2 = 0.22127\ \text{m}^3/\text{kg}$$

Step 6 of 8

(a)

At $P_2 = 300\ \text{kPa}$ from saturated water tables

$T_{sat} = 133.52^0\text{C}$

$v_f = 0.001073\ \text{m}^3/\text{kg}$

$v_{fg} = 0.60474\ \text{m}^3/\text{kg}$

$$v_2 = v_f + x_2 v_{fg}$$
$$0.22127 = 0.001073 + x_2 \times 0.60474$$
$$x_2 = 0.3641$$

Therefore quality at state 2 is

$$\boxed{x_2 = 0.3641}$$

Since the state is in mixture of liquid vapor region the final temperature be saturation temperature at $P_2 = 300\ \text{kPa}$

$$T_2 = T_{sat} = 133.52^0\text{C}$$

$$\boxed{T_2 = 133.52^0\text{C}}$$

Step 7 of 8

Internal energy at state2

$$u_2 = u_{f@300kPa} + x_2 u_{fg@300kPa}$$
$$= 561.11 + 0.3641 \times (1982.1)$$
$$u_2 = 1282.79\ \text{kJ/kg}$$

Step 8 of 8

(b) Heat loss from the tanks

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work energy transfer from the system is zero $W = 0$.

$$-Q_{out} = \Delta U$$
$$-Q_{out} = \Delta U_A + \Delta U_B$$
$$-Q_{out} = m_A(u_2 - u_{1A}) + m_B(u_2 - u_{1B})$$
$$-Q_{out} = 2(1282.79 - 2793.7) + 3(1282.79 - 1595.36)$$
$$= -3021.82 + (-937.71)$$
$$-Q_{out} = -3959.53\ \text{kJ}$$

Therefore heat loss from the tank is

$$\boxed{Q_{out} = 3959.53\ \text{kJ}}$$

Question: 5.045P

In the relation $\Delta u = mc_v \Delta T$, what is the correct unit of c_v :kJ/kg·°C or kJ/kg·K?

Step 1 of 1

Here $\Delta u = mc_{v,avg}\Delta T$

We can use kJ/kg.K or kJ/kg. °C, since the difference in temperature will be same for both the cases.

Question: 5.046P

Is the relation $\Delta u = mc_{v,avg} \Delta T$ restricted to constant-volume processes only, or can it be used for any kind of process of an ideal gas?

Step 1 of 1

The relation $\Delta u = mc_{v,avg}\Delta T$ can be used for any kind of process of an ideal gas.

Reason:

It shows that the internal energy of an ideal gas is also a function of temperature only.

So the process can be used for any kind of process of an ideal gas.

Question: 5.047P

Is the relation $\Delta h = mcp,\text{avg}\,\Delta T$ restricted to constant-pressure processes only, or can it be used for any kind of process of an ideal gas?

Step 1 of 1

The relation $\Delta h = mc_{p,avg}\Delta T$ follows that the enthalpy of an ideal gas is also a function of temperature only. So the process can be used for any kind of process of an ideal gas.

Question: 5.048P

Is the energy required to heat air from 295 to 305 K the same as the energy required to heat it from 345 to 355 K? Assume the pressure remains constant in both cases.

Step 1 of 1

We know that $Q = mc_p\Delta T$

Specific gas constant c_p remains constant as pressure remains constant then the heat transfer will be same since dT is same.

If we consider the change c_p with temperature then the heat required will be higher for 345K to 355K.

Question: 5.049P

A fixed mass of an ideal gas is heated from 50 to 80°C at a constant pressure of (*a*) 1 atm and (*b*) 3 atm. For which case do you think the energy required will be greater? Why?

Step 1 of 1

We know that $Q = mc_p\Delta T$

For both the cases it will be same since c_p is function of temperature only and c_p of ideal gas does not vary with pressure.

Question: 5.050P

A fixed mass of an ideal gas is heated from 50 to 80°C at a constant volume of (*a*) 1 m3 and (*b*) 3 m3. For which case do you think the energy required will be greater? Why?

Step 1 of 1

We know that $Q = mc_v\Delta T$

For both the cases it will be same since c_v is function of temperature only and for ideal gas it doesn't vary with volume.

Question: 5.051P

When undergoing a given temperature change, which of the two gases—air or oxygen—experiences the largest change in its (*a*) enthalpy, *h* and (*b*) internal energy, *u* ?

Step 1 of 1

We know $\Delta u = mc_{v,avg}\Delta T$

$\Delta h = mc_{p,avg}\Delta T$

Both undergoing the same temperature change, we also know that

$c_p = kc_v$ Where k = specific heat ratio

Since value of k will be more for air than oxygen

The air experiences more change for enthalpy h and internal energy u.

Question: 5.052P

Show that for an ideal gas $\bar{c}_p = \bar{c}_v + R_u$.

Step 1 of 1

A special relationship between c_p and c_v for ideal gases can be obtained by differentiating the relation

$$h = u + RT$$
$$dh = du + RdT$$

We know that $dh = c_p dT$, $du = c_v dT$ on substituting in the above equation

$$c_p dT = c_v dT + RdT$$

Dividing with dT on both sides

$$c_p - c_v = R$$

Multiplying both sides by molar mass M

$$\boxed{\bar{c}_p - \bar{c}_v = R_u}$$

Question: 5.053P

What is the change in the enthalpy, in kJ/kg, of oxygen as its temperature changes from 150 to 200°C? Is there any difference if the temperature change were from 0 to 50°C? Does the pressure at the beginning and end of this process have any effect on the enthalpy change?

Step 1 of 3

Given:

Initially temperature $T_1 = 150\,^{\circ}C$

Final temperature $T_2 = 200\,^{\circ}C$

Step 2 of 3

Enthalpy equation is

$$\Delta H = mC_p\Delta t$$

From the molar mass constants of various gases, we have

For oxygen:

$$C_{P\text{oxygen}} = 0.918\frac{kJ}{kg\cdot K}$$

$$\Delta t_1 = (200-150)$$

$$= 50^{\circ}C$$

$$\Delta H = 0.918\times 50$$

$$= 45.9\frac{kJ}{kg}$$

Step 3 of 3

No there is no change in enthalpy when temperature changes from 0 to 50°C because C_p remains same. Also,

$$\Delta t_2 = 50^{\circ}C = \Delta t_1$$

No, pressure at the beginning and end of this process does not have any effect on the enthalpy change because C_p of an ideal gas does not vary with pressure.

Question: 5.054P

Air is compressed from 20 psia and 70°F to 150 psia in a compressor. The compressor is operated such that the air temperature remains constant. Calculate the change in the specific volume of air as it passes through this compressor.

Step 1 of 3

Find the initial specific volume by using ideal gas equation of state.

$$P_1 v_1 = RT_1$$

$$v_1 = \frac{RT_1}{P_1}$$

Here, P_1 is initial pressure, T_1 is initial temperature, and R is gas constant.

Substitute $20\,\text{psia}$ for P_1, $(70+460)\,\text{R}$ for T_1, and $0.3704\,\frac{\text{psia.ft}^3}{\text{lbm.R}}$ for R.

$$v_1 = \frac{0.3704\times(70+460)}{20}$$

$$= 9.8156\,\text{ft}^3/\text{lbm}$$

Step 2 of 3

Similarly, find the specific volume at the compressor exit.

$$v_2 = \frac{RT_2}{P_2}$$

Here, P_2 is exit pressure and T_2 is exit temperature.

Substitute $150\,\text{psia}$ for P_1, $(70+460)\,\text{R}$ for T_1, and $0.3704\,\frac{\text{psia.ft}^3}{\text{lbm.R}}$ for R.

$$v_2 = \frac{0.3704\times(70+460)}{150}$$

$$= 1.3087\,\text{ft}^3/\text{lbm}$$

Step 3 of 3

Find the change in specific volume by using the following equation:

$$\Delta v = v_1 - v_2$$

Substitute $1.3087\,\text{ft}^3/\text{lbm}$ for v_2 and $9.8156\,\text{ft}^3/\text{lbm}$ for v_1

$$\Delta v = 9.8156 - 1.3087$$

$$= 8.5069\,\text{ft}^3/\text{lbm}$$

Therefore, the change in specific volume is $\boxed{8.5069\,\text{ft}^3/\text{lbm}}$

Question: 5.055P

The temperature of 2 kg of neon is increased from 20 to 180°C. Calculate the change in the total internal energy of the neon, in kJ. Would the internal energy change be any different if the neon were replaced with argon?

Step 1 of 2

Obtain the specific heat of neon at constant volume from the table A-2, "Ideal-gas specific heats of various common gases" in the textbook.

$$c_v = 0.6179 \text{ kJ/kg} \cdot \text{K}$$

Calculate the change in internal energy of neon as follows:

$$\Delta U = mc_v(T_2 - T_1)$$

Here, m is the mass of the neon, T_1 is the initial temperature, and T_2 is the final temperature.

Substitute 2 kg for m, $0.6179 \text{ kJ/kg} \cdot \text{K}$ for c_v, $20°\text{C}$ for T_1, and $180°\text{C}$ for T_2.

$$\begin{aligned}\Delta U &= 2 \times 0.6179 \times (180 - 20) \\ &= 197.728 \text{ kJ}\end{aligned}$$

Therefore, the change in internal energy for neon is $\boxed{197.728 \text{ kJ}}$.

Step 2 of 2

Obtain the specific heat of argon at constant volume from the table A-2, "Ideal-gas specific heats of various common gases" in the textbook.

$$c_v = 0.0756 \text{ kJ/kg} \cdot \text{K}$$

Calculate the change in internal energy of argon as follows:

$$\Delta U = mc_v(T_2 - T_1)$$

Substitute 2 kg for m, $0.6179 \text{ kJ/kg} \cdot \text{K}$ for c_v, $20°\text{C}$ for T_1, and $180°\text{C}$ for T_2.

$$\begin{aligned}\Delta U &= 2 \times 0.0756 \times (180 - 20) \\ &= 24.192 \text{ kJ}\end{aligned}$$

Therefore, the change in internal energy for argon is $\boxed{24.192 \text{ kJ}}$.

Question: 5.056P

Calculate the change in the enthalpy of argon, in kJ/kg, when it is cooled from 100 to 25°C. If neon had undergone this same change of temperature, would its enthalpy change have been any different?

Step 1 of 4

Given:
Working fluid: argon
Initial temperature $T_1 = 100\,^{\circ}\mathrm{C}$
Final temperature $T_2 = 25\,^{\circ}\mathrm{C}$

Step 2 of 4

From the molar mass and specific heat of various gases
For argon

$$C_{P,Ar} = 0.5203 \frac{\mathrm{kJ}}{\mathrm{kg \cdot K}}$$

For neon

$$C_{P,Ne} = 1.0299 \frac{\mathrm{kJ}}{\mathrm{kg \cdot K}}$$

Step 3 of 4

At specified conditions, Neon and Argon behave like ideal gases.
The enthalpy change in argon is

$$\begin{aligned} \Delta h_{argon} &= C_P \Delta T \\ &= (0.5203) \times (100 - 25) \\ &= 39.0225 \frac{\mathrm{kJ}}{\mathrm{kg}} \end{aligned}$$

Step 4 of 4

The enthalpy change in Neon is

$$\begin{aligned} \Delta h_{neon} &= C_{P,Ne} \times \Delta T \\ &= (1.0299) \times (100 - 25) \\ &= 77.243 \frac{\mathrm{kJ}}{\mathrm{kg}} \end{aligned}$$

So if Ne had undergone this change of temperature, enthalpy change will be different.

Question: 5.057P

Determine the enthalpy change Δh of oxygen, in Btu/lbm, as it is heated from 800 to 1500 R, using (*a*) the empirical specific heat equation as a function of temperature (Table A-2E *c*), (*b*) the *cp* value at the average temperature (Table A - 2E *b*) , and (*c*) the *cp* value at room temperature (Table A-2E *a*).

Step 1 of 1

2760-4-62P

SA: 2981
SR: 3246

(a) The empirical specific heat equation (Table A-2Ec)

$$c_p(T) = a + bT + cT^2 + dT^3$$

Where a = 6.085, b = 0.2017×10^{-2}, c = -0.05275×10^{-5}, and d = 0.05372×10^{-9}.
The enthalpy of an ideal gas is expressed as

$$\Delta\bar{h} = \int_1^2 c_p(T)\,dT \text{ Btu/lbm}$$

$$\Delta\bar{h} = \int_1^2 (a + bT + cT^2 + dT^3)\,dT \text{ Btu/lbm}$$

$$\Delta\bar{h} = a(T_2 - T_1) + 1/2b\left(T^2{}_2 - T_1^2\right) + 1/3c\left(T^3{}_2 - T_1^3\right) + 1/4d\left(T^4{}_2 - T_1^4\right) \text{ kJ/kg}$$

$$\Delta\bar{h} = 6.085(1500-800) + 1/2\left(0.2017\times10^{-2}\right)\left(1500^2 - 800^2\right) + 1/3\left(-0.05275\times10^{-5}\right)\left(1500^3 - 800^3\right)$$

$$+1/4\left(0.05372\times10^{-9}\right)\left(1500^4 - 800^4\right) \text{ Btu/lbmol.R}$$

$$\Delta\bar{h} = 5442 \text{ Btu/lbmol.R}$$

From tables of Molar mass, gas constant, and critical-point properties.
Molecular weight of oxygen $M_{o_2} = 31.999$lbm/lbmol

Specific enthalpy $\Delta h = \dfrac{\Delta\bar{h}}{M_{o_2}} = \dfrac{5442}{31.999} = 170.1$Btu/lbm

$\boxed{\Delta h = 170.1 \text{Btu/lbm}}$

(b)
The constant *cp* value from Table of ideal gas specific heats of various common gases at the average temperature of 1150 R

$$c_p = 0.255 \text{ Btu/lbm.R}$$

The enthalpy of an ideal gas is expressed as

$$\Delta h = c_p \Delta T \text{ Btu/lbm}$$

$$\Delta h = 0.255(1500-800) \text{ Btu/lbm}$$

$\boxed{\Delta h = 178.5 \text{ Btu/lbm}}$

(c) The constant c_p value from Table of ideal gas specific heats of various common gases at room temperature

$$c_p = 0.219 \text{ Btu/lbm.R}$$

The enthalpy of an ideal gas is expressed as

$$\Delta h = c_p \Delta T \text{ Btu/lbm}$$

$$\Delta h = 0.219(1500-800) \text{ Btu/lbm}$$

$\boxed{\Delta h = 153.3 \text{ Btu/lbm}}$

Question: 5.058P

Determine the internal energy change Δu of hydrogen, in kJ/kg, as it is heated from 200 to 800 K, using (*a*) the empirical specific heat equation as a function of temperature (Table A-2*c*), (*b*) the *c* v value at the average temperature (Table A-2 *b*) , and (*c*) the *c* v value at room temperature (Table A-2*a*).

Step 1 of 3

The internal energy change of hydrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

(a)

Using the empirical relation for $\overline{c}_p(T)$

From ideal gas specific heat of various gases

As a function of temperature

For hydrogen

$a = 29.11$

$b = -0.1916\times10^{-2}$

$c = 0.4003\times10^{-5}$

$d = -0.8704\times10^{-9}$

Relating it to $\overline{c}_v(T)$

$$\overline{c}_v(T) = \overline{c}_p - R_u$$

$$= (a - R_u) + bT + cT^2 + dT^3$$

Then,

$$\Delta\overline{u} = \int_1^2 \overline{c}_v(T)\,dT$$

$$= \int_1^2 \left[(a - R_u) + bT + cT^2 + dT^3\right] dT$$

$$= (a - R_u)(T_2 - T_1) + \frac{1}{2}b\left(T_2^2 + T_1^2\right) + \frac{1}{3}c\left(T_2^3 - T_1^3\right) + \frac{1}{4}d\left(T_2^4 - T_1^4\right)$$

$$= (29.11 - 8.314)(800 - 200) - \frac{1}{2}(0.1961\times10^{-2})(800^2 - 200^2)$$

$$+ \frac{1}{3}(0.4003\times10^{-5})(800^3 - 200^2) - \frac{1}{4}(0.8704\times10^{-9})(800^4 - 200^4)$$

$$= 12{,}487\,\frac{\text{kJ}}{\text{kmol}}$$

Step 2 of 3

Calculate the internal energy as follows:

$$\Delta u = \frac{\Delta\overline{u}}{M}$$

$$= \frac{12{,}487}{2.016}$$

$$= 6194\,\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{\Delta u = 6194\,\frac{\text{kJ}}{\text{kg}}}$$

(b)

Using a constant c_p value

Average temperature

$$T_{avg} = \frac{200 + 800}{2}$$

$$= 500\,\text{K}$$

From ideal gas specific heat of various gases

At various temperature table

At the average temperature of 500 K

$$c_{v,avg} = C_{v@500k}$$

$$= 10.389\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$\Delta u = c_{v,avg}\left(T_2 - T_1\right)$$

$$= (10.389)(800 - 200)$$

$$= 6233\,\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{\Delta u = 6233\,\frac{\text{kJ}}{\text{kg}}}$$

Step 3 of 3

(c)

Using a constant c_p value at room temperature

Take room temperature

$$T_{room} = 27^\circ\text{C}$$

$$= 27 + 273$$

$$= 300\text{K}$$

$$c_{v,avg} = C_{v@300k}$$

$$= 10.183\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Calculate the internal energy as follows:

$$\Delta u = c_{v,avg}\left(T_2 - T_1\right)$$

$$= (10.183)(800 - 200)$$

$$= 6110\,\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{\Delta u = 6110\,\frac{\text{kJ}}{\text{kg}}}$$

Question: 5.059P

Is it possible to compress an ideal gas isothermally in an adiabatic piston-cylinder device? Explain.

Step 1 of 1

No,

Energy balance for any system undergoing any kind of process was expressed as

$$E_{in} - E_{out} = \Delta E_{system} \quad \text{(kJ)} \text{ ----------(1)}$$

In the process $\Delta PE = \Delta KE = 0$

In a isothermal process $\Delta u = 0$, for adiabatic process $Q = 0$

From the above equation

$$Q - W = \Delta U$$

$$W = 0$$

So it is not possible to compress air at constant temperature in an adiabatic piston cylinder process.

Question: 5.060P

1-kg of oxygen is heated from 20 to 120°C. Determine the amount of heat transfer required when this is done during a (*a*) constant-volume process and (*b*) isobaric process.

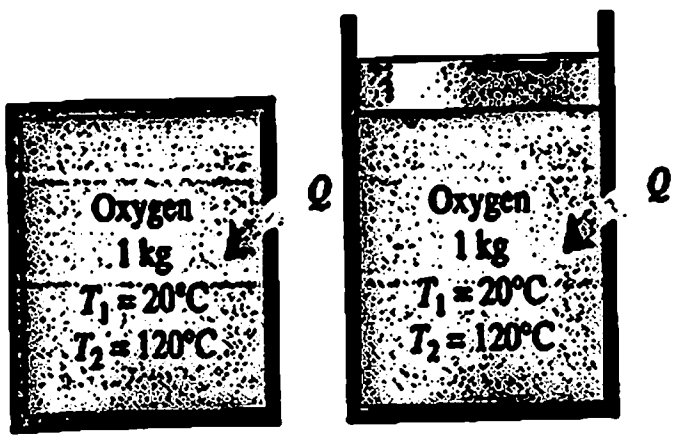


FIGURE P5–60

Step 1 of 6

Given:

Mass of the oxygen $m = 1\text{kg}$

Initially temperature $T_1 = 20°C$

Final temperature $T_2 = 120°C$

Step 2 of 6

The average temperature of oxygen is

$$T_{avg} = \frac{(20+120)}{2}$$
$$= 70°C$$
$$= 70+273$$
$$= 343\text{K}$$

Step 3 of 6

The specific heats of oxygen

From the ideal gases and specific heat of various gases

For oxygen

$$C_p = 0.926\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$C_v = 0.664\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 4 of 6

The energy balance for a constant – volume process is

$$Q = \Delta u + W$$

$$W = 0$$

Since constant volume work done is zero

$$Q_{in} = \Delta U$$
$$= mC_v(T_2 - T_1)$$

The energy balance during a constant pressure process can be expressed as

$$Q_{in} = \Delta H$$
$$= mC_p(T_2 - T_1)$$

Step 5 of 6

(a) Heat transfer in constant volume process is

$$Q_{in,\,v-const} = mC_v(T_2 - T_1)$$
$$= (1)(0.664)(120-20)$$
$$= 66.4\text{kJ}$$

$$\boxed{Q_{in,\,v-const} = 66.4\text{kJ}}$$

Step 6 of 6

(b) Heat transfer in constant pressure process is

$$Q_{in,\,p-const} = mC_p(T_2 - T_1)$$
$$= (1)(0.926)(120-20)$$
$$= 92.6\text{kJ}$$

$$\boxed{Q_{in,\,p-const} = 92.6\text{kJ}}$$

Question: 5.061P

A rigid tank contains 20 lbm of air at 50 psia and 80°E The air is now heated until its pressure doubles. Determine (*a*) the volume of the tank and (*b*) the amount of heat transfer.

Step 1 of 4

Given

Mass of air in the rigid tank $m = 20$ lbm

Initial pressure $P_1 = 50$ psia

Initial temperature $T_1 = 80°F = 80+460 = 540R$

Final pressure $P_2 = 100$ psia

Step 2 of 4

(a)

The volume of the tank

The gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R (From table Molar mass, gas constant, and critical-point properties).

From ideal relation

$$PV = mRT$$

$$V = \frac{mRT_1}{P_1} \text{ ft}^3$$

$$V = \frac{20(0.3704)(80+460)}{50} \text{ ft}^3$$

$$\boxed{V = 80 \text{ ft}^3}$$

Step 3 of 4

(b) The amount of heat transfer

Energy balance for any system undergoing any kind of process was expressed as

$$E_{in} - E_{out} = \Delta E_{system} \quad \text{(Btu)} \quad \text{----------(1)}$$

In the process $\Delta PE = \Delta KE = W = 0$

Therefore equation (1) $Q_{in} + W_{in} = \Delta U$

$$Q_{in} = \Delta U \qquad \left(\text{since } \Delta U = m(u_2 - u_1)\right)$$

$$Q_{in} = m(u_2 - u_1)$$

Step 4 of 4

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

$$T_2 = \frac{T_1}{P_1} \times P_2$$

$$T_2 = \frac{540}{50} \times 100$$

$$T_2 = 1080\text{R}$$

From ideal-gas properties of air

At $T_1 = 540$R

$u_1 = 92.04$ Btu/lbm

At $T_2 = 1080$ R

$u_2 = 186.93$ Btu/lbm

Substituting in above relation

$$Q_{in} = 20(186.93 - 92.04)$$

$$\boxed{Q_{in} = 1898 \text{ Btu}}$$

Question: 5.062P

Determine the internal energy change of air in a rigid container as its temperature is increased by transferring 50 Btu/lbm of heat.

Step 1 of 2

We know the first law of thermodynamics as:

$$Q = \Delta U + W$$

$$W = 0$$

$$Q = \Delta U$$

Since rigid container volume remains constant so work done is zero.

Step 2 of 2

We take the air as the system. This is a closed system since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$Q = \Delta U$$

$$= mC_v\left(T_2 - T_1\right)$$

Since temperature is increasing heat is coming in

$$\Delta U = q_{in}$$

$$= 50\frac{\text{Btu}}{\text{lbm}}$$

$$\boxed{\Delta U = 50\frac{\text{Btu}}{\text{lbm}}}$$

Question: 5.063P

Nitrogen gas to 20 psia and 100°F initially occupies a volume of 1 ft3 in a rigid container equipped with a stirring paddle wheel. After 5000 lbf·ft of paddle wheel work is done on nitrogen, what is its final temperature?

Step 1 of 3

From the table A-1E for nitrogen gas, the gas constant value is,

$$R = 0.3830 \text{ psia.ft}^3/\text{lbm.R}$$

The temperature conversion of °F to R scale ,

$$100°\text{F} = 560\text{R}$$

Step 2 of 3

From the Ideal gas equation

$$P_1V_1 = mRT_1$$

$$m = \frac{P_1V_1}{RT_1}$$

Here P_1 is the initial pressure of the gas, V_1 is the volume of the gas, R is the gas constant, T_1 is the initial temperature of the gas, m is the mass of the gas.

Substitute 20 psia for P_1, 1 ft^3 for V_1, 0.3830 psia.ft^3/lbm.R for R and 560R for T_1.

$$m = \frac{P_1V_1}{RT_1}$$

$$= \frac{20 \times 1}{(0.3830 \times 560)}$$

$$= 0.0932 \text{ lbm}$$

Step 3 of 3

The paddle work done over the gas is,

$$W = mc_v(T_2 - T_1)$$

Consider the work done to be 5000lbf·ft,

But,

$$1\text{lbf} \cdot \text{ft} = 1.28507 \times 10^{-3}\text{Btu}$$

So,

$$5000\text{lbf} \cdot \text{ft} = 6.43\text{Btu}$$

Substitute 6.43Btu for W, 0.177 Btu/lbm.R for c_v, and 560 for T_1.

$$W = mc_v(T_2 - T_1)$$

$$6.43 = 0.0932 \times 0.177 \times (T_2 - 560)$$

$$T_2 = 949.78\text{R}$$

$$= 489°\text{F}$$

Therefore the final temperature after the paddling of the nitrogen gas is, $\boxed{489°\text{F}}$.

Question: 5.064P

Consider a piston-cylinder device that contains nitrogen gas as the system. Initially, the system is at 1 MPa and 427°C. It now undergoes an isobaric process until its temperature is 27°C. Determine the final pressure and the heat transfer, in kJ/kg, associated with this process.

Step 1 of 5

Obtain the properties from the ideal gas specific heats of various common gases table.
For nitrogen:

Gas constant, $R = 0.2968 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$

Specific heat at constant pressure, $C_p = 1.039 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$

Specific heat at constant volume, $C_v = 0.743 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$

Step 2 of 5

In an isobaric process, the pressure remains constant.
Thus, the final pressure will be the same as the initial pressure.

$$P_2 = P_1$$
$$= 1\text{MPa}$$

Therefore, the final pressure is $\boxed{1\text{MPa}}$.

Step 3 of 5

Consider the energy balance equation for this system.

$$\Delta Q = \Delta U + W_{b,out}$$
$$Q_{in} - Q_{out} = mC_v(T_2 - T_1) + W_{b,out}$$
$$0 - Q_{out} = mC_v(T_2 - T_1) + W_{b,out}$$
$$Q_{out} = -mC_v(T_2 - T_1) - W_{b,out} \quad \text{..... (1)}$$

Step 4 of 5

Calculate the boundary work during the isobaric process by using the equation as shown below.

$$W_{b,out} = P(v_2 - v_1)$$
$$= R(T_2 - T_1)$$

Here, the gas constant is R, the initial temperature is T_1 and the final temperature is T_2.

Substitute $0.2968 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$ for R, 27°C for T_2 and 427°C for T_1.

$$W_{b,out} = 0.2968(27 - 427)$$
$$= -118.72 \frac{\text{kJ}}{\text{kg}} \quad \text{..... (2)}$$

Step 5 of 5

Calculate the heat transfer in the process.

Substitute equation (2) in equation (1).

$$Q_{out} = -[0.743(27 - 427)] - (-118.72)$$
$$= 297.2 + 118.72$$
$$= 415.921 \frac{\text{kJ}}{\text{kg}}$$

Therefore, the heat transferred is $\boxed{416 \frac{\text{kJ}}{\text{kg}}}$.

Question: 5.065P

A 4-m × 5-m × 6-m room is to be heated by a baseboard resistance heater. It is desired that the resistance heater be able to raise the air temperature in the room from 5 to 25°C within 11 min. Assuming no heat losses from the room and an atmospheric pressure of 100 kPa, determine the required power of the resistance heater. Assume constant specific heats at room temperature.

Step 1 of 5

Given:

Dimension of the room $= 4\,\text{m} \times 5\,\text{m} \times 6\,\text{m}$

Initially temperature $T_1 = 5^\circ\text{C}$

Final temperature $T_2 = 25^\circ\text{C}$

Time $\Delta t = 11\,\text{min}$

Atmospheric pressure $P_{atm} = 100\,\text{kPa}$

Step 2 of 5

From the ideal gas properties of the various gases

For air

The gas constant of air is

$$R = 0.287\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$C_v = 0.718\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 3 of 5

The energy balance for this stationary constant volume closed system can be expressed as

$E_{in} - E_{out} = \Delta E_{system}$

Since there is no heat losses, So

$W_{e,in} = \Delta U$

$= mC_v(T_2 - T_1)$

$W_{e,in} \times \Delta t = mC_v(T_2 - T_1) \ldots\ldots(1)$

Step 4 of 5

The mass of the air is

$V = 4 \times 5 \times 6$

$= 120\,\text{m}^3$

$$m = \frac{P_1 V}{R T_1}$$

$$= \frac{100 \times 120}{(0.287 \times 278)}$$

$= 150.4\,\text{kg}$

Step 5 of 5

Substituting values in equation 1 we get the power rating of the heater becomes

$$W_{e,in} = \frac{(150.4)(0.718)(25-5)}{(11 \times 60)}$$

$= 3.272\,\text{kW}$

$\boxed{W_{e,in} = 3.272\,\text{kW}}$

Question: 5.066P

A student living in a 4-m × 6-m × 6-m dormitory room turns on her 150-W fan before she leaves the room on a summer day, hoping that the room will be cooler when she comes back in the evening. Assuming all the doors and windows are tightly closed and disregarding any heat transfer through the walls and the windows, determine the temperature in the room when she comes back 10 h later. Use specific heat values at room temperature, and assume the room to be at 100 kPa and 15°C in the morning when she leaves.

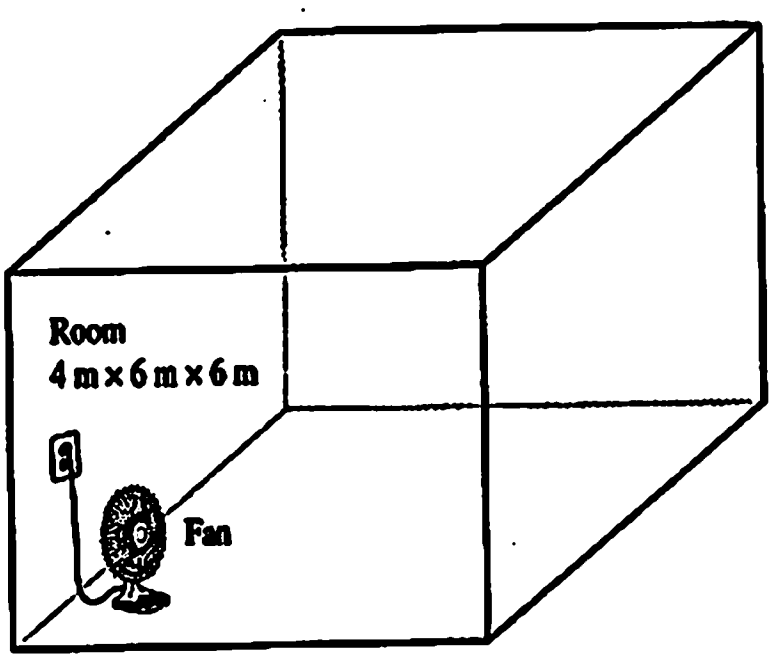


FIGURE P5–66

Step 1 of 4

Given

Volume of the room $V = 4 \times 6 \times 6 = 144\ \text{m}^3$

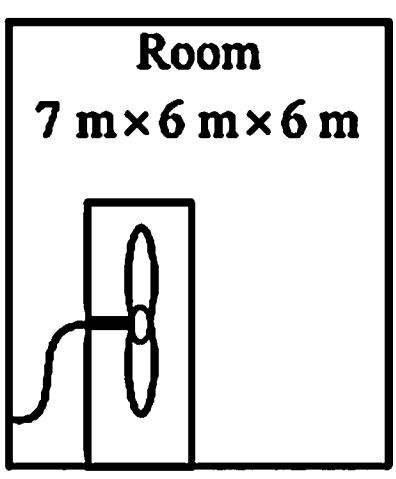
Initial temperature $T_1 = 15°\text{C}$

Atmospheric pressure $P_1 = 100\ \text{kPa}$

Time = 10 hr

Power of the fan P = 150 W

Step 2 of 4



Step 3 of 4

From ideal gas specific heats of various common gases

The gas constant of air R = 0.287 kJ/kg.k, $c_v = 0.718$ kJ/kg.K

The room is a closed system and applying energy balance for a closed system

$$E_{in} - E_{out} = \Delta E_{system}$$

Here $\Delta PE = \Delta KE = Q = 0$

$$W_{in} = \Delta U \qquad \left(\text{Since } \Delta U = m(u_2 - u_1)\right)$$

$$W_{in} = mc_v(T_2 - T_1)$$

The mass of air is expressed $m = \dfrac{P_1 V}{R T_1}$ kg

$$m = \frac{100(144)}{(0.287)(15+273)} = 174.2\ \text{kg}$$

Step 4 of 4

The work supplied to the heater is

$$W_{in} = (174.2)(0.718)(T_2 - 15)$$

The power supplied is

$$P = \frac{W_{in}}{t}$$

$$150 = \frac{174.2(0.718 \times 1000)(T_2 - 15)}{10 \times 60 \times 60}$$

The final temperature of the room

$$\boxed{T_2 = 58.2°\text{C}}$$

10×60×60

The final temperature of the room

$$\boxed{T_2 = 58.2°\text{C}}$$

Question: 5.067P

A 3-ft3 adiabatic rigid container is divided into two equal volumes by a thin membrane, as shown in Fig. P5-67E. Initially, one of these chambers is filled with air at 100 psia and 100°F while the other chamber is evacuated. Determine the internal energy change of the air when the membrane is ruptured. Also determine the final air pressure in the container.

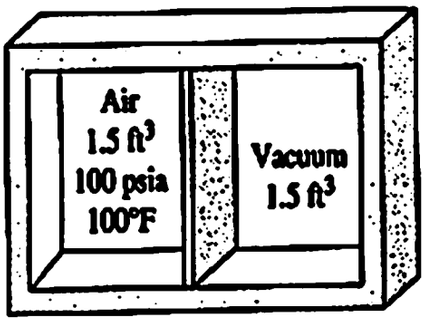


FIGURE P5–67E

Step 1 of 3

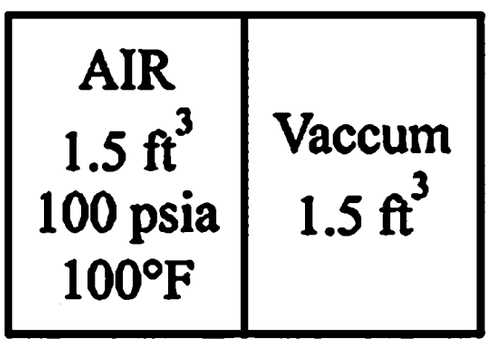
Given

Volume of the room $V = 3\ \text{ft}^3$

Initial pressure $P_1 = 100$ psia

Initial Temperature $T_1 = 100°\text{F}$

Step 2 of 3



Step 3 of 3

The container is a closed system and applying energy balance for a closed system

$$E_{in} - E_{out} = \Delta E_{system}$$

Here $\Delta PE = \Delta KE = W = Q = 0$

$$0 = \Delta U$$

So the internal energy of the rigid body does not change

Applying ideal gas equation

$$P_1V_1 = P_2V_2 \qquad (T_1 = T_2)$$

$$100(1.5) = P_2(3)$$

Final pressure $\boxed{P_2 = 50 \text{ psia}}$

Question:

5.068P

A closed system containing 2-kg of air undergoes an isothermal process from 600 kPa and 200°C to 80 kPa. Determine the initial volume of this system, the work done, and the heat transfer during this process.

Step 1 of 4

Given

Mass of air $m = 2$ kg

Initial pressure $P_1 = 600$ kPa

Final pressure $P_2 = 80$ kPa

Isothermal process $T_1 = T_2 = 200°C$

Step 2 of 4

Initial volume of the system

We know that $PV = mRT$

From molar mass, gas constant, and critical point properties table

Question:

A closed system containing 2-kg of air undergoes an isothermal process from 600 kPa and 200°C to 80 kPa. Determine the initial volume of this system, the work done, and the heat transfer during this process.

Step 1 of 4

Given

Mass of air $m = 2$ kg

Initial pressure $P_1 = 600$ kPa

Final pressure $P_2 = 80$ kPa

Isothermal process $T_1 = T_2 = 200°C$

Step 2 of 4

Initial volume of the system

We know that $PV = mRT$

From molar mass, gas constant, and critical point properties table

$$Q_{in} - W_{out} = \Delta U = mc_v(T_2 - T_1)$$

$$Q_{in} - W_{out} = 0 \qquad \text{Since } T_1 = T_2$$

$$Q_{in} = W_{out}$$

$$\boxed{Q_{in} = 547 \text{ kJ}}$$

$$Q_{in} - W_{out} = \Delta U = mc_v(T_2 - T_1)$$

$$Q_{in} - W_{out} = 0 \qquad \text{Since } T_1 = T_2$$

$$Q_{in} = W_{out}$$

$$\boxed{Q_{in} = 547 \text{ kJ}}$$

Question: 5.069P

A piston-cylinder device containing argon gas as the system undergoes an isothermal process from 200 kPa and 100°C to 50 kPa. During the process, 1500 kJ of heat is transferred to the system. Determine the mass of this system and the amount of work produced.

Step 1 of 3

Given

Initial pressure $P_1 = 200$ kPa

Final pressure $P_2 = 50$ kPa

Isothermal process $T_1 = T_2 = 100°C$

Heat supplied $Q_{in} = 1500$ kJ

Step 2 of 3

The work done for an isothermal process is expressed as

$$W_{out} = m\int_1^2 pdV = mRT\ln\frac{P_1}{P_2}\ \text{kJ}$$

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} - W_{out} = \Delta U = mc_v(T_2 - T_1)$$

$$Q_{in} - W_{out} = 0 \qquad \text{Since } T_1 = T_2$$

$$Q_{in} = W_{out}$$

The amount of work produced is

$$\boxed{W_{out} = 1500\ \text{kJ}}$$

Step 3 of 3

The mass of the system

$$m = \frac{W_{in}}{RT\ln\frac{P_1}{P_2}}$$

$$m = \frac{1500}{(0.2081\times 373)\ln\frac{200}{50}}$$

$$\boxed{m = 13.94\ \text{kg}}$$

Question: 5.070P

Argon is compressed in a polytropic process with n = 1.2 from 120 kPa and 10°C to 800 kPa in a piston-cylinder device. Determine the work produced and heat transferred during this compression process, in kJ/kg.

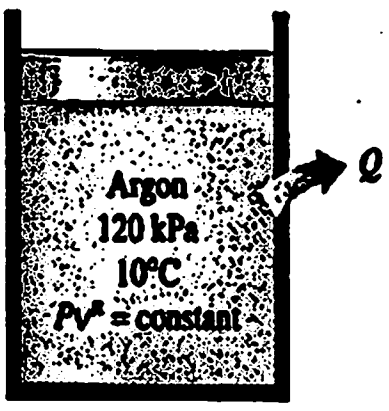


FIGURE P5–70

Step 1 of 5

From properties of ideal gases, the gas constant value of Argon is,

$$R = 0.2081 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$$

Specific heat value at constant volume is,

$$c_v = 0.3122 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$$

Step 2 of 5

Consider a closed system of Argon, since no mass crosses the boundaries of the system the energy balance for the system is,

$$Q - W = \Delta U$$

Here Q is the amount of the heat interaction of the system, W is the work interaction of the system, and ΔU is the change in internal energy.

The internal energy for a closed system per unit mass is,

$$\Delta U = c_v (T_2 - T_1)$$

Step 3 of 5

The temperature at the final state is,

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

Here, T_1 is the initial temperature of the system, T_2 is the final temperature of the system, P_1 is the initial pressure condition, P_2 is the final pressure condition and n is the polytropic index.

Substitute 120 kPa for P_1, 800 kPa for P_2, 1.2 for n and 10°C = 283K for T_1.

$$\begin{aligned} T_2 &= T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \\ &= 283 \left(\frac{800}{120} \right)^{\frac{1.2-1}{1.2}} \\ &= 388.49 \text{K} \end{aligned}$$

Step 4 of 5

Find the work done in a polytropic process by using the expression,

$$\begin{aligned} W &= \frac{R(T_2 - T_1)}{1-n} \\ &= \frac{0.2081 \times (388.5 - 283)}{1 - 1.2} \\ &= -109.77 \text{kJ/kg} \end{aligned}$$

Step 5 of 5

Find the heat transfer by using energy balance equation,

$$\begin{aligned} Q - W &= \Delta U \\ Q &= W + \Delta U \\ &= W + c_v (T_2 - T_1) \\ &= -109.76 + (0.3122)(388.49 - 283) \\ &= -76.83 \text{kJ/kg} \end{aligned}$$

The negative sign indicates that the heat is dissipated from the system of Argon.

Therefore the work done on the system of Argon is, $\boxed{109.77 \text{kJ/kg}}$ and heat dissipated is, $\boxed{76.83 \text{kJ/kg}}$.

Question: 5.071P

A spring-loaded piston-cylinder device contains 1 kg of carbon dioxide. This system is heated from 100 kPa and 25°C to 1000 kPa and 300°C. Determine the total heat transfer to and work produced by this system.

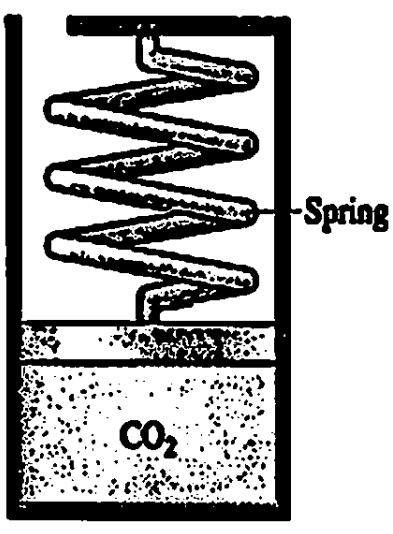


FIGURE P5–71

Step 1 of 5

Obtain the properties of carbon dioxide from Table A-2, "Ideal-gas specific heats of various common gases," from the text book.

Gas constant for carbon dioxide, $R = 0.1889\ \text{kJ/kg·K}$

Specific heat value at constant volume, $c_v = 0.657\ \text{kJ/kg·K}$

Write the ideal gas equation at initial state.

$$P_1V_1 = mRT_1$$

$$V_1 = \frac{mRT_1}{P_1}$$

Here, P_1 is the initial pressure of the system, V_1 is the initial volume of the system, m is the mass of the system, and T_1 is the initial temperature of the system.

Substitute $100\ \text{kPa}$ for P_1, $25°\text{C}$ for T_1, $1\ \text{kg}$ for m, and $0.1889\ \text{kJ/kg·K}$ for R.

$$V_1 = \frac{(1)(0.1889)(25°\text{C}+273)\ \text{K}}{100}$$
$$= 0.562922\ \text{m}^3$$

Step 2 of 5

Write the ideal gas equation at final state.

$$P_2V_2 = mRT_2$$

$$V_2 = \frac{mRT_2}{P_2}$$

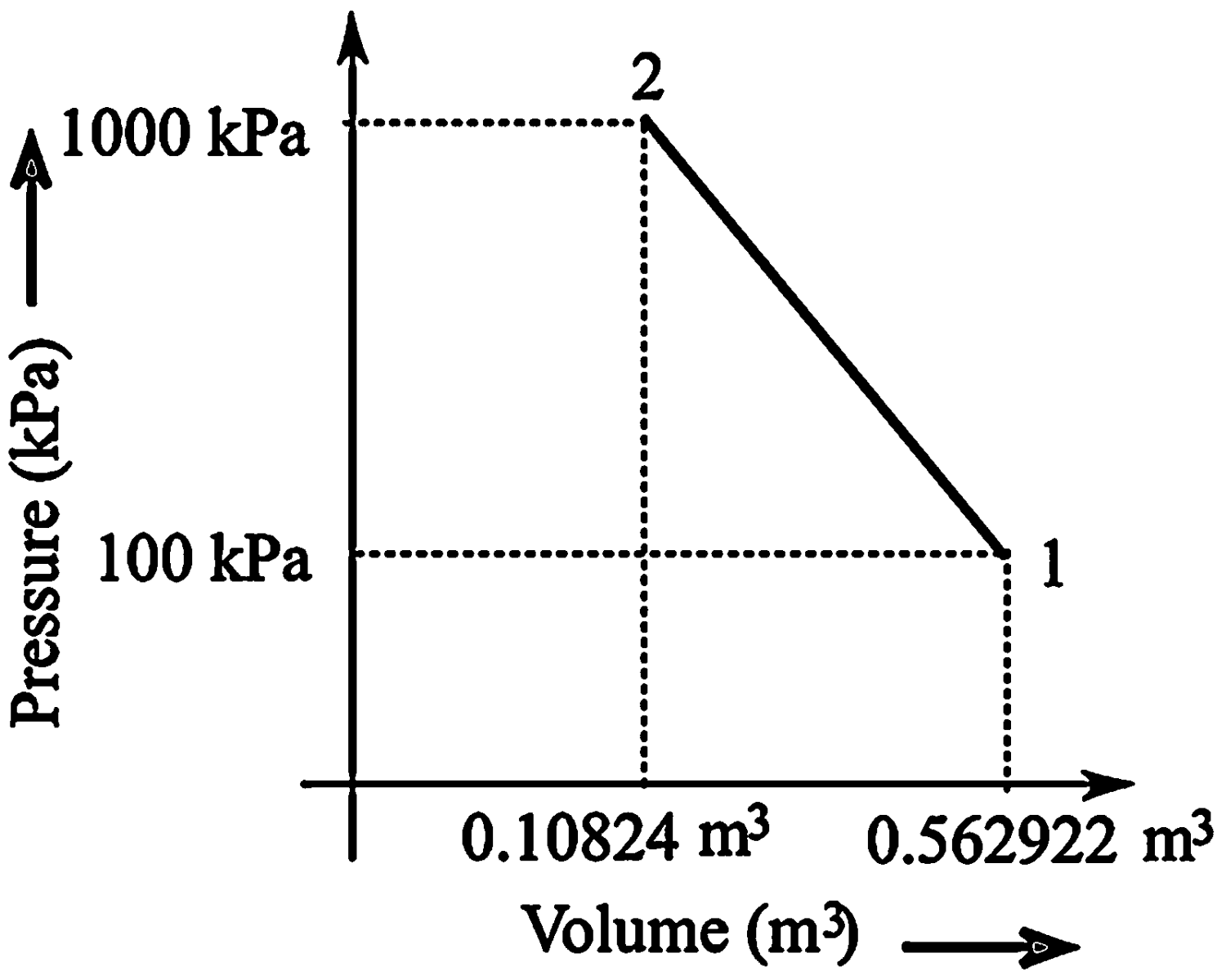
Here, P_2 is the final pressure of the system, V_2 is the final volume of the system, and T_2 is the final temperature of the system.

Substitute $1000\ \text{kPa}$ for P_2, $300°\text{C}$ for T_2, $1\ \text{kg}$ for m, and $0.1889\ \text{kJ/kg·K}$ for R.

$$V_2 = \frac{(1)(0.1889)(300°\text{C}+273)\ \text{K}}{1000}$$
$$= 0.10824\ \text{m}^3$$

Step 3 of 5

Draw the $P-V$ diagram for the system of gas.



Step 4 of 5

The pressure changes linearly with volume, and the work done is equal to the area under the process line 1-2.

Write the relation for the work done during the process by using the following relation:

$$W = \left(\frac{P_1+P_2}{2}\right)(V_2-V_1)$$

Substitute $100\ \text{kPa}$ for P_1, $1000\ \text{kPa}$ for P_2, $0.10824\ \text{m}^3$ for V_2, and $0.562922\ \text{m}^3$ for V_1.

$$W = \left(\frac{100+1000}{2}\right)(0.10824-0.562922)$$
$$= -250.08\ \text{kJ}$$

Here, negative sign indicates that the work is produced by the system.

Therefore, the work produced by the system is $\boxed{250.08\ \text{kJ}}$.

Step 5 of 5

Calculate the heat transfer by using the following relation:

$$Q = W + \Delta U$$

Here, ΔU is the change in internal energy.

Substitute $mc_v(T_2-T_1)$ for ΔU.

$$Q = W + mc_v(T_2-T_1)$$

Substitute $-250.08\ \text{kJ}$ for W, $1\ \text{kg}$ for m, $0.657\ \text{kJ/kg·K}$ for c_v, $300°\text{C}$ for T_2, and $25°\text{C}$ for T_1.

$$Q = -250.08+(1)(0.657)(300-25)$$
$$= -250.08+180.675$$
$$= -69.405\ \text{kJ}$$

Therefore, the heat dissipated from the system is $\boxed{69.405\ \text{kJ}}$.

Question: 5.072P

Air is contained in a variable-load piston-cylinder device equipped with a paddle wheel. Initially, air is at 500 kPa and 27°C. The paddle wheel is now turned by an external electric motor until 50 kJ/kg of work has been transferred to air. During this process, heat is transferred to maintain a constant air temperature while allowing the gas volume to triple. Calculate the required amount of heat transfer, in kJ/kg.

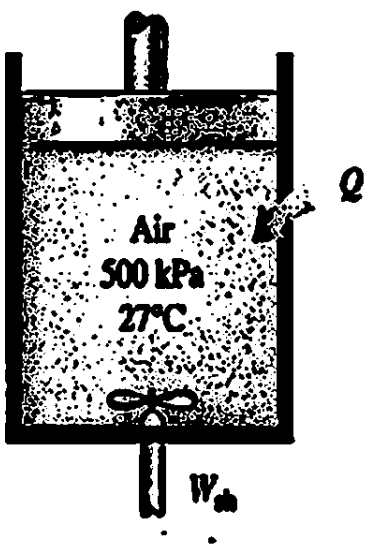
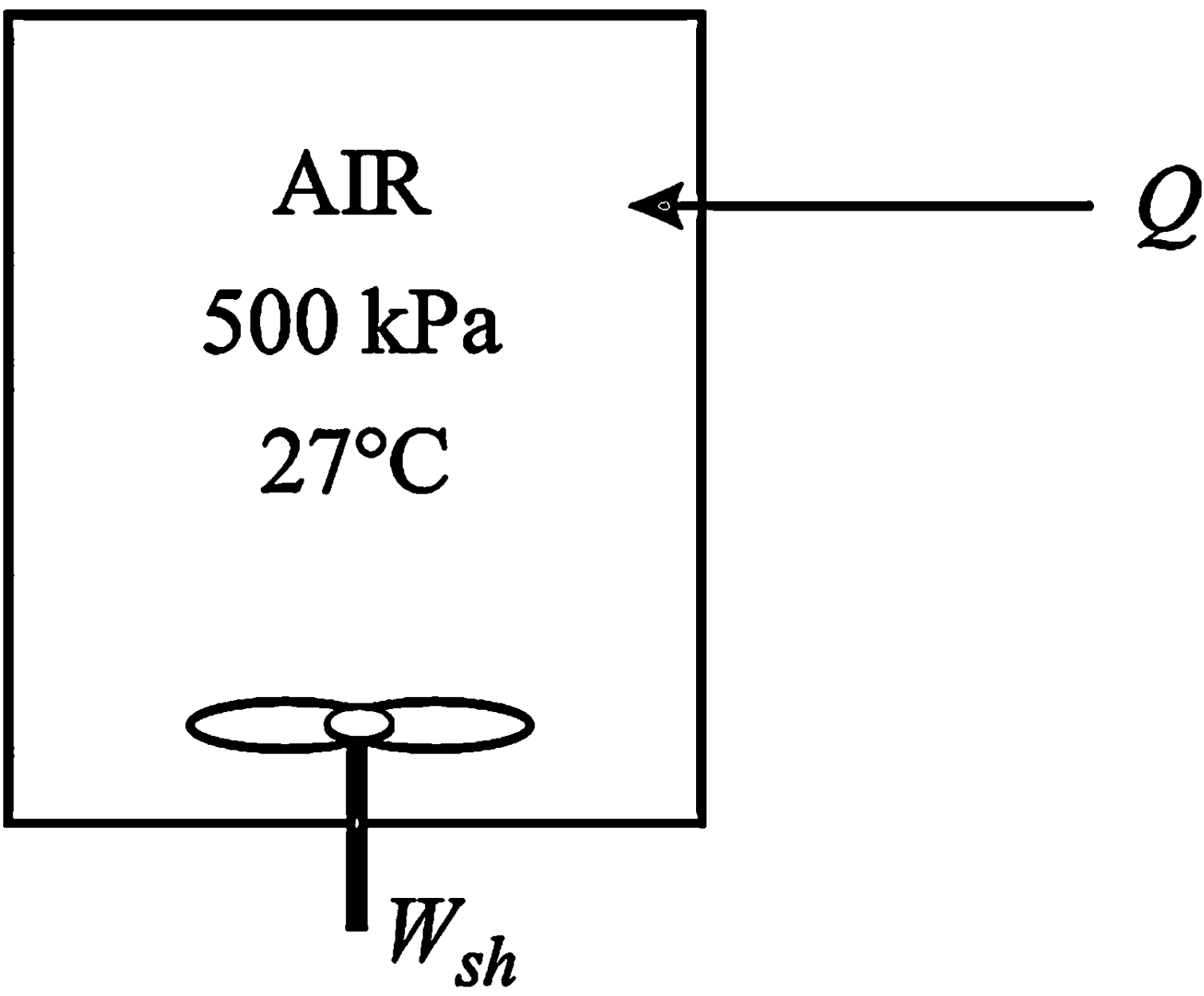


FIGURE P5–72

Step 1 of 4

Schematic diagram of the variable-load piston-cylinder device is shown below.



Step 2 of 4

Calculate the initial specific volume of the system using the ideal gas equation

$$P_1 v_1 = RT_1$$
$$v_1 = \frac{RT_1}{P_1}$$

Here, R is the gas constant and P_1 is the initial pressure.

Substitute $287\ \text{J/kg.K}$ for R $(27+273)\text{K}$ for T_1 and $500\times10^3\ \text{Pa}$ for P_1

$$v_1 = \frac{287(27+273)}{500\times10^3}$$
$$= 0.1722\ \text{m}^3/\text{kg}$$

Calculate the final specific volume of the system using the following relation

$$v_2 = 3v_1$$
$$= 3\times0.1722$$
$$= 0.5166\ \text{m}^3/\text{kg}$$

Step 3 of 4

Calculate the boundary work for isothermal process

$$w_{out} = P_1 v_1 \ln\left(\frac{v_2}{v_1}\right)$$
$$= 500\times10^3\times0.1722\times\ln\left(\frac{0.5166}{0.1722}\right)$$
$$= 94.59\ \text{kJ/kg}$$

Here, the kinetic and potential energy changes are negligible.

$$\Delta PE = \Delta KE = 0$$

Step 4 of 4

Apply the energy balance equation for the system

$$E_{in} - E_{out} = \Delta E_{system}$$

$$\Delta U = W_{in} - Q_{out} - W_{out}$$
$$m(u_2 - u_1) = W_{in} - Q_{out} - W_{out}$$
$$mc_v(T_2 - T_1) = W_{in} - Q_{out} - W_{out}$$

Here, initial temperature T_1 is equal to final temperature T_2

$$W_{in} - Q_{out} - W_{out} = 0$$
$$Q_{out} = W_{in} - W_{out}$$
$$q_{out} = w_{sh,in} - w_{out}$$

Here, $w_{sh,in}$ is the work input and w_{out} is the work output

$$q_{out} = w_{sh,in} - w_{out}$$
$$= 50 - 94.59$$
$$= -44.59\ \text{kJ/kg}$$

Here, negative sign indicates heat is transferred from system to air.

Therefore, the required amount of heat transfer is $\boxed{-44.59\ \text{kJ/kg}}$.

Question: 5.073P

A mass of 15 kg of air in a piston-cylinder device is heated from 25 to 77°C by passing current through a resistance heater inside the cylinder. The pressure inside the cylinder is held constant at 300 kPa during the process, and a heat loss of 60 kJ occurs. Determine the electric energy supplied, in kWh.

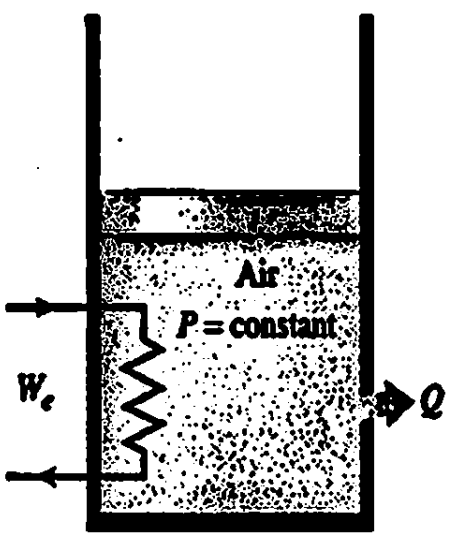
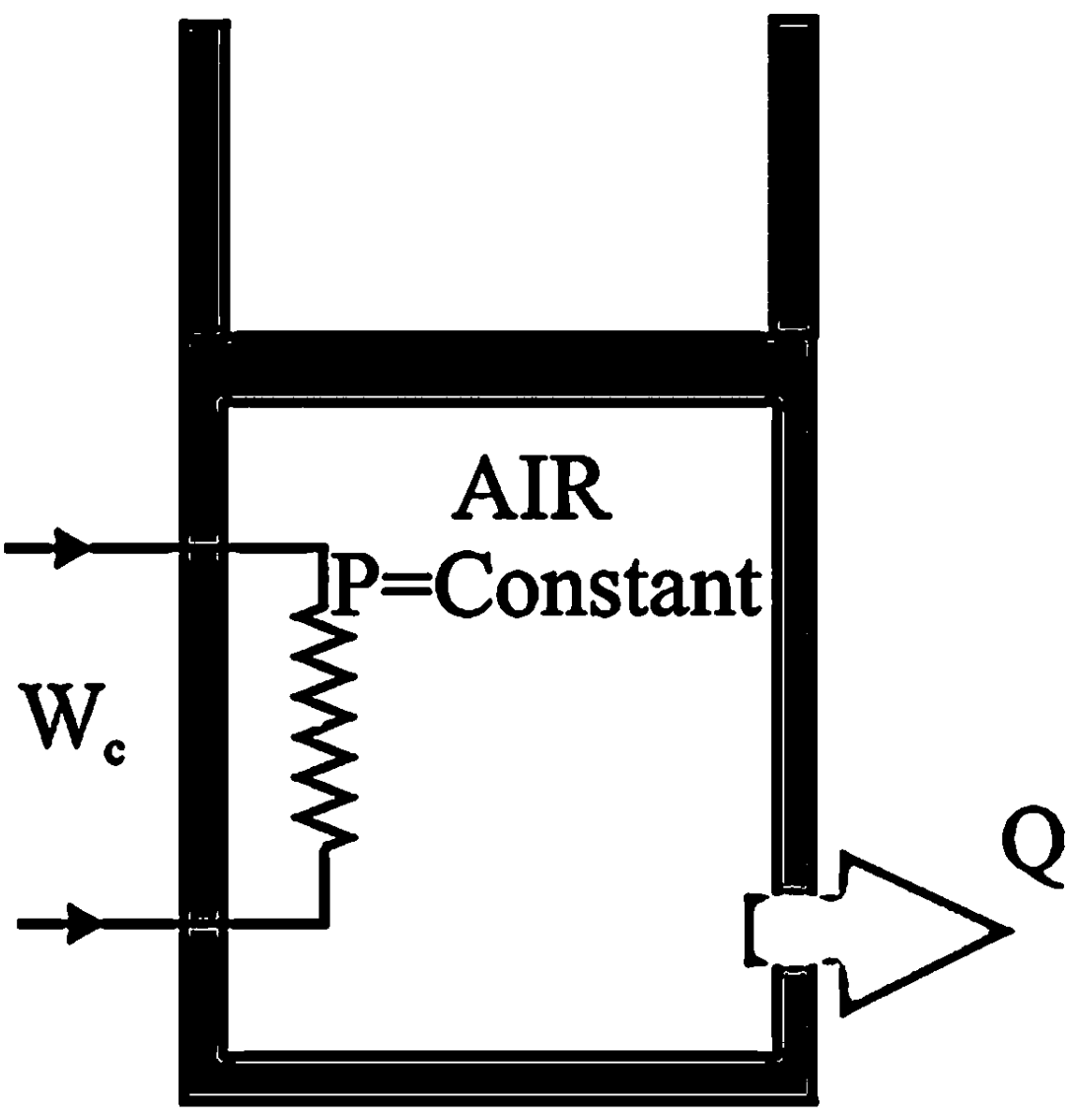


FIGURE P5–73

Step 1 of 4

Draw the schematic diagram.



Step 2 of 4

Obtain the properties of air from Table A1, "Molar mass, gas constant, and critical-point properties," from the text book.

Gas constant, $R = 0.287\ \text{kJ/kg.K}$

Write the energy balance of the system.

$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

Since the system is stationary, the change in potential and kinetic energies are equal to zero.

Rewrite the energy balance equation.

$$W_{c,in} - Q_{out} - W_{b,out} = \Delta U$$
$$W_{c,in} - Q_{out} = \Delta U + W_{b,out}$$

Here, $W_{c,in}$ is the work done on the system, $W_{b,out}$ is the work done by the system, Q_{out} is the heat flowing out of the system, and ΔU is the change in internal energy of the system.

Substitute ΔH for $\Delta U + W_{b,out}$.

$$W_{c,in} - Q_{out} = \Delta H$$
$$W_{c,in} = \Delta H + Q_{out}$$

Here, ΔH is the change in enthalpy.

Substitute $mc_p(T_2 - T_1)$ for ΔH.

$$W_{c,in} = mc_p(T_2 - T_1) + Q_{out} \quad \text{..... (1)}$$

Here, m is the mass of the air, T_1, T_2 are the temperatures at the initial and the final temperatures.

Step 3 of 4

Calculate the mean temperature.

$$T_{mean} = \frac{T_1 + T_2}{2}$$

Substitute $77°C$ for T_1 and $25°C$ for T_2.

$$T_{mean} = \frac{77 + 25}{2}$$
$$= (51°C + 273)\ \text{K}$$
$$= 324\ \text{K}$$

Obtain the specific heat constants for air at $324\ \text{K}$ temperature from table A-2(b), "Ideal gas specific heats of various common gases," from the text book by interpolation.

Specific heat at constant pressure is as follows:

$$\frac{1.008 - c_p}{1.008 - 1.005} = \frac{350 - 324}{350 - 300}$$
$$c_p = 1.00644\ \text{kJ/kg·K}$$

Specific heat at constant volume is as follows:

$$\frac{0.721 - c_v}{0.721 - 0.718} = \frac{350 - 324}{350 - 300}$$
$$c_p = 1.00644\ \text{kJ/kg·K}$$
$$c_v = 0.71944\ \text{kJ/kg·K}$$

Step 4 of 4

Substitute $15\ \text{kg}$ for m, $1.00644\ \text{kJ/kg·K}$ for c_p, $77°C$ for T_2, $25°C$ for T_1, and $60\ \text{kJ}$ for Q in equation (1).

$$W_{c,in} = mc_p(T_2 - T_1) + Q_{out}$$
$$= 15 \times (1.00644) \times \left([(77°C) + 273]\ \text{K} - [(25°C) + 273]\ \text{K} \right) + 60$$
$$= 845.0232\ \text{kJ} \times \frac{1\ \text{kWh}}{3600}$$
$$= 0.235\ \text{kWh}$$

Therefore, the electrical energy supplied to the system is $\boxed{0.235\ \text{kWh}}$.

Question: 5.074P

A piston-cylinder device contains 1.5 kg of nitrogen initially at 100 kPa and 17°C. The nitrogen is now compressed slowly in a polytropic process during which $PV1.3$ = constant until the volume is reduced by one-half. Determine the work done and the heat transfer for this process.

Step 1 of 5

For a polytropic process,

$$P_2V_2^n = P_1V_1^n$$

Here, P_1 is Initial pressure, V_1 is the initial volume, P_2 is the final pressure and V_2 is the final volume and n is the polytropic index.

Substitute 100 kPa for P_1 and $V_2 = 0.5V_1$ as final volume is half of initial volume, 1.3 for n.

$$P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} \times P_1$$
$$= 2^{1.3} \times (100)$$
$$= 246.22 \text{ kPa}$$

Step 2 of 5

From ideal gas equation,

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$T_2 = \frac{P_2V_2 \times T_1}{P_1V_1}$$

Here, T_1 is the initial temperature, T_2 is the final temperature of the system.

$$T_2 = \frac{P_2V_2 \times T_1}{P_1V_1}$$
$$= \frac{246.22 \times 0.5 \times 290}{(100)}$$
$$= 357.031 \text{K}$$

Step 3 of 5

Find the properties of N_2 at the average temperature

$$T_{avg} = \left(\frac{357.034 + 290}{2}\right)$$
$$= 323.517 \text{K}$$

From properties of ideal gases for Nitrogen gas, the gas constant value is,

$$R = 0.2968 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$$

Specific heat value at constant volume is,

$$C_v = 0.7435 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}$$

Step 4 of 5

Determine the boundary work for this polytropic process by using the expression,

$$W = \frac{mR(T_2 - T_1)}{n-1}$$

Here m is the mass of the gas

Substitute 1.5 kg for m

$$W = \frac{mR(T_2 - T_1)}{n-1}$$
$$= \frac{(1.5 \times 0.2968)(357.031 - 290)}{1.3 - 1}$$
$$= 99.5 \text{kJ}$$

Step 5 of 5

Find the amount of heat transfer by using the energy balance:

$$Q = W + mc_v(T_2 - T_1)$$
$$= -99.5 + (1.5)(0.7435)(357.031 - 290)$$
$$= -24.7 \text{ kJ}$$

Therefore, the work done by the gas is, $\boxed{99.5\text{kJ}}$ and the amount of heat dissipated from the system is, $\boxed{24.7 \text{ kJ}}$.

Question: 5.075P



Reconsider Prob. 5–74. Using EES (or other) software, plot the process described in the problem on a *P*- V diagram, and investigate the effect of the polytropic exponent *n* on the boundary work and heat transfer. Let the polytropic exponent vary from 1.1 to 1.6. Plot the boundary work and the heat transfer versus the polytropic exponent, and discuss the results.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

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Question:

5.076P

A piston-cylinder device, with a set of stops on the top, initially contains 3 kg of air at 200 kPa and 27°C. Heat is now transferred to the air, and the piston rises until it hits the stops, at which point the volume is twice the initial volume. More heat is transferred until the pressure inside the cylinder also doubles. Determine the work done and the amount of heat transfer for this process. Also, show the process on a *P*- v diagram.

Step 1 of 7

Given

Mass of air $m = 3\,\text{kg}$

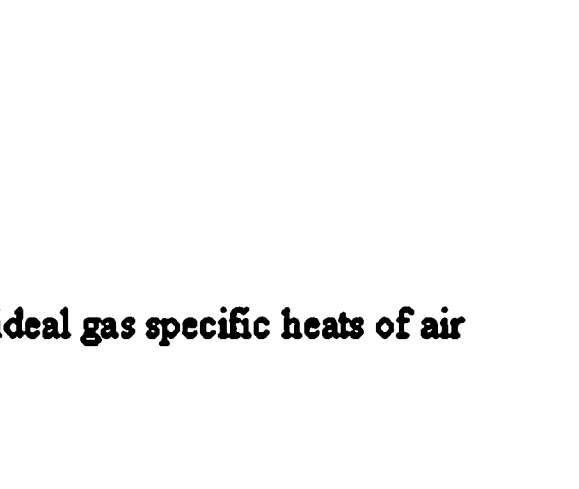
Initial pressure $P_1 = 200\ \text{kPa}$

Initial temperature $T_1 = 27°\text{C}$

Volume at state 2 is $V_2 = 2V_1$

Pressure at state 3 is $P_3 = 2P_2$

Step 2 of 7



Step 3 of 7

By using ideal gas equation temperature at state 3 is

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3}$$

$$T_3 = \frac{P_3V_3}{P_1V_1} \times T_1$$

Since $V_3 = V_2$ and $P_2 = P_1$

$$T_3 = \frac{P_3V_2}{P_2V_1} \times T_1$$

$$T_3 = \frac{2P_2 \times 2V_1}{P_2V_1} \times T_1$$

$$T_3 = 4 \times T_1$$

$$T_3 = 4 \times (27 + 273)$$

$$T_3 = 1200\text{K}$$

Step 4 of 7

At average temperature

$$T_{avg} = \frac{T_1 + T_3}{2}$$

$$T_{avg} = \frac{300 + 1200}{2}$$

$$T_{avg} = 750\text{K}$$

At $T_{avg} = 750\text{K}$ from table ideal gas specific heats of air

$c_v = 0.800\text{kJ/kg.K}$

$R = 0.287\text{kJ/kg.K}$

Step 5 of 7

Initial volume at state 1 is

$$P_1V_1 = mRT_1$$

$$V_1 = \frac{mRT_1}{P_1}$$

$$V_1 = \frac{3 \times (0.287) \times (27 + 273)}{200}$$

$$V_1 = 1.2915\ \text{m}^3$$

Volume at state 2 is

$$V_2 = 2V_1$$

$$V_2 = 2(1.2915)$$

$$V_2 = 2.583\ \text{m}^3$$

Step 6 of 7

The boundary work done is expressed as

$$W_{net} = \int_1^3 pdV$$

$$= W_{1-2} + W_{2-3}$$

$$W_{net} = P_1(V_2 - V_1) + 0$$

$$W_{net} = 200(2.583 - 1.2915)$$

$$\boxed{W_{net} = 258.3\ \text{kJ}}$$

Step 7 of 7

The amount of heat supplied is

$$Q = W_{net} + m(u_3 - u_1)$$

$$Q = W_{net} + mc_v(T_3 - T_1)$$

$$Q = 258.3 + 3(0.800)(1200 - 300)$$

$$\boxed{Q = 2418.3\ \text{kJ}}$$

Question: 5.077P

Air is contained in a cylinder device fitted with a piston-cylinder. The piston initially rests on a set of stops, and a pressure of 300 kPa is required to move the piston. Initially, the air is at 100 kPa and 27°C and occupies a volume of 0.4 m3. Determine the amount of heat transferred to the air, in kJ, while increasing the temperature to 1200 K. Assume air has constant specific heats evaluated at 300 K.

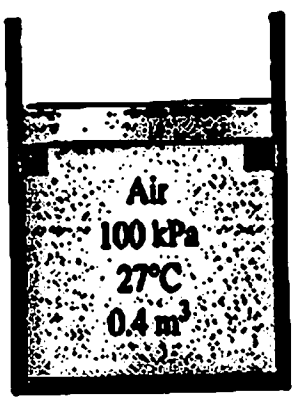


FIGURE P5–77

Step 1 of 6

Given:

Initially pressure $P_1 = 100\text{kPa}$

Initially temperature $T_1 = 27\,^{\circ}\text{C}$

Initially volume $V_1 = 0.4\,\text{m}^3$

Required pressure $P = 300\text{kPa}$

Final temperature $T_2 = 1200\,\text{K}$

Step 2 of 6

From the molar gas, constant and critical point properties

For air

$$R = 0.287\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Specific heat of air $C_v = 0.718\,\dfrac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Step 3 of 6

Mass of the air in the system is

We know that ideal gas equation

$$P_1V_1 = mRT_1$$

$$\Rightarrow m = \frac{P_1V_1}{RT_1}$$

$$= \frac{(0.4)(100)}{0.287\times 300}$$

$$= 0.464\text{kg}$$

Final volume

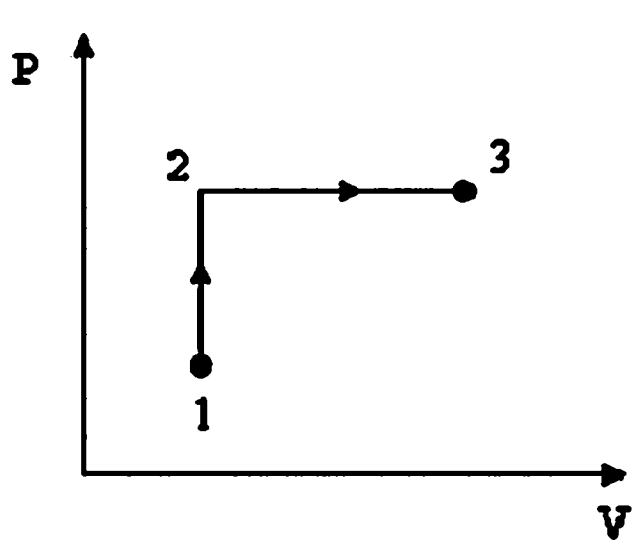
$$V_3 = \frac{mRT_3}{P_3}$$

$$= \frac{0.464\times 0.287\times 1200}{300}$$

$$= 0.533\text{m}^3$$

Step 4 of 6

P-v Diagram:



Step 5 of 6

No work is done during process 1-2 since $V_1 = V_2$.

The pressure remains constant during process 2-3 and

Work done during the process is

$$W_{b,out} = \int_1^2 P\,dV$$

$$= P_2(V_3 - V_2)$$

$$= 300\times(0.533-0.4)$$

$$= 39.6\text{kJ}$$

Step 6 of 6

Then from the energy balance

We take the air in the cylinder as the system.

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

$$Q_{in} = \Delta U + W_{b,out}$$

$$= mC_v(T_3 - T_1) + 40.14$$

$$= [0.464\times 0.718(1200-300)] + 39.9$$

$$= 339.73\text{kJ}$$

$$\boxed{Q_{in} = 339.73\text{kJ}}$$

Question: 5.078P

Air is contained in a piston-cylinder device at 600 kPa and 927°C, and occupies a volume of 0.8 m3. The air undergoes and isothermal (constant temperature) process until the pressure in reduced to 300 kPa. The piston is now fixed in place and not allowed to move while a heat transfer process takes place until the air reaches 27°C.

(*a*) Sketch the system showing the energies crossing the boundary and the *P*- V diagram for the combined processes.

(*b*) For the combined processes determine the net amount of heat transfer, in kJ, and its direction.

Assume air has constant specific heats evaluated at 300 K.

Step 1 of 7

Given:
Process: isothermal
Initially pressure $P_1 = 600\text{kPa}$
Initially temperature $T_1 = 927^\circ\text{C}$
Initially volume $V_1 = 0.8\text{m}^3$
Final pressure $P_2 = P_3 = 300\text{kPa}$
Final temperature $T_3 = 27^\circ\text{C}$

Step 2 of 7

From the molar gas, constant and critical point properties
For air

$$R = 0.287\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

We take the air as the system.
The energy balance equation can be expressed as

$$Q_{in} - W_{b,out} = \Delta U$$
$$= mC_v(T_2 - T_1)$$

Since $T_2 = T_1$

$$Q_{in} = W_{b,out}$$

Step 3 of 7

Mass of the air in a system

$$m = \frac{P_1V_1}{RT_1}$$
$$= \frac{600\times0.8}{0.287\times1200}$$
$$= 1.394\text{kg}$$

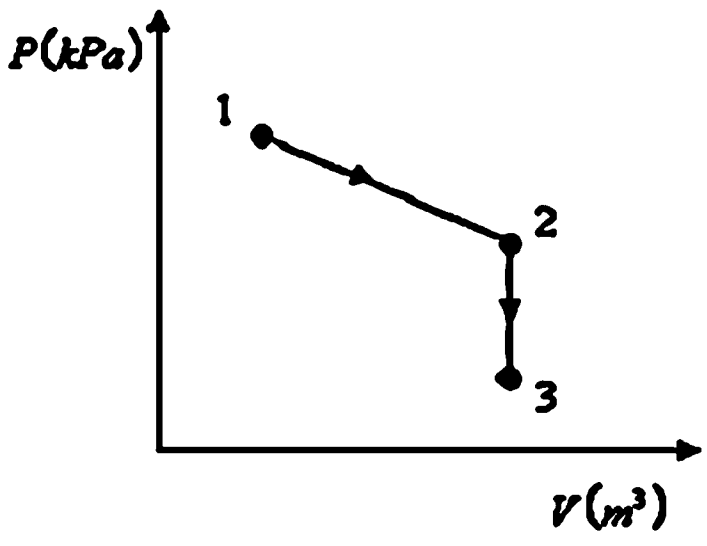
Step 4 of 7

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,out} = mRT\ln\left(\frac{P_1}{P_2}\right)$$
$$= 1.394\times0.287\times1200\ln\left(\frac{600}{300}\right)$$
$$= 332.78\text{kJ}$$

Step 5 of 7

A)



Step 6 of 7

(b) $(Q_{in})_{1-2} = W_{b,out}$
$= 332.78\text{kJ}$
$(Q_{in})_{2-3} = \Delta U$
$= mC_v(T_3 - T_2)$
$= 1.394\times0.718(27-927)$
$= -900.803\text{kJ}$

Step 7 of 7

Net amount of heat transfer
$Q_{net} = W_{in} + \Delta U$
$Q_{net} = 332.78 - 900.803$
$= -568.023\text{kJ}$

$\boxed{Q_{net} = -568.023\text{kJ}}$

Negative sign indicates that heat rejection take place from the system to surrounding.

Question: 5.079P

Argon is contained in a cylinder device fitted with a piston. Initially, the argon is at 100 kPa and 27°C and occupies a volume of 0.4 m3. The argon is first compressed while the temperature is held constant until the volume is 0.2 m3. Then the argon expands while the pressure is held constant until the volume is 0.6 m3.

(*a*) Sketch both processes on a single *P*- V diagram showing the direction of the processes and label the end states as 1, 2, and 3.

(*b*) Determine the total amount of net heat transferred to the argon, in kJ, during the combined processes.

Step 1 of 12

Given:

Working fluid: argon

Initially pressure $P_1 = 100\text{kPa}$

Initially temperature $T_1 = 27\,^\circ\text{C}$

Initially volume $V_1 = 0.4\text{m}^3$

Volume at state 2 $V_2 = 0.2\text{m}^3$

Final volume $V_3 = 0.6\text{m}^3$

Step 2 of 12

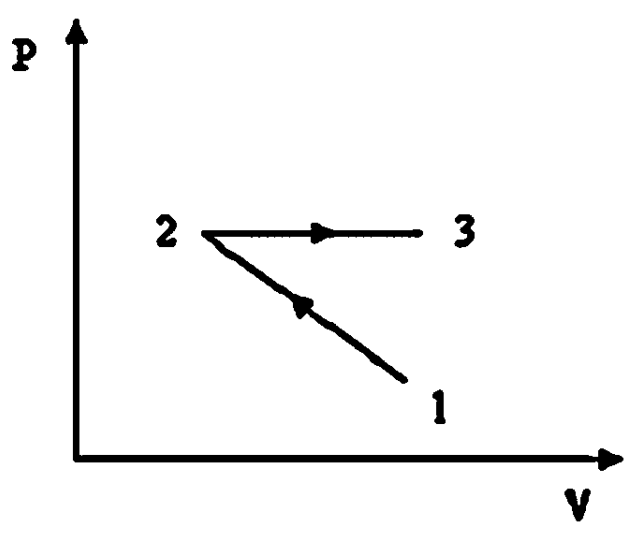
From the molar gas, constant and critical point properties

For argon

$$R = 0.2081\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 3 of 12

(a) P-v Diagram:



Step 4 of 12

(b) The energy balance equation is

$$Q_{in} - W_{b,out} = \Delta U$$
$$= mC_v(T_2 - T_1)$$

Since process 1 to 2 is isothermal $T_2 = T_1$

$$Q_{in} = W_{b,out}$$

$$W_{b,out} = mRT\ln\left(\frac{V_2}{V_1}\right)$$

Step 5 of 12

Mass of the argon in the system is

$$m = \frac{P_1V_1}{RT_1}$$
$$= \frac{100\times0.4}{0.2081\times300}$$
$$= 0.641\text{kg}$$

Step 6 of 12

Work done is

$$W_{b,out} = mRT\ln\left(\frac{V_2}{V_1}\right)$$

$$W_{b,out} = 0.641\times0.2081\times300\ln\left(\frac{0.2}{0.4}\right)$$
$$= -27.74\text{kJ}$$

Since process 1 to 2 is isothermal $T_2 = T_1$

$$Q_{in} = W_{b,out}$$

$$(Q_{in})_{1-2} = -27.74\text{kJ}$$

Step 7 of 12

Process 2 to 3 is isobaric

$$V_2T_3 = V_3T_2$$

$$\Rightarrow T_3 = \frac{V_3T_2}{V_2}$$
$$= \frac{0.6\times300}{0.2}$$
$$= 900\text{K}$$

Step 8 of 12

We know that

$$P_1V_1 = P_2V_2$$

$$P_2 = \frac{P_1V_1}{V_2}$$
$$= \frac{100\times0.4}{0.2}$$
$$= 200\text{kPa}$$

Step 9 of 12

$$W_{b,out} = P_2(V_3 - V_2)$$

$$W_{b,out} = 200(0.6-0.2)$$
$$= 80\text{kJ}$$

Step 10 of 12

Internal energy is

$$\Delta U = mC_v(T_3 - T_2)$$
$$= 0.641\times0.3122(900-300)$$
$$= 120.072\text{kJ}$$

Step 11 of 12

Heat transfer is

$$(Q_{in})_{2-3} = W_{b,out} + mC_v(T_3 - T_2)$$

$$(Q_{in})_{2-3} = 80 + 120.072$$
$$= 200.072\text{kJ}$$

Step 12 of 12

Total amount of heat transferred to Argon

$$Q_{net} = Q_{1-2} + Q_{2-3}$$

$$Q_{net} = 200.072 - 27.74$$
$$= 172.332\text{kJ}$$

$$\boxed{Q_{net} = 172.332\text{kJ}}$$

Question: 5.080P

An ideal gas contained in a piston-cylinder device undergoes an isothermal compression process which begins with an initial pressure and volume of 100 kPa and 0.6 m3, respectively. During the process there is a heat transfer of 60 kJ from the ideal gas to the surroundings. Determine the volume and pressure at the end of the process.

Step 1 of 3

Given:
Process: isothermal compression
Initially pressure $P_1 = 100\text{kPa}$
Initially volume $V_1 = 0.6\text{m}^3$
Heat transfer $Q = 60\text{kJ}$

Step 2 of 3

The energy balance equation is
$Q_{in} = W_{b,out}$
Since temperature remains constant so internal energy is zero.
Work done isothermal process is

$$-60 = P_1V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$\ln\left(\frac{V_2}{V_1}\right) = \frac{-60}{P_1V_1}$$

$$= \frac{-60}{(100\times 06)}$$

$$\Rightarrow V_2 = 0.221\text{m}^3$$

$$\boxed{V_2 = 0.221\text{m}^3}$$

Step 3 of 3

And also we know that
For isothermal process control equation

$$P_1V_1 = P_2V_2$$

$$P_2 = \frac{P_1V_1}{V_2}$$

$$= \frac{100\times 0.6}{0.221}$$

$$= 271.49\text{kPa}$$

$$\boxed{P_2 = 271.49\text{kPa}}$$

Question: 5.081P

A 1-kg block of iron is heated from 25 to 75°C. What is the change in the iron's total internal energy and enthalpy?

Step 1 of 3

Given:

Mass of the iron block $m = 1\text{kg}$

Initially temperature $T_1 = 25^\circ\text{C}$

Final temperature $T_2 = 75^\circ\text{C}$

Step 2 of 3

Iron is an incompressible substance with a constant specific heat.

From the properties of common liquids, solids and foods

For iron

The specific heat of iron is

$$C_p = 0.45\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 3 of 3

The internal energy and enthalpy changes are equal for a solid.

Then,

$$\Delta H = \Delta U = mC_p\Delta T$$

$$= 1(0.45)(75-25)$$

$$= 22.5\text{kJ}$$

$$\boxed{\Delta H = \Delta U = 22.5\text{kJ}}$$

Question: 5.082P

The state of liquid water is changed from 50 psia and 50°F to 2000 psia and 100°F. Determine the change in the internal energy and enthalpy of water on the basis of the (*a*) compressed liquid tables, (*b*) incompressible substance approximation and property tables, and (*c*) specific-heat model.

Step 1 of 6

a)

Obtain the properties of the water from EES software at $T_1 = 50°F$ and $p_1 = 50\,\text{psia}$

Internal energy, $u_1 = u_f = 18.07\ \text{Btu/lbm}$,

Enthalpy, $h_f = 18.07\ \text{Btu/lbm}$

At $T_1 = 50°F$, saturation pressure is given as $P_{sat} = 0.17812\ \text{psia}$

Calculate the enthalpy of compressed liquid as follows:

$$h_1 = h_f + v_f(P_1 - P_{sat})$$
$$= 18.07\ \text{Btu/lbm} + 0.01602\ \text{ft}^3/\text{lbm} \times (50\,\text{psia} - 0.17812\ \text{psia}) \times \frac{\text{Btu}}{\text{psia.ft}^3}$$
$$= 18.87\ \text{Btu/lbm}$$

Step 2 of 6

Obtain the properties of water from the table A-7E "compressed liquid water" at $P_2 = 2000\ \text{psia}$ and $T_2 = 100°F$.

Internal energy, $u_2 = 67.36\ \text{Btu/lbm}$.

Enthalpy, $h_2 = 73.30\ \text{Btu/lbm}$

Calculate the change in internal energy.

Calculate the change in internal energy.

$$\Delta u = u_2 - u_1$$

Substitute $67.36\ \text{Btu/lbm}$ for u_2 and $18.07\ \text{Btu/lbm}$ for u_1 .

$$\Delta u = 67.36\ \text{Btu/lbm} - 18.07\ \text{Btu/lbm}$$
$$= 49.29\ \text{Btu/lbm}$$

Therefore, the change in internal energy is $\boxed{49.29\ \text{Btu/lbm}}$.

Step 3 of 6

Calculate the change in Enthalpy.

$$\Delta h = h_2 - h_1$$

$$\Delta h = 73.3\ \text{Btu/lbm} - 18.87\ \text{Btu/lbm}$$
$$= 54.43\ \text{Btu/lbm}$$

Therefore, the change in enthalpy is $\boxed{54.43\ \text{Btu/lbm}}$.

Step 4 of 6

(b)

Consider the incompressible substance approximation

Neglecting the pressure condition, using the saturated liquid properties at given temperature

$u_1 = u_f = 18.07\ \text{Btu/lbm}$, $h_1 = h_f = 18.07\ \text{Btu/lbm}$

$u_2 = u_f = 68.03\ \text{Btu/lbm}$, $h_2 = h_f = 68.03\ \text{Btu/lbm}$

Calculate the change in internal energy.

Calculate the change in internal energy.

$$\Delta u = u_2 - u_1$$
$$= 68.03\ \text{Btu/lbm} - 18.07\ \text{Btu/lbm}$$
$$= 49.96\ \text{Btu/lbm}$$

Therefore, the change in enthalpy is $\boxed{49.96\ \text{Btu/lbm}}$.

Step 5 of 6

Calculate the change in enthalpy.

$$\Delta h = h_2 - h_1$$
$$= 68.03\ \text{Btu/lbm} - 18.07\ \text{Btu/lbm}$$
$$= 49.96\ \text{Btu/lbm}$$

Therefore, the change in enthalpy is $\boxed{49.96\ \text{Btu/lbm}}$.

Step 6 of 6

(c)

Consider the Specific heat model.

Obtain the properties from the table of 'properties common liquids, solids and foods' for water

The specific heat of water is $1\ \text{Btu/lbm} \cdot \text{R}$

Calculate the enthalpy and internal energy are same for a solid.

$$\Delta h = \Delta u = c\Delta T$$
$$= \left(1 \frac{\text{Btu}}{\text{lbm.R}}\right) \times (100 - 50)\,\text{R}$$
$$= 50\ \text{Btu/lbm}$$

Therefore change in internal energy and enthalpies are $\boxed{50\ \text{Btu/lbm}}$ and $\boxed{50\ \text{Btu/lbm}}$.

Question: 5.083P

During a picnic on a hot summer day, all the cold drinks disappeared quickly, and the only available drinks were those at the ambient temperature of 75°F. In an effort to cool a 12-fluid-oz drink in a can, a person grabs the can and starts shaking it in the iced water of the chest at 32°F. Using the properties of water for the drink, determine the mass of ice that will melt by the time the canned drink cools to 45°F.

Step 1 of 5

Given

12 oz of fluid drink at ambient temperature $T_1 = 75°F$

Canned drink cools at temperature $T_2 = 45°F$

Step 2 of 5

The specific heat of water is 1 Btu/lbm.R

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work done on the system is zero $W = 0$.

$-Q_{out} = \Delta U$

$-Q_{out} = m(u_2 - u_1)$

$-Q_{out} = mc(T_2 - T_1)$

Step 3 of 5

The mass of water is

$m = \rho V$ lbm/can

$1 \text{ oz} = \frac{1}{957.5}$ ft^3 And

Density of water $\rho = 62.34$ lbm/ft^3

$m = \rho V$ lbm/can

$m = 62.34\left(\frac{12}{957.5}\right)$ lbm/can

$m = 0.7812$ lbm/can

Step 4 of 5

Therefore

$-Q_{out} = mc(T_2 - T_1)$

$-Q_{out} = (0.7812)(1)(45-75)$

$Q_{out} = 23.43$ Btu/lbm

Step 5 of 5

The amount of heat required to melt one lbm of ice is heat of fusion of water

$h_{fusion} = 143.5$ Btu/lbm

Therefore the amount of ice melted in order to dissipate 23.43 Btu/lbm of heat is

$m_{ice} = \frac{Q_{out}}{h_{fusion}}$

$m_{ice} = \frac{23.43}{143.5}$

$\boxed{m_{ice} = 0.163 \text{ lbm}}$

The mass of ice melted to cool one can is 0.163 lbm

Question: 5.084P

Consider a 1000-W iron whose base plate is made of 0.5-cm-thick aluminum alloy 2024-T6 (ρ = 2770 kg/m3 and cp = 875 J/kg·°C). The base plate has a surface area of 0.03 m2. Initially, the iron is in thermal equilibrium with the ambient air at 22°C. Assuming 90 percent of the heat generated in the resistance wires is transferred to the plate, determine the minimum time needed for the plate temperature to reach 200°C.

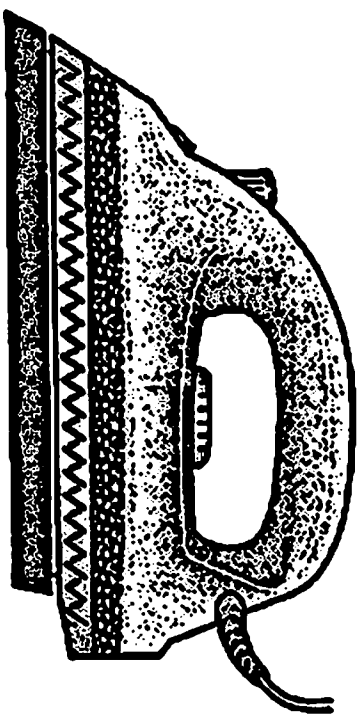


FIGURE P5–84

Step 1 of 4

Given:

Capacity of iron $\dot{Q}_{in} = 1000\,\text{W}$

Thickness of base plate $t = 0.5\,\text{cm}$

Density of aluminum alloy $\rho = 2770\,\frac{\text{kg}}{\text{m}^3}$

Specific heat of aluminum alloy $C_p = 875\,\frac{\text{J}}{\text{kg}\cdot^\circ\text{C}}$

Ambient temperature $T_1 = 22^\circ\text{C}$

Efficiency $\eta = 0.9$

Plate temperature $T_2 = 200^\circ\text{C}$

Surface area of the base plate $A = 0.03\,\text{m}^2$

Step 2 of 4

Mass of the iron is

$$m = \rho \times t \times A$$

$$= \left(2770\,\frac{\text{kg}}{\text{m}^3} \times 0.005\text{m} \times 0.03\text{m}^2\right)$$

$$= 0.4155\text{kg}$$

Step 3 of 4

Noting that only 90% of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{in} = 0.9 \times 1000\,\text{W}$$

$$= 900\,\text{W}$$

Step 4 of 4

We take plate to be the system

The energy balance for this closed system can be expressed as

$$\dot{Q}_{in} = \Delta U_{plate}$$

$$= m(u_2 - u_1)$$

$$\dot{Q}_{in}\Delta t = mc_p(T_2 - T_1)$$

Solving for Δt and substituting

$$\Delta t = \frac{mC\Delta T_{plate}}{\dot{Q}_{in}}$$

$$= \frac{(0.4155)(875)(200-22)}{900}$$

$$= 71.904\,\text{sec}$$

$$\boxed{\Delta t = 71.904\,\text{sec}}$$

Question: 5.085P

Stainless steel ball bearings (ρ = 8085 kg/m3 and cp = 0.480 kJ/kg·°C) having a diameter of 1.2 cm are to be quenched in water at a rate of 800 per minute. The balls leave the oven at a uniform temperature of 900°C and are exposed to air at 25 °C for a while before they are dropped into the water. If the temperature of the balls drops to 850°C prior to quenching, determine the rate of heat transfer from the balls to the air.

Step 1 of 5

Given

Diameter of ball bearings $d = 1.2\text{ cm} = 0.012\text{m}^3$

Ambient air temperature $T_{amb} = 25°\text{C}$

Initial uniform temperature $T_1 = 900°\text{C}$

Final heated temperature $T_2 = 850°\text{C}$

Number of steel ball bearings quenched $n = 800\text{per/min}$

Density of steel ball bearings $\rho = 8085\text{ kg/m}^3$

Specific heat of steel ball bearings $c_p = 0.48\text{ kJ/kg.°C}$

Step 2 of 5

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work done on the system is zero W = 0.

$-Q_{out} = \Delta U$

$-Q_{out} = m(u_2 - u_1)$

$-Q_{out} = mc_p(T_2 - T_1)\text{ kJ}$

Step 3 of 5

The mass of steel ball is

$m = \rho V\text{ kg}$

$m = \rho \times \frac{\pi}{6} d^3$

$m = 8085 \times \left(\frac{\pi}{6}(0.012)^3\right)$

$m = 0.007315\text{ kg}$

Step 4 of 5

The amount of heat transfer from the balls to the air

$-Q_{out} = mc_p(T_2 - T_1)\text{ kJ}$

$-\dot{Q}_{out} = 0.007315 \times (0.48) \times (850 - 900)$

$Q_{out} = 0.17556\text{ kJ}$

Step 5 of 5

Total rate of heat transfer from the balls to the air

$\dot{Q}_{out} = n \times Q_{out}$

$\dot{Q}_{out} = 800 \times 0.17556\text{ kJ/min}$

$\dot{Q}_{out} = 140.448\text{ kJ/min}$

$\dot{Q}_{out} = 140.448 \times \frac{1}{60}$

$\boxed{\dot{Q}_{out} = 2.3408\text{ kW}}$

Question: 5.086P

Carbon steel balls (ρ = 7833 kg/m3 and cp = 0.465 kJ/kg·°C) 8 mm in diameter are annealed by heating them first to 900°C in a furnace, and then allowing them to cool slowly to 100°C in ambient air at 35°C. If 2500 balls are to be annealed per hour, determine the total rate of heat transfer from the balls to the ambient air.

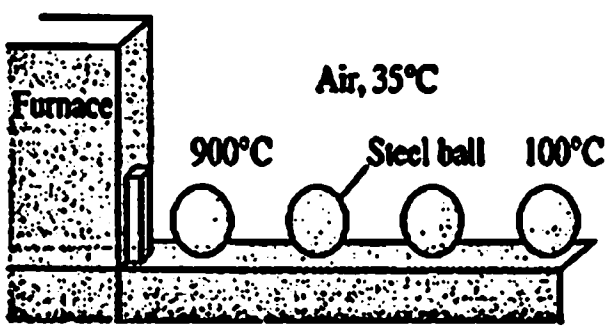


FIGURE P5–86

Step 1 of 5

Given

Diameter of steel ball $d = 8$ mm

Density of steel ball $\rho = 7833\ \text{kg/m}^3$

Specific heat of steel ball $c_p = 0.465$ kJ/kg.°C

Ambient air temperature $T_a = 35°C$

Initial uniform temperature $T_1 = 900°C$

Final heated temperature $T_2 = 100°C$

Number of steel ball bearings annealed per hour $n = 2500$

Step 2 of 5

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work done on the system is zero W = 0.

$-Q_{out} = \Delta U$

$-Q_{out} = m(u_2 - u_1)$

$-Q_{out} = mc_p(T_2 - T_1)$

Step 3 of 5

The mass of steel ball is

$m = \rho V$ kg

Given π ,3

Diameter of steel ball $d = 8$ mm

Density of steel ball $\rho = 7833\ \text{kg/m}^3$

Specific heat of steel ball $c_p = 0.465$ kJ/kg.°C

Ambient air temperature $T_a = 35°C$

Initial uniform temperature $T_1 = 900°C$

Final heated temperature $T_2 = 100°C$

Number of steel ball bearings annealed per hour $n = 2500$

Step 2 of 5

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work done on the system is zero W = 0.

$-Q_{out} = \Delta U$

$-Q_{out} = m(u_2 - u_1)$

$-Q_{out} = mc_p(T_2 - T_1)$

Step 3 of 5

The mass of steel ball is

$m = \rho V$ kg

π ,3

Question: 5.087P

An ordinary egg can be approximated as a 5.5-cm-diameter sphere. The egg is initially at a uniform temperature of 8°C and is dropped into boiling water at 97°C. Taking the properties of the egg to be ρ = 1020 kg/m3 and *cp* = 3.32 kJ/kg·°C, determine how much heat is transferred to the egg by the time the average temperature of the egg rises to 80°C.

Step 1 of 4

Given

Diameter of egg $d = 5.5\ \text{cm} = 0.055\ \text{m}$

Initial temperature of egg $T_1 = 8°\text{C}$

Final temperature of egg $T_2 = 80°\text{C}$

Density of the egg $\rho = 1020\ \text{kg/m}^3$

Specific heat of the egg $c_p = 3.32\ \text{kJ/kg.°C}$

Step 2 of 4

Question:

An ordinary egg can be approximated as a 5.5-cm-diameter sphere. The egg is initially at a uniform temperature of 8°C and is dropped into boiling water at 97°C. Taking the properties of the egg to be ρ = 1020 kg/m3 and *cp* = 3.32 kJ/kg·°C, determine how much heat is transferred to the egg by the time the average temperature of the egg rises to 80°C.

Step 1 of 4

Given

Diameter of egg $d = 5.5\ \text{cm} = 0.055\ \text{m}$

Initial temperature of egg $T_1 = 8°\text{C}$

Final temperature of egg $T_2 = 80°\text{C}$

Density of the egg $\rho = 1020\ \text{kg/m}^3$

Specific heat of the egg $c_p = 3.32\ \text{kJ/kg.°C}$

Step 2 of 4

Step 4 of 4

The amount of heat transfer

$$Q_{in} = mc_p(T_2 - T_1)$$

$$Q_{in} = 0.089(3.32)(80-8)$$

$$\boxed{Q_{in} = 21.274\ \text{kJ}}$$

Question: 5.088P

In a production facility, 1.2-in-thick 2-ft × 2-ft square brass plates (ρ = 532.5 lbm/ft3 and cp = 0.091 Btu/lbm. °F) that are initially at a uniform temperature of 75°F are heated by passing them through an oven at 1300°F at a rate of 300 per minute. If the plates remain in the oven until their average temperature rises to 1000°F, determine the rate of heat transfer to the plates in the furnace.

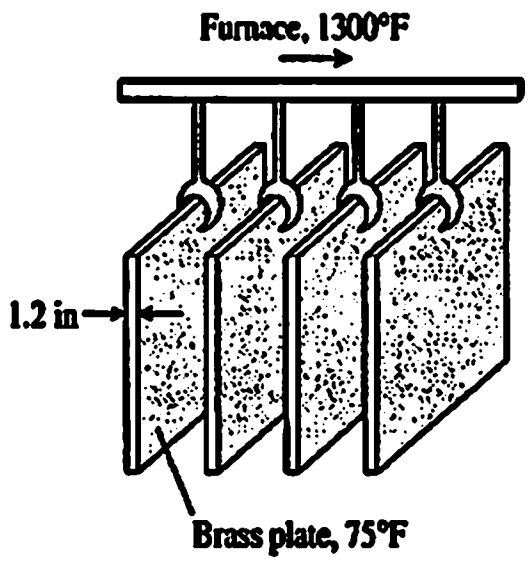


FIGURE P5–88E

Step 1 of 5

Given

Thickness of plate $t = 1.2$ in ,

Cross section of the brass plate $2\text{ft} \times 2\text{ft}$

Initial temperature of the plate $T_1 = 75°\text{F}$

Final temperature of the plate $T_2 = 1000°\text{F}$

Density of the brass plate $\rho = 532.5\text{lbm/ft}^3$

Specific heat of the brass plate $c_p = 0.091\text{Btu/lbm.}^0\text{F}$

Number of plates moving per minute is $N = 300\text{plates/min}$

Step 2 of 5

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Work done on the system is zero $W = 0$.

$Q_{in} = \Delta U$

$Q_{in} = m(u_2 - u_1)$

$Q_{in} = mc_p(T_2 - T_1)$ kJ

Step 3 of 5

The mass of each aluminum plate

$m = \rho V$

$m = \rho AL$

$m = 532.5 \times (2 \times 2) \times \left(\frac{1.2}{12}\right)$

$m = 213$ lbm

Step 4 of 5

The amount of heat transfer

$Q_{in} = mc_p(T_2 - T_1)$ kJ

$Q_{in} = 213 \times (0.091) \times (1000 - 75)$

$Q_{in} = 17929.275$ Btu/plate

Step 5 of 5

The total rate of heat transfer for 300 plates per minute

$\dot{Q}_{in} = \frac{300}{60} \times 17929.275$ Btu/s

$\boxed{\dot{Q}_{in} = 89646.375\text{ Btu/s}}$

Question: 5.089P

Long cylindrical steel rods (ρ = 7833 kg/m3 and cp = 0.465 kJ/kg·°C) of 8-cm diameter are heat-treated by drawing them at a velocity of 2 m/min through an oven maintained at 900°C. If the rods enter the oven at 30°C and leave at a mean temperature of 700°C, determine the rate of heat transfer to the rods in the oven.

Step 1 of 4

Given:

Density of ball bearing $\rho = 7833\frac{\text{kg}}{\text{m}^3}$

Specific heat $c_p = 0.465\frac{\text{kJ}}{\text{kg}\cdot^\circ\text{C}}$

Diameter of ball $D = 8\text{cm}$

Velocity $V = 2\frac{\text{m}}{\text{min}}$

Temperature of the oven $T_2 = 700^\circ\text{C}$

Inlet temperature rod $T_1 = 30^\circ\text{C}$

Step 2 of 4

The energy balance for this closed system can be expressed as:-

$Q_{in} = mC(T_1 - T_2)$

The mass of rod heated in 1 minute is

$m = \rho VA$

$= \rho V\left(\frac{\pi D^2}{4}\right)$

$= (7833)(2)\left[\frac{\pi(0.08)^2}{4}\right]$

$= 78.745\text{kg}$

Step 3 of 4

We take the 2m section of the rod in the oven as the system

$Q_{in} = mC(T_2 - T_1)$

$= (78.74)(0.465)(700-30)$

$= 24533.286\text{kJ}$

Step 4 of 4

Noting that this much heat is transferred in 1 min., the rate of heat transfer to the rod becomes

$Q_{in} = \frac{Q_{in}}{\Delta t}$

$= \frac{24533.2861\text{kJ}}{1\text{min}}$

$= \frac{408.88\text{kJ}}{60\text{sec}}$

$= 408.88\text{kW}$

$\boxed{Q_{in} = 408.88\text{kW}}$

Question: 5.090P

An electronic device dissipating 25 W has a mass of 20 g and a specific heat of 850 J/kg·°C. The device is lightly used, and it is on for 5 min and then off for several hours, during which it cools to the ambient temperature of 25°C. Determine the highest possible temperature of the device at the end of the 5-min operating period. What would your answer be if the device were attached to a 0.5-kg aluminum heat sink? Assume the device and the heat sink to be nearly isothermal.

Step 1 of 5

Given:

Heat dispatching rate $Q = 25\,\text{W}$

Mass of heater $m = 20\,\text{g}$

Specific heat $c_p = 850\,\frac{\text{J}}{\text{kg}\cdot^\circ\text{C}}$

Working time $\Delta t = 5\,\text{min}$

Ambient temperature $T_1 = 25^\circ\text{C}$

Mass of the heat sink $m_H = 0.5\text{kg}$

Step 2 of 5

Take the specific heat of aluminum
From the properties of common liquids and solids and gases
At room temperature of 300K is

$$c_{p,al} = 902\,\frac{\text{J}}{\text{kg}\cdot^\circ\text{C}}$$

Step 3 of 5

The energy balance for this closed system can be expressed as

$$W_{e,in} = \Delta U_{device} = m(u_2 - u_1)$$

$$W_{e,in} = \Delta T_{device} = mc_p(T_2 - T_1)$$

$$Q \times \Delta t = mc_p(T_2 - T_1)$$

Step 4 of 5

Substituting values in above expression
the temperature of the deice at the end of the process is

$$25\text{W} \times (5 \times 60\,\text{sec}) = (0.02\text{kg})\left(850\,\frac{\text{J}}{\text{kg}\cdot^\circ\text{C}}\right)(T_2 - 25)^\circ\text{C}$$

$$\Rightarrow T_2 = 466.18^\circ\text{C}$$

$$\boxed{T_2 = 466.18^\circ\text{C}}$$

Step 5 of 5

Case II:-
When a heat sink is attached the energy balance can be expressed as

$$W_{e,in} = \Delta U_{device} + \Delta U_{heat\,sink}$$

$$W_{e,in}\Delta t = mc_p(T_2 - T_1)_{device} + m_{al}c_{p,al}(T_2 - T_1)_{sink}$$

$$(25) \times (5 \times 60) = (0.02)(850)(T_2 - 25) + (0.5)(902)(T_2 - 25)$$

$$\Rightarrow T_2 = 41.03^\circ\text{C}$$

$$\boxed{T_2 = 41.03^\circ\text{C with heat sink}}$$

Question: 5.091P

Reconsider Prob. 5–90. Using EES (or other) software, investigate the effect of the mass of the heat sink on the maximum device temperature. Let the mass of heat sink vary from 0 to 1 kg. Plot the maximum temperature against the mass of heat sink, and discuss the results.

Step 1 of 7

Given:
Given:
Heat dispatching rate $Q = 25\,\text{W}$
Mass of heater $m = 20\,\text{g}$
Specific heat $c_p = 850\,\frac{\text{J}}{\text{kg}\cdot{}^\circ\text{C}}$
Working time $\Delta t = 5\,\text{min}$
Ambient temperature $T_1 = 25^\circ\text{C}$
Mass of the heat sink varies from $m_H = 0\text{kg}$ to 1kg

Step 2 of 7

"Knowns:"
"T_1 is the maximum temperature of the device"
Q_dot_out = 25 [W]
m_device=20 [g]
Cp_device=850 [J/kg-C]
A=5 [cm^2]
DELTAt=5 [min]
T_amb=25 [C]
{m_sink=0.5 [kg]}
"Cp_al taken from Table A-3(b) at 300K"
Cp_al=0.902 [kJ/kg-C]
T_2=T_amb

Step 3 of 7

"Solution:"
"The device without the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot, we neglect DELTA KE and DELTA PE for the system, the device."
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in =0
E_dot_out = Q_dot_out
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot= m_device*convert(g,kg)*Cp_device*(T_2-
_1_device)/(DELTAt*convert(min,s))
"The device with the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"

Step 4 of 7

"E_dot_in - E_dot_out = DELTAE_dot, we neglect DELTA KE and DELTA PE for the device with the heat sink."
E_dot_in - E_dot_out = DELTAE_dot_combined
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot_combined= (m_device*convert(g,kg)*Cp_device*(T_2-
T_1_device&sink)+m_sink*Cp_al*(T_2-
T_1_device&sink)*convert(kJ,J))/(DELTAt*convert(min,s))

Step 5 of 7

Run the parameter table on menu bar.
Then select the m_sink and $T_{1\text{device\&sink}}$ then click ok

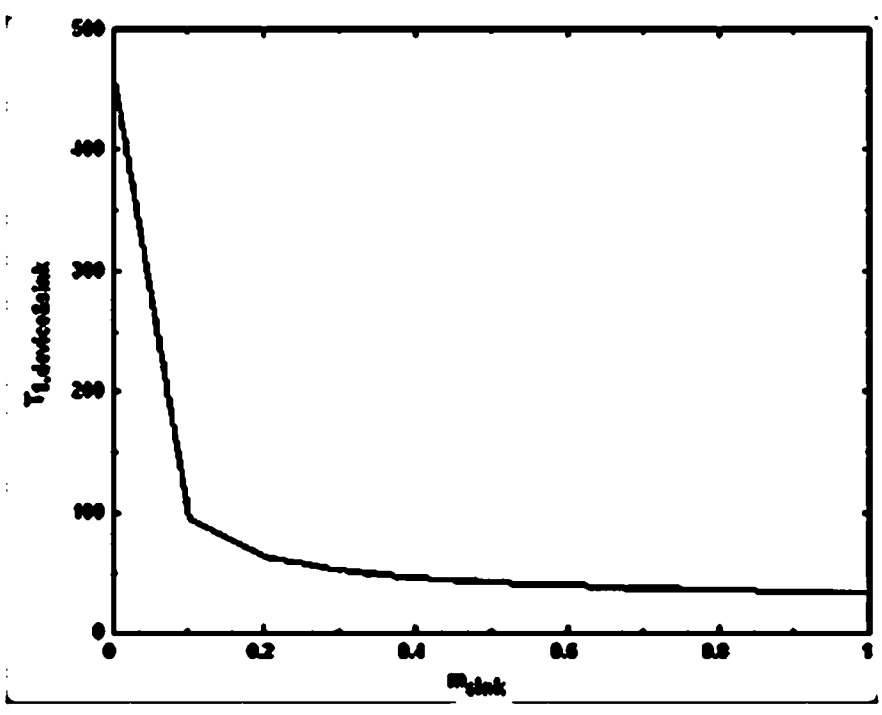
Step 6 of 7

Solution table :

1..11	m_{sink}	$T_{1,device\&sink}$
Run 1	0	466.2
Run 2	0.1	94.96
Run 3	0.2	62.99
Run 4	0.3	51.08
Run 5	0.4	44.85
Run 6	0.5	41.03
Run 7	0.6	38.44
Run 8	0.7	36.57
Run 9	0.8	35.15
Run 10	0.9	34.05
Run 11	1	33.16

Step 7 of 7

Graph between m_sink nad T_sinkdevice



Question: 5.092P

If you ever slapped someone or got slapped yourself, you probably remember the burning sensation. Imagine you had the unfortunate occasion of being slapped by an angry person, which caused the temperature of the affected area of your face to rise by 1.8°C (ouch!). Assuming the slapping hand has a mass of 1.2 kg and about 0.150 kg of the tissue on the face and the hand is affected by the incident, estimate the velocity of the hand just before impact. Take the specific heat of the tissue to be 3.8 kJ/kg·K.

Step 1 of 2

Given:

Rise in temperature $\Delta T = 1.8^{\circ}\mathrm{C}$

Mass of the hands $m_1 = 1.2\,\mathrm{kg}$

Mass of the tissue on face $m_2 = 0.150\,\mathrm{kg}$

Specific heat of tissue $c_p = 3.8\,\dfrac{\mathrm{kJ}}{\mathrm{kg}\cdot\mathrm{K}}$

Step 2 of 2

In this process kinetic energy of hand is converted into heat energy

$$\frac{1}{2}m_1V^2 = m_2c_p\Delta t$$

$$\frac{1}{2}\times(1.2)(V^2) = 0.15\times3.8\times10^3\times1.8$$

$$\Rightarrow V = 41.352\,\frac{\mathrm{m}}{s}$$

$$\boxed{V = 41.352\,\frac{\mathrm{m}}{s}}$$

Question: 5.093P

Does the concept of a specific heat property (either at constant volume or constant pressure) have a meaning for substances that are undergoing a phase change? Why or why not?

Step 1 of 2

For a substance undergoing phase change, the temperature during the phase change remains constant at constant pressure process.

Therefore we know that specific heat at constant pressure is

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p$$

$$= \left(\frac{\partial h}{0}\right)_p$$

$$c_p = \infty$$

So specific heat property undergoing phase change at constant pressure process has no meaning.

Step 2 of 2

For specific heat at constant volume

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v$$

It gives some finite value and it has some meaning since at constant volume process during phase change there might be finite amount of heat transfer to the system.

Question: 5.094P

10-kg of nitrogen is heated from 20°C to 250°C. Determine the total amount of heat required when this is done in (*a*) a constant-volume process and (*b*) an isobaric process.

Step 1 of 4

Given

Mass of nitrogen $m = 10\ \text{kg}$

Initial temperature $T_1 = 20°\text{C}$

Final temperature $T_2 = 250°\text{C}$

Step 2 of 4

For nitrogen the specific heats at average temperatures

$$T_{avg} = \frac{T_1 + T_2}{2}$$

$$= \frac{20 + 250}{2}$$

$$T_{avg} = 135°\text{C} = 408\text{K}$$

At $T_{avg} = 135°\text{C} = 408\text{K}$ from table ideal gas specific heats of common gases

$$c_p = 1.044 + (1.049 - 1.044)\left(\frac{408 - 400}{450 - 400}\right)$$

$$c_p = 1.0448\ \text{kJ/kg.K}$$

$$c_v = 0.747 + (0.752 - 0.747)\left(\frac{408 - 400}{450 - 400}\right)$$

$$c_v = 0.7478\ \text{kJ/kg.K}$$

Step 3 of 4

(a) If the process is constant volume process

By applying energy equation and neglecting kinetic and potential energies

$$Q = W + \Delta U$$

Since the process is constant volume process work done is zero then

$$Q_{in} = \Delta U$$

$$Q_{in} = mc_v(T_2 - T_1)$$

$$Q_{in} = 10(0.7478)(250 - 20)$$

$$\boxed{Q_{in} = 1719.94\ \text{kJ}}$$

Step 4 of 4

(b) If the process is an isobaric process

$$Q = W + \Delta U$$

Where work done during constant pressure process is

$$W = P(V_2 - V_1)$$

Then the amount of heat required is

$$Q_{in} = P(V_2 - V_1) + mc_v(T_2 - T_1)$$

$$Q_{in} = mR(T_2 - T_1) + mc_v(T_2 - T_1)$$

$$Q_{in} = m(T_2 - T_1)(R + c_v)$$

$$Q_{in} = m(T_2 - T_1)c_p$$

$$Q_{in} = mc_p(T_2 - T_1)$$

$$Q_{in} = 10(1.0448)(250 - 20)$$

$$\boxed{Q_{in} = 2403.04\ \text{kJ}}$$

Question: 5.095P

Consider a classroom that is losing heat to the outdoors at a rate of 12,000 kJ/h. If there are 40 students in class, each dissipating sensible heat at a rate of 84 W, determine if it is necessary to turn the heater in the classroom on to prevent the room temperature from dropping.

Step 1 of 2

Given:

Number of students $n = 40$

Rate of heat dissipation of classroom $Q_l = 12000 \ \frac{\text{kJ}}{\text{h}}$

Rate of heat dissipation of sensible heat $Q = 84\text{W}$

Step 2 of 2

Rate of heat dissipation of classroom

$$Q_l = 12000 \ \frac{\text{kJ}}{\text{h}}$$

$$Q_l = \frac{12000}{3600} \frac{\text{kJ}}{\text{sec}}$$

$$= 3.333\text{kW}$$

Rate of heat produced by students

$$Q_{prod} = n \times Q$$

$$= 40 \times 0.084\text{kW}$$

$$= 3.36\text{kW}$$

So, rate of heat production in room is greater than rate of heat dissipation to the outdoors. So it is not necessary to turn the heater on in the classroom to prevent the room temperature from dropping.

Question: 5.096P

Air in the amount of 2 lbm is contained in a well-insulated, rigid vessel equipped with a stirring paddle wheel. The initial state of this air is 30 psia and 60°F. How much work, in Btu, must be transferred to the air with the paddle wheel to raise the air pressure to 40 psia? Also, what is the final temperature of air?

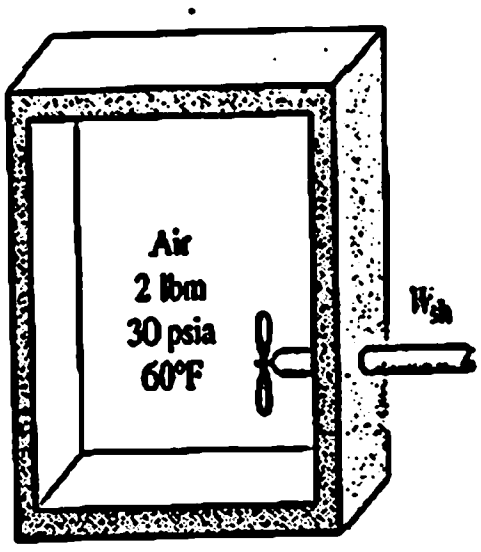


FIGURE P5–96E

Step 1 of 6

Given:

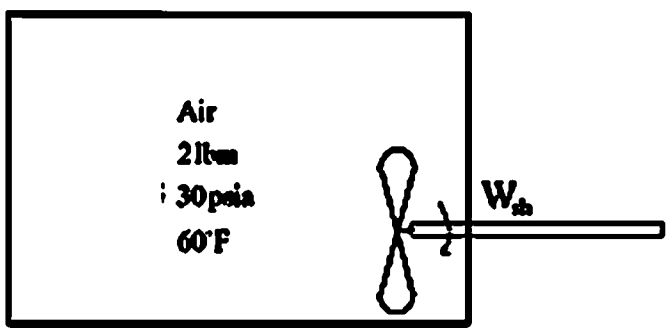
Mass of the air $m = 2\text{lbm}$

Initially pressure $P_1 = 30\text{psia}$

Initially temperature $T_1 = 60^\circ\text{F}$

Final pressure $P_2 = 40\text{psia}$

Step 2 of 6



Step 3 of 6

From the ideal gas specific heat of various common gases

For air

The specific heats of air at room temperature are

$$C_p = 0.240\frac{\text{Btu}}{\text{lbm}\cdot\text{R}} \text{ And}$$

$$C_v = 0.171\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$$

Step 4 of 6

The energy balance for this system can be expressed as

$Q = W + \Delta U$

Since tank is insulated so heat transfer is zero,and work done on the system so take as negative.

$$-\left(-W_{sh,in}\right) = \Delta U$$

$$W_{sh,in} = \Delta U$$

$$= mC_v\left(T_2 - T_1\right)$$

Step 5 of 6

As the specific volume remains constant during this process, the ideal gas equation gives

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)$$

$$= (60+459.67\text{R})\left(\frac{40}{30}\right)$$

$$= 692.893\text{R}$$

$$= 692.893\text{-}459.67\text{R}$$

$$= 233.233^\circ\text{F}$$

$$\boxed{T_2 = 233.233^\circ\text{F}}$$

Step 6 of 6

Substituting,

Work input is

$$W_{sh,in} = mC_v\left(T_2 - T_1\right)$$

$$= (2)(0.171)(692.893 - 519.67)$$

$$= 59.24\text{Btu}$$

$$\boxed{W_{sh,in} = 59.24\text{Btu}}$$

Question: 5.097P

Air is expanded in a polytropic process with n =1.5 from 2 MPa and 300°C to 80 kPa in a piston-cylinder device. Determine the final temperature of the air.

Step 1 of 3

Write the governing law for a polytropic process.

$$P_1V_1^n = P_2V_2^n$$

Here, p_1 is the initial pressure, p_2 is the final pressure, T_1 is the initial temperature and T_2 is the final temperature.

Also, for an ideal gas like air:

$$PV = mRT$$

$$V = \frac{mRT}{P}$$

Since m and R are constants:

$$V \propto \frac{T}{P}$$

Step 2 of 3

Hence, for a polytropic process:

$$P_1\left(\frac{T_1}{P_1}\right)^n = P_2\left(\frac{T_2}{P_2}\right)^n$$

$$P_1^{1-n}T_1^n = P_2^{1-n}T_2^n$$

$$P_1^{n-1}T_1^n = P_2^{n-1}T_2^n$$

$$\left(\frac{T_2}{T_1}\right)^n = \left(\frac{P_2}{P_1}\right)^{n-1}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Step 3 of 3

Convert the initial temperature from Celsius to Kelvin.

$$T_1 = 300°C$$

$$T_1 = (300+273)\ K$$

$$T_1 = 573\ K$$

Substitute 573 K for T_1, 80 kPa for P_2 and 2000 kPa for P_1.

$$T_2 = 573\times\left(\frac{80}{2000}\right)^{\frac{1.5-1}{1.5}}$$

$$= 195.96\ K$$

Therefore, the final temperature of air is $\boxed{195.96K}$.

Question: 5.098P

Nitrogen at 100 kPa and 25 °C in a rigid vessel is heated until its pressure is 300 kPa. Calculate the work done and the heat transferred during this process, in kJ/kg.

Step 1 of 4

Loading solution. Please wait ...

Step 2 of 4

Since the vessel is rigid, so volume remains constant.
So work done

$$W = PdV$$
$$= P(0-0)$$
$$= 0$$
$$\boxed{W = 0}$$

Step 3 of 4

From the ideal gas relation

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$
$$\Rightarrow T_2 = T_1\left(\frac{P_2}{P_1}\right)$$
$$= 298\left(\frac{300}{100}\right)$$
$$= 894\text{K}$$

Step 4 of 4

c_v is the specific heat of nitrogen
From the ideal gas specific heat of various gases
For nitrogen

$$c_v = 0.743\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Heat transferred

$$Q = c_v(T_2 - T_1)$$
$$= 0.743(894-298)$$
$$= 442.828\frac{\text{kJ}}{\text{kg}}$$
$$\boxed{Q = 442.828\frac{\text{kJ}}{\text{kg}}}$$

Question: 5.099P

A well-insulated rigid vessel contains 3 kg of saturated liquid water at 40°C. The vessel also contains an electrical resistor that draws 10 amperes when 50 volts are applied. Determine the final temperature in the vessel after the resistor has been operating for 30 minutes.

Step 1 of 3

Given data:

Mass of the saturated water $m = 3\,\text{kg}$

Initial temperature of the water $T_1 = 40°\text{C} = 313\,\text{K}$

Current drawn by the resister, $I = 10\,\text{A}$

Voltage rating of the resistor, $V = 50\,\text{V}$

Operation time for heat application, $t = 30\,\text{min} = 1800\,\text{sec}$

Internal energy of the saturated water at given conditions, $u_1 = 167.53\,\text{kJ/kg}$

Step 2 of 3

Heat added to the system due to electric resistance is given by,

$$\begin{aligned} Q_{in} &= V \times I \times t \\ &= 50 \times 10 \times 1800 \\ &= 900000\,\text{J} \\ &= 900\,\text{kJ} \end{aligned}$$

The energy balance is given by the formula,

$$Q - W = \Delta E$$

Since the given tank is well-insulated heat lost $Q_{out} = 0$,

The tank is said to be rigid therefore $W = 0$

Step 3 of 3

The entire change in energy is given by change in internal energy alone,

$$\begin{aligned} \Delta E &= m(\Delta u) \\ &= m(u_2 - u_1) \\ &= 3(u_2 - 167.53) \end{aligned}$$

On substitution of the obtained data in the energy balance equation we have,

$$Q - W = \Delta E$$

$$Q_{in} - 0 = m(\Delta u)$$

$$900 - 0 = 3(u_2 - 167.53)$$

$$u_2 = 467.53\,\text{kJ}$$

The water at its initial condition is in saturated state the further increase in its temperature will cause compression of water. Therefore for the internal energy 467.53 kJ we obtain the final temperature of the water as $\boxed{T_2 = 118.9°\text{C}}$ for the compressed water tables of EES.

Question: 5.100P

A frictionless piston-cylinder device initially contains air at 100 kPa and 0.15 m3. At this state, a linear spring ($F \propto x$) is touching the piston but exerts no force on it. The air is now heated to a final state of 0.45 m3 and 800 kPa. Determine (*a*) the total work done by the air and (*b*) the work done against the spring. Also, show the process on a *P*- v diagram.

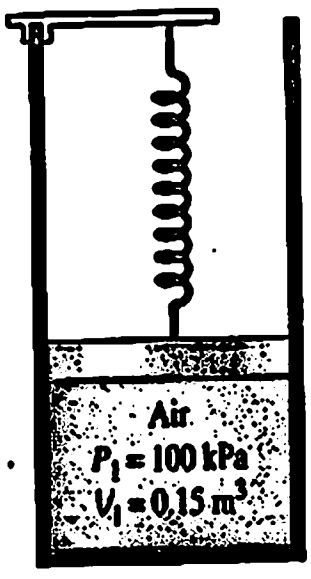


FIGURE P5–100

Step 1 of 5

Given:

Initially pressure $P_1 = 100\text{kPa}$

Initially volume $V_1 = 0.15\text{m}^3$

Final pressure. $P_2 = 800\text{kPa}$

Final volume $V_2 = 0.45\text{m}^3$

Step 2 of 5

P-v Diagram:

Step 3 of 5

(a) The boundary work during this process is simply the area under the process curve Here Displacement is a linear function of fore . Thus,

$W_{b,out}$ = Area under the Pv Diagram

$$= \left(\frac{P_1 + P_2}{2}\right)(V_2 - V_1)$$

$$= \frac{(100+800)}{2}(0.45-0.15)$$

$$= 135\text{kJ}$$

$$\boxed{W_{b,out} = 135\text{kJ}}$$

Step 4 of 5

(b) If there were no spring, we would have a constant process at $P = 100kPa$. The work done during this process is

$$W_{b,out,no\ spring} = \int_1^2 PdV$$

$$= P(V_2 - V_1)$$

$$= 100(0.45-0.15)$$

$$= 30\text{kJ}$$

Step 5 of 5

Thus,

$$W_{spring} = W_b - W_{b,no\ spring}$$

$$= 135 - 30$$

$$= 105\text{kJ}$$

$$\boxed{W_{spring} = 105\text{kJ}}$$

Question: 5.101P

A mass of 5 kg of saturated liquid–vapor mixture of water is contained in a piston-cylinder device at 125 kPa. Initially, 2 kg of the water is in the liquid phase and the rest is in the vapor phase. Heat is now transferred to the water, and the piston, which is resting on a set of stops, starts moving when the pressure inside reaches 300 kPa. Heat transfer continues until the total volume increases by 20 percent. Determine (*a*) the initial and final temperatures, (*b*) the mass of liquid water when the piston first starts moving, and (*c*) the work done during this process. Also, show the process on a P-v diagram.

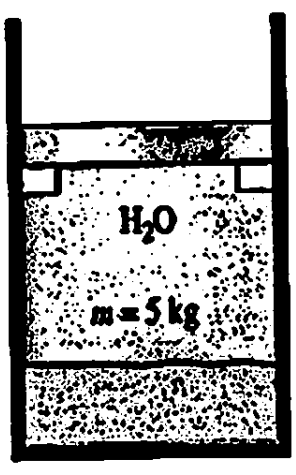


FIGURE P5–101

Step 1 of 9

Given

Mass of saturated vapor liquid $m = 5$ kg

Initial mass of liquid water $m_f = 2$ kg

Initial mass of vapor $m_g = 3$ kg

Initial pressure $P_1 = 125$ kPa

Final pressure $P_3 = 300$ kPa

Volume $V_1 = V_2, V_3 = 1.2V_1$

Step 2 of 9

(a)

At $P_1 = 125$ kPa from saturated water tables

$v_f = 0.001048\ \text{m}^3/\text{kg}$

$v_g = 1.375\ \text{m}^3/\text{kg}$

$T_{sat} = 105.97^0\text{C}$

Since initial state is in wet region then the initial temperature will be the saturation temperature at pressure $P_1 = 125$ kPa

$T_{sat} = T_1$

$\boxed{T_1 = 105.97^\circ\text{C}}$

Step 3 of 9

The initial volume is expressed as

$V_1 = m_f v_f + m_g v_g$

$V_1 = 2(0.001048) + 3(1.375)$

$V_1 = 4.127\ \text{m}^3$

Step 4 of 9

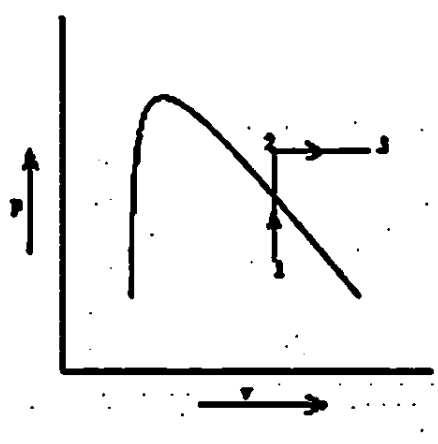
Piston final volume is

$V_3 = 1.2V_1$

$V_3 = 1.2(4.127)$

$V_3 = 4.95\ \text{m}^3$

Step 5 of 9



Step 6 of 9

Therefore the final specific volume will be

$$v_3 = \frac{V_3}{m}$$

$$= \frac{4.95}{5}$$

$v_3 = 0.99\ \text{m}^3/\text{kg}$

Step 7 of 9

So at $P_3 = 300$ kPa and $v_3 = 0.99\ \text{m}^3/\text{kg}$ from superheated water tables by using interpolation

$$T_3 = 300 + (400 - 300)\left(\frac{0.99 - 0.87535}{1.03155 - 0.87535}\right)$$

$\boxed{T_3 = 373.4^0\text{C}}$

Step 8 of 9

(b) The mass of liquid water

When the piston first starts moving $P_2 = 300$ kPa, $V_1 = V_2$, The mass of liquid will be converted into gaseous phase

$\boxed{m_f = 0\ \text{kg}}$

Step 9 of 9

(c) The work done during the process

No work is done during process 1-2 since it's a constant volume process. The pressure remains constant during process 2-3

Therefore the work done in constant pressure process is

$W = W_{1-2} + W_{2-3}$

$W = 0 + W_{2-3}$

$W = \int P dV$

$W = P_2(V_3 - V_2)$

$W = 300(4.95 - 4.127)$

$\boxed{W = 247\ \text{kJ}}$

Question: 5.102P

A spherical balloon contains 7 lbm of air at 30 psia and 600 R. The balloon material is such that the pressure inside is always proportional to the square of the diameter. Determine the work done when the volume of the balloon doubles as a result of heat transfer.

Step 1 of 7

Given:

Mass of the air $m = 7\text{lbm}$

Initially pressure $P_1 = 30\text{psia}$

Initially temperature $T_1 = 600\text{R}$

Pressure is directly proportional to square of the diameter $P \propto D^2$

Step 2 of 7

Step 3 of 7

From the molar mass, gas constant and critical points properties
For air

The gas constant of air is $\text{R} = 0.06855 \dfrac{\text{Btu}}{\text{lbm} \cdot \text{R}}$

The dependence of pressure on volume can be expressed as

$$V = \frac{1}{6}\pi D^3$$

$$\Rightarrow D = \left(\frac{6V}{\pi}\right)^{1/3}$$

Step 4 of 7

$$P \propto D^2$$

$$\Rightarrow P = kD^2$$

$$= k\left(\frac{6V}{\pi}\right)^{2/3}$$

$$k\left(\frac{6}{\pi}\right)^{2/3} = P_1V_1^{-2/3} = P_2V_2^{-2/3}$$

Also,

$$\frac{P_2}{P_1} = \left(\frac{V_2}{V_1}\right)^{2/3}$$

$$= 2^{2/3}$$

$$= 1.587$$

Step 5 of 7

And
From the ideal gas relationship ,we know that

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\Rightarrow T_2 = \frac{P_2V_2}{P_1V_1}T_1$$

$$= 1.587 \times 2 \times (600\text{R})$$

$$= 1904.4\text{R}$$

Step 6 of 7

Thus,
The work done in this process is

$$W = \int_1^2 P\,dV$$

$$W = \int C\left(\frac{6V}{\pi}\right)^{\frac{2}{3}} dV$$

$$W = \int_1^2 C\left(\frac{6V}{\pi}\right)^{\frac{2}{3}} dV$$

$$W = C\left(\frac{6}{\pi}\right)^{\frac{2}{3}} \frac{\left(V_2^{5/3} - V_1^{5/3}\right)}{\frac{5}{3}}$$

$$W = C\left(\frac{6}{\pi}\right)^{\frac{2}{3}} \times \frac{3}{5} \times \left(V_2^{5/3} - V_1^{5/3}\right)$$

$$W = \left(\frac{6}{\pi}\right)^{\frac{2}{3}} \times \frac{3}{5} \times \left(CV_2^{5/3} - CV_1^{5/3}\right)$$

$$W = \left(\frac{6}{\pi}\right)^{\frac{2}{3}} \times \frac{3}{5} \times \left(\frac{P_2}{\left(\frac{6}{\pi}\right)^{2/3}(V_2)^{\frac{2}{3}}} V_2^{5/3} - \frac{P_1}{\left(\frac{6}{\pi}\right)^{2/3}(V_1)^{\frac{2}{3}}} V_1^{5/3}\right)$$

$$W = \frac{3}{5} \times (P_2V_2 - P_1V_1)$$

$$W = \frac{3}{5} \times mR(T_2 - T_1)$$

Step 7 of 7

Substitute the values in above expression

$$W_b = \frac{3}{5}(7)(0.06855)(1904.4 - 600)$$

$$= 375.55\text{Btu}$$

$$\boxed{W_b = 375.55\text{Btu}}$$

Question: 5.103P

Reconsider Prob. 5-102E. Using the integration feature of the EES software, determine the work done. Compare the result with your "hand-calculated" result.

Step 1 of 6

Given:
Given:
Mass of the air $m = 7\,\text{lbm}$
Initially pressure $P_1 = 30\,\text{psia}$
Initially temperature $T_1 = 600\text{R}$
Pressure is directly proportional to square of the diameter $P \propto D^2$

Step 2 of 6

The problem is solved using EES, and the solution is given below.

```
"N=2"
m=7 [lbm]
P_1=30 [psia]
T_1=600 [R]
V_2=2*V_1
R=R_u/molarmass(air)
R=1545 [ft-lbf/lbmol-R]
P_1*Convert(psia,lbf/ft^2)*V_1=m*R*T_1
V_1=4*pi*(D_1/2)^3/3"[ft^3]"
C=P_1/D_1^N
(D_1/D_2)^3=V_1/V_2
P_2=C*D_2^N"[psia]"
P_2*Convert(psia,lbf/ft^2)*V_2=m*R*T_2
```

Step 3 of 6

```
P=C*D^N*Convert(psia,lbf/ft^2)"[ft^2]"
V=4*pi*(D/2)^3/3"[ft^3]"
W_boundary_EES=integral(P,V,V_1,V_2)*convert(ft-lbf,Btu)"[Btu]"
W_boundary_HAND=pi*C/(2*(N+3))*(D_2^(N+3)-D_1^(N+3))*convert(ft-lbf,Btu)*convert(ft^2,in^2)"[Btu]"
```

Step 4 of 6

Run the parameter table on menu bar.
Then select the N and $W_{boundary,EES}$ then click ok

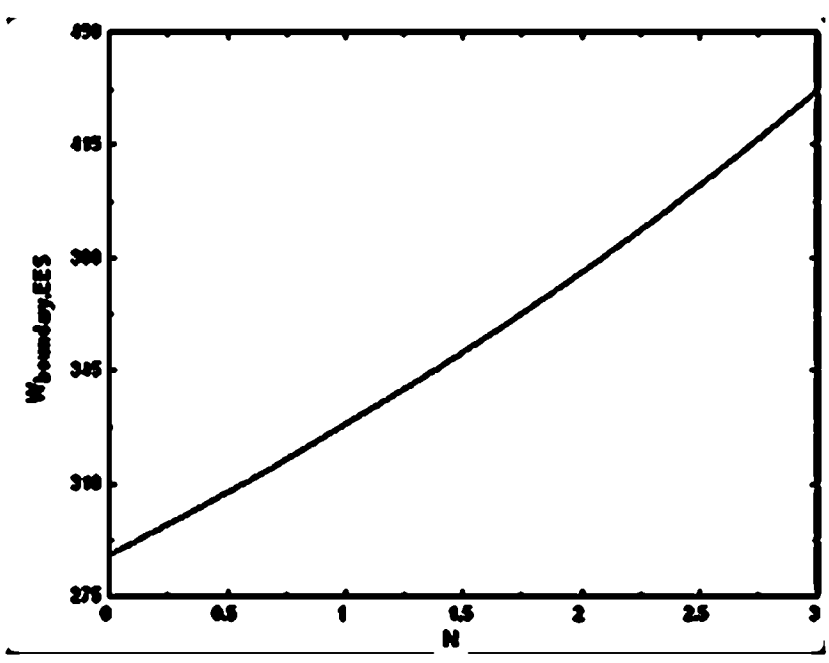
Step 5 of 6

Solution table:

1..10	N	$W_{boundary,HAND}$
Run 1	0	287.9
Run 2	0.3333	300.6
Run 3	0.6667	314
Run 4	1	328.1
Run 5	1.333	343.1
Run 6	1.667	359.9
Run 7	2	375.6
Run 8	2.333	393.3
Run 9	2.667	412
Run 10	3	431.8

Step 6 of 6

Graph between the N and $W_{boundary,EES}$



Question: 5.104P

A mass of 12 kg of saturated refrigerant-134a vapor is contained in a piston-cylinder device at 240 kPa. Now 300 kJ of heat is transferred to the refrigerant at constant pressure while a 110-V source supplies current to a resistor within the cylinder for 6 min. Determine the current supplied if the final temperature is 70°C. Also, show the process on a *T*- v diagram with respect to the saturation lines.

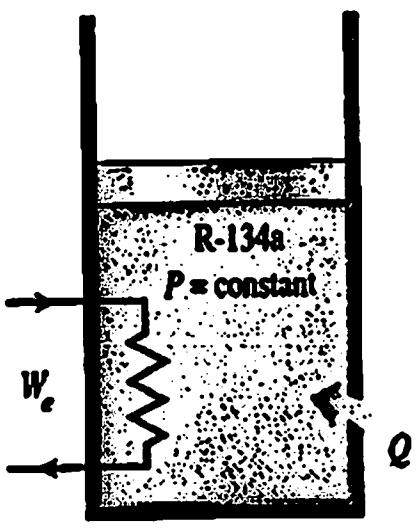


FIGURE P5–104

Step 1 of 5

Given

Mass of saturated vapor refrigerant-134a $m = 12\ \text{kg}$

Saturated vapor in the cylinder $P_1 = 240\ \text{kPa}$

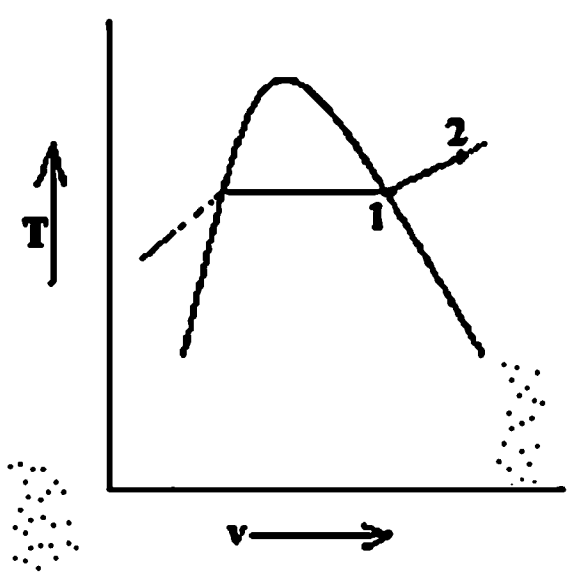
Amount of heat transferred to the refrigerant $Q_{in} = 300\ \text{kJ}$

Voltage supplied $V = 110\ \text{V}$

Voltage supplied for a period of $t = 6\ \text{min}$

Final temperature $T_2 = 70°\text{C}$

Step 2 of 5



Step 3 of 5

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$Q_{in} + W_{in} = \Delta H$

$Q_{in} + VI\Delta t = m(h_2 - h_1)$ (Since $W_{in} = VI\Delta t$)

Step 4 of 5

At pressure $P_1 = 240\ \text{kPa}$ from saturated R-134a tables

$h_1 = h_g = 247.28\ \text{kJ/kg}$

Since the process is constant pressure process

At $P_2 = P_1 = 240\ \text{kPa}$ and $T_2 = 70°\text{C}$ from superheated R-134a tables

$h_2 = 314.51\ \text{kJ/kg}$

Step 5 of 5

Substituting known values in the above equation

$Q_{in} + VI\Delta t = m(h_2 - h_1)$

$300 \times 1000 + 110I(6 \times 60) = 12(314.51 - 247.28) \times 1000$

$300000 + 39600I = 806760$

$39600I = 506760$

$I = 12.796\ \text{A}$

Therefore the current supplied is

$\boxed{I = 12.796\ \text{A}}$

Question:

5.105P

Saturated water vapor at 200°C is condensed to a saturated liquid at 50°C in a spring-loaded piston-cylinder device. Determine the heat transfer for this process, in kJ/kg.

Step 1 of 6

Given:

Initially temperature $T_1 = 200°C$

Final temperature $T_2 = 50°C$

Step 2 of 6

P-v Daigram:



Question:

Saturated water vapor at 200°C is condensed to a saturated liquid at 50°C in a spring-loaded piston-cylinder device. Determine the heat transfer for this process, in kJ/kg.

Step 1 of 6

Given:

Initially temperature $T_1 = 200°C$

Final temperature $T_2 = 50°C$

Step 2 of 6

P-v Daigram:



$v_2 = v_f = 0.001012 \frac{m^3}{kg}$

$u_2 = u_g = 209.33 \frac{kJ}{kg}$

From the ideal gas constants specific heat properties of various gases
For steam

$C_v = 1.4108 \frac{kJ}{kg \cdot K}$

Step 5 of 6

The energy balance for this stationary system can be expressed as

$Q_{in} + W_{b,out} = \Delta U$

Boundary work will be the area under the process 1-2 projected on volume axis.

$$W_{b,out} = \frac{1}{2}(P_1 + P_2)(v_2 - v_1)$$
$$= \frac{1}{2}(1554.9 + 12.352)(0.001012 - 0.12721)$$
$$= -98.892 \frac{kJ}{kg}$$

Step 6 of 6

Heat transferred is

$$Q_{in} = \Delta U - W_{b,out}$$
$$= C_v (T_2 - T_1) - (-98.892)$$
$$= 1.4108(50 - 200) + 98.892$$
$$= -112.728 \frac{kJ}{kg}$$

$$\boxed{Q_{in} = -112.728 \frac{kJ}{kg}}$$

Question: 5.106P

A piston-cylinder device contains helium gas initially at 150 kPa, 20°C, and 0.5 m3. The helium is now compressed in a polytropic process (PV^n = constant) to 400 kPa and 140°C. Determine the heat loss or gain during this process.

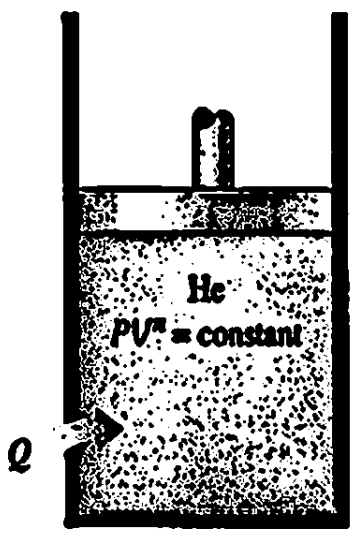


FIGURE P5–106

Step 1 of 8

Given
Piston cylinder contains Helium
Initial pressure $P_1 = 150\ \text{kPa}$
Initial temperature $T_1 = 20°\text{C} = 293\text{K}$
Initial volume $V_1 = 0.5\ \text{m}^3$
Final pressure $P_2 = 400\ \text{kPa}$
Final temperature $T_2 = 140°\text{C} = 413\text{K}$

Step 2 of 8

From table ideal gas specific heats of common gases
Properties of helium are
$R = 2.0769\ \text{kPa.m}^3/\text{kg.K}$
$c_v = 3.1156\ \text{kJ/kg.K}$

Step 3 of 8

The boundary work input for a polytropic process is expressed as

$$W_{b,in} = \int_1^2 P dV$$
$$= \frac{P_2V_2 - P_1V_1}{n-1}$$
$$W_{b,in} = \frac{mR(T_2 - T_1)}{n-1}$$

Step 4 of 8

The mass of helium is determined by

$$m = \frac{P_1V_1}{RT_1}$$
$$m = \frac{(150)(0.5)}{(2.0769)(293)}$$
$$m = 0.123\ \text{kg}$$

Step 5 of 8

The volume after compression is determined by using Ideal gas relation

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$\frac{(150)\times 0.5}{293} = \frac{(400)\times V_2}{413}$$
$$V_2 = 0.264\ \text{m}^3$$

Step 6 of 8

The power index n is determined by polytropic process

$$P_1V_1^n = P_2V_2^n$$
$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n$$
$$\frac{150}{400} = \left(\frac{0.264}{0.5}\right)^n$$
$$0.375 = 0.528^n$$

Taking logarithm on both sides

$$\ln 0.375 = n \times \ln 0.528$$
$$n = 1.536$$

Step 7 of 8

Substituting all the known values in above work relation
The boundary work is obtained as

$$W_{b,in} = \frac{mR(T_2 - T_1)}{n-1}$$
$$W_{b,in} = \frac{(0.123)(2.0769)(413-293)}{1.536-1}$$
$$W_{b,in} = 57.19\ \text{kJ}$$

Step 8 of 8

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$
Therefore

$$Q_{in} + W_{in} = \Delta U$$
$$Q_{in} + W_{in} = m(u_2 - u_1)$$
$$Q_{in} + W_{in} = mc_v(T_2 - T_1)$$
$$Q_{in} + 57.19 = 0.123(3.1156)(413-293)$$
$$Q_{in} + 57.19 = 45.986$$
$$Q_{in} = -11.204\ \text{kJ}$$

Therefore the heat transfer is

$$\boxed{Q_{in} = -11.2\ \text{kJ}}$$

The negative symbol indicates that heat loss from the system.

Question: 5.107P

Nitrogen gas is expanded in a polytropic process with $n = 1.45$ from 2MPa and 1200 K to 200 kPa in a piston-cylinder device. How much work is produced and heat is transferred during this expansion process, in kJ/kg?

Step 1 of 4

Obtain the properties of nitrogen from the ideal gas specific heat of various common gases

$$R = 0.2968\,\text{kJ/kg}\cdot\text{K}$$

$$C_v = 0.743\ \text{kJ/kg}\cdot\text{K}$$

Here, R is the universal gas constant of nitrogen and C_v is the specific heat at constant volume of nitrogen.

Step 2 of 4

Calculate the work done by using the boundary work relation for the polytropic process of an ideal gas.

$$w_{b,out} = \frac{RT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Here, $W_{b,out}$ is the work done, T_1 is the initial temperature, n is the polytropic index, P_2 is the final temperature and P_1 is the initial temperature.

Substitute $0.2968\ \text{kJ/kg}\cdot\text{K}$ for R, $1200\,\text{K}$ for T_1, 1.45 for n, $200\,\text{kPa}$ for P_2 and $2000\,\text{kPa}$ for P_1.

$$w_{b,out} = \frac{(0.2968)(1200)}{1-1.45}\left[\left(\frac{200}{2000}\right)^{\frac{1.45-1}{1.45}} - 1\right]$$
$$= 404.13\,\text{kJ/kg}$$

Therefore, the work produced is $\boxed{404.13\,\text{kJ/kg}}$.

Step 3 of 4

Calculate the temperature at the final state by using the equation,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Substitute $1200\,\text{K}$ for T_1, $200\,\text{kPa}$ for P_2 and $2000\,\text{kPa}$ for P_1.

$$T_2 = 1200\times\left(\frac{200}{2000}\right)^{\frac{1.45-1}{1.45}}$$
$$= 587.275\,\text{K}$$

Step 4 of 4

Calculate the heat transferred by using the energy balance for this system.

$$q_{in} = w_{b,out} + C_v\left(T_2 - T_1\right)$$

Substitute $404.13\,\text{kJ/kg}$ for $w_{b,out}$, $0.743\,\text{kJ/kg}\cdot\text{K}$ for C_v, $587.275\,\text{K}$ for T_2 and $1200\,\text{K}$ for T_1.

$$q_{in} = 404.13 + (0.743)(587.275 - 1200)$$
$$= -51.125\,\text{kJ/kg}$$

Therefore, the amount of heat transferred is $\boxed{51.125\,\text{kJ/kg}}$.

Question: 5.108P

A frictionless piston-cylinder device and a rigid tank initially contain 12 kg of an ideal gas each at the same temperature, pressure, and volume. It is desired to raise the temperatures of both systems by 15°C. Determine the amount of extra heat that must be supplied to the gas in the cylinder which is maintained at constant pressure to achieve this result. Assume the molar mass of the gas is 25.

Step 1 of 3

Given

Mass of ideal gas $m = 12$ kg

Change in temperature $\Delta t = 15°C$

Molar mass of the gas $M = 25$

Step 2 of 3

The gas constant for the ideal gas is determined by

$$R = \frac{R_u}{M}$$

$$= \frac{8.314}{25}$$

$$R = 0.33256 \text{ kJ/kg.K}$$

Step 3 of 3

The process in the piston cylinder is constant pressure process and the process in rigid tank is constant volume process

If there is no work interaction the heat transfer to the cylinder is constant enthalpy process and the heat transfer to the rigid tank is constant internal energy process

Therefore the extra amount of heat must be supplied to the gas in the cylinder is determined by

$$Q_{in} = \Delta H - \Delta U$$

$$Q_{in} = mc_p\Delta T - mc_v\Delta T$$

$$Q_{in} = m\Delta T\left(c_p - c_v\right) \qquad \text{(Since } R = c_p - c_v\text{)}$$

$$Q_{in} = mR\Delta T$$

$$Q_{in} = (12)(0.33256)(15)$$

$$\boxed{Q_{in} = 59.86 \text{ kJ}}$$

Question: 5.109P

A passive solar house that is losing heat to the outdoors at an average rate of 50,000 kJ/h is maintained at 22°C at all times during a winter night for 10 h. The house is to be heated by 50 glass containers each containing 20 L of water that is heated to 80°C during the day by absorbing solar energy. A thermostat-controlled 15-kW back-up electric resistance heater turns on whenever necessary to keep the house at 22°C.

(*a*) How long did the electric heating system run that night?

(*b*) How long would the electric heater run that night if the house incorporated no solar heating?

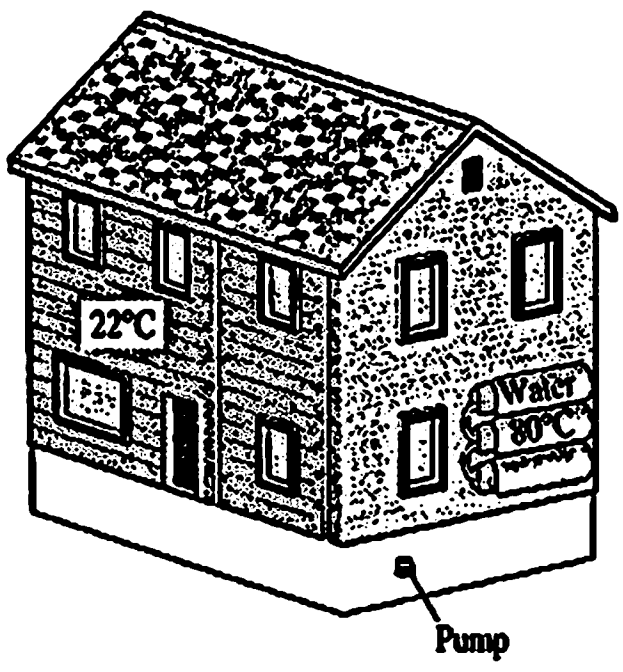


FIGURE P5–109

Step 1 of 4

Given

The amount of heat loss from the solar house $Q_{out} = 50,000\ \text{kJ/h}$

The temperature maintained in the house $T_{room} = 22°\text{C}$

Winter night duration $t = 10\ \text{h}$

Volume contained in glass container $V = 20\ \text{L} = 20 \times \frac{1}{1000}\ \text{m}^3$

Number of glass containers $n = 50$

Final temperature $T_2 = 80°\text{C}$

The capacity of heater is 15kW

Step 2 of 4

Density of water $\rho = 1000\ \text{kg/m}^3$

Specific heat of water $c = 4.18\ \text{kJ/kg.°C}$

(a) Time taken for the electric heating system run that night

The total mass of system

$$m_{water} = n\rho V$$

$$= 50 \times 1000 \times 20 \times \frac{1}{1000}$$

$$m_{water} = 1000\ \text{kg}$$

Step 3 of 4

Considering the house as the system

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$$W_{in} - Q_{out} = \Delta U$$

$$W_{in} - Q_{out} = m(u_2 - u_1)$$

$$W_{in} - Q_{out} = mc(T_2 - T_1)$$

$$\dot{W}_{in} \times t - \dot{Q}_{out} \times t = mc(T_2 - T_1)$$

$$15 \times \Delta t - 50000 \times 10 = -1000 \times 4.18 \times (80 - 22)$$

$$15 \times \Delta t - 500000 = -242440$$

$$15 \times \Delta t = 257560$$

$$\Delta t = 17170.66\ \text{s}$$

$$= \frac{17170.66}{60 \times 60}$$

$$= 4.77h$$

Therefore the electric heating system run that night is

$$\boxed{\Delta t = 4.77h}$$

Step 4 of 4

(b)

If there is no solar heating there will be no water supply then the energy equation is written as

$$W_{in} - Q_{out} = \Delta U$$

$$\dot{W}_{in} \times t - \dot{Q}_{out} \times t = 0$$

$$15 \times \Delta t - 50000 \times 10 = 0$$

$$\Delta t = 33333.33\ \text{s}$$

$$= \frac{33333.33}{60 \times 60}$$

$$= 9.26\ \text{h}$$

Therefore the electric heater run that night if the house incorporated no solar heating is

$$\boxed{\Delta t = 9.26\ \text{h}}$$

Question:

5.110P

An 1800-W electric resistance heating element is immersed in 40 kg of water initially at 20°C. Determine how long it will take for this heater to raise the water temperature to 80°C.

Step 1 of 3

Given

Mass of water $m = 40\ \text{kg}$

Initial temperature of water $T_1 = 20°\text{C}$

Final temperature of water $T_2 = 80°\text{C}$

Capacity of electric heater $\dot{W} = 1800\ \text{W} = 1.8\ \text{kW}$

Step 2 of 3

Density of water $\rho = 1000\ \text{kg/m}^3$

Specific heat of water $c = 4.18\ \text{kJ/kg.°C}$

Applying energy balance to the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$$W_{in} - Q_{out} = \Delta U$$

$$W_{in} - Q_{out} = m(u_2 - u_1)$$

$$W_{in} - Q_{out} = mc(T_2 - T_1)$$

Step 3 of 3

Assuming there is no heat loss during heating $Q_{out} = 0$

$$W_{in} - Q_{out} = mc(T_2 - T_1)$$

$$\dot{W}\Delta t - Q_{out} = mc(T_2 - T_1)$$

$$1.8\Delta t - 0 = 40(4.18)(80 - 20)$$

$$1.8\Delta t = 10032$$

$$\Delta t = 5573.33\ s$$

$$= \frac{5573.33}{60 \times 60}$$

$$= 1.548\ \text{h}$$

Therefore the time taken for the heater to raise the temperature is

$$\boxed{\Delta t = 1.548\ \text{h}}$$

Question: 5.111P

One ton (1000 kg) of liquid water at 50°C is brought into a well-insulated and well-sealed 4-m × 5-m × 6-m room initially at 15°C and 95 kPa. Assuming constant specific heats for both air and water at room temperature, determine the final equilibrium temperature in the room.

Step 1 of 5

Given:

Mass of liquid water $m = 1000\,\text{kg}$

Temperature of water $T_1 = 50^\circ\text{C}$

Dimension of the room $V = 4\text{m} \times 5\text{m} \times 6\text{m}$

Initially temperature of air $T_{air} = 15^\circ\text{C}$

Pressure of air $P_1 = 95\,\text{kPa}$

Step 2 of 5

From the molar mass gas constant and critical properties

The gas constant of air is

$$R = 0.287\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

The specific heat of water at room temperature is

From the properties of liquids and solids gases

For water

$$C = 4.18\frac{\text{kJ}}{\text{kg}\cdot^\circ\text{C}}$$

Step 3 of 5

The volume and the mass of the air in the room are

$$V = 4\times5\times6$$
$$= 120\text{m}^3$$

We know that

Ideal gas equation

$$P_1V_1 = mRT_1$$
$$\Rightarrow m_{air} = \frac{P_1V_1}{RT_1}$$
$$= \frac{(95)\times(120)}{(0.2870)\times(288)}$$
$$= 137.921\text{kg}$$

Step 4 of 5

The energy balance can be written as

$$E_{in} - E_{out} = \Delta E_{system}$$

Since well insulated and sealed so amount of heat transfer is zero.

$$0 = \Delta U = (\Delta U)_{water} + (\Delta U)_{air}$$
$$\left[mC(T_2 - T_1)\right]_{water} + \left[mC_v(T_2 - T_1)\right]_{air} = 0$$
$$\left[mC(T_f - T_1)\right]_{water} + \left[mC_v(T_f - T_1)\right]_{air} = 0$$

Where T_f is the final equilibrium temperature in the room.

Step 5 of 5

Substituting

$$(1000)(4.180)(T_f - 50) + (137.92)(0.718)(T_f - 15) = 0$$

Solving for T_f, gives

$$T_f = 49.19^\circ\text{C}$$

$$\boxed{T_f = 49.19^\circ\text{C}}$$

Question: 5.112P

A 4-m × 5-m × 6-m room is to be heated by one ton (1000 kg) of liquid water contained in a tank that is placed in the room. The room is losing heat to the outside at an average rate of 8000 kJ/h. The room is initially at 20°C and 100 kPa and is maintained at an average temperature of 20°C at all times. If the hot water is to meet the heating requirements of this room for a 24-h period, determine the minimum temperature of the water when it is first brought into the room. Assume constant specific heats for both air and water at room temperature.

Step 1 of 3

Given

Mass of water $m_w = 1000$ kg

Initial temperature of room $T_1 = 20°C$

The amount of heat lost from the room $\dot{Q}_{out} = 8000$ kJ/h

Initial temperature of air $T_{air} = 20^0$ C

Initial pressure of air $P_{air} = 100$ kPa

Volume of the room $V = 4 \times 5 \times 6 = 120$ m^3

Final equilibrium temperature of air and water $T_2 = 20^0$ C

Step 2 of 3

From properties table

Specific heat of water $c = 4.18$ kJ/kg.°C

The heat is lost for 24 hr from the room, so heat lost during that period is determined by

$$Q_{out} = \dot{Q}_{out} \times t$$
$$= 8000 \times 24$$
$$Q_{out} = 192,000 \text{ kJ}$$

Step 3 of 3

Applying energy balance to the room and water as the system

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

And there in work transfer the energy balance of the system is

$$-Q_{out} = \Delta U$$
$$-Q_{out} = \left[mc(T_2 - T_1)\right]_{water} + \left[mc(T_2 - T_1)\right]_{air}$$
$$-Q_{out} = \left[mc(T_2 - T_1)\right]_{water} + 0$$
$$-192000 = \left[1000 \times (4.18) \times (20 - T_1)\right]$$
$$-45.933 = 20 - T_1$$
$$T_1 = 65.933^0 C$$

Therefore the minimum (initial) temperature of the water when it is first brought into the room

$$\boxed{T_1 = 65.933°C}$$

Question: 5.113P

The energy content of a certain food is to be determined in a bomb calorimeter that contains 3 kg of water by burning a 2-g sample of it in the presence of 100 g of air in the reaction chamber. If the water temperature rises by 3.2°C when equilibrium is established, determine the energy content of the food, in kJ/kg, by neglecting the thermal energy stored in the reaction chamber and the energy supplied by the mixer. What is a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber?

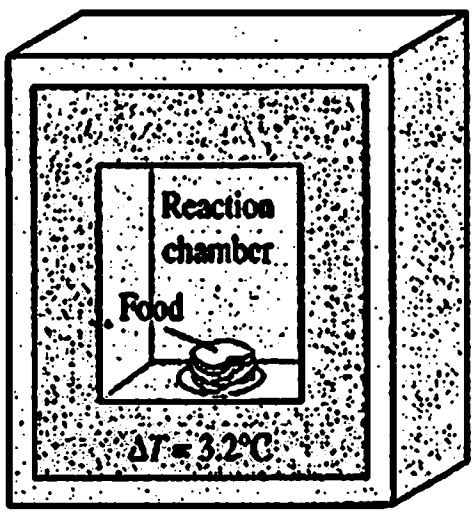


FIGURE P5–113

Step 1 of 4

Given

Mass of water $m_w = 3\,\text{kg}$

Temperature rise of water $\Delta T = 3.2°\text{C}$

Mass of sample food $m = 2\,\text{g}$

Mass of air $m_{air} = 100\,\text{g}$

Step 2 of 4

From properties table

Specific heat of water $c = 4.18\,\text{kJ/kg.°C}$

Specific heat of air $c = 0.718\,\text{kJ/kg.°C}$

Gas constant of air $R = 0.287\,\text{kJ/kg.K}$

Step 3 of 4

The chemical energy produced during the combustion of the sample is transferred to the water as heat. The energy content of the food is simply the heat transferred to the water.

Considering the water as our system,

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$Q_{in} = \Delta U$ As there in no work involved

$$Q_{in} = \left[mc(T_2 - T_1)\right]_{water}$$

Therefore

$$Q_{in} = \left[3(4.18)(3.2)\right]_{water}$$

$$Q_{in} = 40.13\,\text{kJ}$$

For 2-g sample the energy content of the food is

$$q_{in} = \frac{40.13}{2\,\text{g}}\left(\frac{1000\,\text{g}}{1\,\text{kg}}\right)$$

$$\boxed{q_{in} = 20065\,\text{kJ/kg}}$$

Step 4 of 4

We can estimate the error by considering air as the working medium where heat transfer occurred

$$Q_{in} = \left[mc(T_2 - T_1)\right]_{air}$$

$$Q_{in} = \left[\frac{100\,\text{g}}{1000\,\text{kg}} \times (0.718) \times (3.2)\right]_{air}$$

$$Q_{in} = 0.23\,\text{kJ}$$

Internal change of air $(Q_{in} = 0.23\,\text{kJ})$ is very much less than the internal energy change of water $(Q_{in} = 40.13\,\text{kJ})$ so we can neglect the internal energy change of air

Question:

5.114P

A 68-kg man whose average body temperature is 39°C drinks 1 L of cold water at 3°C in an effort to cool down. Taking the average specific heat of the human body to be 3.6 kJ/kg·°C, determine the drop in the average body temperature of this person under the influence of this cold water.

Step 1 of 4

Given

Mass of man $m = 68\ \text{kg}$

The average body temperature of the man $T_{man} = 39°\text{C}$

The temperature of the cold water $T_{cold} = 3°\text{C}$

The volume of the cold water $V = 1\ \text{L} = 1 \times \frac{1}{1000}\ \text{m}^3$

Average specific heat of the human body $c = 3.6\text{kJ/kg.}^0\text{C}$

Step 2 of 4

From properties of water

Specific heat of water $c = 4.18\ \text{kJ/kg.°C}$

Density of water $\rho = 1000\text{kg/m}^3$

The total mass of water

$$m_{water} = \rho V$$

$$= 1000 \times 1 \times \frac{1}{1000}$$

$$m_{water} = 1\ \text{kg}$$

Step 3 of 4

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

Consider body of the man as system, where there is no work and heat transfer involved in the system, only there is a change in the internal energy of the system due to transfer of temperature due to drinking cold water

Therefore

$$(\Delta U)_{water} + (\Delta U)_{man} = 0$$

$$\left[mc(T_2 - T_1)\right]_{water} + \left[mc(T_2 - T_1)\right]_{man} = 0$$

$$\left[1 \times (4.18) \times (T_f - 3)\right]_{water} + \left[68 \times (3.6) \times (T_f - 39)\right]_{man} = 0$$

$$4.18T_f - 12.54 + 244.8T_f - 9547.2 = 0$$

$$248.98T_f - 9559.74 = 0$$

$$T_f = 38.4^0\text{C}$$

The final equilibrium temperature is

$$T_f = 38.4°\text{C}$$

Step 4 of 4

The drop in the average body temperature is

$$\Delta T = T_{man} - T_f$$

$$\Delta T = 39 - 38.4$$

$$= 0.6°\text{C}$$

$$\boxed{\Delta T = 0.6°\text{C}}$$

Question: 5.115P

A 0.3-L glass of water at 20°C is to be cooled with ice to 5°C. Determine how much ice needs to be added to the water, in grams, if the ice is at (*a*) 0°C and (*b*) -20°C. Also determine how much water would be needed if the cooling is to be done with cold water at 0°C. The melting temperature and the heat of fusion of ice at atmospheric pressure are 0°C and 333.7 kJ/kg, respectively, and the density of water is 1 kg/L.

Step 1 of 8

Loading solution. Please wait ...

Step 2 of 8

The specific heat of water at room temperature
From the properties of liquids solids and foods

$$c = 4.18 \frac{kJ}{kg \cdot K}$$

The specific heat of ice at about 0°C is

$$c_{ice} = 2.11 \frac{kJ}{kg \cdot K}$$

Step 3 of 8

(a) The mass of the water is

$$m_w = \rho V$$
$$= (1)(0.3)$$
$$= 0.3kg$$

Step 4 of 8

We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as
Net energy transferby heat, work, and mass

=Change in internal, kinetic, potential, etc. energies

$$E_{in} - E_{out} = \Delta E_{system}$$

Since steady state system.

$$0 = \Delta U$$
$$0 = \Delta U_{ice} + \Delta U_{water}$$
$$\left[mc(0°C - T_1)_{solid} + mh_{if} + mc(T_2 - 0°C)_{liquid}\right]_{ice} + \left[mc(T_2 - T_1)\right]_{water} = 0$$

Step 5 of 8

Noting that $T_{1,ice} = 0°C$ and $T_2 = 5°C$

Substituting gives

$$m[0 + 333.7 + (4.18)(5-0)] + (0.3)(4.18)(5-20) = 0$$
$$m = 0.053kg$$
$$\boxed{m = 0.053kg}$$

Step 6 of 8

(b) When $T_{1,ice} = -20°C$ instead of 0°C,

$$\left[mc(0°C - T_1)_{solid} + mh_{if} + mc(T_2 - 0°C)_{liquid}\right]_{ice} + \left[mc(T_2 - T_1)\right]_{water} = 0$$

Substituting gives

$$m[(2.11)[0-(-20)] + 333.7 + (4.18)(5-0)] + (0.3)(4.18)(5-20) = 0$$
$$m = 0.0474kg$$
$$\boxed{m = 0.0474kg}$$

Step 7 of 8

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C

$$(\Delta U)_{cold\ water} + (\Delta U)_{water} = 0$$
$$\left[mc(T_2 - T_1)\right]_{cold\ water} + \left[mc(T_2 - T_1)\right]_{water} = 0$$

Step 8 of 8

Substituting,

$$\left[m_{cold\ water}(4.18)(5-0)\right] + (0.3)(4.18)(5-20) = 0$$

It gives

$$m_{Cold\ water} = 0.9kg$$
$$= 900gm$$
$$\boxed{m_{Cold\ water} = 900gm}$$

Question: 5.116P

Reconsider Prob. 5–115. Using EES (or other) software, investigate the effect of the initial temperature of the ice on the final mass required. Let the ice temperature vary from −26 to 0°C. Plot the mass of ice against the initial temperature of ice, and discuss the results.

Step 1 of 8

Given:

The density of water is $\rho_w = 1\ \frac{kg}{L}$

The melting temperature $T = 0°C$

The heat of fusion of ice at $L = 333.7\ \frac{kJ}{kg}$

Capacity of glass water $m = 0.3L$

Temperature of water $T_w = 20°C$

Temperature of the ice varies from $-26°C$ to $0°C$

Step 2 of 8

The problem is solved using EES, and the solution is given below.

```
"Knowns"
rho_water = 1 [kg/L]
V = 0.3 [L]
T_1_ice = 0 [C]
T_1 = 20 [C]
T_2 = 5 [C]
C_ice = 2.11 [kJ/kg-C]
C_water = 4.18 [kJ/kg-C]
h_if = 333.7 [kJ/kg]
T_1_ColdWater = 0 [C]
```

Step 3 of 8

```
"The mass of the water is:"
m_water = rho_water*V "[kg]"
"The system is the water plus the ice. Assume a short time period and neglect any heat
and mass transfer. The energy balance becomes:"
E_in - E_out = DELTAE_sys "[kJ]"
E_in = 0 "[kJ]"
E_out = 0"[kJ]"
```

Step 4 of 8

```
DELTAE_sys = DELTAU_water+DELTAU_ice"[kJ]"
DELTAU_water = m_water*C_water*(T_2 - T_1)"[kJ]"
DELTAU_ice = DELTAU_solid_ice+DELTAU_melted_ice"[kJ]"
DELTAU_solid_ice =m_ice*C_ice*(0-T_1_ice) + m_ice*h_if"[kJ]"
DELTAU_melted_ice=m_ice*C_water*(T_2 - 0)"[kJ]"
m_ice_grams=m_ice*convert(kg,g)"[g]"
```

Step 5 of 8

```
"Cooling with Cold Water:"
DELTAE_sys = DELTAU_water+DELTAU_ColdWater"[kJ]"
DELTAU_water = m_water*C_water*(T_2_ColdWater - T_1)"[kJ]"
DELTAU_ColdWater = m_ColdWater*C_water*(T_2_ColdWater -
T_1_ColdWater)"[kJ]"
m_ColdWater_grams=m_ColdWater*convert(kg,g)"[g]"
```

Step 6 of 8

select the table in menu bar then select the parameter table .
then select the T_1ice and m_ice grams then cleck ok.
Then run the table from the $-26°C$ to $0°C$

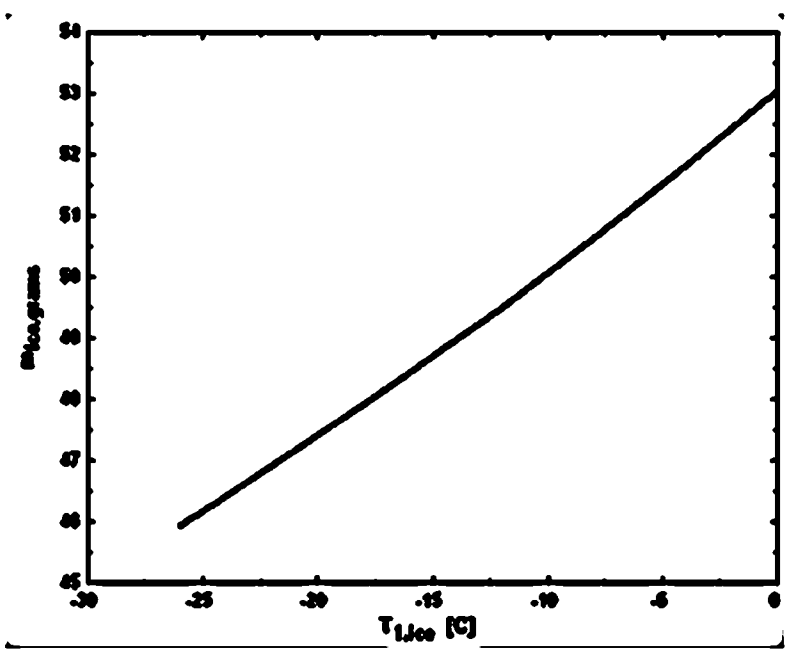
Step 7 of 8

Solution table:

D 1_14	$T_{1,ice}$ [C]	$m_{ice,grams}$
Run 1	-26	45.94
Run 2	-24	46.42
Run 3	-22	46.91
Run 4	-20	47.4
Run 5	-18	47.91
Run 6	-16	48.43
Run 7	-14	48.97
Run 8	-12	49.51
Run 9	-10	50.07
Run 10	-8	50.64
Run 11	-6	51.22
Run 12	-4	51.81
Run 13	-2	52.42
Run 14	0	53.05

Step 8 of 8

Graph between the T_1 ice and m_ice grams



Question: 5.117P

A rigid tank containing 0.4 m3 of air at 400 kPa and 30°C is connected by a valve to a piston-cylinder device with zero clearance. The mass of the piston is such that a pressure of 200 kPa is required to raise the piston. The valve is now opened slightly, and air is allowed to flow into the cylinder until the pressure in the tank drops to 200 kPa. During this process, heat is exchanged with the surroundings such that the entire air remains at 30°C at all times. Determine the heat transfer for this process.

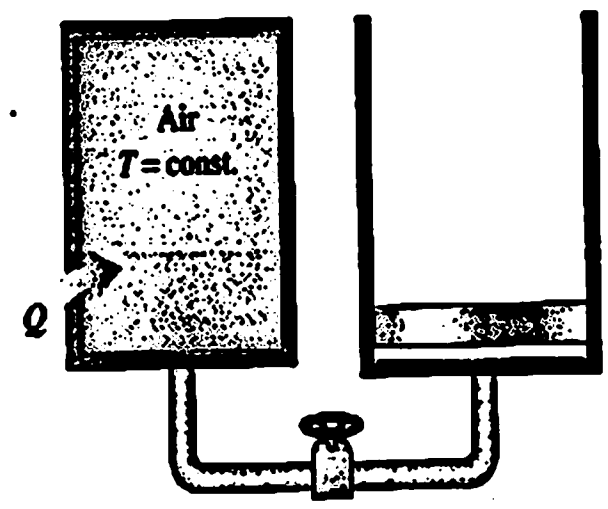


FIGURE P5–117

Step 1 of 2

Obtain the *R* value of air from the table A-1, "Molar mass, gas constant, and critical-point properties" in the textbook.

$$R = 0.287 \text{ kJ/kg.K}$$

Consider the tank and cylinder as a system.

Apply the energy balance equation of the system.

$$E_{in} - E_{out} = \Delta E_{system}$$

Neglect the kinetic and potential energy changes.

$$Q_{in} - W_{out} = \Delta U$$

Since, the temperature remains constant therefore, the internal energy is zero.

$$Q_{in} - W_{out} = \Delta U = 0$$
$$Q_{in} = W_{out}$$

Step 2 of 2

Calculate the volume by using Ideal gas relation.

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

Here, P_1 is the initial pressure, V_1 is the initial volume, T_1 is the initial temperature, P_2 is the final pressure, V_2 is the final volume, and T_2 is the final temperature.

Substitute 400 kPa for P_1, 0.4 m^3 for V_1, $(30+273)$ K for T_1, $(30+273)$ K for T_2, and 200 kPa for P_2.

$$\frac{400 \times 0.4}{(30+273)} = \frac{200 \times V_2}{(30+273)}$$
$$V_2 = 0.8 \text{ m}^3$$

Since, the pressure remains constant the boundary work for this process is expressed as

$$W_{b,out} = \int_1^2 P dV$$
$$= P_2 (V_2 - V_1)$$

Substitute 200 kPa for P_2, 0.4 m^3 for V_1, and 0.8 m^3 for V_2.

$$W_{b,out} = 200(0.8 - 0.4)$$
$$W_{b,out} = 80 \text{ kJ}$$

From energy balance of the system the heat transfer for this process is

$$Q_{in} = W_{out}$$
$$= 80 \text{ kJ}$$

Therefore, the heat transfer for this process is **80 kJ**.

Question: 5.118P

A well-insulated 4-m × 4-m × 5-m room initially at 10°C is heated by the radiator of a steam heating system. The radiator has a volume of 15 L and is filled with superheated vapor at 200 kPa and 200°C. At this moment both the inlet and the exit valves to the radiator are closed. A 120-W fan is used to distribute the air in the room. The pressure of the steam is observed to drop to 100 kPa after 30 min as a result of heat transfer to the room. Assuming constant specific heats for air at room temperature, determine the average temperature of air in 30 min. Assume the air pressure in the room remains constant at 100 kPa.

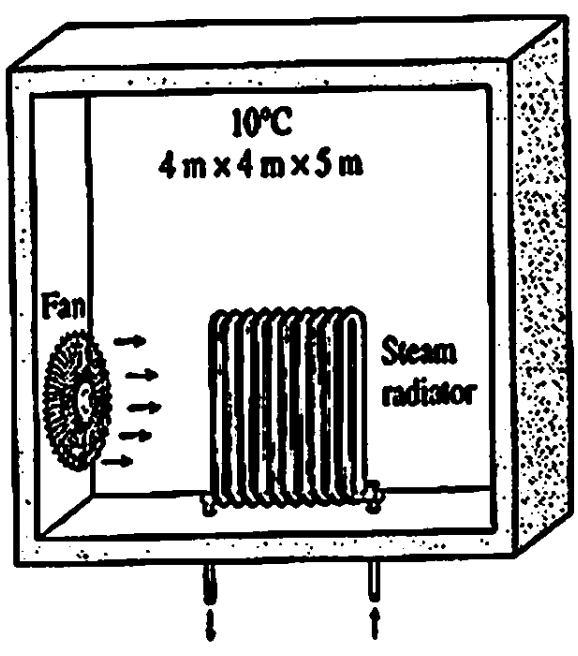


FIGURE P5–118

Step 1 of 9

Given

Volume of the room $V_{room} = 4\times4\times5 = 80\ \text{m}^3$

Initial temperature of the room $T_{room} = 10°\text{C}$

The volume of the radiator $V_{rad} = 15L$

The pressure of the super heated vapor in radiator $P_1 = 200\ \text{kPa}$

The temperature of the super heated vapor in radiator $T_1 = 200°\text{C}$

The final pressure of the vapor $P_2 = 200\ \text{kPa}$

The power of the fan used $\dot{W}_{fan} = 120\ \text{W}$

The time for the pressure to fall down $t = 30\ \text{min} = 30\times60\ \text{sec}$

Step 2 of 9

The gas constant of air is

$R = 0.287\ \text{kJ/kg.K}$

Specific heat of air

$c_v = 1.005\ \text{kJ/kg}$

Considering radiator alone as the system and both the inlet and exit valves to the radiator are closed

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$-Q_{out} = \Delta U = 0$

$-Q_{out} = m(u_2 - u_1)$

Step 3 of 9

At $P_1 = 200\ \text{kPa}$ and $T_1 = 200°\text{C}$ from superheated water tables

Specific volume $v_1 = 1.08049\ \text{m}^3/\text{kg}$

Internal energy $u_1 = 2654.6\ \text{kJ/kg}$

At $P_2 = 200\ \text{kPa}$ from saturated water tables

$v_f = 0.001043\ \text{m}^3/\text{kg}$

$v_g = 1.6941\ \text{m}^3/\text{kg}$

$u_f = 417.4\ \text{kJ/kg}$

$u_{fg} = 2088.2\ \text{kJ/kg}$

Step 4 of 9

Since $v_1 = v_2$

$v_1 = v_f + x_2(v_g - v_f)$

$1.08049 = 0.001043 + x_2(1.6941 - 0.001043)$

$x_2 = 0.6376$

Specific volume at state 2

$u_2 = u_f + xu_{fg}$

$u_2 = 417.4 + (0.6376)\times(2088.2)$

$u_2 = 1748.83\ \text{kJ/kg}$

Step 5 of 9

Mass of the steam is

$$m = \frac{V_{rad}}{v_1}$$

$$= \frac{0.015}{1.08049}$$

$m = 0.014\ \text{kg}$

Step 6 of 9

Substituting the energy relation

$-Q_{out} = m(u_2 - u_1)$

$-Q_{out} = (0.014)(1748.83 - 2654.6)$

$= -12.68\ \text{kJ}$

$Q_{out} = 12.68\ \text{kJ}$

Step 7 of 9

The mass of air in the room is determined by ideal gas relation

$$m_{air} = \frac{P_1V_1}{RT_1}$$

$$m_{air} = \frac{(100)(80)}{(0.287)(283)}$$

$m_{air} = 98.5\ \text{kg}$

Step 8 of 9

Consider room air as the system, the energy balance for a closed system is expressed as

$Q_{in} + W_{fan} - W_{out} = \Delta U$

$Q_{in} + W_{fan} = \Delta U + W_{out}$

Since the air pressure in the room remains constant

$Q_{in} + W_{fan} = mc_v(T_2 - T_1) + P(V_2 - V_1)$

$Q_{in} + W_{fan} = mc_v(T_2 - T_1) + mR(T_2 - T_1)$

$Q_{in} + W_{fan} = (T_2 - T_1)(c_v + R)$

$Q_{in} + W_{fan} = m(T_2 - T_1)c_p$

$Q_{in} + W_{fan} = mc_p(T_2 - T_1)$

Step 9 of 9

Here the heat rejected by the radiator will be gained by the air in the room

$Q_{in} = Q_{out} = 12.68\ \text{kJ}$

Then by using energy balance equation

$Q_{in} + W_{fan} = mc_p(T_2 - T_1)$

$Q_{in} + \dot{W}_{fan} \times t = mc_p(T_2 - T_1)$

$12.68 + 0.12\times30\times60 = 98.5\times(1.005)\times(T_2 - 10)$

$228.68 = 98.9925\times(T_2 - 10)$

$2.31007 = (T_2 - 10)$

$T_2 = 12.31007^0\text{C}$

The final air temperature of air in the room in 30 minutes is

$\boxed{T_2 = 12.31°\text{C}}$

Question:

5.119P

Consider a well-insulated horizontal rigid cylinder that is divided into two compartments by a piston that is free to move but does not allow either gas to leak into the other side. Initially, one side of the piston contains 1 m3 of N2 gas at 500 kPa and 120°C while the other side contains 1 m3 of He gas at 500 kPa and 40°C Now thermal equilibrium is established in the cylinder as a result of heat transfer through the piston. Using constant specific heats at room temperature, determine the final equilibrium temperature in the cylinder. What would your answer be if the piston were not free to move?

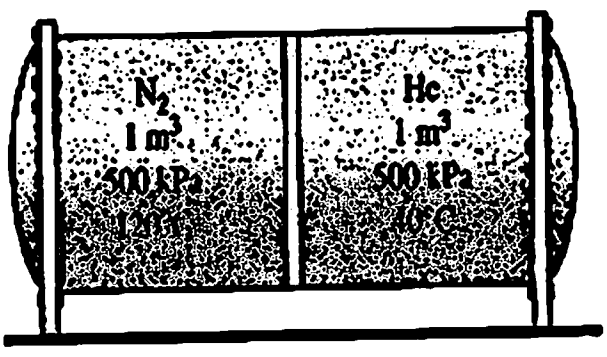


FIGURE P5–119

Step 1 of 6

Given:
Compartment Nitrogen:
Volume $V_1 = 1\text{m}^3$
Pressure $P_1 = 500\text{kPa}$
Temperature $T = 120^\circ\text{C}$
Compartment of helium:
Volume $V_{1H} = 1\text{m}^3$
Pressure $P_{1H} = 500\text{kPa}$
Temperature $T_{1H} = 40^\circ\text{C}$

Step 2 of 6

From the ideal gas specific heat of various common gases
For Nitrogen
The gas constants and the constant volume specific heats are

$$R = 0.2968\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$c_v = 0.743\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

From the ideal gas specific heat of various common gases
For helium

$$R = 2.0769\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$c_v = 3.1156\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 3 of 6

We know that
$PV = mRT$
The mass of the nitrogen is

$$m_{N_2} = \left(\frac{P_1V_1}{RT_1}\right)_{N_2}$$
$$= \frac{(500)(1)}{(0.2968)(393)}$$
$$= 4.29\text{kg}$$

Step 4 of 6

Mass of the helium

$$m_{He} = \left(\frac{P_{1H}V_{1H}}{RT_{1H}}\right)_{He}$$
$$= \frac{(500)(1)}{(2.0769)(313)}$$
$$= 0.769\text{kg}$$

Step 5 of 6

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = \left[mc_v(T_2 - T_1)\right]_{N_2} + \left[mc_v(T_2 - T_1)\right]_{He}$$

Since $T_2 = T_f$

$$0 = \left[mc_v(T_f - T_1)\right]_{N_2} + \left[mc_v(T_f - T_1)\right]_{He}$$

Step 6 of 6

Substituting,

$$(4.29)(0.743)(T_f - 120) + (0.769)(3.1156)(T_f - 40) = 0$$

It gives

$$T_f = 85.67^\circ\text{C}$$

$$\boxed{T_f = 85.67^\circ\text{C}}$$

Where T_f is the final equilibrium temperature in the cylinder.

The answer would be the same if the piston were not free to move since it would effect only pressure and not the specific heats.

Question: 5.120P

Repeat Prob. 5–119 by assuming the piston is made of 8 kg of copper initially at the average temperature of the two gases on both sides.

Step 1 of 6

Given:
Compartment Nitrogen:
Volume $V_1 = 1\text{m}^3$
Pressure $P_1 = 500\text{kPa}$
Temperature $T = 120^\circ\text{C}$
Compartment of helium:
Volume $V_{1H} = 1\text{m}^3$
Pressure $P_{1H} = 500\text{kPa}$
Temperature $T_{1H} = 40^\circ\text{C}$
Average temperature of cooper $T_{1,Cu} = 80^\circ\text{C}$

Step 2 of 6

From the ideal gas specific heat of various common gases
For Nitrogen
The gas constants and the constant volume specific heats are

$$R = 0.2968\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$c_v = 0.743\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

From the ideal gas specific heat of various common gases
For helium

$$R = 2.0769\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$c_v = 3.1156\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

From the properties of common liquids, solids and foods
At room temperature
The specific heat of copper piston

$$c = 0.386\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 3 of 6

We know that
$PV = mRT$
The mass of the nitrogen is
The mass of the nitrogen is

$$m_{N_2} = \left(\frac{P_1V_1}{RT_1}\right)_{N_2}$$
$$= \frac{(500)(1)}{(0.2968)(393)}$$
$$= 4.29\text{kg}$$

Step 4 of 6

Mass of the helium

$$m_{He} = \left(\frac{P_{1H}V_{1H}}{RT_{1H}}\right)_{He}$$
$$= \frac{(500)(1)}{(2.0769)(313)}$$
$$= 0.769\text{kg}$$

Step 5 of 6

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = \left[mc_v(T_2 - T_1)\right]_{N_2} + \left[mc_v(T_2 - T_1)\right]_{He} + \left[mc(T_2 - T_1)\right]_{Cu}$$

$$0 = \left[mc_v(T_f - T_1)\right]_{N_2} + \left[mc_v(T_f - T_1)\right]_{He} + \left[mc(T_f - T_1)\right]_{Cu}$$

Where T_f is the final equilibrium temperature in the cylinder.

Step 6 of 6

Where
Average temperature of the cooper is

$$T_{1,Cu} = \frac{120+40}{2}$$
$$= 80^\circ\text{C}$$

Substituting,

$$(4.29)(0.743)(T_f - 120) + (0.769)(3.1156)(T_f - 40) + (8)(0.386)(T_f - 80) = 0$$

It gives

$$T_f = 83.7^\circ\text{C}$$

$$\boxed{T_f = 83.7^\circ\text{C}}$$

Question: 5.121P

Reconsider Prob. 5–120. Using EES (or other) software, investigate the effect of the mass of the copper piston on the final equilibrium temperature. Let the mass of piston vary from 1 to 10 kg. Plot the final temperature against the mass of piston, and discuss the results.

Step 1 of 8

Given:
Working fluid: helium
Inlet pressure of the compressor $P_1 = 100\,\text{kPa}$
Inlet temperature of the compressor $T_1 = 17^\circ\text{C}$
Final pressure of the compressor $P_2 = 400\,\text{kPa}$
Inlet temperature of the turbine $T_3 = 20^\circ\text{C}$
Mass flow rate $\dot{m} = 0.2\frac{\text{kg}}{\text{s}}$
Mass of the piston varies from 1 to 10 kg.

Step 2 of 8

The problem is solved using EES, and the solution is given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=1 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=120 [C]
P_N2[1]=500 [kPa]
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=40 [C]
P_He[1]=500 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
"m_Pist=8 [kg]"
Cv_Pist=0.386 [kJ/kg-K]  "Use Cp for Copper from Table A-3(b) at 27C"
```

Step 3 of 8

```
"Solution:"

"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

"The entire cylinder is considered to be a closed system, neglecting the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_negPist,  we neglect DELTA KE and DELTA PE for the
cylinder."
E_in - E_out = DELTAE_neglPist
E_in =0 [kJ]
E_out = 0 [kJ]
```

Step 4 of 8

```
"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_neglPist= m_N2*Cv_N2*(T_2_neglPist-
T_N2[1])+m_He*Cv_He*(T_2_neglPist-
T_He[1])
"The entire cylinder is considered to be a closed system, including the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_withPist,  we neglect DELTA KE and DELTA PE for the
cylinder."
E_in - E_out = DELTAE_withPist
```

Step 5 of 8

```
"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_withPist= m_N2*Cv_N2*(T_2_withPist-
T_N2[1])+m_He*Cv_He*(T_2_withPist-
T_He[1])+m_Pist*Cv_Pist*(T_2_withPist-T_Pist[1])
T_Pist[1]=(T_N2[1]+T_He[1])/2
"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
"Final pressure at equilibrium:"
"Neglecting effect of piston, P_2 is:"
P_2_neglPist*V_total=N_total*R_u*(T_2_neglPist+273)

"Including effect of piston, P_2 is:"
N_total=m_N2/molarmass(nitrogen)+m_He/molarmass(Helium)
P_2_withPist*V_total=N_total*R_u*(T_2_withPist+273)
```

Step 6 of 8

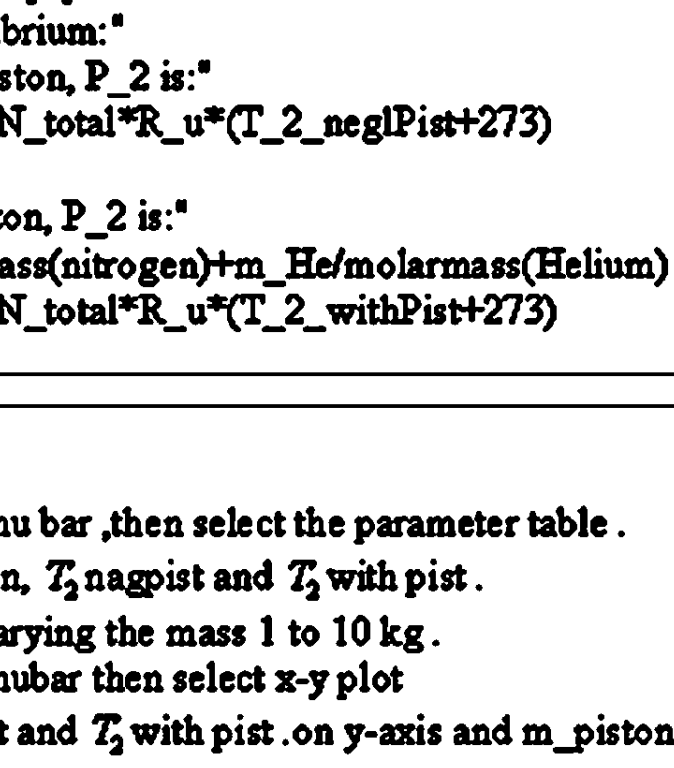
Now click tables in menu bar ,then select the parameter table .
Then select the m_piston, T_2 nagpist and T_2 with pist .
Then run the table by varying the mass 1 to 10 kg .
Then plot From the menubar then select x-y plot
Then click on T_2 nagpist and T_2 with pist .on y-axis and m_piston x-axis.

Step 7 of 8

▷ 1..10	m_{Pist} [kg]	$T_{2,negPist}$	$T_{2,withPist}$
Run 1	1	85.65	85.29
Run 2	2	85.65	84.96
Run 3	3	85.65	84.68
Run 4	4	85.65	84.43
Run 5	5	85.65	84.2
Run 6	6	85.65	83.99
Run 7	7	85.65	83.81
Run 8	8	85.65	83.64
Run 9	9	85.65	83.48
Run 10	10	85.65	83.34

Step 8 of 8

Graph between the T_2 nagpist and T_2 with pist and m_ piston



Question: 5.122P

An insulated piston-cylinder device initially contains 1.4-kg saturated liquid water at 200°C. Now an electric resistor placed in the tank is turned on for 20 min until the volume quadruples. Determine (*a*) the volume of the tank, (*b*) the final temperature, and (*c*) the electrical power rating of the resistor.

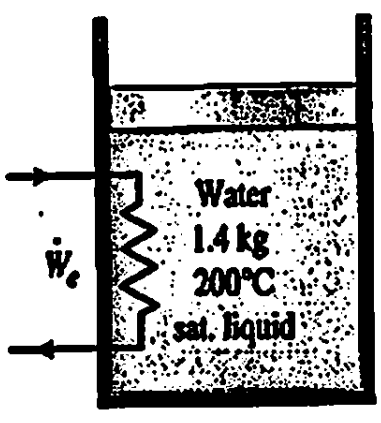


FIGURE P5–122

Step 1 of 12

Given

Mass of the saturated liquid water $m = 1.4$ kg

Temperature of water $T_1 = 200°C$

Final volume $V_2 = 4V_1$

Electric resistor placed in the tank is turned on for 20 minutes

Step 2 of 12

(a) Final volume of the tank

At $T_1 = 200°C$ from saturated water tables

$P_1 = P_{sat} = 1554.9$ kPa

$v_1 = v_f = 0.001157\ m^3/kg$

$h_1 = h_f = 852.26$ kJ/kg

Step 3 of 12

The initial volume of the tank is

$$V_1 = mv_1$$
$$= 1.4 \times (0.001157)$$
$$V_1 = 0.00162\ m^3$$

Step 4 of 12

Final volume of the tank is

$$V_2 = 4V_1$$
$$= 4(0.00162)$$
$$\boxed{V_2 = 0.00648\ m^3}$$

Step 5 of 12

(b) The final temperature

The saturated mixture in the final state has same pressure and temperature.

Therefore final temperature is

$$\boxed{T_2 = T_1 = 200°C}$$

Step 6 of 12

(c) The electrical power rating of the resistor

Considering the piston cylinder as a system, and doesn't involve any heat transfer.

From energy balance of the system

$$E_{in} - E_{out} = \Delta E_{system}$$
$$W_{in} - W_{out} = \Delta U$$
$$W_{in} = \Delta U + W_{out}$$
$$W_{in} = \Delta H$$
$$W_{in} = m(h_2 - h_1)\ \text{kJ}$$

Step 7 of 12

At $T_2 = 200°C$ from saturated water tables

$v_f = 0.001157\ m^3/kg$

$v_g = 0.12721\ m^3/kg$

$h_f = 852.26$ kJ/kg

$h_{fg} = 1939.8$ kJ/kg

Step 8 of 12

Specific volume at state 2

$$v_2 = \frac{V_2}{m}$$
$$= \frac{0.00648}{1.4}$$
$$v_2 = 0.00463\ m^3/kg$$

Step 9 of 12

We know that

$$v_2 = v_f + x_2 v_{fg}$$
$$W_{in} = m(h_2 - h_1)\ \text{kJ}$$

Step 7 of 12

At $T_2 = 200°C$ from saturated water tables

$v_f = 0.001157\ m^3/kg$

$v_g = 0.12721\ m^3/kg$

$h_f = 852.26$ kJ/kg

$h_{fg} = 1939.8$ kJ/kg

Step 8 of 12

Specific volume at state 2

$$v_2 = \frac{V_2}{m}$$
$$= \frac{0.00648}{1.4}$$
$$v_2 = 0.00463\ m^3/kg$$

Step 9 of 12

We know that

$$v_2 = v_f + x_2 v_{fg}$$
$$\dot{W}_{in} = \frac{m(h_2 - h_1)\ \text{kJ}}{20 \times 60}$$
$$= 0.0623\ \text{kW}$$
$$\boxed{\dot{W}_{in} = 0.0623\ \text{kW}}$$

Question: 5.123P

A vertical 12-cm diameter piston-cylinder device contains an ideal gas at the ambient conditons of 1 bar and 24°C. Initially, the inner face of the piston is 20 cm from the base of the cylinder. Now an external shaft connected to the piston exerts a force corresponding to a boundary work input of 0.1 kJ. The temperature of the gas remains constant during the process. Determine (*a*) the amount of heat transfer, (*b*) the final pressure in the cylinder, and (*c*) the distance that the piston is displaced.

Step 1 of 7

Given

The diameter of the piston $d = 12$ cm

The pressure inside the cylinder $P_1 = 1$ bar

Initial temperature in the cylinder $T_1 = 24°C$

The boundary input to the system $W_{in} = 0.1$ kJ

At initial condition the inner face of the piston from base of the cylinder $L_1 = 20$ cm

Step 2 of 7

(a) The amount of heat transfer

Considering piston cylinder arrangement as closed system

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$W_{in} - Q_{out} = \Delta U$

Since the temperature remains constant $T_1 = T_2$

Change in internal energy is zero $\Delta U = 0$

$W_{in} - Q_{out} = \Delta U = 0$

$Q_{out} = W_{in}$

$Q_{out} = 0.1$kJ

The amount of heat transfer

$\boxed{Q_{out} = 0.1\,\text{kJ}}$

Step 3 of 7

(b) The final pressure in the cylinder can be calculated by the relation, for a isothermal process

$P_1V_1 = P_2V_2$

Where initial volume

$$V_1 = \frac{\pi D^2}{4} L_1$$

Step 2 of 7

(a) The amount of heat transfer

Considering piston cylinder arrangement as closed system

From energy balance of the system

$E_{in} - E_{out} = \Delta E_{system}$

The kinetic and potential energy changes are negligible $\Delta PE = \Delta KE = 0$

$W_{in} - Q_{out} = \Delta U$

Since the temperature remains constant $T_1 = T_2$

Change in internal energy is zero $\Delta U = 0$

$W_{in} - Q_{out} = \Delta U = 0$

$Q_{out} = W_{in}$

$Q_{out} = 0.1$kJ

The amount of heat transfer

$\boxed{Q_{out} = 0.1\,\text{kJ}}$

Step 3 of 7

(b) The final pressure in the cylinder can be calculated by the relation, for a isothermal process

$P_1V_1 = P_2V_2$

Where initial volume

$$V_1 = \frac{\pi D^2}{4} L_1$$

Step 4 of 7

(c) The distance moved by the piston

We know the final volume is

$$V_2 = \frac{\pi D^2}{4} L_2$$

$$0.00145 = \frac{\pi (0.12)^2}{4} L_2$$

$$L_2 = 0.1282 \text{ m}$$

Step 7 of 7

The distance that the piston is displaced

$\Delta L = L_1 - L_2$

$= 0.2 - 0.1282$

$= 0.0718$ m

$\boxed{\Delta L = 7.18 \text{ cm}}$

Question: 5.124P pg 1

A piston-cylinder device initially contains 0.35-kg steam at 3.5 MPa, superheated by 7.4°C. Now the steam loses heat to the surroundings and the piston moves down, hitting a set of stops at which point the cylinder contains saturated liquid water. The cooling continues until the cylinder contains water at 200°C. Determine (*a*) the final pressure and the quality (if mixture), (*b*) the boundary work, (*c*) the amount of heat transfer when the piston first hits the stops, (*d*) and the total heat transfer.

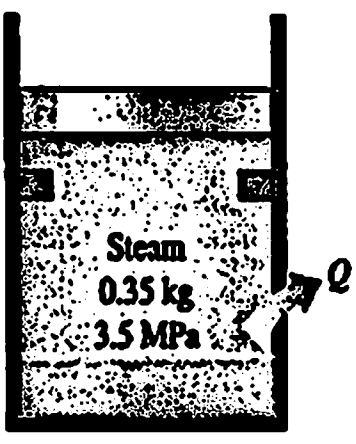


FIGURE P5–124

Step 1 of 10

(a)

Write the saturated temperature at initial state from Table A-5: Saturated water-Pressure table corresponding to pressure 3.5 MPa.

$$T_{sat@3.5\,MPa} = 242.56°C$$

Calculate the superheated temperature at initial state.

$$T_1 = T_{sat} + \Delta T$$

Here, saturated temperature at initial state is T_{sat} and superheated temperature is ΔT.

Substitute 242.56°C for T_{sat} and 7.4°C for ΔT.

$$T_1 = 242.56 + 7.4$$
$$= 249.96°C$$

Step 2 of 10

Write the specific volume of superheated water from Table A-6 corresponding to pressure of 3.5 MPa and 249.96°C by using interpolation.

$$\frac{v_1 - v_{@242.56°C}}{v_{@250°C} - v_{@242.56°C}} = \frac{T_1 - T_{@242.56°C}}{T_{@250°C} - T_{@242.56°C}}$$

Substitute 0.05706 m^3/kg for $v_{@242.56°C}$, 0.05876 m^3/kg for $v_{@250°C}$ and 249.96°C for T_1.

$$\frac{v_1 - 0.05706}{0.05876 - 0.05706} = \frac{249.96 - 242.56}{250 - 242.56}$$
$$v_1 - 0.05706 = \frac{7.4 \times 0.0017}{7.44}$$
$$v_1 = 0.00169 + 0.05706$$
$$v_1 = 0.05875\ m^3/kg$$

Step 3 of 10

Write the internal energy of superheated water from Table A-6 corresponding to pressure of 3.5 MPa and 249.96°C by using interpolation.

$$\frac{u_1 - u_{@242.56°C}}{u_{@250°C} - u_{@242.56°C}} = \frac{T_1 - T_{@242.56°C}}{T_{@250°C} - T_{@242.56°C}}$$

Substitute 0.05706 m^3/kg for $u_{@242.56°C}$, 0.05876 m^3/kg for $u_{@250°C}$ and 249.96°C for T_1.

$$\frac{u_1 - 2603.0}{2624.0 - 2603.0} = \frac{249.96 - 242.56}{250 - 242.56}$$
$$u_1 - 2603.0 = \frac{7.4 \times 21}{7.44}$$
$$u_1 = 20.887 + 2603.0$$
$$u_1 = 2623.9\ kJ/kg$$

Step 4 of 10

Calculate the pressure of steam at state 2 when the piston is hitting the stops.

$$P_2 = P_1$$

Substitute 3.5 MPa for P_1.

$$P_2 = 3.5\ MPa$$

Write the specific volume of steam at state 2 corresponding to pressure of 3.5 MPa and dryness fraction of 0.

$$v_2 = 0.001235\ m^3/kg$$

Write the internal energy of steam at state 2 corresponding to pressure of 3.5 MPa and dryness fraction of 0.

$$u_2 = 1045.4\ kJ/kg$$

Step 5 of 10

Calculate the specific volume of steam at state 3 when the piston stops.

$$v_3 = v_2$$

Here, specific volume of steam at state 2 is v_2 and specific volume of steam at state 3 is v_3.

Substitute 0.001235 m^3/kg for v_2.

$$v_3 = 0.001235\ m^3/kg$$

Calculate the saturated pressure of steam at state 3 by using saturated water-temperature table.

$$P_3 = P_{sat@200°C}$$
$$= 1554.9\ kPa$$

Therefore, the final pressure of steam is $\boxed{1554.9\ kPa}$.

Step 6 of 10

Calculate the dryness fraction of steam at state 3 by using saturated water-temperature table.

$$v_3 = v_f + x_3 v_{fg}$$

Calculate the dryness fraction of steam at state 3 by using saturated water-temperature table.

$$v_3 = v_f + x_3 v_{fg}$$
$$= v_f + x_3 (v_g - v_f)$$

Here, specific volume of saturated liquid is v_f, specific volume of saturated vapor is v_g and dryness fraction is x_3.

Substitute $0.001235\,\text{m}^3/\text{kg}$ for v_3, $0.001157\,\text{m}^3/\text{kg}$ for v_f and $0.12721\,\text{m}^3/\text{kg}$ for v_g.

$$0.001235 = 0.001157 + x_3 (0.12721 - 0.001157)$$
$$0.126 x_3 = 0.000078$$
$$x_3 = 0.000619$$

Therefore, the dryness fraction of steam at exit is $\boxed{0.000619}$.

Step 7 of 10

(b)

Calculate the internal energy of steam at state 3 by using saturated water-temperature table.

$$u_3 = u_f + x_3 u_{fg}$$

Here, internal energy of saturated liquid is u_f, internal energy of evaporation is u_{fg} and dryness fraction is x_3.

Substitute 0.000619 for x_3, $850.46\,\text{kJ/kg}$ for u_f and $1743.7\,\text{kJ/kg}$ for u_{fg}.

$$u_3 = 850.46 + (0.000619 \times 1743.7)$$
$$= 850.46 + 1.079$$
$$= 851.54\,\text{kJ/kg}$$

Step 8 of 10

Calculate the boundary work done by the steam.

$$W_b = mP_1 (v_1 - v_2)$$

Here, mass of steam is m.

Substitute $0.35\,\text{kg}$ for m, $3500\,\text{kPa}$ for P_1, $0.05875\,\text{m}^3/\text{kg}$ for v_1 and $0.001235\,\text{m}^3/\text{kg}$ for v_2.

$$W_b = 0.35 \times 3500 \times (0.05875 - 0.001235)$$
$$= 70.45\,\text{kJ}$$

Therefore, the boundary work is $\boxed{70.45\,\text{kJ}}$.

Step 9 of 10

(c)

Calculate the heat transfer at stop position by applying energy balance equation.

$$W_b - Q_{out\,1\text{-}2} = m(u_2 - u_1)$$

Here, heat lost during process $1-2$ is $Q_{out\,1\text{-}2}$, internal energy at state 2 is u_2 and internal energy at state 1 is u_1.

Substitute $70.45\,\text{kJ}$ for W_b, $0.35\,\text{kg}$ for m, $1045.4\,\text{kJ/kg}$ for u_2 and $2623.9\,\text{kJ/kg}$ for u_1.

$$Q_{out,1\text{-}2} = 70.45 - [0.35(1045.4 - 2623.9)]$$
$$= 622.925\,\text{kJ}$$

Therefore, heat transfer at stop position is $\boxed{622.925\,\text{kJ}}$.

Step 10 of 10

(d)

Calculate the total heat transfer.

$$W_b - Q_{out\,1\text{-}3} = m(u_3 - u_1)$$

Here, total heat lost during process $1-3$ is $Q_{out\,1\text{-}3}$, internal energy at state 3 is u_3 and internal energy at state 1 is u_1.

Substitute $70.45\,\text{kJ}$ for W_b, $0.35\,\text{kg}$ for m, $851.537\,\text{kJ/kg}$ for u_3 and $2623.9\,\text{kJ/kg}$ for u_1.

$$Q_{out,1\text{-}3} = 70.45 - [0.35(851.537 - 2623.9)]$$
$$= 690.78\,\text{kJ}$$

Therefore, the total heat transfer is $\boxed{690.78\,\text{kJ}}$.

CHAPTER 06

SOLUTIONS

Question: 6.001P

Define mass and volume flow rates. How are they related to each other?

Step 1 of 1

Mass Flow Rate:

Mass flow rate is defined as mass per unit time.

It is represented by $\dot{m}$. The SI units of mass flow rate is $\mathrm{kg/s}$.

Volumetric Flow Rate:

Volumetric flow rate is defined as the volume of fluid per unit time.

It is represented by $\dot{V}$. The SI units of volumetric flow rate is $\mathrm{m^3/s}$.

Relation between mass flow rate and volumetric flow rate:

Mass flow rate is directly proportional to the volumetric flow rate.

$$\dot{m} \propto \dot{V}$$

$$\dot{m} = \rho \dot{V} \qquad \left(\text{since } \rho = \frac{1}{v}\right)$$

$$= \frac{\dot{V}}{v}$$

Here, ρ is the density and v is the specific volume of the fluid.

Question: 6.002P

When is the flow through a control volume steady?

Step 1 of 1

If the properties in a control volume involve no change with time at any fixed position, then the flow through the control volume is said to be steady.

Question: 6.003P

Does the amount of mass entering a control volume have to be equal to the amount of mass leaving during an unsteady-flow process?

Step 1 of 1

No, the amount of mass entering a control volume does not have to be equal to the amount of mass leaving during an unsteady-flow process.

Question: 6.004P

Consider a device with one inlet and one outlet. If the volume flow rates at the inlet and at the outlet are the same, is the flow through this device necessarily steady? Why?

Step 1 of 1

No, the flow through the device is not steady unless the mass flow rate at inlet and outlet should be same

Question: 6.005P

A pneumatic accumulator arranged to maintain a constant pressure as air enters or leaves it is set for 200 psia. Initially, the volume is 0.2 ft3 and the temperature is 80°F. Air is now added to the accumulator until its volume is 1 ft3 and its temperature is 80°F. How much air has been added to the accumulator?

Step 1 of 3

Given

The initial pressure of the pneumatic accumulator $P_1 = 200$ psia

Initial volume, $V_1 = 0.2\ \text{ft}^3$

Initial temperature, $T_1 = 80°\text{F}$

$T_1 = 540\text{R}$

The final pressure of the pneumatic accumulator $P_2 = 200$ psia

Final volume, $V_2 = 1\ \text{ft}^3$

Finial temperature, $T_2 = 80°\text{F}$

$T_2 = 540\text{R}$

Step 2 of 3

From ideal gas properties of air, gas constant of air is $R = 0.3704$ psia·ft^3/lbm·R

The amount of mass added is $\Delta m = m_2 - m_1$

From the ideal gas equation, $P_1V_1 = m_1RT_1$,

$$m_1 = \frac{P_1V_1}{RT_1}$$

$$m_1 = \frac{(200)(0.2)}{(0.3704)(540)}$$

$$m_1 = 0.2\ \text{lbm}$$

The initial mass of air contained in the container is $m_1 = 0.2$ lbm

Step 3 of 3

From the ideal gas equation, $P_2V_2 = m_2RT_2$,

The final mass of air in the container,

$$m_2 = \frac{P_2V_2}{RT_2}$$

$$m_2 = \frac{(200)(1)}{(0.3704)(540)}$$

$$m_2 = 1\ \text{lbm}$$

Therefore the air added to the accumulator is

$$\Delta m = m_2 - m_1$$

$$\Delta m = 1 - 0.2$$

$$\Delta m = 0.8\ \text{lbm}$$

Hence air added to the accumulator is $\boxed{\Delta m = 0.8\ \text{lbm}}$

Question: 6.006P

A steady-flow compressor is used to compress helium from 15 psia and 70°F at the inlet to 200 psia and 600°F at the outlet. The outlet area and velocity are 0.01 ft2and 100 ft/s, respectively, and the inlet velocity is 50 ft/s. Determine the mass flow rate and the inlet area.

Step 1 of 3

Given

The initial pressure of the helium, $P_1 = 15$ psia

Initial temperature, $T_1 = 70°F$

$T_1 = 530R$

Inlet velocity, $V_1 = 50$ ft/s

Final pressure, $P_2 = 200$ psia

Finial temperature, $T_2 = 600°F$

$T_2 = 1060°F$

Outlet area, $A_2 = 0.01\ ft^2$

Outlet velocity, $V_2 = 100$ ft/s

Step 2 of 3

From ideal gas properties of air table, $R = 2.6809$ psia·ft^3/lbm·R

The mass flow rate is expressed as $\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2$

From the ideal gas equation, $P_2 = \rho_2 R T_2$

$$\rho_2 = \frac{P_2}{R T_2}$$

Therefore, $\dot{m} = \frac{P_2}{R T_2} A_2 V_2$

$$\dot{m} = \frac{200}{(2.6809)1060}(0.01)(100)$$

$$\dot{m} = 0.0704 \text{ lbm/s}$$

Hence mass flow rate is $\boxed{\dot{m} = 0.0704 \text{ lbm/s}}$

Step 3 of 3

From the ideal gas equation, $P_1 = \rho_1 R T_1$

$$\rho_1 = \frac{P_1}{R T_1}$$

Therefore, $\dot{m} = \frac{P_1}{R T_1} A_1 V_1$

$$A_1 = \frac{\dot{m} R T_1}{P_1 V_1}$$

$$A_1 = \frac{0.0704 \times 2.6809 \times 530}{15 \times 50}$$

$$A_1 = 0.133\ ft^2$$

Hence inlet area is $\boxed{A_1 = 0.133\ ft^2}$

Question: 6.007P

Air enters a nozzle steadily at 2.21 kg/m3 and 40 m/s and leaves at 0.762 kg/m3 and 180 m/s. If the inlet area of the nozzle is 90 cm2, determine (a) the mass flow rate through the nozzle, and (b) the exit area of the nozzle.

Step 1 of 3

Given data

The density at the inlet of the nozzle $\rho_1 = 2.21\ \text{kg/m}^3$

The velocity at the inlet is $V_1 = 40\ \text{m/s}$

The inlet area $A_1 = 90\ \text{cm}^2 = 90\times10^{-4}\ \text{m}^2$

The density at the outlet of the nozzle $\rho_2 = 0.762\ \text{kg/m}^3$

The velocity at the outlet is $V_2 = 180\ \text{m/s}$

Step 2 of 3

(a)

The mass flow rate

$$\dot{m} = \rho_1 A_1 V_1$$
$$= (2.21)\left(90\times10^{-4}\right)(40)$$
$$= 0.7956\ \text{kg/s}$$

$$\boxed{\dot{m} = 0.796\ \text{kg/s}}$$

Step 3 of 3

(b) The exit area of the nozzle

Since the process is steady mass flow rate remains constant in the nozzle, the exit area can be determined as

$$A_2 = \frac{\dot{m}}{\rho_2 V_2}$$
$$= \frac{0.796}{0.762(180)}$$
$$= 5.8\times10^{-3}\ \text{m}^2$$

$$\boxed{A_2 = 58\ \text{cm}^2}$$

Question: 6.008P

Water enters the constant 130-mm inside-diameter tubes of a boiler at 7 MPa and 65°C, and leaves the tubes at 6 MPa and 450°C with a velocity of 80 m/s. Calculate the velocity of the water at the tube inlet and the inlet volume flow rate.

Step 1 of 4

Given

Inside diameter of boiler tube $d = 130 \text{ mm} = 0.13 \text{ m}$

Inlet pressure $P_1 = 7 \text{ MPa}$

Inlet temperature $T_1 = 65°C$

Exit pressure $P_2 = 6 \text{ MPa}$

Exit temperature $T_2 = 450°C$

Exit velocity $V_2 = 80 \text{ m/s}$

Step 2 of 4

At $P_1 = 7 \text{ MPa}$ and $T_1 = 65°C$ state is in sub cooled region so we can take specific volume at $T_1 = 65°C$ from saturated water tables

$v_1 = v_f = 0.001020 \text{ m}^3/\text{kg}$

At $P_2 = 6 \text{ MPa}$ and $T_2 = 450°C$ from superheated steam tables

Specific volume $v_2 = 0.05217 \text{ m}^3/\text{kg}$

Step 3 of 4

For a steady state the mass flow rate remains constant.

$$\dot{m} = \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

Since it is a constant area duct the area is constant.

The inlet velocity is

$$\frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

$$V_1 = \frac{v_1}{v_2} V_2$$

$$= \frac{0.001020}{0.05217} \times 80$$

$$\boxed{V_1 = 1.56 \text{ m/s}}$$

Step 4 of 4

The inlet volume flow rate

$$\dot{V}_1 = A V_1$$

$$= \frac{\pi}{4}\left(0.13^2\right) 1.56$$

$$\boxed{\dot{V}_1 = 0.0207 \text{ m}^3/\text{s}}$$

Question: 6.009P

A hair dryer is basically a duct of constant diameter in which a few layers of electric resistors are placed. A small fan pulls the air in and forces it through the resistors where it is heated. If the density of air is 1.20 kg/m3 at the inlet and 0.95 kg/m3 at the exit, determine the percent increase in the velocity of air as it flows through the dryer.

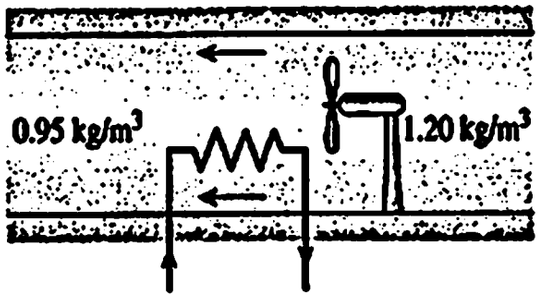


FIGURE P6–9

Step 1 of 3

Given

The density of air at inlet, $\rho_1 = 1.2\ \text{kg/m}^3$

The density of air at exit, $\rho_2 = 0.95\ \text{kg/m}^3$

Step 2 of 3

The mass flow rate is expressed as $\dot{m} = \rho A V$

$$A = \frac{\dot{m}}{\rho V}$$

For a constant area duct, $A_1 = A_2$

$$\frac{\dot{m}_1}{\rho_1 V_1} = \frac{\dot{m}_2}{\rho_2 V_2}$$

But the mass flow rate is constant, $\dot{m}_1 = \dot{m}_2$. Hence

$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2}$$

$$\frac{V_2}{V_1} = \frac{1.2}{0.95}$$

$$\frac{V_2}{V_1} = 1.263$$

Step 3 of 3

Percentage increase in volume is $\%\Delta V = \dfrac{V_2 - V_1}{V_1} \times 100$

$$\%\Delta V = \left(\frac{V_2}{V_1} - 1\right) \times 100$$

$$\%\Delta V = (1.263 - 1) \times 100$$

$$\%\Delta V = 26.3\%$$

Percentage increase in volume is $\boxed{\%\Delta V = 26.3\%}$

Question: 6.010P

A 1-m3 rigid tank initially contains air whose density is 1.18 kg/m3. The tank is connected to a high-pressure supply line through a valve. The valve is opened, and air is allowed to enter the tank until the density in the tank rises to 7.20 kg/m3. Determine the mass of air that has entered the tank.

Step 1 of 3

Given data

The volume of the rigid tank $V = 1\ \text{m}^3$

The density of air $\rho_1 = 1.18\ \text{kg/m}^3$

The density of air in the tank at state 2 $\rho_2 = 7.20\ \text{kg/m}^3$

Step 2 of 3

By applying mass balance to the system

$$\Delta m_{system} = m_{in} - m_{out}$$

Since

$$\rho = \frac{m}{V}$$

$$m = \rho V$$

Step 3 of 3

Therefore mass balance in terms of densities we can write it as

$$\Delta m_{system} = \rho_2 V - \rho_1 V$$

$$= (\rho_2 - \rho_1) V$$

$$= (7.20 - 1.18)(1)$$

The mass of air entering the system is

$$\boxed{\Delta m_{system} = 6.02\ \text{kg}}$$

Question: 6.011P

Air enters the 1-m2 inlet of an aircraft engine at 100 kPa and 20°C with a velocity of 180 m/s. Determine the volume flow rate, in m3/s, at the engine's inlet and the mass flow rate, in kg/s, at the engine's exit.

Step 1 of 4

Given

Inlet cross-section area $A_1 = 1\ \text{m}^2$

Inlet pressure $P_1 = 100\ \text{kPa}$

Inlet temperature $T_1 = 20°\text{C} = 293\text{K}$

Inlet velocity $V_1 = 180\ \text{m/s}$

Step 2 of 4

The volume flow rate at the inlet

$$\dot{V}_1 = A_1 V_1$$
$$= (1)(180)$$
$$\boxed{\dot{V}_1 = 180\ \text{m}^3/\text{s}}$$

Step 3 of 4

The mass flow rate remains constant since the flow is steady.

Therefore

$$\dot{m} = \rho_1 \dot{V}_1$$

From ideal gas equation

$$\rho_1 = \frac{P_1}{RT_1}$$

Where R is the gas constant for air $R = 0.287\ \text{kJ/kg.K}$

Then

$$\rho_1 = \frac{100}{(0.287)(293)}$$
$$\rho_1 = 1.189\ \text{kg/m}^3$$

Step 4 of 4

The mass flow rate at the exit is

$$\dot{m} = \rho_1 \dot{V}_1$$
$$= 1.189(180)\ \text{kg/s}$$
$$\boxed{\dot{m} = 214.02\ \text{kg/s}}$$

Question: 6.012P

A pump increases the water pressure from 70 kPa at the inlet to 700 kPa at the outlet. Water enters this pump at 15°C through a 1-cm-diameter opening and exits through a 1.5-cm-diameter opening. Determine the velocity of the water at the inlet and outlet when the mass flow rate through the pump is 0.5 kg/s. Will these velocities change significantly if the inlet temperature is raised to 40°C?

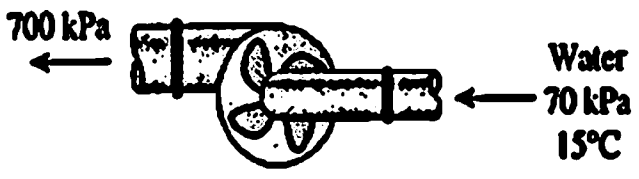


FIGURE P6–12

Step 1 of 6

Given

Inlet pressure $P_1 = 70\ \text{kPa}$

Inlet temperature $T_1 = 15°\text{C} = 288\ \text{K}$

Inlet diameter $d_1 = 1\ \text{cm} = 0.01\ \text{m}$

Exit pressure $P_2 = 700\ \text{kPa}$

Exit diameter $d_2 = 1.5\ \text{cm} = 0.015\ \text{m}$

Mass flow rate of water $\dot{m}_w = 0.5\ \text{kg/s}$

The increased inlet temperature is $T_1' = 40°\text{C} = 313\ \text{K}$

Step 2 of 6

From properties of water tables

At $T_1 = 15°\text{C}$

Specific volume $v_1 = v_f = 0.001001\ \text{m}^3/\text{kg}$

At $T_1' = 40°\text{C}$

Specific volume $v_1' = v_f = 0.001008\ \text{m}^3/\text{kg}$

Step 3 of 6

The velocity at the inlet

$$\dot{m} = \rho_1 A_1 V_1$$

$$\dot{m} = \frac{A_1 V_1}{v_1}$$

$$V_1 = \frac{\dot{m} v_1}{A_1}$$

$$= \frac{(0.5)(0.001001)}{\pi\left(\frac{0.01^2}{4}\right)}$$

$$\boxed{V_1 = 6.37\ \text{m/s}}$$

Step 4 of 6

The velocity at the outlet

Since the specific volume and mass flow rate remains constant

$$\dot{m} = \frac{A_1 V_1}{v} = \frac{A_2 V_2}{v}$$

$$A_1 V_1 = A_2 V_2$$

$$V_2 = V_1 \frac{A_1}{A_2}$$

$$= 6.37 \times \frac{(\pi/4)0.01^2}{(\pi/4)0.015^2}$$

$$\boxed{V_2 = 2.83\ \text{m/s}}$$

Step 5 of 6

Velocity of the water at inlet when the temperature is raised to 40^0C

$$\dot{m} = \rho_1 A_1 V_1$$

$$\dot{m} = \frac{A_1 V_1'}{v_1'}$$

$$V_1' = \frac{\dot{m} v_1'}{A_1}$$

$$= \frac{(0.5)(0.001008)}{\pi\left(\frac{0.01^2}{4}\right)}$$

$$\boxed{V_1' = 6.417\ \text{m/s}}$$

Step 6 of 6

Percentage change in velocity

$$\% = \frac{V_1' - V_1}{V_1} \times 100$$

$$= \frac{6.417 - 6.37}{6.37} \times 100$$

$$\% = 0.7378\%$$

If the inlet temperature changes from 15^0C to 40^0C the velocity increased by only 0.73%

Question: 6.013P

Refrigerant-134a enters a 28-cm diameter pipe steadily at 200 kPa and 20°C with a velocity of 5 m/s. The refrigerant gains heat as it flows and leaves the pipe at 180 kPa and 40°C. Determine (a) the volume flow rate of the refrigerant at the inlet, (b) the mass flow rate of the refrigerant, and (c) the velocity and volume flow rate at the exit.

Step 1 of 6

Loading solution. Please wait ...

Step 2 of 6

At $P_1 = 200$ kPa and $T_1 = 20°C$ from superheated R-134a tables

Specific volume $v_1 = 0.11418\ m^3/kg$

At $P_2 = 180$ kPa and $T_2 = 40°C$ from superheated R-134a tables

Specific volume $v_2 = 0.13741\ m^3/kg$

Step 2 of 6

At $P_1 = 200$ kPa and $T_1 = 20°C$ from superheated R-134a tables

Specific volume $v_1 = 0.11418\ m^3/kg$

At $P_2 = 180$ kPa and $T_2 = 40°C$ from superheated R-134a tables

Specific volume $v_2 = 0.13741\ m^3/kg$

Step 3 of 6

(a)

The inlet volume flow rate

$$\dot{V}_1 = AV_1$$

$$= \frac{\pi}{4}(0.28^2)5$$

$$\boxed{\dot{V}_1 = 0.3078\ m^3/s}$$

Step 4 of 6

(b)

The mass flow rate of the refrigerant

$$\dot{m} = \rho_1 \dot{V}_1$$

$$\dot{m} = \frac{\dot{V}_1}{v_1}$$

$$= \frac{0.3078}{0.11418}$$

$$\boxed{\dot{m} = 2.695\ kg/s}$$

Step 5 of 6

(c)

Since mass flow rate remains constant

$$\dot{m} = \dot{m}_1 = \dot{m}_2$$

$$\frac{V_1}{v_1} = \frac{V_2}{v_2}$$

$$V_2 = \frac{v_2}{v_1} V_1$$

$$= \frac{0.13741}{0.11418} \times 5$$

$$\boxed{V_1 = 6.017\ m/s}$$

Step 6 of 6

The exit volume flow rate

$$\dot{V}_2 = \dot{m} v_2$$

$$= (2.695)(0.13741)$$

$$\boxed{\dot{V}_2 = 0.3703\ m^3/s}$$

Question: 6.014P

A smoking lounge is to accommodate 15 heavy smokers. The minimum fresh air requirement for smoking lounges is specified to be 30 L/s per person (ASHRAE, Standard 62, 1989). Determine the minimum required flow rate of fresh air that needs to be supplied to the lounge, and the diameter of the duct if the air velocity is not to exceed 8 m/s.

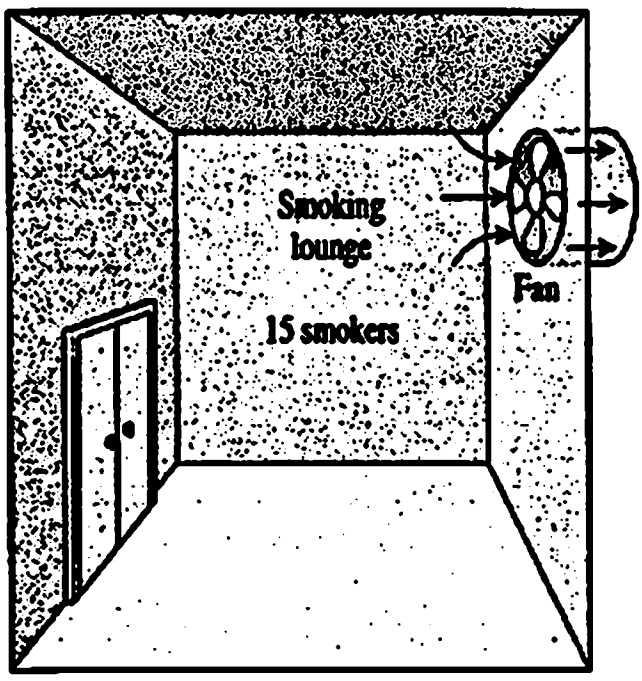


FIGURE P6–14

Step 1 of 3

Given data

The flow rate of the fresh air per person is $\dot{V} = 30\ \text{L/s}$

The velocity of air in to the lounge $V = 8\ \text{m/s}$

Number of persons $N = 15$

Step 2 of 3

The total volume flow rate of fresh air required in to the lounge is

$$\dot{V}_{total} = N\dot{V}$$

$$\dot{V}_{total} = 15 \times 30$$

$$= 450\ \text{L/s}$$

$$= 450 \times 10^{-3}\ \text{m}^3/\text{s} \qquad (1\ \text{L} = 10^{-3}\ \text{m}^3)$$

$$\boxed{\dot{V}_{total} = 0.45\ \text{m}^3/\text{s}}$$

Step 3 of 3

We know $\dot{V}_{total} = A \times V$

The area of the duct

$$A = \frac{\dot{V}_{total}}{V}$$

$$\pi\frac{d^2}{4} = \frac{0.45}{8}$$

$$d^2 = \frac{0.45 \times 4}{8 \times \pi}$$

$$d^2 = 0.0716\ \text{m}^2$$

$$\boxed{d = 0.267\ \text{m}}$$

Question: 6.015P

The minimum fresh air requirement of a residential building is specified to be 0.35 air change per hour (ASHRAE, Standard 62, 1989). That is, 35 percent of the entire air contained in a residence should be replaced by fresh outdoor air every hour. If the ventilation requirement of a 3.0-m-high, 200-m2 residence is to be met entirely by a fan, determine the flow capacity in L/min of the fan that needs to be installed. Also determine the diameter of the duct if the air velocity is not to exceed 4 m/s.

Step 1 of 3

Given

The area of the residence, $A = 200\ \text{m}^2$

The height of the building, $h = 3\ \text{m}$

Percentage of air that should be replaced by fresh air is 35%

The maximum velocity of air in the duct, $V = 4\ \text{m/s}$

Step 2 of 3

The volume of the entire building, $Vol = 3 \times 200\ \text{m}^3$

$$Vol = 600\ \text{m}^3$$

The flow capacity of the fan is $\dot{V} = \dfrac{(0.35)(600)}{60}\ \text{m}^3/\text{min}$

$$\dot{V} = 3.5\ \text{m}^3/\text{min}$$

But $1\ \text{L} = 10^{-3}\ \text{m}^3$, Hence $\dot{V} = 3500\ \text{L}/\text{min}$

Therefore the minimum flow rate should be $\boxed{\dot{V} = 3500\ \text{L}/\text{min}}$

Step 3 of 3

Volume flow rate of fresh air is $\dot{V} = A \times V$

$$\dot{V} = \frac{\pi d^2}{4} \times V$$

The diameter of the duct, $d = \sqrt{\dfrac{4\dot{V}}{\pi V}}$

$$d = \sqrt{\frac{4 \times 3.5\ \text{m}^3/\text{min}}{\pi \times 4\ \text{m/s}}}$$

$$d = \sqrt{\frac{4 \times 3.5\ \text{m}^3/\text{s}}{\pi \times 4 \times 60\ \text{m/s}}}$$

$$d = \sqrt{0.01857\ \text{m}^2}$$

$$d = 0.136\ \text{m}$$

Hence diameter of duct should be $\boxed{d = 0.136\ \text{m}}$

Question: 6.016P

What are the different mechanisms for transferring energy to or from a control volume?

Step 1 of 1

The different mechanisms which transfer energy to or from control volume are heat, work and mass.

Question: 6.017P

What is flow energy? Do fluids at rest possess any flow energy?

Step 1 of 1

Control volumes involve mass flow across their boundaries, and some work is required to push the mass into or out of the control volume. This work is known as the flow work, or flow energy.

No, fluids at rest do not posses any flow energy.

Question: 6.018P

How do the energies of a flowing fluid and a fluid at rest compare? Name the specific forms of energy associated with each case.

Step 1 of 2

The fluid at rest has total energy

$$e = u + ke + pe$$

$$e = u + \frac{v^2}{2} + gz \qquad \text{(kJ/kg)}$$

The total energy of fluid at rest consists of internal, kinetic and potential energies.

Step 2 of 2

The fluid entering or leaving a control volume possess an additional form of energy, i.e. the flow energy. Then the total energy of a flowing fluid on a unit mass basis becomes

Question:

6.019P

A water pump increases the water pressure from 10 psia to 50 psia. Determine the flow work, in Btu/lbm, required by the pump.

Step 1 of 2

Given

Initial pressure $p_1 = 10\,\text{psia}$

Final pressure $p_2 = 50\,\text{psia}$

Step 2 of 2

From properties of saturated water tables at $p_1 = 10\,\text{psia}$

$v = v_f = 0.01659\ \text{ft}^3/\text{lbm}$

Here the working substance is water and the specific volume remains constant then the flow work becomes

Question:

A water pump increases the water pressure from 10 psia to 50 psia. Determine the flow work, in Btu/lbm, required by the pump.

Question: 6.020P

An air compressor compresses 6 L of air at 120 kPa and 20°C to 1000 kPa and 400°C. Determine the flow work, in kJ/kg, required by the compressor.

Step 1 of 1

From ideal gas properties of air table, gas constant of air, $R = 0.287\ \text{kJ/kg}\cdot\text{K}$

From ideal gas equation, $Pv = RT$

Calculate the flow work,

$$\begin{aligned} W &= P_2 v_2 - P_1 v_1 \\ &= R(T_2 - T_1) \end{aligned}$$

Here,

Initial Temperature of air, T_1

Final Temperature of air, T_2

Gas constant of air, R

Inlet pressure of the air, P_1

Outlet pressure of the air, P_2

Substitute 400°C or 673K for T_2 , 0.287 kJ/kg·K for R , 20°C or 293K for T_1 , 1000 kpa for P_2 ,

$$\begin{aligned} W &= 0.287(673 - 293) \\ &= 109\ \text{kJ/kg} \end{aligned}$$

Therefore, flow work is $\boxed{109\ \text{kJ/kg}}$

Question: 6.021P

Steam is leaving a pressure cooker whose operating pressure is 20 psia. It is observed that the amount of liquid in the cooker has decreased by 0.6 gal in 45 minutes after the steady operating conditions are established, and the cross-sectional area of the exit opening is 0.15 in2. Determine (a) the mass flow rate of the steam and the exit velocity, (b) the total and flow energies of the steam per unit mass, and (c) the rate at which energy is leaving the cooker by steam.

Step 1 of 6

(a)

Calculate the mass of water that converted into steam.

$$m = \frac{V}{v_f}$$

Here, m is the mass of water that converted into steam, V is the volume of the steam, and v_f is the specific volume of the liquid.

Take the values of steam from the table "properties of saturated water" recorded at pressure $20\ \text{psia}$ at saturated state

Specific volume, $v_f = 0.01683\ \text{ft}^3/\text{lbm}$.

Substitute $0.01683\ \text{ft}^3/\text{lbm}$ for v_f and $0.6\ \text{gal}$ for V.

$$m = \frac{0.6\text{gal}}{0.01683\ \text{ft}^3/\text{lbm}}$$
$$= \frac{0.6 \times 0.1336\text{ft}^3}{0.01683\ \text{ft}^3/\text{lbm}}$$
$$= 4.762\text{lbm}$$

Step 2 of 6

Calculate the mass flow rate of the steam leaving the pressure cooker.

Water that converted into steam leaves the pressure cooker from the valve.

$$\dot{m} = \frac{m}{t}$$

Here, $\dot{m}$ is the mass flow rate of the steam and t is time.

$$\dot{m} = \frac{4.762}{45 \times 60}$$
$$\dot{m} = 1.7637 \times 10^{-3}\ \text{lbm/sec}$$

Thus, the mass flow rate of the steam is $\boxed{1.7637 \times 10^{-3}\ \text{lbm/sec}}$.

Step 3 of 6

Calculate the exit velocity of the steam.

Consider the equation of continuity.

$$\dot{m} = \frac{AV'}{v_g}$$

Here, A is the cross sectional area of the exit opening, v_g is the specific volume of vapor, and V' is the velocity of the steam.

Here, v_g is the specific volume of vapor is considered since the steam leave as vapor from the valve.

$$V' = \frac{\dot{m}v_g}{A}$$

Take the values of steam from the table "properties of super-heated steam," recorded at pressure $20\ \text{psia}$ at saturated state.

Specific volume, $v_g = 20.093\ \text{ft}^3/\text{lbm}$.

Substitute $20.093\ \text{ft}^3/\text{lbm}$ for v_g , $1.477 \times 10^{-6}\ \text{lbm/sec}$ for $\dot{m}$, and $0.15\ \text{in}^2$ for A.

$$V' = \frac{1.7637 \times 10^{-3} \times 20.093}{0.15\left(\frac{1}{12}\right)^2}$$
$$V' = 34.020\ \text{ft/s}$$

Thus, the exit velocity of the steam is $\boxed{34.020\ \text{ft/s}}$.

Step 4 of 6

(b)

Calculate the flow energy of the steam per unit mass.

$$\text{flow energy, } E_f = PV$$

Write the relation between specific enthalpy, h and internal energy, u.

$$h = u + PV$$
$$PV = h - u$$
$$E_f = h - u$$

Take the values of steam from the table "properties of saturated water" recorded at pressure $20\ \text{psia}$ at saturated state.

Specific volume, $v_g = 20.093\ \text{ft}^3/\text{lbm}$.

Specific internal energy, $u = 1081.8\ \text{Btu/lbm}$.

Specific enthalpy, $h = 1156.2\ \text{Btu/lbm}$.

Substitute $1156.2\ \text{Btu/lbm}$ for h and $1081.8\ \text{Btu/lbm}$ for u.

$$E_f = 1156.2 - 1081.8$$
$$E_f = 74.4\ \text{Btu/lbm}$$

Thus, the flow energy of the steam per unit mass is $\boxed{74.4\ \text{Btu/lbm}}$.

Step 5 of 6

Calculate the total energy of steam per unit mass.

$$\theta = h + ke + pe$$

Here, θ is the total energy of steam per unit mass, ke is the kinetic energy, and pe is the potential energy.

$$\theta = h + \frac{1}{2}(V')^2 + pe$$

Substitute $1156.2\ \text{Btu/lbm}$ for h, $0.02849\ \text{ft/s}$ for V' , and 0 for pe .

$$\theta = 1156.2\ \text{Btu/lbm} + \frac{1}{2}(34.020\ \text{ft/s})^2 + 0$$
$$\theta = 1156.2\ \text{Btu/lbm} + 578.6802 \times \frac{1}{25037}\ \text{Btu/lbm}$$
$$\theta = 1156.2231\ \text{Btu/lbm}$$

Thus, the total energy of the steam per unit mass is $\boxed{1156.2231\ \text{Btu/lbm}}$.

Step 6 of 6

(c)

Calculate the total energy flow rate.

$$\dot{E} = \dot{m}\theta$$

Here, $\dot{E}$ is the total energy flow rate.

Substitute $1156.2231\ \text{Btu/lbm}$ for θ and $1.7637 \times 10^{-3}\ \text{lbm/sec}$ for $\dot{m}$.

$$\dot{E} = 1156.2231 \times 1.7637 \times 10^{-3}$$
$$\dot{E} = 2.039\ \text{Btu/s}$$

Thus, the total energy flow rate is $\boxed{2.039\ \text{Btu/s}}$.

Question: 6.022P

Air flows steadily in a pipe at 300 kPa, 77°C, and 25 m/s at a rate of 18 kg/min. Determine (a) the diameter of the pipe, (b) the rate of flow energy, (c) the rate of energy transport by mass, and (d) the error involved in part (c) if the kinetic energy is neglected.

Step 1 of 8

Given

Pressure $P = 300$ kpa

Temperature $T = 77°C = 350K$

Velocity $V = 25$ m/s

Mass flow rate $\dot{m} = 18$ kg/min

$$= \frac{18}{60}$$

$$= 0.3 \text{ kg/s}$$

Step 2 of 8

From table ideal gas specific heats of air at $T = 350K$,

$c_p = 1.008$ kJ/kg.K

$c_v = 0.721$ kJ/kg.K

$R = C_p - C_v$

$= 1.008 - 0.721$

$R = 0.287$ kJ/kg.K

Step 3 of 8

From ideal gas equation

$$\frac{P}{\rho} = RT$$

$$\rho = \frac{P}{RT}$$

$$= \frac{300}{(0.287)(350)}$$

$\rho = 2.98656$ kg/m^3

Step 4 of 8

(a)

The diameter of the pipe

We know the mass flow rate is

$$\dot{m} = \rho A V$$

$$A = \frac{\dot{m}}{\rho V}$$

$$\pi \frac{d^2}{4} = \frac{\dot{m}}{\rho V}$$

$$d^2 = \frac{4\dot{m}}{\pi \rho V}$$

$$d = \sqrt{\frac{4}{\pi}\left(\frac{0.3}{2.98656 \times 25}\right)}$$

$\boxed{d = 0.0715 \text{ m}}$

Step 5 of 8

(b)

The rate of flow energy

$$\dot{W} = \dot{m} P v \qquad \left(\text{since } v = \frac{1}{\rho}\right)$$

$$\dot{W} = \frac{\dot{m} P}{\rho}$$

$$= (0.3)(300)\left(\frac{1}{2.98656}\right)$$

$\boxed{\dot{W} = 30.13 \text{ kW}}$

Step 6 of 8

(c)

The rate of energy transport by mass

$$\dot{E}_{mass} = \dot{m}\theta = \dot{m}\left(h + \frac{V^2}{2} + gz\right)$$

Neglecting potential energy and substituting $h = c_p T$

Then

$$\dot{E}_{mass} = \dot{m}\left(c_p T + \frac{V^2}{2}\right)$$

$$\dot{E}_{mass} = 0.3\left(1.008 \times 10^3 (350) + \frac{25^2}{2}\right)$$

$$= 0.3(352800 + 312.5)$$

$$= 105933.75 \text{W}$$

$\boxed{\dot{E}_{mass} = 105.93 \text{ kW}}$

Step 7 of 8

(d)

If the kinetic energy is neglected

The rate of energy transport by mass

$$\dot{E}_{mass} = \dot{m} h$$

$$= \dot{m} c_p T$$

$$= 0.3\left(1.008 \times 10^3\right)(350)$$

$$= 105840 \text{ W}$$

$\dot{E}_{mass} = 105.84$ kW

Step 8 of 8

The error involved in part (c) is

error $= 105.93 - 105.84$

$\boxed{\text{error} = 0.09}$

i.e. 9% of error will occur if we neglect Kinetic energy.

Question: 6.023P

Can a steady-flow system involve boundary work?

Step 1 of 1

No, in a steady flow system the fluid properties at an inlet or exit remain constant, so there will be no boundary work involved.

Question: 6.024P

A diffuser is an adiabatic device that decreases that kinetic energy of the fluid by slowing it down. What happens to this *lost* kinetic energy?

Step 1 of 1

A diffuser is a device that increases the pressure of a fluid by slowing it down. The lost kinetic energy is converted as internal energy.

Question: 6.025P

The kinetic energy of a fluid increases as it is accelerated in an adiabatic nozzle. Where does this energy come from?

Step 1 of 1

The nozzle is a device that increases the velocity of the fluid at the expense of pressure.
The internal energy of the fluid will be converted into kinetic energy.

Question: 6.026P

Is heat transfer to or from the fluid desirable as it flows through a nozzle? How will heat transfer affect the fluid velocity at the nozzle exit?

Step 1 of 1

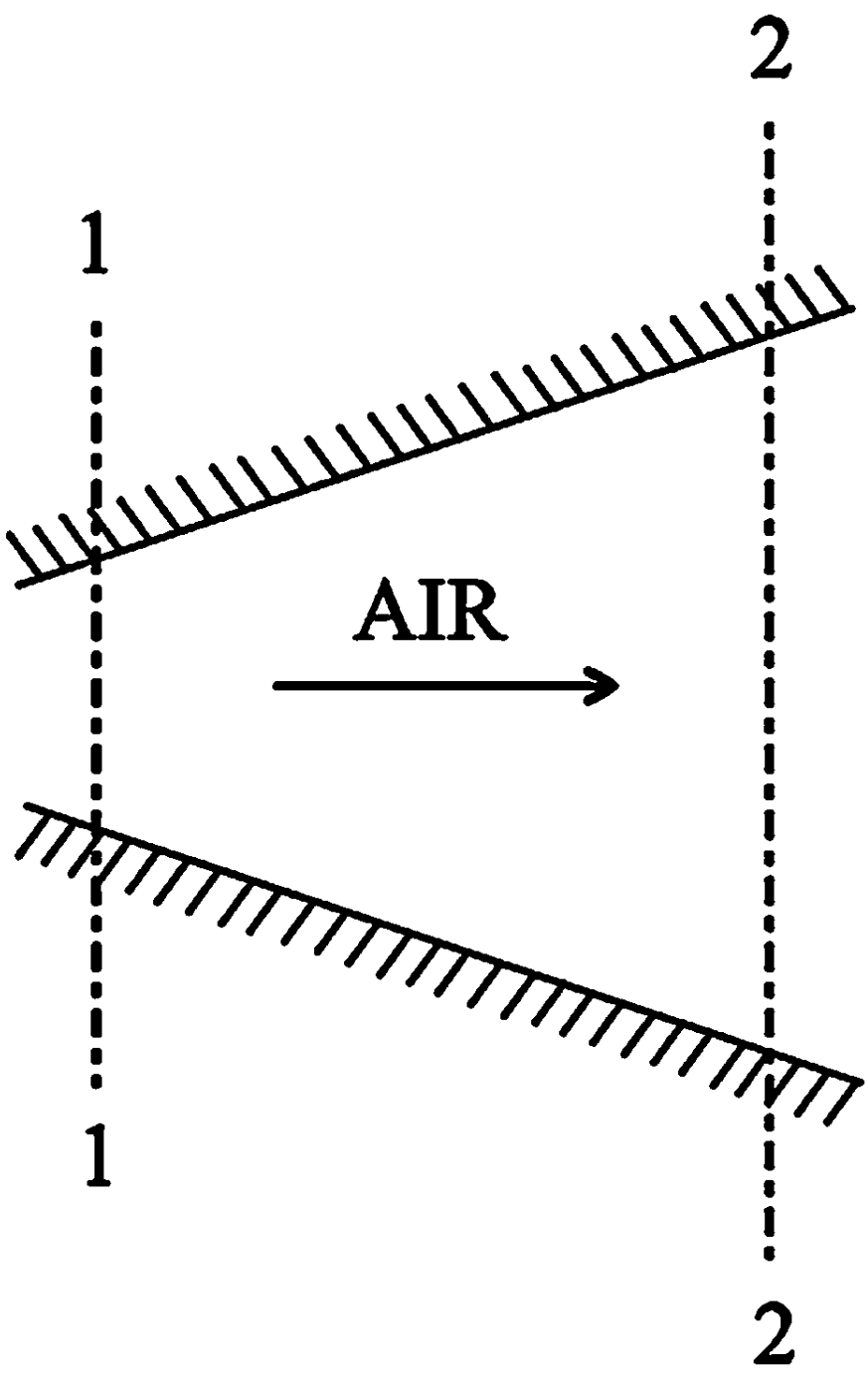
Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity

Question: 6.027P

Air at 80 kPa and 127°C enters an adiabatic diffuser steadily at a rate of 6000 kg/h and leaves at 100 kPa. The velocity of the air stream is decreased from 230 to 30 m/s as it passes through the diffuser. Find (a) the exit temperature of the air and (b) the exit area of the diffuser.

Step 1 of 5

Draw the schematic diagram of the diffuser.



Step 2 of 5

Assumptions:

1. The flow of air is a steady-flow process

2. Device is adiabatic and heat transfer is negligible

3. Potential energy changes are negligible

4. Work interactions are negligible

Determine the enthalpy of air at state 1 corresponding to a temperature of 400K from ideal gas properties of air.

$$h_1 = 400.98 \text{ kJ/kg}$$

Step 3 of 5

Write the conversion factor to convert kg/h to kg/s.

$$1 \text{ kg/h} = 2.78\times10^{-4} \text{ kg/s}$$

Convert the mass flow rate from 6000 kg/h to kg/s.

$$6000 \text{ kg/h} = 6000 \text{ kg/h} \times \frac{2.78\times10^{-4} \text{ kg/s}}{1 \text{ kg/h}}$$
$$= 1.67 \text{ kg/s}$$

Step 4 of 5

(a)

Calculate the enthalpy at state 2 by using energy balance equation per unit mass for steady state process.

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here, heat supplied to the system is q .work done by the system is w , enthalpy at state 1 is h_1 , enthalpy at state 2 is h_2 , velocity of air at state 1 is V_1 , velocity of air at state 2 is V_2 , potential head at state 1 is z_1 and potential head at state 2 is z_2.

Substitute 230 m/s for V_1, 30 m/s for V_2, $400.98\times10^3 \text{ J/kg}$ for h_1, 9.81 m/s^2 for g, 0 for q, 0 for w and $z_1 = z_2$.

$$0 - 0 = h_2 - (400.98\times10^3) + \frac{(30)^2 - (230)^2}{2} + (9.81\times0)$$
$$h_2 = (400.98\times10^3) + 26000$$
$$h_2 = 426.98 \text{ kJ/kg}$$

Step 5 of 5

Calculate the temperature at state 2 by use of interpolation of values from ideal gas properties of air.

$$\frac{T_{@426.98\text{ kJ/kg}} - T_{@421.26\text{ kJ/kg}}}{T_{@431.43\text{ kJ/kg}} - T_{@421.26\text{ kJ/kg}}} = \frac{h_{@426.98\text{ kJ/kg}} - h_{@421.26\text{ kJ/kg}}}{h_{@431.43\text{ kJ/kg}} - h_{@421.26\text{ kJ/kg}}}$$
$$T_2 = 420 + (430 - 420)\left(\frac{426.98 - 421.26}{431.43 - 421.26}\right)$$
$$T_2 = 425.62 \text{ K}$$

Therefore, the temperature at exit is $\boxed{425.62 \text{ K}}$.

(b)

Calculate the density of air at the exit of the diffuser from ideal gas equation.

$$\rho_2 = \frac{P_2}{RT_2}$$

Here, pressure of air at exit is P_2, gas constant for air is R and temperature of air at exit is T_2.

Substitute $0.287 \text{ kJ/kg}\cdot\text{K}$ for R, 100 kPa for P_2 and T_2 for 425.62 K.

$$\rho_2 = \frac{100}{(0.287)\times(425.62)}$$
$$= 0.818 \text{ kg/m}^3$$

Calculate the exit area of the diffuser by maintain the mass flow rate constant.

$$\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

$$A_2 = \frac{\dot{m}}{V_2 \rho_2}$$

Here, mass flow rate of air is $\dot{m}$, area of the diffuser at inlet is A_1, density of air at inlet is ρ_1, density of air at exit is ρ_2, velocity of air at state 1 is V_1 and velocity of air at state 2 is V_2.

Substitute 1.67 kg/s for $\dot{m}$, 0.818 kg/m^3 for ρ_2 and 30 m/s for V_2.

$$A_2 = \frac{1.67}{30\times0.818}$$
$$= 0.068 \text{ m}^2$$

Therefore, the exit area of the diffuser is $\boxed{0.068 \text{ m}^2}$.

Question: 6.028P

Air enters an adiabatic nozzle steadily at 300 kPa, 200°C, and 45 m/s and leaves at 100 kPa and 180 m/s. The inlet area of the nozzle is 110 cm2. Determine (*a*) the mass flow rate through the nozzle, (b) the exit temperature of the air, and (*c*) the exit area of the nozzle.

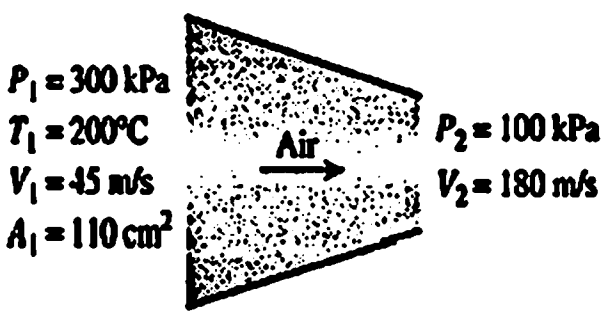


FIGURE P6–28

Step 1 of 5

Given

Inlet Pressure of air, $P_1 = 300$ kPa

Inlet Temperature of air, $T_1 = 200°C$

$$T_1 = 473K$$

Inlet Velocity of air, $V_1 = 45$ m/s

Inlet area of nozzle, $A_1 = 110$ cm^2

$$A_1 = 0.011\ \text{m}^2$$

Exit Pressure of air, $P_2 = 100$ kPa

Exit velocity of air, $V_2 = 180$ m/s

Ideal gas properties of air table,

The gas constant for air $R = 0.287$ kJ/kg.K and

Specific heat of air, $C_p = 1.02$ kJ/kg.K

Step 2 of 5

(a)

From ideal gas equation, $P = \rho RT$

Density of air, $\rho = \dfrac{P}{RT}$

$$\rho_1 = \frac{P_1}{RT_1}$$

$$\rho_1 = \frac{300}{(0.287)(473)}$$

$$\rho_1 = 2.21\ \text{kg/m}^3$$

Mass flow rate, $\dot{m} = \rho A_1 V_1$

$$\dot{m} = (2.21)(0.011)(45)\ \text{kg/s}$$

$$\dot{m} = 1.094\ \text{kg/s}$$

Therefore mass flow rate is $\boxed{\dot{m} = 1.094\ \text{kg/s}}$

Step 3 of 5

(b)

The energy balance equation per unit mass for a steady state process is

$$(q - w) = (h_2 - h_1) + \left(\frac{V_2^2 - V_1^2}{2}\right) + g(z_2 - z_1)$$

As the process is adiabatic, heat transfer is $q = 0$

And external work done on this process is $w = 0$

Also the nozzle entrance and exit are at same level. So $z_1 = z_2$

Enthalpy $h = C_p T$

Therefore $0 = C_p(T_2 - T_1) + \dfrac{V_2^2 - V_1^2}{2}$

$$(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2C_p}$$

$$T_2 = T_1 - \left(\frac{V_2^2 - V_1^2}{2C_p}\right)$$

$$T_2 = 200 - \left(\frac{180^2 - 45^2}{2 \times 1.02 \times 10^3\ \text{J/kg}}\right)$$

$$T_2 = 200 - (14.89)$$

$$T_2 = 185.11°C$$

The exit temperature of the air $\boxed{T_2 = 185.11°C}$

Step 4 of 5

(c)

From ideal gas equation, $P = \rho RT$

$$\rho_2 = \frac{P_2}{RT_2}$$

$$\rho_2 = \frac{100}{(0.287)(185.11 + 273)}$$

$$\rho_2 = 0.762\ \text{kg/m}^3$$

Step 5 of 5

Mass flow rate, $\dot{m} = \rho AV$

Mass flow rate remains constant throughout the nozzle, $\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2$

Therefore $A_2 = \dfrac{\dot{m}}{\rho_2 V_2}$

$$A_2 = \frac{1.1}{0.762 \times 180}$$

$$A_2 = 80 \times 10^{-4}\ \text{m}^2$$

$$A_2 = 80\ \text{cm}^2$$

The exit area of the nozzle $\boxed{A_2 = 80\ \text{cm}^2}$

Question: 6.029P pg 1

Reconsider Prob. 6-28. Using EES (or other) IIII software, investigate the effect of the inlet area on the mass flow rate, exit temperature, and the exit area. Let the inlet area vary from 50 cm2 to 150 cm2. Plot the final results against the inlet area, and discuss the results.

Step 1 of 8

a)The ideal gas equation,

$$P = \rho RT$$

The Density of air,

$$\rho = \frac{P}{RT}$$

$$\rho_1 = \frac{P_1}{RT_1}$$

Here, the Inlet Pressure of air is P_1, the Inlet Temperature of air is T_1 and the gas constant for air R.

Substitute $300\ \text{kPa}$ for P_1 . $200+273 = 473\text{K}$ for T_1 .

From Ideal gas properties of air table,

The gas constant for air is $R = 0.287\,\text{kJ/kg.K}$ and Specific heat of air is $C_p = 1.02\,\text{kJ/kg.K}$

$$\rho_1 = \frac{300}{(0.287)(473)}$$
$$= 2.21\ \text{kg/m}^3$$

The Mass flow rate of air,

$$\dot{m} = \rho A_1 V_1$$

Here, the Inlet area of nozzle is A_1 and the Inlet Velocity of air is V_1 .

Substitute $0.011\,\text{m}^2$ for A_1 and $45\,\text{m/s}$ for V_1

$$\dot{m} = \rho A_1 V_1$$
$$= (2.21)(0.011)(45)$$
$$= 1.094\,\text{kg/s}$$

The mass flow rate of air $\dot{m}$ is $\boxed{1.094\,\text{kg/s}}$

Step 2 of 8

b) The energy balance equation per unit mass for a steady state process is

$$(q - w) = (h_2 - h_1) + \left(\frac{V_2^2 - V_1^2}{2}\right) + g(z_2 - z_1)$$

As the process is adiabatic, heat transfer is $q = 0$, the external work done on this process is $w = 0$ and Also the nozzle entrance and exit are at same level. So $z_1 = z_2$

Enthalpy $h = C_p T$

$$0 = C_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$
$$(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2C_p}$$
$$T_2 = T_1 - \left(\frac{V_2^2 - V_1^2}{2C_p}\right)$$

Here, the exit Velocity of air is V_2 and Specific heat of air is C_p.

Substitute $1.02\,\text{kJ/kg.K}$ for C_p and $180\,\text{m/s}$ for V_2.

$$T_2 = T_1 - \left(\frac{V_2^2 - V_1^2}{2C_p}\right)$$
$$= 200 - \left(\frac{180^2 - 45^2}{2\times 1.02\times 10^3}\right)$$
$$= 200 - (14.89)$$
$$= 185.11°\text{C}$$

The exit temperature of the air T_2 is $\boxed{185.11°\text{C}}$

Step 3 of 8

c) Ideal gas equation, $P = \rho RT$

$$\rho = \frac{P}{RT}$$
$$\rho_2 = \frac{P_2}{RT_2}$$

Here, the exit Pressure of air is P_2, the exit Temperature of air is T_2 and the gas constant for air R.

Substitute $100\ \text{kPa}$ for P_2 . $185.11°\text{C} + 273 = 458.11\text{K}$ for T_2.

$$\rho_2 = \frac{P_2}{RT_2}$$
$$\rho_2 = \frac{100}{(0.287)(185.11+273)}$$
$$\rho_2 = 0.762\ \text{kg/m}^3$$

Step 4 of 8

Mass flow rate,

$$\dot{m} = \rho AV$$

Mass flow rate remains constant throughout the nozzle,

Mass flow rate,

$$\dot{m} = \rho A V$$

Mass flow rate remains constant throughout the nozzle,

$$\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

$$A_2 = \frac{\dot{m}}{\rho_2 V_2}$$

$$= \frac{1.1}{0.762 \times 180}$$

$$= 80 \times 10^{-4}\ \text{m}^2$$

$$= 80\ \text{cm}^2$$

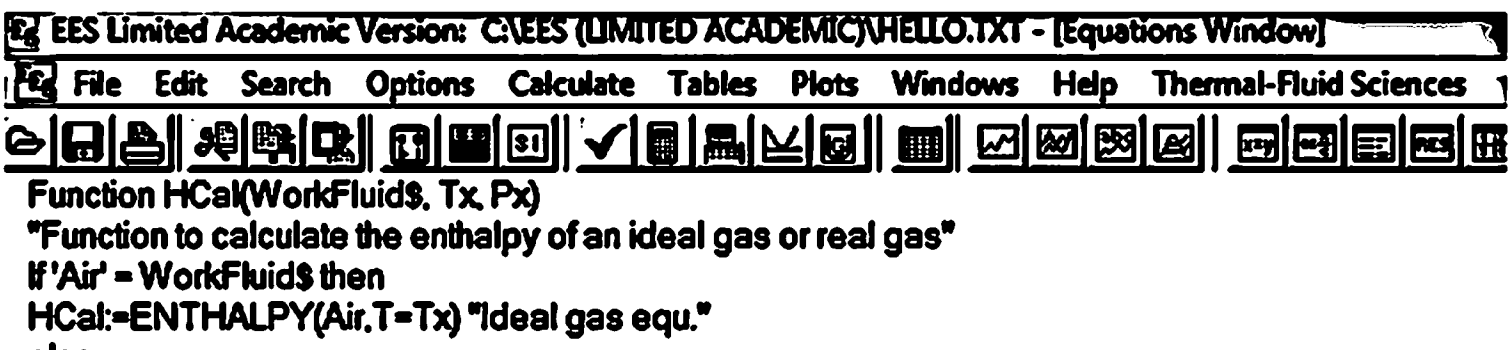
The exit area of the nozzle A_2 is $\boxed{80\ \text{cm}^2}$

The given problem can be solved using EES software by following the below steps.

1. Open EES software

2. Create a new file by clicking on the 'New' of the 'File menu'

3. Now type the following code in dailog box



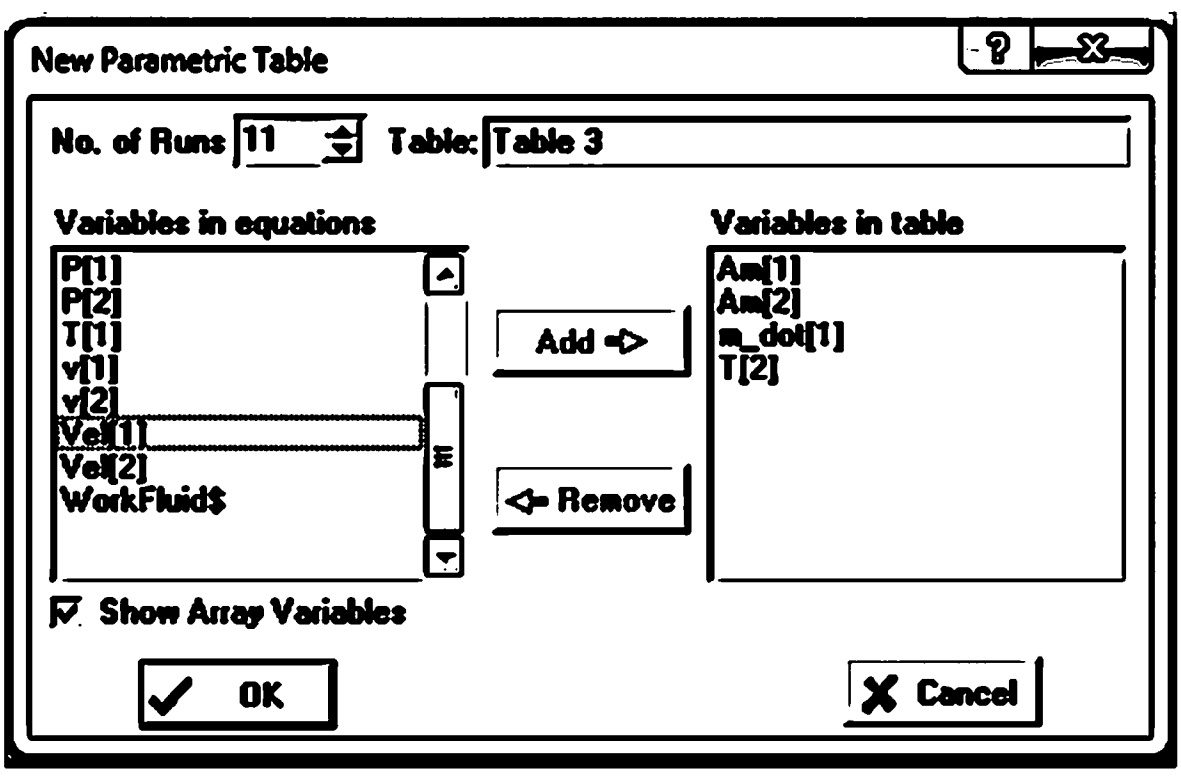
Step 5 of 8

Now to solve the equation for a range of Area values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.

A dailog box will open.

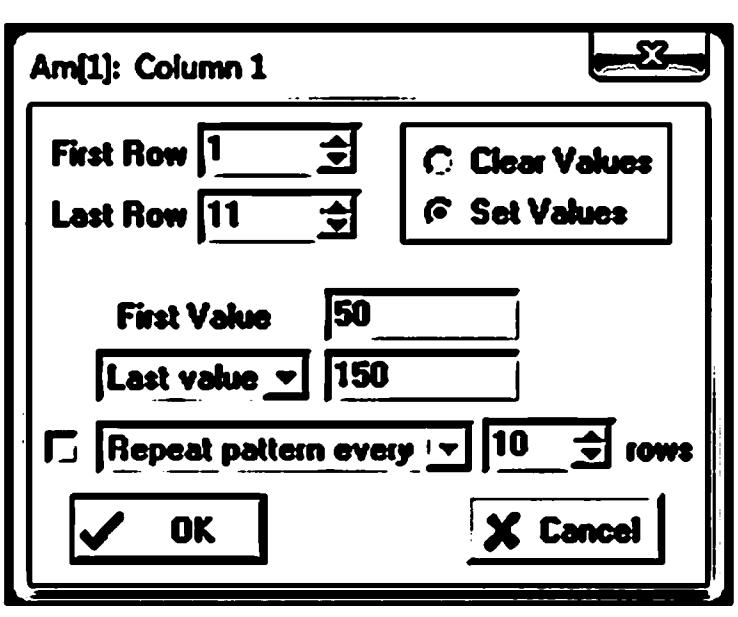
Select no of runs = 11.

Select variables in equations as "Am[1]" "Am[2]" "m_doT[1]"and "T[2]" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 6 of 8

To enter the values in the "Am[1]" column click on the drop tab in "Am[1]" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Now click on 'Solve table' of 'Calculate' menu.

The Solution will appear as shown below:

1..11	1 A_1	2 A_2	3 $\dot{m}_1$	4 T_2 [C]
Run 1	50	36.32	0.497	185.2
Run 2	60	43.59	0.5964	185.2
Run 3	70	50.85	0.6958	185.2
Run 4	80	58.12	0.7952	185.2
Run 5	90	65.38	0.8946	185.2
Run 6	100	72.65	0.9941	185.2
Run 7	110	79.91	1.093	185.2
Run 8	120	87.18	1.193	185.2

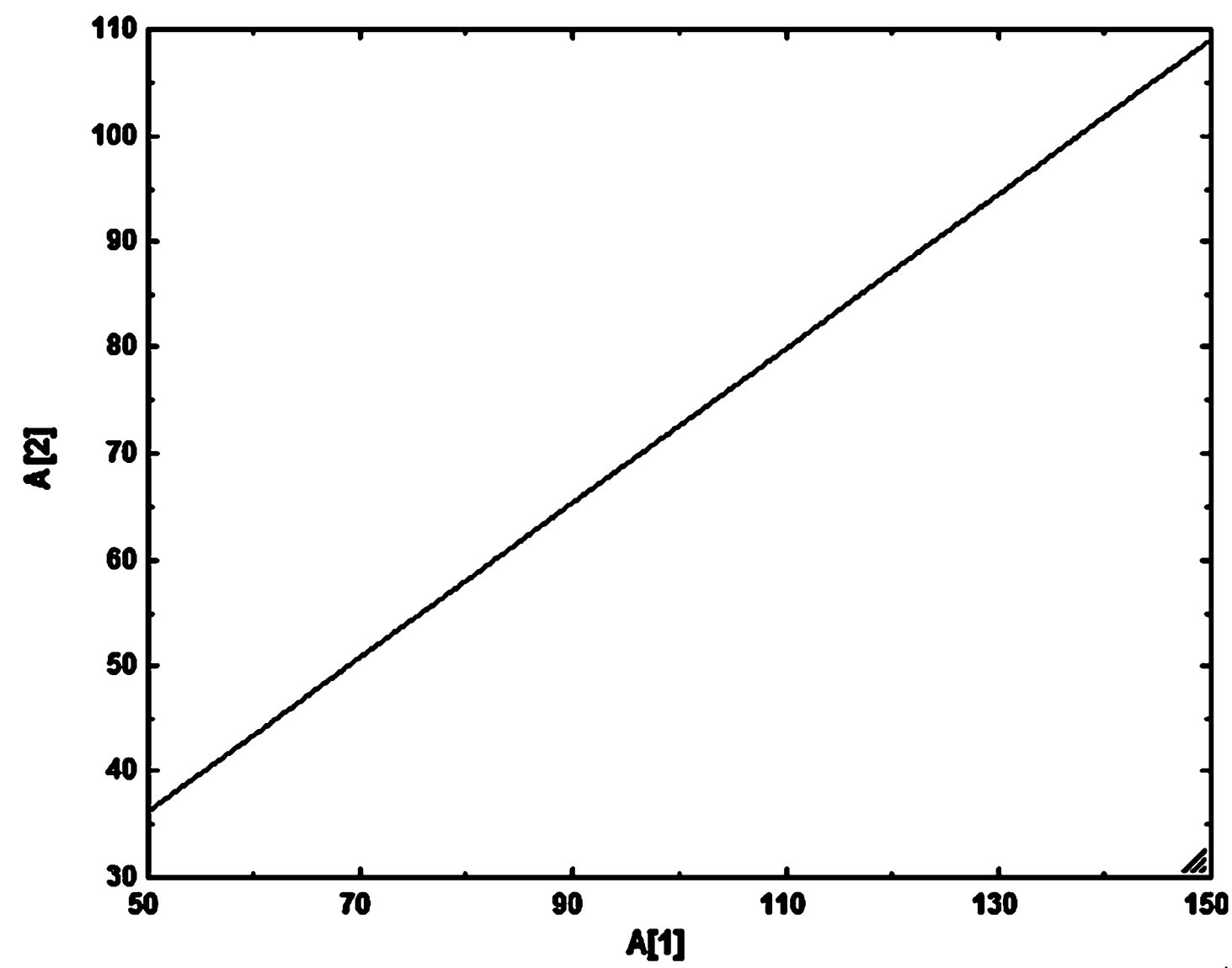
Run 4	80	58.12	0.7952	185.2
Run 5	90	65.38	0.8946	185.2
Run 6	100	72.65	0.9941	185.2
Run 7	110	79.91	1.093	185.2
Run 8	120	87.18	1.193	185.2
Run 9	130	94.44	1.292	185.2
Run 10	140	101.7	1.392	185.2
Run 11	150	109	1.491	185.2

Step 7 of 8

Inorder to plot the area A1 versus the area A2, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as "A[1]" and Y-axis as "A[2]" and press 'OK'

The Graph will be plotted as shown below:

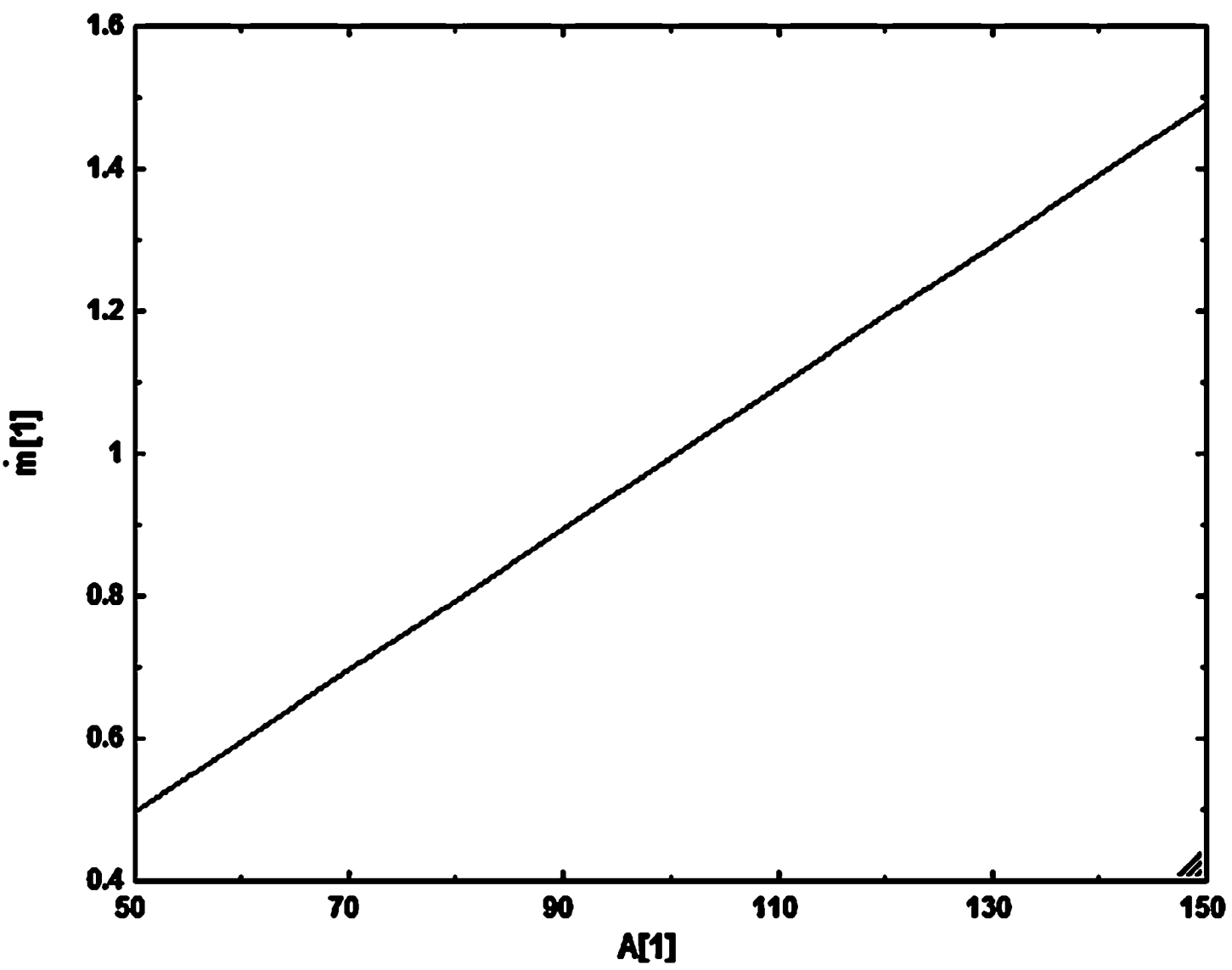


Step 8 of 8

Inorder to plot the area A1 versus the Mass flow rate $\dot{m}[1]$, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as "A[1]" and Y-axis as "$\dot{m}[1]$" and press 'OK'

The Graph will be plotted as shown below:



Question: 6.030P

The stators in a gas turbine are designed to increase the kinetic energy of the gas passing through them adiabatically. Air enters a set of these nozzles at 300 psia and 700°F with a velocity of 80 ft/s and exits at 250 psia and 645°F. Calculate the velocity at the exit of the nozzles.

Step 1 of 4

Given

Inlet pressure $P_1 = 300$ psia

Inlet temperature $T_1 = 700°F = 1160^0R$

Inlet velocity $V_1 = 80$ ft/s

Exit pressure $P_2 = 250$ psia

Exit temperature $T_2 = 645°F = 1105^0R$

Step 2 of 4

The energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here $q = w = 0$ and $z_1 = z_2$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \text{ ---------- (1)}$$

Step 3 of 4

At $T_1 = 700°F = 1160^0R$ from ideal gas properties of air

$h_1 = 281.14$ Btu/lbm

At $T_2 = 645°F = 1105^0R$ from ideal gas properties of air

$$h_2 = 260.97 + (271.03 - 260.97)\left(\frac{1105 - 1080}{1120 - 1080}\right)$$

$h_2 = 267.2575$ Btu/lbm

Step 4 of 4

Then from equation (1)

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (267.2575 - 281.14) \times 25047 + \frac{V_2^2 - 80^2}{2}$$

$V_2^2 = 701829.955 ft^2/s^2$

$\boxed{V_2 = 837.752 ft/s}$

Question: 6.031P

The diffuser in a jet engine is designed to decrease the kinetic energy of the air entering the engine compressor without any work or heat interactions. Calculate the velocity at the exit of a diffuser when air at 100 kPa and 20°C enters it with a velocity of 500 m/s and the exit state is 200 kPa and 90°C.

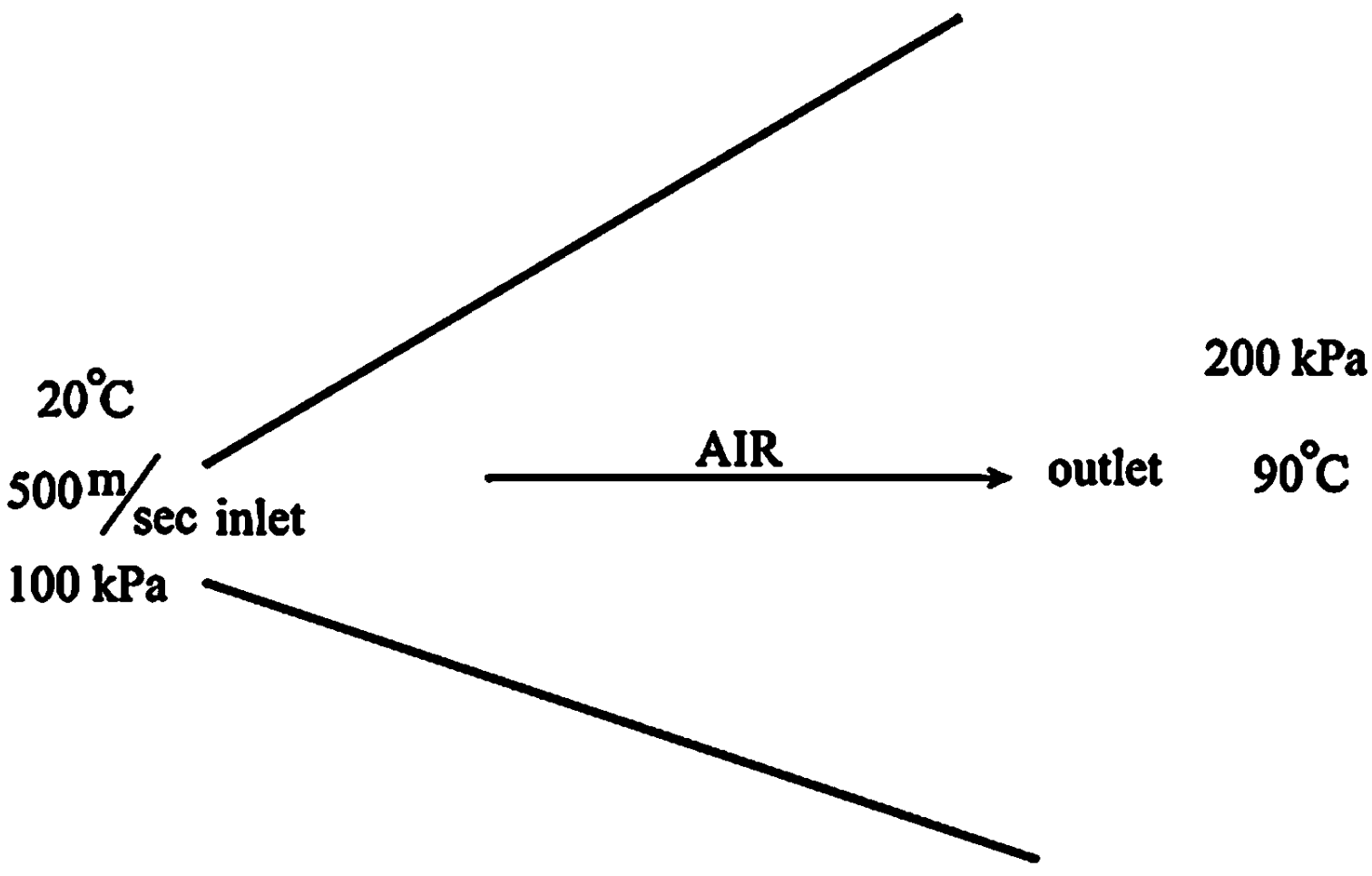


FIGURE P6–31

Step 1 of 6

Sketch the system as follows.

Consider the diffuser as the control volume system, as the mass crosses the boundary.



Step 2 of 6

The energy balance equation per unit mass for a steady state process is below.

Rate of net energy transfer by heat, work and mass is equal to the rate of change of internal kinetic, potential energies.

Therefore, the energy balance equation per unit mass for a steady state process is as follows.

$$\dot{E}_{in}-\dot{E}_{out}=\frac{\Delta \dot{E}_{system}}{dt}=0$$

Since it is a steady state process, hence the above equation becomes as follows.

$$\dot{E}_{in}-\dot{E}_{out}=0$$
$$\dot{m}_1\left(h_1+\frac{V_1^2}{2}\right)=\dot{m}_2\left(h_2+\frac{V_2^2}{2}\right)$$

Step 3 of 6

Since there is only one entry and exit the mass flow rate at the inlet and the outlet is equal.

$$\dot{m}_1=\dot{m}_2=\dot{m}$$

Therefore, write the above equation as follows.

$$\left(h_1+\frac{V_1^2}{2}\right)=\left(h_2+\frac{V_2^2}{2}\right)$$

Rewrite the above equation for V_2 as follows.

$$V_2=\left[V_1^2+2(h_1-h_2)\right]^{0.5}$$

Since $h_1=h_2=c_p(T)_{1,2}$, V_2 becomes as follows.

$$V_2=\left[V_1^2+2c_p(T_1-T_2)\right]^{0.5} \quad \text{..... (1)}$$

Here, c_p is the specific heat capacity, V_1 is the initial velocity, T_1 and T_2 are the initial and final temperatures.

Step 4 of 6

Calculate the average temperatures.

The average temperature of the air is below.

$$T=\frac{T_1+T_2}{2}$$

Here, T_1 and T_2 are the initial and final temperatures.

Substitute $293\,\text{K}$ for T_1 and $363\,\text{K}$ for T_2.

$$T=\frac{293+363}{2}$$
$$=328\,\text{K}$$

Step 5 of 6

Calculate the value for the c_p.

Obtain the value for the value c_p at the average temperature of the air from the table A-2 of ideal gas specific heats of various common gases at various temperatures.

Calculate the specific heat capacity, for the temperature values at $T_2=350\,\text{K}$ and $T_1=300\,\text{K}$ by interpolating the specific heat capacities values at $c_{p2}=1.008$ and $c_{p1}=1.005$

$$(c_p)=c_{p,300\text{K}}+(c_{p,350\text{K}}-c_{p,350\text{K}})\left(\frac{328-300}{350-300}\right)$$
$$=1.007+(1.008-1.007)(0.56)$$
$$=1.007+5.6\times10^{-4}$$
$$=1.00756\ \text{kJ/kg}\cdot\text{k}$$

Step 6 of 6

Substitute $1.00756\ \text{kJ/kg}\cdot\text{k}$ for c_p, $500\ \text{m}^2/\text{sec}^2$ for V_1 and $293\,\text{K}$ for T_1 and $363\,\text{K}$ for T_2 in equation (1).

$$V_2=\left[V_1^2+2c_p(T_1-T_2)\right]^{0.5}$$
$$=\left[500^2+2\times1.00756(20-90)\,\text{K}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\right]^{0.5}$$
$$=330.06\ \text{m/s}$$

Therefore, velocity at the exit V_2 is $\boxed{330.06\ \text{m/s}}$.

Question: 6.032P

Steam enters a nozzle at 400°C and 800 kPa with a velocity of 10 m/s, and leaves at 300°C and 200 kPa while losing heat at a rate of 25 kW. For an inlet area of 800 cm2, determine the velocity and the volume flow rate of the steam at the nozzle exit.

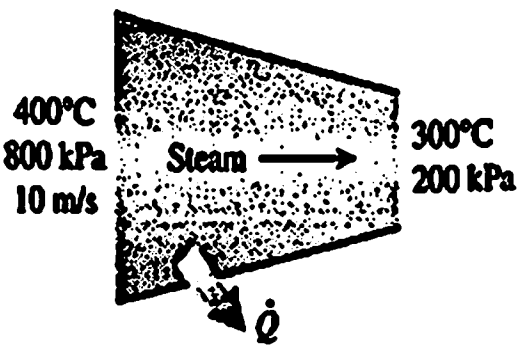


FIGURE P6–32

Step 1 of 5

Given

Inlet pressure $P_1 = 800$ kPa

Inlet temperature $T_1 = 400°C$

Inlet velocity $V_1 = 10$ m/s

Inlet cross-sectional area $A_1 = 800\ \text{cm}^2 = 0.08\ \text{m}^2$

Exit pressure $p_2 = 200$ kPa

Exit temperature $T_2 = 300°C$

The rate of heat loss $\dot{Q} = -25$ kW

Step 2 of 5

At $P_1 = 800$ kPa , $T_1 = 400°C$ from super heated steam tables

$v_1 = 0.38429\ \text{m}^3/\text{kg}$ And

$h_1 = 3267.7$ kJ/kg

At $P_2 = 200$ kPa , $T_2 = 300°C$

$v_2 = 1.31623\ \text{m}^3/\text{kg}$ and

$h_2 = 3072.1$ kJ/kg

Step 3 of 5

(a) The exit velocity

The energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here $w = 0$ and $z_1 = z_2$ and $q = \frac{\dot{Q}}{\dot{m}}$

The mass flow rate of steam can be obtained by using

$$\dot{m} = \frac{A_1 V_1}{v_1}$$

$$= \frac{(0.08)(10)}{0.38429}$$

$$\dot{m} = 2.08\ \text{kg/s}$$

Step 4 of 5

Therefore

$$-\frac{\dot{Q}}{\dot{m}} = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

$$-\frac{25\times10^3}{2.08} = (3072.1 - 3267.7)\times10^3 + \frac{V_2^2 - 10^2}{2}$$

$$-12.019\times10^3 = -195.6\times10^3 + \frac{V_2^2 - 10^2}{2}$$

$$V_2^2 = 367162 + 10^2$$

$$\boxed{V_2 = 606.02\ \text{m/s}}$$

Step 5 of 5

(b) The volume flow rate at the exit.

The volume flow rate is expressed as

$$\dot{V}_2 = \dot{m} v_2$$

$$= 2.08(1.31623)$$

$$\boxed{\dot{V} = 2.737\ \text{m}^3/\text{s}}$$

Question: 6.033P

Steam at 3 MPa and 400°C enters an adiabatic nozzle steadily with a velocity of 40 m/s and leaves at 2.5 MPa and 300 m/s. Determine (a) the exit temperature and (b) the ratio of the inlet to exit area *A1/A2*.

Step 1 of 6

Given

Inlet pressure $P_1 = 3\ \text{MPa}$

Inlet temperature $T_1 = 400°\text{C}$

Inlet velocity $V_1 = 40\ \text{m/s}$

Exit pressure $P_2 = 2.5\ \text{MPa}$

Exit velocity $V_2 = 300\ \text{m/s}$

Step 2 of 6

At $P_1 = 3\ \text{MPa}$, $T_1 = 400°\text{C}$ from superheated steam tables

$v_1 = 0.09938\ \text{m}^3/\text{kg}$

$h_1 = 3231.7\ \text{kJ/kg}$

Step 3 of 6

(a)

The exit temperature can be obtained by using energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here $q = w = 0$ and $z_1 = z_2$

Therefore

$$0 = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (h_2 - 3231.7) + \frac{300^2 - 40^2}{2}\left(\frac{1}{1000}\right)$$

$$h_2 = 3187.5\ \text{kJ/kg}$$

Step 4 of 6

At $P_2 = 2.5\ \text{MPa}$ and $h_2 = 3187.5\ \text{kJ/kg}$ from superheated steam tables by using interpolation

$$T_2 = 350 + (400 - 350)\left(\frac{3187.5 - 3127.0}{3240.1 - 3127.0}\right)$$

$$\boxed{T_2 = 376.746°\text{C}}$$

Step 5 of 6

(b) The ratio of the inlet to exit area A_1 / A_2

We know that mass flow rate remains constant

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\frac{A_1}{A_2} = \frac{V_2 v_1}{V_1 v_2}$$

At $P_2 = 2.5\ \text{MPa}$ and $T_2 = 376.746°\text{C}$ from superheated steam tables

$$v_2 = 0.10979 + (0.12012 - 0.10979)\left(\frac{376.746 - 350}{400 - 350}\right)$$

$$v_2 = 0.11531\ \text{m}^3/\text{kg}$$

Step 6 of 6

Then

$$\frac{A_1}{A_2} = \frac{V_2 v_1}{V_1 v_2}$$

$$\frac{A_1}{A_2} = \frac{300(0.09938)}{40(0.11531)}$$

$$\boxed{\frac{A_1}{A_2} = 6.463}$$

Question: 6.034P

Air at 13 psia and 50°F enters an adiabatic diffuser steadily with a velocity of 600 ft/s and leaves with a low velocity at a pressure of 14.5 psia. The exit area of the diffuser is 4 times the inlet area. Determine (a) the exit temperature and (b) the exit velocity of the air.

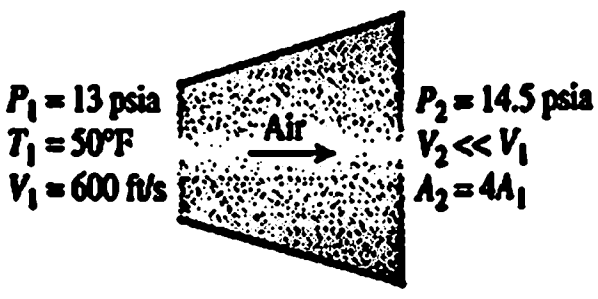


FIGURE P6–34E

Step 1 of 3

(a)

Apply energy balance equation per unit mass for a steady state diffuser

$$(q-w)=(h_2-h_1)+\left(\frac{V_2^2-V_1^2}{2}\right)+g(z_2-z_1)$$

Here, q is the amount of heat transfer, w is the work transfer; h_1, h_2 are the enthalpies at diffuser entrance and exit; V_1, V_2 are the velocities at diffuser entrance and exit; z_1, z_2 are the datum heads at diffuser entrance and exit and g is acceleration due to gravity.

Since Diffuser is adiabatic diffuser, there is no change in heat transfer $q=0$, there is no work transfer during the process $w=0$

Also the diffuser entrance and exit are at same level, so $z_1 = z_2$

Substitute all those known values in energy equation,

$$(q-w)=(h_2-h_1)+\left(\frac{V_2^2-V_1^2}{2}\right)+g(z_2-z_1)$$

$$0-0=(h_2-h_1)+\left(\frac{V_2^2-V_1^2}{2}\right)+g(0)$$

$$0=(h_2-h_1)+\left(\frac{V_2^2-V_1^2}{2}\right)$$

$$h_2=h_1-\left(\frac{V_2^2-V_1^2}{2}\right)$$

At the diffuser exit, velocity is very much less than the velocity at diffuser inlet. So neglect the exit velocity then the energy equation becomes,

$$h_2=h_1+\frac{V_1^2}{2} \qquad \text{..... (1)}$$

Step 2 of 3

Obtain enthalpy at diffuser inlet temperature $T_1 = 50°F = 510°R$ from "ideal gas properties of air table" by using interpolation,

$$h_1=119.48+(124.27-119.48)\left(\frac{510-500}{520-500}\right)$$

$$=121.875\,\text{Btu/lbm}$$

Substitute 600 ft/s for V_1 in equation (1)

$$h_2=h_1+\frac{V_1^2}{2}$$

$$=(121.88\ \text{Btu/lbm})+\frac{600^2}{2}\text{ft}^2/\text{s}^2\times\frac{1\,\text{Btu/lbm}}{25047\,\text{ft}^2/\text{s}^2}$$

$$=121.88+7.186$$

$$=129.066\,\text{Btu/lbm}$$

Obtain the exit temperature from table "ideal gas properties of air" at exit enthalpy $h_2 = 129.066$ Btu/lbm

$$T_2=540\text{R}$$

Therefore, the exit temperature of diffuser is $\boxed{540\text{R}}$

Step 3 of 3

(b)

Find the exit velocity of air by using the mass flow rate.

Mass flow rate remains constant throughout the diffuser,

$$\dot{m}=\frac{A_1V_1}{v_1}=\frac{A_2V_2}{v_2}$$

Here, A_1, A_2 is the cross sectional areas at inlet and exit of the diffuser V_1, V_2 are the velocities at inlet and exit of the diffuser and v_1, v_2 are the specific volumes at state 1 and state 2.

Obtain exit velocity from the mass flow rate,

$$V_2=\frac{A_1}{A_2}\frac{v_2}{v_1}V_1$$

Substitute $\frac{RT}{P}$ for v by using ideal gas equation, $Pv=RT$

$$V_2=\frac{A_1}{A_2}\times\frac{\frac{RT_2}{P_2}}{\frac{RT_1}{P_1}}\times V_1$$

$$=\frac{A_1}{A_2}\times\frac{P_1T_2}{P_2T_1}V_1$$

Substitute $4A_1$ for A_2, 14.5 psia for P_2 and 13 psia for P_1

$$V_2=\frac{A_1}{4\times A_1}\frac{(540\text{R})\times(13\,\text{psia})}{(50+460\text{R})\times(14.5\,\text{psia})}\times 600\,\text{ft/s}$$

$$=\frac{1}{4}\times\frac{7020}{7395}\times 600\,\text{ft/s}$$

$$=142.39\,\text{ft/s}$$

Therefore, the exit velocity of the air is $\boxed{142.39\,\text{ft/s}}$

Question: 6.035P

Carbon dioxide enters an adiabatic nozzle steadily at 1 MPa and 500°C with a mass flow rate of 6000 kg/h and leaves at 100 kPa and 450 m/s. The inlet area of the nozzle is 40 cm2. Determine (a) the inlet velocity and (b) the exit temperature.

Step 1 of 6

Given

Inlet pressure $P_1 = 1\ \text{MPa}$

Inlet temperature $T_1 = 500°\text{C}$

Inlet cross sectional area $A_1 = 40\ \text{cm}^2 = 0.004\ \text{m}^2$

Exit pressure $P_2 = 100\ \text{kPa}$

Exit velocity $V_2 = 450\ \text{m/s}$

Mass flow rate of carbon dioxide $\dot{m} = 6000\ \text{kg/h}$

$$= \frac{6000}{60 \times 60}$$

$$\dot{m} = 1.67\ \text{kg/s}$$

Step 2 of 6

From the table molar mass, gas constant and critical point properties

The gas constant of Carbon dioxide is $R = 0.1889\ \text{kJ/kg.K}$

Molar mass $M = 44\ \text{kg/kmol}$

Step 3 of 6

(a) The inlet velocity

The mass flow rate is expressed as

$$\dot{m} = \rho_1 A_1 V_1$$

From ideal gas equation

$$\rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{1000}{(0.1889)(500+273)}$$

$$\rho_1 = 6.848\ \text{kg/m}^3$$

Step 4 of 6

Therefore the inlet velocity is

$$V_1 = \frac{\dot{m}}{\rho_1 A_1}$$

$$= \frac{1.67}{6.848 \times 0.004}$$

$$\boxed{V_1 = 60.96\ \text{m/s}}$$

Step 5 of 6

(b) The exit temperature

The energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here $q = w = 0$ and $z_1 = z_2$ (Since adiabatic and no boundary work)

And $h = c_p T$

Step 6 of 6

Therefore

$$0 = c_p (T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

From ideal gas specific heats of common gases at $T_1 = 500^0\text{C}$

$c_p = 1.156\ \text{kJ/kg.K}$

Then

$$0 = 1.156(T_2 - 500) + \frac{450^2 - 60.96^2}{2}\left(\frac{1}{1000}\right)$$

$$T_2 - 500 = -85.979$$

$$T_2 = 414.021^0\text{C}$$

$$T_2 = 687.021\text{K}$$

The exit temperature $\boxed{T_2 = 687.021\text{K}}$

Question: 6.036P

Refrigerant-134a at 700 kPa and 120°C enters an adiabatic nozzle steadily with a velocity of 20 m/s and leaves at 400 kPa and 30°C. Determine (a) the exit velocity and (b) the ratio of the inlet to exit area A1/A2.

Step 1 of 4

Given

Inlet pressure $P_1 = 700$ kPa

Inlet temperature $T_1 = 120°C$

Inlet velocity $V_1 = 20$ m/s

Exit pressure $P_2 = 400$ kPa

Exit temperature $T_2 = 30°C$

Step 2 of 4

At $P_1 = 700$ kPa , $T_1 = 120°C$ from superheated R-134a tables

$v_1 = 0.043358\ m^3/kg$ and

$h_1 = 358.9$ kJ/kg

At $P_2 = 400$ kPa , $T_2 = 30°C$ from super heated R-134a tables

$v_2 = 0.056796\ m^3/kg$ and

$h_2 = 275.07$ kJ/kg

Step 3 of 4

(a)

The exit velocity can be obtained by applying energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

$q = w = 0$ and $z_1 = z_2$ Since the process is adiabatic and there is no boundary work

$$0 = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (275.07 - 358.9) + \frac{V_2^2 - 20^2}{2}\left(\frac{1}{1000}\right)$$

$$0 = -83.83 + \frac{V_2^2 - 20^2}{2}\left(\frac{1}{1000}\right)$$

$$V_2^2 = 167660 + 20^2$$

$$\boxed{V_2 = 409.951\ m/s}$$

Step 4 of 4

(b)

The ratio of inlet to exit area

We know that mass flow rate remains constant throughout the nozzle

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\frac{A_1}{A_2} = \frac{V_2 v_1}{V_1 v_2}$$

$$\frac{A_1}{A_2} = \frac{409.951 \times 0.043358}{20 \times 0.056796}$$

$$\boxed{\frac{A_1}{A_2} = 15.647}$$

Question: 6.037P

Refrigerant-134a enters a diffuser steadily as saturated vapor at 800 kPa with a velocity of 120 m/s, and it leaves at 900 kPa and 40°C. The refrigerant is gaining heat at a rate of 2 kJ/s as it passes through the diffuser. If the exit area is 80 percent greater than the inlet area, determine (a) the exit velocity and (b) the mass flow rate of the refrigerant.

Step 1 of 4

Given

Inlet pressure $P_1 = 800$ kPa

Inlet velocity $V_1 = 120$ m/s

Exit pressure $P_2 = 900$ kPa

Exit temperature $T_2 = 40°C$

The rate of heat gain $\dot{Q} = 2$ kJ/s

Exit area is 80% greater than the inlet area $A_2 = A_1 + 80\% A_1$

$$A_2 = 1.8 A_1$$

Step 2 of 4

From the refrigerant R-134a tables

At $P_1 = 800$ kPa

$v_1 = v_g = 0.025621 \text{ m}^3/\text{kg}$ and

$h_1 = h_g = 267.29$ kJ/kg

At $P_2 = 900$ kPa and $T_2 = 40°C$ from super heated R-134a table

$v_2 = 0.023375 \text{ m}^3/\text{kg}$ and

$h_2 = 274.17$ kJ/kg

Step 3 of 4

(a)

The exit velocity

We know that mass flow rate remains constant throughout the diffuser

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$V_2 = \frac{A_1}{A_2}\frac{v_2}{v_1} V_1$$

$$= \frac{1(0.023375)}{1.8(0.025621)} 120$$

$$\boxed{V_2 = 60.8 \text{ m/s}}$$

Step 4 of 4

(b) The mass flow rate of the refrigerant

The energy balance equation per unit mass for a steady state process is

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here $w = 0$ and $z_1 = z_2$ and $q = \frac{\dot{Q}}{\dot{m}}$

$$\frac{\dot{Q}}{\dot{m}} = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}$$

$$\frac{2}{\dot{m}} = (274.17 - 267.29) + \frac{60.8^2 - 120^2}{2}\left(\frac{1}{1000}\right)$$

$$\frac{2}{\dot{m}} = 6.88 + (-5.35168)$$

$$\dot{m} = \frac{2}{1.528}$$

$$\boxed{\dot{m} = 1.308 \text{ kg/s}}$$

Question: 6.038P

Steam at 4 MPa and 400°C enters a nozzle steadily with a velocity of 60 m/s, and it leaves at 2 MPa and 300°C. The inlet area of the nozzle is 50 cm2, and heat is being lost at a rate of 75 kJ/s. Determine (a) the mass flow rate of the steam, (b) the exit velocity of the steam, and (*c*) the exit area of the nozzle.

Step 1 of 6

Given

Inlet Pressure of steam, $P_1 = 4\ \text{MPa}$

Inlet Temperature of steam, $T_1 = 400°\text{C}$

Inlet Velocity of steam, $V_1 = 60\ \text{m/s}$

Area of inlet, $A_1 = 50\ \text{cm}^2$

$$A_1 = 0.005\ \text{m}^2$$

Exit Pressure of steam, $P_2 = 2\ \text{MPa}$

Exit Temperature of steam, $T_2 = 300°\text{C}$

The rate of heat loss, $\dot{Q} = 75\ \text{kJ/s}$

Step 2 of 6

From the superheated steam tables, at 4 MPa and 400°C

Specific volume of steam, $v_1 = 0.07343\ \text{m}^3/\text{kg}$

Enthalpy of steam, $h_1 = 3214.5\ \text{kJ/kg}$

From the superheated steam tables, At 2 MPa and 300°C

Specific volume of steam, $v_2 = 0.12551\ \text{m}^3/\text{kg}$

Enthalpy of steam, $h_2 = 3024.2\ \text{kJ/kg}$

Step 3 of 6

(a)

The mass flow rate, $\dot{m} = \dfrac{A_1 V_1}{v_1}$

$$\dot{m} = \frac{(0.005)(60)}{0.07343}$$

$$\dot{m} = 4.085\ \text{kg/s}$$

The mass flow rate of the steam, $\boxed{\dot{m} = 4.085\ \text{kg/s}}$

Step 4 of 6

(b)

From the energy balance equation per unit mass for a steady state process,

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

Here boundary work is $w = 0$

Entrance and exit are at same levels, $z_1 = z_2$

Heat transfer per unit mass, $q = \dfrac{\dot{Q}}{\dot{m}}$

And Enthalpy, $h = C_p T$

Step 5 of 6

Hence energy balance equation becomes,

$$-\frac{\dot{Q}}{\dot{m}} = (h_2 - h_1) + \left(\frac{V_2^2 - V_1^2}{2}\right)$$

$$\left[-\frac{75\times10^3}{4.085}\right] = \left[(3024.2 - 3214.5)\times10^3\right] + \left[\frac{V_2^2 - 60^2}{2}\right]$$

$$-18359.85 = -190300 + \left[\frac{V_2^2 - 60^2}{2}\right]$$

$$\frac{V_2^2 - 60^2}{2} = 171940.15$$

$$V_2^2 = 343880.3 + 60^2$$

$$V_2 = 589.5\ \text{m/s}$$

The exit velocity $\boxed{V_2 = 589.5\ \text{m/s}}$

Step 6 of 6

(c)

Mass flow rate, $\dot{m} = \dfrac{A_2 V_2}{v_2}$

Area at exit, $A_2 = \dfrac{\dot{m} \times v_2}{V_2}$

$$A_2 = \frac{4.0855\times0.12551}{589.5}$$

$$A_2 = 8.7\times10^{-4}\ \text{m}^2$$

The exit area of the nozzle, $\boxed{A_2 = 8.7\ \text{cm}^2}$

Question: 6.039P

Consider an adiabatic turbine operating steadily. Does the work output of the turbine have to be equal to the decrease in the energy of the steam flowing through it?

Step 1 of 1

Yes, the work output of the turbine has to be equal to the decrease in the energy of the steam flowing thought it. During the adiabatic process heat transfer is zero

From the energy equation

$$W = \dot{m}\left(h_1 + \frac{V_1^2}{2} + z_1\right) - \dot{m}\left(h_2 + \frac{V_2^2}{2} + z_2\right)$$

Question: 6.040P

Consider an air compressor operating steadily. How would you compare the volume flow rates of the air at the compressor inlet and exit?

Step 1 of 1

Compressor is a device which increases the pressure of the fluid at the cost of energy input. As pressure increases volume decreases, so at the inlet the volume flow rate is more compared to outlet as specific volume decreases along the compressor.

Question: 6.041P

Will the temperature of air rise as it is compressed by an adiabatic compressor? Why?

Step 1 of 1

Yes, the temperature and pressure increases during an adiabatic compression process.
Since energy is added to the working fluid which raises the temperature.

Question: 6.042P

Somebody proposes the following system to cool a house in the summer: Compress the regular outdoor air, let it cool back to the outdoor temperature, pass it through a turbine, and discharge the cold air leaving the turbine into the house. From a thermodynamic point of view, is the proposed system sound?

Step 1 of 1

No, since compressing process involves the raise of temperature of the air.

Question: 6.043P

Refrigerant-134a enters a compressor at 100 kPa and −24°C with a flow rate of 1.35m3/min and leaves at 800 kPa and 60°C. Determine the mass flow rate of R-134a and the power input to the compressor.

Step 1 of 4

Given

Inlet Pressure, $P_1 = 100\ \text{kPa}$

Inlet Temperature, $T_1 = -24°\text{C}$

Volume flow rate, $\dot{V}_1 = 1.35\ \text{m}^3/\text{min}$,

Exit Pressure, $P_2 = 800\ \text{kPa}$

Exit Temperature, $T_2 = 60°\text{C}$

Step 2 of 4

From the property table of refrigerant R-134a table, at 100 kPa and −24°C

Specific volume, $v_1 = 0.1947\ \text{m}^3/\text{kg}$

Enthalpy, $h_1 = 236.34\ \text{kJ/kg}$

From the property table of refrigerant R-134a table, at 800 kPa and 60°C

Enthalpy $h_2 = 296.81\ \text{kJ/kg}$

Step 3 of 4

The mass flow rate, $\dot{m} = \dfrac{\dot{V}_1}{v_1}$

$$\dot{m} = \frac{(1.35/60)}{0.1947}$$

$$\dot{m} = 0.1155\ \text{kg/s}$$

The mass flow rate of the refrigerator, $\boxed{\dot{m} = 0.1155\ \text{kg/s}}$

Step 4 of 4

Work input, $\dot{W}_{in} = \dot{m}(h_2 - h_1)$

$$\dot{W}_{in} = (0.1155)(296.81 - 236.34)$$

$$\dot{W}_{in} = 6.9843\ \text{kW}$$

The power output $\boxed{\dot{W}_{in} = 6.9843\ \text{kW}}$

Question: 6.044P

Refrigerant-134a enters a compressor at 180 kPa as a saturated vapor with a flow rate of 0.35 m3/min and leaves at 700 kPa. The power supplied to the refrigerant during compression process is 2.35 kW. What is the temperature of R-134a at the exit of the compressor?

Step 1 of 4

Given

Inlet Pressure $P_1 = 180\ \text{kPa}$

Volume flow rate, $\dot{V}_1 = 0.35\ \text{m}^3/\text{min}$

Exit Pressure, $P_2 = 700\ \text{kPa}$

The power supplied to the refrigerator $\dot{W}_{in} = 2.35\ \text{kW}$

Step 2 of 4

The mass flow rate, $\dot{m} = \dfrac{\dot{V}_1}{v_1}$

$$\dot{m} = \frac{(0.35/60)}{0.1104}$$

$$\dot{m} = 0.0528\ \text{kg/s}$$

The mass flow rate of the refrigerator, $\dot{m} = 0.0528\ \text{kg/s}$

Step 3 of 4

From the property table of refrigerant R-134a table at 180 kPa, saturated vapor has

Specific volume, $v_1 = 0.1104\ \text{m}^3/\text{kg}$

Enthalpy, $h_1 = 242.86\ \text{kJ/kg}$

Work input, $\dot{W}_{in} = \dot{m}(h_2 - h_1)$

Enthalpy at exit, $h_2 = h_1 + \dfrac{\dot{W}_{in}}{\dot{m}}$

$$h_2 = 242.86 + \frac{2.35}{0.0528}$$

$$h_2 = 287.4\ \text{kJ/kg}$$

Step 4 of 4

From the property table of refrigerant R-134a table at 700 kPa

$h = 278.57\ \text{kJ/kg}$, $T = 40^\circ\text{C}$

$h = 288.53\ \text{kJ/kg}$, $T = 50^\circ\text{C}$

At $h_2 = 287.4\ \text{kJ/kg}$, $P_2 = 700\ \text{kPa}$ By interpolation

$$\frac{T_2 - 40}{50 - 40} = \frac{287.4 - 278.57}{288.53 - 278.57}$$

$$\frac{T_2 - 40}{10} = 0.8865$$

$$\frac{T_2 - 40}{10} = 0.8865$$

$$T_2 = 48.86\ ^0\text{C}$$

So at $h_2 = 287.4\ \text{kJ/kg}$, $T_2 = 48.86\ ^0\text{C}$

Therefore exit temperature is $\boxed{T_2 = 48.86\ ^0\text{C}}$

Question: 6.045P

Steam enters an adiabatic turbine at 10 MPa and 500°C and leaves at 10 kPa with a quality of 90 percent. Neglecting the changes in kinetic and potential energies, determine the mass flow rate required for a power output of 5 MW.

Step 1 of 4

Given

Inlet pressure $P_1 = 10$ MPa

Inlet temperature $T_1 = 500°C$

Outlet pressure $P_2 = 10$ kPa

Outlet velocity $V_2 = 50$ m/s

Exit quality $x_2 = 0.90$

Power output $\dot{W} = 5$ MW

Step 2 of 4

At $P_1 = 10$ MPa and $T_1 = 500°C$ from super heated steam tables

$h_1 = 3375.1$ kJ/kg

At $P_2 = 10$ kPa from saturated water tables

$h_f = 191.81$kJ/kg

$h_{fg} = 2392.1$kJ/kg

$h_2 = h_f + x_2 h_{fg}$

$h_2 = 191.81 + 0.9 \times 2392.1$

$h_2 = 2344.7$kJ/kg

Step 3 of 4

The energy balance equation for a steady state process is

$$\dot{Q} - \dot{W} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right)$$

Since the process is adiabatic $q = 0$

And neglecting the kinetic and potential energies $ke \approx 0, pe \approx 0$

Step 4 of 4

Therefore

$$0 - \dot{W} = \dot{m}(h_2 - h_1) + 0 + 0$$

$$-5000 = \dot{m}(2344.7 - 3375.1)$$

$$\dot{m} = -4.852 \text{kg/s}$$

The mass flow rate is

$$\boxed{\dot{m} = 4.852 \text{ kg/s}}$$

Question: 6.046P

Steam flows steadily through a turbine at a rate of 45,000 lbm/h, entering at 1000 psia and 900°F and leaving at 5 psia as saturated vapor. If the power generated by the turbine is 4 MW, determine the rate of heat loss from the steam.

Step 1 of 3

Given

Inlet pressure $P_1 = 1000$ psia

Inlet temperature $T_1 = 900°F$

Outlet pressure as saturated vapor $P_2 = 5$ psia

Power output $\dot{W} = 4$ MW $\quad (1\ \text{Btu/s} = 1055\ \text{J/s})$

$$\dot{W} = \frac{12\times10^9}{1055}\ \text{Btu/s}$$

Mass flow rate of the steam $\dot{m} = 45000$ lbm/h

$$= \frac{45000}{3600}$$

$$\dot{m} = 12.5\ \text{lbm/s}$$

Step 2 of 3

At $P_1 = 1000$ psia, $T_1 = 900°F$ from superheated steam tables

$h_1 = 1448.6$ Btu/lbm

At $P_2 = 5$ psia from saturated water tables

$h_2 = h_g = 1130.7$ Btu/lbm

Step 3 of 3

The energy balance equation for a steady state process is

$$\dot{Q} - \dot{W} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right)$$

Assuming there is no change in kinetic and potential energy changes

$$\dot{Q} - \dot{W} = \dot{m}(h_2 - h_1)$$

$$\dot{Q} - \frac{4\times10^6}{1055} = 12.5(1130.7 - 1448.6)$$

$$\dot{Q} - 3791.469 = -3973.75$$

$$\boxed{\dot{Q} = -182.281\ \text{Btu/s}} \text{ or}$$

$$\dot{Q} = -182.281\times1055$$

$$\boxed{\dot{Q} = -192.306\ \text{MW}}$$

Negative sign indicates heat lost to the surroundings

Question: 6.047P

An adiabatic air compressor compresses 10 L/s of air at 120 kPa and 20°C to 1000 kPa and 300°C. Determine (a) the work required by the compressor, in kJ/kg, and (b) the power required to drive the air compressor, in kW.

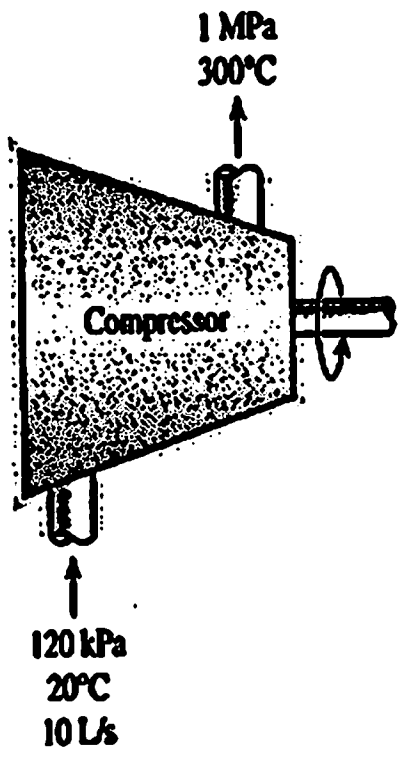


FIGURE P6–47

Step 1 of 4

Hence, the steady state equation for compressor is reduced to,

$$\dot{W} = \dot{m}(h_2 - h_1)$$

$$w_{in} = c_p(T_2 - T_1) \text{ ..... (1)}$$

Note the value for specific heat which is taken at average temperature of the air,

$$T_{avg} = \frac{T_1 + T_2}{2}$$

Here, temperature at inlet is T_1 and temperature at outlet is T_2.

Substitute, $20°C$ for T_1 and $300°C$ for T_2 in relation above and calculate average temperature.

$$T_{avg} = \frac{20 + 300}{2}$$
$$= 160^0C$$
$$= 433K$$

Step 2 of 4

Note the value for specific heat at $433K$ from ideal gas specific heats of common gases

$$\frac{c_p - 1.013}{(1.02 - 1.013)} = \left(\frac{433 - 400}{450 - 400}\right)$$
$$c_p = 1.01762 kJ/kg \cdot K$$

Substitute, $1.01762 kJ/kg \cdot K$ for c_p, $300°C$ for T_2 and $20°C$ for T_1 in equation (1) and calculate the work required by the compressor.

$$w_{in} = 1.01762(300 - 20)$$
$$= 284.933 kJ/kg$$

Therefore the work input to the compressor (w_{in}) is $\boxed{284.933 kJ/kg}$.

Step 3 of 4

(b)

Calculate the power required to drive the air compressor using relation.

$$\dot{W} = \dot{m} w$$

Where the mass flow rate is expressed as

$$\dot{m} = \rho_1 \dot{V}_1$$

From ideal gas equation, calculate the density using relation.

$$\rho_1 = \frac{P_1}{RT_1}$$

Here, density of air is ρ_1, pressure at inlet in Pascal's is P_1, gas constant is R and temperature at inlet in kelvin is T_1.

Substitute, (120×10^3) Pa for P_1, $287 J/kg \cdot K$ for R and $293 K$ for T_1 in relation above and calculate ρ_1.

$$\rho_1 = \frac{120 \times 10^3}{(0.287 \times 10^3)(20 + 273)}$$
$$= 1.427 kg/m^3$$

Step 4 of 4

Calculate the power required to drive the air compressor using relation,

$$\dot{W} = \rho_1 \dot{V}_1 w_{in} \text{ ..... (2)}$$

Here, volumetric flow rate of air is $\dot{V}_1$.

Substitute, $1.427 kg/m^3$ for ρ_1, $284.933 kJ/kg$ for w_{in} and $(10 \times 10^{-3}) m^3/s$ for $\dot{V}_1$ in equation (2) and calculate the power required.

$$\dot{W} = 1.427 \times 0.01 \times 284.933$$
$$= 4.0659 kW$$

Therefore the power required to drive the air compressor $(\dot{W})$ is $\boxed{4.066 kW}$.

Question: 6.048P

Argon gas enters an adiabatic turbine steadily at 1600 kPa and 450°C with a velocity of 55 m/s and leaves at 150 kPa with a velocity of 150 m/s. The inlet area of the turbine is 60 cm2. If the power output of the turbine is 190 kW, determine the exit temperature of the argon.

Step 1 of 5

Given

Inlet Pressure, $P_1 = 1600\ \text{kPa}$

Inlet Temperature, $T_1 = 450°C$

Inlet Velocity, $V_1 = 55\ \text{m/s}$

Inlet Area, $A_1 = 60\ \text{cm}^2$

$$A_1 = 0.006\ \text{m}^2$$

Exit Pressure, $P_2 = 150\ \text{kPa}$

Exit Velocity, $V_2 = 150\ \text{m/s}$

Power output, $\dot{W} = 190\ \text{kW}$

Step 2 of 5

From ideal gas equation, $P = \rho RT$

$$\rho = \frac{P}{RT}$$

$$\rho_1 = \frac{P_1}{RT_1}$$

From ideal gas property table of Argon,

Gas constant, $R = 0.2081\ \text{kJ/kg.K}$

Specific heat, $C_p = 0.5203\ \text{kJ/kg.K}$

Step 3 of 5

Hence density, $\rho_1 = \dfrac{1600}{(0.2081)(450+273)}$

$$\rho_1 = 10.63\ \text{kg/m}^3$$

Therefore mass flow rate, $\dot{m} = \rho_1 A_1 V_1$

$$\dot{m} = (10.63)(0.006)(55)\ \text{kg/s}$$

$$\dot{m} = 3.51\ \text{kg/s}$$

Step 4 of 5

From the energy balance equation for a steady state process,

$$\dot{Q} - \dot{W} = \dot{m}\left[(h_2 - h_1) + \left(\frac{V_2^2 - V_1^2}{2}\right) + g(z_2 - z_1)\right]$$

As the process is adiabatic, $\dot{Q} = 0$

Also potential energy change, $pe \approx 0$ and

Enthalpy, $h = C_p T$

Step 5 of 5

Therefore $0 - \dot{W} = \dot{m}\left[C_p(T_2 - T_1) + \left(\dfrac{V_2^2 - V_1^2}{2}\right)\left(\dfrac{1}{1000}\right) + 0\right]$

$$-190 = 3.51\left[0.5203(T_2 - 450) + \left(\frac{150^2 - 55^2}{2000}\right)\right]$$

$$(T_2 - 450) = \frac{-54.13 - 9.7375}{0.5203}$$

$$T_2 = 327.24\ ^0\text{C}$$

The exit temperature, $\boxed{T_2 = 327.24\ ^0\text{C}}$

Question: 6.049P

Helium is to be compressed from 120 kPa and 310 K to 700 kPa and 430 K. A heat loss of 20 kJ/kg occurs during the compression process. Neglecting kinetic energy changes, determine the power input required for a mass flow rate of 90 kg/min.

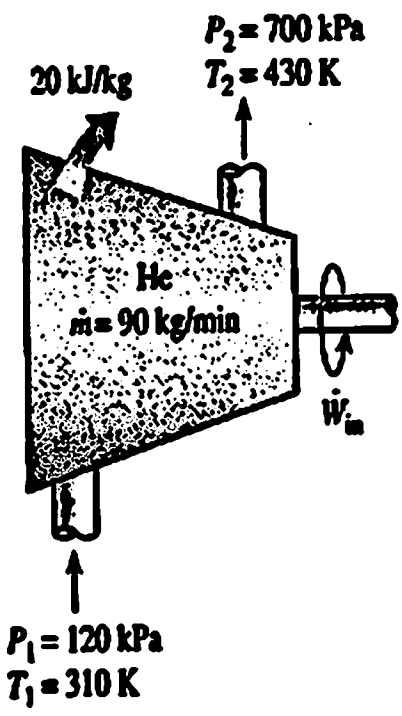


FIGURE P6–49

Step 1 of 5

Given

Working substance is Helium

Inlet pressure $P_1 = 120$ kPa

Inlet temperature $T_1 = 310$K

Mass flow rate $\dot{m} = 90$ kg/min

$$= \frac{90}{60}$$

$$\dot{m} = 1.5 \text{ kg/s}$$

Final pressure $P_2 = 700$ kPa

Final temperature $T_2 = 430$K

Heat loss $q = -20$ kJ/kg

Step 2 of 5

From the table ideal gas specific heats of various common gases

The gas constant for Helium is $R = 2.0769$ kJ/kg.K

Specific heat $c_p = 5.1926$ kJ/kg.K

Step 3 of 5

The energy balance equation for a steady state process is

$$q - w = \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \right)$$

Assuming there is no change in kinetic and potential energy changes $ke \approx 0, pe \approx 0$

and $h = c_p T$

Step 4 of 5

Therefore

$$q - w = \left(c_p (T_2 - T_1) + 0 + 0 \right)$$

$$q - w = \left(c_p (T_2 - T_1) \right)$$

$$\left(-20 \times 10^3 \right) - w = \left(5.1926 \times 10^3 (430 - 310) \right)$$

$$\left(-20 \times 10^3 \right) - w = 623.112 \times 10^3$$

$$w = -643112 \text{ J/kg}$$

Step 5 of 5

Power input to the compressor

$$\dot{W} = \dot{m} w$$

$$= 1.5 \times (-643112)$$

$$\boxed{\dot{W} = -964.668 \text{ kW}}$$

Negative sign indicates power supplied to compressor

Question: 6.050P

Carbon dioxide enters an adiabatic compressor at 100 kPa and 300 K at a rate of 0.5 kg/s and leaves at 600 kPa and 450 K. Neglecting kinetic energy changes, determine (a) the volume flow rate of the carbon dioxide at the compressor inlet and (b) the power input to the compressor.

Step 1 of 4

Obtain the gas constant for carbon dioxide at 300 K from the table A-2, "ideal gas specific heats of common gases" from the textbook.

The gas constant for Carbon dioxide gas, $R = 0.1889\ \text{kJ/kg.K}$.

Calculate the average temperature.

$$T_{av} = \frac{T_1 + T_2}{2}$$

Here, T_1 is the inlet temperature, and T_2 is the outlet temperature.

Substitute 300 K for T_1 , and 450 K for T_2 .

$$T_{av} = \frac{300 + 450}{2}$$
$$= 375\,\text{K}$$

Step 2 of 4

Obtain the specific heat at constant pressure for carbon dioxide at 350 K, and 400 K from the table A-2, "ideal gas specific heats of common gases" from the textbook.

At 350 K, $c_p = 0.895\ \text{kJ/kg} \cdot \text{K}$

At 400 K, $c_p = 0.939\ \text{kJ/kg} \cdot \text{K}$

Calculate the Specific heat at $T_{av} = 375\,\text{K}$ using interpolation.

$$c_p = 0.895 + (0.939 - 0.895)\left(\frac{375 - 350}{400 - 350}\right)$$
$$c_p = 0.917\ \text{kJ/kg.K}$$

Step 3 of 4

(a)

Calculate the volume flow rate using the following expression.

$$\dot{V} = \frac{\dot{m}}{\rho} \quad \text{..... (1)}$$

Here, $\dot{m}$ is the mas flow rate, and ρ is the density.

From ideal gas equation,

$$\rho_1 = \frac{P_1}{RT_1}$$

Substitute 300 K for T_1 , $0.1889\ \text{kJ/kg.K}$ for R, and 100 kPa for P_1 .

$$\rho_1 = \frac{100}{(0.1889)(300)}$$
$$= 1.765\ \text{kg/m}^3$$

Substitute $1.765\ \text{kg/m}^3$ for ρ , and 0.5 kg/s for $\dot{m}$ in equation 1.

$$\dot{V}_1 = \frac{\dot{m}}{\rho_1}$$
$$= \frac{0.5}{1.765}$$
$$= 0.283\ \text{m}^3/\text{s}$$

Therefore volume flow rate is $\boxed{\dot{V}_1 = 0.283\ \text{m}^3/\text{s}}$.

Step 4 of 4

(b)

Calculate the power input to the compressor by applying the energy balance equation for a steady state process.

$$\dot{Q} - \dot{W} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right)$$

Since the compressor is adiabatic, $\dot{Q} = 0$.

As there is no change in kinetic and potential energies $ke \approx 0, pe \approx 0$

Also, $h = c_p T$

Then the equation simplifies to,

$$0 - \dot{W}_{in} = \dot{m}\left(c_p(T_2 - T_1) + 0 + 0\right)$$

Substitute $0.917\ \text{kJ/kg.K}$ for c_p , 300 K for T_1 , 0.5 kg/s for $\dot{m}$, and $450\ \text{K}$ for T_2

$$-\dot{W} = 0.5\left(0.917 \times 10^3 (450 - 300)\right)$$
$$W = -68.775 \times 10^3$$

Negative sign indicates power input to compressor

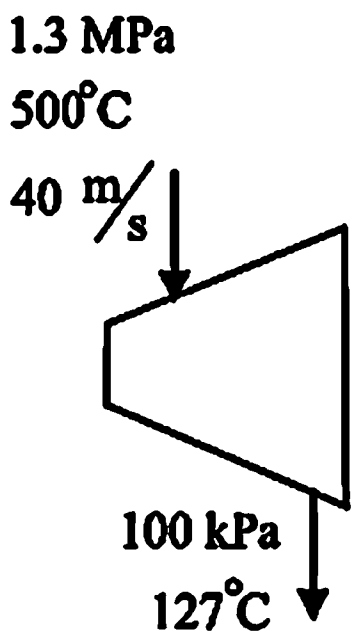
Therefore, the power input required for the compressor is $\boxed{68.775\ \text{kW}}$.

Question: 6.051P pg 1

An adiabatic gas turbine expands air at 1300 kPa and 500°C to 100 kPa and 127°C. Air enters the turbine through a 0.2-m2 opening with an average velocity of 40 m/s, and exhausts through a 1-m2 opening. Determine (a) the mass flow rate of air through the turbine and (b) the power produced by the turbine.

Step 1 of 9

Sketch the turbine as follows.
Consider the turbine as the control volume system, as the mass crosses the boundary.



Step 2 of 9

The energy balance equation per unit mass for a steady state process is below.
Rate of net energy transfer by heat, work and mass is equal to the rate of change of internal kinetic, potential energies.
Therefore, the energy balance equation per unit mass for a steady state process is as follows.

$$\dot{E}_{in}-\dot{E}_{out}=\frac{\Delta \dot{E}_{system}}{dt}=0$$

Since and there is only one entry and exit the mass flow rate at the inlet and the outlet is equal.

$$\dot{m}_1=\dot{m}_2=\dot{m}$$

It is a steady state process, hence the above equation becomes as follows.

$$\dot{E}_{in}-\dot{E}_{out}=0$$

$$\dot{m}\left(h_1+\frac{V_1^2}{2}\right)=\dot{m}\left(h_2+\frac{V_2^2}{2}\right)+\dot{W}_{out}$$

$$\dot{W}_{out}=\dot{m}\left(h_1-h_2+\frac{V_1^2-V_2^2}{2}\right)$$

Step 3 of 9

Since $h_1=h_2=c_p\ T_{1,2}$, the above equation becomes as follows.

$$\dot{W}_{out}=\dot{m}\left(c_p\ T_1-T_2+\frac{V_1^2-V_2^2}{2}\right) \quad \text{..... (1)}$$

Here, c_p is the specific heat capacity, V_1 is the initial velocity, T_1 and T_2 are the initial and final temperatures, $\dot{m}$ is the mass flow rate and $\dot{W}_{out}$ is the power output produced by the turbine.

Calculate the average temperatures.
The average temperature of the air is below.

$$T=\frac{T_1+T_2}{2}$$

Here, T_1 and T_2 are the initial and final temperatures.
Substitute 500°C for T_1 and 127°C for T_2.

$$T=\frac{500+127}{2}$$
$$=314°C$$
$$=527\ K$$

Step 4 of 9

Calculate the value for the c_p.
Obtain the value for the value c_p at the average temperature of the air from the table A-2 of ideal gas specific heats of various common gases at various temperatures.
Calculate the specific heat capacity, for the temperature values at $T_2=600\ K$ and $T_1=550\ K$ by interpolating the specific heat capacities values at $c_{p1}-1.040\ kJ/kg\cdot k$ and $c_{p2}=1.051\ kJ/kg\cdot k$.

$$(c_p)=c_{p,550K}+(c_{p,600K}-c_{p,550K})\left(\frac{587-550}{600-550}\right)$$
$$=1.040+(1.051-1.040)(0.74)$$
$$=1.040+8.14\times10^{-3}$$
$$=1.04814\ kJ/kg\text{-}k$$

Step 5 of 9

Calculate the density at point 1.
From the ideal gas property table of air the value for the universal gas constant R is 0.287 kJ/kg.K.
The ideal gas equation is below.

$$P=\rho RT$$

Rewrite the above equation for density at the inlet.

$$\rho_1=\frac{P_1}{RT_1}$$

Here, P_1 is the pressure at the inlet, R is the universal gas constant; T_1 is the temperature at inlet and ρ_1 is the density of the air at inlet.

Step 6 of 9

Substitute the value 1300 kPa for P_1, 0.287 kJ/kg.K for R and 773 K for T_1 in the above equation.

$$\rho_1=\frac{1300}{(0.287)(500+273)}\ kg/m^3$$

Substitute the value 1300 kPa for P_1, 0.287 kJ/kg.K for R and 773 K for T_1 in the above equation.

$$\rho_1 = \frac{1300}{(0.287)(500+273)} \text{ kg/m}^3$$
$$= 5.86 \text{ kg/m}^3$$

a)

Calculate the mass flow rate.

$$\dot{m} = \rho_1 A_1 V_1$$

Here, ρ_1 is the density of the air at inlet, A_1 is the area at inlet and V_1 is the velocity at inlet.

Substitute 0.2m^2 for A_1, 5.86 kg/m^3 for ρ_1 and 40 m/s for V_1.

$$\dot{m} = (5.86)(0.2)(40)$$
$$= 46.88 \text{ kg/s}$$

Therefore, the mass flow rate $\dot{m}$ is $\boxed{46.88 \text{ kg/s}}$.

b)

Calculate the density at the outlet.

$$\rho_2 = \frac{P_2}{RT_2}$$

Here, P_2 is the pressure at the outlet, R is the universal gas constant; T_2 is the temperature at outlet and ρ_2 is the density of the air at outlet.

Substitute 100 kPa for P_2, 0.287 kJ/kg.K for R and 400 K for T_2.

$$\rho_2 = \frac{100}{(0.287)(400)}$$
$$= 0.871 \text{ kg/m}^3$$

Calculate the exit velocity.

The formula for the mass flow rate is below.

$$\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

Rewrite the above equation for exit velocity V_2.

Substitute 46.88 kg/s for $\dot{m}$, 0.871 kg/m^3 for ρ_2 and 1m^2 for A_2.

$$V_2 = \frac{\dot{m}}{\rho_2 A_2}$$
$$= \frac{46.88}{0.871 \times (1)}$$
$$= 53.82 \text{ m/s}$$

Substitute 53.82 m/s for V_2, 40 m/s for V_1, 400 K for T_2, 773 K for T_1 and 1.04814 kJ/kg·k for c_p and 46.88 kg/s for $\dot{m}$ in equation (1).

$$\dot{W}_{out} = 46.88 \text{ kg/s} \left(1.04814 \text{kJ/kg} \cdot \text{k} \ 500 - 127 + \frac{40^2 - 53.82^2}{2} \left(\frac{1 \text{kJ/kg}}{1000 \text{m}^2/\text{s}^2} \right) \right)$$
$$= 18295.187 \text{ kW}$$

Therefore, work output from the turbine is $\boxed{18295.187 \text{ kW}}$

Question: 6.052P

Air is expanded by an adiabatic gas turbine from 500 psia and 800°F to 60 psia and 250°F. If the volume flow rate at the exit is 50 ft3/s; the inlet area is 0.6 ft2; and the outlet area is 1.2 ft2; determine the power produced by this turbine.

Step 1 of 5

From the table molar mass, gas constant and critical point properties,

The gas constant for air $R = 0.3704\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$

Find the density of air at the exit by using ideal gas equation

$$\rho_2 = \frac{P_2}{RT_2}$$

Here, P_2 is exit pressure, T_2 is exit temperature of air.

Substitute $60\,\text{psia}$ for P_2 and $(250+460)°\text{R}$ for T_2

$$\rho_2 = \frac{60}{(0.3704)(250+460)}$$
$$= 0.228\,\text{lbm/ft}^3$$

Step 2 of 5

Find the mass flow rate of air by using the equation,

$$\dot{m} = \rho_2 \dot{V}_2$$

Here, $\dot{V}_2$ is volume flow rate at the exit and $\dot{m}$ is mass flow rate of air

Substitute $50\,\text{ft}^3/\text{s}$ for $\dot{V}_2$

$$\dot{m} = (0.228)(50)$$
$$= 11.4\,\text{lbm/s}$$

Find the velocity of air at exit by using the equation,

$$\dot{m} = \rho_2 A_2 V_2$$

Here, A_2 is exit area

$$V_2 = \frac{\dot{m}}{\rho_2 A_2}$$

Substitute $1.2\,\text{ft}^2$ for A_2

$$V_2 = \frac{11.4}{0.228 \times 1.2}$$
$$= 41.67\,\text{ft/s}$$

Step 3 of 5

Find the inlet density of air by using ideal gas equation,

$$\rho_1 = \frac{P_1}{RT_1}$$

Here, P_1 is inlet pressure and T_1 is inlet temperature.

Substitute $500\,\text{psia}$ for P_1 and $(800+460)°\text{R}$ for T_1

$$\rho_1 = \frac{500}{(0.3704)(800+460)}$$
$$= 1.0713\,\text{lbm/ft}^3$$

Step 4 of 5

Find the inlet velocity by using the mass flow rate equation,

$$V_1 = \frac{\dot{m}}{\rho_1 A_1}$$

Here, A_1 is inlet area

Substitute $0.6\,\text{ft}^2$ for A_1

$$V_1 = \frac{11.4}{1.07 \times 0.6}$$
$$= 17.75\,\text{ft/s}$$

Step 5 of 5

Take the properties of air average temperature of inlet and exit temperatures of air,

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{800+250}{2}$$
$$= 525°\text{F}$$

Take the specific heat of air at temperature $T_{avg} = 525°\text{F}$, from ideal gas properties of air,

$$c_p = 0.2472 + (0.2503 - 0.2472)\left(\frac{525-500}{600-500}\right)$$
$$= 0.2479\,\text{Btu/lbm} \cdot \text{R}$$

Calculate the power produced by the turbine by using the energy balance equation for a steady state process:

$$\dot{Q} - \dot{W} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right)$$

Since the process is adiabatic, heat transfer $\dot{Q} = 0$

And assuming there is no change in potential energy, then $g(z_2 - z_1) = 0$

Then the above equation becomes,

$$\dot{Q} - \dot{W} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right)$$
$$0 - \dot{W} = \dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} + 0\right)$$
$$-\dot{W} = \dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$

Substitute all the known values in energy equation,

$$-\dot{W} = 11.41\left(0.2479 \times (250-800) + \frac{41.67^2 - 17.75^2}{2} \times \frac{1}{25047}\right)$$

(Since $1\,\text{Btu/lbm} = 25047\,\text{ft}^2/\text{sec}^2$)

$$-\dot{W} = 11.41(-136.345 + 0.02837)$$
$$\dot{W} = 1555.372\,\text{Btu/s}$$

Therefore, the power produced by the turbine is **1555.372 Btu/s**

Question: 6.053P

Steam flows steadily into a turbine with a mass flow rate of 26 kg/s and a negligible velocity at 6 MPa and 600°C. The steam leaves the turbine at 0.5 MPa and 200°C with a velocity of 180 m/s. The rate of work done by the steam in the turbine is measured to be 20 MW. If the elevation change between the turbine inlet and exit is negligible, determine the rate of heat transfer associated with this process.

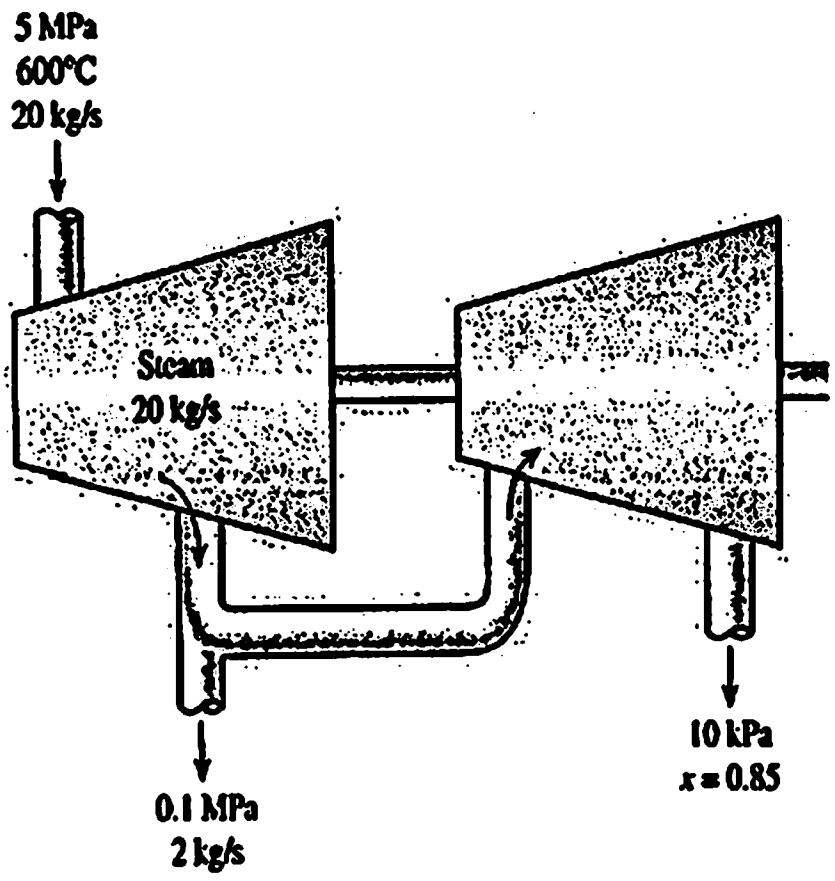


FIGURE P6–53

Step 1 of 2

Obtain properties of steam at inlet temperature, $T_1 = 600°C$ and inlet pressure, $p_1 = 6\,MPa$ from table A-6, "Properties of superheated water tables" in the text book.

Specific enthalpy, $h_1 = 3658.8\ kJ/kg$

Similarly, obtain properties of steam at exit temperature, $T_2 = 200°C$ and exit pressure, $p_2 = 0.5\,MPa$ from table A-6, "Properties of superheated water tables" in the text book.

Specific enthalpy, $h_2 = 2855.8\ kJ/kg$

Apply energy balance equation for a steady state flow process.

$$\dot{Q} + \dot{m}h_1 + \dot{m}\frac{V_1^2}{2} + \dot{m}gz_1 = \dot{m}h_2 + \dot{m}\frac{V_2^2}{2} + \dot{m}gz_2 + \dot{W}$$

$$\dot{Q} = \dot{W} + \dot{m}(h_2 - h_1) + \dot{m}\left(\frac{V_2^2 - V_1^2}{2}\right) + \dot{m}g(z_2 - z_1)$$

Here, $\dot{m}$ is mass flow rate of steam, $\dot{Q}$ is rate of heat transfer, $\dot{W}$ is rate of work transfer, V_1 is velocity of steam at inlet, V_2 is velocity of steam at exit, z_1 is datum head at turbine inlet, z_2 is datum head at turbine exit, and g is acceleration due to gravity.

Since there is no change in datum head, difference at inlet and exit, $(z_2 - z_1) = 0$

Hence, the energy balance equation becomes,

$$\dot{Q} = \dot{W} + \dot{m}(h_2 - h_1) + \dot{m}\left(\frac{V_2^2 - V_1^2}{2}\right) + 0$$

$$\dot{Q} = \dot{W} + \dot{m}(h_2 - h_1) + \dot{m}\left(\frac{V_2^2 - V_1^2}{2}\right) \quad \text{..... (1)}$$

Step 2 of 2

Substitute $3658.8\ kJ/kg$ for h_1, $2855.8\ kJ/kg$ for h_2, 26 kg/s for $\dot{m}$, $(20\times10^3)kW$ for $\dot{W}$, 0 m/s for V_1, 180 m/s for V_2 in equation (1).

$$\dot{Q} = \dot{W} + \dot{m}(h_2 - h_1) + \dot{m}\left(\frac{V_2^2 - V_1^2}{2}\right)$$

$$= 20\times10^3 + 26\times(2855.8 - 3658.8) + 26\times\left(\frac{180^2 - 0^2}{2}\right)\times\frac{1}{1000}$$

$$= 20\times10^3 - 20878 + 421.2$$

$$= -456.8\,kW$$

Negative sign indicates heat rejected from the turbine.

Therefore, the amount of heat transfer during the process is $\boxed{456.8\,kW}$.

Question: 6.054P

Determine the power input for a compressor that compresses helium from 150 kPa and 20°C to 400 kPa and 200°C. Helium enters this compressor through a 0.1-m2 pipe at a velocity of 15 m/s. What is the increase in the flow power during this process?

Step 1 of 4

Given

Inlet pressure of Helium, $P_1 = 150\ \text{kPa}$

Inlet temperature of Helium, $T_1 = 20°\text{C}$

$T_1 = 293\text{K}$

Inlet velocity of Helium, $V_1 = 0.1\ \text{m}^2$

Inlet Area of Compressor, $A_1 = 0.1\ \text{m}^2$

Exit pressure of Helium, $P_2 = 400\ \text{kPa}$

Final temperature of Helium, $T_2 = 200°\text{C}$

$T_2 = 473\text{K}$

Step 2 of 4

From the property table of ideal gasses for Helium,

The gas constant, $R = 2.0769\ \text{kJ/kg.K}$

Specific heat, $C_p = 5.1926\ \text{kJ/kg.K}$

From ideal gas equation $P = \rho RT$

Hence $\rho_1 = \dfrac{P_1}{RT_1}$

$$\rho_1 = \frac{150}{(2.0769)(293)}$$

$$\rho_1 = 0.2465\ \text{kg/m}^3$$

Step 3 of 4

The mass flow rate $\dot{m} = \rho AV$

$$\dot{m} = (0.2465)(0.1)(15)$$

$$\dot{m} = 0.37\ \text{kg/s}$$

Work input for compressor, $\dot{W}_{in} = \dot{m}(h_2 - h_1)$

But Enthalpy, $h = C_p T$. Hence

$$\dot{W}_{in} = \dot{m}C_p(T_2 - T_1)$$

$$\dot{W}_{in} = 0.37 \times 5.1926 \times (200 - 20)$$

$$\dot{W}_{in} = 346\ \text{kW}$$

Hence power input to the compressor is $\boxed{\dot{W}_{in} = 346\ \text{kW}}$

Step 4 of 4

The increase in flow power,

$$\dot{W}_f = \dot{m}(P_2 v_2 - P_1 v_1)$$

$$\dot{W}_f = \dot{m}R(T_2 - T_1)$$

$$\dot{W}_f = (0.37)(2.0769)(200 - 20)$$

$$\dot{W}_f = 138.22\ \text{kW}$$

Increase in flow power is $\boxed{\dot{W}_f = 138.22\ \text{kW}}$

Question: 6.055P

Someone claims, based on temperature measurements, that the temperature of a fluid rises during a throttling process in a well-insulated valve with negligible friction, How do you evaluate this claim? Does this process violate thermodynamic laws?

Step 1 of 1

During throttling process, the temperature of the fluid can increase, decrease, or remain the same. Therefore, this claim is valid since no thermodynamic laws are violated

Question: 6.056P

Would you expect the temperature of air to drop as it undergoes a steady-flow throttling process? Explain.

Step 1 of 1

No, since the temperature of an ideal gas does not change during a throttling (h =constant) process because enthalpy is a function of temperature only

$h = h(T)$

Question: 6.057P

Would you expect the temperature of a liquid to change as it is throttled? Explain.

Step 1 of 1

If the fluid remains in the liquid phase, then there will not be any change in temperature of the liquid. But if some part of the liquid vaporizes during throttling, change in the temperature may take place.

Question: 6.058P

During a throttling process, the temperature of a fluid drops from 30 to −20°C. Can this process occur adiabatically?

Step 1 of 1

Yes, the process can occur adiabatically.

The throttling devices are very small and the flow through them is assumed as adiabatic, since there is neither sufficient time, nor a large enough area for an effective heat transfer. During throttling, a significant pressure drop occurs in the fluid, along with a large temperature drop. The magnitude of temperature drop during a throttling process is governed by a property called Joule-Thomson coefficient.

Joule-Thomson coefficient μ is defined as follows:

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h$$

Question: 6.059P

An adiabatic capillary tube is used in some refrigeration systems to drop the pressure of the refrigerant from the condenser level to the evaporator level. The R-134a enters the capillary tube as a saturated liquid at 50°C, and leaves at −20°C. Determine the quality of the refrigerant at the inlet of the evaporator.

Step 1 of 3

Given

Temperature of R-134a at inlet of capillary tube, $T_1 = 50°C$

Temperature of R-134a at exit of capillary tube, $T_2 = -20°C$

Step 2 of 3

From saturated R-134a tables, at 50°C,

Saturated liquid- Enthalpy, $h_1 = h_f = 123.49$ kJ/kg

From saturated R-134a tables, at −20°C,

Saturated liquid- Enthalpy, $h_f = 25.49$ kJ/kg

Enthalpy of evaporation, $h_{fg} = 212.91$ kJ/kg

Step 3 of 3

The process in adiabatic capillary tube is isenthalpic. So, $h_2 = h_1$

$h_2 = 123.49$ kJ/kg

The quality at the exit, $x_2 = \dfrac{h_2 - h_f}{h_{fg}}$

$$x_2 = \frac{123.49 - 25.49}{212.91}$$

$$x_2 = 0.46$$

The steam at exit of capillary tube enters evaporator. So

Quality of steam at inlet of evaporator is $\boxed{x_2 = 0.46}$

Question: 6.060P

Saturated liquid-vapor mixture of water, called wet steam, in a steam line at 2000 kPa is throttled to 100 kPa and 120°C. What is the quality in the steam line?

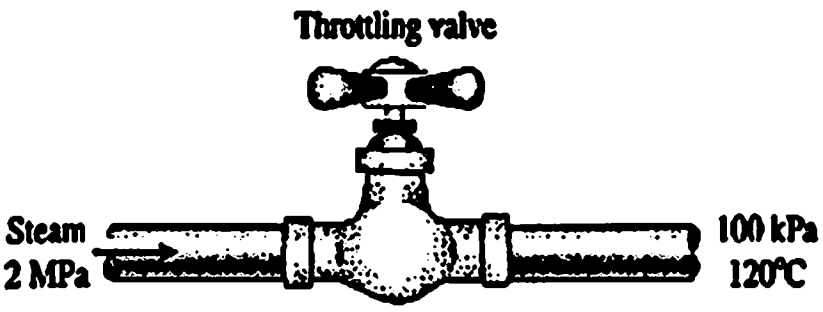


FIGURE P6–60

Step 1 of 4

Loading solution. Please wait ...

Step 2 of 4

Throttling valves are usually small devices, and the flow through them may be assumed to be adiabatic $(q \approx 0)$, there is no work done $(w \approx 0)$, and changes in kinetic and potential energies are insignificant $(ke \approx 0, pe \approx 0)$.

So the conservation of energy equation for this single steam steady flow devices reduces to $h_2 = h_1$ kJ/kg

Step 3 of 4

From superheated water tables

At $P_2 = 100$ kPa, $T_2 = 120°C$

$$h_2 = 2675.8 + (2776.6 - 2675.8)\left(\frac{120-100}{150-100}\right)$$

$h_2 = 2716.12$ kJ/kg

At $P_1 = 2000$ kPa from saturated water tables

$h_f = 908.47$ kJ/kg

$h_{fg} = 1889.8$ kJ/kg

Step 4 of 4

Since $h_2 = h_1$

$h_2 = h_1 = 2716.12$ kJ/kg

We know that $h_1 = h_f + x_1 h_{fg}$

Therefore the quality at inlet is

$$x_1 = \frac{h_1 - h_f}{h_{fg}}$$

$$x_1 = \frac{2716.12 - 908.47}{1889.8}$$

$$\boxed{x_1 = 0.9565}$$

Question: 6.061P

Refrigerant-134a at 800 kPa and 25°C is throttled to a temperature of −20°C. Determine the pressure and the internal energy of the refrigerant at the final state.

Step 1 of 6

Given

Inlet pressure $P_1 = 800$ kPa

Inlet temperature $T_1 = 25°C$

Exit temperature $T_2 = -20°C$

Step 2 of 6

Throttling valves are usually small devices, and the flow through them may be assumed to be adiabatic $(q \approx 0)$, there is no work done $(w \approx 0)$, and changes in kinetic and potential energies are insignificant $(ke \approx 0, pe \approx 0)$.

So the conservation of energy equation for this steady flow device reduces to

$h_2 = h_1$ kJ/kg

Step 3 of 6

At $P_1 = 800$ kPa and $T_1 = 25°C$ the state is in wet condition

So from saturated R-134a temperature table at $T_1 = 25°C$

$$h_1 = h_f = 84.98 + (87.83 - 84.98)\left(\frac{25-24}{26-24}\right)$$

$h_1 = h_f = 86.405$ kJ/kg

At $T_2 = -20°C$ from saturated R-134a tables

$h_f = 25.49$ kJ/kg

$h_{fg} = 212.91$ kJ/kg

$u_f = 25.39$ kJ/kg

$u_{fg} = 193.45$ kJ/kg

Step 4 of 6

The pressure at the exit will be the saturation pressure at $T_2 = -20°C$

So from saturated R-134a tables

$P_2 = P_{sat} = 132.82$ kPa

$\boxed{P_2 = 132.82 \text{ kPa}}$

Step 5 of 6

The exit quality can be obtained from

$h_2 = h_f + x_2 h_{fg}$ (Since $h_1 = h_2$)

$$x_2 = \frac{h_2 - h_f}{h_{fg}}$$

$$x_2 = \frac{86.405 - 25.49}{212.91}$$

$x_2 = 0.2861$

Step 6 of 6

The exit internal energy can be determined as

$u_2 = u_f + x_2 u_{fg}$

$u_2 = 25.39 + (0.2861)(193.45)$

$\boxed{u_2 = 80.736 \text{ kJ/kg}}$

Question:

6.062P

A well-insulated valve is used to throttle steam from 8 MPa and 350°C to 2 MPa. Determine the final temperature of the steam.

Step 1 of 1

In throttling process 1-2, enthalpy remains constant.

$$h_1 = h_2$$

Obtain the properties of steam at inlet pressure $P_1 = 8$ MPa and inlet temperature $T_1 = 350°C$ from superheated water tables.

Enthalpy, $h_1 = 2988.1$ kJ/kg

Since enthalpy remains constant in throttling process, enthalpy at state 2 is

$$h_2 = 2988.1 \text{ kJ/kg}$$

Now obtain the exit temperature of steam at final pressure $P_2 = 2$ MPa and enthalpy $h_2 = 2988.1$ kJ/kg from superheated water tables by using linear interpolation,

$$T_2 = 250 + (300 - 250)\left(\frac{2988.1 - 2903.3}{3024.2 - 2903.3}\right)$$

$$= 285.07°C$$

Hence, exit temperature of the steam is $\boxed{285.07°C}$

Question:

Reconsider Prob. 6–62. Using EES (or other) software, investigate the effect of the exit pressure of steam on the exit temperature after throttling. Let the exit pressure vary from 6 to 1 MPa. Plot the exit temperature of steam against the exit pressure, and discuss the results.

Step 1 of 7

State 1

From superheated steam tables,

The inlet pressure of steam at 8 MPa and $250°C$.

The Enthalpy is, $h_1 = 2988.1 kJ/kg$

State 2

From superheated steam tables,

The exit pressure of steam at 2 MPa,

The temperature at $T = 250°C$ the Enthalpy is , $h = 2903.3\ kJ/kg$.

The temperature at $T = 300°C$ the Enthalpy is , $h = 3024.2\ kJ/kg$.

Step 2 of 7

Throttling valves are devices wherein the process is isenthalpic. So, $h_2 = h_1$

$h_2 = 2988.1\ kJ/kg$ And $P_2 = 2\ MPa$

Use the method of interpolation,

$$\frac{T_2 - 250}{300 - 250} = \frac{2988.1 - 2903.3}{3024.2 - 2903.3}$$

$$T_2 = 250 + (0.7014 \times 50)$$

$$T_2 = 285.07°C$$

Hence temperature at exit T_2 is $\boxed{285.07°C}$

Step 3 of 7

The given problem can be solved using EES software by following the below steps.

1. Open EES software

2. Create a new file by clicking on the 'New' of the 'File menu'

3. Now type the following code in dailog box

Equations Window

```
WorkingFluid$='Steam_iapws' "WorkingFluid can be changed to ammonia or other fluids"
P_in=8000 [kPa]
T_in=350 [C]
"P_out=2000 "
"Analysis"
m_dot_in=m_dot_out "steady-state mass balance"
m_dot_in=1 "mass flow rate is arbitrary"
m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0 "steady-state energy balance"
Q_dot=0 "assume the throttle to operate adiabatically"
W_dot=0 "throttles do not have any means of producing power"
h_in=enthalpy(WorkingFluid$,T=T_in,P=P_in) "property table lookup"
T_out=temperature(WorkingFluid$,P=P_out,h=h_out) "property table lookup"
X_out=quality(WorkingFluid$,P=P_out,h=h_out) "x_out is the quality at the outlet"
P[1]=P_in; P[2]=P_out; h[1]=h_in; h[2]=h_out "use arrays to place points on property plot"
```

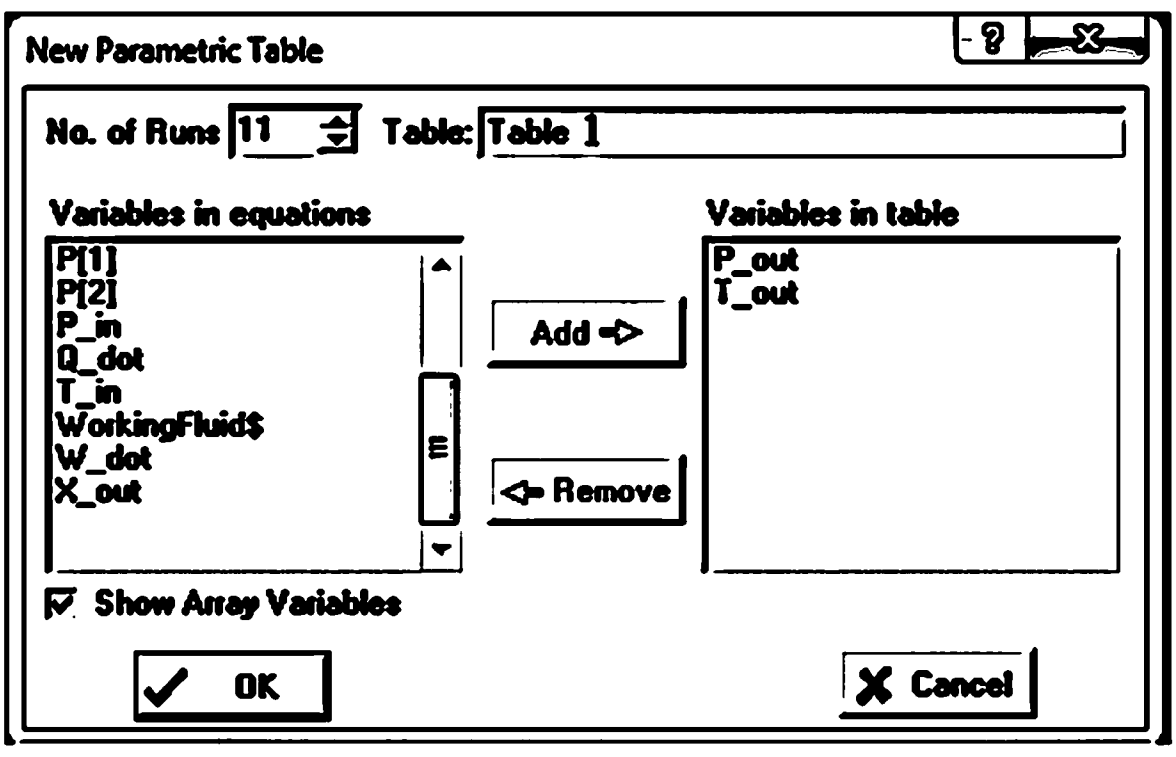
Step 4 of 7

Now to solve the equation for a range of pressure values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.

A dailog box will open.

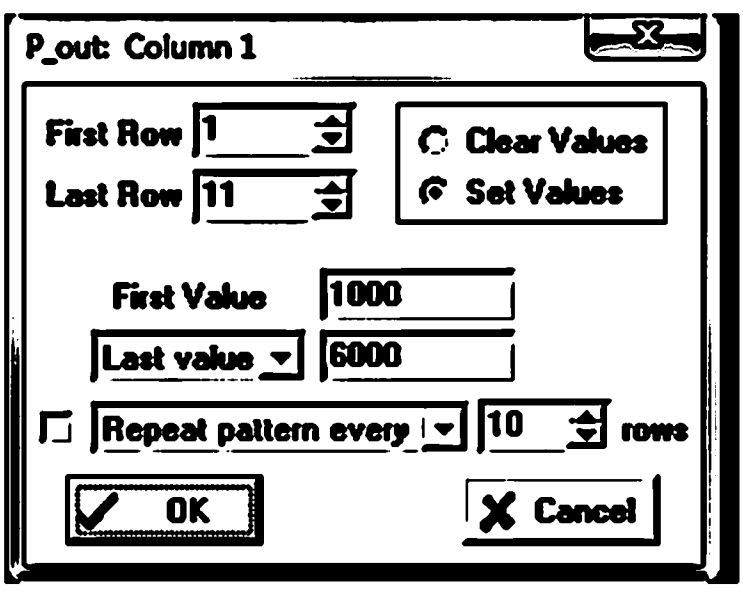
Select no of runs = 11.

Select variables in equations as "P_out" and"T_out" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 5 of 7

To enter the values in the "P_out" column click on the drop tab in "P_out" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Now click on 'Solve table' of 'Calculate' menu.



6.06P pg 2

Now click on 'Solve table' of 'Calculate' menu.

The Solution will appear as shown below:

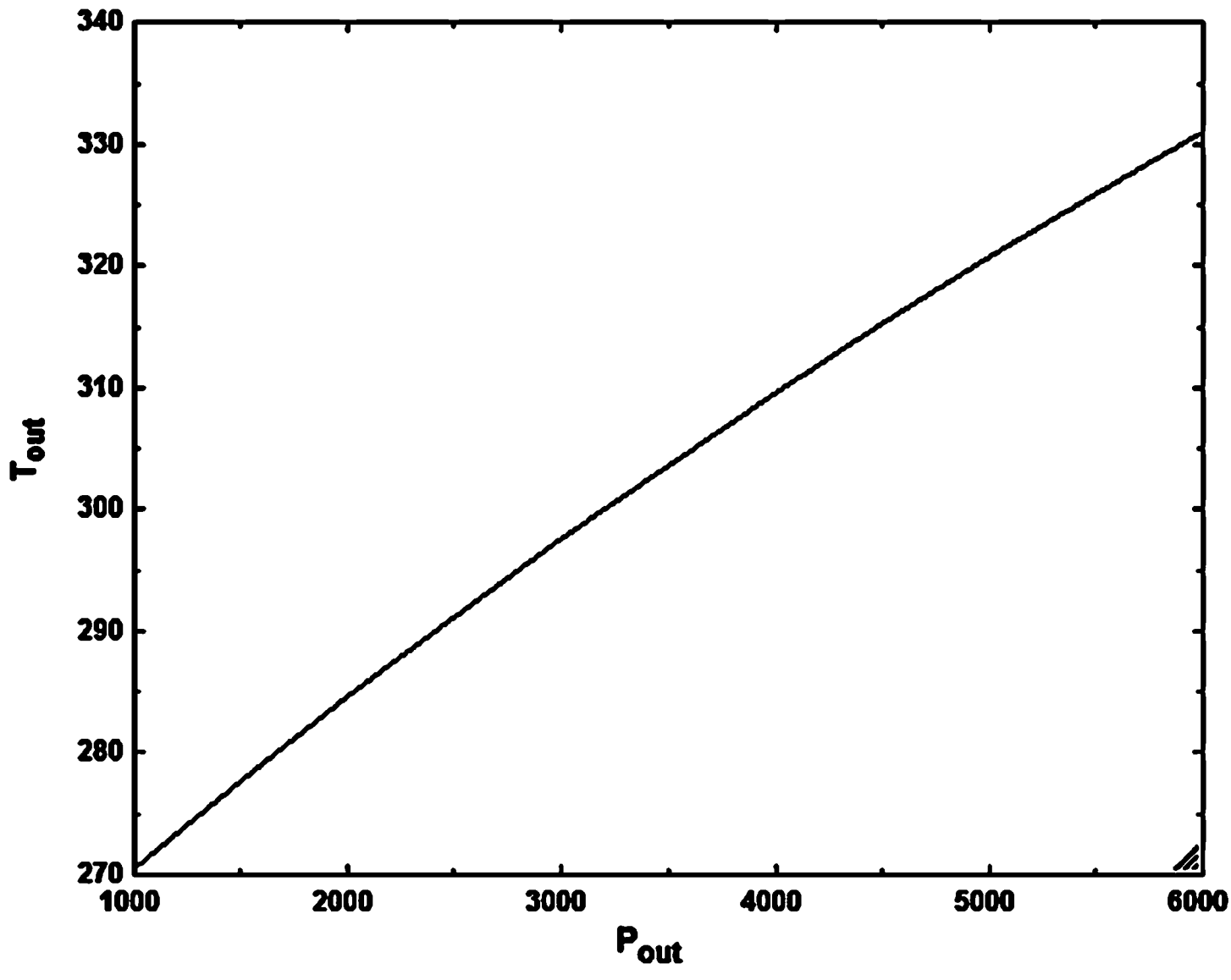
1..11	1 P_{out}	2 T_{out}
Run 1	1000	270.5
Run 2	1500	277.7
Run 3	2000	284.6
Run 4	2500	291.2
Run 5	3000	297.6
Run 6	3500	303.7
Run 7	4000	309.5
Run 8	4500	315.2
Run 9	5000	320.7
Run 10	5500	325.9
Run 11	6000	331

Step 6 of 7

Inorder to plot the pressure verses the temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as "P_out" and Y-axis as "T_out" and press 'OK'

The Graph will be plotted as shown below:

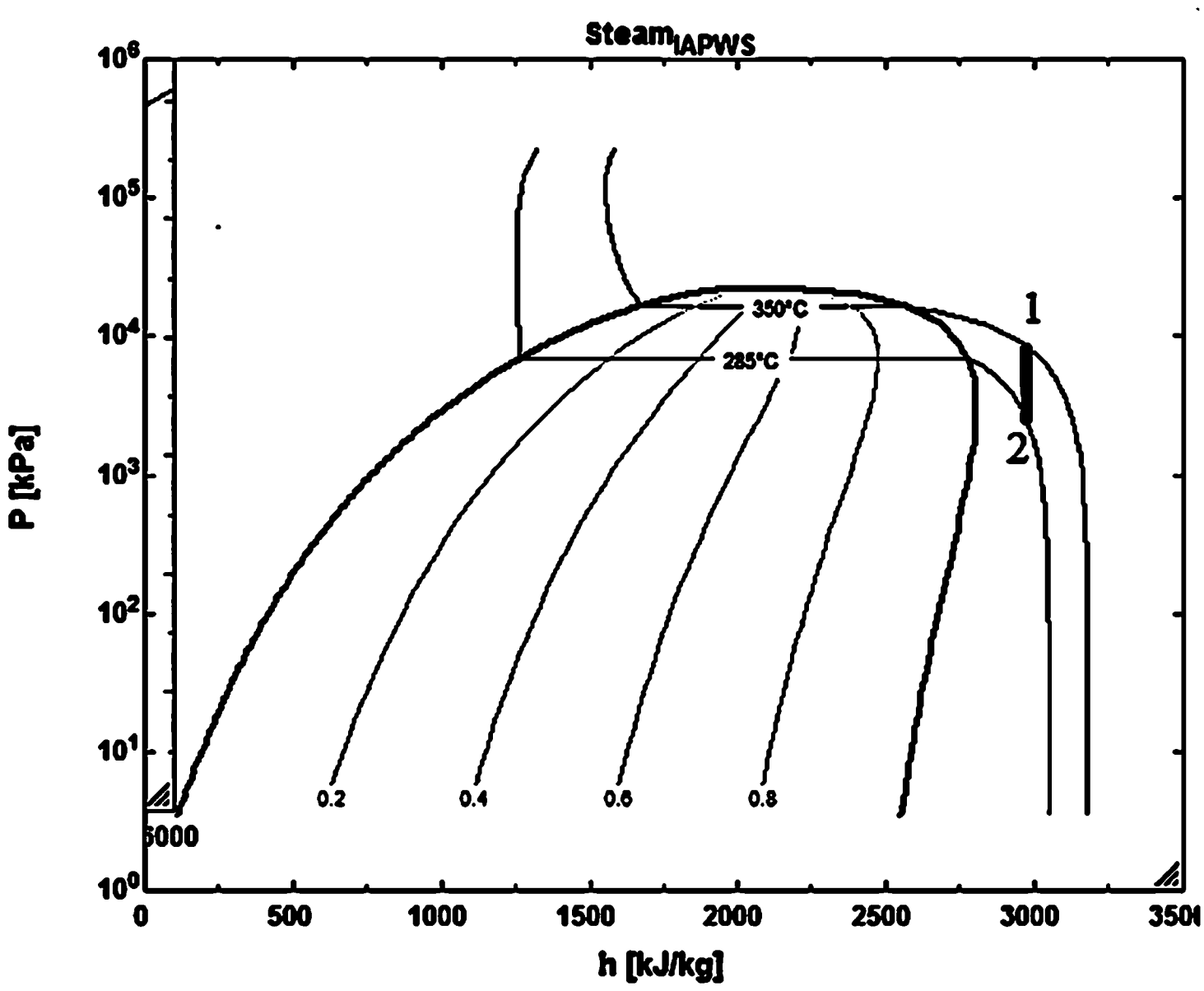


Step 7 of 7

Inorder to plot the cable enthalpy verses the pressure, click on the 'properties Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as "h" and Y-axis as "P" and press 'OK'

The Graph will be plotted as shown below:



Question: 6.064P

Refrigerant-134a enters the expansion valve of a refrigeration system at 160 psia as a saturated liquid and leaves at 30 psia. Determine the temperature and internal energy changes across the valve.

Step 1 of 5

Obtain the following properties from the table of A-12E "saturated refrigerant-134a pressure tables" at pressure 160 psia.

Enthalpy at inlet,

$$h_1 = h_f$$
$$= 48.519 \text{ Btu/lbm}$$

Internal energy at inlet,

$$u_1 = u_f$$
$$= 48.101 \text{ Btu/lbm}$$

Saturated temperature is,

$$T_1 = T_{sat}$$
$$= 109.5°\text{F}$$

Since, the enthalpy at inlet is equal to the enthalpy at outlet in expansion, hence

$$h_2 = h_1$$

Substitute 48.519 Btu/lbm for h_1.

$$h_2 = h_1$$
$$= 48.519 \text{ Btu/lbm}$$

Step 2 of 5

Obtain the following properties from the table of A-12E "saturated refrigerant-134a pressure tables" at pressure 160 psia.

Saturated liquid enthalpy, $h_{2f} = 17.006 \text{ Btu/lbm}$

Saturated vapor enthalpy, $h_{2g} = 105.32 \text{ Btu/lbm}$

Saturated liquid internal energy, $u_{2f} = 16.939 \text{ Btu/lbm}$

Saturated vapor internal energy, $u_{2g} = 96.72 \text{ Btu/lbm}$

Saturated temperature is,

$$T_2 = T_{sat}$$
$$= 15.37°\text{F}$$

Step 3 of 5

Find the quality at state using the following expression.

$$h_2 = h_{2f} + x\left(h_g - h_f\right)$$
$$x = \frac{\left(h_2 - h_{2f}\right)}{\left(h_g - h_f\right)}$$

Substitute 48.519 Btu/lbm for h_2, 17.006 Btu/lbm for h_{2f}, and 105.32 Btu/lbm for h_{2g}.

$$x = \frac{\left(h_2 - h_{2f}\right)}{\left(h_g - h_f\right)}$$
$$= \frac{48.519 - 17.006}{105.32 - 17.006}$$
$$= 0.356$$

Step 4 of 5

Find the internal energy at outlet using the equation.

Find the Internal energy at exit using the following expression.

$$u_2 = u_{2f} + x\left(u_g - u_f\right)$$
$$= 16.939 + 0.356\left(96.72 - 16.939\right)$$
$$= 45.34 \text{ Btu/lbm}$$

Find the change in temperatures using the equation.

$$\Delta T = T_2 - T_1$$

Substitute $15.37°\text{F}$ for T_2 and $109.5°\text{F}$ for T_1.

$$\Delta T = T_2 - T_1$$
$$= 109.5 - 15.37$$
$$= 94.13°\text{F}$$

Therefore, the required change in temperature is $\boxed{94.13°\text{F}}$.

Step 5 of 5

Find the change in internal energy using the equation.

$$\Delta u = u_2 - u_1$$

Substitute 48.101 Btu/lbm for u_1, and 45.34 Btu/lbm for u_2.

$$\Delta u = u_2 - u_1$$
$$= 45.34 - 48.101$$
$$= -2.761 \text{ Btu/lbm}$$

Therefore, the change in internal energy is $\boxed{-2.761 \text{ Btu/lbm}}$.

Question: 6.065P

Consider a steady-flow mixing process. Under what conditions will the energy transported into the control volume by the incoming streams be equal to the energy transported out of it by the outgoing stream?

Step 1 of 1

Mixing chambers are usually well insulated ($q = 0$) and usually do not involve any kind of work ($w = 0$). Also, the kinetic and potential energies of the fluid streams are usually negligible ($ke = 0, pe = 0$).

Under all these conditions the energy transport in and out of the control volume are same.

Question: 6.066P

Consider a steady-flow heat exchanger involving two different fluid streams. Under what conditions will the amount of heat lost by one fluid be equal to the amount of heat gained by the other?

Step 1 of 1

Heat exchangers typically involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream.

Under these conditions the amount of heat lost by one fluid is equal to the amount of heat gained by the other.

Question: 6.067P

When two fluid streams are mixed in a mixing chamber, can the mixture temperature be lower than the temperature of both streams? Explain.

Step 1 of 1

Answer: Yes.

When the mixing chamber loses heat to the surrounding medium, the mixture temperature will be lower than the temperature of both streams. But usually, mixing chambers are well insulated. Therefore, the case violates the statement that the mixing chambers are insulated.

Question: 6.068P

Hot and cold streams of a fluid are mixed in a rigid mixing chamber. The hot fluid flows into the chamber at a mass flow rate of 5 kg/s with an energy in the amount of 150 kJ/kg. The cold fluid flows into the chamber with a mass flow rate of 15 kg/s and carries energy in the amount of 50 kJ/kg. There is heat transfer to the surroundings from the mixing chamber in the amount of 5.5 kW. The mixing chamber operates in a steady-flow manner and does not gain or loose energy or mass with time. Determine the energy carried from the mixing chamber by the fluid mixture per unit mass of fluid, in kJ/kg.

Step 1 of 2

Apply mass balance for the mixing chamber system,

$$\dot{m}_{out} = \dot{m}_{in}$$

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Here, $\dot{m}_1$ is mass flow rate of hot fluid stream, $\dot{m}_2$ is mass flow rate of cold fluid stream and $\dot{m}_3$ is mass flow rate of mixture.

Substitute $5\,\text{kg/s}$ for $\dot{m}_1$, $15\,\text{kg/s}$ for $\dot{m}_2$

$$\dot{m}_3 = 5+15$$

$$= 20\,\text{kg/s}$$

Step 2 of 2

Apply energy balance to the mixing chamber system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} e_{in} = \dot{m}_{out} e_{out} + \dot{Q}_{out}$$

$$\dot{m}_1 e_1 + \dot{m}_2 e_2 = \dot{m}_3 e_3 + \dot{Q}_{out}$$

Here, e_1 is the amount of energy of hot fluid, e_2 is the amount of energy of cold fluid and $\dot{Q}_{out}$ is the amount of heat transfer to the surroundings from the mixing chamber.

Substitute $150\,\text{kJ/kg}$ for e_1, $50\,\text{kJ/kg}$ for e_2 and $5.5\,\text{kW}$ for $\dot{Q}_{out}$

$$5\times150+15\times50 = 20\times e_3 + 5.5$$

$$1500 = 20\times e_3 + 5.5$$

$$e_3 = 74.725\,\text{kJ/kg}$$

Therefore, the energy carried from the mixing chamber is $\boxed{74.725\,\text{kJ/kg}}$

Question: 6.069P

A hot-water stream at 80°C enters a mixing chamber with a mass flow rate of 0.5 kg/s where it is mixed with a stream of cold water at 20°C. If it is desired that the mixture leave the chamber at 42°C, determine the mass flow rate of the cold-water stream. Assume all the streams are at a pressure of 250 kPa.

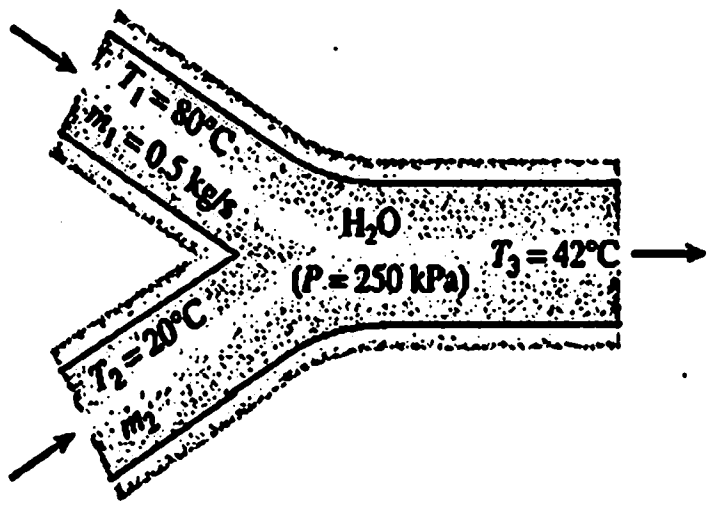


FIGURE P6–69

Step 1 of 5

Given

Hot water stream temperature $T_1 = 80°C$

Mass flow rate of hot water stream $\dot{m}_1 = 0.5$ kg/s

Cold water stream temperature $T_2 = 20°C$

Mixture temperature $T_3 = 42°C$

Pressure of all the streams $P = 250$ kPa

Step 2 of 5

Mixing chambers are usually well insulated ($q = 0$) and usually do not involve any kind of work ($w = 0$). Also, the kinetic and potential energies of the fluid streams are usually negligible ($ke = 0, pe = 0$).

Applying mass balance and energy balance for the mixing chamber system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 5

Here there are two inlet streams and one outlet stream

By mass balance

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 \quad \text{---------- (1)}$$

By Energy balance

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad \text{---------- (2)}$$

Substituting equation 1 in equation 2

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \left(\dot{m}_1 + \dot{m}_2\right) h_3$$

$$\dot{m}_1 h_3 - \dot{m}_1 h_1 = \dot{m}_2 h_2 - \dot{m}_2 h_3$$

$$\dot{m}_1 (h_3 - h_1) = \dot{m}_2 (h_2 - h_3)$$

$$\dot{m}_2 = \frac{h_3 - h_1}{h_2 - h_3} \dot{m}_1$$

Step 4 of 5

At $T_1 = 80°C$ from saturated water tables

$h_1 = h_f = 335.02$ kJ/kg

At $T_2 = 20°C$ from saturated water tables

$h_2 = h_f = 83.915$ kJ/kg

At $T_3 = 42°C$ from saturated water tables

$$h_3 = h_f = 167.53 + (188.44 - 167.53)\left(\frac{42-40}{45-40}\right)$$

$h_f = h_3 = 175.894$ kJ/kg

Step 5 of 5

Therefore

$$\dot{m}_2 = \frac{h_3 - h_1}{h_2 - h_3} \dot{m}_1$$

$$\dot{m}_2 = \frac{175.894 - 335.02}{83.915 - 175.894} \times 0.5$$

$$\boxed{\dot{m}_2 = 0.865}$$

Question: 6.070P

Water at 65 °F and 20 psia is heated in a chamber by mixing it with saturated water vapor at 20 psia. If both streams enter the mixing chamber at the same mass flow rate, determine the temperature and the quality of the exiting stream.

Step 1 of 6

Apply mass balance to the mixing chamber system,

$$\dot{m}_{out} = \dot{m}_{in}$$

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Here, $\dot{m}_1$ is mass flow rate of water, $\dot{m}_2$ is mass flow rate of water vapor and $\dot{m}_3$ is mass flow rate of mixture

But $\dot{m}_1 = \dot{m}_2$ then,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_1$$

$$\dot{m}_3 = 2\dot{m}_1$$

Step 2 of 6

Assume mixing chambers are usually well insulated, so $\dot{Q}_{out} = 0$ and they do not involve any kind of work so $\dot{W} = 0$.

Also, the changes in kinetic and potential energies of the fluid streams are usually negligible in the chamber. So,

$$ke = 0,$$

$$pe = 0$$

Apply energy balance to the mixing chamber system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$$

$$\dot{m}_1 h_1 + \dot{m}_1 h_2 = 2\dot{m}_1 h_3$$

Here, h_1 is enthalpy of water, h_2 is enthalpy of vapor and h_3 is enthalpy of the mixture.

$$h_1 + h_2 = 2h_3 \qquad \text{..... (1)}$$

Step 3 of 6

From saturated water tables at temperature $T_1 = 65°\text{F}$,

Saturated enthalpy of liquid,

$$h_1 = h_f$$

$$h_1 = 33.08 \text{ Btu/lbm}$$

From saturated water tables at pressure $P_2 = 20 \text{ psia}$

Saturated enthalpy of vapor,

$$h_2 = h_g$$

$$h_2 = 1156.2 \text{ Btu/lbm}$$

Step 4 of 6

Substitute the known enthalpy values in equation (1),

$$h_1 + h_2 = 2h_3$$

$$33.08 + 1156.2 = 2h_3$$

$$h_3 = 594.64 \text{ Btu/lbm}$$

Assume the mixture pressure is also at pressure $P_3 = 20 \text{ psia}$

Obtain the properties of enthalpy values from saturated water tables at pressure $P_3 = 20 \text{ psia}$

Enthalpy of saturated water, $h_f = 196.27 \text{ Btu/lbm}$

Enthalpy of evaporation, $h_{fg} = 959.93 \text{ Btu/lbm}$

Step 5 of 6

Find the exit quality of steam by using enthalpy value at state 3

$$h_3 = h_f + x_3 h_{fg}$$

$$x_3 = \frac{h_3 - h_f}{h_{fg}}$$

$$= \frac{594.64 - 196.27}{959.93}$$

$$= 0.415$$

Therefore, the quality of the mixture, x_3 is $\boxed{0.415}$

Step 6 of 6

The exit flow is a saturated mixture, because the enthalpy value (h_3) lies between saturated enthalpy values of vapor, h_g and liquid, h_f

Therefore, the temperature of the exiting stream is equal to the saturation temperature at pressure 20 psia

Now obtain the temperature from saturated water tables at pressure $P_3 = 20 \text{ psia}$

$$T_{sat} = T_3 = 227.92°\text{F}$$

Hence, the temperature of the exiting stream is $\boxed{227.92°\text{F}}$

Question: 6.071P

A stream of refrigerant-134a at 1 MPa and 20°C is mixed with another stream at 1 MPa and 80°C. If the mass flow rate of the cold stream is twice that of the hot one, determine the temperature and the quality of the exit stream.

Step 1 of 5

Apply mass balance equation to the mixing chamber system

$$\dot{m}_{out} = \dot{m}_{in}$$

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Here, $\dot{m}_1$ is mass flow rate of cold stream 1 and $\dot{m}_2$ is mass flow rate of hot stream and $\dot{m}_3$ is mass flow rate of mixing stream.

Substitute $2\dot{m}_2$ for $\dot{m}_1$

$$\dot{m}_3 = 2\dot{m}_2 + \dot{m}_2$$

$$= 3\dot{m}_2$$

Step 2 of 5

Assume the Mixing chambers are usually well insulated, $q = 0$ and they do not involve any kind of work, $w = 0$

Also, the changes in kinetic and potential energies of the fluid streams are usually negligible, $ke = 0, pe = 0$

Then energy balance equation for the mixing chamber becomes,

$$\dot{m}_{out} h_{out} = \dot{m}_{in} h_{in}$$

$$\dot{m}_3 h_3 = \dot{m}_1 h_1 + \dot{m}_2 h_2$$

$$3\dot{m}_2 h_3 = 2\dot{m}_2 h_1 + \dot{m}_2 h_2$$

$$3h_3 = 2h_1 + h_2$$

Here, h_1 is enthalpy of cold stream, h_2 is enthalpy of hot stream and h_3 is enthalpy of mixture.

Step 3 of 5

Obtain enthalpy at state 1 from saturated R-134a tables at temperature $T_1 = 20°C$

$$h_1 = h_f = 79.32 \text{ kJ/kg}$$

Obtain enthalpy at state 2 from superheated R-134a tables at pressure $P_2 = 1\text{MPa}$ and temperature $T_2 = 80°C$

$$h_2 = 314.25 \text{ kJ/kg}$$

Substitute these enthalpy values in the equation,

$$3h_3 = 2h_1 + h_2$$

$$3h_3 = (2 \times 79.32) + (314.25)$$

$$h_3 = \frac{472.89}{3}$$

$$h_3 = 157.63 \text{kJ/kg}$$

Step 4 of 5

Assume mixture is also at same inlet and exit pressure, $P_3 = 1 \text{ MPa}$

Obtain the properties of saturated R-134a tables at mixture pressure $P_3 = 1 \text{ MPa}$

Enthalpy of saturated liquid R-134a, $h_f = 107.32 \text{ kJ/kg}$

Enthalpy of evaporation of R-134a, $h_{fg} = 163.67 \text{ kJ/kg}$

Enthalpy of saturated vapor R-134a, $h_g = 270.99 \text{ kJ/kg}$

The exit flow is a saturated mixture since the enthalpy value (h_3) lies between saturated enthalpy values of vapor (h_g) and fluid (h_f).

Find the dryness fraction at state 3 by using enthalpy equation,

$$h_3 = h_f + x_3 h_{fg}$$

$$157.63 = 107.32 + x_3 \times 163.67$$

$$x_3 = 0.307$$

The quality of the mixture is $\boxed{0.3079}$

Step 5 of 5

The temperature at exit is the saturation temperature at mixture pressure $P_3 = 1 \text{ MPa}$

Obtain the temperature from saturated R-134a tables at $P_3 = 1 \text{ MPa}$

Saturation temperature, $T_{sat} = T_3 = 39.37°C$

Hence, temperature of the exit stream is $\boxed{39.37°C}$

Question:

Reconsider Prob. 6–71. Using EES (or other)software, investigate the effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream. Let the ratio of the mass flow rate of the cold stream to that of the hot stream vary from 1 to 4. Plot the mixture temperature and quality against the cold-to-hot mass flow rate ratio, and discuss the results.

Step 1 of 7

Mixing chambers are usually well insulated, $q = 0$ and usually do not involve any kind of work, $w = 0$.

Also, the changes in kinetic and potential energies of the fluid streams are usually negligible, $ke = 0, pe = 0$

Applying mass balance for the mixing chamber system

$$\dot{m}_{out} = \dot{m}_{in}$$
$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

Mass flow rate relation, $\dot{m}_1 = 2\dot{m}_2$

$$\dot{m}_3 = 2\dot{m}_2 + \dot{m}_2$$
$$\dot{m}_3 = 3\dot{m}_2$$

Applying energy balance for the mixing chamber system,

$$\dot{m}_{out} h_{out} = \dot{m}_{in} h_{in}$$
$$\dot{m}_3 h_3 = \dot{m}_1 h_1 + \dot{m}_2 h_2$$
$$3\dot{m}_2 h_3 = 2\dot{m}_2 h_1 + \dot{m}_2 h_2$$
$$3h_3 = 2h_1 + h_2$$

Step 2 of 7

From Saturated R-134a tables,

Inlet Temperature of Cold R-134a stream, $T_1 = 20°C$ The Enthalpy is $h_1 = h_f = 79.32\,kJ/kg$

From superheated R-134a tables,

Outlet Temperature of Hot R-134a stream, $T_2 = 80°C$ The Enthalpy is,

$$h_2 = 314.25\,kJ/kg$$

$$3h_3 = 2h_1 + h_2$$
$$3h_3 = (2 \times 79.32) + (314.25)$$
$$h_3 = \frac{427.89}{3}$$
$$h_3 = 157.63\,kJ/kg$$

Step 3 of 7

The mixture is also at same pressure. So, $P_3 = 1\ MPa$

From Saturated R-134a tables, at $P_3 = 1\ MPa$

Enthalpy of saturated liquid R-134a, $h_f = 107.32\,kJ/kg$

Enthalpy of evaporation of R-134a, $h_{fg} = 163.67\,kJ/kg$

Enthalpy of saturated vapor R-134a, $h_g = 270.99\,kJ/kg$

In saturated region,

$$x_3 = \frac{h_3 - h_f}{h_{fg}}$$
$$= \frac{157.63 - 107.32}{163.67}$$
$$= 0.3079$$

The quality of the mixture at outlet x_3 is $\boxed{0.3079}$

The exit flow is a saturated mixture since the enthalpy value (h_3) lies between enthalpy values of vapor (h_g) and fluid (h_f)

Therefore the temperature at exit is the saturation temperature,

At $P_3 = 1\ MPa$, from saturated R-134a tables, saturation temperature is $T_3 = 39.37°C$

Hence temperature at outlet T_3 is $\boxed{39.37°C}$

Step 4 of 7

The given problem can be solved using EES software by following the below steps.

1. Open EES software

2. Create a new file by clicking on the 'New' of the 'File menu'

3.Now type the following code in dailog box

Equations Window

```
"Input Data"
"m_frac = 2" "m_frac =m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"
T[1]=20 [C]
P[1]=1000 [kPa]
T[2]=80 [C]
P[2]=1000 [kPa]
m_dot_1=m_frac*m_dot_2
P[3]=1000 [kPa]
m_dot_1=1
"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"
m_dot_1+ m_dot_2 =m_dot_3
"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "Steady-flow requirement"
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]
"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
```

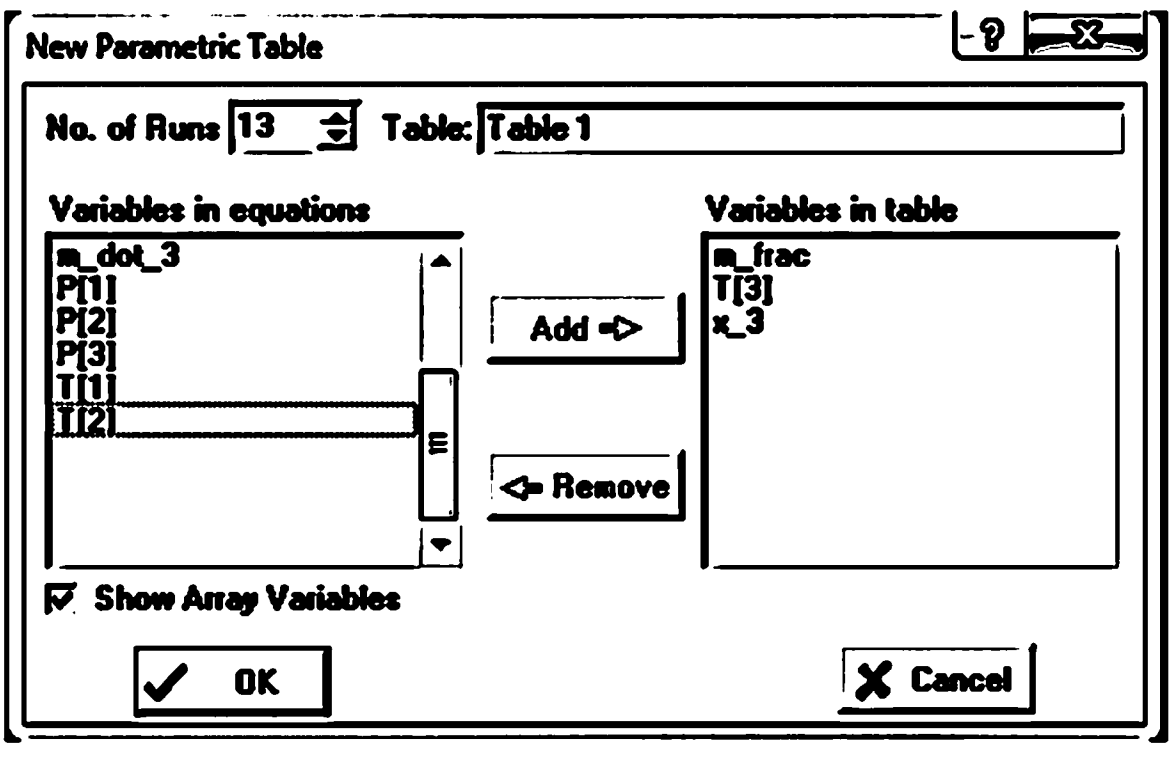
```
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]
"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
x_3=QUALITY(R134a,h=h[3],P=P[3])
```

Now to solve the equation for a range of m fraction, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.

A dailog box will open.

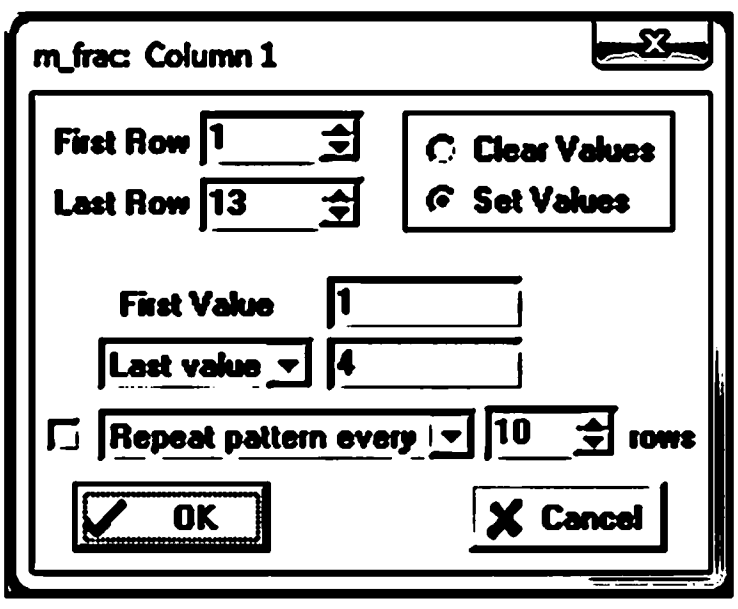
Select no of runs = 13.

Select variables in equations as "P_out" and"T_out" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 5 of 7

To enter the values in the "m fraction" column click on the drop tab in "m fraction" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 6 of 7

Now click on 'Solve table' of 'Calculate' menu.

The Solution will appear as shown below:

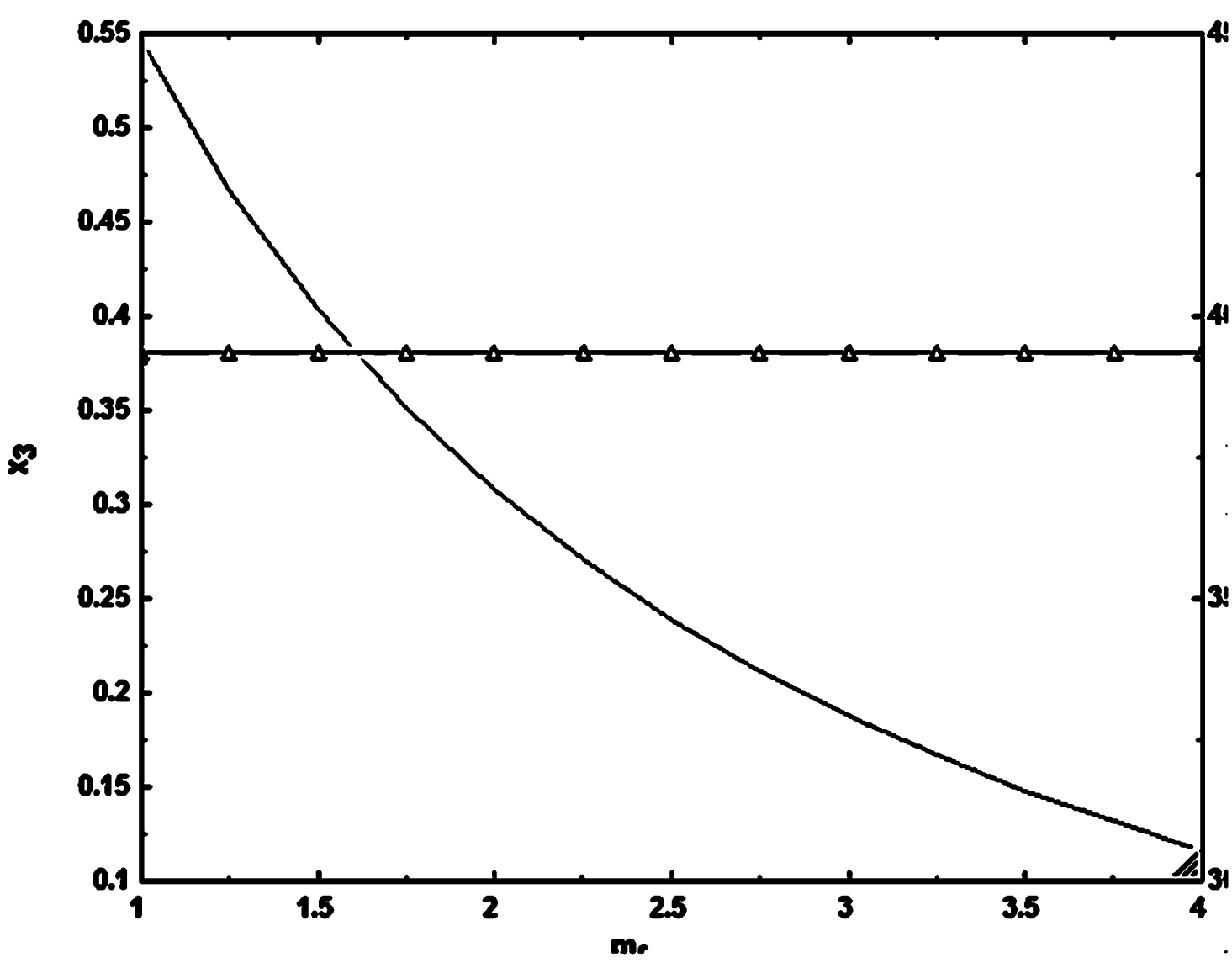
1..13	1 m_{frac}	2 T_3 [C]	3 x_3
Run 1	1	39.37	0.5467
Run 2	1.25	39.37	0.4669
Run 3	1.5	39.37	0.4032
Run 4	1.75	39.37	0.351
Run 5	2	39.37	0.3075
Run 6	2.25	39.37	0.2707
Run 7	2.5	39.37	0.2392
Run 8	2.75	39.37	0.2118
Run 9	3	39.37	0.1879
Run 10	3.25	39.37	0.1668
Run 11	3.5	39.37	0.1481
Run 12	3.75	39.37	0.1313
Run 13	4	39.37	0.1162

Step 7 of 7

Inorder to plot the m fraction versus the dryness fraction and temperature , click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu. And again use overlay plot.

Select X-axis as "m fraction" and Y-axis as "X_3 and T3" and press 'OK'

The Graph will be plotted as shown below:



6.072Pm.pg 3

Question: 6.073P

Steam is to be condensed on the shell side of a heat exchanger at 85°F. Cooling water enters the tubes at 60°F at a rate of 138 lbm/s and leaves at 73°F. Assuming the heat exchanger to be well-insulated, determine the rate of heat transfer in the heat exchanger and the rate of condensation of the steam.

Step 1 of 5

Given

Inlet temperature of cooling water $T_1 = 60°F$

Exit temperature of cooling water $T_2 = 73°F$

The mass flow rate of cooling water $\dot{m}_{cw} = 138\ \text{lbm/s}$

The temperature of steam in heat exchanger $T = 85°F$

Step 2 of 5

Heat exchangers typically involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream.

Applying mass balance and energy balance for the heat exchanger system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 5

The specific heat of water $c_p = 1\ \text{Btu/lbm.°F}$

Applying energy equation to the cooling water stream

Therefore

$$\dot{Q} + \dot{m}_w h_1 = \dot{m}_w h_2$$

$$\dot{Q} = \dot{m}_w (h_2 - h_1) \qquad \text{(since } h = c_p T\text{)}$$

$$\dot{Q} = \dot{m}_w c_{pw} (T_2 - T_1)$$

Substituting the values

$$\dot{Q} = 138(1)(73 - 60)$$

The rate of heat transfer

$$\boxed{\dot{Q} = 1794\ \text{Btu/s}}$$

Step 4 of 5

Heat lost by the hot fluid is equal to the heat gained by the cold fluid

Since the temperature of steam remains constant it changes from saturated vapor to saturated liquid

Therefore

$$\dot{Q}_{water} = -\dot{Q}_{steam}$$

Applying energy equation to steam

$$\dot{Q}_{steam} + \dot{m}_s h_3 = \dot{m}_s h_4$$

$$\dot{Q}_{steam} + \dot{m}_s h_g = \dot{m}_s h_f$$

$$\dot{Q}_{out} = \dot{m}_s (h_f - h_g)$$

Where h_f *and* h_g are taken from $T = 85°F$

At $T = 85°F$ from saturated water tables

$h_f = 53.06\ \text{Btu/lbm}$

$h_g = 1098.3\ \text{Btu/lbm}$

Step 5 of 5

$$\dot{Q}_{out} = \dot{m}_s (h_f - h_g)$$

$$-1794 = \dot{m}_s (53.06 - 1098.3)$$

$$\boxed{\dot{m}_s = 1.7163\ \text{lbm/s}}$$

Question: 6.074P

Steam enters the condenser of a steam power plant at 20 kPa and a quality of 95 percent with a mass flow rate of 20,000 kg/h. It is to be cooled by water from a nearby river by circulating the water through the tubes within the condenser. To prevent thermal pollution, the river water is not allowed to experience a temperature rise above 10°C. If the steam is to leave the condenser as saturated liquid at 20 kPa, determine the mass flow rate of the cooling water required.

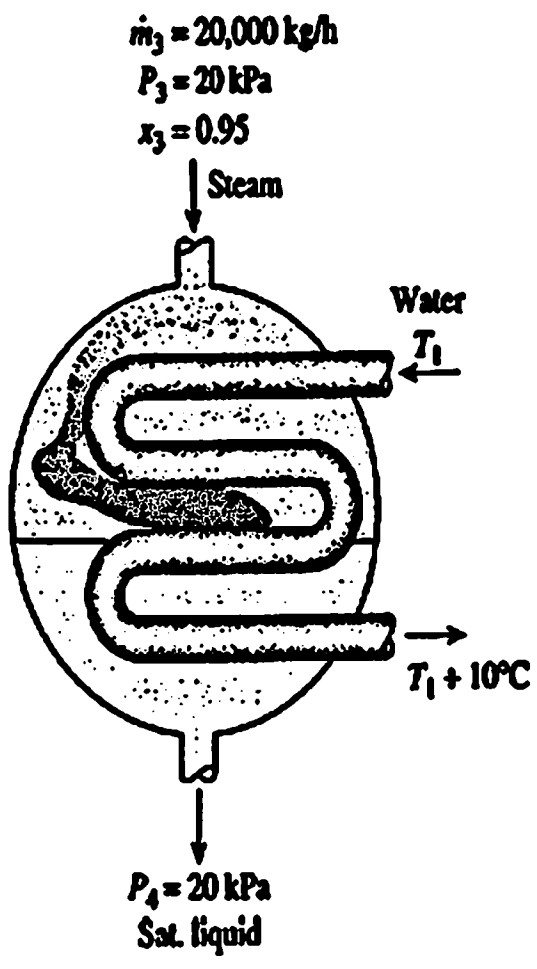


FIGURE P6–74

Step 1 of 5

Given

Properties of steam

Inlet pressure $P_3 = 20$ kPa

Inlet quality $x_3 = 0.95$

Exit pressure as saturated liquid $P_4 = 20$ kPa

Mass flow rate of steam $\dot{m}_s = 20000$ kg/hr

$$= \frac{20000}{3600}$$

$$\dot{m}_s = 5.56 \text{ kg/s}$$

Inlet water temperature is T_1

Exit water temperature is $T_2 = T_1 + 10°C$

Step 2 of 5

Mixing chambers are usually well insulated ($q = 0$) and usually do not involve any kind of work ($w = 0$). Also, the kinetic and potential energies of the fluid streams are usually negligible ($ke = 0, pe = 0$).

Applying mass balance and energy balance for the mixing chamber system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 5

In our problem we have two inlets and two exits, the mass flow rate of water and steam remains same at inlet and exit.

Therefore from above energy equation

$$\dot{m}_w h_1 + \dot{m}_s h_3 = \dot{m}_w h_2 + \dot{m}_s h_4$$

$$\dot{m}_w (h_1 - h_2) = \dot{m}_s (h_4 - h_3) \quad \text{---------- (1)}$$

Step 4 of 5

At $P_3 = 20$ kPa from properties of saturated liquid water

$$h_f = 251.42 \text{ kJ/kg}$$

$$h_{fg} = 2357.5 \text{ kJ/kg}$$

$$h_3 = h_f + x_3 h_{fg}$$

$$h_3 = 251.42 + (0.95)2357.5$$

$$h_3 = 2491.045 \text{ kJ/kg}$$

At $P_4 = 20$ kPa from properties of saturated liquid water

$$h_f = h_4 = 251.42 \text{ kJ/kg}$$

Assume the specific heat of water is $C_p = 4.18$ kJ/kg.K

Step 5 of 5

In equation 1 the enthalpy values of water can be write it as $h = c_p T$ and substituting all the known values in equation 1

$$\dot{m}_w (h_1 - h_2) = \dot{m}_s (h_4 - h_3)$$

$$\dot{m}_w c_p (T_1 - T_2) = \dot{m}_s (h_4 - h_3)$$

$$\dot{m}_w \times 4.18 (T_1 - (T_1 + 10)) = 5.56 (251.42 - 2491.045)$$

$$\dot{m}_w \times 4.18 (-10) = 5.56 (251.42 - 2491.045)$$

$$\dot{m}_w = \frac{2239.6}{10 \times 4.18} \times 5.56$$

$$= 297.89 \text{ kg/s}$$

The mass flow rate of water is

$$\boxed{\dot{m}_w = 297.89 \text{ kg/sec}}$$

Question: 6.075P

A heat exchanger is to cool ethylene glycol (cp = 2.56 kJ/kg•°C) flowing at a rate of 3.2 kg/s from 80°C to 40°C by water (cp = 4.18 kJ/kg•°C) that enters at 20°C and leaves at 70°C. Determine (*a*) the rate of heat transfer and (b) the mass flow rate of water.

Step 1 of 3

Given

The mass flow rate of ethylene glycol, $\dot{m}_{gly} = 3.2$ kg/s

The temperatures of glycol at inlet, $T_1 = 80°C$

The temperatures of glycol at exit, $T_2 = 40°C$

The specific heat value of water, $(C_p)_{water} = 4.18$ kJ/kg°C

The specific heat value of glycol, $(C_p)_{gly} = 2.56$ kJ/kg°C

The temperature of water at inlet, $T_3 = 20°C$

The temperature of water at exit, $T_4 = 70°C$

Step 2 of 3

Heat exchangers typically involve no work interactions. So, $w = 0$

The changes in kinetic and potential energies are negligible. So, $ke = 0, pe = 0$

Applying mass balance for the heat exchanger system

$$\dot{m}_{in} = \dot{m}_{out}$$

Applying energy balance for the heat exchanger system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} + \dot{Q}$$

and energy balance for the heat exchanger system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

i.e. $\dot{m}_{in} h_{in} = \dot{Q} + \dot{m}_{out} h_{out}$ $(w = 0 \text{ and } ke = 0, pe = 0)$

(a)

Taking glycol as a system, $\dot{m}_{gl} h_1 = \dot{Q} + \dot{m}_{gl} h_2$

Hence heat transferred, $\dot{Q} = \dot{m}_{gl}(h_1 - h_2)$

But enthalpy, $h = C_p T$, Hence

$$\dot{Q} = \dot{m}_{gl} C_p (T_1 - T_2)$$

$$\dot{Q} = 3.2 \times 2.56 (80 - 40)$$

$$\dot{Q} = 327.68 \text{ kJ/kg}$$

The rate of heat transfer, $\boxed{\dot{Q} = 327.68 \text{ kJ/kg}}$

Step 3 of 3

(b)

Taking water as a system, $\dot{m}_w h_1 = \dot{Q} + \dot{m}_w h_2$

But enthalpy, $h = C_p T$, Hence

$$\dot{Q} = \dot{m}_w C_p (T_1 - T_2)$$

Hence mass flow rate, $\dot{m}_w = \dfrac{\dot{Q}}{C_p (T_1 - T_2)}$

But heat transferred to water is from glycol therefore $\dot{Q} = 327.68$ kJ/kg

$$\dot{m}_w = \frac{327.68}{(4.18)(70 - 20)}$$

$$\dot{m}_w = 1.568 \text{ kg/s}$$

The mass flow rate of water is $\boxed{\dot{m}_w = 1.568 \text{ kg/s}}$

Question: 6.076P

Reconsider Prob. 6–75. Using EES (or other) software, investigate the effect of the inlet temperature of cooling water on the mass flow rate of water. Let the inlet temperature vary from 10 to 40°C, and assume the exit temperature to remain constant. Plot the mass flow rate of water against the inlet temperature, and discuss the results.

Step 1 of 4

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click Calculate on menu bar, select solve.

Finally, Solution window displayed the unknown parameters

Step 2 of 4

Equation window:

"Input Data"

"T_w[1]=20 [C]"

T_w[2]=70 [C] "w: water"

m_dot_eg=2 [kg/s] "eg: ethylene glycol"

T_eg[1]=80[C]

T_eg[2]=40 [C] _

C_p_w=4.18 [kJ/kg-K]

C_p_eg=2.56 [kJ/kg-K]

"Conservation of mass for the water: m_dot_w_in=m_dot_w_out=m_dot_w"

"Conservation of mass for the ethylene glycol: m_dot_eg_in=m_dot_eg_out=m_dot_eg"

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass steam"

"We assume no heat transfer and no work occur across the control surface."

E_dot_in - E_dot_out = DELTAE_dot_cv

DELTAE_dot_cv=0 "Steady-flow requirement"

E_dot_in=m_dot_w*h_w[1] + m_dot_eg*h_eg[1]

E_dot_out=m_dot_w*h_w[2] + m_dot_eg*h_eg[2]

Q_exchanged =m_dot_eg*h_eg[1] - m_dot_eg*h_eg[2] "Property data are given by:"

h_w[1] =C_p_w*T_w[1] "liquid approximation applied for water and ethylene glycol"

h_w[2] =C_p_w*T_w[2]

h_eg[1] =C_p_eg*T_eg[1]

h_eg[2] =C_p_eg*T_eg[2]

Step 3 of 4

Table for $\dot{m}_w$ as a function of $T_{w,1}$:

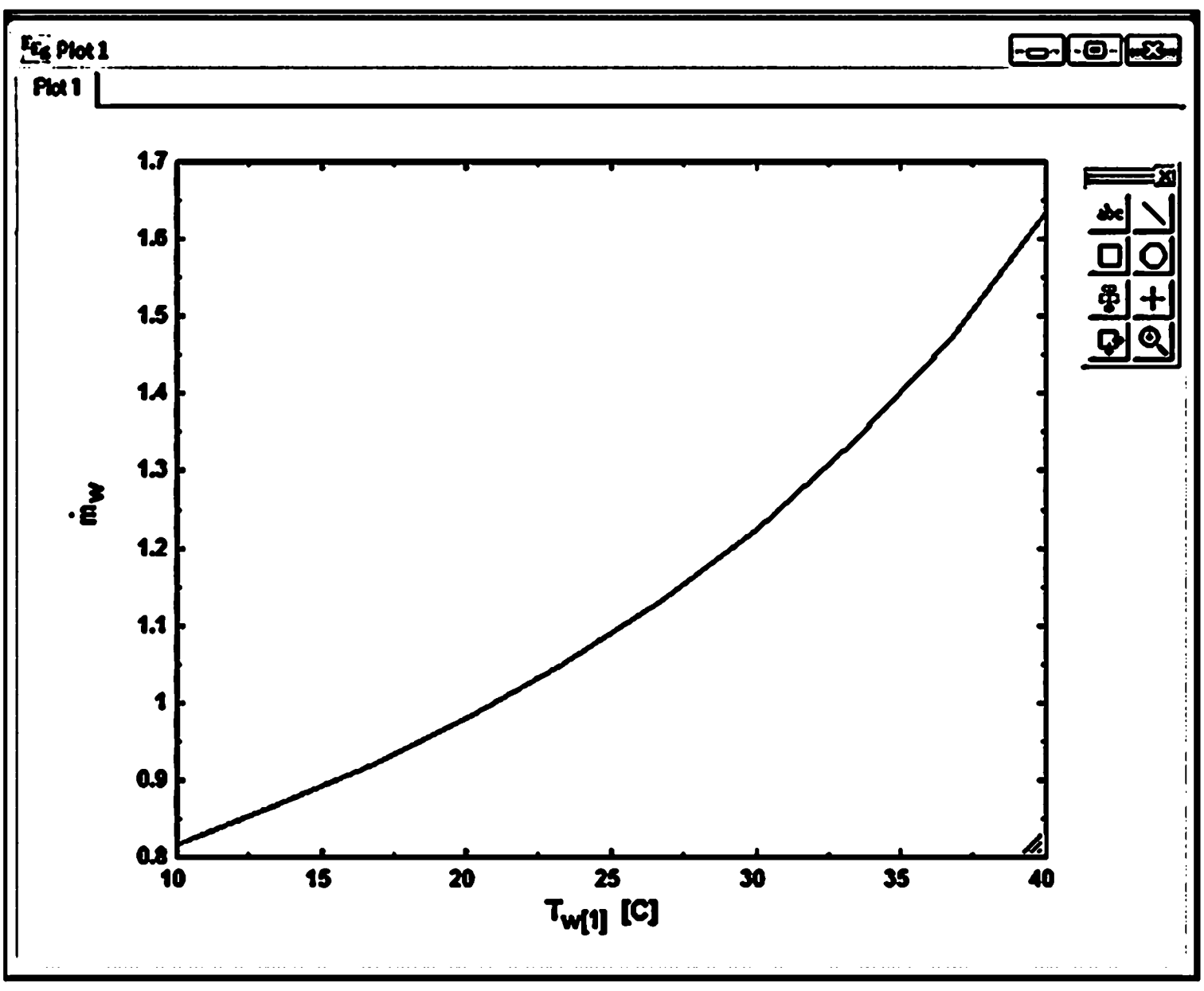
Parametric Table

Table 1 | Table 2

1..10	$T_{w,1}$ [C]	$\dot{m}_w$
Run 1	10	0.8166
Run 2	13.33	0.8646
Run 3	16.67	0.9187
Run 4	20	0.9799
Run 5	23.33	1.05
Run 6	26.67	1.131
Run 7	30	1.225
Run 8	33.33	1.336
Run 9	36.67	1.47
Run 10	40	1.633

Step 4 of 4

Graph between the $\dot{m}_w$ Vs. $T_{w,1}$:



Question: 6.077P

Cold water (cp = 4.18 kJ/kg•°C) leading to a shower enters a thin-walled double-pipe counter-flow heat exchanger at 15°C at a rate of 0.60 kg/s and is heated to 45°C by hot water (cp = 4.19 kJ/kg•°C) that enters at 100°C at a rate of 3 kg/s. Determine the rate of heat transfer in the heat exchanger and the exit temperature of the hot water.

Step 1 of 4

Given

The mass flow rate of cold water $\dot{m}_{cw} = 0.6$ kg/s

Inlet temperature of cold water $T_1 = 15°C$

Exit temperature of cold water $T_2 = 45°C$

Specific heat of cold water $c_{p_{cw}} = 4.18$ kJ/kg.°C

Specific heat of hot water $c_{p_{hw}} = 4.19$ kJ/kg.°C

Inlet temperature of hot water $T_3 = 100°C$

Mass flow rate of hot water $\dot{m}_{hw} = 3$ kg/s

Step 2 of 4

Heat exchangers typically involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream.

Applying mass balance and energy balance for the heat exchanger system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 4

Applying energy equation to cold water

$$\dot{Q} + \dot{m}_{cw} h_1 = \dot{m}_{cw} h_2$$

$$\dot{Q} = \dot{m}_{cw}(h_2 - h_1) \qquad \text{(since } h = c_p T\text{)}$$

$$\dot{Q} = \dot{m}_{cw} c_{p_{cw}} (T_2 - T_1)$$

Substituting the values

$$\dot{Q} = 0.6(4.18)(45 - 15)$$

$$\boxed{\dot{Q} = 75.24 \text{ kW}}$$

Step 4 of 4

All the heat loss by the hot fluid will be equal to heat gained by the cold fluid.
So the rate of heat transfer of the hot water will be same as above.
Therefore

$$-\dot{Q} = \dot{m}_{hw} c_{p_{hw}} (T_4 - T_3)$$

$$-75.24 = (3)(4.19)(T_4 - 100)$$

$$-5.985 = T_4 - 100$$

$$T_4 = 94.015°C$$

The exit temperature of hot water is

$$\boxed{T_4 = 94.015°C}$$

Question: 6.078P

Air (cp = 1.005 kJ/kg•°C) is to be preheated by hot exhaust gases in a cross-flow heat exchanger before it enters the furnace. Air enters the heat exchanger at 95 kPa and 20°C at a rate of 0.6 m3/s. The combustion gases (*cp* = 1.10 kJ/kg•°C) enter at 160°C at a rate of 0.95 kg/s and leave at 95°C. Determine the rate of heat transfer to the air and its outlet temperature.

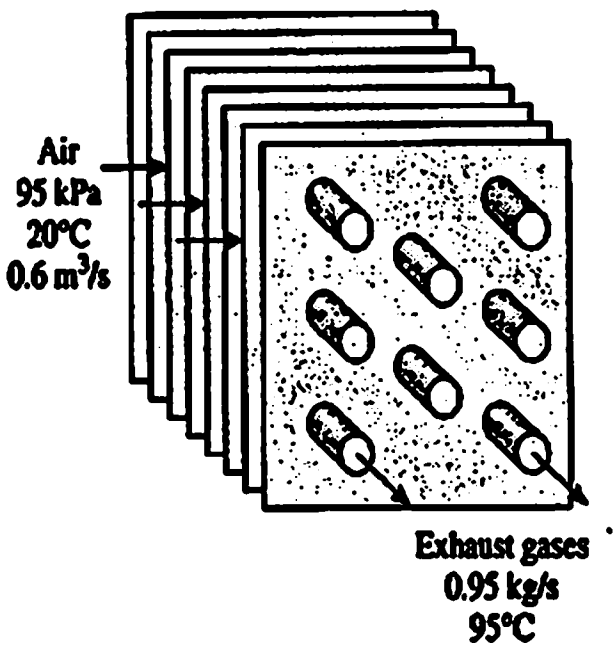


FIGURE P6–78

Step 1 of 4

Given

The volume flow rate of air at inlet, $\dot{V}_{air} = 0.6\ m^3/s$

The temperatures of air at inlet, $T_1 = 20°C$

The pressure of inlet air at inlet, $P_1 = 95\ kPa$

The specific heat values of air, $(C_P)_{air} = 1.005\ kJ/kg°C$

The specific heat values of gases, $(C_P)_{gases} = 1.10\ kJ/kg°C$

The temperatures of combustion gases at inlet, $T_3 = 160°C$

The temperatures of combustion gases at exit, $T_4 = 95°C$

Mass flow rate of gases $\dot{m}_{gas} = 0.95\ kg/s$

Step 2 of 4

(a)

Heat exchangers involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream. Hence by applying mass balance, $\dot{m}_{in} = \dot{m}_{out}$

and from energy balance for the heat exchanger system, we obtain

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

$$\dot{Q}_{in} + \dot{m}_{gas} h_4 = \dot{m}_{gas} h_3$$

$$\dot{Q}_{in} = \dot{m}_{gas} (h_3 - h_4)$$

$$\dot{Q}_{in} = \dot{m}_{gas} (C_P)_{gas} (T_3 - T_4)$$

$$\dot{Q}_{in} = 0.95(1.1)(160 - 95)$$

$$\dot{Q}_{in} = 67.925\ kW$$

Hence heat transfer rate is $\boxed{\dot{Q}_{in} = 67.925\ kW}$

Step 3 of 4

(b)

From Ideal gas equation, $P = \rho RT$

$$\rho_1 = \frac{P_1}{RT_1}$$

$$\rho_1 = \frac{95}{0.287 \times (20 + 273)}$$

$$\rho_1 = 1.129\ kg/m^3$$

Mass of air, $\dot{m}_{air} = \rho \dot{V}$

$$\dot{m}_{air} = 1.129 \times 0.6$$

$$\dot{m}_{air} = 0.677\ kg/s$$

Step 4 of 4

In an ideal heat exchanger, all the heat lost by the working fluid will be equal to heat gained by the cooling fluid.

$$\dot{Q}_{in} = \dot{m}_{air} (C_P)_{air} (T_2 - T_1)$$

$$(T_2 - T_1) = \frac{\dot{Q}_{in}}{\dot{m}_{air} (C_P)_{air}}$$

$$T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}_{air} (C_P)_{air}}$$

$$T_2 = 20 + \frac{67.93}{0.677 \times 1.005}$$

$$T_2 = 20 + 99.84$$

$$T_2 = 119.84\ C$$

Temperature at exit is $\boxed{T_2 = 119.84\ C}$

Question: 6.079P

An open feedwater heater heats the feedwater by mixing it with hot steam. Consider an electrical power plant with an open feedwater heater that mixes 0.1 lbm/s of steam at 10 psia and 200°F with 2.0 lbm/s of feedwater at 10 psia and 100°F to produce 10 psia and 120°F feedwater at the outlet. The diameter of the outlet pipe is 0.5 ft. Determine the mass flow rate and feedwater velocity at the outlet. Would the outlet flow rate and velocity be significantly different if the temperature at the outlet were 180°F?

Step 1 of 6

Given

Inlet conditions of cool feed water

Mass flow rate $\dot{m}_w = 2$ lbm/s

Pressure $P_1 = 10$ Psia

Temperature $T_1 = 100°F$

Steam inlet conditions

Mass flow rate $\dot{m}_{steam} = 0.1$ lbm/s

Pressure $P_2 = 10$ Psia

Temperature $T_2 = 200°F$

Properties of warm feed water at exit

Pressure $P_3 = 10$ Psia

Temperature $T_3 = 120°F$

The outlet diameter of the pipe $d = 0.5$ ft

Step 2 of 6

The outlet mass flow rate is the sum of both feed water and steam mass flow rates respectively

By mass balance

$$\dot{m}_{in} = \dot{m}_{out}$$

Therefore

$$\dot{m}_{outlet} = \dot{m}_w + \dot{m}_s$$

$$\dot{m}_{outlet} = 2 + 0.1$$

$$\boxed{\dot{m}_{outlet} = 2.1 \text{ lbm/s}}$$

Step 3 of 6

The mass flow rate is expressed as

$$\dot{m} = \frac{AV}{v} \text{ lbm/s}$$

At $T_3 = 120°F$ from saturated water tables

$$v_3 = v_f = 0.0162 \text{ ft}^3/\text{lbm}$$

And area

$$A = \pi \frac{d^2}{4}$$

$$A = \pi \frac{0.5^2}{4}$$

Step 4 of 6

Therefore the outlet velocity is

$$V_3 = \frac{\dot{m}_{outlet} v_3}{A_3}$$

$$= \frac{2.1 \times 0.0162}{\pi \frac{0.5^2}{4}}$$

$$\boxed{V_3 = 0.173 \text{ ft/s}}$$

Step 5 of 6

If the temperature at outlet is $T_3 = 180°F$

The mass flow rate remains the same value $\dot{m}_{outlet} = 2.1$ lbm/s

The outlet velocity changes due to change in specific volume

At $T_3 = 180°F$ from saturated water tables

$$v_3 = 0.01651 \text{ ft}^3/\text{lbm}$$

Step 6 of 6

$$V_3 = \frac{\dot{m} v_2}{A}$$

$$= \frac{2.1 \times 0.01651}{\pi \frac{0.5^2}{4}}$$

$$\boxed{V_3 = 0.1765 \text{ ft/s}}$$

If the temperature changes from $T_3 = 120^0 F$ to $T_3 = 180^0 F$ there is no significant change in velocity

Question: 6.080P

In a steam heating system, air is heated by being passed over some tubes through which steam flows steadily. Steam enters the heat exchanger at 30 psia and 400°F at a rate of 15 lbm/min and leaves at 25 psia and 212°F. Air enters at 14.7 psia and 80°F and leaves at 130°F. Determine the volume flow rate of air at the inlet.

Step 1 of 8

Given

Properties of steam

Inlet pressure $P_1 = 30$ psia

Inlet temperature $T_1 = 400°F$

Exit pressure $P_2 = 25$ psia

Exit temperature $T_2 = 212°F$

Mass flow rate of steam $\dot{m}_s = 15$ lbm/min

$$= \frac{15}{60}$$

$$\dot{m}_s = 0.25 \text{ lbm/s}$$

Properties of air

Inlet pressure $P_3 = 14.7$ psia

Inlet temperature $T_3 = 80°F$

Exit temperature $T_4 = 130°F$

Step 2 of 8

Mixing chambers are usually well insulated ($q = 0$) and usually do not involve any kind of work ($w = 0$). Also, the kinetic and potential energies of the fluid streams are usually negligible ($ke = 0, pe = 0$).

Applying mass balance and energy balance for the mixing chamber system, we obtain the equations as

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 8

In our problem we have two inlets and two exits, the mass flow rate of air and steam remains same at inlet and exit.

Therefore from above energy equation

$$\dot{m}_a h_3 + \dot{m}_s h_1 = \dot{m}_a h_4 + \dot{m}_s h_2$$

$$\dot{m}_a (h_3 - h_4) = \dot{m}_s (h_2 - h_1) \quad \text{---------- (1)}$$

Step 4 of 8

At $P_1 = 30$ psia and $T_1 = 400°F$ from superheated water tables by using interpolation

$$h_1 = 1239.3 + (1236.5 - 1239.3)\left(\frac{30-20}{40-20}\right)$$

$$h_1 = 1237.9 \text{ Btu/lbm}$$

At $P_2 = 25$ psia from saturated water tables

$$T_{sat} = 240.03^0 F$$

And given $T_2 < T_{sat}$

So at $T_2 = 212°F$ from saturated water tables

$$h_2 = h_f = 180.21 \text{ Btu/lbm}$$

Step 5 of 8

The specific heat at constant pressure of air can be obtained at average temperature of air

$$T_{avg} = \frac{T_3 + T_4}{2}$$

$$T_{avg} = \frac{80 + 130}{2}$$

$$T_{avg} = 105^0 F$$

At $T_{avg} = 105^0 F$ from ideal gas specific heats of common gases

$$c_p = 0.24 \text{ Btu/lbm.R}$$

From molar mass, gas constant table

The gas constant for air $R = 0.3704$ psia.ft^3/lbm.R

Step 6 of 8

In equation 1 the enthalpy values we can write it as $h = c_p T$, on substituting all the known values in equation 1

$$\dot{m}_a (h_3 - h_4) = \dot{m}_s (h_2 - h_1)$$

$$\dot{m}_a c_p (T_3 - T_4) = \dot{m}_s (h_2 - h_1)$$

$$\dot{m}_a 0.24(80 - 130) = 0.25(180.21 - 1237.9)$$

$$\dot{m}_a = \frac{1057.69}{50 \times 0.24} \times 0.25$$

$$\dot{m}_a = 22.035 \text{ lbm/s}$$

Step 7 of 8

We know that volume flow rate of air is expressed as

$$\dot{V} = \frac{\dot{m}}{\rho} \text{ ft}^3/\text{s}$$

And density of air at inlet can be determined by using ideal gas equation

$$\frac{P}{\rho} = RT$$

$$\rho_3 = \frac{P_3}{RT_3}$$

$$= \frac{14.7}{(0.3704)(80+460)}$$

$$\rho_3 = 0.0735 \text{ lbm/ft}^3$$

Step 8 of 8

Therefore volume flow rate of air at the inlet is

$$\dot{V}_3 = \frac{22.035}{0.0735}$$

$$\boxed{\dot{V}_3 = 299.795 \text{ ft}^3/\text{s}}$$

Question: 6.081P

Hot exhaust gases of an internal combustion engine are to be used to produce saturated water vapor at 2 MPa pressure. The exhaust gases enter the heat exchanger at 400°C at a rate of 32 kg/min while water enters at 15°C. The heat exchanger is not well insulated, and it is estimated that 10 percent of heat given up by the exhaust gases is lost to the surroundings. If the mass flow rate of the exhaust gases is 15 times that of the water, determine (a) the temperature of the exhaust gases at the heat exchanger exit and (b) the rate of heat transfer to the water. Use the constant specific heat properties of air for the exhaust gases.

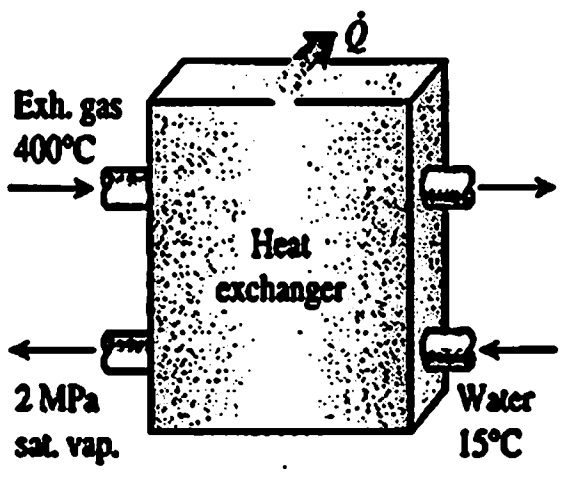


FIGURE P6–81

Step 1 of 4

Calculate the mass flow rate of exhaust gases as follows:

$$\dot{m}_{gases} = 32 \text{ kg/min}$$
$$= \frac{32}{60}$$
$$\dot{m}_{gases} = 0.534 \text{ kg/s}$$

Heat lost to the surroundings $\dot{Q}_{surr} = 0.1\dot{Q}_{gases}$

Calculate the mass flow rate of the water as follows:

$$\dot{m}_w = \frac{\dot{m}_{gases}}{15}$$

Substitute 0.534 kg/s for $\dot{m}_{gases}$.

$$\dot{m}_w = \frac{(0.534)}{15}$$
$$= 0.0356 \text{ kg/s}$$

Step 2 of 4

(b)

Obtain the enthalpy of water at inlet from the table A-4, "Saturated water-Temperature table" in the textbook.

$$h_1 = 62.98 \text{ kJ/kg}$$

Obtain the enthalpy of water at inlet from the table A-5, "Saturated water-Pressure table" in the textbook.

$$h_2 = 2798.3 \text{ kJ/kg}$$

Calculate the rate of heat transfer to the water as follows:

$$\dot{Q}_w = \dot{m}_w(h_2 - h_1)$$

Substitute 0.0356 kg/s for $\dot{m}_w$, 62.98 kJ/kg for h_1, and 2798.3 kJ/kg for h_2.

$$\dot{Q}_w = (0.0355)(2798.3 - 62.98)$$
$$= 97.1 \text{ kW}$$

Therefore, the rate of heat transfer to the water is $\boxed{97.1 \text{ kW}}$.

Step 3 of 4

(a)

Since, there is a 10 percent heat is lost to surroundings and the remaining 90 percent will be absorbed by the water.

Therefore,

$$Q_{exhaust} = Q_{surr} + Q_w$$
$$Q_{exhaust} = 0.1Q_{exhaust} + Q_w$$
$$Q_{exhaust} = \frac{Q_w}{0.9}$$

Substitute 97.1 kW for Q_w.

$$Q_{exhaust} = \frac{97.1}{0.9}$$
$$= 107.8 \text{ kW}$$

Step 4 of 4

Write the equation for heat transfer rate in exhaust gases as follows:

$$Q_{exhaust} = \dot{m}_{gases} c_p (t_{in} - t_{out})$$

Substitute $1.005 \text{ kJ/kg} \cdot \text{K}$ for c_p, 0.534 kg/s for $\dot{m}_{gases}$, 107.8 kW for $Q_{exhaust}$, and $400°\text{C}$ for t_{in}.

$$107.8 \times 10^3 = 0.534 \times 1.005 \times (400 - t_{out})$$
$$t_{out} = 199.14°\text{C}$$

Therefore, the temperature of exhaust gases exit is $\boxed{199.14°\text{C}}$.

Question: 6.082P

A chilled-water heat-exchange unit is designed to cool 5 m3/s of air at 100 kPa and 30°C to 100 kPa and 18°C by using water at 8°C. Determine the maximum water outlet temperature when the mass flow rate of the water is 2 kg/s.

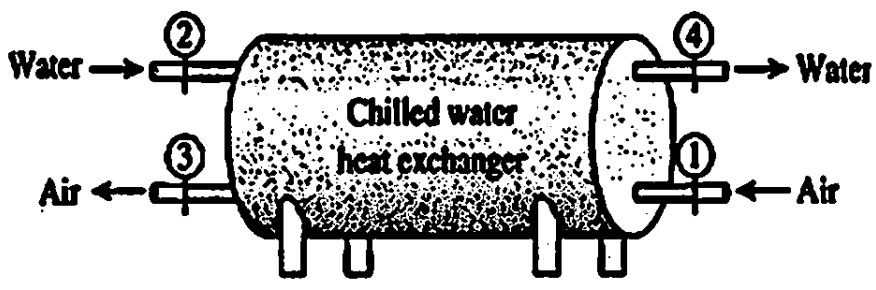


FIGURE P6–82

Step 1 of 4

Heat exchangers typically involve no work interactions ($w=0$) and negligible kinetic and potential energy changes ($ke=0, pe=0$) for each fluid stream.

Apply mass balance for the heat exchanger system, to obtain the equation:

$$\dot{m}_{in}=\dot{m}_{out}$$

Here, input and output mass flow rates are $\dot{m}_{in}$ and $\dot{m}_{out}$ respectively.

Apply energy balance for the heat exchanger system, to obtain the equation:

$$\dot{E}_{in}=\dot{E}_{out}$$
$$\dot{Q}+\dot{m}_{in}h_{in}=\dot{m}_{out}h_{out}$$

Here, input and output energy flow rate are $\dot{E}_{in}$ and $\dot{E}_{out}$ respectively; heat flow rate is $\dot{Q}$; input and output enthalpy are h_{in} and h_{out} .

Step 2 of 4

Consider the mass flow rate of air:

$$\dot{m}_{air}=\rho_1\dot{V}_{air} \quad \text{..... (1)}$$

Here, the density of air is ρ_1 , and volume flow rate of air is $\dot{V}_{air}$.

Consider the ideal gas equation:

$$\rho_1=\frac{P_1}{RT_1}$$

Assume gas constant for air as;

$$R=0.287\ \text{kJ/kg.K}$$

Here, inlet pressure of air is P_1 , and inlet temperature of air is T_1 .

Substitute $100\ \text{kPa}$ for P_1 , $30°\text{C}$ for T_1 , $0.287\ \text{kJ/kg.K}$ for R:

$$\rho_1=\frac{100}{(0.287)(30+273)}$$
$$=1.15\ \text{kg/m}^3$$

Substitute $1.15\ \text{kg/m}^3$ for ρ_1 , and $5\ \text{m}^3/\text{s}$ for $\dot{V}_{air}$ in equation (1):

$$\dot{m}_{air}=\rho_1\dot{V}_{air}$$
$$=(1.15)(5)$$
$$=5.75\ \text{kg/s}$$

Step 3 of 4

Take the specific heat value at average temperature of air:

$$T_{avg}=\frac{T_1+T_3}{2}$$

Here, the exit temperature is T_3 .

Substitute $30°\text{C}$ for T_1 , and $18°\text{C}$ for T_3 :

$$T_{avg}=\frac{30+18}{2}$$
$$=24°\text{C}$$
$$=297\ \text{K}$$

At $T_{avg}=297\text{K}$, consider the ideal gas properties of air:

$$c_{p_{air}}=1.003+(1.005-1.003)\left(\frac{297-250}{300-250}\right)$$
$$=1.00488\ \text{kJ/kg.K}$$

Consider the rate of transfer in the air stream:

$$\dot{Q}_{air}+\dot{m}_{air}h_1=\dot{m}_{air}h_3$$
$$\dot{Q}_{air}=\dot{m}_{air}(h_3-h_1)$$
$$=\dot{m}_{air}c_{p_{air}}(T_3-T_1)$$

Substitute $1.00488\ \text{kJ/kg.K}$ for $c_{p_{air}}$, $5.75\ \text{kg/s}$ for $\dot{m}_{air}$, $30°\text{C}$ for T_1 , and $18°\text{C}$ for T_3 :

$$\dot{Q}_{air}=5.75\times1.00488(18-30)$$
$$=-69.336\ \text{kW}$$

Step 4 of 4

Heat lost by the air stream is equal to the heat gained by the water stream:

$$\dot{Q}_{water}=\dot{m}_w(h_4-h_2)$$
$$=\dot{m}_wc_{p_w}(T_4-T_2)$$

Here, mass flow rate of water is $\dot{m}_w$.

Substitute $69.336\ \text{kW}$ for $\dot{Q}_{water}$, $2\ \text{kg/s}$ for $\dot{m}_w$, and $8°\text{C}$ for T_2 :

$$69.336=2\times4.18(T_4-8)$$
$$T_4=16.29°\text{C}$$

Therefore, the exit temperature of water is $\boxed{16.29°\text{C}}$.

Question: 6.083P

The condenser of a refrigeration cycle is basically a heat exchanger in which a refrigerant is condensed by rejecting heat to a cooling fluid. Refrigerant-134a enters a condenser at 1200 kPa and 85°C with a flow rate of 0.042 kg/s and leaves at the same pressure subcooled by 6.3°C. The condensation is done by cooling water which experiences a temperature rise of 12°C in the condenser. Determine (a) the rate of heat transferred to the water in the condenser, in kJ/min and (b) the mass flow rate of water, in kg/min.

Step 1 of 5

Given

Inlet pressure of R-134a, $P_1 = 1200\ \text{kPa}$

Inlet temperature of R-134a, $T_1 = 85°\text{C}$

Mass flow rate of R-134a $\dot{m}_{ref} = 0.042\ \text{kg/s}$

Exit pressure of R-134a, $P_2 = 1200\ \text{kPa}$

Exit state of R-134a is sub cooled by $6.3°\text{C}$

The raise of water temperature, $\Delta T_w = 12°\text{C}$

Step 2 of 5

Condenser typically involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream.

Applying mass balance,

$$\dot{m}_{in} = \dot{m}_{out}$$

Energy balance for the condenser system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{Q} + \dot{m}_{out} h_{out}$$

Step 3 of 5

(a)

The properties of R-134a from superheated tables,

At 1200 kPa and $85°\text{C}$, Enthalpy $h_1 = 316.73\ \text{kJ/kg}$

The properties of R-134a from saturated tables,

At 1200 kPa, saturation temperature, $T_{sat} = T_1 = 46.29°\text{C}$

At final state, R-134a is sub cooled by 3.6, $T_2 = T_1 - 6.3$

$$T_2 = 46.29 - 6.3$$

$$T_2 = 40°\text{C}$$

At 1200 kPa and $40°\text{C}$, Enthalpy $h_2 = h_f = 108.26\ \text{kJ/kg}$

Step 4 of 5

The rate of heat rejected to water is

$$\dot{Q}_{out} = \dot{m}_{ref}(h_1 - h_2)$$

$$\dot{Q}_{out} = 0.042 \times (316.72 - 108.26)$$

$$\dot{Q}_{out} = 8.76\ \text{kJ/s}$$

$$\dot{Q}_{out} = 8.76 \times 60\ \text{kJ/min}$$

$$\dot{Q}_{out} = 525.32\ \text{kJ/min}$$

Hence rate of heat rejected to water is $\boxed{\dot{Q}_{out} = 525.32\ \text{kJ/min}}$

Step 5 of 5

(b)

Heat transferred to water is $\dot{Q}_{out} = \dot{m}_w C_p \Delta T_w$

$$\dot{m}_w = \frac{\dot{Q}_{out}}{C_p \Delta T_w}$$

$$\dot{m}_w = \frac{525.32}{4.18 \times 12}$$

$$\dot{m}_w = 10.48\ \text{kg/min}$$

Hence mass flow rate of water is $\boxed{\dot{m}_w = 10.48\ \text{kg/min}}$

Question: 6.084P

The evaporator of a refrigeration cycle is basically a heat exchanger in which a refrigerant is evaporated by absorbing heat from a fluid. Refrigerant-22 enters an evaporator at 200 kPa with a quality of 22 percent and a flow rate of 2.25 L/h. R-22 leaves the evaporator at the same pressure superheated by 5°C. The refrigerant is evaporated by absorbing heat from air whose flow rate is 0.5 kg/s. Determine (a) the rate of heat absorbed from the air and (b) the temperature change of air. The properties of R-22 at the inlet and exit of the condenser are h1 = 220.2 kJ/kg, *V*1 = 0.0253 m3/kg, and h2 = 398.0 kJ/kg.

Step 1 of 5

Given

Inlet pressure of R-22, $P_1 = 200\ \text{kPa}$

Inlet quality of R-22, $x_1 = 0.22$

Inlet volume flow rate of R-22, $\dot{V} = 2.25\ \text{L/h}$

$$\dot{V} = \frac{2.25\times10^{-3}}{3600}\ \text{m}^3/\text{s}$$

$$\dot{V} = 6.25\times10^{-7}\ \text{m}^3/\text{s}$$

Exit state of R-22 is superheated by 5°C, $T_2 = T_1 + 5$

Mass flow rate of air, $\dot{m}_{air} = 0.5\ \text{kg/s}$

Step 2 of 5

Evaporator typically involve no work interactions ($w = 0$) and negligible kinetic and potential energy changes ($ke = 0, pe = 0$) for each fluid stream.

Applying mass balance,

$$\dot{m}_{in} = \dot{m}_{out}$$

Energy balance for the Evaporator system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in}h_{in} = \dot{Q} + \dot{m}_{out}h_{out}$$

Step 3 of 5

(a)

The properties of R-22 from saturated tables,

At 200 kPa and $x_1 = 0.22$, Enthalpy $h_1 = 220.2\ \text{kJ/kg}$

Saturation temperature, $T_{sat} = T_1$

Specific volume, $v_1 = 0.0253\ \text{m}^3/\text{kg}$

The properties of R-22 from superheated tables,

At 200 kPa and T_2 Exit enthalpy $h_2 = 398\ \text{kJ/kg}$

Step 4 of 5

The mass flow rate of refrigerant is expressed as

$$\dot{m}_{ref} = \frac{\dot{V}}{v_1}$$

$$\dot{m}_{ref} = \frac{6.25\times10^{-7}}{0.0253}$$

$$\dot{m}_{ref} = 2.47\times10^{-5}\ \text{kg/s}$$

The rate of heat absorbed from air is

$$\dot{Q}_{in} = \dot{m}_{ref}(h_2 - h_1)$$

$$\dot{Q}_{in} = \left(2.47\times10^{-5}\right)\times(398 - 220.2)$$

$$\dot{Q}_{in} = 4.39\times10^{-3}\ \text{kJ/s}$$

Hence rate of heat absorbed is $\boxed{\dot{Q}_{in} = 4.39\times10^{-3}\ \text{kJ/s}}$

Step 5 of 5

(b)

Heat transferred from air is $\dot{Q}_{in} = \dot{m}_{air}C_p\Delta T_{air}$

$$\Delta T_{air} = \frac{\dot{Q}_{in}}{\dot{m}_{air}C_p}$$

$$\Delta T_{air} = 8.73\times10^{-3}\ ^\circ\text{C}$$

The change in temperature for air $\boxed{\Delta T_{air} = 8.73\times10^{-3}\ ^\circ\text{C}}$

Question: 6.085P

Argon steadily flows into a constant pressure heater at 300 K and 100 kPa with a mass flow rate of 6.24 kg/s. Heat transfer in the rate of 150 kW is supplied to the argon as it flows through the heater, (a)Determine the argon temperature at the heater exit, in °C. (b)Determine the argon volume flow rate at the heater exit, in m3/s.

Step 1 of 4

Given

Inlet pressure of Argon, $P_1 = 100\ \text{kPa}$

Inlet temperature of Argon, $T_1 = 300\ \text{K}$

The mass flow rate of Argon, $\dot{m}_{Argon} = 6.24\ \text{kg/s}$

Heat transfer rate to Argon, $\dot{Q}_{in} = 150\ \text{kW}$

Step 2 of 4

(a)

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

But Enthalpy, $h = C_p T$. So

$$\dot{Q} + \dot{m}_{Argon} C_p T_1 = \dot{m}_{Argon} C_p T_2$$

$$\dot{Q} = \dot{m}_{Argon} C_p (T_2 - T_1) \qquad \text{---- (1)}$$

Step 3 of 4

From ideal gas properties table for Argon,

$$R = 0.2801\ \text{kJ/kgK}$$

$$C_p = 0.5203\ \text{kJ/kg°C}$$

Substituting all the values in equation (1),

$$150 = 6.24 \times 0.5203 \times (T_2 - 300)$$

$$(T_2 - 300) = 46.2$$

$$T_2 = 46.2 + 300$$

$$T_2 = 346.2\ ^0\text{C}$$

Hence temperature of the Argon at the exit is $\boxed{T_2 = 346.2\ ^0\text{C}}$

Step 4 of 4

(b)

From Ideal gas equation, $Pv = RT$. We get

$$v_2 = \frac{RT_2}{P_2}$$

$$v_2 = \frac{0.2081 \times 346.2}{100}$$

$$v_2 = 0.7204\ \text{m}^3/\text{kg}$$

Mass flow rate of argon is $\dot{m} = \frac{\dot{V}_2}{v_2}$. Therfore,

$$\dot{V}_2 = \dot{m} v_2$$

$$\dot{V}_2 = 6.24 \times 0.7204$$

$$\dot{V}_2 = 4.5\ \text{m}^3/\text{s}$$

Hence final volume of argon at exit is $\boxed{\dot{V}_2 = 4.5\ \text{m}^3/\text{s}}$

Question: 6.086P

Saturated liquid water is heated in a steady-flow steam boiler at a constant pressure of 2 MPa at a rate of 4 kg/s to an outlet temperature of 250°C. Determine the rate of heat transfer in the boiler.

Step 1 of 3

Given

The inlet pressure of the saturated water, $P = 2\ \text{MPa}$

The inlet mass flow rate of the saturated water, $\dot{m}_w = 4\ \text{kg/s}$

The exit pressure of the superheated water, $P = 2\ \text{MPa}$

The exit temperature of the superheated water, $T_2 = 250°\text{C}$

Step 2 of 3

From mass balance equation,

$$\dot{m}_{in} = \dot{m}_{out}$$

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

$$\dot{Q} = \dot{m}(h_{out} - h_{in}) \qquad \text{---- (1)}$$

Step 3 of 3

From saturated water tables, at 2 MPa,

Saturated liquid enthalpy, $h_1 = h_f = 908.47\ \text{kJ/kg}$

From superheated water tables, at 2 MPa, and 250°C,

Enthalpy, $h_2 = 2903.3\ \text{kJ/kg}$

Substituting all the values in equation (1), we get

$$\dot{Q} = 4 \times (2903.3 - 908.47)$$

$$\dot{Q} = 7979.32\ \text{kW}$$

$$\dot{Q} = 7.98\ \text{MW}$$

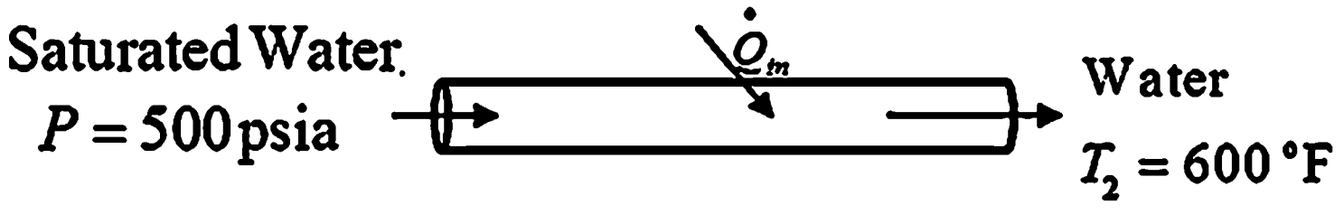
Hence rate of heat transfer in the boiler is $\boxed{\dot{Q} = 7.98\ \text{MW}}$

Question: 6.087P

Water enters a boiler at 500 psia as a saturated liquid and leaves at 600°F at the same pressure. Calculate the heat transfer per unit mass of water.

Step 1 of 3

Draw the flow process through pipe system as below:



Here, pressure of the boiler is P and exit temperature of the water is T_2.

Step 2 of 3

Obtain the properties of steam from saturated water tables at pressure $P_1 = 500\,\text{psia}$ and quality and quality $x_1 = 0$

Enthalpy, $h_1 = h_f$
$= 449.51\,\text{kJ/kg}$

Obtain the properties of steam from the Super - heated water tables at $P_2 = 500\,\text{psia}$ and $T_2 = 600°\text{F}$

$h_2 = 1298.6\,\text{Btu/lbm}$

Step 3 of 3

Consider control volume as the steam pipe as the system, since mass crosses the boundary. There is only one inlet and one exit. Hence, mass entering is equal to mass leaving.

$\dot{m}_1 = \dot{m}_2$

Apply energy conservation for the system

For a steady state condition energy stored within the system is zero.

$\dot{E}_{in} = \dot{E}_{out}$

$\dot{m}\left(h_1 + V_1^2/2\right)_{in} + \dot{Q}_{in} = \dot{Q}\left(h_2 + V_2^2/2\right)_{out}$

$\dot{m}h_1 + 0 + \dot{Q}_{in} = \dot{m}h_2$

$\dot{Q}_{in} = \dot{m}(h_2 - h_1)$

$= (h_2 - h_1)$

Calculate the heat transfer per unit mass of water as shown below:

$\dot{Q}_{in} = (h_2 - h_1)$

$= 1298.6\,\text{Btu/lbm} - 449.5\,\text{Btu/lbm}$

$= 849.1\,\text{Btu/lbm}$

Therefore, the heat transfer per unit mass of water is $\boxed{849.1\,\text{Btu/lbm}}$.

Question: 6.088P

A 110-volt electrical heater is used to warm 0.3 m3/s of air at 100 kPa and 15°C to 100 kPa and 30°C. How much current in amperes must be supplied to this heater?

Step 1 of 5

Given

The volume flow rate of air $\dot{V}_1 = 0.3\ m^3/s$

Inlet temperature of air $T_1 = 15°C$

Exit temperature of air $T_2 = 30°C$

Inlet and exit pressures of air $P_1 = P_2 = 100$ kPa

The voltage for an electrical heater $V = 110$ volts

Step 2 of 5

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 5

The mass flow rate of air is expressed as

$$\dot{m} = \rho \dot{V}_1$$

From ideal gas equation

$$\rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{100}{(0.287)(15+273)} \qquad \text{(For air } R = 0.287 \text{ kJ/kg.K)}$$

$$\rho_1 = 1.21\ kg/m^3$$

Therefore

$$\dot{m} = \rho \dot{V}_1$$

$$\dot{m}_{air} = (1.21)(0.3)$$

$$\dot{m}_{air} = 0.363\ kg/s$$

Step 4 of 5

Taking specific heat of air $c_p = 1.005$ kJ/kg°C

And enthalpy we can write it as $h = c_p T$

By using mass balance

$\dot{m}_{in} = \dot{m}_{out}$

Then the energy equation becomes

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

$$\dot{Q} = 0.363(1.005)(30-15)$$

$$\dot{Q} = 5.47\ kW$$

Step 5 of 5

The amount of electrical energy supplied will be converted into heat.

i.e. $\dot{Q} = VI$

Where

V is voltage and

I is current supplied to electrical heater.

Therefore the current I is

$$I = \frac{\dot{Q}}{V}$$

$$= \frac{5.47 \times 10^3}{110}$$

$$\boxed{I = 49.727\ A}$$

Question: 6.089P

The fan on a personal computer draws 0.5 ft3/s of air at 14.7 psia and 70°F through the box containing the CPU and other components. Air leaves at 14.7 psia and 80°F. Calculate the electrical power, in kW, dissipated by the PC components.



FIGURE P6–89E

© Vol. 55/PhotoDisc/Getty Images.

Step 1 of 4

Given

The volume flow rate of air $\dot{V}_{air} = 0.5\ ft^3/s$

Inlet temperature of air $T_1 = 70°F$

Exit temperature of air $T_2 = 80°F$

Inlet pressure of air $P_1 = 14.7$ psia

Exit pressure of air $P_2 = 14.7$ psia

Step 2 of 4

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

Step 3 of 4

The mass flow rate is expressed as

$$\dot{m} = \rho \dot{V}_{air}$$

From ideal gas equation

$$\rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{14.7}{(0.3704)(70+460)} \qquad (\text{For air } R = 0.3704 \text{ psia.ft}^3/\text{lbm.R})$$

$$\rho_1 = 0.0748\ \text{lbm/ft}^3$$

Therefore

$$\dot{m} = \rho \dot{V}_{air}$$

$$\dot{m}_{air} = (0.0748)(0.5)$$

$$\dot{m}_{air} = 0.0374\ \text{kg/s}$$

Step 4 of 4

Taking specific heat of air $c_p = 0.24$ Btu/lbm°F

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

From energy equation

$$\dot{W} = \dot{m}_{out}(h_{out} - h_{in})$$

$$\dot{W} = \dot{m}_{air} c_p (T_2 - T_1)$$

$$\dot{W} = 0.0374(0.24)(80-70)$$

$$\dot{W} = 0.08976\ \text{Btu/s}$$

Since $(1\ \text{Btu/s} = 1.055\ \text{kW})$

$$\dot{W} = 0.08976 \times 1.055$$

$$\boxed{\dot{W} = 0.0946\ \text{kW}}$$

Question: 6.090P

A desktop computer is to be cooled by a fan. The electronic components of the computer consume 60 W of power under full-load conditions. The computer is to operate in environments at temperatures up to 45°C and at elevations up to 3400 m where the average atmospheric pressure is 66.63 kPa. The exit temperature of air is not to exceed 60°C to meet the reliability requirements. Also, the average velocity of air is not to exceed 110 m/min at the exit of the computer case where the fan is installed to keep the noise level down. Determine the flow rate of the fan that needs to be installed and the diameter of the casing of the fan.

Step 1 of 7

Given

The power consumed by electronic components $\dot{W} = -60\ \text{W} = -0.06\ \text{kW}$

Inlet temperature of air $T_1 = 45°\text{C}$

Exit temperature of air $T_2 = 60°\text{C}$

The atmospheric pressure of air $P = 66.63\ \text{kPa}$

The average velocity of air $V = 110\ \text{m/min}$

$$= \frac{110}{60}$$

$$V = 1.8334\ \text{m/s}$$

Step 2 of 7

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} + \dot{W} \qquad \text{(Assuming } q = 0 \text{ and } ke = 0, pe = 0\text{)}$$

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out}$$

Then energy equation becomes

$$\dot{m}_{in}\left(h_{in} - h_{out}\right) = \dot{W}$$

We know that $h = c_p T$

$$\dot{m}_{in}\, c_{p_{air}}\left(T_1 - T_2\right) = \dot{W}$$

Step 3 of 7

At average temperature of air

$$T_{avg} = \left(\frac{T_1 + T_2}{2}\right)$$

$$= \left(\frac{45 + 60}{2}\right)$$

$$T_{avg} = 52.5^0\text{C} = 325.5\text{K}$$

At $T_{avg} = 325.5\text{K}$ from ideal gas properties of air

$$c_{p_{air}} = 1.005 + (1.008 - 1.005)\left(\frac{325.5 - 300}{350 - 300}\right)$$

$$c_{p_{air}} = 1.0065\ \text{kJ/kg.K}$$

Step 4 of 7

Substituting all the values in energy equation we get

$$\dot{m}_{in}\, c_{p_{air}}\left(T_1 - T_2\right) = \dot{W}$$

$$\dot{m}(1.0065)(45 - 60) = -0.06$$

$$\dot{m} = 0.00397\ \text{kg/s}$$

Step 5 of 7

The mass flow rate is expressed as

$$\dot{m} = \rho \dot{V}_{air}$$

From ideal gas equation

$$\rho_2 = \frac{P}{RT_2}$$

$$= \frac{66.63}{(0.287)(60 + 273)} \qquad \text{(For air } R = 0.287\ \text{kJ/kg.K)}$$

$$\rho_2 = 0.697\ \text{kg/m}^3$$

Step 6 of 7

Therefore volume flow rate of fan which is installed at the exit of the CPU so,

$$\dot{V}_{air} = \frac{\dot{m}}{\rho}$$

$$\dot{V}_{air} = \frac{0.00397}{0.697}$$

$$= 0.00569\ \text{m}^3\text{/s}$$

$$\boxed{\dot{V}_{fan} = 0.00569\ \text{m}^3\text{/s}}$$

Step 7 of 7

The diameter of the fan casing is obtained as

$$\dot{m} = \rho A V$$

Therefore

$$A_2 = \frac{\dot{m}}{V \rho_2}$$

$$\pi \frac{d^2}{4} = \frac{0.00397}{1.8334(0.697)}$$

$$\boxed{d_2 = 0.0628\text{m}}$$

Question: 6.091P

Repeat Prob. 6–90 for a computer that consumes 100 W of power.

Step 1 of 7

Given

The power consumed by electronic components $\dot{W} = -100\ \text{W} = -0.1\text{kW}$

Inlet temperature of air $T_1 = 45°\text{C}$

Exit temperature of air $T_2 = 60°\text{C}$

The atmospheric pressure of air $P = 66.63\ \text{kPa}$

The average velocity of air $V = 110\ \text{m/min}$

$$= \frac{110}{60}$$

$$V = 1.8334\ \text{m/s}$$

Step 2 of 7

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} + \dot{W} \qquad \text{(Assuming } q = 0 \text{ and } ke = 0, pe = 0\text{)}$$

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out}$$

Then energy equation becomes

$$\dot{m}_{in}\left(h_{in} - h_{out}\right) = \dot{W}$$

We know that $h = c_p T$

$$\dot{m}_{in}\, c_{p_{air}}\left(T_1 - T_2\right) = \dot{W}$$

Step 3 of 7

At average temperature of air

$$T_{avg} = \left(\frac{T_1 + T_2}{2}\right)$$

$$= \left(\frac{45+60}{2}\right)$$

$$T_{avg} = 52.5^0\text{C} = 325.5\text{K}$$

At $T_{avg} = 325.5\text{K}$ from ideal gas properties of air

$$c_{p_{air}} = 1.005 + (1.008 - 1.005)\left(\frac{325.5-300}{350-300}\right)$$

$$c_{p_{air}} = 1.0065\ \text{kJ/kg.K}$$

Step 4 of 7

Substituting all the values in energy equation we get

$$\dot{m}_{in}\, c_{p_{air}}\left(T_1 - T_2\right) = \dot{W}$$

$$\dot{m}(1.0065)(45 - 60) = -0.1$$

$$\dot{m} = 0.00662\ \text{kg/s}$$

Step 5 of 7

The mass flow rate is expressed as

$$\dot{m} = \rho \dot{V}_{air}$$

From ideal gas equation

$$\rho_2 = \frac{P}{RT_2}$$

$$= \frac{66.63}{(0.287)(60+273)} \qquad \text{(For air } R = 0.287\ \text{kJ/kg.K)}$$

$$\rho_2 = 0.697\ \text{kg/m}^3$$

Step 6 of 7

Therefore volume flow rate of fan which is installed at the exit of the CPU so,

$$\dot{V}_{air} = \frac{\dot{m}}{\rho}$$

$$\dot{V}_{air} = \frac{0.00662}{0.697}$$

$$= 0.00949\ \text{m}^3\text{/s}$$

$$\boxed{\dot{V}_{fan} = 0.00949\ \text{m}^3\text{/s}}$$

Step 7 of 7

The diameter of the fan casing is obtained as

$$\dot{m} = \rho A V$$

Therefore

$$A_2 = \frac{\dot{m}}{V \rho_2}$$

$$\pi \frac{d^2}{4} = \frac{0.00662}{1.8334(0.697)}$$

$$\boxed{d_2 = 0.0812\text{m}}$$

Question:

6.092P

Water enters the tubes of a cold plate at 70°F with an average velocity of 40 ft/min and leaves at 105°F. The diameter of the tubes is 0.25 in. Assuming 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation, and the remaining 85 percent is removed by the cooling water, determine the amount of heat generated by the electronic devices mounted on the cold plate.

Step 1 of 4

Given

The inlet temperatures of water, $T_1 = 70°F$

The exit temperatures of water, $T_2 = 105°F$

The diameter of the tubes, $d = 0.25\ \text{in}$

The average velocity of water, $V = 40\ \text{ft/min}$

$$V = \frac{40}{60}$$

$$V = 0.667\ \text{ft/s}$$

The amount of the generated from the electronic device be $\dot{Q}$

Question:

Water enters the tubes of a cold plate at 70°F with an average velocity of 40 ft/min and leaves at 105°F. The diameter of the tubes is 0.25 in. Assuming 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation, and the remaining 85 percent is removed by the cooling water, determine the amount of heat generated by the electronic devices mounted on the cold plate.

Step 1 of 4

Given

The inlet temperatures of water, $T_1 = 70°F$

The exit temperatures of water, $T_2 = 105°F$

The diameter of the tubes, $d = 0.25\ \text{in}$

The average velocity of water, $V = 40\ \text{ft/min}$

$$V = \frac{40}{60}$$

$$V = 0.667\ \text{ft/s}$$

The amount of the generated from the electronic device be $\dot{Q}$

Also $1\ \text{Btu/hr} = 0.000293\ \text{kW}$. Hence

$$\dot{Q}_{water} = 1779.05 \times 0.000293\ \text{kW}$$

$$\dot{Q}_{water} = 0.52126\ \text{kW}$$

Step 4 of 4

Energy received by water is 85% of total energy. Hence

$$\dot{Q}_{water} = \frac{\dot{Q}_{total}}{100} \times 85$$

$$\dot{Q}_{total} = \frac{\dot{Q}_{water}}{0.85}$$

$$\dot{Q}_{total} = \frac{0.52126}{0.85}$$

$$\dot{Q}_{total} = 0.6132\ \text{kW}$$

$$\dot{Q}_{total} = 613.2\ \text{W}$$

Hence total energy generated is $\boxed{\dot{Q}_{total} = 613.2\ \text{W}}$

Question: 6.093P

The components of an electronic system dissipating 180 W are located in a 1.4-m-long horizontal duct whose cross section is 20 cm × 20 cm. The components in the duct are cooled by forced air that enters the duct at 30°C and 1 atm at a rate of 0.6 m3/min and leaves at 40°C. Determine the rate of heat transfer from the outer surfaces of the duct to the ambient.

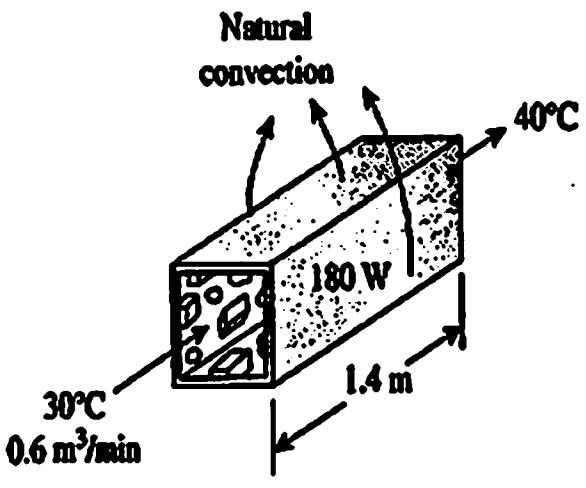


FIGURE P6–93

Step 1 of 6

Given

The heat dissipated by electronic components $\dot{Q}_{ele} = 180\ \text{W}$

The inlet temperature of air $T_1 = 30°\text{C}$

Exit temperature of air $T_2 = 40°\text{C}$

The atmospheric pressures of air $P = 1\ \text{atm} = 101.325\ \text{kPa}$

The volume flow rate of air at inlet $\dot{V}_1 = 0.6\ \text{m}^3/\text{min}$

Length of the duct $L = 1.4\,\text{m}$

Cross section of the duct $A = 20 \times 20\,\text{cm}^2$

Step 2 of 6

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} \quad (w = 0 \text{ And } \Delta ke = 0, \Delta pe = 0)$$

Using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

Then

$$\dot{Q} = \dot{m}_{air}(h_{out} - h_{in})$$

We know that $h = c_p T$

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

Step 3 of 6

Taking specific heat of air at average temperature

$$T_{avg} = \frac{40+30}{2}$$

$$T_{avg} = 35^0\text{C} = 308\,\text{K}$$

At $T_{avg} = 308\,\text{K}$ from ideal gas properties of air

$$c_p = 1.005 + (1.008 - 1.005)\left(\frac{308-300}{350-300}\right)$$

$$c_p = 1.00548\ \text{kJ/kg°C}$$

Step 4 of 6

Substituting all the values in above equation we get the amount of heat transferred occurred due to forced air

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

The mass flow rate of air is

$$\dot{m} = \rho \dot{V}_{air}$$

From ideal gas equation

$$\rho_1 = \frac{P}{RT_1}$$

$$= \frac{101.325}{(0.287)(30+273)} \qquad (\text{For air } R = 0.287\ \text{kJ/kg.K})$$

$$\rho_1 = 1.165\ \text{kg/m}^3$$

Step 5 of 6

Therefore rate of heat transfer due to forced air is

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

$$\dot{Q} = \rho_1 \dot{V}_{air} c_p (T_2 - T_1)$$

$$\dot{Q} = (1.165)\left(\frac{0.6}{60}\right)(1.00548)(40-30)$$

$$\dot{Q} = 0.117\ \text{kW}$$

$$\dot{Q} = 117\ \text{W}$$

Step 6 of 6

Total rate of heat transfer=rate of heat transfer due to forced air + rate of heat transfer from the outer surface of the duct

$$\dot{Q}_{ele} = \dot{Q} + \dot{Q}_{atm}$$

$$\dot{Q}_{atm} = \dot{Q}_{ele} - \dot{Q}$$

$$\dot{Q}_{atm} = 180 - 117$$

$$= 63\ \text{W}$$

$$\boxed{\dot{Q}_{atm} = 63\ \text{W}}$$

Question: 6.094P

Repeat Prob. 6–93 for a circular horizontal duct of diameter 10 cm.

Step 1 of 6

Given

The heat dissipated by electronic components $\dot{Q}_{ele} = 180\ \text{W}$

The inlet temperature of air $T_1 = 30°\text{C}$

Exit temperature of air $T_2 = 40°\text{C}$

The atmospheric pressures of air $P = 1\ \text{atm} = 101.325\ \text{kPa}$

The volume flow rate of air at inlet $\dot{V}_1 = 0.6\ \text{m}^3/\text{min}$

Length of the duct $L = 1.4\,\text{m}$

Diameter of the duct $A = 20 \times 20\,\text{cm}^2$

Step 2 of 6

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} \quad (w = 0 \text{ And } \Delta ke = 0, \Delta pe = 0)$$

Using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

Then

$$\dot{Q} = \dot{m}_{air}(h_{out} - h_{in})$$

We know that $h = c_p T$

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

Step 3 of 6

Taking specific heat of air at average temperature

$$T_{avg} = \frac{40+30}{2}$$

$$T_{avg} = 35^0\text{C} = 308\,\text{K}$$

At $T_{avg} = 308\,\text{K}$ from ideal gas properties of air

$$c_p = 1.005 + (1.008 - 1.005)\left(\frac{308-300}{350-300}\right)$$

$$c_p = 1.00548\ \text{kJ/kg°C}$$

Step 4 of 6

Substituting all the values in above equation we get the amount of heat transferred occurred due to forced air

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

The mass flow rate of air is

$$\dot{m} = \rho \dot{V}_{air}$$

From ideal gas equation

$$\rho_1 = \frac{P}{RT_1}$$

$$= \frac{101.325}{(0.287)(30+273)} \qquad (\text{For air } R = 0.287\ \text{kJ/kg.K})$$

$$\rho_1 = 1.165\ \text{kg/m}^3$$

Step 5 of 6

Therefore rate of heat transfer due to forced air is

$$\dot{Q} = \dot{m}_{air} c_p (T_2 - T_1)$$

$$\dot{Q} = \rho_1 \dot{V}_{air} c_p (T_2 - T_1)$$

$$\dot{Q} = (1.165)\left(\frac{0.6}{60}\right)(1.00548)(40-30)$$

$$\dot{Q} = 0.117\ \text{kW}$$

$$\dot{Q} = 117\ \text{W}$$

Step 6 of 6

Total rate of heat transfer=rate of heat transfer due to forced air + rate of heat transfer from the outer surface of the duct

$$\dot{Q}_{elec} = \dot{Q} + \dot{Q}_{atm}$$

$$\dot{Q}_{atm} = \dot{Q}_{ele} - \dot{Q}$$

$$\dot{Q}_{atm} = 180 - 117$$

$$= 63\ \text{W}$$

$$\boxed{\dot{Q}_{atm} = 63\ \text{W}}$$

If it is circular duct, or square duct the rate of heat from the outer surface of the duct remains same. Since the flow rate doesn't change.

Question: 6.095P

Consider a hollow-core printed circuit board 9 cm high and 18 cm long, dissipating a total of 15 W. The width of the air gap in the middle of the PCB is 0.25 cm. If the cooling air enters the 12-cm-wide core at 25°C and 1 atm at a rate of 0.8 L/s, determine the average temperature at which the air leaves the hollow core.

Step 1 of 4

Given

The total heat dissipated by a circuit board $\dot{Q} = 15\ \text{W}$

The temperature of air at inlet, $T_1 = 25°\text{C}$

Atmospheric pressures of air, $P = 1\ \text{atm}$

$$P = 101.325\ \text{kPa}$$

The volume flow rate of air, $\dot{V} = 0.8\ \text{L/s}$

$$\dot{V} = 0.8 \times 10^{-3}\ \text{m}^3/\text{s}$$

Step 2 of 4

From ideal gas equation $P = \rho RT$

$$\rho = \frac{P}{RT}$$

$$\rho = \frac{101.325}{0.287 \times (25+273)}$$

$$\rho = 1.1847\ \text{kg/m}^3$$

Mass flow rate of air is $\dot{m} = \rho \dot{V}$

$$\dot{m} = (1.1847) \times (0.8 \times 10^{-3})$$

$$\dot{m} = 9.4776 \times 10^{-4}\ \text{kg/s}$$

Step 3 of 4

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$. Hence

$$\dot{Q} = \dot{m}(h_{out} - h_{in})$$

But enthalpy, $h = C_p T$, and $C_p = 1.005\ \text{kJ/kg°C}$ for air.

Also from properties of ideal gasses table for air, $R = 0.287\ \text{kJ/kg.K}$

Step 4 of 4

Hence by substituting the values,

$$\dot{Q} = \dot{m}_{out} C_p (T_{out} - T_{in})$$

$$15 = (9.4776 \times 10^{-4}) \times (1.005 \times 10^{3}) \times (T_2 - 25)$$

$$(T_2 - 25) = \frac{15}{0.9525}$$

$$T_2 = 15.7 + 25$$

$$T_2 = 40.7^0\text{C}$$

Hence final temperature of air is $\boxed{T_2 = 40.7^0\text{C}}$

Question: 6.096P

A computer cooled by a fan contains eight PCBs, each dissipating 10 W power. The height of the PCBs is 12 cm and the length is 18 cm. The cooling air is supplied by a 25-W fan mounted at the inlet. If the temperature rise of air as it flows through the case of the computer is not to exceed 10°C, determine (*a*) the flow rate of the air that the fan needs to deliver and (b) the fraction of the temperature rise of air that is due to the heat generated by the fan and its motor.

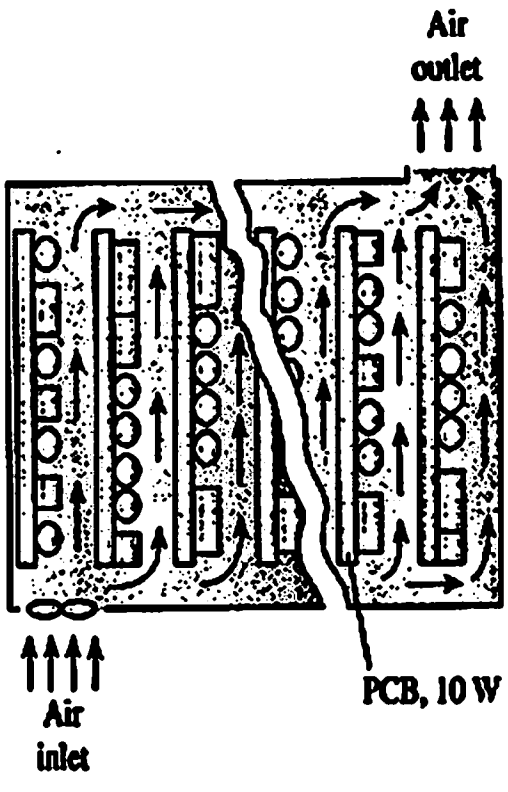


FIGURE P6–96

Step 1 of 5

Given

Number of PCBs in a computer is N=8

Amount of heat dissipated by each PCB $\dot{Q} = 10\,\text{W}$

The power supplied to the fan $\dot{W} = -25\ \text{W}$

The temperature rise of the air $\Delta T = 10^0\,\text{C}$

Step 2 of 5

The amount of heat dissipated by a complete circuit board

$$\dot{Q}_T = N \times \dot{Q}$$

$$\dot{Q}_T = 8 \times 10$$

$$\dot{Q}_T = 80\ \text{W}$$

Step 3 of 5

The energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_T + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} + \dot{W} \qquad (\text{since } \Delta ke = 0, \Delta pe = 0)$$

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

Then energy equation becomes

$$\dot{Q}_T = \dot{m}_{air}\left(h_{out} - h_{in}\right) + \dot{W}$$

We know that $h = c_p T$

$$\dot{Q}_T = \dot{m}_{air}\, c_{p_{air}}\left(T_{out} - T_{in}\right) + \dot{W}$$

Step 4 of 5

Taking specific heat of air $c_{p_{air}} = 1.005\ \text{kJ/kg°C}$

Gas constant $R = 0.287\ \text{kJ/kg.K}$

Substituting the known values in energy equation

$$\dot{Q}_T = \dot{m}_{air}\, c_{p_{air}}\left(T_{out} - T_{in}\right) + \dot{W}$$

$$80 = \dot{m}_{air} \times 1.005 \times\times 10^3 \times 10 + (-25)$$

$$\boxed{\dot{m}_{air} = 0.0104\,\text{kg/s}}$$

Step 5 of 5

b) The amount of heat generated by fan neglecting all the losses is

$$\dot{Q}_{fan} = \dot{W} = \dot{m}_{air} c_p \Delta T$$

$$25 = 0.0104(1005)\Delta T$$

$$\Delta T = 2.391$$

$$\Delta T = 2.4$$

The fraction of temperature rise is

$$\text{fraction} = \frac{2.4}{10} \times 100$$

$$\boxed{\text{fraction} = 24\%}$$

Question: 6.097P

Hot water at 90°C enters a 22-m section of a cast iron pipe whose inner diameter is 2.5 cm at an average velocity of 0.6 m/s. The outer surface of the pipe is exposed to the cold air at 10°C in a basement. If water leaves the basement at 88°C, determine the rate of heat loss from the 1 water.

Step 1 of 3

Given

The temperature of water at inlet, $T_1 = 90°C$

The temperature of water at exit, $T_2 = 88°C$

The diameter of the pipe, $d = 2.5$ cm

$d = 0.025$ m

The average velocity of water, $V = 0.6$ m/s

Step 2 of 3

The density of water, $\rho = 1000\ kg/m^3$

The mass flow rate, $\dot{m}_w = \rho AV$

$$\dot{m}_w = \rho \times \frac{\pi d^2}{4} \times V$$

$$\dot{m}_w = 1000 \times \frac{\pi (0.025)^2}{4} \times 0.6$$

$$\dot{m}_w = 0.2945\ kg/s$$

Step 3 of 3

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{Q} + \dot{m}_{out} h_{out}$$

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$

$$\dot{Q} = \dot{m}_w (h_{in} - h_{out})$$

Also Enthalpy, $h = C_p T$. Where, $C_p = 4.18$ kJ/kg°C for water. Hence

$$\dot{Q} = \dot{m}_w C_p (T_{in} - T_{out})$$

$$\dot{Q} = \dot{m}_w C_p (T_1 - T_2)$$

$$\dot{Q} = 0.2945 \times 4.18 \times (90 - 88)$$

$$\dot{Q} = 2.462\ kW$$

Hence Rate of heat loss from water, $\boxed{\dot{Q} = 2.462\ kW}$

Question: 6.098P

Reconsider Prob. 6–97. Using EES (or other) software, investigate the effect of the inner pipe diameter on the rate of heat loss. Let the pipe diameter vary from 1.5 to 7.5 cm. Plot the rate of heat loss against the diameter, and discuss the results.

Step 1 of 4

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click Calculate on menu bar, select solve.

Finally, Solution window displayed the unknown parameters

Step 2 of 4

Equation window:

"Knowns:"

D = 0.025 [m]

rho = 965 [kg/m^3]

Vel = 0.6 [m/s]

T_1 = 90 [C]

T_2 = 88 [C]

c_p = 4.21 [kJ/kg-C]

"Analysis:" "The mass flow rate of water is:"

Area = pi*D^2/4

m_dot = rho*Area*Vel

"We take the section of the pipe in the basement to be the system, which is a control volume. The energy

Balance for this steady-flow system can be expressed in the rate form as"

E_dot_in - E_dot_out = DELTAE_dot_sys

DELTAE_dot_sys = 0 "Steady-flow assumption"

E_dot_in = m_dot*h_in

doE_dot_out =Q_dot_out+m_dot*h_out

h_out = c_p * T_1

h_out = c_p * T_2

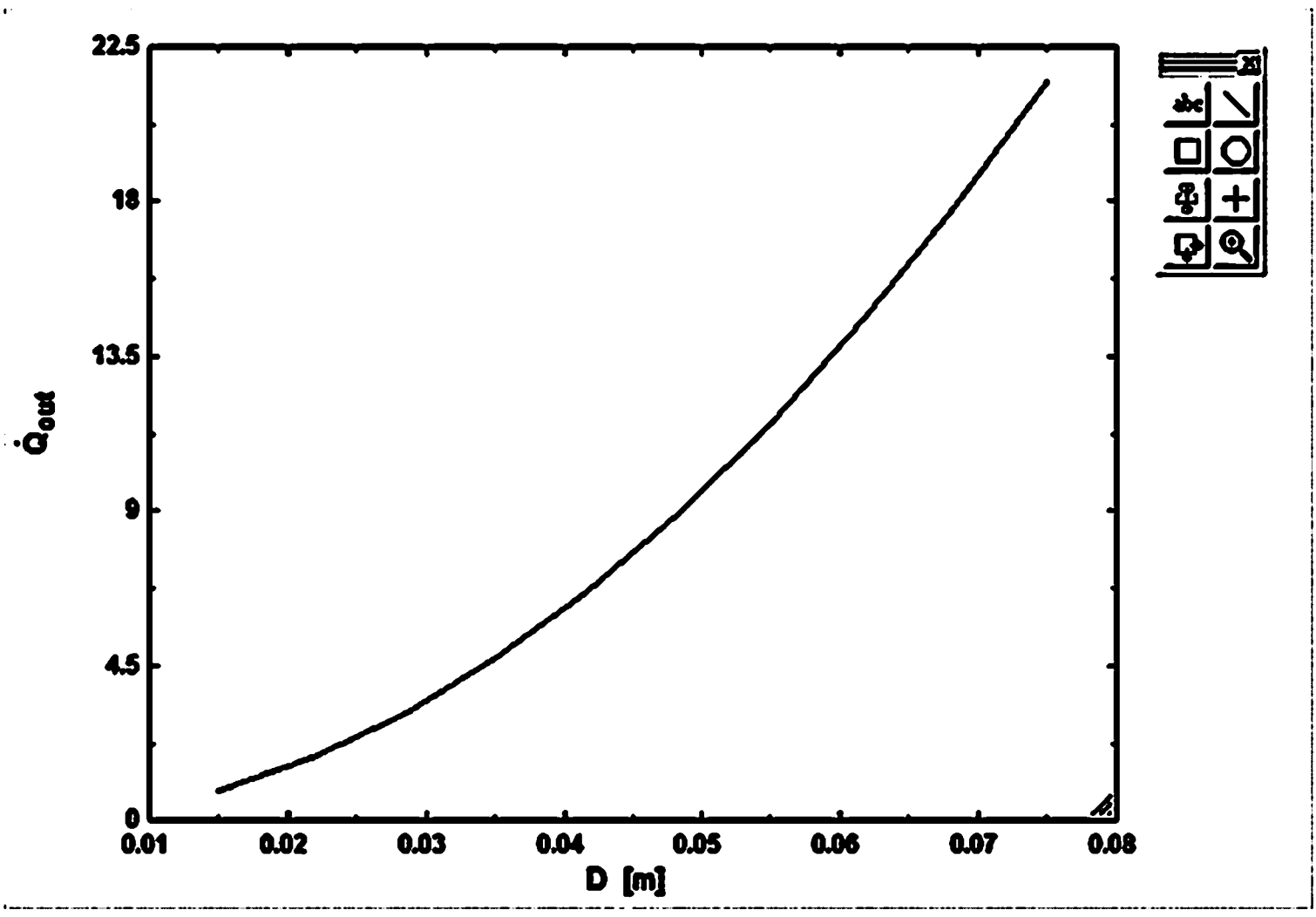
Step 3 of 4

Table for $\dot{Q}_{out}$ as a function of *D*:

$\triangleright_{1..10}$	1 D [m]	2 $\dot{Q}_{out}$
Run 1	0.015	0.8615
Run 2	0.02167	1.797
Run 3	0.02833	3.074
Run 4	0.035	4.69
Run 5	0.04167	6.647
Run 6	0.04833	8.945
Run 7	0.055	11.58
Run 8	0.06167	14.56
Run 9	0.06833	17.88
Run 10	0.075	21.54

Step 4 of 4

Graph between the $\dot{Q}_{out}$ Vs. D :



Question: 6.099P

A 5-m × 6-m × 8-m room is to be heated by an electric resistance heater placed in a short duct in the room. Initially, the room is at 15°C, and the local atmospheric pressure is 98 kPa. The room is losing heat steadily to the outside at a rate of 200 kJ/min. A 200-W fan circulates the air steadily through the duct and the electric heater at an average mass flow rate of 50 kg/min. The duct can be assumed to be adiabatic, and there is no air leaking in or out of the room. If it takes 15 min for the room air to reach an average temperature of 25°C, find (*a*) the power rating of the electric heater and (*b*) the temperature rise that the air experiences each time it passes through the heater.

Step 1 of 6

Given

The volume of the room $V = 5\times6\times8$

$$V = 240\ \text{m}^3$$

The initial temperature of the room $T_1 = 15°\text{C}$

The local atmospheric pressure $P = 98\ \text{kPa}$

The amount of heat lost to the surroundings $\dot{Q} = -200\ \text{kJ/min}$

$$= -\frac{200}{60}$$

$$\dot{Q} = -3.334\ \text{kJ/s}$$

The work input to the fan $\dot{W}_{fan} = -200\ \text{W} = -0.2\text{kW}$

Mass flow rate of air $\dot{m}_{air} = 50\ \text{kg/min}$

The final temperature of the room $T_2 = 25°\text{C}$

The time taken to reach the final temperature is t=15 min

Step 2 of 6

(a) The power rating of the electrical heater

Taking the gas constant $R = 0.287\ \text{kJ/kg.K}$

Specific heat of gas at constant pressure is $c_p = 1.005\ \text{kJ/kg.K}$

Specific heat of gas at constant volume $c_v = 0.718\ \text{kJ/kg.K}$

From first law of thermodynamics for a closed system

$Q = W + \Delta E$

Here considering room as a closed system, the system is stationary so $\Delta E = \Delta U$.

$$\dot{Q} = \dot{W}_{ele} + \dot{W}_{fan} + \frac{m}{t}\left(c_v\left(T_2 - T_1\right)\right) \text{ ---------- (1)}$$

Step 3 of 6

The density of air at initial condition

$$\rho_1 = \frac{P_1}{RT_1}$$

$$\rho_1 = \frac{98}{(0.287)(15+273)}$$

$$\rho_1 = 1.185\ \text{kg/m}^3$$

Mass flow rate of air

$m = \rho_1 V$

$= 1.185(240)$

$m = 284.5\ \text{kg}$

Step 4 of 6

Substituting all the values in equation (1)

$$\dot{Q} = \dot{W}_{ele} + \dot{W}_{fan} + \frac{m}{t}\left(c_v\left(T_2 - T_1\right)\right)$$

$$-3.334 = \dot{W}_{ele} - 0.200 + \frac{284.5}{15\times60}\times0.718(25-15)$$

$$-3.334 = \dot{W}_{ele} - 0.2 + 2.2696$$

$$\boxed{\dot{W}_{ele} = -5.403\ \text{kW}}$$

Step 5 of 6

(b) The rise of temperature of the heater

Here taking only the duct as a system, so for a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{W} + \dot{m}_{out} h_{out} \quad (Q = 0 \text{ And } ke = 0, pe = 0)$$

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

Then

$$0 = \dot{W}_{fan} + \dot{W}_{ele} + \dot{m}_{air}\left(h_2 - h_1\right)$$

We know that $h = c_p T$

$$0 = \dot{W}_{fan} + \dot{W}_{ele} + \dot{m}_{air} c_p \left(T_2 - T_1\right)$$

Step 6 of 6

Substituting all the values in above equation we get

$$0 = -0.2 - 5.403 + \frac{50}{60}\left(1.005\right)\left(T_2 - T_1\right)$$

$$0 = -0.2 - 5.403 + 0.8375\left(T_2 - T_1\right)$$

$$T_2 - T_1 = 6.69^0\text{C}$$

The rise in temperature is

$$\boxed{T_2 - T_1 = 6.69°\text{C}}$$

Question: 6.100P

The hot-water needs of a household are to be met by heating water at 55°F to 180°F by a parabolic solar collector at a rate of 4 lbm/s. Water flows through a 1.25-in-diameter thin aluminum tube whose outer surface is black-anodized in order to maximize its solar absorption ability. The centerline of the tube coincides with the focal line of the collector, and a glass sleeve is placed outside the tube to minimize the heat losses. If solar energy is transferred to water at a net rate of 400 Btu/h per ft length of the tube, determine the required length of the parabolic collector to meet the hot-water requirements of this house.

Step 1 of 3

Given

Inlet temperature of water $T_1 = 55°F$

Exit temperature of water $T_2 = 180°F$

The diameter of the aluminum tube $d = 1.25\,\text{in}$

The mass flow rate of water $\dot{m} = 4\ \text{lbm/s}$

Solar energy transferred to water per ft length of the tube $\frac{\dot{Q}}{L} = 400\,\text{Btu/h}$

$$= \frac{400}{3600}$$

$$\frac{\dot{Q}}{L} = 0.1111\,\text{Btu/s}$$

Step 2 of 3

For a steady flow device such as pipe or duct, the energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out} \qquad (w = 0, \Delta ke = 0, \Delta pe = 0)$$

Since $\dot{m}_{in} = \dot{m}_{out} = \dot{m}_w$ by using mass balance

$$\dot{Q} = \dot{m}_w (h_{out} - h_{in})$$

We know that $h = c_p T$

$$\dot{Q} = \dot{m}_w c_p (T_2 - T_1)$$

Taking specific heat of water $c_p = 1$ Btu/lbm.°F

Therefore

$$\dot{Q} = (4)(1)(180 - 55)$$

$$\dot{Q} = 500\ \text{Btu/s}$$

Step 3 of 3

The total length of the tube is determined as

$$\frac{\dot{Q}}{L} = 0.1111$$

$$L = \frac{\dot{Q}}{0.1111}$$

$$= \frac{500}{0.1111}$$

$$\boxed{L = 4500.45\ \text{ft}}$$

Question: 6.101P

A house has an electric heating system that consists of a 300-W fan and an electric resistance heating element placed in a duct. Air flows steadily through the duct at a rate of 0.6 kg/s and experiences a temperature rise of 7°C. The rate of heat loss from the air in the duct is estimated to be 300 W. Determine the power rating of the electric resistance heating element.

Step 1 of 3

Given

The work input to the fan $\dot{W}_{fan} = -300$ W

The mass flow rate of air in the duct $\dot{m}_{air} = 0.6$ kg/s

The rise in temperature of the air in the duct $\Delta T = 7$

The amount of heat lost to the surroundings $\dot{Q} = -300$ W

Step 2 of 3

The energy balance equation for the conservation of energy is written as

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{W} + \dot{m}_{out} h_{out} \quad (\text{Since } \Delta ke = 0, \Delta pe = 0)$$

By using mass balance

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{air}$$

$$\dot{Q} = \dot{W}_{fan} + \dot{W}_{elec} + \dot{m}_{air}(h_{out} - h_{in})$$

Step 3 of 3

Substituting the values in above equation

$$\dot{Q} = \dot{W}_{fan} + \dot{W}_{elec} + \dot{m}_{air}(h_{out} - h_{in})$$

We know that $h = c_p T$

Taking specific heat of air $c_p = 1.005$ kJ/kg°C

$$-0.3 = -0.3 + \dot{W}_{elec} + 0.6 \times 1.005 \times 7$$

$$\boxed{\dot{W}_{elec} = -4.221 \text{kW}}$$

Question: 6.102P

Steam enters a long, horizontal pipe with an inlet diameter of D1= 12 cm at 2 MPa and 300°C with a velocity of 3 m/s. Farther downstream, the conditions are 800 kPa and 250°C, and the diameter is *D*2 = 10 cm. Determine (a) the mass flow rate of the steam and (b) the rate of heat transfer.

Step 1 of 3

Given

Inlet Pressure of air, $P_1 = 2\,\text{MPa}$

Inlet temperature of air, $T_1 = 300°\text{C}$

Inlet diameter of pipe, $d_1 = 12\ \text{cm}$

Inlet velocity of air, $V_1 = 3\ \text{m/s}$

Exit pressure of air, $P_2 = 800\ \text{kPa}$

Exit temperature of air, $T_2 = 250°\text{C}$

Exit diameter of pipe, $D_2 = 10\ \text{cm}$

Step 2 of 3

(a)

From superheated steam tables, at 2MPa and 300°C

Specific volume, $v_1 = 0.12551\ \text{m}^3/\text{kg}$

Enthalpy, $h_1 = 3024.2\ \text{kJ/kg}$

From superheated steam tables, at 800kPa and 250°C

Enthalpy, $h_2 = 2950.4\ \text{kJ/kg}$

The mass flow rate of steam, $\dot{m} = \dfrac{A_1 V_1}{v_1}$

$$\dot{m} = \frac{\pi d_1^2 V_1}{4 v_1}$$

$$\dot{m} = \frac{\pi \times (0.12)^2 \times 3}{4 \times 0.12551}$$

$$\dot{m} = 0.27\ \text{kg/s}$$

Hence mass flow a rate of steam is $\boxed{\dot{m} = 0.27\ \text{kg/s}}$

Step 3 of 3

(b)

From the energy balance equation for the conservation of energy

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{Q} + \dot{m}_{out} h_{out}$$

$$\dot{Q} = \dot{m}_{in} (h_{in} - h_{out})$$

$$\dot{Q} = \dot{m}(h_1 - h_2)$$

$$\dot{Q} = 0.27 \times (3024.2 - 2950.4)$$

$$\dot{Q} = 19.92\ \text{kJ/s}$$

The rate of heat transfer $\boxed{\dot{Q} = 19.92\ \text{kJ/s}}$

Question: 6.103P

Refrigerant-134a enters the condenser of a refrigerator at 900 kPa and 60°C, and leaves as a saturated liquid at the same pressure. Determine the heat transfer from the refrigerant per unit mass.

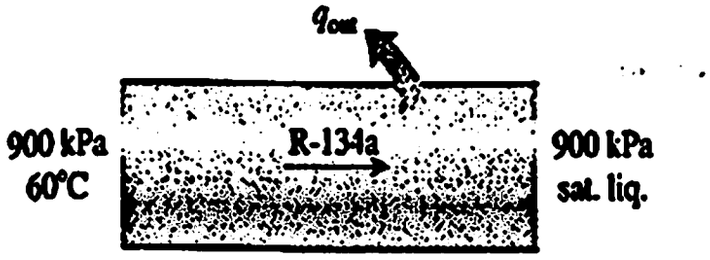


FIGURE P6–103

Step 1 of 3

Given

Inlet Pressure of R-134a, $P_1 = 900$ kPa

Inlet temperature of R-134a, $T_1 = 60°C$

Exit pressure of R-134a, $P_2 = P_1$

$P_2 = 900$ kPa

Exit state of R-134a is saturated liquid $(x_2 = 0)$.

Step 2 of 3

From superheated tables of R-134a at 900 kPa and 60°C ,

Enthalpy, $h_1 = 295.13$ kJ/kg

From saturated tables of R-134a at 900 kPa,

Enthalpy, $h_2 = h_f = 101.61$ kJ/kg

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in} h_{in} = \dot{Q} + \dot{m}_{out} h_{out}$$

Step 3 of 3

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$. So

$$\dot{Q} = \dot{m}(h_1 - h_2)$$

$$\frac{\dot{Q}}{\dot{m}} = (h_1 - h_2)$$

$$\dot{q} = (h_1 - h_2)$$

$$\dot{q} = (295.13 - 101.61)$$

$$\dot{q} = 193.52 \text{ kJ/kg}$$

Hence the heat transfer per unit mass is $\boxed{\dot{q} = 193.52 \text{ kJ/kg}}$

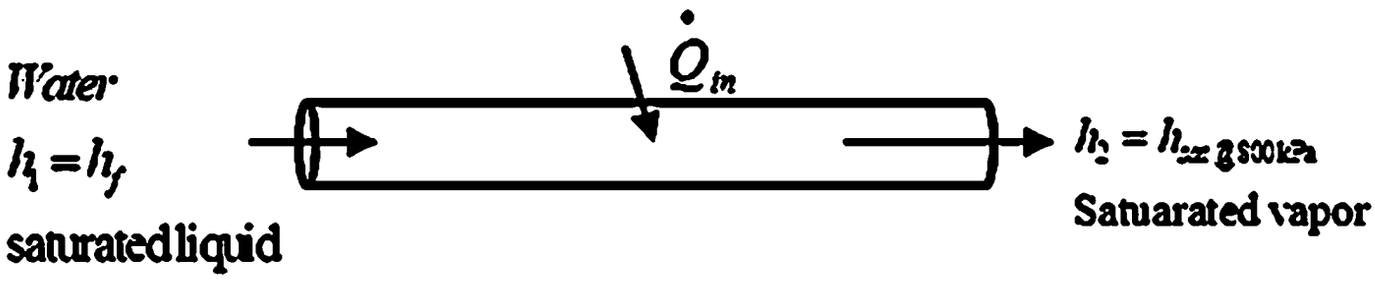
6.104P

Question:

Saturated liquid water is heated at constant pressure in a steady-flow device until it is a saturated vapor. Calculate the heat transfer, in kJ/kg, when the vaporization is done at a pressure of 800 kPa.

Step 1 of 2

Draw the flow diagram as shown:



Step 2 of 2

Obtain the value of enthalpy from the table: "Saturated water – Pressure table" at $P_1 = 800\,\text{kPa}$

$$h_{fg} = 2047.5\,\text{kJ/kg}$$

Consider control volume as the steam pipe as the system, since mass crosses the boundary. There is only one inlet and one exit, so mass entering is equal to mass leaving.

$$\dot{m}_1 = \dot{m}_2$$

Apply energy conservation for the system.

For a steady state condition energy stored within the system is zero.

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + V_1^2/2\right)_{in} + \dot{Q}_{in} + \dot{W}_{in} = \dot{m}\left(h_2 + V_2^2/2\right)_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2$$

$$\dot{Q}_{in} = \dot{m}(h_2 - h_1)$$

$$\dot{q}_{in} = (h_2 - h_1)$$

Here, $\dot{m}$ is the mass flow rate, V_1, V_2 are the mass flows rate at inlet and exit respectively and $\dot{q}_{out}$ is the heat transfer per unit mass.

$$\begin{aligned}\dot{q}_{in} &= (h_1 - h_2)\\ &= h_{fg}\\ &= 2047.5\ \text{kJ/kg}\end{aligned}$$

Therefore, the heat transfer per unit mass of water is $\boxed{2047.5\,\text{kJ/kg}}$

Question: 6.105P

Air at 300 K and 100 kPa steadily flows into a hair dryer having electrical work input of 1500 W. Because of the size of the air intake, the inlet velocity of the air is negligible. The air temperature and velocity at the hair dryer exit are 80°C and 21 m/s, respectively. The flow process is both constant pressure and adiabatic. Assume air has constant specific heats evaluated at 300 K. (a) Determine the air mass flow rate into the hair dryer, in kg/s. (b) Determine the air volume flow rate at the hair dryer exit, in m3/s.

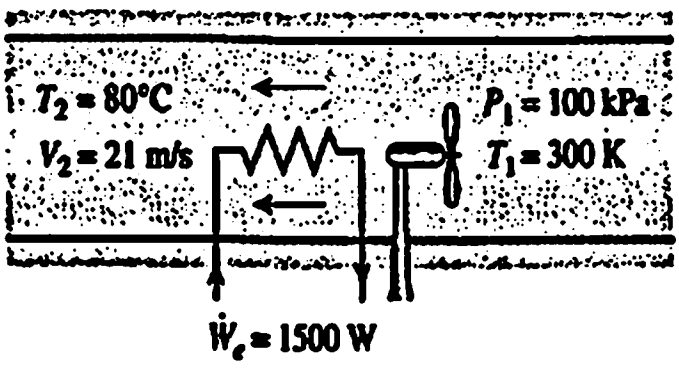


FIGURE P6–105

Step 1 of 3

(a)

The steady state energy equation is,

$$\dot{Q}-\dot{W}=\dot{m}\left[(h_1-h_2)+\left(\frac{V_1^2-V_2^2}{2000}\right)+g(z_1-z_2)\right]$$

Here, $\dot{Q}$ is the amount of heat interaction, $\dot{W}$ is the word done, $\dot{m}$ is the mass flow, h_1 is the initial enthalpy, h_2 is the final enthalpy, V_1 is the initial velocity, V_2 is the final velocity, g is the acceleration due to gravity, z_1 is the level from datum to initial point or inlet point and z_2 is the level from datum to the final point or outlet.

Because the process takes place along same level, so the potential energy is same and assuming the process to be adiabatic, the heat interaction is nullified.
So,

$$\dot{W}=\dot{m}\left[(h_2-h_1)+\left(\frac{V_2^2-V_1^2}{2000}\right)\right]$$

But Enthalpy is $h=c_pT$. So,

$$\dot{W}=\dot{m}\left[c_p(T_2-T_1)+\left(\frac{V_2^2-V_1^2}{2000}\right)\right] \quad \text{..... (1)}$$

Here, c_p is the Specific heat at constant pressure.

Step 2 of 3

Substitute 100 kPa for P_1, 300K for T_1, 1500 W for $\dot{W}$, 353K for T_2, 21 m/s for V_1, 0 m/s for V_2 and 1.005 kJ/kg.K for c_p in (1).

$$\dot{W}=\dot{m}\left[c_p(T_2-T_1)+\left(\frac{V_2^2-V_1^2}{2000}\right)\right]$$

$$1.5=\dot{m}\left[1.005(353-300)+\left(\frac{21^2-0^2}{2000}\right)\right]$$

$$1.5\text{ kW}=\dot{m}\times(53.4855\text{ kJ/kg})$$

$$\dot{m}=\frac{1.5\text{ kW}}{53.4855\text{ kJ/kg}}$$

$$\dot{m}=0.028044\text{ kg/s}$$

Therefore mass flow rate of air drier is, $\boxed{0.28044\text{ kg/s}}$

Step 3 of 3

(b)

From properties of ideal gases, the gas constant of air is, $R=0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$
Find the density at final condition by using the ideal gas equation,

$$\rho_2=\frac{P_2}{RT_2}$$

$$=\frac{100}{0.287\times353}$$

$$=0.987\text{ kg/m}^3$$

The mass flow rate is the product of the density and the discharge,

$$\dot{m}=\rho_2\dot{Q}_2$$

Here

$$\dot{Q}_2=\frac{\dot{m}}{\rho_2}$$

$$=\frac{0.028044}{0.987}$$

$$=0.02841\text{ m}^3/\text{s}$$

The volume flow rate is, $\boxed{0.02841\text{ m}^3/\text{s}}$

Question: 6.106P

Reconsider Prob. 6–105. Using EES (or other) software, investigate the effect of the exit velocity on the mass flow rate and the exit volume flow rate. Let the exit velocity vary from 5 to 25 m/s. Plot the mass flow rate and exit volume flow rate against the exit velocity, and discuss the results.

Step 1 of 5

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click Calculate on menu bar, select solve.

Finally, Solution window displayed the unknown parameters

Step 2 of 5

Equation window:

"Given"

T1=300 [K]

P=100 [kPa]

Vel_1=0 [m/s]

W_dot_e_in=1.5 [kW]

T2=(80+273) [K]

Vel_2=21 [m/s]

"Properties"

c_p=1.005 [kJ/kg-K]

R=0.287 [kJ/kg-K]

"Analysis"

W_dot_e_in=m_dot*c_p*(T2-T1)+m_dot*(Vel_2^2-Vel_1^2)*Convert(m^2/s^2,kJ/kg) "energy balance on hair dryer"

v2=(R*T2)/P

Vol_dot_2=m_dot*v2

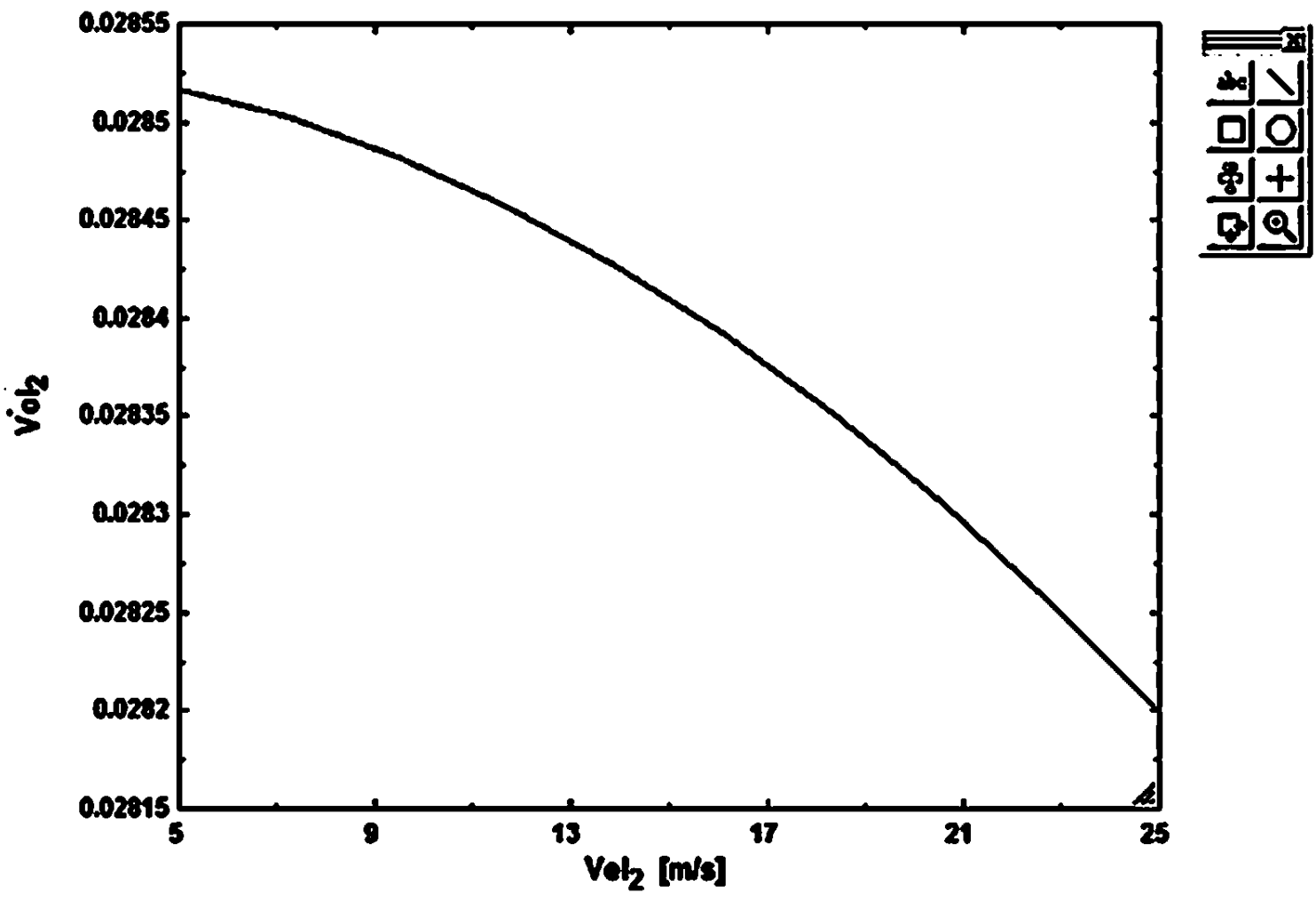
Step 3 of 5

Table for $\dot{Q}_{out}$ as a function of *D*:

1..10	1 Vel_2 [m/s]	2 $\dot{m}$	3 $\dot{Vol}_2$
Run 1	5	0.02815	0.02852
Run 2	7.222	0.02813	0.0285
Run 3	9.444	0.02811	0.02848
Run 4	11.67	0.02809	0.02846
Run 5	13.89	0.02806	0.02843
Run 6	16.11	0.02802	0.02839
Run 7	18.33	0.02798	0.02835
Run 8	20.56	0.02794	0.02831
Run 9	22.78	0.02789	0.02826
Run 10	25	0.02783	0.0282

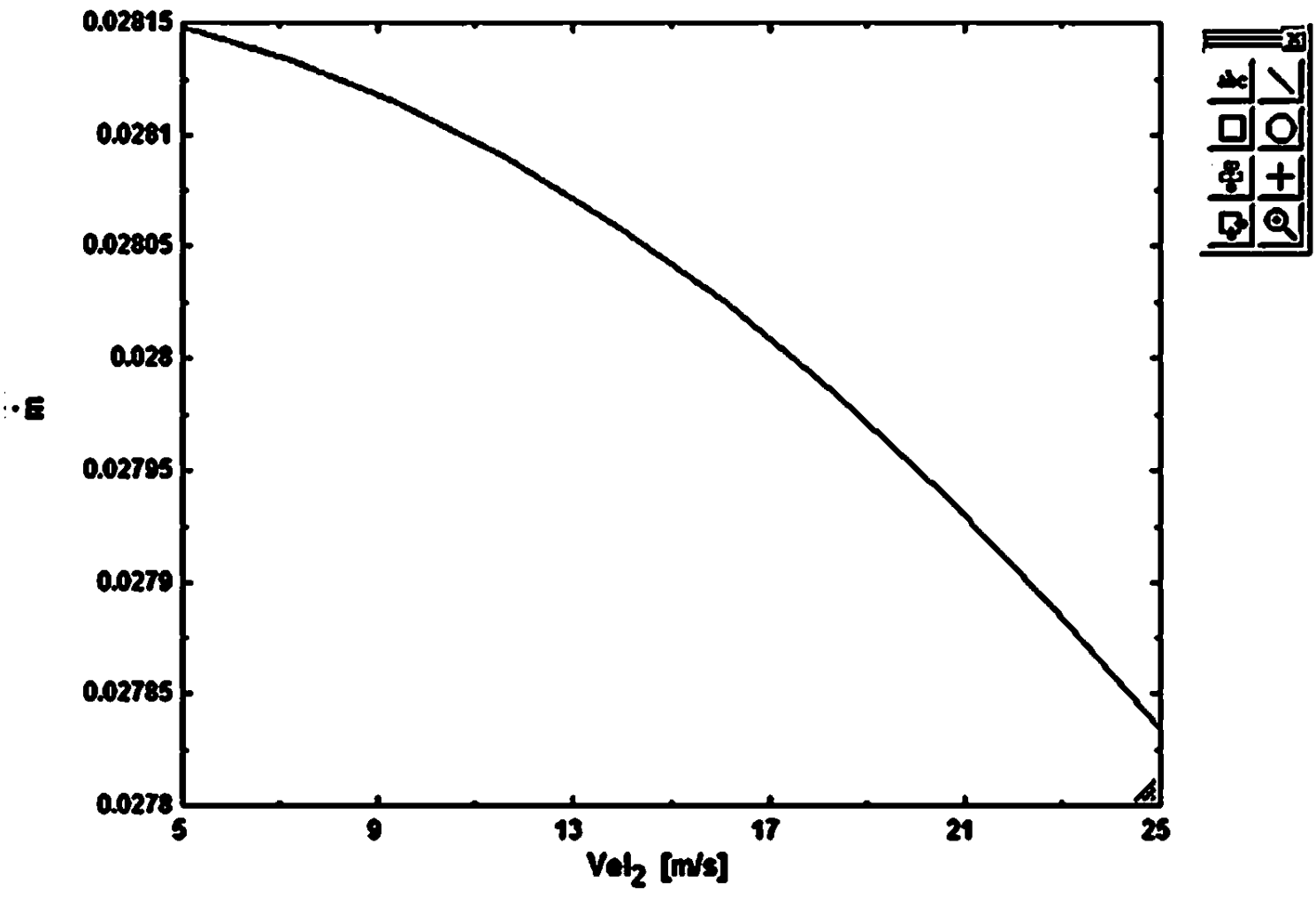
Step 4 of 5

Graph between the $\dot{Vol}_2$ Vs. Vel_2 :



Step 5 of 5

Graph between the $\dot{m}$ Vs. Vel_2 :

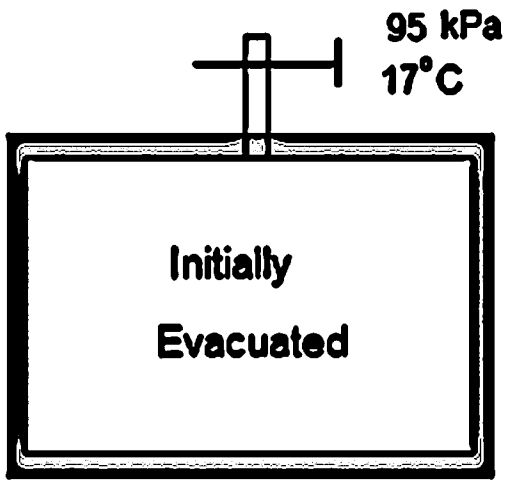


Question: 6.107P

An insulated rigid tank is initially evacuated. A valve is opened, and atmospheric air at 95 kPa and 17°C enters the tank until the pressure in the tank reaches 95 kPa, at which point the valve is closed. Determine the final temperature of the air in the tank. Assume constant specific heats.

Step 1 of 4

Figure shows the insulated rigid tank:



Step 2 of 4

From table A-2a, ratio of specific heats of air is

$$R = 0.287 \text{ kJ/kg°K}$$

The mass balance is

$$m_{in} - m_{out} = \Delta m_{sys}$$

For steady flow $\Delta m_{sys} = 0$ and so

$$m_{in} - m_{out} = 0$$

$$m_1 - m_2 = 0$$

$$m_1 = m_2 \qquad \text{..... (1)}$$

As the tank valve is opened to atmosphere, the initial temperature of the bottle is same as the atmospheric pressure & temperature.

$$T_1 = T_{atm}$$

The energy balance is given by

$$E_{in} - E_{out} = E_{sys}$$

Here, initial energy of system is E_{in}

The final energy of the system is E_{out}

The system energy is E_{sys}

For a steady process, $E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

Step 3 of 4

As there is flow of air initially, initial energy is given by enthalpy and as there is no flow of air finally, final energy is given by internal energy.

$$m_1 h_1 = m_2 u_2$$

Here initial mass of the air is m_1

Final mass of the air is m_2

Enthalpy of the air at Initial state is h_1

Internal energy of the air at Final state is u_2

Substitute equation (1) in the above equation

$$m_1 h_1 = m_1 u_2$$

$$h_1 = u_2$$

Step 4 of 4

$$h_1 = u_2$$

$$c_p \times T_1 = c_v \times T_2$$

Here specific heat at constant pressure is c_p

Specific heat at constant volume is c_v

Temperature at Initial state is T_1

Temperature at final state is T_2

$$T_2 = \frac{c_p \times T_1}{c_v}$$

$$T_2 = \frac{c_p}{c_v} \times T_1$$

Since specific heat ratio $\frac{c_p}{c_v} = k$

$$T_2 = k \times T_1$$

$$T_2 = 1.4 \times 290$$

Therefore final temperature in the tank is $\boxed{T_2 = 406 \text{ K}}$

Question: 6.108P

A rigid, insulated tank that is initially evacuated is connected through a valve to a supply line that carries helium at 200 kPa and 120°C. Now the valve is opened, and helium is allowed to flow into the tank until the pressure reaches 200 kPa, at which point the valve is closed. Determine the flow work of the helium in the supply line and the final temperature of the helium in the tank.

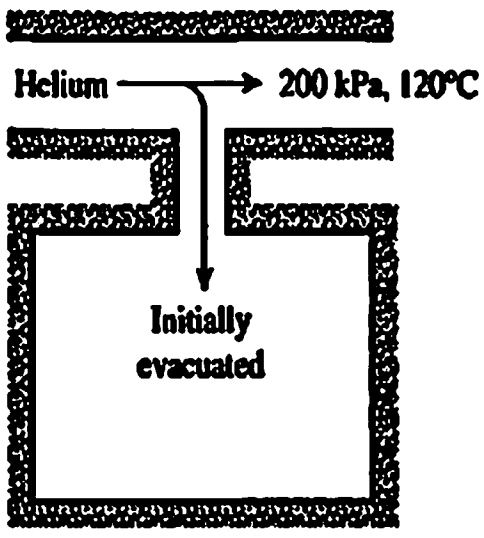
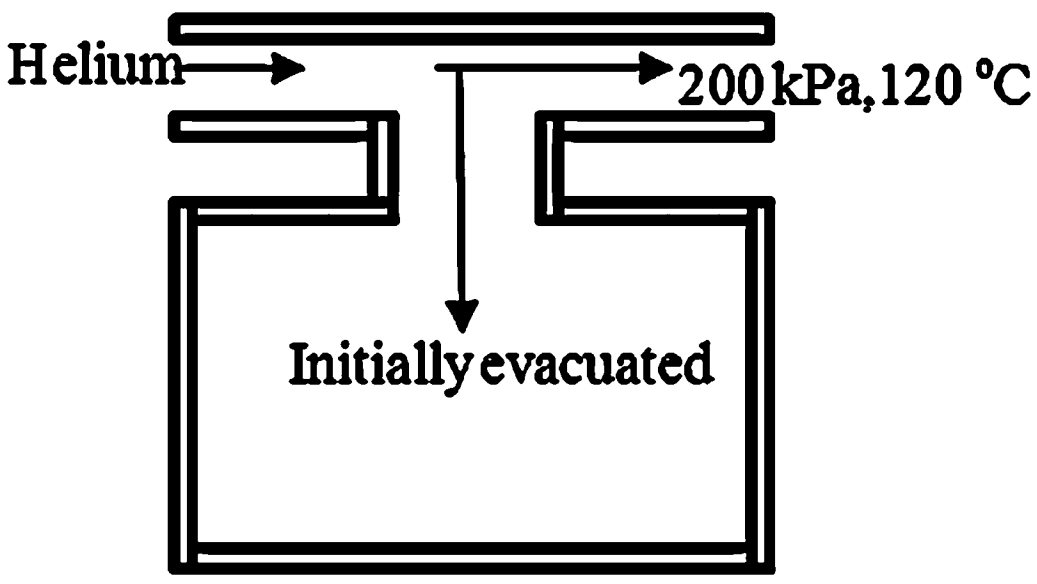


FIGURE P6–108

Step 1 of 3

Schematic diagram of the flow



Step 2 of 3

From the properties of gases in the helium column, obtain the following values

$R = 2.0769$ kJ/kg-K,

$C_p = 5.1926$ kJ/kg-K,

$C_v = 3.1156$ kJ/kg-K

Find the specific volume of the helium gas as shown below:

$$v = \frac{RT}{P}$$

Here R is the universal gas constant, T is temperature and P is the pressure.

Substitute $R = 2.0769$ kJ/kg-K , $P = 200$ kPa and $T = 120°c = (120+273)$ K

$$v = \frac{2.0769(\text{kJ/kg-K}) \times (120+273)\text{K}}{200\text{kPa}}$$
$$= 4.08\,\text{m}^3/\text{kg}$$

Find the flow work of gas as shown below:

$$W_{flow} = Pv$$

Substitute $P = 200$ kPa and $v = 4.08\,\text{m}^3/\text{kg}$ in above formula,

$$W_{flow} = 200(\text{kPa}) \times 4.08(\text{m}^3/\text{kg})$$
$$= 816\,\text{kJ/kg}$$

Therefore, the flow work is $\boxed{816\,\text{kJ/kg}}$

Step 3 of 3

Since the flow work in the supply line is converted to sensible internal energy in the tank.

Find the final helium temperature in the tank as follows

$$u_{tank} = h_{line}$$
$$h_{line} = C_p T_{line}$$
$$u_{tank} = C_v T_{tank}$$

Find the temperature of the helium in the tank is

$$T_{tank} = \frac{C_p T_{line}}{C_v}$$
$$= \frac{5.1926(\text{kJ/kg-K})}{3.1156(\text{kJ/kg-K})} \times 393(\text{K})$$
$$= 655\,\text{K}$$

Therefore, the final temperature of the helium is $\boxed{655\,\text{K}}$

Question: 6.109P

A 2-m3 rigid tank initially contains air at 100 kPa and 22°C. The tank is connected to a supply line through a valve. Air is flowing in the supply line at 600 kPa and 22°C. The valve is opened, and air is allowed to enter the tank until the pressure in the tank reaches the line pressure, at which point the valve is closed. A thermometer placed in the tank indicates that the air temperature at the final state is 77°C. Determine (a) the mass of air that has entered the tank and (b) the amount of heat transfer.

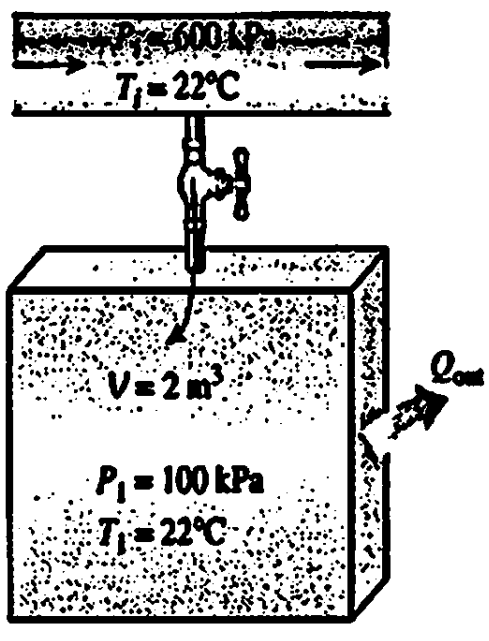
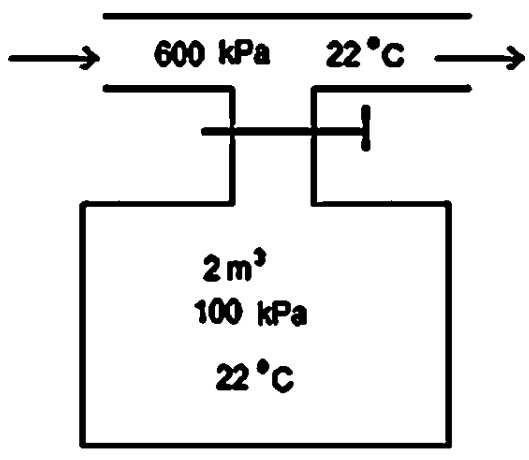


FIGURE P6–109

Step 1 of 6

Figure shows the rigid tank



Step 2 of 6

From table A-2a, gas constant of air is

$$R = 0.287 \text{ kJ/kg°K}$$

From table A-17, at temperature $T_1 = 295$ K specific internal energy of air is

$$u_1 = 210.49 \text{ kJ/kg}$$

From table A-17, at temperature $T_2 = 350$ K specific internal energy of air is

$$u_2 = 250.02 \text{ kJ/kg}$$

From table A-17, at temperature $T_s = 295$ K specific enthalpy of air is

$$h_s = 295.17 \text{ kJ/kg}$$

Step 3 of 6

Calculate the Initial mass in the tank by using the following equation

$$m_1 = \frac{P_1 \times V}{T_1 \times R}$$

Here, volume of the tank is V

Initial pressure of the air P_1

Initial temperature of the air T_1

Substitute, 100 kPa for P_1, 295 K for T_1 and 0.287 kJ/kg°K for R

$$m_1 = \frac{100 \times 2}{295 \times 0.287}$$

$$m_1 = \frac{200}{84.665}$$

$$m_1 = 2.362 \text{ kg}$$

Step 4 of 6

Calculate the Final mass by using the following equation

$$m_2 = \frac{P_2 \times V}{T_2 \times R}$$

Here,

Final pressure of the air P_2

Final temperature of the air T_2

Substitute, 600 kPa for P_2, 350 K for T_2 and 0.287 kJ/kg°K for R

$$m_2 = \frac{600 \times 2}{350 \times 0.287}$$

$$m_2 = \frac{1200}{100.45}$$

$$m_2 = 11.946 \text{ kg}$$

Step 5 of 6

Calculate the mass of air by using the mass balance

$$m_{in} - m_{out} = \Delta m_{sys}$$

$$m_s - 0 = m_2 - m_1$$

Substitute, 11.946 kg for m_2 and 2.362 kg for m_1

$$m_s = m_2 - m_1$$

$$m_s = 11.946 - 2.362$$

$$\boxed{m_s = 9.584 \text{ kg}}$$

Step 6 of 6

Calculate heat transfer by using the following energy balance

$$E_{in} - E_{out} = E_{sys}$$

For a steady process, $E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$m_s h_s - Q_{out} = m_2 u_2 - m_1 u_1$$

$$Q_{out} = m_s h_s - m_2 u_2 + m_1 u_1$$

$$Q_{out} = (9.584 \times 295.17) - (11.946 \times 250.02) + (2.362 \times 210.49)$$

$$Q_{out} = 2828.909 - 2986.739 + 497.177$$

$$\boxed{Q_{out} = 339.347 \text{ kJ}}$$

Therefore, the mass of air that has entered the tank is $\boxed{9.584 \text{ kg}}$

The heat transferred out through the wall of the tank is $\boxed{339.347 \text{ kJ}}$

Question: 6.110P

A 0.2-m3 rigid tank equipped with a pressure regulator contains steam at 2 MPa and 300°C. The steam in the tank is now heated. The regulator keeps the steam pressure constant by letting out some steam, but the temperature inside rises. Determine the amount of heat transferred when the steam temperature reaches 500°C.

Step 1 of 2

Calculate the initial mass using the following equation

$$\begin{aligned} m_t &= m_i - m_f \\ &= \frac{V}{v_i} - \frac{V}{v_f} \end{aligned}$$

Here, the total mass, initial mass and final mass are denote by m_t, m_i and m_f respectively.

The volume is denoted by V

Substitute $0.2\,\mathrm{m}^3$ for V

$$\begin{aligned} m_i &= \frac{0.2}{0.12551} - \frac{0.2}{0.17568} \\ &= 0.45\,\mathrm{kg} \end{aligned}$$

Assume that at exit, the enthalpy of the steam maintains an average value.

Calculate the enthalpy at exit using the following equation

$$\begin{aligned} h_t &= \frac{h_i + h_f}{2} \\ &= \frac{3024.20 + 3468.30}{2}\,\frac{\mathrm{kJ}}{\mathrm{kg}} \\ &= 3246.25\,\frac{\mathrm{kJ}}{\mathrm{kg}} \end{aligned}$$

Step 2 of 2

Consider the energy balance equation

$$Q - m_t h_t = m_f u_f - m_i u_i$$

$$Q - m_t h_t = \frac{V}{v_f} \times u_f - \frac{V}{v_i} u_i$$

$$Q - 0.45 \times 3246.25 = \left(\frac{0.2}{0.1757} \times 3116.9\right) - \left(\frac{0.2}{0.1255} \times 2773.2\right)$$

$$Q - 0.45 \times 3246.25_t = (0.138 \times 3116.9) - (1.59 \times 2773.2)$$

$$Q = 599\,\mathrm{kJ/kg}$$

Therefore, the total heat transfer is $\boxed{599\ \mathrm{kJ}}$.

Question: 6.111P

An insulated, vertical piston–cylinder device initially contains 10 kg of water, 6 kg of which is in the vapor phase. The mass of the piston is such that it maintains a constant pressure of 200 kPa inside the cylinder. Now steam at 0.5 MPa and 350°C is allowed to enter the cylinder from a supply line until all the liquid in the cylinder has vaporized. Determine (a) the final temperature in the cylinder and (b) the mass of the steam that has entered.

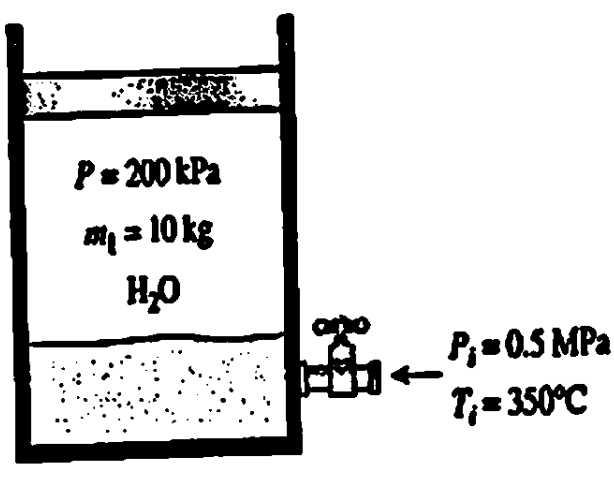
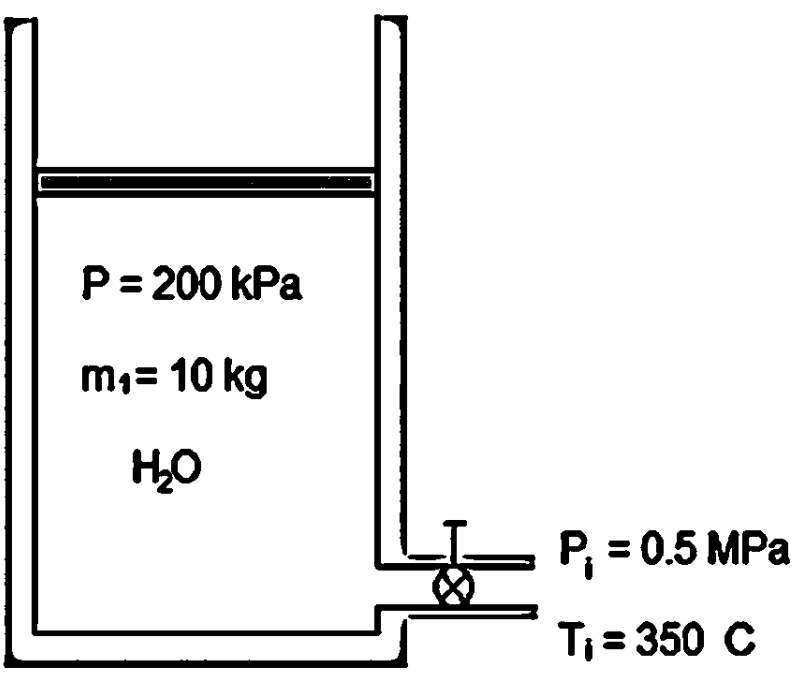


FIGURE P6–111

Step 1 of 8

Figure shows the rigid tank:



Step 2 of 8

Calculate the initial quality of steam in the device by using the following equation: $x_1 = \frac{m_v}{m_1}$ $x_1 = \frac{6}{10}$

$x_1 = 0.6$

Step 3 of 8

From table properties of water, at pressure $P_1 = 200\ \text{kPa}$: The specific enthalpy of saturated liquid at initial state is $h_{f1} = 504.71\ \text{kJ/kg}$. The specific enthalpy of saturated vapor at initial state is $h_{fg1} = 2201.6\ \text{kJ/kg}$. From table A-5, at pressure $P_2 = 200\ \text{kPa}$ and saturated vapor condition enthalpy of water are The specific enthalpy of saturated vapor at final state is, $h_2 = h_{g2} = 2706.3\ \text{kJ/kg}$. From table A-6, at pressure $P_s = 0.5\ \text{MPa}$ and temperature $T_s = 350\ ^\circ\text{C}$ properties of water:

$h_s = 3168.1\ \text{kJ/kg}$

Step 4 of 8

Calculate the Initial specific enthalpy of water by using the following equation:

$$h_1 = h_{f1} + \left(x_1 \times h_{fg1}\right)$$

Substitute, $504.71\ \text{kJ/kg}$ for h_{f1} and $2201.6\ \text{kJ/kg}$ for h_{fg1}:

$$h_1 = 504.71 + \left(0.6 \times 2201.6\right)$$

$$h_1 = 504.71 + 1320.96$$

$$h_1 = 1825.67\ \text{kJ/kg}$$

Step 5 of 8

(a)

As the final condition of water in the device is saturated vapor, the final temperature can be taken as the saturated temperature at that pressure.

From table properties of water are, at pressure $P_2 = 200\ \text{kPa}$.

Therefore, saturated temperature is $\boxed{T_2 = 120.21\ ^\circ\text{C}}$.

Step 6 of 8

(b)

Calculate mass of water that has entered the device mass balance:

$$m_{in} - m_{out} = \Delta m_{sys}$$
$$m_s - 0 = m_2 - m_1$$

$$m_s = m_2 - m_1 \quad \ldots\ldots (1)$$

Step 7 of 8

Calculate the Final mass of the air by using the following energy balance:

$$E_{in} - E_{out} = E_{sys}$$

$m_s \times h_s - 0 = \left(m_2 \times h_2\right) - \left(m_1 \times h_1\right)$ Substitute equation (1) in the above equation:

$$m_s \times h_s = \left(m_2 \times h_2\right) - \left(m_1 \times h_1\right)$$
$$\left(m_2 - m_1\right) \times h_s = \left(m_2 \times h_2\right) - \left(m_1 \times h_1\right)$$
$$\left(m_2 \times h_s\right) - \left(m_1 \times h_s\right) = \left(m_2 \times h_2\right) - \left(m_1 \times h_1\right)$$
$$\left(m_1 \times h_1\right) - \left(m_1 \times h_s\right) = \left(m_2 \times h_2\right) - \left(m_2 \times h_s\right)$$
$$m_1 \times \left(h_1 - h_s\right) = m_2 \times \left(h_2 - h_s\right)$$
$$m_2 = \frac{m_1 \times \left(h_1 - h_s\right)}{\left(h_2 - h_s\right)}$$

Step 8 of 8

Substitute 10 kg for m_1 , $1825.67\ \text{kJ/kg}$ for h_1 , $3168.1\ \text{kJ/kg}$ for h_s :

$$m_2 = \frac{10 \times \left(1825.67 - 3168.1\right)}{\left(2706.3 - 3168.1\right)}$$
$$= \frac{10 \times \left(-1342.43\right)}{\left(-461.8\right)}$$
$$= \frac{-13424.3}{-461.8}$$
$$= 29.069\ \text{kg}$$

From equation (1),

$$m_s = m_2 - m_1$$

Substitute, $29.069\ \text{kg}$ for m_2, $10\ \text{kg}$ for m_1 :

$$m_s = 29.069 - 10$$
$$= 19.069\ \text{kg}$$

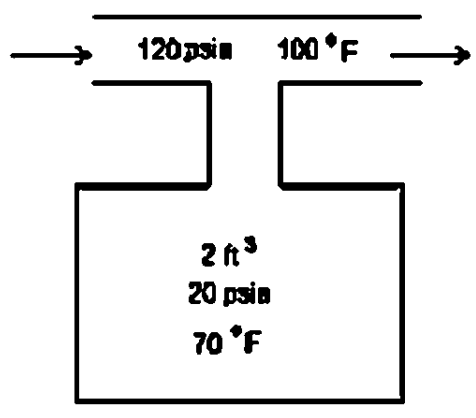
Therefore, mass of water that has entered the device is $\boxed{19.069\ \text{kg}}$.

Question: 6.112P

A scuba diver's 2-ft3 air tank is to be filled with air from a compressed air line at 120 psia and 100°F.Initially, the air in this tank is at 20 psia and 70°F. Presuming that the tank is well insulated, determine the temperature and mass in the tank when it is filled to 120 psia.

Step 1 of 8

Figure shows the scuba divers tank:



Step 2 of 8

From table A-1E, gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R

From table A-2E (a), specific heats of air are

Specific heat at constant pressure is $c_p = 0.240$ Btu/lbm.R

Specific heat at constant volume is $c_v = 0.171$ Btu/lbm.R

Step 3 of 8

Calculate the Initial mass of air by using the following equation

$$m_1 = \frac{P_1 \times V}{T_1 \times R}$$

Here, volume of the tank is V

Initial pressure of the air P_1

Initial temperature of the air T_1

Substitute, 20 psia for P_2, 530 R for T_1 and 0.3704 psia.ft^3/lbm.R for R

$$m_1 = \frac{20 \times 2}{530 \times 0.3704}$$

$$m_1 = \frac{40}{196.312}$$

$$m_1 = 0.20375 \text{ lbm}$$

Calculate the Final mass by using the following equation

$$m_2 = \frac{P_2 \times V}{T_2 \times R}$$

Here,

Final pressure of the air P_2

Final temperature of the air T_2

Substitute, 120 psia for P_2 and 0.3704 psia.ft^3/lbm.R for R

$$m_2 = \frac{120 \times 2}{T_2 \times 0.3704}$$

$$m_2 = \frac{240}{T_2 \times 0.3704}$$

$$m_2 = \frac{647.948}{T_2} \quad \text{..... (1)}$$

Step 4 of 8

The mass balance is given by

$$m_{in} - m_{out} = \Delta m_{sys}$$

$$m_s - 0 = m_2 - m_1$$

$$m_s = m_2 - m_1 \quad \text{..... (2)}$$

The energy balance is given by

$$E_{in} - E_{out} = E_{sys}$$

$$m_s \times h_s - 0 = (m_2 \times u_2) - (m_1 \times u_1)$$

$$m_s \times h_s = (m_2 \times u_2) - (m_1 \times u_1)$$

$$m_s \times c_p \times T_s = (m_2 \times c_v \times T_2) - (m_1 \times c_v \times T_1)$$

Step 5 of 8

Substituting equation (2) in the above equation

$$(m_2 - m_1) \times c_p \times T_s = (m_2 \times c_v \times T_2) - (m_1 \times c_v \times T_1)$$

$$(m_2 \times c_p \times T_s) - (m_1 \times c_p \times T_s) = (m_2 \times c_v \times T_2) - (m_1 \times c_v \times T_1)$$

$$(m_1 \times c_v \times T_1) - (m_1 \times c_p \times T_s) = (m_2 \times c_v \times T_2) - (m_2 \times c_p \times T_s)$$

$$m_1 \times ((c_v \times T_1) - (c_p \times T_s)) = m_2 \times ((c_v \times T_2) - (c_p \times T_s))$$

Step 6 of 8

Substitute 0.20375 lbm for m_1, 0.240 Btu/lbm.R for c_p, 0.171 Btu/lbm.R for c_v, $\frac{647.948}{T_2}$ for m_2, 560 R for the values in the above equation we get,

$$0.20375 \times ((0.171 \times 530) - (0.24 \times 560)) = \frac{647.948}{T_2} \times ((0.171 \times T_2) - (0.24 \times 560))$$

$$0.20375 \times (90.63 - 134.4) = \frac{647.948}{T_2} \times (0.171 \times T_2 - 134.4)$$

$$0.20375 \times (-43.77) = \left(\frac{647.948}{T_2} \times 0.171 \times T_2\right) - \left(\frac{647.948}{T_2} \times 134.4\right)$$

$$-8.918 = 110.799 - \left(\frac{87084.2112}{T_2}\right)$$

$$-119.717 = -\frac{87084.2112}{T_2}$$

Step 7 of 8

Calculate the final temperature by using the following equation

$$119.717 = \frac{87084.2112}{T_2}$$

$$T_2 = \frac{87084.2112}{119.717}$$

$$T_2 = 727.417 \text{ R}$$

$$= 267.417 \text{°F}$$

Therefore the final temperature of air is $\boxed{T_2 = 267.417 \text{ °F}}$

Step 8 of 8

From equation (1), calculate the final mass of the air in the tank

$$m_2 = \frac{647.948}{T_2}$$

$$m_2 = \frac{647.948}{727.417}$$

$$m_2 = 0.89075 \text{ lbm}$$

Therefore, final mass of air in the tank is $\boxed{0.89075 \text{ lbm}}$

Question: 6.113P

An air-conditioning system is to be filled from a rigid container that initially contains 5 kg of liquid R-134a at 24°C. The valve connecting this container to the air-conditioning system is now opened until the mass in the container is 0.25 kg, at which time the valve is closed. During this time, only liquid R-134a flows from the container. Presuming that the process is isothermal while the valve is open, determine the final quality of the R-134a in the container and the total heat transfer.

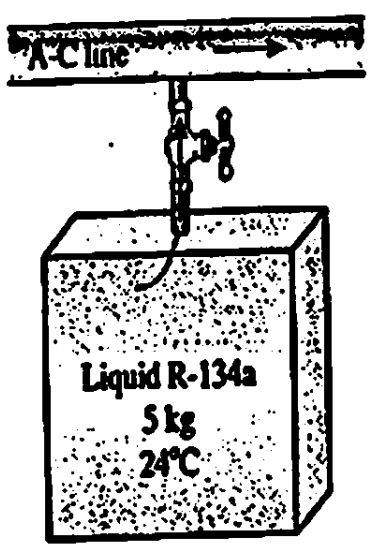
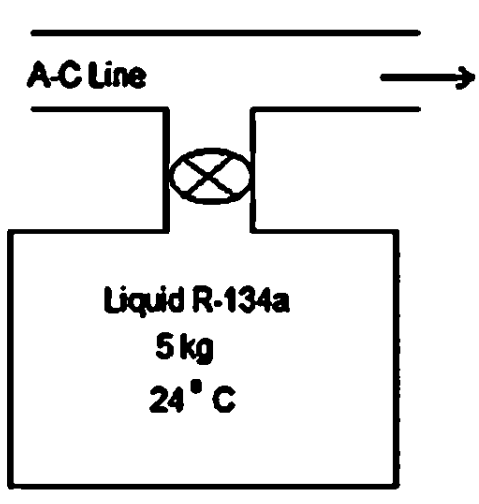


FIGURE P6–113

Step 1 of 10

Figure shows the air conditioning tank:



Step 2 of 10

From table A-11, at temperature $T_1 = 24\ ^\circ C$ properties of R-134a are

The specific volume of saturated liquid at initial state is $v_1 = v_f = 0.0008261\ m^3/kg$

The specific volume of saturated vapor at initial state is $v_g = 0.031834\ m^3/kg$

The internal energy of saturated liquid at initial state is $u_1 = u_f = 84.44\ kJ/kg$

The internal energy of saturated liquid-vapor at initial state is $u_{fg} = 158.65\ kJ/kg$

The enthalpy is $h_e = 84.98\ kJ/kg$

Step 3 of 10

Calculate the Volume of R-134a by using the following equation

$$V = m_1 \times v_1$$

Here initial mass of refrigerant is m_1, specific volume of the refrigerant at initial state is v_1

Substitute, $5\ kg$ for m_1, $0.0008261\ m^3/kg$ for v_1

$$V = 5 \times 0.0008261$$
$$V = 0.0041305\ m^3$$

Step 4 of 10

Calculate the final specific volume of R-134a by using the following equation

$$v_2 = \frac{V}{m_2}$$

Here the final mass of the refrigerant is m_2

Substitute, $0.0041305\ m^3$ for V, $0.25\ kg$ for m_2

$$v_2 = \frac{0.0041305}{0.25}$$
$$v_2 = 0.01652\ m^3/kg$$

Step 5 of 10

(a) Calculate the dryness fraction by using the Final specific volume equation

$$v_2 = v_f + x_2 \times v_{fg}$$
$$v_2 = v_f + \left[x_2 \times \left(v_g - v_f\right)\right]$$

Substitute, $0.0008261\ m^3/kg$ for v_f, $0.01652\ m^3/kg$ for v_2, $0.031834\ m^3/kg$ for v_g

$$0.01652 = 0.0008261 + \left[x_2 \times (0.031834 - 0.0008261)\right]$$
$$0.01652 = 0.0008261 + (x_2 \times 0.031008)$$

So, final quality of R-134a is given by

$$x_2 = \frac{0.01652 - 0.0008261}{0.031008}$$
$$x_2 = \frac{0.015694}{0.031008}$$
$$\boxed{x_2 = 0.50613}$$

Step 6 of 10

Calculate the Final specific internal energy by using the following equation

$$u_2 = u_f + \left(x_2 \times u_{fg}\right)$$

Substitute, $84.44\ kJ/kg$ for u_f, 0.50613 for x_2, $158.65\ kJ/kg$ for u_{fg}

$$u_2 = 84.44 + (0.50613 \times 158.65)$$
$$u_2 = 84.44 + 80.297$$
$$u_2 = 164.737\ kJ/kg$$

Step 7 of 10

(b) The mass balance is

$$m_{in} - m_{out} = \Delta m_{sys}$$
$$0 - m_e = m_2 - m_1$$
$$-m_e = m_2 - m_1$$
$$m_e = m_1 - m_2 \qquad \ldots\ldots (1)$$

Step 8 of 10

The energy balance is given by

$$E_{in} - E_{out} = E_{sys}$$
$$Q_{in} - m_e \times h_e = m_2 \times u_2 - m_1 \times u_1$$
$$Q_{in} = (m_2 \times u_2) - (m_1 \times u_1) + (m_e \times h_e)$$

Step 9 of 10

Substitute equation (1) in the above equation

$$Q_{in} = (m_2 \times u_2) - (m_1 \times u_1) + (m_1 - m_2) \times h_e$$
$$Q_{in} = (0.25 \times 164.737) - (5 \times 84.44) + \left[(5 - 0.25) \times 84.98\right]$$

Step 10 of 10

$$Q_{in} = (0.25 \times 164.737) - (5 \times 84.44) + (4.75 \times 84.98)$$
$$Q_{in} = 41.184 - 422.2 + 403.655$$
$$\boxed{Q_{in} = 22.639\ kJ}$$

Therefore, the final quality of R-134a is $\boxed{0.50613}$ and the total heat transfer is $\boxed{22.639\ kJ}$

Question: 6.114P

Oxygen is supplied to a medical facility from ten 1.5-ft3 compressed oxygen tanks. Initially, these tanks are at 1500 psia and 80°F. The oxygen is removed from these tanks slowly enough that the temperature in the tanks remains at 80°F. After two weeks, the pressure in the tanks is 300 psia. Determine the mass of oxygen used and the total heat transfer to the tanks.

Step 1 of 4

Given

Total volume of the oxygen tank, $V = 10 \times \left(1.5\ \text{ft}^3\right)$

$$V = 15\ \text{ft}^3$$

Initial pressure of oxygen, $P_2 = 1500$ psia

Initial temperature of oxygen, $T_2 = 80°\text{F}$

$$T_2 = 540\text{R}$$

Final pressure of oxygen, $P_2 = 300$ psia

Final temperature of oxygen, $T_2 = 80°\text{F}$

$$T_2 = 540\text{R}$$

Step 2 of 4

From ideal gas properties table for oxygen, $R = 0.3553$ psia·ft^3/lbm·R

From ideal gas equation, $PV = mRT$

$$m_1 = \frac{P_1 V_1}{RT_1}$$

$$m_1 = \frac{1500 \times 15}{0.3353 \times 540}$$

$$m_1 = 124.27\ \text{lbm}$$

Similarly, $m_2 = \dfrac{P_2 V_2}{RT_2}$

$$m_2 = \frac{300 \times 15}{0.3353 \times 540}$$

$$m_2 = 24.85\ \text{lbm}$$

Step 3 of 4

From mass balance, $\dot{m}_{in} - \dot{m}_{out} = \Delta m_{system}$

$$m_{exit} = \dot{m}_{in} - \dot{m}_{out}$$

$$m_{exit} = \dot{m}_1 - \dot{m}_2$$

$$m_{exit} = 124.27 - 24.85$$

$$m_{exit} = 99.41\ \text{lbm}$$

The mass of oxygen used is $\boxed{m_{exit} = 99.41\ \text{lbm}}$

Step 4 of 4

From energy balance, $E_{in} - E_{out} = \Delta E_{system}$

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1$$

But $h = c_p T$ and $u = c_v T$. Also

From ideal gas properties table for oxygen,

Specific heat at constant pressure, $c_p = 0.219$ BTu/lbm.R

Specific heat at constant volume, $c_v = 0.157$ BTu/lbm.R

Substituting the values in the energy equation,

$$Q_{in} - m_e c_p T_e = m_2 c_v T_2 - m_1 c_v T_1$$

$$Q_{in} - (99.41 \times 0.219 \times 540) = (24.85 \times 0.157 \times 540) - (124.27 \times 0.157 \times 540)$$

$$Q_{in} - 11756.2266 = 2106.783 - 10535.6$$

$$Q_{in} = 3327.4\ \text{Btu}$$

Hence net heat transfer in the tank $\boxed{Q_{in} = 3327.4\ \text{Btu}}$

Question: 6.115P

A 0.03-m3 rigid tank initially contains refrigerant-134a at 1.4 MPa and 100 percent quality. The tank is connected by a valve to a supply line that carries refrigerant-134a at 1.6 MPa and 36°C. Now the valve is opened, and the refrigerant is allowed to enter the tank. The valve is closed when it is observed that the tank contains saturated liquid at 1.6 MPa. Determine (*a*) the mass of the refrigerant that has entered the tank and (b) the amount of heat transfer.

Step 1 of 3

(a)

Obtain the properties of refrigerant R-134a from "properties of R-134a tables"

At pressure of 1.4 MPa :

Internal energy, $u_1 = u_g = 256.37 \text{ kJ/kg}$

Specific volume, $v_1 = v_g = 0.01411 \text{ m}^3/\text{kg}$

At pressure of 1.6 MPa :

Enthalpy, $h_{in} = h_f = 102.33 \text{ kJ/kg}$ **(at** $T_{in} = 36°\text{C}$**)**

Internal energy, $u_2 = u_f = 134.43 \text{ kJ/kg}$

Specific volume, $v_2 = v_f = 0.00094 \text{ m}^3/\text{kg}$

Step 2 of 3

The change of mass in the system is,

$$\Delta m_{system} = m_2 - m_1$$

Here, m_2 **is the final mass in the system, and** m_1 **is the initial mass in the system.**

The mass of refrigerant is the ratio of the volume of tank to the specific volume of gas at the particular parameters.

$$m = \frac{V}{v}$$

Initial mass of gas in the tank is,

$$m_1 = \frac{V_1}{v_1}$$

Substitute 0.03 m^3 **for** V_1 **, and** $v_1 = 0.01411 \text{ m}^3/\text{kg}$**.**

$$m_1 = \frac{V_1}{v_1}$$
$$= \frac{0.03}{0.01411}$$
$$= 2.126 \text{ kg}$$

Final mass of gas in the tank is,

$$m_2 = \frac{V_2}{v_2}$$

Substitute 0.03 m^3 **for** V_2 **and** v_2 **for** $0.00094 \text{ m}^3/\text{kg}$**.**

$$m_2 = \frac{V_2}{v_2}$$
$$= \frac{0.03}{0.00094}$$
$$= 31.92 \text{ kg}$$

Therefore the change in the mass of the system is,

$$\Delta m_{system} = m_2 - m_1$$
$$= 31.92 - 2.126$$
$$= 29.79 \text{ kg}$$
$$= 29.8 \text{ kg}$$

Therefore the mass entered the tank is $\boxed{29.8 \text{ kg}}$

Step 3 of 3

(b)

The change of energy in the system is,

$$\Delta E_{system} = E_2 - E_1$$

Here, E_1 **is the initial energy of the system, and** E_2 **is the final energy of the system.**

The above energy equation can be rewritten as,

$$Q + m_{in} h_{in} = m_2 u_2 - m_1 u_1$$

Here, Q **is the heat transferred in the system.**

Substitute the corresponding values in the above equation.

$$Q + m_{in} h_{in} = m_2 u_2 - m_1 u_1$$
$$= m_2 u_2 - m_1 u_1 - m_{in} h_{in}$$
$$= (31.92)(134.43) - (2.127)(256.37) - (29.8)(102.33)$$
$$= 696.266 \text{ kJ}$$

The amount of heat transfer $\boxed{Q = 696.266 \text{ kJ}}$

Question: 6.116P

A 0.3-m3 rigid tank is filled with saturated liquid water at 200°C. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank. Heat is transferred to the water such that the temperature in the tank remains constant. Determine the amount of heat that must be transferred by the time one-half of the total mass has been withdrawn.

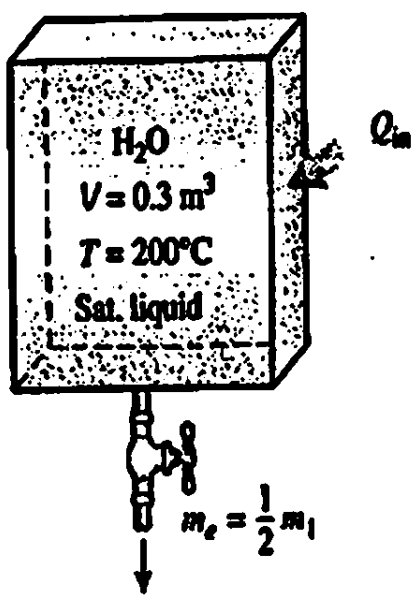


FIGURE P6–116

Step 1 of 5

From tables, at temperature of water in the tank $T = 200\ °C$ and saturated liquid condition, properties of water are

$$v_1 = v_f = 0.001157\ m^3/kg$$

$$u_1 = u_f = 850.46\ kJ/kg$$

$$h_1 = h_f = 852.26\ kJ/kg$$

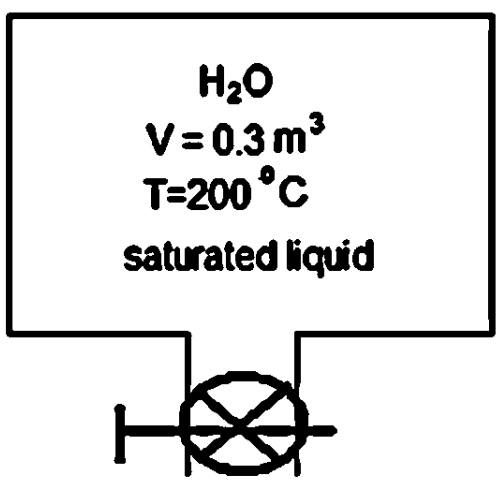
From tables, at temperature $T = 200\ °C$, properties of water are

$$v_g = 0.12721\ m^3/kg$$

$$u_{fg} = 1743.7\ kJ/kg$$

Step 2 of 5

Draw the figure representing the given system,



Step 3 of 5

Initial mass in the tank is calculated by dividing volume of the tank V by initial specific volume

$$m_1 = \frac{V}{v_1}$$

Where,
V is the volume of the tank.

Substitute $0.3\ m^3$ for V in the above equation,

$$m_1 = \frac{0.3}{0.001157}$$
$$= 259.29\ kg$$

Calculate the final mass of the given sytem,

$$m_2 = \frac{m_1}{2}$$

Where,
m_1 is the initial mass of the system,

$$m_2 = \frac{259.29}{2}$$
$$= 129.645\ kg$$

Step 4 of 5

Apply the mass balance to the given system,

$$m_e = m_1 - m_2$$
$$= 259.29 - 129.645$$
$$= 129.645\ kg$$

Calculate the final specific volume of the given system,

$$v_2 = \frac{V}{m_2}$$
$$= \frac{0.3}{129.645}$$
$$= 0.00231\ m^3/kg$$

Calculate the final quality of the given system,

$$x_2 = \frac{v_2 - v_f}{v_{fg}}$$
$$= \frac{v_2 - v_f}{v_g - v_f}$$
$$= \frac{0.00231 - 0.001157}{0.12721 - 0.001157}$$
$$= \frac{0.001153}{0.126053}$$
$$= 0.009147$$

Step 5 of 5

Calculate the final internal energy of the given system,

$$u_2 = u_f + x_2 \times u_{fg}$$
$$= 850.46 + 0.009147 \times 1743.7$$
$$= 850.46 + 15.9496$$
$$= 866.4096\ kJ/kg$$

Apply the energy balance to the given system,

$$E_{in} - E_{out} = E_{sys}$$

$$Q_{in} - m_e \times h_e = m_2 \times u_2 - m_1 \times u_1$$

$$Q_{in} = m_e \times h_e + m_2 \times u_2 - m_1 \times u_1$$
$$= 129.645 \times 852.26 + 129.645 \times 866.4096 - 259.29 \times 850.46$$
$$= 110491.247 + 112325.672 - 220515.773$$
$$= 2301.146\ kJ$$

Therefore, the heat transferred into the tank is $\boxed{2301.146\ kJ}$

Question: 6.117P

A 0.12-m3 rigid tank contains saturated refrigerant-134a at 800 kPa. Initially, 25 percent of the volume is occupied by liquid and the rest by vapor. A valve at the bottom of the tank is now opened, and liquid is withdrawn from the tank. Heat is transferred to the refrigerant such that the pressure inside the tank remains constant. The valve is closed when no liquid is left in the tank and vapor starts to come out. Determine the total heat transfer for this process.

Step 1 of 7

From tables, at pressure of R-134a in the tank $P=800\ \text{kPa}$ properties of R-134a are

$$v_f = 0.0008458\ \text{m}^3/\text{kg}$$

$$v_g = 0.025621\ \text{m}^3/\text{kg}$$

$$u_f = 94.79\ \text{kJ/kg}$$

$$u_g = 246.79\ \text{kJ/kg}$$

From tables, at pressure of R-134a in the tank $P=800\ \text{kPa}$ and saturated vapor condition, properties of R-134a are

$$v_2 = v_g = 0.025621\ \text{m}^3/\text{kg}$$

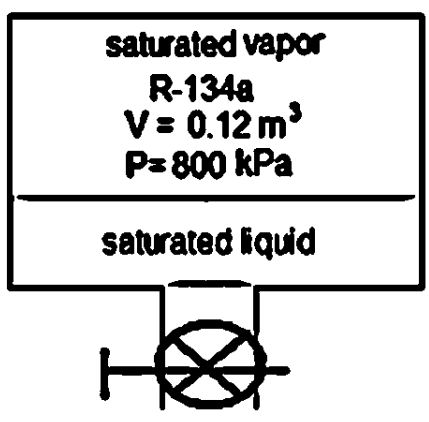
$$u_2 = u_g = 246.79\ \text{kJ/kg}$$

From tables, at pressure of R-134a in the tank $P=800\ \text{kPa}$ and saturated liquid condition, properties of R-134a are

$$h_1 = h_f = 95.47\ \text{kJ/kg}$$

Step 2 of 7

Draw the figure representing the given system,



Step 3 of 7

As 25% of refrigerant is in liquid phase, volume of liquid refrigerant is

$$V_f = 0.25 \times V$$

Where,
V is the volume of the tank.
Substitute $\left(0.12\ \text{m}^3\right)$ for V in the above equation,

$$V_f = 0.25 \times 0.12$$

$$= 0.03\ \text{m}^3$$

Step 4 of 7

As 75% of refrigerant is in vapor phase, volume of vapor refrigerant is

$$V_g = 0.75 \times V$$

Where,
V is the volume of the tank.
Substitute $\left(0.12\ \text{m}^3\right)$ for V in the above equation,

$$V_g = 0.75 \times 0.12$$

$$= 0.09\ \text{m}^3$$

Initial mass in the tank is calculated by adding the mass of liquid and vapor phases

$$m_1 = m_f + m_g$$

Mass of liquid refrigerant is calculated by dividing volume with specific volume and so

$$m_1 = \frac{V_f}{v_f} + \frac{V_g}{v_g}$$

$$= \frac{0.03}{0.0008458} + \frac{0.09}{0.025621}$$

$$= 35.4694 + 3.5127$$

$$= 38.9821\ \text{kg}$$

Step 5 of 7

Calculate the initial internal energy of the given system,

$$U_1 = m_1 \times u_1$$

$$= m_f \times u_f + m_g \times u_g$$

$$= \frac{V_f}{v_f} \times u_f + \frac{V_g}{v_g} \times u_g$$

$$= \frac{0.03}{0.0008458} \times 94.79 + \frac{0.09}{0.025621} \times 246.79$$

$$= 35.4694 \times 94.79 + 3.5127 \times 246.79$$

$$= 3362.144 + 866.899$$

$$= 4229.043\ \text{kJ}$$

Calculate the final mass of refrigerant,

$$m_2 = \frac{V}{v_2}$$

$$= \frac{0.12}{0.025621}$$

$$= 4.6837\ \text{kg}$$

Step 6 of 7

Apply the mass balance to the given system,

$$m_e = m_1 - m_2$$

$$= 38.9821 - 4.6837$$

$$= 34.2984\ \text{kg}$$

Apply the energy balance to the given system,

$$E_{in} - E_{out} = E_{sys}$$

$$Q_{in} - m_e \times h_e = m_2 \times u_2 - m_1 \times u_1$$

$$Q_{in} = m_e \times h_e + m_2 \times u_2 - U_1$$

Step 7 of 7

Substitute the respective values in the above equation,

$$Q_{in} = 34.2984 \times 95.47 + 4.6837 \times 246.79 - 4229.043$$

$$= 3274.468 + 1155.89 - 4229.043$$

$$= 201.315\ \text{kJ}$$

Therefore, the heat transferred into the tank is $\boxed{201.315\ \text{kJ}}$

Question: 6.118P

A 2-ft3 rigid tank contains saturated refrigerant-134a at 160 psia. Initially, 5 percent of the volume is occupied by liquid and the rest by vapor. A valve at the top of the tank is now opened, and vapor is allowed to escape slowly from the tank. Heat is transferred to the refrigerant such that the pressure inside the tank remains constant. The valve is closed when the last drop of liquid in the tank is vaporized. Determine the total heat transfer for this process.

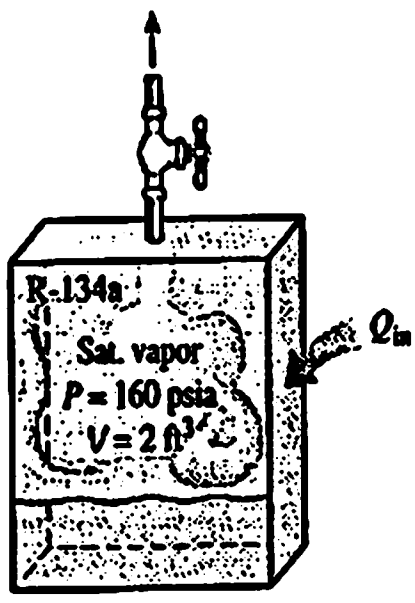


FIGURE P6–118E

Step 1 of 5

Given

The volume of the rigid tank, $V = 2\ \text{ft}^3$

Initial pressure of R-134a, $P_1 = 160$ psia

5% volume is liquid 95% is vapor of refrigerant R-134a

Step 2 of 5

From saturated R-134a tables, at 160 psia,

Specific volume of liquid, $v_f = 0.01413\ \text{ft}^3/\text{lbm}$

Specific volume of vapor, $v_g = 0.29316\ \text{ft}^3/\text{lbm}$

Internal energy of liquid, $u_f = 48.1$ Btu/lbm

Internal energy of vapor, $u_g = 108.5$ Btu/lbm

Enthalpy of vapor, $h_g = 117.18$ Btu/lbm

Step 3 of 5

The initial mass of refrigerant, $m_1 = m_f + m_g$

$$m_1 = \frac{V_f}{v_f} + \frac{V_g}{v_g}$$

$$m_1 = \frac{0.05 \times 2}{0.01413} + \frac{0.95 \times 2}{0.29316}$$

$$m_1 = 7.077 + 6.48$$

$$m_1 = 13.557\ \text{lbm}$$

Final mass of refrigerant, $m_2 = \frac{V}{v_2}$

$$m_2 = \frac{V}{v_g}$$

$$m_2 = \frac{2}{0.29316}$$

$$m_2 = 6.82\ \text{lbm}$$

Step 4 of 5

From mass balance, $m_{in} - m_{out} = \Delta m_{system}$

$$m_1 - m_2 = m_e$$

$$m_e = m_1 - m_2$$

$$m_e = 13.56 - 6.822$$

$$m_e = 6.738\ \text{lbm}$$

The initial internal energy, $U_1 = m_1 u_1$

$$U_1 = m_f u_f + m_g u_g$$

$$U_1 = (7.077)48.1 + (6.48)(108.5)$$

$$U_1 = 340.4 + 703.08$$

$$U_1 = 1043.48\ \text{Btu}$$

Step 5 of 5

The final internal energy, $U_2 = m_2 u_2$

$$U_2 = 6.82 \times 108.5$$

$$U_2 = 739.97\ \text{Btu}$$

From Energy balance, $E_{in} - E_{out} = \Delta E_{system}$

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$Q_{in} = m_e h_e + U_2 - U_1$$

$$Q_{in} = (6.738 \times 117.18) + 739.97 - 1043.48$$

$$Q_{in} = 486\ \text{Btu}$$

Hence, the heat that must be transferred is $\boxed{Q_{in} = 486\ \text{Btu}}$

Question: 6.119P

A 0.4-m3 rigid tank initially contains refrigerant-134a at 14°C. At this state, 70 percent of the mass is in the vapor phase, and the rest is in the liquid phase. The tank is connected by a valve to a supply line where refrigerant at 1 MPa and 100°C flows steadily. Now the valve is opened slightly, and the refrigerant is allowed to enter the tank. When the pressure in the tank reaches 700 kPa, the entire refrigerant in the tank exists in the vapor phase only. At this point the valve is closed. Determine (a) the final temperature in the tank, (b) the mass of refrigerant that has entered the tank, and (*c*) the heat transfer between the system and the surroundings.

Step 1 of 6

Given

Volume of the rigid tank, $V = 0.4\ \text{m}^3$

Initial temperature of R-134a, $T_1 = 14°\text{C}$

Initial quality of R-134a, $x_1 = 70\%$

Pressure of R-134a in supply line, $P_{in} = 1\ \text{MPa}$

Temperature of R-134a in supply line, $T_{in} = 100°\text{C}$

Final pressure of R-134a, $P_2 = 700\ \text{kPa}$

Final quality of R-134a, $x_2 = 100\%$

Step 2 of 6

(a)

From saturated R-134a tables, at 14°C

Specific volume of liquid, $v_f = 0.0008\ \text{m}^3/\text{kg}$

Specific volume of vapor, $v_g = 0.04342\ \text{kJ/kg}$

Specific volume initially, $v_1 = v_f + xv_{fg}$

$$v_1 = v_f + x\left(v_g - v_f\right)$$

$$v_1 = 0.0008 + 0.7(0.04342 - 0.0008)$$

$$v_1 = 0.0008 + 0.02983$$

$$v_1 = 0.03063\ \text{m}^3/\text{kg}$$

Step 3 of 6

From saturated R-134a tables, at 14°C

Internal energy of liquid, $u_f = 70.57\ \text{kJ/kg}$

Internal energy for vaporization, $u_{fg} = 167.26\ \text{kJ/kg}$

Internal energy initially, $u_1 = u_f + xu_{fg}$

$$u_1 = 70.57 + (0.7 \times 167.26)$$

$$u_1 = 70.57 + 117.082$$

$$u_1 = 187.652\ \text{kJ/kg}$$

Step 4 of 6

From saturated R-134a tables, at 700kPa

Specific volume of vapor, $v_2 = v_g = 0.02936\ \text{m}^3/\text{kg}$

Internal energy of vapor, $u_2 = u_g = 244.48\ \text{kJ/kg}$

From superheated R-134a tables, at 1 MPa and 100°C

Enthalpy of vapor, $h_i = 335.06\ \text{kJ/kg}$

Saturation temperature, $T_{sat} = 26.69°\text{C}$

As the tank contains saturated vapor, the temperature also corresponds to saturation temperature. Hence final temperature of the tank is $\boxed{T_2 = 26.69°\text{C}}$

Step 5 of 6

(b)

The initial mass in tank, $m_1 = \dfrac{V_1}{v_1}$

$$m_1 = \frac{0.4}{0.0306}$$

$$m_1 = 13.06\ \text{kg}$$

The final mass in tank, $m_2 = \dfrac{V_2}{v_2}$

$$m_2 = \frac{0.4}{0.02936}$$

$$m_2 = 13.62\ \text{kg}$$

From Mass balance, $m_{in} - m_{out} = \Delta m_{system}$

$$m_1 - m_2 = m_{in}$$

$$m_{in} = m_1 - m_2$$

$$m_{in} = 13.62 - 13.06$$

$$m_{in} = 0.56\ \text{kg}$$

Therefore the mass entered to the tank $\boxed{m_{in} = 0.56\ \text{kg}}$

Step 6 of 6

(c)

From energy balance, $E_{in} - E_{out} = \Delta E_{system}$

$$Q_{in} + m_{in}h_{in} = m_2u_2 - m_1u_1$$

$$Q_{in} = m_2u_2 - m_1u_1 - m_{in}h_{in}$$

$$Q_{in} = m_2u_2 - m_1u_1 - m_{in}h_{in}$$

$$Q_{in} = (13.62)(244.48) - (13.06)(187.65) - (0.56)(335.06)$$

$$Q_{in} = 3329.8 - 2450.71 - 187.6336$$

$$Q_{in} = 691.4565\ \text{kJ}$$

The amount of heat transfer $\boxed{Q_{in} = 691.4565\ \text{kJ}}$

Question: 6.120P

The air-release flap on a hot-air balloon is used to release hot air from the balloon when appropriate. On one hot-air balloon, the air release opening has an area of 0.5 m2, and the filling opening has an area of 1 m2. During a two minute adiabatic flight maneuver, hot air enters the balloon at 100 kPa and 35°C with a velocity of 2 m/s; the air in the balloon remains at 100 kPa and 35°C; and air leaves the balloon through the air-release flap at velocity 1 m/s. At the start of this maneuver, the volume of the balloon is 75 m3. Determine the final volume of the balloon and work produced by the air inside the balloon as it expands the balloon skin.



FIGURE P6–120

© Vol. 55/PhotoDisc/Getty Images.

Step 1 of 6

From tables, universal gas constant of air,

$$R = 0.287\ \text{kJ/kg}\cdot\text{K}$$

Specific volume of air at inlet & exit of balloon is given by gas equation, where T is temperature and P is the pressure inside the balloon

$$v = \frac{R \times T}{P}$$

$$= 0.88396\ \text{m}^3/\text{kg}$$

Substitute 35°C for T and 100 Pa for P in the above equation,

$$v = \frac{R \times T}{P}$$

$$= \frac{0.287 \times (35+273)}{100}$$

$$= \frac{0.287 \times 308}{100}$$

$$= 0.88396\ \text{m}^3/\text{kg}$$

Step 2 of 6

Mass flow rate at the entrance of the balloon is given by

$$m_i = \frac{A_i \times C_i}{v}$$

where A_i is the area of cross section and C_i is the velocity of air at entrance

Substitute 1m^2 for A_i, 2 m/s for C_i and 2 kg/s for v in the above equation,

$$m_i = \frac{1 \times 2}{0.88396}$$

$$= \frac{2}{0.88396}$$

$$= 2.249\ \text{kg/s}$$

Step 3 of 6

Mass flow rate at the exit of the balloon is given by

$$m_e = \frac{A_e \times C_e}{v}$$

where A_e is the area of cross section and C_e is the velocity of air at entrance

Substitute 0.5m^2 for A_i, 1 m/s for C_i and .88396 kg/s for v in the above equation,

$$m_e = \frac{0.5 \times 1}{0.88396}$$

$$= \frac{0.5}{0.88396}$$

$$= 0.5656\ \text{kg/s}$$

Step 4 of 6

Apply mass balance to the given system,

$$m_{in} - m_{out} = \Delta m_{sys}$$

In order to convert mass flow rate into mass we multiply it with time (Δt) and so

$$m_i \times \Delta t - m_e \times \Delta t = m_2 - m_1$$

$$m_2 - m_1 = (m_i - m_e) \times \Delta t$$

Substitute the respective values in the above equation,

$$m_2 - m_1 = (2.249 - 0.5656) \times (2 \times 60)$$

$$= 1.6834 \times 120$$

$$= 202.008\ \text{kg}$$

Step 5 of 6

Calculate the change in the volume of the balloon,

$$\Delta V = (m_2 - m_1) \times v$$

$$= (202.008) \times 0.88396$$

$$= 178.567\ \text{m}^3$$

Calculate the final volume of air in the balloon,

$$V_2 = V_1 + \Delta V$$

where V_1 is the initial volume of air in the balloon

Substitute the respective values in the above equation,

$$V_2 = 75 + 178.567$$

$$= 253.567\ \text{m}^3$$

Step 6 of 6

Calculate the work done by the air,

$$W = P \times \Delta V$$

$$= 100 \times 178.567$$

$$= 17856.7\ \text{kJ}$$

Therefore, the final volume of air in the balloon is $\boxed{V_2 = 253.567\ \text{m}^3}$ and the work done by the air is $\boxed{W = 17856.7\ \text{kJ}}$

Question: 6.121P

An insulated 0.15-m3 tank contains helium at 3 MPa and 130°C. A valve is now opened, allowing some helium to escape. The valve is closed when one-half of the initial mass has escaped. Determine the final temperature and pressure in the tank.

Step 1 of 4

Given

Volume of the insulated tank $V = 0.4\ \text{m}^3$

Initial temperature $T_1 = 130°C = 130+273 = 403K$, Initial pressure $P_1 = 3\ \text{MPa}$

$$m_2 = \frac{m_1}{2}$$

Step 2 of 4

Mass balance

$$\dot{m}_{in} - \dot{m}_{out} = \Delta m_{system}$$

$$m_e = m_1 - m_2$$

Energy balance

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} - m_e(h_e) = m_2(u_2) - m_1u_1$$

(Since $w = 0$ and $ke = 0, pe = 0$)

Step 3 of 4

From mass balance

$$m_e = m_1 - \frac{m_1}{2} = \frac{m_1}{2}$$

Energy balance

Since insulated $Q_{in} = 0$

$$0 - \frac{m_1}{2}(h_e) = \frac{m_1}{2}(u_2) - m_1u_1$$

We have $h = c_pT, u = c_vT$

Removing m_1 $$0 - \frac{1}{2}\left(c_p\left(\frac{T_1+T_2}{2}\right)\right) = \frac{1}{2}(c_vT_2) - c_vT_1$$

We know gas constant $k = \frac{c_p}{c_v}$, here for helium k = 1.667 (table A-2)

$$0 - \frac{1}{2}\left(kc_v\left(\frac{T_1+T_2}{2}\right)\right) = \frac{1}{2}(c_vT_2) - c_vT_1$$

Removing c_v $$0 - \frac{1}{2}\left((1.667)\left(\frac{403+T_2}{2}\right)\right) = \frac{1}{2}(T_2) - 403$$

The final temperature $\boxed{T_2 = 257K}$

Step 4 of 4

The final pressure in the tank

$$P_1V = m_1RT_1, P_2V = m_2RT_2$$

$$\frac{P_2}{P_1} = \frac{m_2T_2}{m_1T_1}$$

$$\frac{P_2}{3000} = \frac{1(257)}{2(403)}$$

$$\boxed{P_2 = 956.5\ \text{kPa}}$$

Question: 6.122P

An insulated 60-ft3 rigid tank contains air at 75 psia and 120°F. A valve connected to the tank is now opened, and air is allowed to escape until the pressure inside drops to 30 psia. The air temperature during this process is maintained constant by an electric resistance heater placed in the tank. Determine the electrical work done during this process.

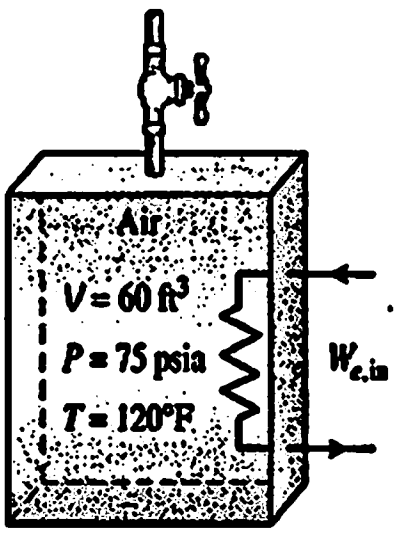


FIGURE P6–122E

Step 1 of 5

From tables, gas constant of air is

$$R = 0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$$

From tables, at temperature of air in the tank $T = 120°\text{F} = 580 \text{ R}$, properties of air are

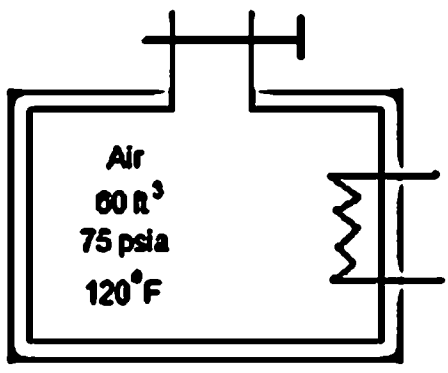
$$h_1 = 138.66 \text{ Btu/lbm}$$

$$u_1 = 98.9 \text{ Btu/lbm}$$

$$u_2 = 98.9 \text{ Btu/lbm}$$

Step 2 of 5

Draw the figure representing the given system,



Step 3 of 5

From the gas equation, initial mass of air in the tank,

$$m_1 = \frac{P_1 \times V}{T_1 \times R}$$

Substituting the value of the volume of the tank V and initial pressure P_1 in the above equation we get

$$m_1 = \frac{75 \times 60}{580 \times 0.3704}$$
$$= \frac{4500}{214.832}$$
$$= 20.947 \text{ kg}$$

Step 4 of 5

From the gas equation, final mass of air in the tank is given by

$$m_2 = \frac{P_2 \times V}{T_2 \times R}$$

Substituting the value of the volume of the tank V and initial pressure P_1 in the above equation we get

$$m_2 = \frac{30 \times 60}{580 \times 0.3704}$$
$$= \frac{1800}{214.832}$$
$$= 8.379 \text{ kg}$$

Step 5 of 5

Apply the mass balance to the given system,

$$m_e = m_1 - m_2$$
$$= 20.947 - 8.379$$
$$= 12.568 \text{ lbm}$$

Apply the energy balance to the given system,

$$E_{in} - E_{out} = E_{sys}$$
$$W_{in} - m_e \times h_e = m_2 \times u_2 - m_1 \times u_1$$
$$W_{in} = m_e \times h_e + m_2 \times u_2 - m_1 \times u_1$$

Substitute the respective values in the above equation,

$$W_{in} = 12.568 \times 138.66 + 8.379 \times 98.9 - 20.947 \times 98.9$$
$$= 1742.679 + 828.683 - 2071.658$$
$$= 499.704 \text{ Btu}$$

Therefore, the work done on the tank is $\boxed{499.704 \text{ Btu}}$

Question: 6.123P

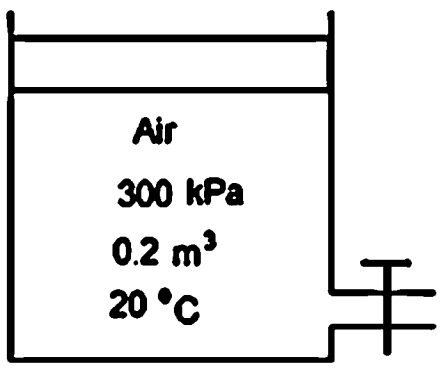
A vertical piston–cylinder device initially contains 0.2 m3 of air at 20°C. The mass of the piston is such that it maintains a constant pressure of 300 kPa inside. Now a valve connected to the cylinder is opened, and air is allowed to escape until the volume inside the cylinder is decreased by one-half. Heat transfer takes place during the process so that the temperature of the air in the cylinder remains constant. Determine (a) the amount of air that has left the cylinder and (b) the amount of heat transfer.

Step 1 of 5

From tables, gas constant of air is

$$R = 0.287 \text{ kJ/kg°K}$$

Draw the figure representing the given system,



Step 2 of 5

Initial mass of air in the cylinder is given by the gas equation

$$m_1 = \frac{P_1 \times V_1}{R \times T_1}$$

where P_1 is the initial pressure, T_1 is the initial temperature and V_1 is the initial volume of air in the cylinder

Substitute 300 kPa for P_1, 0.2 m³ for V_1, 20°C for T_1 and the respective values in the above equation,

$$\begin{aligned} m_1 &= \frac{300 \times 0.2}{0.287 \times (20+273)} \\ &= \frac{300 \times 0.2}{0.287 \times 293} \\ &= \frac{60}{84.091} \\ &= 0.7134 \text{ kg} \end{aligned}$$

Step 3 of 5

The final volume is half of the initial volume and so

$$V_2 = \frac{V_1}{2}$$

Final mass of air in the cylinder is given by the gas equation

$$m_2 = \frac{P_2 \times V_2}{R \times T_2}$$

where P_2 is the final pressure, T_2 is the final temperature and V_2 is the final volume of air in the cylinder

Substitute 300 kPa for P_2, 0.2 m³ for V_1, 20°C for T_1 and the respective values in the above equation,

$$\begin{aligned} m_2 &= \frac{P_2}{R \times T_2} \times \frac{V_1}{2} \\ &= \frac{300}{0.287 \times (20+273)} \times \frac{0.2}{2} \\ &= \frac{300 \times 0.1}{0.287 \times 293} \\ &= 0.3567 \text{ kg} \end{aligned}$$

Step 4 of 5

(a)

Apply the mass balance to the given system,

$$\begin{aligned} m_{in} - m_{out} &= \Delta m_{sys} \\ 0 - m_e &= m_2 - m_1 \\ -m_e &= m_2 - m_1 \\ m_e &= m_1 - m_2 \\ &= 0.7134 - 0.3567 \\ &= 0.3657 \text{ kg} \end{aligned}$$

Step 5 of 5

As the there is no temperature of air in the cylinder the enthalpies will be same and so

$$h_1 = h_2 = h_e$$

(b)

Apply the energy balance to the given system,

$$\begin{aligned} E_{in} - E_{out} &= \Delta E_{sys} \\ Q_{in} - m_e \times h_e &= m_2 \times h_2 - m_1 \times h_1 \\ Q_{in} &= m_e \times h_e + m_2 \times h_2 - m_1 \times h_1 \\ Q_{in} &= m_e \times h_1 + m_2 \times h_1 - m_1 \times h_1 \\ &= (m_e + m_2 - m_1) \times h_1 \\ &= (0.7134 + 0.3657 - 0.3657) \times h_1 \\ &= 0 \times h_1 \\ &= 0 \end{aligned}$$

Therefore, the amount of air left in the cylinder is $\boxed{m_e = 0.3657 \text{ kg}}$ and the heat transferred into the cylinder is $\boxed{Q_{in} = 0}$

Question: 6.124P pg 1

An insulated vertical piston–cylinder device initially contains 0.8 m3 of refrigerant-134a at 1.2 MPa and 120°C. A linear spring at this point applies full force to the piston. A valve connected to the cylinder is now opened, and refrigerant is allowed to escape. The spring unwinds as the piston moves down, and the pressure and volume drop to 0.6 MPa and 0.5 m3 at the end of the process. Determine (a) the amount of refrigerant that has escaped and (b) the final temperature of the refrigerant.

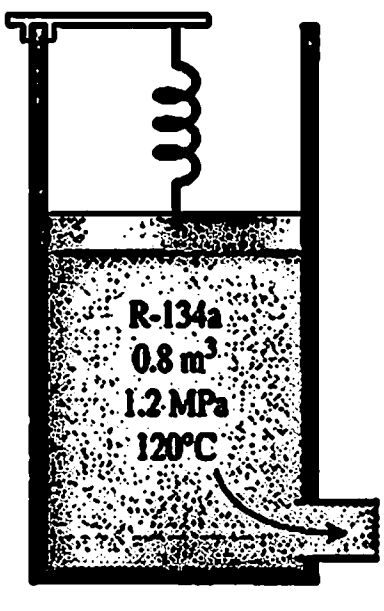
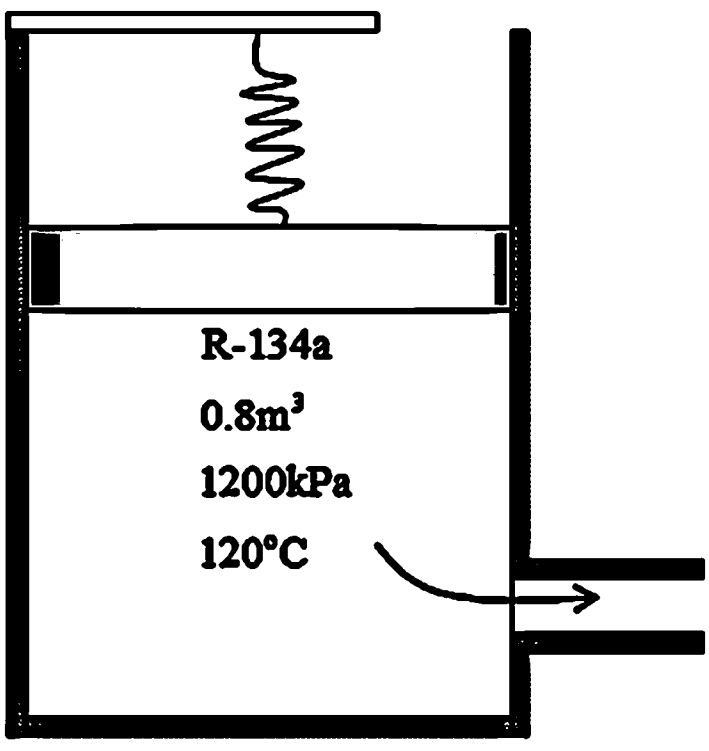


FIGURE P6–124

Step 1 of 12

Draw the Schematic figure of the piston cylinder.



Step 2 of 12

Obtain the properties of R-134a at a Pressure of 1200 kPa and temperature $120°C$ from the table A-13 "saturated refrigerant -134 a".

Specific volume at state-1 is $v_1 = 0.02423 m^3/kg$,

Specific internal energy at state-1is $u_1 = 325.03 kJ/kg$

Specific enthalpy at state-1is $h_1 = 354.11 kJ/kg$

Step 3 of 12

Calculate the initial mass of the refrigerant inside the cylinder.

$$m_1 = \frac{V_1}{v_1}$$

Here, the initial volume of the cylinder is V_1 and the Specific volume at state-1 is v_1 .

Substitute $0.8 m^3$ for V_1 and $0.02423 m^3/kg$ for v_1 .

$$m_1 = \frac{V_1}{v_1}$$
$$= \frac{0.8(m^3)}{0.02423(m^3/kg)}$$
$$= 33.02 kg$$

Step 4 of 12

Calculate the final mass of the refrigerant inside the cylinder,

$$m_2 = \frac{V_2}{v_2}$$

Here, the final volume of the cylinder is V_2 and the Specific volume at state-2 is v_2 .

Substitute $0.5 m^3$ for .

$$m_2 = \frac{V_2}{v_2}$$
$$= \frac{0.5}{v_2}$$

Apply mass balance to the system.

For a steady flow device,

$$\dot{m}_{in} - \dot{m}_{out} = \Delta m_{system}$$
$$m_e = m_1 - m_2$$

Substitute 33.02 kg for m_2 and $\frac{0.5}{v_2}$ for m_1 .

$$m_e = m_2 - m_1$$
$$= \left(33.02 - \frac{0.5}{v_2}\right) \quad \text{..... (1)}$$

Step 5 of 12

Calculate the boundary work during the process assuming average pressure as shown below:

Step 5 of 12

Calculate the boundary work during the process assuming average pressure as shown below:

$$W_{b,in} = \frac{P_1 + P_2}{2}(V_1 - V_2)$$

Here, the initial pressure inside the cylinder is P_1 and the final pressure inside the cylinder is P_2.

Substitute $1200\,\text{kPa}$ for P_1, $600\,\text{kPa}$ for P_2, $0.8\,\text{m}^3$ for V_1 and $0.5\,\text{m}^3$ for V_2,

$$\begin{aligned} W_{b,in} &= \frac{P_1 + P_2}{2}(V_1 - V_2) \\ &= \frac{(1200+600)}{2}(\text{kPa}) \times (0.8-0.5)(\text{m}^3) \\ &= 270\,\text{kJ} \end{aligned}$$

Step 6 of 12

The enthalpy of fluid at the exit is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder. Therefore,

$$h_e = \frac{h_1 + h_2}{2}$$

Substitute $354.11\,\text{kJ/kg}$ for h_1 .

$$h_e = \frac{(354.11 + h_2)}{2}(\text{kJ/kg})$$

Step 7 of 12

Apply energy balance to the system.

$$E_{in} - E_{out} = \Delta E_{system}$$

$$W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1$$

Substitute 270kJ for $W_{b,in}$, $\left(33.02 - \frac{0.5}{v_2}\right)$ for m_e , $\frac{(354.11 + h_2)}{2}$ for h_e , $\left(\frac{0.5}{v_2}\right)$ for m_2 ,

$325.03\,\text{kJ/kg}$ for u_1 , and 33.02 kg for m_1 .

$$270 - \left(33.02 - \frac{0.5}{v_2}\right)\left[\frac{(354.11 + h_2)}{2}\right] = \left(\frac{0.5}{v_2}\right)u_2 - 33.02 \times 325.03 \quad \ldots\ldots (3)$$

Final state properties of the refrigerant R-134a h_2, u_2 and v_2 are all functions of final pressure (known) and temperature (unknown).

The solution may be obtained by a trial-error approach by trying different final state temperatures or by using EES software until equation (3) is satisfied.

Step 8 of 12

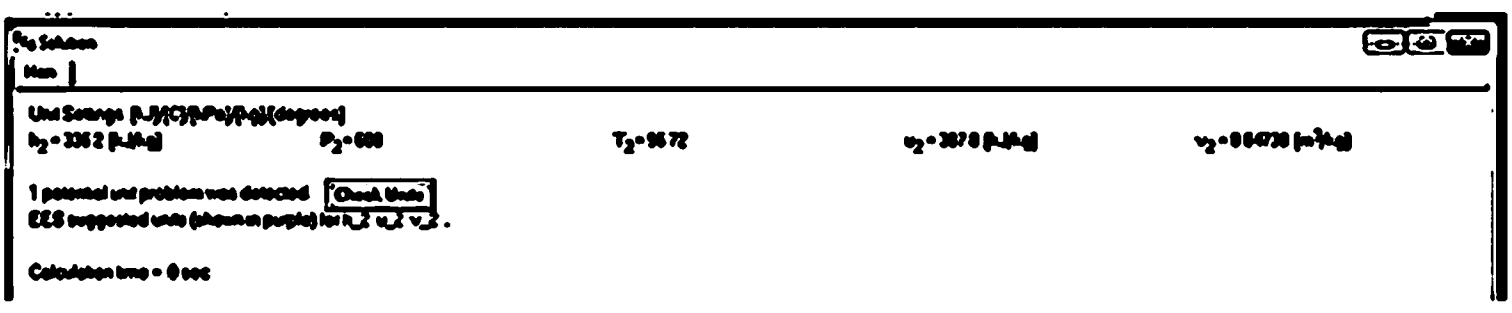
Write the equations in EES software as shown below.

Equations Window

```
h_2=Enthalpy(R134a,T=T_2,P=P_2)
u_2=IntEnergy(R134a,T=T_2,P=P_2)
v_2=Volume(R134a,T=T_2,P=P_2)
270-((33.02-(0.5/v_2)))*((354.11+h_2)/2)=((0.5/v_2)*u_2)-(33.02*325.03)
P_2=0.6*1000
```

Step 9 of 12

The results obtained by solving the above equation are shown below:



Step 10 of 12

The properties at the final state are:

The temperature at final state is $T_2 = 96.72°\text{C}$

The enthalpy at final state is $h_2 = 336.20\ \text{kJ/kg}$

The internal energy at final state is $u_2 = 307.8\ \text{kJ/kg}$

The specific volume at final state is $v_2 = 0.04738\ \text{m}^3/\text{kg}$

Step 11 of 12

(a)

Calculate the amount of refrigerant that has escaped.

$$m_e = \left(33.02 - \frac{0.5}{v_2}\right)$$

Substitute $0.04739\ \text{m}^3/\text{kg}$ for v_2 in equation (1).

$$\begin{aligned} m_e &= \left(33.02 - \frac{0.5}{0.047438}\right) \\ &= 22.46\ \text{kg} \end{aligned}$$

Therefore, the amount of refrigerant escaped is $\boxed{22.46\ \text{kg}}$

Step 12 of 12

(b)

The final temperature of the refrigerant is $\boxed{96.72°\text{C}}$.

Step 12 of 12

(b)

The final temperature of the refrigerant is $\boxed{96.72^{\circ}C}$.

Question: 6.125P pg 1

A piston-cylinder device initially contains 0.6 kg of steam with a volume of 0.1 m3. The mass of the piston is such that it maintains a constant pressure of 800 kPa. The cylinder is connected through a valve to a supply line that carries steam at 5 MPa and 500°C. Now the valve is opened and steam is allowed to flow slowly into the cylinder until the volume of the cylinder doubles and the temperature in the cylinder reaches 250°C, at which point the valve is closed. Determine (a) the mass of steam that has entered and (b) the amount of heat transfer.

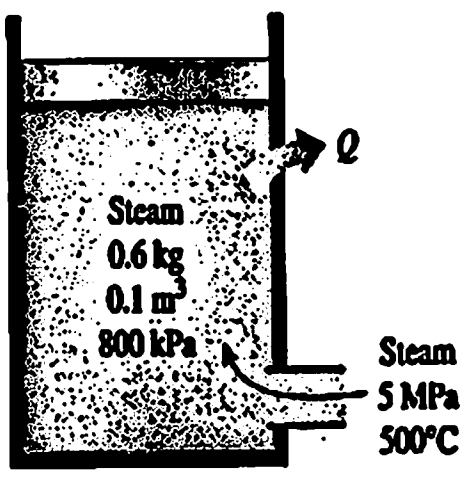


FIGURE P6–125

Step 1 of 10

Initial specific volume of steam in the cylinder is,

$v_1 = \frac{V_1}{m_1}$ Here, the initial volume and initial mass of steam in the cylinder respectively are V_1 and m_1.

Step 2 of 10

Substitute, $0.1\ m^3$ for V_1 and, 0.6 kg for m_1 in the equation,

$$\begin{aligned} v_1 &= \frac{V_1}{m_1} \\ &= \frac{0.1}{0.6} \\ &= 0.1667\ m^3 \end{aligned}$$

Step 3 of 10

Initial specific volume is also given by

$$\begin{aligned} v_1 &= v_{f1} + x_1 \times v_{fg1} \\ &= v_{f1} + x_1 \times (v_{g1} - v_{f1}) \end{aligned}$$

Here, the saturated liquid and saturated vapor specific volumes at that pressure and x_1 is the initial quality of steam are v_{f1} and v_{g1}.

From saturated water pressure table, at initial pressure of steam 800 kPa. Substitute, $0.1667\ m^3$ for v_1, $0.001115\ m^3$ for v_{f1} and, $0.24035\ m^3$ for v_{g1} in the equation,

$$\begin{aligned} v_1 &= v_{f1} + x_1 \times (v_{g1} - v_{f1}) \\ 0.1667 &= 0.001115 + x_1 \times (0.24035 - 0.001115) \\ 0.1667 &= 0.001115 + x_1 \times 0.239235 \\ x_1 &= \frac{0.1667 - 0.001115}{0.239235} \\ &= \frac{0.165585}{0.239235} \\ &= 0.6921 \end{aligned}$$

Step 4 of 10

Initial internal energy is, $u_1 = u_{f1} + x_1 \times u_{fg1}$ Here, the saturated liquid and vaporization specific volumes at that pressure are u_{f1} and u_{fg1}.

Step 5 of 10

From saturated water pressure table, at initial pressure of steam 800 kPa. Substitute, $710.97\ kJ/kg$ for u_{f1}, $1856.1\ kJ/kg$ for u_{fg1} and, 0.6921 for x_1 in the equation,

$$\begin{aligned} u_1 &= u_{f1} + x_1 \times u_{fg1} \\ &= 710.97 + 0.6921 \times 1856.1 \\ &= 710.97 + 1284.607 \\ &= 1995.577\ kJ/kg \end{aligned}$$

Step 6 of 10

The final volume in the cylinder is double the initial volume

$V_2 = 2 \times V_1$

Here, the initial volume and initial mass of steam in the cylinder respectively are V_1 and m_1.

Substitute, $0.1\ m^3$ for V_1 and, 0.6 kg for m_1 in the equation,

$$\begin{aligned} V_2 &= 2 \times V_1 \\ &= 2 \times 0.1 \\ &= 0.2\ m^3 \end{aligned}$$

Step 7 of 10

Final mass of steam in the cylinder is

$m_2 = \frac{V_2}{v_2}$

Here, specific volume at stage 2 is v_2.

Since the cylinder pressure is constant from superheated water table, at pressure $800\ kPa$ and, temperature $T_2 = 250\ °C$, specific volume and internal energy are $0.29321\ m^3/kg$

Substitute, $0.29321\ m^3/kg$ for v_2 and, $0.2\ m^3$ for V_2 in the equation,

$$\begin{aligned} m_2 &= \frac{V_2}{v_2} \\ &= \frac{0.2}{0.29321} \end{aligned}$$

$$m_2 = \frac{V_2}{v_2}$$
$$= \frac{0.2}{0.29321}$$
$$= 0.6821\,\text{kg}$$

Mass of steam entered the cylinder is

$$m_f = m_2 - m_1$$

Here, mass of steam entered in the cylinder at stage 1 is m_1 and, mass of steam entered in the cylinder at stage 2 is m_2

Substitute, 0.6 kg for m_1 and, 0.6821 kg for m_2 in the equation,

$$m_f = m_2 - m_1$$
$$= 0.6821 - 0.6$$
$$= 0.0821\,\text{kg}$$

The net mass entered the cylinder is $\boxed{0.0821\,\text{kg}}$.

Step 8 of 10

Work output is given by the formula,

$$W_{out} = P \times (V_2 - V_1)$$

Here, pressure is denoted by *P*, volume at stage 1 is V_1 and, volume at stage 2 is V_2.

Substitute, 800 kPa for *P*, $0.1\,\text{m}^3$ for V_1 and, $0.2\,\text{m}^3$ for V_2 in the equation,

$$W_{out} = P \times (V_2 - V_1)$$
$$= 800 \times (0.2 - 0.1)$$
$$= 800 \times 0.1$$
$$= 80\,\text{kJ}$$

Step 9 of 10

Energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$
$$Q_{in} + m_f \times h_f - W_{out} = m_2 \times u_2 - m_1 \times u_1$$
$$Q_{in} = m_2 \times u_2 - m_1 \times u_1 - m_f \times h_f + W_{out}$$

Here, mass of the steam at stage 1 is m_1 , internal energy of the steam at stage 1 is u_1 , mass of the steam at stage 2 is m_2 , internal energy of the steam at stage 2 is u_2, net mass of steam entered m_f , enthalpy of steam is denoted as h_f and, work output be W_{out}.

Since the cylinder pressure is constant from superheated water table, at pressure $800\,\text{kPa}$ and, temperature $T_2 = 250\,^\circ\text{C}$, internal energy is obtained as 2715.9 kJ/kg. From superheated water table, at pressure in the pipe $P_f = 5\,\text{MPa}$ and temperature in the pipe $T_f = 500\,^\circ\text{C}$, enthalpy of steam is 3434.7 kJ/kg.

Step 10 of 10

Substitute, 0.6821 kg for m_2, 2715.9 kJ/kg for u_2, 0.6 kg for m_1, 1995.577 kg for u_1, 0.0821 kg for m_f, 3434.7 kJ/kg for h_f and, 80 kJ for W_{out} in the equation,

$$Q_{in} = m_2 \times u_2 - m_1 \times u_1 - m_f \times h_f + W_{out}$$
$$= 0.6821 \times 2715.9 - 0.6 \times 1995.577 - 0.0821 \times 3434.7 + 80$$
$$= 1852.515 - 1197.346 - 281.989 + 80$$
$$= 453.18\,\text{kJ}$$

Therefore, mass of steam that entered is $\boxed{m_f = 0.0821\,\text{kg}}$ and the heat input to the cylinder is $\boxed{Q_{in} = 453.18\,\text{kJ}}$

Question: 6.126P

An adiabatic piston-cylinder device equipped with a spring maintains the pressure inside at 300 kPa when the volume is 0, and 3000 kPa when the volume is 5 m3. The device is connected to a steam line maintained at 1500 kPa and 200°C, and initially the volume is 0. Determine the final temperature (and quality if appropriate) when valve is opened and steam in the line is allowed to enter the cylinder until the pressure inside matches that in the line. Also determine the total work produced during this adiabatic filling process.

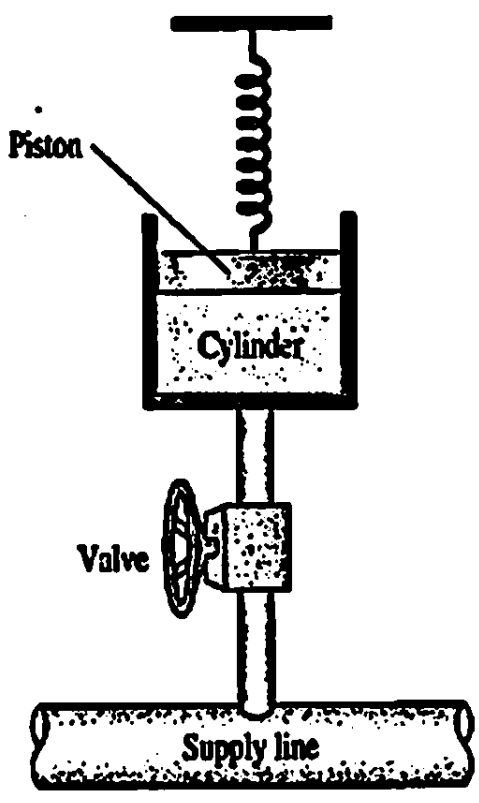
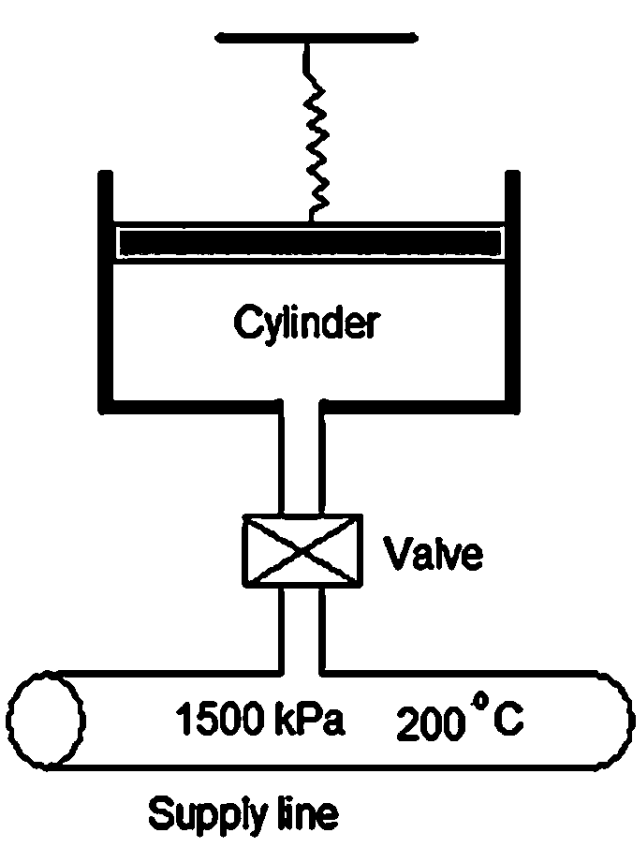


FIGURE P6–126

Step 1 of 5

A piston-cylinder device equipped with a spring:



Step 2 of 5

At pressure $300\ \text{kPa}$, volume in the cylinder is $0\ \text{m}^3$ and at pressure $3000\ \text{kPa}$, volume in the cylinder is $5\ \text{m}^3$. So for the pressure difference of $P_{diff} = 2700\ \text{kPa}$ (i.e., $3000-300$) the difference in the volume of cylinder will be

$$V_{diff} = 5-0$$
$$= 5\ \text{m}^3$$

Step 3 of 5

Find the final vapor volume using the relation between pressure and volume is $P_2 - P_1 = \frac{P_{diff}}{V_{diff}} \times (V_2 - V_1)$

Where P_1 and P_2 are the initial and final pressures in the cylinder and V_1 & V_2 are the initial and final volumes of the cylinder Substitute 1500 kPa for P_2, 300 kPa for P_1, $5\ \text{m}^3$ for V_{diff}, and $2700\ \text{kPa}$ for P_{diff}

$$1500-300 = \frac{2700}{5} \times (V_2 - 0)$$

in the above equation

$$1200 = 540 \times V_2$$
$$V_2 = \frac{1200}{540}$$
$$= 2.22\ \text{m}^3$$

Step 4 of 5

Find the work needed to compress the spring:

$$W = \int_{V_1}^{V_2} P \times dV$$
$$= \int_0^{V_2} \left(\frac{2700}{5} \times V + 300 \right) \times dV$$
$$= \left(\frac{2700}{5 \times 2} \times V^2 + 300 \times V \right)_0^{V_2}$$
$$= \left(\frac{2700}{5 \times 2} \times V_2^2 + 300 \times V_2 \right)$$
$$= 270 \times 2.22^2 + 300 \times 2.222$$
$$= 2000\ \text{kJ}$$

Therefore, the work needed to compress the spring is $\boxed{2000\ \text{kJ}}$.

Step 5 of 5

As the cylinder is evacuated initially ($m_1 = 0$) the final mass of air is equal to the inlet mass of air

$$m_2 = m_i$$

From ideal gas specific heats table, gas constant, specific heat at constant pressure and specific heat at constant volume of air respectively are

$R = 0.287\ \text{kJ/kg°K}$ $c_p = 1.005\ \text{kJ/kg°K}$ $c_v = 0.718\ \text{kJ/kg°K}$ From the table, super-heated water

The enthalpy of steam at the inlet for $P = 1500\ \text{kPa}$ and $T = 200°\text{C}$ is $h_i = 2796.0$

Apply Energy balance to the given system:

$$E_{in} - E_{out} = \Delta E_{sys}$$
$$m_i \times h_i - W = m_2 \times u_2$$
$$W = m_i \times h_i - m_2 \times u_2$$
$$= m_2 (h_i - u_2)$$
$$= \frac{V_2}{v_2}(h_i - u_2)$$
$$2000 = \frac{2.222}{v_2}(2796.0 - u_2)$$

Now we need to find the temperature from tables such that the internal energy and specific volume satisfies the relation obtained above. By trial and error method, the value of temperature is found to be around

$\boxed{T_2 = 233.2\,°\text{C}}$

The properties at this temperature are

$T_2 = 233.2°\text{C}$

$u_2 = 2664.8\ \text{kJ/kg}$

$v_2 = 0.1458\ \text{m}^3/\text{kg}$

Therefore, the temperature of the cylinder is $\boxed{233.2\ °\text{C}}$

Question: 6.127P

Repeat Prob. 6–126 when the line is filled with air that is maintained at 2000 kPa and 327°C.

Step 1 of 5

Given

Initial pressure, $P_1 = 300$ kPa

Initial volume, $V_1 = 0\ \text{m}^3$

Final pressure, $P_2 = 3000$ kPa

Final volume, $V_2 = 5\ \text{m}^3$

Supply line pressure, $P_i = 2000$ kPa

Supply line temperature, $T_i = 327°\text{C}$

$$T_i = 600\ \text{K}$$

From properties of air table,

Gas constant, $R = 0.287$ kJ/kg.K

Specific heat at constant pressure, $c_p = 1.005$ kJ/kg.K

Specific heat at constant volume, $c_v = 0.718$ kJ/kg.K

Step 2 of 5

Assuming the pressure of spring to be linear, we can write

$$(P-300)5 = (3000-300)V_2$$

$$(P-300)5 = 2700 \times V_2$$

$$P = (540 \times V_2) + 300$$

Also, $(3000-300)V_2 = (2000-300)5$

$$V_2 = \frac{(1700)5}{(2700)}$$

$$V_2 = 3.148\ \text{m}^3$$

Step 3 of 5

The work needed to compress the spring by the air is

$$W_{out} = \int PdV$$

$$W_{out} = \int_0^{V_2} \left[(540 \times V_2) + 300\right] dV$$

$$W_{out} = 540\frac{V_2^2}{2} + 300V_2$$

$$W_{out} = 540\frac{(3.148)^2}{2} + 300(3.148)$$

$$W_{out} = 3620\ \text{kJ}$$

Step 4 of 5

Considering the piston-cylinder system, $q = 0$ as the gas is filled adiabatically. Also, the kinetic and potential energies of the fluid stream are negligible so $ke = 0, pe = 0$

Applying mass balance for this system,

$$\dot{m}_{in} = \dot{m}_{out}$$

$$\dot{m}_i + \dot{m}_1 = \dot{m}_2$$

As the initial volume is zero, the initial mass is also zero,

$$\dot{m}_i = \dot{m}_2$$

Applying energy balance for this system,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_{in}h_{in} - W_{out} = \dot{m}_{out}u_{out}$$

$$\dot{m}_i h_i - W_{out} = \dot{m}_2 u_2$$

$$W_{out} = \dot{m}_i h_i - \dot{m}_2 u_2$$

$$W_{out} = \dot{m}_2 (h_i - u_2)$$

Step 5 of 5

Substituting the values, we get

$$W_{out} = \dot{m}_2 (h_i - u_2)$$

But $h = c_p T$ and $u = c_v T$. Hence

$$W_{out} = \dot{m}_2 (c_p T_i - c_v T_2)$$

$$W_{out} = \frac{P_2 V_2}{RT_2}(c_p T_i - c_v T_2)$$

$$3620 = \frac{2000 \times 3.148}{0.287 \times T_2}\left[(1.005 \times 600) - (0.718 \times T_2)\right]$$

$$(0.165 \times T_2) = 603 - (0.718 \times T_2)$$

$$(0.883 \times T_2) = 603$$

$$T_2 = 682.899\ \text{K}$$

Therefore the final temperature is $\boxed{T_2 = 682.899\ \text{K}}$

Question: 6.128P

Carbon dioxide gas enters a throttling valve at 5 MPa and 100°C and leaves at 100 kPa. Determine the temperature change during this process if CO2 is assumed to be (a) an ideal gas and (b) a real gas. Real gas properties of CO2 may be obtained from EES.

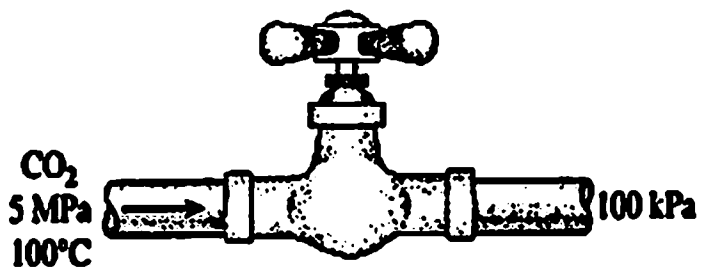
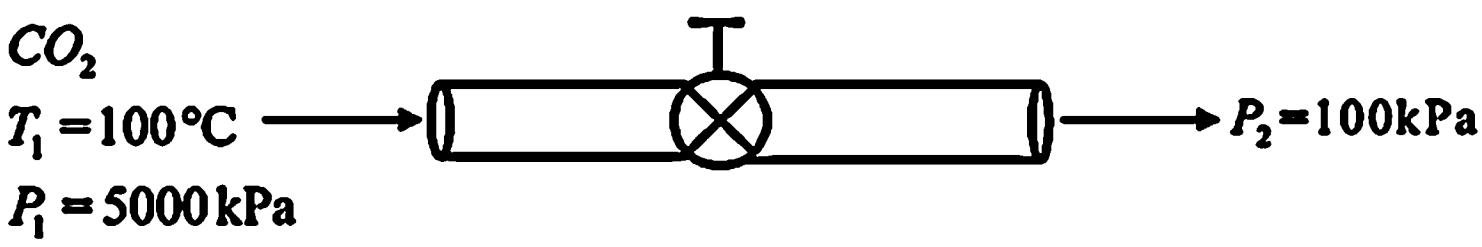


FIGURE P6–128

Step 1 of 3

Schematic diagram of the throttle valve is shown below.



Here, inlet temperature of Carbon dioxide is T_1, inlet pressure of the gas is P_1

and exit pressure of the gas is P_2

Step 2 of 3

Apply the continuity of mass relation

$$m_1 = m_2$$
$$= m$$

Assume the throttle valve as control volume;

Write the energy balance for a steady flow system

Rate of net Energy transfer = Rate of change in the energy of the system

$$E_{in} - E_{out} = \Delta E_{system}$$

For steady state condition, Apply energy conservation law (energy stored within the system is zero).

$$E_{in} = E_{out}$$
$$m\left(h_1 + V_1^2/2\right)_{in} = m\left(h_2 + V_2^2/2\right)_{out}$$

Here, h_1, h_2 are the enthalpies at inlet ant outlet respectively.

$$0 = m\left(h_1 - h_2 + 0\right)$$
$$h_1 = h_2$$

(a)

For an ideal gas, consider the equation for change in enthalpy

$$h_1 - h_1 = c_p\left(T_2 - T_1\right)$$
$$0 = c_p\left(T_2 - T_1\right)$$
$$T_2 = T_1$$

Therefore, for an ideal gas during throttling process the change in temperature is equal to zero.

Step 3 of 3

(b)

For a real gas, using EES software at a pressure $P_1 = 5000\,kPa$ and a temperature $T_1 = 100\,°C$ for CO_2 , obtain the value of enthalpy.

Enthalpy at inlet, $h_1 = 34.77\,kJ/kg$.

For a real gas, using EES software at a pressure $P_2 = 100\,kPa$ and exit enthalpy $h_2 = h_1 = 34.77\,kJ/kg$ for CO_2 , obtain the value of the exit temperature.

Exit temperature, $T_2 = 66\,°C$

Calculate the temperature change during the throttling process:

$$\Delta T = T_1 - T_2$$
$$= 100 - 66$$
$$= 34°C$$

Therefore, when Carbon dioxide is a real gas, the temperature decreases by $\boxed{34\,°C}$

Question: 6.129P

Helium steadily enters a pipe with a mass flow rate of 8 kg/s at 427°C and 100 kPa and leaves the pipe at 27°C. The pressure during the process is constant at 100 kPa. (*a*) Determine the heat transfer for the process, in kW. (*b*) Determine the volume flow rate of the helium at the pipe exit, in m3/s.

Step 1 of 4

Given

Inlet pressure of Helium, $P_1 = 100$ kPa

Inlet temperature of Helium, $T_1 = 427°C$

$T_1 = 700K$

The mass flow rate of Helium, $\dot{m} = 8$ kg/s

Exit pressure of Helium, $P_2 = 100$ kPa

Exit temperature of Helium, $T_2 = 27°C$

$T_2 = 300K$

Step 2 of 4

(a)

From the properties table of helium, at room temperature,

Gas constant, $R = 2.0769$ kJ/kg.K

Specific heat at constant pressure, $c_v = 3.1156$ kJ/kg.K

Specific heat at constant pressure, $c_p = 5.1926$ kJ/kg.K

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

$$\dot{Q} = \dot{m}_{out}(h_{out} - h_{in})$$

Step 3 of 4

But enthalpy, $h = C_p T$. Therefore,

$$\dot{Q} = \dot{m}_{out} C_p (T_{out} - T_{in})$$

$$\dot{Q} = \dot{m} C_p (T_2 - T_1)$$

$$\dot{Q} = 8 \times 5.1926(700 - 300)$$

$$\dot{Q} = 16616.32 \text{ kW}$$

Hence heat transfer for the process is $\boxed{\dot{Q} = 16616.32 \text{ kW}}$

Step 4 of 4

(b)

The specific volume at end, $v_2 = \dfrac{RT_2}{P_2}$

$$v_2 = \frac{2.0769(300)}{100}$$

$$v_2 = 6.2307 \text{ m}^3/\text{kg}$$

The volume flow rate at the end is $\dot{V}_2 = \dot{m} v_2$

$$\dot{V}_2 = 8 \times 6.2307$$

$$\dot{V}_2 = 49.63 \text{ m}^3/\text{s}$$

The volume flow rate at the exit $\boxed{\dot{V}_2 = 49.63 \text{ m}^3/\text{s}}$

Question:

6.130P

Underground water is being pumped into a pool whose cross section is 6 m × 9 m while water is discharged through a 7-cm-diameter orifice at a constant average velocity of 4 m/s. If the water level in the pool rises at a rate of 2.5 cm/min, determine the rate at which water is supplied to the pool, in m3/s.

Step 1 of 3

Given

Cross section area of the of pool, $A_{pool} = 6\text{ m} \times 9\text{ m}$

$$A_{pool} = 54\text{ m}^2$$

Diameter of the orifice, $d = 7\text{ cm}$

$$d = 0.07\text{ m}$$

Velocity of the water in the orifice, $V_e = 4\text{ m/s}$

Change in level of the water in the pool, $\dot{V}_L = 2.5\text{ cm/min}$

$$\dot{V}_L = 4.1667 \times 10^{-4}\text{ m/s}$$

Step 2 of 3

Area of orifice, $A_e = \dfrac{\pi d^2}{4}$

$$A_e = \frac{\pi \times 0.07^2}{4}$$

$$A_e = 3.85 \times 10^{-3}\text{ m}^2$$

Volume flow rate at exit, $\dot{V}_e = A_e V_e$

$$\dot{V}_e = \left(3.85 \times 10^{-3}\right) \times (4)$$

$$\dot{V}_e = 0.0154\text{ m}^3/\text{s}$$

Step 3 of 3

The amount of water collected in the pool,

$$\dot{V}_{pool} = A_{pool} V_L$$

$$\dot{V}_{pool} = 54 \times \left(4.1667 \times 10^{-4}\right)$$

$$\dot{V}_{pool} = 0.0225\text{ m}^3/\text{s}$$

Therefore the rate at which water supplied to the pool is sum of volume flow rate of water discharge and water collected.

$$\dot{V}_{total} = \dot{V}_{pool} + \dot{V}_e$$

$$\dot{V}_{total} = 0.0225 + 0.0154$$

$$\dot{V}_{total} = 0.0379\text{ m}^3/\text{s}$$

The rate of discharge of water, $\boxed{\dot{V}_{total} = 0.0379\text{ m}^3/\text{s}}$

Question: 6.131P

Air at 4.18 kg/m3 enters a nozzle that has an inlet-to-exit area ratio of 2:1 with a velocity of 120 m/s and leaves with a velocity of 380 m/s. Determine the density of air at the exit.

Step 1 of 2

Density of air at nozzle inlet,

$$\rho_1 = 4.18\ \text{kg/m}^3$$

There is only one entry and exit and so the mass flow rate is constant

$$m_1 = m_2$$

$$\rho_1 \times A_1 \times C_1 = \rho_2 \times A_2 \times C_2$$

where ρ is the density of air, A is the area of cross-section, C is the velocity of air, variables with suffix 1 are inlet parameters and variables with suffix 2 are exit parameters.

Step 2 of 2

Calculate the density of air at nozzle exit,

$$\rho_2 = \rho_1 \times \frac{A_1}{A_2} \times \frac{C_1}{C_2}$$

The ratio of area of cross-section inlet to exit is 2,

$$\rho_2 = 4.18 \times 2 \times \frac{120}{380}$$

$$= 4.18 \times 2 \times 0.3158$$

$$= 2.64\ \text{kg/m}^3$$

Therefore, the density of air at the nozzle exit is $\boxed{2.64\ \text{kg/m}^3}$

Question: 6.132P

An air compressor compresses 15 L/s of air at 120 kPa and 20°C to 800 kPa and 300°C while consuming 6.2 kW of power. How much of this power is being used to increase the pressure of the air versus the power needed to move the fluid through the compressor?

Step 1 of 4

Given

Pressure of air at inlet of compressor, $P_1 = 120\ \text{kPa}$

Temperature of air at inlet of compressor, $T_1 = 20°\text{C}$

$$T_1 = 293\ \text{K}$$

The volume flow rate at the inlet, $\dot{V} = 15\ \text{L/s}$

$$\dot{V} = 0.015\ \text{m}^3/\text{s}$$

Pressure of air at exit of compressor, $P_2 = 800\ \text{kPa}$

Temperature of air at exit of compressor, $T_2 = 300°\text{C}$

$$T_2 = 573\ \text{K}$$

Power consumed by the compressor, $\dot{W}_{in} = 6.2\ \text{kW}$

Step 2 of 4

From properties table of air, gas constant of air is $R = 0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$

The specific volume at inlet, $v_1 = \dfrac{RT_1}{P_1}$

$$v_1 = \frac{0.287(293)}{120}$$

$$v_1 = 0.7\ \text{m}^3/\text{kg}$$

The mass flow rate, $\dot{m} = \dfrac{\dot{V}}{v_1}$

$$\dot{m} = \frac{0.015}{0.7}$$

$$\dot{m} = 0.0214\ \text{kg/s}$$

Step 3 of 4

Flow work in compressor, $\dot{W}_{flow} = \dot{m}(P_2 v_2 - P_1 v_1)$

$$\dot{W}_{flow} = \dot{m}(RT_2 - RT_1)$$

$$\dot{W}_{flow} = \dot{m}R(T_2 - T_1)$$

$$\dot{W}_{flow} = 0.0214 \times 0.287 \times (573 - 293)$$

$$\dot{W}_{flow} = 1.72\ \text{kW}$$

Therefore the flow power $\boxed{\dot{W}_{flow} = 1.72\ \text{kW}}$

Step 4 of 4

The power used to increases the pressure in air is

$$\dot{W}_p = \dot{W}_{in} - \dot{W}_f$$

$$\dot{W}_p = 6.2 - 1.72$$

$$\dot{W}_p = 4.48\ \text{kW}$$

Hence the power used to increases the pressure in air is $\boxed{\dot{W}_p = 4.48\ \text{kW}}$

Question: 6.133P

A steam turbine operates with 1.6 MPa and 350°C steam at its inlet and saturated vapor at 30°C at its exit. The mass flow rate of the steam is 16 kg/s, and the turbine produces 9000 kW of power. Determine the rate at which heat is lost through the casing of this turbine.

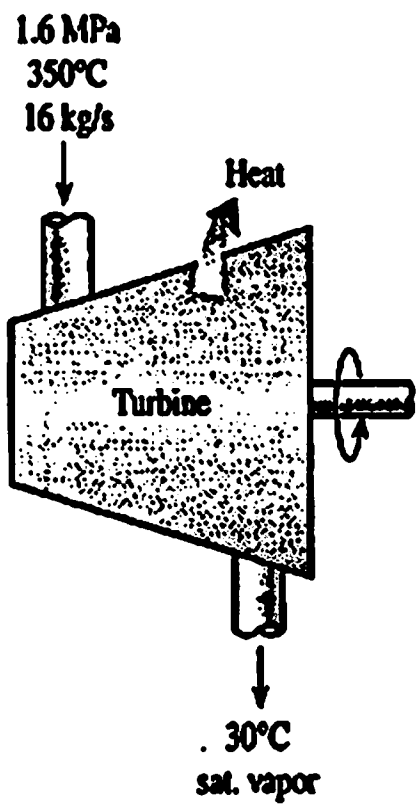


FIGURE P6–133

Step 1 of 4

From the table of superheated steam

At turbine inlet pressure $P_1 = 1.6\ \text{MPa}$ and turbine inlet temperature $T_1 = 350\ ^\circ\text{C}$ specific enthalpy of steam is

$$h_1 = 3146\ \text{kJ/kg}$$

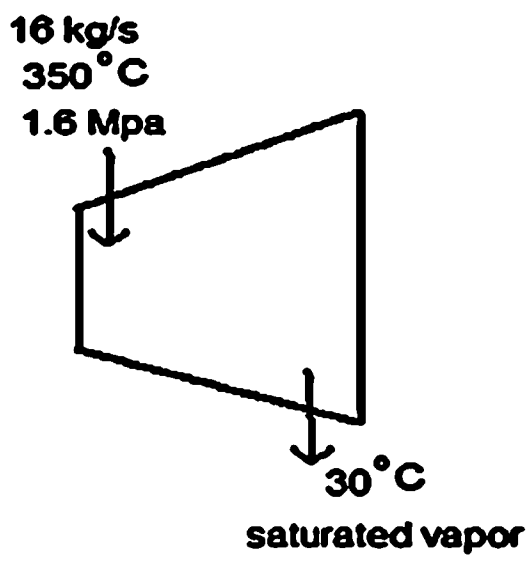
From table of saturated water

At turbine exit temperature $T_2 = 30\ ^\circ\text{C}$ and saturated vapor condition specific enthalpy of steam is

$$h_2 = h_g = 2555.6\ \text{kJ/kg}$$

Step 2 of 4

Schematic diagram of turbine system:



Step 3 of 4

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$m_1 \times h_1 = m_2 \times h_2 + W_{out} + Q_{out}$$

where Q_{out} is heat loss from the turbine and W_{out} is the work done by the turbine.

m_1 is the mass flow rate of steam at inlet and m_2 is the mass flow rate of steam at outlet

As there is only one entry and exit to the turbine, the mass flow rate is constant

$$m_1 = m_2 = m$$

$$m \times h_1 = m \times h_2 + W_{out} + Q_{out}$$

$$Q_{out} = m \times h_1 - m \times h_2 - W_{out}$$

$$Q_{out} = m \times (h_1 - h_2) - W_{out}$$

Step 4 of 4

Substituting 16kg/s for m, 3146kJ/kg for h_1, 2555.6 kJ/kg for h_2, 9000kW for W_{out}

$$Q_{out} = 16 \times (3146 - 2555.6) - 9000$$

$$= 16 \times 590.4 - 9000$$

$$= 9446.4 - 9000$$

$$= 446.4\ \text{kW}$$

Therefore, the rate of heat loss from the turbine is $\boxed{446.4\ \text{kW}}$

Question: 6.134P

Nitrogen gas flows through a long, constant-diameter adiabatic pipe. It enters at 100 psia and 120°F and leaves at 50 psia and 70°F. Calculate the velocity of the nitrogen at the pipe's inlet and outlet.

Step 1 of 7

From table of specific heats of common gases "
For nitrogen
Specific heat of nitrogen is $c_p = 0.248\ \text{Btu/lbm} \cdot \text{R}$
As there is one entry and exit for the pipe, the mass flow rate will be same. So,

$$m_1 = m_2$$
$$\frac{A_1 \times C_1}{v_1} = \frac{A_2 \times C_2}{v_2}$$
$$C_2 = \frac{A_1}{A_2} \times \frac{v_2}{v_1} \times C_1$$

Here,
Mass flow rate of nitrogen at inlet of pipe is m_1
Mass flow rate of nitrogen at inlet of pipe is m_2
Cross sectional area of the section at inlet pipe is A_1
Cross sectional area of the section at exit pipe is A_2
The velocities of nitrogen at entry is C_1
The velocities of nitrogen at exit is C_2
Specific volume of nitrogen at inlet of the pipe is v_1
Specific volume of nitrogen at exit of the pipe is v_2

Step 2 of 7

From the ideal gas equation, ratio of specific volumes is given by the ratio of temperature and pressure

$$C_2 = \frac{A_1}{A_2} \times \frac{T_2}{T_1} \times \frac{P_1}{P_2} \times C_1$$

Here,
Temperature of the nitrogen at inlet of pipe is T_1
Temperature of the nitrogen at exit of pipe is T_2
As the diameter of the pipe is constant, ratio of area of cross-section of the pipe will be unity.

$$C_2 = 1 \times \frac{T_2}{T_1} \times \frac{P_1}{P_2} \times C_1$$
$$C_2 = \frac{T_2}{T_1} \times \frac{P_1}{P_2} \times C_1 \qquad \text{..... (1)}$$

Step 3 of 7

Write the energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$m_1\left(h_1 + \frac{C_1^2}{2}\right) = m_2\left(h_2 + \frac{C_2^2}{2}\right)$$

As $m_1 = m_2$ the above equation becomes,

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$
$$\frac{C_2^2}{2} - \frac{C_1^2}{2} = h_1 - h_2$$
$$\frac{C_2^2 - C_1^2}{2} = c_p \times T_1 - c_p \times T_2$$
$$\frac{C_2^2 - C_1^2}{2} = c_p \times (T_1 - T_2)$$

Step 4 of 7

Substituting equation (1) in the above equation,

$$\frac{\left(\frac{T_2}{T_1} \times \frac{P_1}{P_2} \times C_1\right)^2 - C_1^2}{2} = c_p \times (T_1 - T_2)$$
$$\left(\frac{T_2}{T_1} \times \frac{P_1}{P_2}\right)^2 \times C_1^2 - C_1^2 = 2 \times c_p \times (T_1 - T_2)$$
$$\left[\left(\frac{T_2}{T_1} \times \frac{P_1}{P_2}\right)^2 - 1\right] \times C_1^2 = 2 \times c_p \times (T_1 - T_2)$$
$$C_1^2 = \frac{2 \times c_p \times (T_1 - T_2)}{\left(\frac{T_2}{T_1} \times \frac{P_1}{P_2}\right)^2 - 1}$$

Step 5 of 7

Substituting $0.248\ \text{Btu/lbm} \cdot \text{R}$ for c_p, $120°\text{F}$ for T_1, $70°\text{F}$ for T_2, $100\ \text{psia}$ for P_1, $50\ \text{psia}$ for P_2

$$C_1^2 = \frac{2 \times 0.248 \times (120 - 70)}{\left[\frac{(70 + 460)}{(120 + 460)} \times \frac{100}{50}\right]^2 - 1}$$

In order to convert Btu/lbm into ft^2/s^2 we have to multiply numerator of RHS with 25037

$$C_1^2 = \frac{2 \times 0.248 \times (120 - 70) \times 25037}{\left[\frac{530}{580} \times \frac{100}{50}\right]^2 - 1}$$
$$C_1^2 = \frac{2 \times 0.248 \times 50 \times 25037}{[0.9138 \times 2]^2 - 1}$$

Step 6 of 7

$$C_1^2 = \frac{620917.6}{2.34}$$
$$C_1^2 = \frac{620917.6}{2.34}$$
$$C_1^2 = 265349.4$$
$$C_1 = \sqrt{265349.4}$$

Hence, the velocities of nitrogen at entry is $\boxed{515.12\ \text{ft/s}}$

Step 7 of 7

From equation (1) we get,

$$C_2 = \frac{T_2}{T_1} \times \frac{P_1}{P_2} \times C_1$$
$$= \frac{530}{580} \times \frac{100}{50} \times 515.12$$
$$= 0.9138 \times 2 \times 515.12$$
$$= 941.43\ \text{ft/s}$$

Hence, the velocities of nitrogen at exit is $\boxed{941.43\ \text{ft/s}}$

Question: 6.135P

A 110-V electric hot-water heater warms 0.1 L/s of water from 18 to 30°C. Calculate the current in amperes that must be supplied to this heater.

Step 1 of 3

Given

Inlet temperature of water, $T_1 = 18°C$

The volume flow rate of water, $\dot{V} = 0.1\ L/s$

$\dot{V} = 0.0001\ m^3/s$

Exit temperature of water, $T_2 = 30°C$

Voltage supplied to heater, $V = 110\ V$

Specific heat of water, $c_p = 4.18\ kJ/kg°C$

Step 2 of 3

From the mass balance equation for the conservation of mass,

$$\dot{m}_{in} = \dot{m}_{out}$$

Mass, $\dot{m} = \rho \dot{V}$

$$\dot{m} = 1000 \times (0.0001)$$

$$\dot{m} = 0.1\ kg/s$$

From the energy balance equation for the conservation of energy,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}_{in} h_{in} = \dot{m}_{out} h_{out}$$

$$\dot{W}_{in} = \dot{m}(h_{out} - h_{in})$$

Step 3 of 3

But $h = c_p T$ and $\dot{W}_{in} = VI$ hence

$$VI = \dot{m} c_p (T_{out} - T_{in})$$

$$I = \frac{\dot{m} c_p}{V}(T_{out} - T_{in})$$

$$I = \frac{0.1 \times 4.8}{100}(30 - 18)$$

$$I = \frac{0.1 \times (4.18 \times 1000)}{110}(30 - 18)$$

$$I = 45.6\ A$$

The current supplied to the heater in amperes, $\boxed{I = 45.6\ A}$

Question: 6.136P

Water is boiled at 100°C electrically by a 3-kW resistance wire. Determine the rate of evaporation of water.

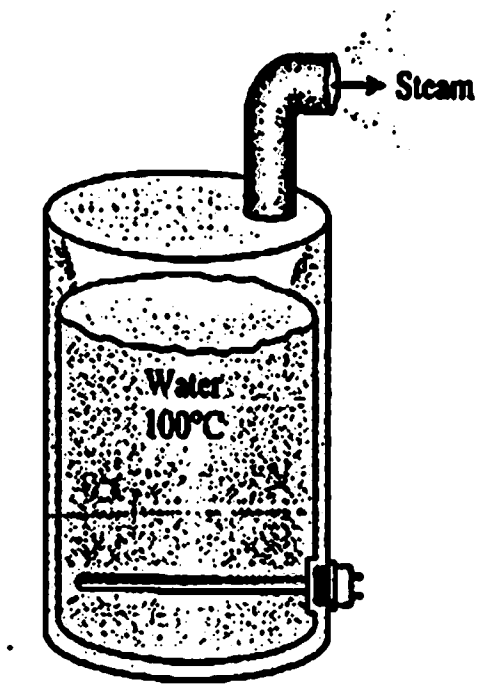


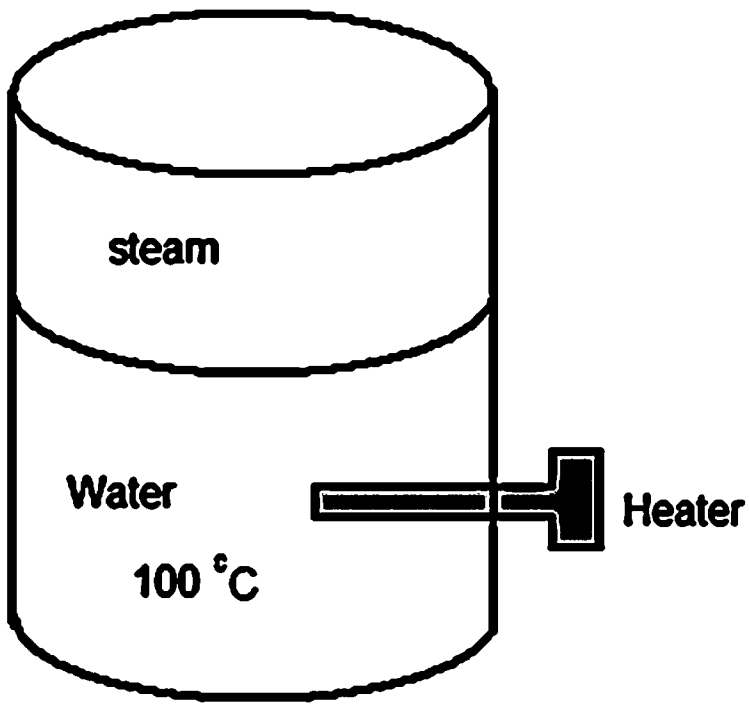
FIGURE P6–136

Step 1 of 3

From the table of Saturation water table "
At boiling temperature of water $T = 100\ ^\circ C$
Latent heat vaporization is $h_{fg} = 2256.4\ kJ/kg^\circ \cdot K$

Step 2 of 3

Schematic diagram water heated in tank system:



Step 3 of 3

Calculate the rate of water evaporated is

$$W_{boiling} = h_{fg} \times m_{evap}$$

Where m_{evap} mass of water evaporated.

$$m_{evap} = \frac{W_{boiling}}{h_{fg}}$$

Substitute 3kW for W, 2256.4kJ/kg for h_{fg}

$$m_{evap} = \frac{3}{2256.4}$$
$$= 0.001329\ kg/s$$

Therefore, rate of evaporation of water is $\boxed{0.001329\ kg/s}$

Question: 6.137P

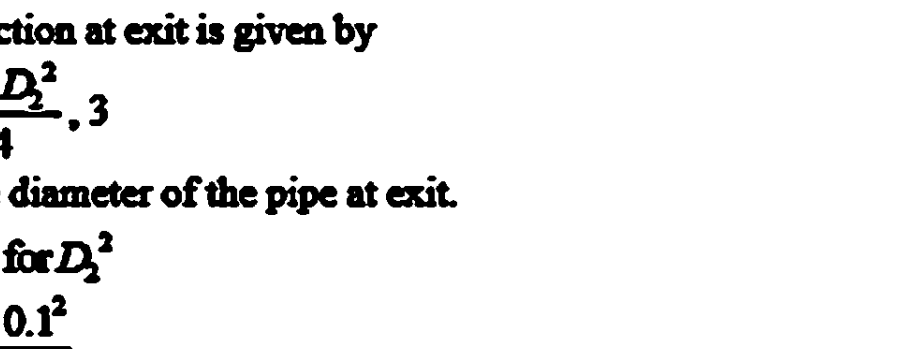
Steam enters a long, insulated pipe at 1400 kPa, 350°C, and 10 m/s, and exits at 1000 kPa. The diameter of the pipe is 0.15 m at the inlet, and 0.1 m at the exit. Calculate the mass flow rate of the steam and its speed at the pipe outlet.

Step 1 of 8

From table of properties of steam,
at entrance pressure $P_1 = 1400\ \text{kPa}$ and entrance temperature $T_1 = 350\ ^\circ\text{C}$ properties of steam are
Specific volume is $v_1 = 0.20029\ \text{m}^3/\text{kg}$
Enthalpy is $h_1 = 3150.1\ \text{kJ/kg}$

Step 2 of 8

Schematic diagram flow system:



Step 3 of 8

Area of cross-section at entrance is given by

$$A_1 = \frac{\pi \times D_1^2}{4},$$

Where D_1 is the diameter of the pipe at entrance
Substitute $0.15\,\text{m}$ for D^2

$$A_1 = \frac{\pi \times 0.15^2}{4}$$
$$= \frac{\pi \times 0.0225}{4}$$
$$= \frac{0.0707}{4}$$
$$= 0.017675\ \text{m}^2$$

Step 4 of 8

Area of cross-section at exit is given by

$$A_2 = \frac{\pi \times D_2^2}{4}, 3$$

Where D_2 is the diameter of the pipe at exit.
Substitute $0.1\,\text{m}$ for D_2^2

$$A_2 = \frac{\pi \times 0.1^2}{4}$$
$$= \frac{\pi \times 0.01}{4}$$
$$= \frac{0.0314}{4}$$
$$= 0.00785\ \text{m}^2$$

Step 5 of 8

Write the energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$m_{in} h_1 = m_{out} h_2$$

There is only one entrance and one exit. So, mass flow rate of steam is constant $m_{in} = m_{out}$ and so the equation becomes

$$h_1 = h_2$$

Step 6 of 8

From table of properties of steam ,
At pressure $P_2 = 1000\ \text{kPa}$ and enthalpy $h_2 = 3150.1\ \text{kJ/kg}$,
Specific volume of steam is interpolated as

$$\frac{v_2 - 0.25799}{0.28250 - 0.25799} = \frac{3150.1 - 3051.6}{3158.2 - 3051.6}$$
$$\frac{v_2 - 0.25799}{0.02451} = \frac{98.5}{106.6}$$
$$v_2 = 0.924 \times 0.02451 + 0.25799$$
$$v_2 = 0.02265 + 0.25799$$
$$v_2 = 0.28064\ \text{m}^3/\text{kg}$$

Step 7 of 8

Calculate mass flow rate is

$$m = \frac{A_1 \times C_1}{v_1}$$

Where C_1 is the velocity and v_1 is the specific volume at entrance of the pipe

$$m = \frac{0.017675 \times 10}{0.20029}$$
$$= \frac{0.17675}{0.20029}$$
$$= 0.88247\ \text{kg/s}$$

Hence, mass flow rate is $\boxed{0.88247\ \text{kg/s}}$

Step 8 of 8

Calculate exit velocity is

$$C_2 = \frac{v_2 \times m}{A_2}$$

Substitute $0.28064\,\text{kg/m}^3$ for v_2, $0.88247\,\text{kg/s}$ for m, $0.00785\ \text{m}^2$ for A_2

$$C_2 = \frac{0.28064 \times 0.88247}{0.00785}$$
$$= \frac{0.24765}{0.00785}$$
$$= 31.547\ \text{m/s}$$

Therefore, the mass flow rate of steam at exit velocity is $\boxed{31.547\ \text{m/s}}$

Question: 6.138P pg 1

The air flow in a compressed air line is divided into two equal streams by a T-fitting in the line. The compressed air enters this 2.5-cm diameter fitting at 1.6 MPa and 40°C with a velocity of 50 m/s. Each outlet has the same diameter as the inlet, and the air at these outlets has a pressure of 1.4 MPa and a temperature of 36°C. Determine the velocity of the air at the outlets and the rate of change of flow energy (flow power) across the T-fitting.

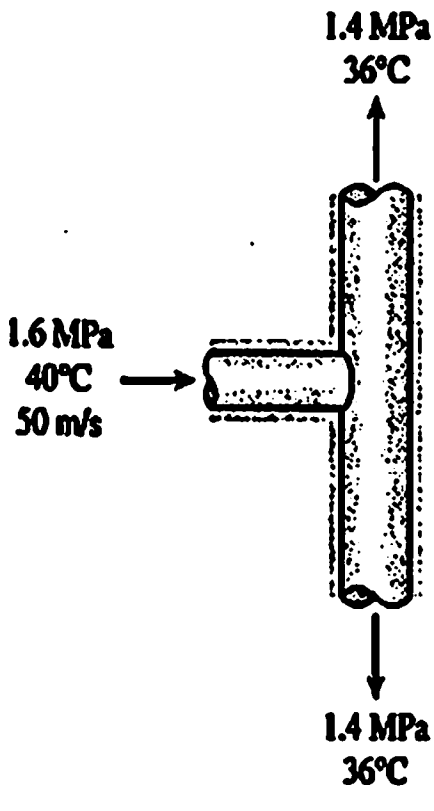
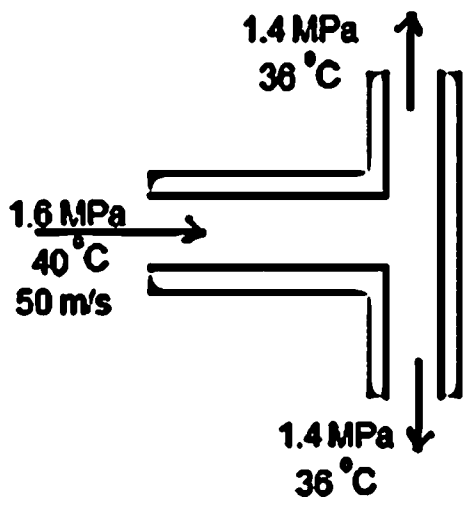


FIGURE P6–138

Step 1 of 8

Schematic diagram of mixing chamber:



Step 2 of 8

From table of properties of common gases
For gas constant of air is

$$R = 0.287 \text{ kJ/kg} \cdot ^\circ\text{K}$$

Specific volume at inlet of line is given by

$$v_1 = \frac{R \times T_1}{P_1},$$

Where T_1 is the temperature and P_1 is the pressure at inlet of line
Substitute $0.287 \text{ kJ/kg} \cdot ^\circ\text{K}$ for R, 313K for T_1, 1600 kPA for P_1

$$v_1 = \frac{0.287 \times (313)}{1600}$$
$$= \frac{0.287 \times 313}{1600}$$
$$= \frac{89.831}{1600}$$
$$= 0.05614 \text{ m}^3/\text{kg}$$

Step 3 of 8

Let us assume that air is divided evenly after the T-junction.
Specific volume at both the exits of line are given by

$$v_2 = v_3 = \frac{R \times T_2}{P_2},$$

Where T_2 is the temperature and P_2 is the pressure at exit of line
Substitute $0.287 \text{ kJ/kg} \cdot ^\circ\text{K}$ for R, 309K for T_2, 1400 kPA for P_1

$$v_2 = \frac{0.287 \times (36+273)}{1400}$$
$$= \frac{0.287 \times 309}{1400}$$
$$= \frac{88.683}{1400}$$
$$v_2 = v_3 = 0.06334 \text{ m}^3/\text{kg}$$

Step 4 of 8

As the diameters of line at inlet and exits is same, area of cross-section will also be same

$$A = \frac{\pi \times D^2}{4},$$

where D is the diameter of the pipe
Substitute 0.025m for D

$$A = \frac{\pi \times 0.025^2}{4}$$
$$= \frac{\pi \times 0.000625}{4}$$
$$= \frac{0.001963}{4}$$
$$= 0.00049075 \text{ m}^2$$

Step 5 of 8

As there is one inlet and two exits, mass flow rate at inlet is double the mass flow rate at exits

$$m_1 = 2m_2$$
$$\frac{A_1 \times C_1}{v_1} = 2\frac{A_2 \times C_2}{v_2},$$

Where C is the velocity and v is the specific volume

$$C_2 = \frac{A_1 \times C_1 \times v_2}{2 \times v_1 \times A_2}$$

$\frac{A_1 \times C_1}{v_1} = 2\frac{A_2 \times C_2}{v_2}$.

Where C is the velocity and v is the specific volume

$$C_2 = \frac{A_1 \times C_1 \times v_2}{2 \times v_1 \times A_2}$$

$$= \frac{C_1 \times v_2}{2 \times v_1}$$

Substitute 50 m/s for C_1, $0.06334\ m^3/kg$ for v_2, $0.05614\ m^3/kg$ for v_1

$$C_2 = \frac{50 \times 0.06334}{2 \times 0.05614}$$

$$= \frac{3.167}{0.11228}$$

$$= 28.206\ m/s$$

Hence, velocity at outlet is $\boxed{28.206\ m/s}$

Step 6 of 8

Calculate Mass flow rate at inlet is

$$m_1 = \frac{A \times C_1}{v_1}$$

Substitute $0.00049075\ m^2$ for A, 50 m/s for C, $0.05614\ m^3/kg$ for v_1

$$m_1 = \frac{0.00049075 \times 50}{0.05614}$$

$$= \frac{0.0245375}{0.05614}$$

$$= 0.43708\ kg/s$$

Step 7 of 8

Mass flow rates at exit is given by

$$m_2 = \frac{A \times C_2}{v_2}$$

Substitute $0.00049075\ m^2$ for A, 28.206 m/s for C_2, $0.06334\ m^3/kg$ for v_2

$$m_2 = \frac{0.00049075 \times 28.206}{0.06334}$$

$$= \frac{0.01384}{0.06334}$$

$$= 0.21854\ kg/s$$

Step 8 of 8

Calculate Change in work flow energy is

$$W_{flow} = 2 \times E_{out} - E_{in}$$

$$= 2 \times m_2 \times P_2 \times v_2 - m_1 \times P_1 \times v_1$$

$$= 38.758 - 39.2603$$

$$= -0.5023\ kW$$

Hence, the decrease in energy is $\boxed{0.5023\ kW}$

Question: 6.139P

Air enters a pipe at 65°C and 200 kPa and leaves at 60°C and 175 kPa. It is estimated that heat is lost from the pipe in the amount of 3.3 kJ per kg of air flowing in the pipe. The diameter ratio for the pipe is D1/D2 = 1.4. Using constant specific heats for air, determine the inlet and exit velocities of the air.

Step 1 of 4

Obtain properties of air from properties of gases table,

Gas constant is $R = 0.287\ \text{kJ/kg}\cdot\text{K}$

Specific heat is $c_p = 1.005\ \text{kJ/kg}\cdot\text{K}$

Apply mass balance by using the equation.

$$m_{in} - m_{out} = \Delta m_{sys}$$

Here, m_{in} is mass flow of air at inlet, m_{out} is mass flow of air at exit and Δm_{sys} is change in mass of air in the system.

For steady flow, $\Delta m_{sys} = 0$

$$m_{in} - m_{out} = 0$$
$$m_{in} = m_{out}$$
$$\rho_1 \times A_1 \times V_1 = \rho_2 \times A_2 \times V_2$$
$$\frac{P_1}{R \times T_1} \times \frac{\pi}{4} D_1^2 \times V_1 = \frac{P_2}{R \times T_2} \times \frac{\pi}{4} D_2^2 \times V_2$$

Here, P is pressure, T is temperature, D is diameter, V is velocity, ρ is density and R is gas constant. Parameters with suffix 1 are parameters at inlet and parameters with suffix 2 are parameters at exit.

Step 2 of 4

Rearrange the terms and simplify the equation.

$$\frac{P_1}{T_1} \times D_1^2 \times V_1 = \frac{P_2}{T_2} \times D_2^2 \times V_2$$
$$\frac{V_2}{V_1} = \frac{P_1}{P_2} \times \frac{T_2}{T_1} \times \frac{D_1^2}{D_2^2}$$
$$\frac{V_2}{V_1} = \frac{P_1}{P_2} \times \frac{T_2}{T_1} \times \left(\frac{D_1}{D_2}\right)^2$$

Substitute $200\,\text{kPa}$ for P_1, $175\,\text{kPa}$ for P_2, $(60+273 = 333\,\text{K})$ for T_2, $(65+273 = 338\,\text{K})$ for T_1, and 1.8 for $\frac{D_1}{D_2}$.

$$\frac{V_2}{V_1} = \frac{200}{175} \times \frac{(60+273)}{(65+273)} \times (1.4)^2$$
$$= \frac{200}{175} \times \frac{333}{338} \times 1.96$$
$$= 1.1428 \times 0.985 \times 1.96$$
$$= 2.206$$

$$V_2 = 2.206 \times V_1 \quad \text{..... (1)}$$

Step 3 of 4

Apply Energy balance to the system by using the equation,

$$E_{in} - E_{out} = \Delta E_{sys}$$

For steady flow, $\Delta E_{sys} = 0$

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{out}$$
$$c_p \times T_1 + \frac{V_1^2}{2} = c_p \times T_2 + \frac{V_2^2}{2} + q_{out}$$

Here, c_p specific heat of air, and q_{out} is the heat loss from pipe.

$$c_p \times T_1 + \frac{V_1^2}{2} = c_p \times T_2 + \frac{V_2^2}{2} + q_{out}$$

Step 4 of 4

Substitute $1.005 \times 10^3\ \text{J/kg}\cdot\text{K}$ for c_p, $338\,\text{K}$ for T_1, $3.3 \times 10^3\ \text{J/kg}$ for q_{out} and 333 K for T_2.

$$1.005 \times 10^3 \times 338 + \frac{V_1^2}{2} = 1.005 \times 10^3 \times 333 + \frac{V_2^2}{2} + 3.3 \times 10^3$$
$$\frac{V_2^2}{2} - \frac{V_1^2}{2} = 1.725 \times 10^3$$
$$V_2^2 - V_1^2 = 3.45 \times 10^3$$

Substitute $2.206 \times V_1$ for V_2 obtained from equation (1).

$$(2.206 \times V_1)^2 - V_1^2 = 3.45 \times 10^3$$
$$4.8664 \times V_1^2 - V_1^2 = 3.45 \times 10^3$$
$$3.8664 V_1^2 = 3.45 \times 10^3$$
$$V_1 = 29.87\ \text{m/s}$$

Therefore, the velocity of air at inlet is **29.87 m/s**.

Find the velocity of air at exit by using equation (1)

$$V_2 = 2.206 \times V_1$$
$$= 2.206 \times 29.87$$
$$= 65.893\ \text{m/s}$$

Hence, velocity of air at exit is **65.893 m/s**.

Question: 6.140P

Steam enters a nozzle with a low velocity at 150°C and 200 kPa, and leaves as a saturated vapor at 75 kPa. There is a heat transfer from the nozzle to the surroundings in the amount of 26 kJ for every kilogram of steam flowing through the nozzle. Determine (a) the exit velocity of the steam and (b) the mass flow rate of the steam at the nozzle entrance if the nozzle exit area is 0.001 m2.

Step 1 of 5

From superheated water tables take the properties of steam at nozzle inlet pressure $P_1 = 200\ \text{kPa}$ and nozzle inlet temperature $T_1 = 150°\text{C}$.

Specific enthalpy of steam is $h_1 = 2769.1\ \text{kJ/kg}$

From saturated water tables at nozzle exit pressure $P_2 = 75\ \text{kPa}$ and saturated vapor condition

Specific volume $v_2 = v_g = 2.2172\ \text{m}^3/\text{kg}$

Specific enthalpy of steam $h_2 = h_g = 2662.4\ \text{kJ/kg}$

As there is only one entry, one exit for nozzle, mass flow rate of steam will be same

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

Step 2 of 5

(a)

Apply the steady state energy balance

$$E_{in} - E_{out} = \Delta E_{sys}$$

For steady flow process, $\Delta E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$\dot{m} \times \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \times \left(h_2 + \frac{V_2^2}{2} \right) + Q_{out}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{out}$$

Here V_1 and V_2 are the inlet and exit velocities of steam and q_{out} is the specific heat lost from nozzle.

Step 3 of 5

Neglect the inlet velocity of the nozzle, the energy equation becomes,

$$h_1 = h_2 + \frac{V_2^2}{2} + q_{out}$$

Write the exit velocity by rearranging the terms in the above equation,

$$\frac{V_2^2}{2} = h_1 - h_2 - q_{out}$$

$$V_2^2 = 2 \times (h_1 - h_2 - q_{out})$$

$$V_2 = \sqrt{2 \times (h_1 - h_2 - q_{out})}$$

In order to convert m^2/s^2 into kJ/kg we have to multiply RHS with 1000

$$V_2 = \sqrt{2 \times 1000 \times (h_1 - h_2 - q_{out})}$$

$$= \sqrt{2000 \times (h_1 - h_2 - q_{out})}$$

Step 4 of 5

Substitute $26\ \text{kJ/kg}$ for q_{out}.

$$V_2 = \sqrt{2000 \times (2769.1 - 2662.4 - 26)}$$

$$= \sqrt{2000 \times 80.7}$$

$$= \sqrt{161400}$$

$$= 401.75\ \text{m/s}$$

Therefore, exit velocity of the steam from the nozzle is $\boxed{401.75\ \text{m/s}}$

Step 5 of 5

(b)

Find mass flow rate of steam by using the equation,

$$\dot{m} = \frac{A_2 \times V_2}{v_2}$$

Here A_2 is cross-section area of nozzle at exit

Substitute $0.001\ \text{m}^2$ for A_2

$$\dot{m} = \frac{0.001 \times 401.75}{2.2172}$$

$$= \frac{0.40175}{2.2172}$$

$$= 0.18119\ \text{kg/s}$$

Therefore, the mass flow rate of steam is $\boxed{0.18119\ \text{kg/s}}$

Question: 6.141P

In a gas-fired boiler, water is boiled at 150°C by hot gases flowing through a stainless steel pipe submerged in water. If the rate of heat transfer from the hot gases to water is 74 kJ/s, determine the rate of evaporation of water.

Step 1 of 1

From table properties of water,

At temperature $T = 150\ ^\circ C$

Specific enthalpy of evaporation is

$$h_{fg} = 2113.8\ kJ/kg^\circ K$$

Rate of heat transfer is given by

$$Q_{boiling} = h_{fg} \times m_{evap}$$

Where m_{evap} mass of water evaporated.

Calculate the rate at which water is evaporated is

$$m_{evap} = \frac{Q_{boiling}}{h_{fg}}$$

Substitute $75\ kJ/s$ for $Q_{boiling}$, $2113.8 kJ/kg$ for h_{fg}

$$m_{evap} = \frac{74}{2113.8}$$
$$= 0.0355\ kg/s$$

Therefore, the rate of evaporation of water is $\boxed{0.0355\ kg/s}$

Question:

6.142P

Cold water enters a steam generator at 20°C and leaves as saturated vapor at 200°C. Determine the fraction of heat used in the steam generator to preheat the liquid water from 20°C to the saturation temperature of 200°C.

Step 1 of 3

Given

Inlet temperature of cold water, $T_i = 20°C$

Exit temperature of water, $T_{ex} = 200°C$

Specific heat of water, $c_p = 4.18\ kJ/kg°C$

Step 2 of 3

The amount of heat required to raise the temperature of unit mass of water

$$q_{heat} = c_p \Delta T$$

$$q_{heat} = (4.18)(200 - 20)$$

Question:

Cold water enters a steam generator at 20°C and leaves as saturated vapor at 200°C. Determine the fraction of heat used in the steam generator to preheat the liquid water from 20°C to the saturation temperature of 200°C.

Step 1 of 3

Given

Inlet temperature of cold water, $T_i = 20°C$

Exit temperature of water, $T_{ex} = 200°C$

Specific heat of water, $c_p = 4.18\ kJ/kg°C$

Step 2 of 3

The amount of heat required to raise the temperature of unit mass of water

Question: 6.143P

Cold water enters a steam generator at 20°C and leaves as saturated vapor at the boiler pressure. At what pressure will the amount of heat needed to preheat the water to saturation temperature be equal to the heat needed to vaporize the liquid at the boiler pressure?

Step 1 of 4

From table properties of steam,

At temperature of cold water $T_1 = 20\ ^\circ\text{C}$ specific enthalpy of saturated liquid is

$$h_{f1} = 83.915\ \text{kJ/kg°K}$$

Given that, amount of heat needed to preheat the water is equal to heat needed to vaporize the liquid

$$q_{preheat} = q_{boiling}$$

$$h_{f2} - h_{f1} = h_{fg2}$$

Here,

Enthalpy of water is h

$$h_{f2} - 83.915 = h_{fg2}$$

$$h_{f2} - h_{fg2} = 83.915$$

Step 2 of 4

The difference between specific heat of saturate liquid and specific heat of vaporization is calculated by trial and error are

at $T = 310\ ^\circ\text{C}$,

$$h_f - h_{fg} = 1402 - 1325.9$$

$$h_f - h_{fg} = 76.1\ \text{kJ/kg°K}$$

at $T = 315\ ^\circ\text{C}$,

$$h_f - h_{fg} = 1431.6 - 1283.4$$

$$h_f - h_{fg} = 148.2\ \text{kJ/kg°K}$$

Step 3 of 4

By taking the difference of specific heats into consideration, the temperature of water at exit is interpolated as

$$\frac{T_2 - 310}{315 - 310} = \frac{83.915 - 76.1}{148.2 - 76.1}$$

$$\frac{T_2 - 310}{5} = \frac{7.815}{72.1}$$

$$\frac{T_2 - 310}{5} = 0.1084$$

$$T_2 = 0.1084 \times 5 + 310$$

$$h_{f2} - h_{fg2} = 83.915$$

Step 2 of 4

The difference between specific heat of saturate liquid and specific heat of vaporization is calculated by trial and error are

at $T = 310\ ^\circ\text{C}$,

$$h_f - h_{fg} = 1402 - 1325.9$$

$$h_f - h_{fg} = 76.1\ \text{kJ/kg°K}$$

at $T = 315\ ^\circ\text{C}$,

$$h_f - h_{fg} = 1431.6 - 1283.4$$

$$h_f - h_{fg} = 148.2\ \text{kJ/kg°K}$$

Step 3 of 4

By taking the difference of specific heats into consideration, the temperature of water at exit is interpolated as

$$\frac{T_2 - 310}{315 - 310} = \frac{83.915 - 76.1}{148.2 - 76.1}$$

$$\frac{T_2 - 310}{5} = \frac{7.815}{72.1}$$

Question:

6.144P

An ideal gas expands in an adiabatic turbine from 1200 K and 900 kPa to 700 K. Determine the turbine inlet volume flow rate of the gas, in m3/s, required to produce turbine work output at the rate of 350 kW. The average values of the specific heats for this gas over the temperature range and the gas constant are cp = 1.13 kJ/kg•K, cv = 0.83 kJ/kg•K, and R = 0.30 kJ/kg•K.

Step 1 of 3

Given

Inlet pressure of water, $P_1 = 900\ \text{kPa}$

Inlet temperature of water, $T_1 = 1200\ \text{K}$

Exit temperature of water, $T_2 = 700\ \text{K}$

Power produced by the turbine $\dot{W}_{out} = 350\ \text{kW}$

Gas constant, $R = 0.3\ \text{kJ/kg.K}$

Specific heat at constant volume, $c_v = 0.83\ \text{kJ/kg.K}$

Specific heat at constant volume, $c_p = 1.13\ \text{kJ/kg.K}$

Step 2 of 3

Energy from the turbine is $\dot{W}_{out} = \dot{m}(h_1 - h_2)$

$$\dot{W}_{out} = \dot{m}c_p(T_1 - T_2)$$

Hence mass flow rate, $\dot{m} = \dfrac{\dot{W}_{out}}{c_p(T_1 - T_2)}$

$$\dot{m} = \frac{350}{1.13 \times (1200 - 700)}$$

$$\dot{m} = 0.6195\ \text{kg/s}$$

Step 3 of 3

But mass flow rate, $\dot{m} = \dfrac{\dot{V}}{v_1}$

Also from ideal gas equation, $P_1 v_1 = RT_1$

Hence from both the relations,

$$\dot{V} = \frac{\dot{m}RT_1}{P_1}$$

$$\dot{V} = \frac{0.6195 \times 0.3 \times 1200}{900}$$

$$\dot{V} = 0.2478\ \text{m}^3/\text{s}$$

Therefore the volume flow rate is $\boxed{\dot{V} = 0.2478\ \text{m}^3/\text{s}}$

Question: 6.145P

Chickens with an average mass of 2.2 kg and average specific heat of 3.54 kJ/kg•°C are to be cooled by chilled water that enters a continuous-flow-type immersion chiller at 0.5°C. Chickens are dropped into the chiller at a uniform temperature of 15 °C at a rate of 500 chickens per hour and are cooled to an average temperature of 3°C before they are taken out. The chiller gains heat from the surroundings at a rate of 200 kJ/h. Determine (a) the rate of heat removal from the chickens, in kW, and (b) the mass flow rate of water, in kg/s, if the temperature rise of water is not to exceed 2°C.

Step 1 of 3

The mass flow rate of chicken is given by

$$\dot{m}_{chicken} = n \times m_{chicken}$$

Here, $m_{chicken}$ is the average mass of each chicken and n is the number of chickens dropped in the chiller

Substitute 500 Chicken/h for $m_{chicken}$ and 2.2 kg/Chicken for n.

$$\begin{aligned}\dot{m}_{chicken} &= 500 \times 2.2 \\ &= 1100 \text{ kg/h} \\ &= \frac{1100}{3600} \text{ kg/s} \\ &= 0.3056 \text{ kg/s}\end{aligned}$$

Rate of heat gain from surroundings is given as

$$\begin{aligned}Q_{surr} &= 200 \text{kJ/h} \\ &= \frac{200}{3600} \text{kJ/h} \\ &= 0.056 \text{ kJ/s}\end{aligned}$$

Step 2 of 3

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$\begin{aligned}E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out} \\ \dot{m}_{chicken} \times h_1 &= Q_{chicken} + \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times h_1 - \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times (h_1 - h_2) \\ Q_{chicken} &= \dot{m}_{chicken} \times c_p \times (T_1 - T_2)\end{aligned}$$

Here, T_1 & T_2 are the initial and final temperatures of Chicken respectively and c_p is the specific heat of Chicken.

Substitute 3°C for T_1, 15°C for T_2 and 3.54 kJ/kg°C for c_p.

$$\begin{aligned}&= \frac{1100}{3600} \text{ kg/s} \\ &= 0.3056 \text{ kg/s}\end{aligned}$$

Rate of heat gain from surroundings is given as

$$\begin{aligned}Q_{surr} &= 200 \text{kJ/h} \\ &= \frac{200}{3600} \text{kJ/h} \\ &= 0.056 \text{ kJ/s}\end{aligned}$$

Step 2 of 3

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$\begin{aligned}E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out} \\ \dot{m}_{chicken} \times h_1 &= Q_{chicken} + \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times h_1 - \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times (h_1 - h_2) \\ Q_{chicken} &= \dot{m}_{chicken} \times c_p \times (T_1 - T_2)\end{aligned}$$

Here, T_1 & T_2 are the initial and final temperatures of Chicken respectively and c_p is the specific heat of Chicken.

Substitute 3°C for T_1, 15°C for T_2 and 3.54 kJ/kg°C for c_p.

$$\begin{aligned}&= \frac{8.36}{} \text{ kg/s} \\ &= 1.559 \text{kg/s}\end{aligned}$$

Therefore, the rate of heat removal from chicken is $\boxed{12.98 \text{ kW}}$ and the mass flow rate of water is to be at least $\boxed{1.559 \text{ kg/s}}$

$$\begin{aligned}&= \frac{8.36}{} \text{ kg/s} \\ &= 1.559 \text{kg/s}\end{aligned}$$

Therefore, the rate of heat removal from chicken is $\boxed{12.98 \text{ kW}}$ and the mass flow rate of water is to be at least $\boxed{1.559 \text{ kg/s}}$

Question: 6.146P

Repeat Prob. 6–145 assuming heat gain of the chiller is negligible.

Step 1 of 3

The mass flow rate of chicken is given by

$$\dot{m}_{chicken} = n \times m_{chicken}$$

Here, $m_{chicken}$ is the average mass of each chicken and n is the number of chickens dropped in the chiller

Substitute 500 Chicken/h for $m_{chicken}$ and 2.2 kg/Chicken for n.

$$\begin{aligned}\dot{m}_{chicken} &= 500 \times 2.2 \\ &= 1100 \text{ kg/h} \\ &= \frac{1100}{3600} \text{ kg/s} \\ &= 0.3056 \text{ kg/s}\end{aligned}$$

Step 2 of 3

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$\begin{aligned}E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out} \\ \dot{m}_{chicken} \times h_1 &= Q_{chicken} + \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times h_1 - \dot{m}_{chicken} \times h_2 \\ Q_{chicken} &= \dot{m}_{chicken} \times (h_1 - h_2) \\ Q_{chicken} &= \dot{m}_{chicken} \times c_p \times (T_1 - T_2)\end{aligned}$$

Here, $T_1 \& T_2$ are the initial and final temperatures of Chicken respectively and c_p is the specific heat of Chicken.

Substitute $3°C$ for T_1, $15°C$ for T_2 and 3.54 kJ/kg°C for c_p.

$$\begin{aligned}Q_{chicken} &= 0.3056 \times 3.54 \times (15 - 3) \\ &= 0.3056 \times 3.54 \times 12 \\ &= 12.98 \text{ kW}\end{aligned}$$

The rate of heat removal by the chicken $\boxed{12.98 \text{ kW}}$

Step 3 of 3

Total heat gained by water in the chiller is given by

$$\begin{aligned}Q_{water} &= Q_{chicken} + Q_{surr} \\ &= 12.98 + 0 \\ &= 12.98 \text{ kW}\end{aligned}$$

Total heat gained by water is also given by

$$Q_{water} = \dot{m}_{water} \times c_p \times \Delta T_{water}$$

Here, the maximum change in temperature of water is ΔT_{water}, the mass flow rate of water is $\dot{m}_{water}$ and the specific heat of water is c_p

From specific heat of water table the given value is $c_p = 4.18$ kJ/kg°C

Substitute 2°C for ΔT_{water} and 12.98 kW for Q_{water}.

The mass flow rate of water is given by

$$\begin{aligned}\dot{m}_{water} &= \frac{Q_{water}}{c_p \times \Delta T_{water}} \\ &= \frac{12.98}{4.18 \times 2} \\ &= \frac{12.98}{8.36} \\ &= 1.553 \text{kg/s}\end{aligned}$$

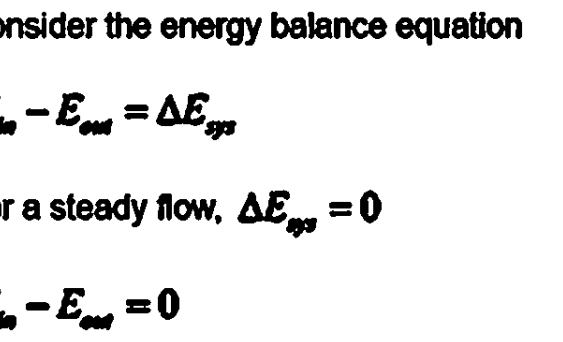
Therefore, the rate of heat removal from chicken is $\boxed{12.98 \text{ kW}}$ and the mass flow rate of water is to be at least $\boxed{1.553 \text{ kg/s}}$

Question: 6.147P

A refrigeration system is being designed to cool eggs (ρ = 67.4 lbm/ft3 and cp = 0.80 Btu/lbm•°F) with an average mass of 0.14 lbm from an initial temperature of 90°F to a final average temperature of 50°F by air at 34°F at a rate of 10,000 eggs per hour. Determine (*a*) the rate of heat removal from the eggs, in Btu/h and (b) the required volume flow rate of air, in ft3/h, if the temperature rise of air is not to exceed 10°F.

Step 1 of 5

Process of air cooling egg is shown below.



Step 2 of 5

Obtain the properties of air from the tables: Molar mass, gas constant, and critical – point properties and Ideal – gas specific heats of various common gases

Specific heat, $c_{p,air} = 0.24$ Btu/lbm°F

Gas constant, $R = 0.3704$ psia $\cdot$ ft^3/lbm $\cdot$ R

Calculate the mass flow rate of egg using the following equation

$$\dot{m}_{egg} = n \times m_{avg}$$

Here, n is the number of eggs cooled per hour and m_{avg} is the average mass of egg

Substitute 10000 eggs/h for n and 0.14 lbm for m_{avg}

$$\begin{aligned}\dot{m}_{egg} &= 10000 \times 0.14 \\ &= 1400 \text{ lbm/h}\end{aligned}$$

Step 3 of 5

Consider the energy balance equation

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady flow, $\Delta E_{sys} = 0$

$$\begin{aligned}E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out}\end{aligned}$$

$$\begin{aligned}\dot{m}_{egg} \times h_1 &= Q_{egg} + \dot{m}_{egg} \times h_2 \\ Q_{egg} &= \dot{m}_{egg} \times h_2 - \dot{m}_{egg} \times h_1 \\ &= \dot{m}_{egg} \times (h_2 - h_1) \\ &= \dot{m}_{egg} \times c_{p,egg} \times (T_1 - T_2)\end{aligned}$$

Here, $c_{p,egg}$ is the specific heat of egg. The initial and final temperatures of eggs are denoted by T_1 and T_2 respectively.

Substitute 0.80 Btu/lbm $\cdot$ °F for $c_{p,egg}$, 90°F for T_1 and 50°F for T_2

$$\begin{aligned}Q_{egg} &= 1400 \times 0.8 \times (90 - 50) \\ &= 1400 \times 0.8 \times 40 \\ &= 44800 \text{ Btu/h}\end{aligned}$$

Therefore the rate heat removal of eggs is **44800 Btu/h**

Step 4 of 5

The heat released from eggs (Q_{egg}) is used to heat the refrigerated air (Q_{air}).

$$\begin{aligned}Q_{air} &= Q_{egg} \\ &= 44800 \text{ Btu/h}\end{aligned}$$

Calculate the mass flow rate of air using the following equation

$$\dot{m}_{air} = \frac{Q_{air}}{c_{p,air} \times \Delta T_{air}}$$

Here, ΔT_{air} is the permissible temperature rise of air

Substitute 10°F for ΔT_{air}

$$\begin{aligned}\dot{m}_{air} &= \frac{44800}{0.24 \times 10} \\ &= \frac{44800}{2.4} \\ &= 18666.67 \text{ lbm/h}\end{aligned}$$

Step 5 of 5

Calculate the density of air using the following equation

$$\rho_{air} = \frac{P}{R \times T_{air}}$$

Here, P is atmospheric pressure and T_{air} is the temperature of the air

Substitute 14.7 lbf/in^2 for P and $(34 + 460)$R for T_{air}

$$\begin{aligned}\rho_{air} &= \frac{14.7}{0.3704 \times (34 + 460)} \\ &= \frac{14.7}{0.3704 \times 494} \\ &= \frac{14.7}{182.9776} \\ &= 0.0803 \text{ lbm/ft}^3\end{aligned}$$

Calculate the volume flow rate of air using the following equation

$$\begin{aligned}\dot{V}_{air} &= \frac{\dot{m}_{air}}{\rho_{air}} \\ &= \frac{18666.67}{0.0803} \\ &= 232461.644 \text{ ft}^3\text{/h}\end{aligned}$$

Therefore, the volumetric flow rate of air is **232461.644 ft^3/h**

Question: 6.148P

A glass bottle washing facility uses a well-agitated hot-water bath at 55°C that is placed on the ground. The bottles enter at a rate of 800 per minute at an ambient temperature of 20°C and leave at the water temperature. Each bottle has a mass of 150 g and removes 0.2 g of water as it leaves the bath wet. Make-up water is supplied at 15°C. Disregarding any heat losses from the outer surfaces of the bath, determine the rate at which (a) water and (b) heat must be supplied to maintain steady operation.

Step 1 of 6

From properties of liquids table, specific heat of water is $c_{p,water} = 4.18\ \text{kJ/kg°C}$ From properties of solids table, specific heat of glass is $c_{p,glass} = 0.8\ \text{kJ/kg°C}$

Step 2 of 6

Mass flow rate of glass bottles through the water bath is given by below equation

$$\dot{m}_{bottle} = m_{bottle} \times Flow_{bottle}$$

Where m_{bottle} is the mass of bottle and $Flow_{bottle}$ is the flow rate of bottle

Substitute $150\,\text{kg}$ for m_{bottle}, $800\,\text{bottles/min}$ for $Flow_{bottle}$

$$\begin{aligned}\dot{m}_{bottle} &= \dot{m}_{bottle} \times Flow_{bottle}\\ &= 150 \times 800\\ &= 120000\ \text{g/min}\\ &= \frac{120000}{1000 \times 60}\ \text{kg/s}\\ &= \frac{120000}{60000}\\ &= 2\ \text{kg/s}\end{aligned}$$

Step 3 of 6

Energy balance of the bottle flow is given by below equation

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady flow, $\Delta E_{sys} = 0$

$$\begin{aligned}&E_{in} - E_{out} = 0\\ &E_{in} = E_{out}\\ &Q_{bottle} + \dot{m}_{bottle} \times h_1 = \dot{m}_{bottle} \times h_2\\ &Q_{bottle} = \dot{m}_{bottle} \times h_2 - \dot{m}_{bottle} \times h_1\\ &\quad = \dot{m}_{bottle} \times (h_2 - h_1)\\ &\quad = \dot{m}_{bottle} \times c_{p,glass} \times (T_2 - T_1)\end{aligned}$$

Where T_1 and T_2 are the ambient and hot water temperatures

Calculate the rate of heat transfer from the bottle

$$\begin{aligned}Q_{bottle} &= 2 \times 0.8 \times (55 - 20)\\ &= 2 \times 0.8 \times 35\\ &= 56\ \text{kW}\end{aligned}$$

Step 4 of 6

Calculate the Amount of water removed by bottle

$$\begin{aligned}\dot{m}_{water,rem} &= Flow_{bottle} \times \text{water removed per bottle}\\ &= 800 \times 0.2\\ &= 160\ \text{g/min}\\ &= \frac{160}{1000 \times 60}\ \text{kg/s}\\ &= \frac{160}{60000}\ \text{kg/s}\\ &= 2.667 \times 10^{-3}\ \text{kg/s}\end{aligned}$$

Therefore the rate at which water must be supplied to maintain steady operation is $\boxed{2.667 \times 10^{-3}\ \text{kg/s}}$

Step 5 of 6

Calculate the rate of heat removal by the water

$$Q_{water,rem} = \dot{m}_{water,rem} \times c_{p,water} \times \Delta T$$

Substitute $2.667 \times 10^{-3}\ \text{kg/s}$ for $\dot{m}_{water,rem}$, $4.18\,\text{kJ/kg} \cdot \text{K}$ for $c_{p,water}$

$$\begin{aligned}Q_{water,rem} &= \dot{m}_{water,rem} \times c_{p,water} \times \Delta T\\ &= 2.667 \times 10^{-3} \times 4.18 \times (55 - 15)\\ &= 2.667 \times 10^{-3} \times 4.18 \times 40\\ &= 0.446\ \text{kW}\end{aligned}$$

Step 6 of 6

Calculate the total amount of heat removed by bottles

$$\begin{aligned}Q_{tot,rem} &= Q_{bottle} + Q_{water,rem}\\ &= 56 + 0.446\\ &= 56.446\ \text{kW}\end{aligned}$$

Therefore, the rate at which heat must be supplied to maintain steady operation is $\boxed{56.446\ \text{kW}}$

Question: 6.149P

Long aluminum wires of diameter 5 mm (ρ = 2702 kg/m3 and cp = 0.896 kJ/kg•°C) are extruded at a temperature of 350°C and are cooled to 50°C in atmospheric air at 25°C. If the wire is extruded at a velocity of 8 m/min, determine the rate of heat transfer from the wire to the extrusion room.

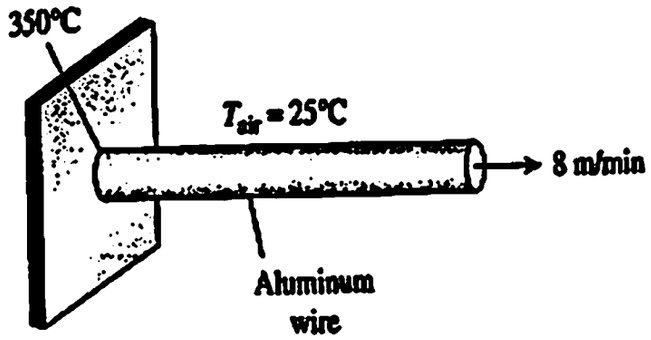


FIGURE P6–149

Step 1 of 3

Given

Diameter of aluminum wire, $d = 5$ mm

Base temperature of wire, $T_1 = 350°C$

Final temperature $T_2 = 50°C$

Velocity of extruded wire, $V = 8$ m/min

The temperature of atmospheric air, $T_{atm} = 25°C$

Density of Aluminum, $\rho = 2702\ kg/m^3$

Specific heat of Aluminum at constant pressure, $c_p = 0.896$ kJ/kg°C

Step 2 of 3

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$

Hence mass flow rate, $\dot{m} = \rho AV$

$$\dot{m} = \rho \times \frac{\pi d^2}{4} \times V$$

$$\dot{m} = 2702 \times \frac{\pi (0.005)^2}{4} \times \left(\frac{8}{60}\right)$$

$$\dot{m} = 0.007\ kg/s$$

Step 3 of 3

The energy balance equation for this process,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

But $h = c_p T$. So,

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

$$\dot{Q}_{out} = 0.007 \times 0.896 \times (350 - 50)$$

$$\dot{Q}_{out} = 1.88\ kW$$

Therefore the rate of heat transfer to the air from wire is $\boxed{\dot{Q}_{out} = 1.88\ kW}$

Question: 6.150P

Repeat Prob. 6–149 for a copper wire (ρ = 8950 kg/m3and cp = 0.383 kJ/kg•°C).

Step 1 of 3

Given

Diameter of copper wire, $d = 5\,\text{mm}$

Base temperature of wire, $T_1 = 350°\text{C}$

Final temperature $T_2 = 50°\text{C}$

Velocity of extruded wire, $V = 8\,\text{m/min}$

The temperature of atmospheric air, $T_{atm} = 25°\text{C}$

Density of copper, $\rho = 8950\,\text{kg/m}^3$

Specific heat of copper at constant pressure, $c_p = 0.383\,\text{kJ/kg°C}$

Step 2 of 3

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$

Hence mass flow rate, $\dot{m} = \rho A V$

$$\dot{m} = \rho \times \frac{\pi d^2}{4} \times V$$

$$\dot{m} = 8950 \times \frac{\pi (0.005)^2}{4} \times \left(\frac{8}{60}\right)$$

$$\dot{m} = 0.0234\,\text{kg/s}$$

Step 3 of 3

The energy balance equation for this process,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

But $h = c_p T$. So,

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

$$\dot{Q}_{out} = 0.0234 \times 0.383 \times (350 - 50)$$

$$\dot{Q}_{out} = 2.688\ \text{kW}$$

Therefore the rate of heat transfer to the air from wire is $\boxed{\dot{Q}_{out} = 2.688\ \text{kW}}$

Question: 6.151P

Steam at 40°C condenses on the outside of a 5-m-long, 3-cm-diameter thin horizontal copper tube by cooling water that enters the tube at 25°C at an average velocity of 2 m/s and leaves at 35°C. Determine the rate of condensation of steam.

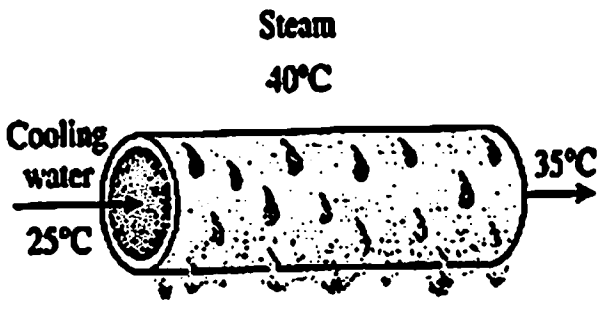
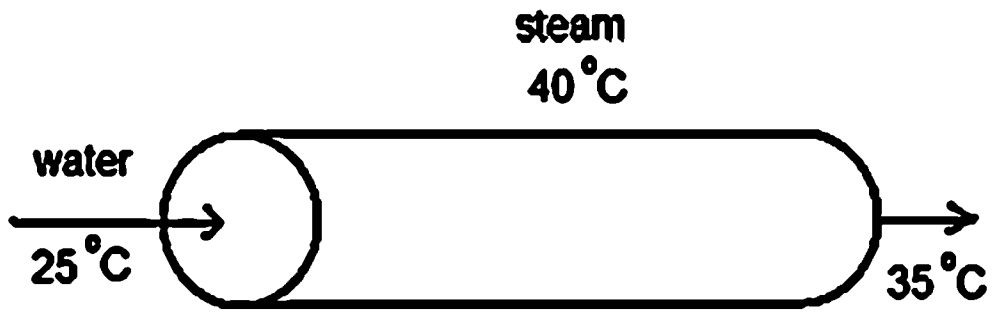


FIGURE P6–151

Step 1 of 4

Schematic Diagram of water flow through a pipe.



Step 2 of 4

The mass flow rate of the extruded wire is given by

$$\dot{m} = \rho \times A \times C$$
$$= \rho \times \frac{\pi \times d^2}{4} \times C$$

Here, A is the area of cross-section of the tube, C is the velocity of water, d is the diameter of cross-section of the tube and ρ is the density of water.

From the properties tables the density of water is $\rho = 997\ \text{kg/m}^3$ and the specific heat $c_p = 4.18\ \text{kJ/kg°C}$

Substitute 0.03 m for d, $997\ \text{kg/m}^3$ for ρ and 2m/min for C.

$$\dot{m} = 997 \times \frac{\pi \times 0.03^2}{4} \times 2$$
$$= 997 \times \frac{2.827 \times 10^{-3}}{4} \times 2$$
$$= 997 \times 7.0675 \times 10^{-4} \times 2$$
$$= 1.409\ \text{kg/s}$$

Step 3 of 4

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$Q_{water} + \dot{m} \times h_1 = \dot{m} \times h_2$$
$$Q_{water} = \dot{m} \times h_2 - \dot{m} \times h_1$$
$$Q_{water} = \dot{m} \times (h_2 - h_1)$$
$$Q_{water} = \dot{m} \times c_p \times (T_2 - T_1)$$

Considering the ideal conditions, heat gained by water is equal to heat lost by steam (heat of condensation) and so

$$Q_{cond} = \dot{m} \times c_p \times (T_2 - T_1)$$

Here, T_1 & T_2 are the initial and final temperatures of copper respectively and c_p is the specific heat of copper.

Substitute 25°C for T_1, 35°C for T_2 and 4.18kJ/kg°C for c_p.

$$Q_{cond} = 1.409 \times 4.18 \times (35 - 25)$$
$$= 1.409 \times 4.18 \times 10$$
$$= 58.896\ \text{kW}$$

Step 4 of 4

Heat of condensation is also given by

$$Q_{cond} = \dot{m}_{cond} \times h_{fg}$$

Here, $\dot{m}_{cond}$ is the mass of steam condensed and h_{fg} is the specific enthalpy of vaporization of steam.

From tables specific enthalpy of vaporization of steam is given as $h_{fg} = 2406\ \text{kJ/kg}$

$$Q_{cond} = \dot{m}_{cond} \times h_{fg}$$
$$58.896 = \dot{m}_{cond} \times 2406$$
$$\dot{m}_{cond} = \frac{58.896}{2406}$$
$$= 0.0245\ \text{kg/s}$$

Therefore, the rate of condensation of steam is $\boxed{0.0245\text{kg/s}}$

Question: 6.152P

Steam at 60 psia and 350°F is mixed with water at 40°F and 60 psia steadily in an adiabatic device. Steam enters the device at a rate of 0.05 lbm/s, while the water enters at 1 lbm/s. Determine the temperature of the mixture leaving this device when the outlet pressure is 60 psia.

Step 1 of 3

Apply the energy balance is

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$m_1 \times h_1 + m_2 \times h_2 = m_3 \times h_3$$

Here, mass and enthalpy of steam is m_1 and h_1. Mass and enthalpy of water is m_2 and h_2.

Mass and enthalpy of the mixture m_3 and h_3

Step 2 of 3

Apply mass balance and find the mass of mixture

$$\begin{aligned} m_3 &= m_1 + m_2 \\ &= 0.05 + 1 \\ &= 1.05 \text{ lbm/s} \end{aligned}$$

From table, properties of water obtain the enthalpies of steam and water

For pressure $P_1 = 60$ psia and entrance temperature $T_1 = 350°F$

Enthalpy $h_1 = 1208.3$ Btu/lbm

For $P_2 = 60$ psia and $T_2 = 40°F$.

Enthalpy $h_2 \cong h_f$
$= 8.032$ Btu/lbm

Step 3 of 3

Find the enthalpy mixture coming out using the below relation

$$\begin{aligned} h_3 &= \frac{m_1 h_1 + m_2 h_2}{m_3} \\ &= \frac{0.05 \times 1208.3 + 1 \times 8.032}{1.05} \\ &= 65.19 \text{ Btu/lbm} \end{aligned}$$

Therefore, the exit pressure is 60 psia and enthalpy is 65.19 Btu/lbm

Using the above pressure and enthalpy determine the final mixture temperature from the table saturated water –temperature table

For pressure $P_3 = 60$ psia and enthalpy $h_3 = 65.19$ Btu/lbm

Final temperature of the mixture is $T_3 = 97.2$ °F
$\cong T_f (h = 65.19$ Btu/lbm$)$

Therefore, temperature of the mixture leaving the device is **97.2 °F**

Question: 6.153P

The average atmospheric pressure in Spokane, Washington (elevation = 2350 ft), is 13.5 psia, and the average winter temperature is 36.5°F. The pressurization test of a 9-ft-high, 4500-ft2 older home revealed that the seasonal average infiltration rate of the house is 2.2 air changes per hour (ACH). That is, the entire air in the house is replaced completely 2.2 times per hour by the outdoor air. It is suggested that the infiltration rate of the house can be reduced by half to 1.1 ACH by winterizing the doors and the windows. If the house is heated by natural gas whose unit cost is $1.24/therm and the heating season can be taken to be six months, determine how much the home owner will save from the heating costs per year by this winterization project. Assume the house is maintained at 72°F at all times and the efficiency of the furnace is 0.92. Also assume the latent heat load during the heating season to be negligible.

Step 1 of 5

Given

Average pressure of atmospheric air, $P_{amb} = 13.5$ psia

Average temperature in winter, $T_{amb} = 36.5°F$

$$T_{amb} = 496.5 \text{ R}$$

Height of the home, $h = 9$ ft

Area of the house, $A = 4500 \text{ ft}^2$

Temperature maintained in the house, $T_{in} = 72°C$

$$T_{in} = 532 \text{ R}$$

Unit cost of natural gas is $ 1.24/therm

Thermal efficiency of the furnace, $\eta = 0.92$

Step 2 of 5

From properties table of air,

Gas constant of air is $R = 0.3704$ psia.ft^3/lbm.R

Specific heat of air is $c = 0.24$ Btu/lbm.°F

The density of outdoor air, from the gas equation $P_0 = \rho_0 R T_0$ psia

$$\rho_0 = \frac{P_0}{RT_0} \text{ lbm/ft}^3$$

$$= \frac{13.5}{(0.3704)(496.5)}$$

$$= 0.0734 \text{ lbm/ft}^3$$

The volume of the house

$$V = Ah$$

$$= (4500)(9)$$

$$= 40500 \text{ ft}^3$$

Step 3 of 5

From mass balance, $\dot{m}_{in} = \dot{m}_{out}$

$$\dot{m} = \rho \dot{V}$$

$$= 1000 \times 20 \times 10^{-3}$$

$$= 20 \text{ kg/s}$$

From energy balance equation,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} = \dot{m}_{out} h_{out} - \dot{m}_{in} h_{in}$$

$$= \dot{m}(h_2 - h_1)$$

$$= \rho \dot{V} c_p (T_2 - T_1)$$

Step 4 of 5

Therefore by substituting the values, we get

$$\dot{Q}_{in} = \rho [V \times ACH] c_p (T_2 - T_1)$$

$$= 0.0734 \times \left[40500(2.2 - 1.1)\right](0.24)(532 - 496.5)$$

$$= 27{,}860 \text{ Btu/h } \frac{1 \text{ therm}}{100{,}000 \text{ Btu}}$$

$$= 0.2786 \text{ therm/h}$$

Step 5 of 5

Energy saved is given by,

$$\dot{E}_{saved} = \frac{\dot{Q}_{in}}{\eta_{furnace}}$$

$$= \frac{0.2789}{0.92}$$

$$= 0.3028 \text{ therm/h}$$

Considering the heating season as 6 months, and the unit cost for natural gas as $ 1.24/therm,

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy})(\text{time period})$$

$$= \left(0.3028 \frac{\text{therm}}{\text{h}}\right)(\$1.24 \text{ /therm})(6 \times 30 \times 24)$$

$$= \$ 1622 \text{ per year}$$

Hence the total money saved per year is $\boxed{\$ 1622}$

Question: 6.154P

Determine the rate of sensible heat loss from a building due to infiltration if the outdoor air at −5°C and 90 kPa enters the building at a rate of 35 L/s when the indoors is maintained at 20°C.

Step 1 of 2

From properties table of air,

Gas constant of air is $R = 0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}$

Specific heat of air is $c_p = 1.005\ \text{kJ/kg}\cdot{}^\circ\text{C}$

Find density of outdoor air, from the gas equation

$$P_o = \rho_0 R T_0 \text{ psia}$$

$$\rho_0 = \frac{P_0}{RT_0} \text{ lbm/ft}^3$$

Here, Average pressure of atmospheric air, P_0 , Average temperature in winter, T_0

Substitute $P_0 = 90\ \text{kPa}$, $T_0 = 268\ \text{K}$ and $R = 0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}$ in the above equation

$$\begin{aligned}\rho_0 &= \frac{P_0}{RT_0} \text{ lbm/ft}^3 \\ &= \frac{90\ \text{kPa}}{(0.287)(268\ \text{K})} \\ &= 1.17\ \text{kg/m}^3\end{aligned}$$

Step 2 of 2

Apply the energy balance equation,

$$E_{in} = E_{out}$$

$$\begin{aligned}Q_{in} &= m_{out}h_{out} - m_{in}h_{in} \\ &= m(h_2 - h_1) \\ &= mc_p(T_2 - T_1)\end{aligned}$$

Find reduction in the sensible infiltration heat lost due to infiltration:

$$\begin{aligned}Q_{in} &= \rho_0 V_{air} c_p (T_2 - T_1) \\ &= 1.17\times[0.035](1.005)(20-(-5)) \\ &= 1.029\ \text{kW}\end{aligned}$$

Therefore, sensible heat will be lost due to infiltration is $\boxed{1.029\ \text{kJ/s}}$.

Question: 6.155P

Consider a large classroom on a hot summer day with 150 students, each dissipating 60 W of sensible heat. All the lights, with 6.0 kW of rated power, are kept on. The room has no external walls, and thus heat gain through the walls and the roof is negligible. Chilled air is available at 15°C, and the temperature of the return air is not to exceed 25°C. Determine the required flow rate of air, in kg/s, that needs to be supplied to the room to keep the average temperature of the room constant.

Step 1 of 4

The rate of heat generation by the people in the classroom

$$Q_{gen} = q_{gen} \times n$$

Here, q_{gen} is the heat generated by each student and n is the number of students in the classroom.

Substitute 60 W/persons for q_{gen} and 150 for n.

$$\begin{aligned} Q_{gen} &= 60 \times 150 \\ &= 9000 \text{ W} \\ &= 9 \text{ kW} \end{aligned}$$

Step 2 of 4

The total heat generated in the classroom is given by

$$Q_{tot} = Q_{gen} + Q_{light}$$

Here, Q_{gen} is the rate of heat generation by the people in the classroom and Q_{light} is the rate of heat generation by the light.

Substitute 9 kW for Q_{gen} and 6 kW for Q_{light}.

$$\begin{aligned} Q_{tot} &= Q_{gen} + Q_{light} \\ Q_{tot} &= 9 + 6 \\ Q_{tot} &= 15 \text{ kW} \end{aligned}$$

Step 3 of 4

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$\begin{aligned} E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out} \\ Q_{in} + \dot{m} \times h_1 &= \dot{m} \times h_2 \\ Q_{in} &= \dot{m} \times h_2 - \dot{m} \times h_1 \\ Q_{in} &= \dot{m} \times (h_2 - h_1) \\ Q_{in} &= \dot{m} \times c_p \times (T_2 - T_1) \end{aligned}$$

Heat input to classroom is from students and lighting and so the heat input is same as total heat generated

$$Q_{out} = \dot{m} \times c_p \times (T_2 - T_1)$$

Step 4 of 4

Mass flow rate of air is given by

$$\dot{m} = \frac{Q_{tot}}{c_p \times (T_2 - T_1)}$$

Here, T_1 & T_2 are the initial and final temperatures of air respectively and c_p is the specific heat ratio.

From specific heat tables at air is $c_p = 1.005$ kJ/kg°C

Substitute 15°C for T_1 and 25°C for T_2.

$$\begin{aligned} \dot{m} &= \frac{Q_{tot}}{c_p \times (T_2 - T_1)} \\ &= \frac{15}{1.005 \times (25 - 15)} \\ &= \frac{15}{1.005 \times 10} \\ &= \frac{15}{10.05} \\ &= 1.492 \text{ kg/s} \end{aligned}$$

Therefore, the mass flow rate of air is $\boxed{1.492 \text{kg/s}}$

Question: 6.156P

A fan is powered by a 0.5-hp motor and delivers air at a rate of 85 m3/min. Determine the highest value for the average velocity of air mobilized by the fan. Take the density of air to be 1.18 kg/m3.

Step 1 of 3

Since mass crosses the system boundary during the process, consider the fan-motor assembly is control volume

Since there is only one inlet and one exit, and thus

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

When all of the entire electrical energy drawn by the motor is converted to kinetic energy and the friction between the air layers is zero then the velocity of air leaving the fan will be high. In this best possible case, no energy will be converted to thermal energy, and thus there is no change in temperature change

$T_1 = T_2$ Since temperature remains constant, enthalpy remains constant $h_1 = h_2$

Step 2 of 3

Find mass flow rate of air is

$$\begin{aligned}\dot{m} &= \rho \dot{V} \\ &= 1.18(\text{kg/m}^3) \times 85(\text{m}^3/\text{min}) \\ &= 100.3\,\text{kg/min} \\ &= 1.67\,\text{kg/s}\end{aligned}$$

Apply energy balance for this steady-flow system:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} = 0$$

$$\dot{E}_{in} - \dot{E}_{out} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}_1 h_1 = \dot{m}_2 \left(h_2 + \frac{V_2^2}{2} \right)$$

$$\dot{W}_{e,in} + \dot{m} h_1 = \dot{m} \left(h_2 + \frac{V_2^2}{2} \right)$$

$$\dot{W}_{e,in} + \dot{m} h_1 = \dot{m} \left(h_1 + \frac{V_2^2}{2} \right)$$

$$\dot{W}_{e,in} = \dot{m} \frac{V_2^2}{2}$$

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}}$$

Step 3 of 3

Substitute the known values in above equation

$$\begin{aligned}V_2 &= \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} \\ &= \sqrt{\frac{2 \times 0.5\,\text{hp}}{1.67\,\text{kg/s}} \times \left(\frac{745.7\,\text{W}}{1\,\text{hp}} \right) \times \left(\frac{1\,\text{kg-m}^2/\text{s}^3}{\text{W}} \right)} \\ &= 21.13\,\text{m/s}\end{aligned}$$

Therefore the average velocity of air is $\boxed{V_2 = 21.13\,\text{m/s}}$

Question: 6.157P

An air-conditioning system requires airflow at the main supply duct at a rate of 130 m3/min. The average velocity of air in the circular duct is not to exceed 8 m/s to avoid excessive vibration and pressure drops. Assuming the fan converts 80 percent of the electrical energy it consumes into kinetic energy of air, determine the size of the electric motor needed to drive the fan and the diameter of the main duct. Take the density of air to be 1.20 kg/m3.

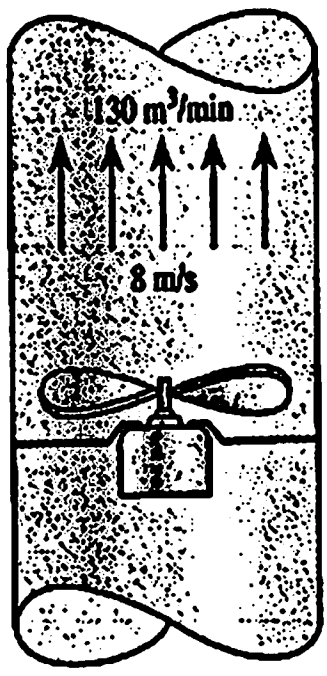


FIGURE P6–157

Step 1 of 3

Given

Volume flow rate of air, $\dot{V} = 130\ \text{m}^3/\text{min}$

The average velocity of air $V_2 = 8\ \text{m/s}$

The efficiency of fan is 80%

From properties table, Gas constant $R = 0.287\ \text{kJ/kg.K}$

Specific heat at constant pressure, $c_p = 1.005\ \text{kJ/kg.K}$

density of air, $\rho = 1.2\ \text{kg/m}^3$

Step 2 of 3

The mass flow rate is expressed as

$$\dot{m} = \rho\dot{V}$$
$$= (1.2)(130/60)$$
$$= 2.6\ \text{kg/s}$$

The energy balance equation for a steady state process is

$$\dot{W} = \dot{Q} - \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} + g(z_1 - z_2)\right)$$

Neglecting potential energy change, enthalpy change and heat losses,

$$\eta\dot{W} = \dot{Q} - \dot{m}\left(h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} + g(z_1 - z_2)\right)$$

$$\eta\dot{W} = 0 - 2.6\left(\frac{0^2 - 8^2}{2} \times \frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$(0.8)\dot{W} = -(0.0832)\ \text{kW}$$

Thus $\dot{W}_{\text{motor}} = 0.104\ \text{kW}$

The electric power consumed by the motor is $\boxed{\dot{W}_{\text{motor}} = 0.104\ \text{kW}}$

Step 3 of 3

From volumetric flow relation,

$$\dot{V} = VA$$

$$\dot{V} = V \times \frac{\pi D^2}{4}$$

Thus the diameter of the main duct is

$$D = \sqrt{\frac{4\dot{V}}{\pi V}}$$
$$= \sqrt{\frac{4(130/60)}{\pi(8)}}$$
$$= \sqrt{0.3448}$$
$$= 0.587\ \text{m}$$

Hence the duct diameter is $\boxed{D = 0.587\ \text{m}}$

Question: 6.158P

An adiabatic air compressor is to be powered by a direct-coupled adiabatic steam turbine that is also driving a generator. Steam enters the turbine at 12.5 MPa and 500°C at rate of 25 kg/s and exits at 10 kPa and a quality of 0.92. Air enters the compressor at 98 kPa and 295 K at a rate of 10 kg/s and exits at 1 MPa and 620 K. Determine the net power delivered to the generator by the turbine.

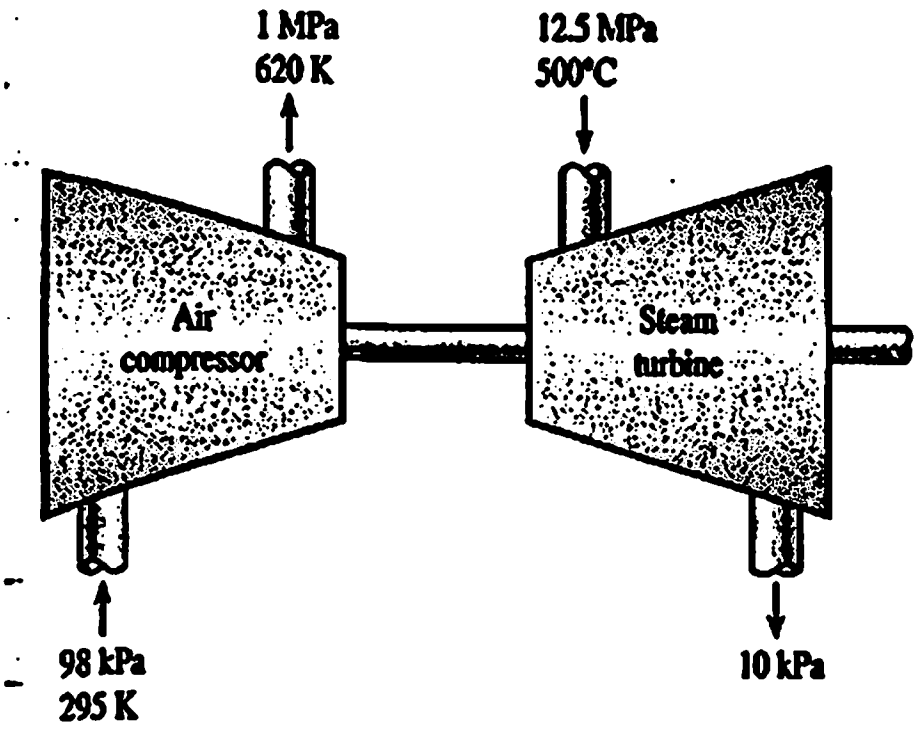


FIGURE P6–158

Step 1 of 4

Take the properties at turbine inlet pressure $P_1 = 12.5$ MPa and turbine inlet temperature $T_1 = 500$ °C from superheated water tables,

Specific enthalpy of steam $h_1 = 3343.6$ kJ/kg

Take the properties of water from saturated water tables at turbine exit pressure $P_2 = 10$ kPa

Specific enthalpy of saturated liquid $h_{f2} = 191.81$kJ/kg

Specific enthalpy of evaporation $h_{fg2} = 2392.1$kJ/kg

Find specific enthalpy of steam by using the equation,

$$h_2 = h_{f2} + x_2 \times h_{fg2}$$

Here, turbine exit quality $x_2 = 0.92$

$$h_2 = 191.81 + 0.92 \times 2392.1$$
$$= 191.81 + 2200.732$$
$$= 2392.542 \text{ kJ/kg}$$

Step 2 of 4

From ideal gas properties of air table at compressor inlet temperature $T_3 = 295$ K specific enthalpy of air is

$$h_3 = 295.17 \text{ kJ/kg}$$

Similarly at compressor exit temperature $T_4 = 620$ K specific enthalpy of air is

$$h_4 = 628.07 \text{ kJ/kg}$$

As there is only one entry and one exit for turbine, mass flow rate will be same

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{steam}$$
$$\dot{m}_{steam} = 25 \text{ kg/s}$$

As there is only one entry and one exit for compressor, mass flow rate will be same

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{air}$$
$$\dot{m}_{air} = 10 \text{ kg/s}$$

Step 3 of 4

Find the work done by turbine by using the equation,

$$\dot{W}_{tur} = \dot{m}_{steam} \times (h_1 - h_2)$$
$$= 25 \times (3343.6 - 2392.542)$$
$$= 25 \times 951.058$$
$$= 23776.45 \text{ kW}$$

Find the work required to run the compressor by using the equation,

$$\dot{W}_{comp} = \dot{m}_{air} \times (h_4 - h_3)$$
$$= 10 \times (628.07 - 295.17)$$
$$= 10 \times 332.9$$
$$= 3329 \text{ kW}$$

Step 4 of 4

Since work done by turbine is used for running compressor and generator

$$\dot{W}_{tur} = \dot{W}_{comp} + \dot{W}_{gen}$$
$$\dot{W}_{gen} = \dot{W}_{tur} - \dot{W}_{comp}$$
$$= 23776.45 - 3329$$
$$= 20447.45 \text{ kW}$$

Therefore, the net power delivered to generator is **20447.45 kW**

Question: 6.159P

Determine the power input for a compressor that compresses helium from 150 kPa and 20°C to 400 kPa and 200°C. Helium enters this compressor through a 0.1-m2 pipe at a velocity of 15 m/s.

Step 1 of 4

From ideal gas properties take the properties of helium

Specific heat at constant pressure $c_p = 5.1926\ \text{kJ/kg}\cdot\text{K}$

Gas constant of Helium $R = 2.0769\ \text{kJ/kg}\cdot\text{K}$

Step 2 of 4

As there is only one entry and one exit for compressor, mass flow rate will be same

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

Calculate the specific volume of helium at inlet of compressor by using the equation,

$$v_1 = \frac{R \times T_1}{P_1}$$

Here T_1 and P_1 are inlet temperature and pressure respectively

Substitute $20°\text{C} = 293\text{K}$ for T_1 and 150 kPa for P_1

$$\begin{aligned} v_1 &= \frac{2.0769 \times (20+273)}{150} \\ &= \frac{2.0769 \times 293}{150} \\ &= \frac{608.5317}{150} \\ &= 4.0569\ \text{m}^3/\text{kg} \end{aligned}$$

Step 3 of 4

Find the mass flow rate of helium by using the expression

$$\dot{m} = \frac{A_1 \times V_1}{v_1}$$

Here A_1 is the area of cross-section and V_1 is the inlet velocity of compressor.

Substitute $15\ \text{m/s}$ for V_1

$$\begin{aligned} \dot{m} &= \frac{0.1 \times 15}{4.0569} \\ &= \frac{1.5}{4.0569} \\ &= 0.3697\ \text{kg/s} \end{aligned}$$

Step 4 of 4

Apply the energy balance equation to the steady flow system,

$$E_{in} - E_{out} = \Delta E_{sys}$$

For steady process, $\Delta E_{sys} = 0$ and so

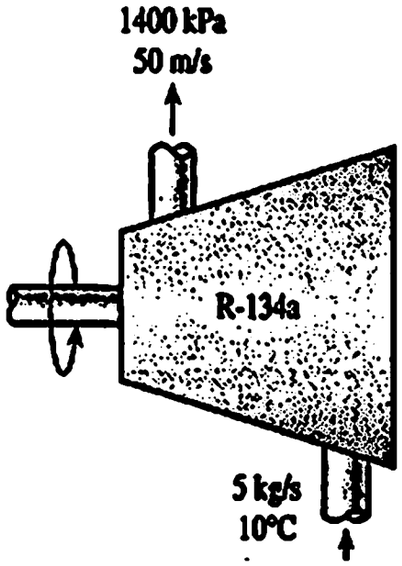
$$\begin{aligned} E_{in} - E_{out} &= 0 \\ E_{in} &= E_{out} \\ W_{in} + \dot{m} \times h_1 &= \dot{m} \times h_2 \\ W_{in} &= \dot{m} \times h_2 - \dot{m} \times h_1 \\ W_{in} &= \dot{m} \times (h_2 - h_1) \\ &= \dot{m} \times c_p \times (T_2 - T_1) \\ &= 0.3697 \times 5.1926 \times (200 - 20) \\ &= 0.3697 \times 5.1926 \times 180 \\ &= 345.55\ \text{kW} \end{aligned}$$

Therefore, the power required to run the compressor is $\boxed{345.55\ \text{kW}}$

Question:

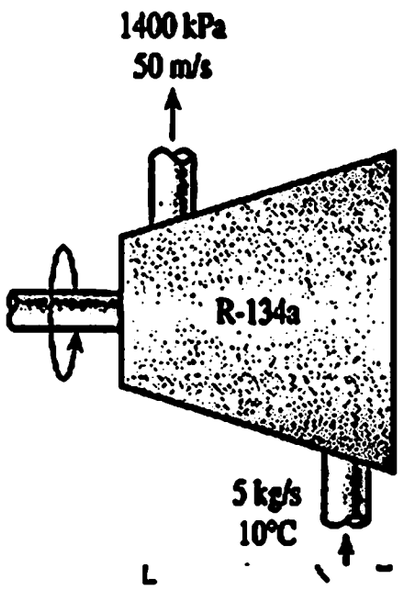
6.160P

Refrigerant 134a enters a compressor with a mass flow rate of 5 kg/s and a negligible velocity. The refrigerant enters the compressor as a saturated vapor at 10°C and leaves the compressor at 1400 kPa with an enthalpy of 281.39 kJ/kg and a velocity of 50 m/s. The rate of work done on the refrigerant is measured to be 132.4 kW. If the elevation change between the compressor inlet and exit is negligible, determine the rate of heat transfer associated with this process, in kW.



Question:

Refrigerant 134a enters a compressor with a mass flow rate of 5 kg/s and a negligible velocity. The refrigerant enters the compressor as a saturated vapor at 10°C and leaves the compressor at 1400 kPa with an enthalpy of 281.39 kJ/kg and a velocity of 50 m/s. The rate of work done on the refrigerant is measured to be 132.4 kW. If the elevation change between the compressor inlet and exit is negligible, determine the rate of heat transfer associated with this process, in kW.



$$= 5\left[(256.16 - 281.39) - \left(\frac{50^2}{2000}\right)\right] + (132.4)$$

$$= 0$$

The heat transfer in the process is $\boxed{\dot{Q}_{out} = 0}$

Therefore it is an adiabatic process.

Question:

6.161P

Submarines change their depth by adding or removing air from rigid ballast tanks, thereby displacing seawater in the tanks. Consider a submarine that has a 700 m3 air-ballast tank originally partially filled with 100 m3 of air at 1500 kPa and 15°C. For the submarine to surface, air at 1500 kPa and 20°C is pumped into the ballast tank, until it is entirely filled with air. The tank is filled so quickly that the process is adiabatic and the seawater leaves the tank at 15°C. Determine the final temperature and mass of the air in the ballast tank.

Step 1 of 4

Obtain properties of air from properties of air table.

Gas constant of air is $R = 0.287\ \text{kPa} \cdot \text{m}^3/\text{kg}$

Specific heat of air is $c_p = 1.005\ \text{kJ/kg} \cdot °\text{C}$

Apply the conservation of mass principle to the air.

$$\frac{dm_a}{dt} = \dot{m}_{in} - \dot{m}_{out}$$

Question:

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Step 1 of 4

Obtain properties of air from properties of air table.

Gas constant of air is $R = 0.287\ \text{kPa} \cdot \text{m}^3/\text{kg}$

Specific heat of air is $c_p = 1.005\ \text{kJ/kg} \cdot °\text{C}$

Apply the conservation of mass principle to the air.

$$\frac{dm_a}{dt} = \dot{m}_{in} - \dot{m}_{out}$$

Step 2 of 4

Apply steady flow energy equation to the system

$$0 = \frac{d(mu)_a}{dt} + \frac{d(mu)_w}{dt} + h_w \dot{m}_{out} - h_a \dot{m}_{in} \quad \cdots\cdots (3)$$

Here, enthalpies of mass water and air are h_w and h_a respectively and u_a and u_w are specific internal energies of air and water respectively.

Substitute equation (1) and (2) in equation (3).

$$0 = \frac{d(mu)_a}{dt} + \frac{d(mu)_w}{dt} - h_w \frac{dm_w}{dt} - h_a \frac{dm_a}{dt}$$

$$d(mu)_a + d(mu)_w = h_w dm_w + h_a dm_a$$

Integrate the equation.

$$\int \left(d(mu)_a + d(mu)_w \right) = h_w \int dm_w + h_a \int dm_a$$

$$\left[(mu)_2 - (mu)_1 \right]_a + \left[(mu)_2 - (mu)_1 \right]_w = h_w (m_2 - m_1)_w + h_a (m_2 - m_1)_a$$

Here, mass of water at state 2 is zero, $m_{2,w} = 0$

Step 3 of 4

Substitute mass of air by using ideal gas equation as $\frac{Pv}{RT}$ for m_a and $c_v T$ for u_a and $c_p T$ for h_a

$$\frac{Pv_2}{RT_2} c_v T_2 - \frac{Pv_1}{RT_1} c_v T_1 + \left(0 - m_{w,1} c_w T_w \right) = \left(0 - m_{w1} c_w T_w \right) + c_p T_{in} \left(\frac{Pv_2}{RT_2} - \frac{Pv_1}{RT_1} \right)$$

$$\frac{Pv_2}{RT_2} c_v T_2 - \frac{Pv_1}{RT_1} c_v T_1 = c_p T_{in} \left(\frac{Pv_2}{RT_2} - \frac{Pv_1}{RT_1} \right)$$

$$c_v \left(\frac{v_2}{T_2} T_2 - \frac{v_1}{T_1} T_1 \right) = c_p T_{in} \left(\frac{v_2}{T_2} - \frac{v_1}{T_1} \right)$$

$$\frac{c_v}{c_p} (v_2 - v_1) = T_{in} \left(\frac{v_2}{T_2} - \frac{v_1}{T_1} \right)$$

$$\frac{1}{kT_{in}} (v_2 - v_1) + \frac{v_1}{T_1} = \frac{v_2}{T_2}$$

$$T_2 = \frac{v_2}{\frac{1}{kT_{in}} (v_2 - v_1) + \frac{v_1}{T_1}} \quad \cdots\cdots (4)$$

Here, k is specific heat ratio and T_{in} is temperature of air entering into the ballast tank

Step 4 of 4

Substitute $700\,\text{m}^3$ for v_2 , $100\,\text{m}^3$ for v_1 , $(15°\text{C} = 288\,\text{K})$ for T_1 , $(20°\text{C} = 293\,\text{K})$ for T_{in} and 1.4 for k in equation (4)

$$T_2 = \frac{v_2}{\frac{v_1}{T_1} + \frac{1}{kT_{in}} (v_2 - v_1)}$$

$$= \frac{700}{\frac{100}{288} + \frac{1}{1.4 \times 293} (700 - 100)}$$

$$= 386.75\ \text{K}$$

The final temperature of the air in the ballast tank is $\boxed{386.75\ \text{K}}$.

Find the final mass of air present in the ballast tank

$$m_{2a} = \frac{Pv_2}{RT_2}$$

Here, P is final pressure of air in the tank and v_2 is final volume of air in the ballast tank.

Substitute 1500 kPa for P, $700\,\text{m}^3$ for v_2 and 386.75 K for T_2

$$m_{2a} = \frac{1500 \times 700}{(0.287)(386.75)}$$

$$= 9{,}459.693\ \text{kg}$$

Therefore, the final mass of air present in the ballast tank is $\boxed{9{,}459.693\ \text{kg}}$

k in equation (4)

Question: 6.162P

In the preceding problem, presume that air is added to the tank in such a way that the temperature and pressure of the air in the tank remain constant. Determine the final mass of the air in the ballast tank under this condition. Also determine the total heat transfer while the tank is being filled in this manner.

Step 1 of 2

Find the initial mass of the air,

$$m_1 = \frac{P_1 V_1}{R T_1}$$

Here,

The pressure, temperature and volume at state-1 is P_1, T_1 and V_1 respectively

Substitute $1500\,\text{kPa}$ for P_1, $100\,\text{m}^3$ for V_1, $0.287\,\text{kJ/kg}\cdot\text{K}$ and $288.15\,\text{K}$ for T_1,

$$m_1 = \frac{P_1 V_1}{R T_1}$$

Question:

In the preceding problem, presume that air is added to the tank in such a way that the temperature and pressure of the air in the tank remain constant. Determine the final mass of the air in the ballast tank under this condition. Also determine the total heat transfer while the tank is being filled in this manner.

Step 1 of 2

Find the initial mass of the air,

$$m_1 = \frac{P_1 V_1}{R T_1}$$

Here,

The pressure, temperature and volume at state-1 is P_1, T_1 and V_1 respectively

Substitute $1500\,\text{kPa}$ for P_1, $100\,\text{m}^3$ for V_1, $0.287\,\text{kJ/kg}\cdot\text{K}$ and $288.15\,\text{K}$ for T_1,

$$\begin{aligned} m_1 &= \frac{P_1 V_1}{R T_1} \\ &= \frac{1500\times 100}{0.287\times 288.15} \\ &= 1814\,\text{kg} \end{aligned}$$

Find the initial mass of the water,

$$m_w = \frac{V_w}{v_1}$$

Here,

The initial volume of the water is V_w

The initial specific volume of the water is v_1

Substitute $600\,\text{m}^3$ for V_w and $0.001\,\text{m}^3/\text{kg}$ for v_1,

$$\begin{aligned} m_w &= \frac{V_w}{v_1} \\ &= \frac{600}{0.001} \\ &= 600000\,\text{kg} \end{aligned}$$

Step 2 of 2

Find the final mass of the air,

$$m_2 = \frac{P_2 V_2}{R T_2}$$

Here,

The pressure, temperature and volume at state-2 is P_2, T_2 and V_2 respectively

Substitute $1500\,\text{kPa}$ for P_2, $700\,\text{m}^3$ for V_2, $0.287\,\text{kJ/kg}\cdot\text{K}$ and $288.15\,\text{K}$ for T_1,

$$m_2 = \frac{P_2 V_2}{R T_2}$$

$$1500 \times 700$$

$600000\,\text{kg}$ for m_w and $62.98\,\text{kJ/kg}$ for h_w,

$$\begin{aligned} Q_{in} &= m_2 c_v T - (m_1 c_v T + m_w h_w) + m_w h_w - (m_2 - m_1) c_p T \\ &= (12696\times 0.718\times 288.15) - (1814\times 0.718\times 288.15 + 600000\times 62.98) \\ &\quad + (600000\times 62.98) - (12696\times 1.005\times 288.15) \\ &= 0\,\text{kJ} \end{aligned}$$

Therefore, the process is adiabatic.

Question: 6.163P

A steel cylinder initially contains nitrogen gas at 200 kPa and 25°C. The cylinder is connected to a supply line containing nitrogen at 800 kPa and 25 °C. A valve is opened, allowing nitrogen to flow into the cylinder until the pressure reaches 800 kPa. The cylinder internal volume is 0.1 m3, its mass (cylinder only) is 50 kg, and its specific heat is 0.43 kJ/kg•K. Find the final mass of nitrogen in the tank and the final temperature of the nitrogen assuming constant specific heats at room temperature for nitrogen, and assuming (a) no heat transfer from the nitrogen to the tank, and (b) rapid heat transfer between the nitrogen and tank such that the cylinder and nitrogen remain in thermal equilibrium during the process (with negligible heat transfer from the cylinder to its external surroundings.)

Step 1 of 5

Calculate the Final mass by using the following equation

$$m_2 = \frac{P_2 \times V}{T_2 \times R}$$

Here, Final pressure of the air P_2 Final temperature of the air T_2

Question:

A steel cylinder initially contains nitrogen gas at 200 kPa and 25°C. The cylinder is connected to a supply line containing nitrogen at 800 kPa and 25 °C. A valve is opened, allowing nitrogen to flow into the cylinder until the pressure reaches 800 kPa. The cylinder internal volume is 0.1 m3, its mass (cylinder only) is 50 kg, and its specific heat is 0.43 kJ/kg•K. Find the final mass of nitrogen in the tank and the final temperature of the nitrogen assuming constant specific heats at room temperature for nitrogen, and assuming (a) no heat transfer from the nitrogen to the tank, and (b) rapid heat transfer between the nitrogen and tank such that the cylinder and nitrogen remain in thermal equilibrium during the process (with negligible heat transfer from the cylinder to its external surroundings.)

Step 1 of 5

Calculate the Final mass by using the following equation

$$m_2 = \frac{P_2 \times V}{T_2 \times R}$$

Here, Final pressure of the air P_2 Final temperature of the air T_2

Substitute, $800\ \text{kPa}$ for P_2, $0.1\ \text{m}^3$ for V and $0.287\ \text{kJ/kg}\cdot\text{K}$ for R

$$m_2 = \frac{800 \times 0.1}{T_2 \times 0.287} = \frac{278.7}{T_2}$$

Step 2 of 5

Calculate the mass of air by using the mass balance

$$m_{in} - m_{out} = \Delta m_{sys}$$

$$m_s - 0 = m_2 - m_1$$

Here, Initial mass in the tank is m_1, Final mass in the tank is m_2, mass entered in the tank is m_s

Substitute, $11.946\ \text{kg}$ for m_2 and $2.362\ \text{kg}$ for m_1

$$m_s = m_2 - m_1$$

Calculate heat transfer by using the following energy balance

$$E_{in} - E_{out} = E_{sys}$$

For a steady process, $E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$m_s h_s = m_2 u_2 - m_1 u_1$$

$$m_s h_s = m_2 c_v T_2 - m_1 c_v T_1$$

Step 3 of 5

(a) No heat transfer from nitrogen to the tank

From the mass balance equation

$$(m_2 - m_1) c_p T_s = m_2 c_v T_2 - m_1 c_v T_1$$

Rearrange the above equation to find the final temperature

$$T_2 = \frac{(m_2 - m_1) c_p T_s + m_1 c_v T_1}{m_2 c_v}$$

Substitute the corresponding values in the above equation

$$T_2 = \frac{\left(\frac{278.7}{T_2} - 0.2261\right)(1.039 \times 298) + (0.2261 \times 0.743 \times 298)}{\frac{278.7}{T_2} \times 0.743}$$

Step 4 of 5

Solve the above equation using Equation solver and find the values of final temperature

$$T_2 = 380.1\ \text{K} = 107.1°\text{C}$$

Therefore, the final temperature of the system $\boxed{107.1°\text{C}}$

Find the final mass of the system as below

$$m_2 = \frac{800 \times 0.1}{T_2 \times 0.287} = \frac{278.7}{380.1} = 0.733\ \text{kg}$$

Therefore, the final mass of the system is $\boxed{0.733\ \text{kg}}$

(b) Rapid heat transfer between nitrogen and tank

Apply the energy balance to the system

$$E_{in} - E_{out} = E_{sys}$$

$$(m_2 - m_1) c_p T_s = (m_{air,2} c_v T_2 + m_{cy} c_{cy} T_2) - (m_{air,1} c_v T_1 + m_{cy} c_{cy} T_1)$$

Therefore, the final mass of the system is $\boxed{0.9226\ \text{kg}}$

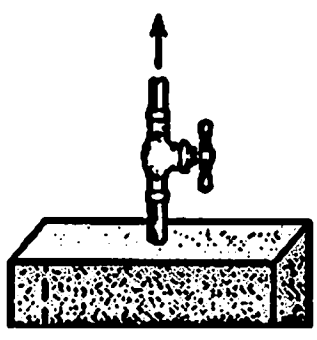
Question: 6.164P pg 1

A tank with an internal volume of 1 m3 contains air at 800 kPa and 25°C. A valve on the tank is opened allowing air to escape and the pressure inside quickly drops to 150 kPa, at which point the valve is closed. Assume there is negligible heat transfer from the tank to the air left in the tank.

(*a*) Using the approximation he ≈ constant = he,avg = 0.5(*h*1 + h2), calculate the mass withdrawn during the process.

(*b*) Consider the same process but broken into two parts. That is, consider an intermediate state at P2 = 400 kPa, calculate the mass removed during the process from *P*1, = 800 kPa to P2 and then the mass removed during the process from P2 to *P*3 = 150 kPa, using the type of approximation used in part (a), and add the two to get the total mass removed.

(*c*) Calculate the mass removed if the variation of he is accounted for.



Question:

A tank with an internal volume of 1 m3 contains air at 800 kPa and 25°C. A valve on the tank is opened allowing air to escape and the pressure inside quickly drops to 150 kPa, at which point the valve is closed. Assume there is negligible heat transfer from the tank to the air left in the tank.

(*a*) Using the approximation he ≈ constant = he,avg = 0.5(*h*1 + h2), calculate the mass withdrawn during the process.

(*b*) Consider the same process but broken into two parts. That is, consider an intermediate state at P2 = 400 kPa, calculate the mass removed during the process from *P*1, = 800 kPa to P2 and then the mass removed during the process from P2 to *P*3 = 150 kPa, using the type of approximation used in part (a), and add the two to get the total mass removed.

(*c*) Calculate the mass removed if the variation of he is accounted for.

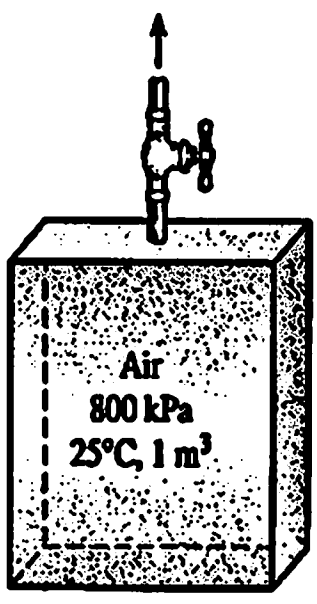


FIGURE P6–164

Step 1 of 12

(a)

Calculate the initial mass in the tank using the ideal gas equation:

$$p_1V_1 = m_1RT_1$$

$$m_1 = \frac{p_1V_1}{RT_1}$$

Here, p is the pressure, T is the temperature, R is the gas constant, V is the volume, and m is the mass.

Substitute $800(10^3)$ Pa for p_1, $1\ \text{m}^3$ for V_1, $(25+273)$ K for T_1, and $287\ \text{J/kg}\cdot\text{K}$ for R.

$$m_1 = \frac{[800(10^3)](1)}{(287)(25+273)}$$
$$= 9.35\ \text{kg}$$

Step 2 of 12

Calculate the final mass in the tank using the ideal gas equation:

$$p_2V_2 = m_2RT_2$$

$$m_2 = \frac{p_2V_2}{RT_2}$$

Substitute $150(10^3)$ Pa for p_2, $1\ \text{m}^3$ for V_2, and $287\ \text{J/kg}\cdot\text{K}$ for R.

$$m_2 = \frac{[150(10^3)](1)}{(287)T_2} \quad \text{..... (1)}$$
$$= \frac{522.64}{T_2}$$

Step 3 of 12

Apply mass balance equation.

$$m_e = m_1 - m_2 \quad \text{..... (2)}$$

Here, m_e is the mass exiting the tank.

[illegible] 2.73 kg [illegible]

Calculate the mass withdrawn from the tank using equation (2).

$$m_e = m_1 - m_2$$
$$= 9.35 - 2.73$$
$$= 6.62\ \text{kg}$$

Therefore, the mass of air extracted is $\boxed{6.62\ \text{kg}}$.

$= 9.35 - 2.73$

$= 6.62 \text{ kg}$

Therefore, the mass of air extracted is $\boxed{6.62 \text{ kg}}$.

Step 4 of 12

(b)

Stage 1 to a: Consider the pressure drop from 800 kPa to 400 kPa.

Calculate the mass during the intermediate stage (m_2) using the ideal gas equation:

$$p_2 V_2 = m_2 R T_2$$

$$m_2 = \frac{p_2 V_2}{R T_2}$$

Here, suffix *a* indicates intermediate stage.

Substitute $400(10^3)$ Pa for p_2, 1 m^3 for V_2, and $287 \text{ J/kg} \cdot \text{K}$ for R.

$$p_2 V_2 = m_2 R T_2$$

$= 2.15 \text{ kg}$

Calculate the mass withdrawn from the tank using equation (2).

$$m_e = m_1 - m_2$$

$= 9.35 - 2.73$

$= 6.62 \text{ kg}$

Therefore, the mass of air extracted is $\boxed{6.62 \text{ kg}}$.

Step 4 of 12

(b)

Stage 1 to a: Consider the pressure drop from 800 kPa to 400 kPa.

Calculate the mass during the intermediate stage (m_2) using the ideal gas equation:

$$p_2 V_2 = m_2 R T_2$$

$$m_2 = \frac{p_2 V_2}{R T_2}$$

Here, suffix *a* indicates intermediate stage.

Substitute $400(10^3)$ Pa for p_2, 1 m^3 for V_2, and $287 \text{ J/kg} \cdot \text{K}$ for R.

$$p_2 V_2 = m_2 R T_2$$

$$m_a = \frac{400(10^3)(1)}{(287) T_2}$$

$$= \frac{1394}{T_2}$$

Step 5 of 12

Apply energy equation for the stage 1-a.

$$\Delta E = E_{in} - E_{out}$$

$$0 - m_e h_e = m_2 u_2 - m_1 u_1$$

$$m_e h_e + m_2 u_2 - m_1 u_1 = 0$$

$$\left[(m_1 - m_2) c_p \left(\frac{T_1 + T_2}{2}\right)\right] + (m_2 c_v T_2) - (m_1 c_v T_1) = 0$$

Substitute 9.35 kg for m_1, $1.005 \text{ kJ/kg} \cdot \text{K}$ for c_p, $0.718 \text{ kJ/kg} \cdot \text{K}$ for c_v, and $(25 + 273)$ K.

$$\left[\left(9.35 - \frac{1394}{T_2}\right)(1.005)\left(\frac{298 + T_2}{2}\right)\right] + \left(\frac{1394}{T_2}(0.718) T_2\right) - (9.35)(0.718)(298) = 0$$

$$T_2 = 245.12 \text{ K}$$

Step 6 of 12

Calculate the mass at the intermediate stage.

$$m_2 = \frac{1394}{T_2}$$

$$= \frac{1394}{245.12}$$

$$= 5.687 \text{ kg}$$

Calculate the mass of air extracted during the stage 1-2 (m_{1-2}) using the following relation:

$$m_{1-2} = m_1 - m_2$$

$= 9.35 - 5.687$

$= 3.66 \text{ kg}$

Step 7 of 12

Stage 2 to 3: Consider the pressure drop from 400 kPa to 150 kPa.

Apply energy equation for the stage 2-3.

$$\Delta E = E_{in} - E_{out}$$

$$0 - m_e h_e = m_3 u_3 - m_2 u_2$$

$$m_e h_e + m_3 u_3 - m_2 u_2 = 0$$

$$\left[(m_2 - m_3) c_p \left(\frac{T_3 + T_2}{2}\right)\right] + (m_3 c_v T_3) - (m_2 c_v T_2) = 0$$

Substitute $\frac{522.64}{T_3}$ for m_3, $1.005 \text{ kJ/kg} \cdot \text{K}$ for c_p, $0.718 \text{ kJ/kg} \cdot \text{K}$ for c_v, and $(25 + 273)$ K.

$$\left[\left[\left(5.687 - \frac{522.64}{T_3}\right)(1.005)\left(\frac{245.12 + T_3}{2}\right)\right] + \left(\frac{522.64}{T_3}(0.718) T_3\right)\right] - 0$$

Substitute $\frac{522.64}{T_3}$ for m_3, $1.005\ \text{kJ/kg·K}$ for c_p, $0.718\ \text{kJ/kg·K}$ for c_v, and $(25+273)\ \text{K}$.

$$\left[\left[\left(5.687-\frac{522.64}{T_3}\right)(1.005)\left(\frac{245.12+T_3}{2}\right)\right]+\left(\frac{522.64}{T_3}(0.718)T_3\right)\right]=0$$
$$-(5.687)(0.718)(245.12)$$

$$T_3=186.47\ \text{K}$$

Step 8 of 12

Calculate the mass m_3 using the following relation:

$$m_3=\frac{522.64}{T_3}$$
$$=\frac{522.64}{186.47}$$
$$=2.80\ \text{kg}$$

Calculate the mass extracted during stage 2-3 using the following relation:

$$m_{2-3}=m_2-m_3$$
$$=5.687-2.80$$
$$=2.887\ \text{kg}$$

Calculate the total mass extracted during the process:

$$m_e=m_{1-2}+m_{2-3}$$
$$=3.66+2.887$$
$$=6.547\ \text{kg}$$

Therefore, the mass of air extracted is $\boxed{6.547\ \text{kg}}$.

Step 9 of 12

(c)

Consider the differential form of mass flow rate

$$\dot{m}_e=\frac{dm}{dt}$$

Apply the relation for the ideal gas equation:

$$pV=mRT$$
$$m=\frac{pV}{RT}$$

Differentiate the equation with respect to time.

$$pV=mRT$$
$$\frac{dm}{dt}=\frac{d}{dt}\left(\frac{pV}{RT}\right)$$
$$=\frac{V}{R}\frac{d(p/T)}{dt}$$

Step 10 of 12

The volume of the tank is constant through the process. Calculate the change in energy during the process as follows:

The change in internal energy (du) in the tank is equal to the rate of enthalpy exiting the tank (h_e).

$$\frac{d(mu)}{dt}=-h_e\dot{m}_e$$
$$c_v\frac{d(mT)}{dt}=c_pT\frac{dm}{dt}$$

Substitute $\frac{V}{R}\frac{d(p/T)}{dt}$ for $\frac{dm}{dt}$.

$$c_v\frac{v}{R}\frac{d\left[(P/T)(T)\right]}{dt}=c_p\left(\frac{dP}{dt}-\frac{P}{T}\frac{dT}{dt}\right)$$
$$c_v\frac{v}{R}\frac{dP}{dt}=c_p\left(\frac{dP}{dt}-\frac{P}{T}\frac{dT}{dt}\right)$$
$$(c_p-c_v)\frac{dP}{P}=c_p\frac{dT}{T}$$
$$\frac{(c_p-c_v)}{c_p}\frac{dP}{P}=\frac{dT}{T}$$
$$\left(1-\frac{c_v}{c_p}\right)\frac{dP}{P}=\frac{dT}{T}$$
$$\left(1-\frac{1}{\gamma}\right)\frac{dP}{P}=\frac{dT}{T}$$
$$\left(\frac{\gamma-1}{\gamma}\right)\frac{dP}{P}=\frac{dT}{T}$$

Step 11 of 12

Integrate the above equation.

$$\frac{(\gamma-1)}{\gamma}\int_{P_1}^{P_2}\left(\frac{dP}{P}\right)=\int_{T_1}^{T_2}\left(\frac{dT}{T}\right)$$
$$\frac{(\gamma-1)}{\gamma}\left[\ln(p_2)-\ln(p_1)\right]=\ln(T_2)-\ln(T_1)$$
$$\frac{(\gamma-1)}{\gamma}\ln\left(\frac{p_2}{p_1}\right)=\ln\left(\frac{T_2}{T_1}\right)$$
$$\ln\left(\frac{p_2}{p_1}\right)^{\frac{(\gamma-1)}{\gamma}}=\ln\left(\frac{T_2}{T_1}\right)$$
$$\left(\frac{T_2}{T_1}\right)=\left(\frac{p_2}{p_1}\right)^{\frac{(\gamma-1)}{\gamma}}$$

$$\ln\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \ln\left(\frac{T_2}{T_1}\right)$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma-1)}{\gamma}}$$

Substitute 298 K for T_1, 800 kPa for p_1, 150 kPa for p_2 and 1.4 for γ.

$$\left(\frac{T_2}{298}\right) = \left(\frac{150}{800}\right)^{\frac{(1.4-1)}{1.4}}$$

$$T_2 = 184.7 \text{ K}$$

$$m_e = m_1 - m_2$$
$$= 9.35 - 2.830$$
$$= 6.52 \text{ kg}$$

Therefore, the mass removed from the tank if the variation in h_e is accounted, is **6.52 kg**.

Question: 6.165P

A liquid R-134a bottle has an internal volume of 0.001 m3. Initially it contains 0.4 kg of R-134a (saturated mixture) at 26°C. A valve is opened and R-134a vapor only (no liquid) is allowed to escape slowly such that temperature remains constant until the mass of R-134a remaining is 0.1 kg. Find the heat transfer necessary with the surroundings to maintain the temperature and pressure of the R-134a constant.

Step 1 of 6

Assuming container as the control volume since mass crosses the boundary.

Applying mass balance for a steady flow device,

$$\dot{m}_{in} - \dot{m}_{out} = \Delta m_{system} \quad \text{..... (1)}$$

Since there is no mass of R-134a is entering into the system

$$\dot{m}_{in} = 0$$

Then equation (1) becomes

$$0 - m_e = m_2 - m_1$$
$$m_e = m_1 - m_2$$

Here, initial mass of the refrigerant inside the container, $m_1 = 0.4\,\text{kg}$, Final mass of the refrigerant inside the container, $m_2 = 0.1\,\text{kg}$

Step 2 of 6

Applying energy balance to the control volume

$$\dot{E}_{in} - \dot{E}_{out} = \Delta E_{system}$$
$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$Q_{in} = m_2 u_2 - m_1 u_1 + m_e h_e$$
$$= m_2 u_2 - m_1 u_1 + m_e h_e$$
$$= m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$

The specific volume at the initial state can be determined as

$$v_1 = \frac{V}{m_1}$$
$$= \frac{0.001(\text{m}^3)}{0.4(\text{kg})}$$
$$= 0.0025\,\text{m}^3/\text{kg}$$

Here, Volume of the tank, $V = 0.001\,\text{m}^3$.

Step 3 of 6

At Temperature inside the container, $T_1 = 26°\text{C}$ from saturated R-134a refrigerant tables

$$v_f = 0.0008313\,\text{m}^3/\text{kg}$$
$$v_g = 0.029976\,\text{m}^3/\text{kg}$$
$$u_f = 87.26\,\text{kJ/kg}$$
$$u_{fg} = 156.87\,\text{kJ/kg}$$

Step 4 of 6

Then specific volume at state1 is

$$v_1 = v_f + x_1 v_{fg}$$
$$v_1 = v_f + x_1 (v_g - v_f)$$

$$x_1 = \frac{v_1 - v_f}{v_g - v_f}$$
$$= \frac{0.0025 - 0.0008313}{0.029976 - 0.0008313}$$
$$= 0.05726$$

Internal energy at state 1 is

$$u_1 = u_f + x_1 u_{fg}$$
$$= 87.26 + 0.05726 \times 156.87$$
$$= 96.24\,\text{kJ/kg}$$

Step 5 of 6

Since saturated vapor escapes slowly by maintaining the temperature remains constant

Therefore from the saturated R-134a tables, enthalpy of saturated vapor refrigerant leaving the bottle is

$$h_e = h_{g@26°\text{C}} = 264.68\,\text{kJ/kg}$$

The final specific volume in the container is

$$v_2 = \frac{V}{m_2}$$
$$= \frac{0.001(\text{m}^3)}{0.1(\text{kg})}$$
$$= 0.01\,\text{m}^3/\text{kg}$$

From saturated R-134a tables at $T_2 = T_1 = 26°\text{C}$

$$v_2 = v_f + x_2 (v_g - v_f)$$

$$x_2 = \frac{v_1 - v_f}{v_g - v_f}$$
$$= \frac{0.01 - 0.0008313}{0.029976 - 0.0008313}$$
$$= 0.3146$$

Step 6 of 6

Internal energy at state 2 is

$$u_2 = u_f + x_2 u_{fg}$$
$$= 87.26 + 0.3146 \times 156.87$$
$$= 136.61\,\text{kJ/kg}$$

Substituting the known values in the energy equation

$$Q_{in} = m_2 u_2 - m_1 u_1 + (m_1 - m_2) h_e$$
$$= 0.1 \times 136.61 - 0.4 \times 96.24 + (0.4 - 0.1) \times 264.68$$
$$= 54.569\,\text{kJ}$$

Therefore heat transfer with the surroundings is $\boxed{54.569\,\text{kJ}}$

Question: 6.166P

Steam enters a turbine steadily at 7 MPa and 600°C with a velocity of 60 m/s and leaves at 25 kPa with a quality of 95 percent. A heat loss of 20 kJ/kg occurs during the process. The inlet area of the turbine is 150 cm2, and the exit area is 1400 cm2. Determine (a)the mass flow rate of the steam, (b) the exit velocity, and (*c*) the power output.

Step 1 of 5

From the properties of superheated water,

At state-1

At temperature $T_1 = 600°C$ and pressure $P_1 = 7\,MPa$

The specific volume is $v_1 = 0.05567\,m^3/kg$

The specific enthalpy is $h_1 = 3650.6\,kJ/kg$

From the properties of saturated water

At state-2

At pressure $P_2 = 25\,kPa$

$v_f = 0.00102\,m^3/kg$ and $v_{fg} = 6.20238\,m^3/kg$

$h_f = 271.96\,kJ/kg$ and $h_{fg} = 2345.5\,kJ/kg$

Step 2 of 5

Find the enthalpy at state-2,

$$h_2 = h_f + x_2 h_{fg}$$

Substitute $271.96\,kJ/kg$ for h_f, 0.95 for x_2 and $2345.5\,kJ/kg$ for h_{fg},

$$\begin{aligned} h_2 &= h_f + x_2 h_{fg} \\ &= 271.96 + (0.95)(2345.5) \\ &= 2500.2\,kJ/kg \end{aligned}$$

Find the specific volume at state-2,

$$v_2 = v_f + x_2 v_{fg}$$

Substitute $0.00102\,m^3/kg$ for v_f, 0.95 for x_2 and $6.20238\,m^3/kg$ for v_{fg},

$$\begin{aligned} v_2 &= v_f + x_2 v_{fg} \\ &= 0.00102 + (0.95)(6.20238) \\ &= 5.8933\,m^3/kg \end{aligned}$$

Step 3 of 5

a)

Calculate the mass flow rate of the steam,

$$\dot{m} = \frac{V_1 A_1}{v_1}$$

Here,

The velocity of steam at state-1 is V_1

The inlet area of the turbine is A_1

Substitute $60\,m/s$ for V_1, $0.015\,m^2$ for A_2 and $0.05567\,m^3/kg$ for v_1,

$$\begin{aligned} \dot{m} &= \frac{V_1 A_1}{v_1} \\ &= \frac{60 \times 0.015}{0.05567} \\ &= 16.17\,kg/s \end{aligned}$$

Therefore, the mass flow rate of the steam is $\boxed{16.17\,kg/s}$

Step 4 of 5

b)

Calculate the exit velocity of the steam,

$$\dot{m} = \frac{V_2 A_2}{v_2}$$

Here,

The velocity of steam at state-2 is V_2

The inlet area of the turbine is A_2

Substitute $16.17\,kg/s$ for $\dot{m}$, $0.14\,m^2$ for A_2 and $5.8933\,m^3/kg$ for v_2,

$$\begin{aligned} 16.17 &= \frac{V_2 \times 0.14}{5.8933} \\ &= 680.6\,m/s \end{aligned}$$

Step 5 of 5

c)

Calculate the power output by using the energy balance equation,

$$\dot{W}_{out} = -\dot{Q}_{out} - \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substitute $16.17\,kg/s$ for $\dot{m}$, $20\,kJ/kg$ for $\dot{Q}_{out}$, $3650.6\,kJ/kg$ for h_1,

$2500.2\,kJ/kg$ for h_2, $680.6\,m/s$ for V_2 and $60\,m/s$ for V_1

$$\begin{aligned} \dot{W}_{out} &= -(16.17 \times 20) - 16.17\left(2500.2 - 2650.6 + \frac{680.6^2 - 60^2}{2}\right) \\ &= 14560\,kW \end{aligned}$$

Therefore, the power output of the turbine is $\boxed{14560\,kW}$

Question: 6.167P

Reconsider Prob. 6–166. Using EES (or other) software, investigate the effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine. Let the exit pressure vary from 10 to 50 kPa (with the same quality), and the exit area to vary from 1000 to 3000 cm2. Plot the exit velocity and the power outlet against the exit pressure for the exit areas of 1000, 2000, and 3000 cm2, and discuss the results.

Step 1 of 3

Open the EES software, then click file on menu bar and select the new file on menu bar.

Then write the program in equation window.

Equations Window

```
Fluid$='Steam_IAPWS'
A[1]=150 [cm^2]
T[1]=600 [C]
P[1]=7000 [kPa]
Vel[1]= 60 [m/s]
A[2]=3000 [cm^2]
"P[2]=25 [kPa]"
q_out = 20 [kJ/kg]
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2)
v[1]=volume(Fluid$, T=T[1], P=P[1]) "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2))
v[2]=volume(Fluid$, x=0.95, P=P[2]) "specific volume of steam at state 2"
T[2]=temperature(Fluid$, P=P[2], v=v[2]) "[C]" "not required, but good to know"
"[conservation of Energy for steady-flow:"
"Ein_dot - Eout_dot = DeltaE_dot" "For steady-flow, DeltaE_dot = 0"
DELTAE_dot=0
"For the turbine as the control volume, neglecting the PE of each flow steam:"
E_dot_in=E_dot_out
h[1]=enthalpy(Fluid$,T=T[1], P=P[1])
E_dot_in=m_dot*(h[1]+ Vel[1]^2/2*Convert(m^2/s^2, kJ/kg))
h[2]=enthalpy(Fluid$,x=0.95, P=P[2])
E_dot_out=m_dot*(h[2]+ Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+ m_dot *q_out+ W_dot_out
Power=W_dot_out
Q_dot_out=m_dot*q_out
```

Step 2 of 3

After writing the equation in equation window, then select table on menu bar, then click new parameter table.

The following results are displayed

Parametric Table

Table 1 | Table 2 | Table 3

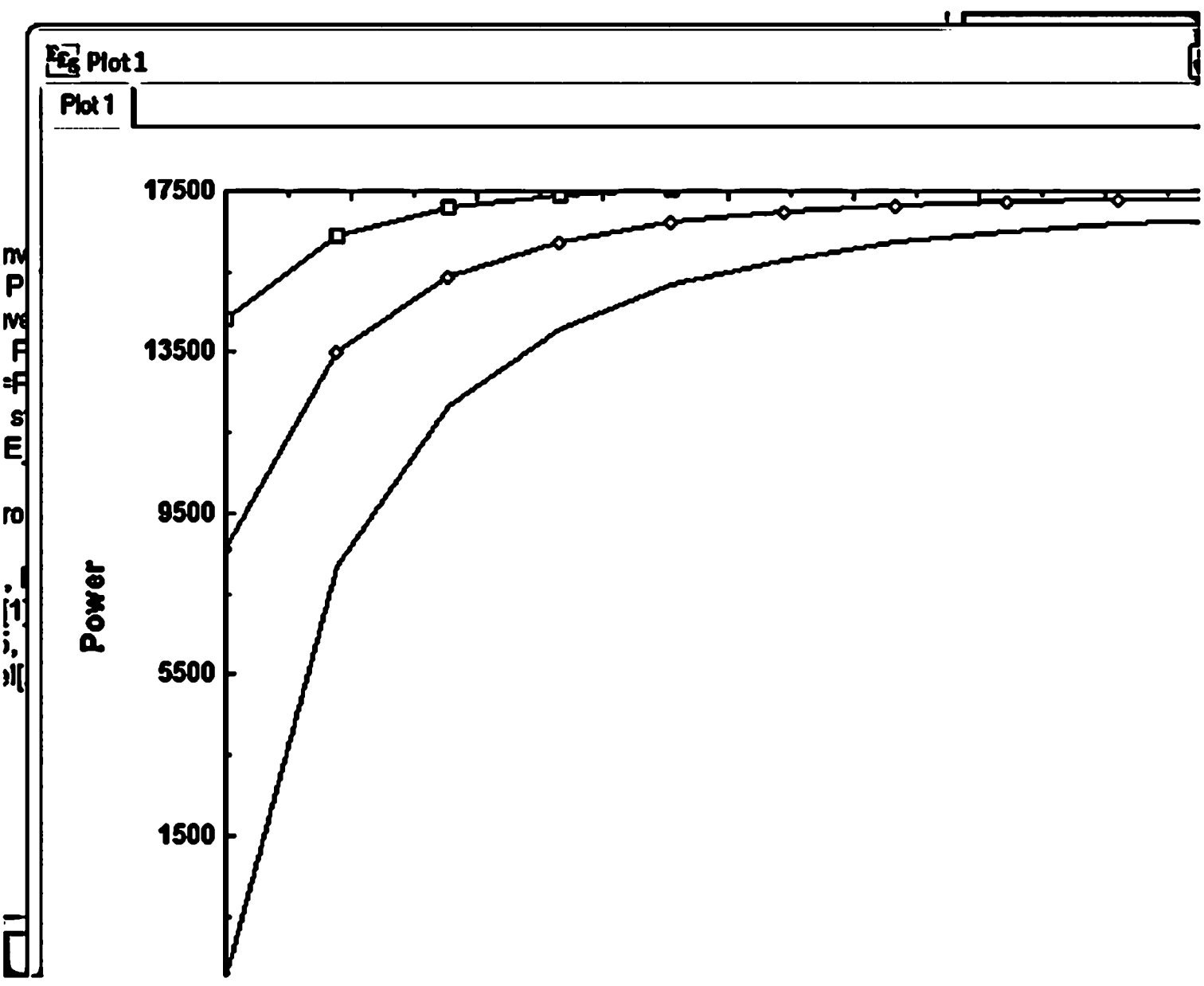
1..10	P_2 [kPa]	v_2	Power
Run 1	10	13.94	14326
Run 2	14.44	9.863	16377
Run 3	18.89	7.666	17110
Run 4	23.33	6.287	17424
Run 5	27.78	5.339	17566
Run 6	32.22	4.645	17627
Run 7	36.67	4.116	17646
Run 8	41.11	3.698	17643
Run 9	45.56	3.359	17626
Run 10	50	3.078	17601

Step 3 of 3

Select the plot on menu bar, then click x-y plot .Dialog box will appeared, then select variables to draw graph, then click ok button.

The graph will appeared

By plotting the graph between Power and P[2]:



Question: 6.168P

In large gas-turbine power plants, air is preheated by the exhaust gases in a heat exchanger called the regeneratorbefore it enters the combustion chamber. Air enters the regenerator at 1 MPa and 550 K at a mass flow rate of 800 kg/min. Heat is transferred to the air at a rate of 3200 kJ/s. Exhaust gases enter the regenerator at 140 kPa and 800 K and leave at 130 kPa and 600 K. Treating the exhaust gases as air, determine (a) the exit temperature of the air and (b) the mass flow rate of exhaust gases.

Step 1 of 6

From properties of air table, gas constant of air is

$$R = 0.287 \text{ kJ/kg-K}$$

From ideal gas properties of air at inlet temperature $T_1 = 550 \text{ K}$, specific enthalpy of air is

$$h_1 = 555.74 \text{ kJ/kg}$$

Let us assume that the properties of exhaust gases are taken as properties of air.

From ideal gas properties of air at exhaust gas inlet temperature $T_3 = 800 \text{ K}$, specific enthalpy of gas is

$$h_3 = 821.95 \text{ kJ/kg}$$

Similarly at exhaust gas exit temperature $T_4 = 600 \text{ K}$, specific enthalpy of gas is

$$h_4 = 607.02 \text{ kJ/kg}$$

Step 2 of 6

(a)

As there is only one entry, one exit for air and as air and exhaust gases are not mixing anywhere, mass flow rate of air will be same

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{air}$$

Apply the energy balance to the steady flow system,

$$E_{in} - E_{out} = \Delta E_{sys}$$

Here, E_{in}, E_{out} are rate of net energy transfer and ΔE_{sys} is rate of change in internal, kinetic and potential energies.

For steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$Q_{in} + \dot{m}_{air} \times h_1 = \dot{m}_{air} \times h_2$$
$$Q_{in} = \dot{m}_{air} \times h_2 - \dot{m}_{air} \times h_1$$

Step 3 of 6

Then find the amount of heat transferred to air by using the expression,

$$Q_{in} = \dot{m}_{air} \times (h_2 - h_1)$$

Substitute 3200kJ/s for Q_{in} and $\frac{800}{60}$ kg/s for $\dot{m}_{air}$

$$3200 = \frac{800}{60} \times (h_2 - 555.74)$$
$$3200 = 13.33 \times (h_2 - 555.74)$$
$$h_2 = \frac{3200}{13.33} + 555.74$$
$$h_2 = 240.06 + 555.74$$
$$= 795.8 \text{ kJ/kg}$$

Step 4 of 6

From ideal gas properties of air table, find exit temperature of air by using interpolation

$$\frac{T_2 - 760}{780 - 760} = \frac{795.8 - 778.18}{800.03 - 778.18}$$
$$\frac{T_2 - 760}{20} = \frac{17.62}{21.85}$$
$$\frac{T_2 - 760}{20} = 0.8064$$
$$T_2 = 0.8064 \times 20 + 760$$
$$T_2 = 776.128 \text{ K}$$

Therefore, exit temperature of air is $\boxed{776.128 \text{K}}$

Step 5 of 6

(b)

As there is only one entry, one exit for air and as air exhaust gases are not mixing anywhere, mass flow rate of exhaust gas will be same

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{gas}$$

Assume exhaust gases behaves as an ideal gas, apply the energy balance of exhaust gas

$$E_{in} - E_{out} = \Delta E_{sys}$$

For steady process, $\Delta E_{sys} = 0$ and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$\dot{m}_{gas} \times h_3 = Q_{out} + \dot{m}_{gas} \times h_4$$
$$Q_{out} = \dot{m}_{gas} \times h_3 - \dot{m}_{gas} \times h_4$$
$$Q_{out} = \dot{m}_{gas} \times (h_3 - h_4)$$
$$\dot{m}_{gas} = \frac{Q_{out}}{(h_3 - h_4)}$$

Step 6 of 6

The amount of heat rejected from exhaust gases is equal to the heat addition to air then the above expression becomes,

$$\dot{m}_{gas} = \frac{Q_{in}}{(h_3 - h_4)}$$
$$= \frac{3200}{(821.95 - 607.02)}$$
$$= \frac{3200}{214.93}$$
$$= 14.89 \text{kg/s}$$

Therefore, the mass flow rate of exhaust gases is $\boxed{14.89 \text{ kg/s}}$

Question: 6.169P

It is proposed to have a water heater that consists of an insulated pipe of 5-cm diameter and an electric resistor inside. Cold water at 20°C enters the heating section steadily at a rate of 30 L/min. If water is to be heated to 55 °C, determine (a) the power rating of the resistance heater and (b) the average velocity of the water in the pipe.

Step 1 of 6

From saturated water tables at average temperature of water $T = 25\ ^\circ\text{C}$, the properties of water is

Density of water is $\rho = 997\ \text{kg/m}^3$

Specific heat of water at constant pressure $c_p = 4.18\ \text{kJ/kg}\cdot\text{K}$

Step 2 of 6

As there is only one entry, one exit for water, mass flow rate of water will be same

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

Find the mass flow rate of water by using the equation,

$$\dot{m} = \rho \times \dot{V}_1$$

Here ρ is the density of water and $\dot{V}_1$ is volume flow rate of water at inlet

Substitute $30\ \text{L/min} = \frac{0.03}{60}\ \text{m}^3/\text{s}$ for $\dot{V}_1$

$$\dot{m} = 997 \times \frac{0.03}{60}$$
$$= 997 \times 0.0005$$
$$= 0.4985\ \text{kg/s}$$

Step 3 of 6

Find the energy balance of steady flow system

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady process, $\Delta E_{sys} = 0$ is zero and so

$$E_{in} - E_{out} = 0$$
$$E_{in} = E_{out}$$
$$W_{in} + \dot{m} \times h_1 = \dot{m} \times h_2$$
$$W_{in} = \dot{m} \times h_2 - \dot{m} \times h_1$$

Step 4 of 6

(a)

Find the power rating of the resistance heater from energy equation,

$$W_{in} = \dot{m} \times (h_2 - h_1)$$
$$W_{in} = \dot{m} \times c_p \times (T_2 - T_1)$$

Here T_1 and T_2 are inlet and exit temperatures of water respectively

Substitute 55°C for T_2 and 20°C for T_1

$$W_{in} = 0.4985 \times 4.18 \times (55 - 20)$$
$$= 0.4985 \times 4.18 \times 35$$
$$= 72.93\ \text{kW}$$

Therefore, power rating of the resistance heater is $\boxed{72.9306\ \text{kW}}$

Step 5 of 6

(b)

Find cross-section area of the pipe by using the equation,

$$A = \pi \times \frac{D^2}{4}$$

Here D is the diameter of cross-section of the pipe

Substitute 0.05 m for D

$$A = \pi \times \frac{0.05^2}{4}$$
$$= \pi \times \frac{0.0025}{4}$$
$$= \pi \times 0.000625$$
$$= 0.0019635\ \text{m}^2$$

Step 6 of 6

Now find velocity of water by using volume flow rate of water,

$$V_1 = \frac{\dot{V}_1}{A}$$
$$= \frac{0.03/60}{0.0019635}$$
$$= \frac{0.0005}{0.0019635}$$
$$= 0.2546\ \text{m/s}$$

Therefore, the velocity of water in the pipe is $\boxed{0.2546\ \text{m/s}}$

Question: 6.170P

An insulated vertical piston-cylinder device initially contains 0.11 m3 of air at 150 kPa and 22°C. At this state, a linear spring touches the piston but exerts no force on it. The cylinder is connected by a valve to a line that supplies air at 700 kPa and 22°C. The valve is opened, and air from the high-pressure line is allowed to enter the cylinder. The valve is turned off when the pressure inside the cylinder reaches 600 kPa. If the enclosed volume inside the cylinder doubles during this process, determine (*a*) the mass of air that entered the cylinder, and (b) the final temperature of the air inside the cylinder.

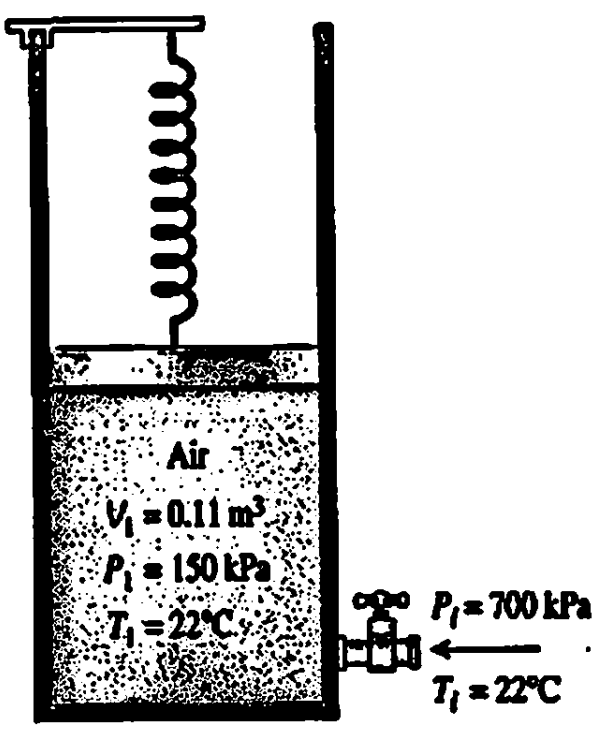


FIGURE P6–170

Step 1 of 6

Apply mass balance equation to the piston cylinder arrangement.

$$m_i + m_1 = m_e + m_2$$

Here, m_i is mass of air entering into the cylinder, m_1 is mass of air in the cylinder at initial state, m_2 is mass of air in the cylinder at final state and m_e is mass of air leaving from the cylinder.

Since there is no mass of air leaving from the cylinder,

$$m_e = 0$$

The mass balance equation becomes.

$$m_i + m_1 = m_e + m_2$$
$$m_i + m_1 = 0 + m_2$$

$$m_i = m_2 - m_1 \quad \text{..... (1)}$$

Step 2 of 6

Apply energy balance equation to the arrangement.

$$m_i h_i + m_1 u_1 = m_e h_e + m_2 u_2 + W_{b,out}$$

Here, *u* is internal energy and *h* is enthalpy.

Since there is no mass of air leaving from the system $m_e = 0$, then energy equation becomes.

$$m_i h_i + m_1 u_1 = 0 \times h_e + m_2 u_2 + W_{b,out}$$
$$m_i h_i = m_2 u_2 - m_1 u_1 + W_{b,out}$$

Substitute $m_2 - m_1$ for m_i .

$$(m_2 - m_1) h_i = m_2 u_2 - m_1 u_1 + W_{b,out}$$

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1 + W_{b,out} \quad \text{..... (2)}$$

Here, c_p is specific heat at constant pressure, c_v is specific heat at constant volume and *T* is temperature.

Step 3 of 6

Find the mass of air at initial state by using ideal gas equation,

$$P_1 V_1 = m_1 R T_1$$
$$m_1 = \frac{P_1 V_1}{R T_1}$$

Here, T_1 is initial temperature, V_1 is initial volume of the cylinder and *R* is gas constant.

Substitute 150 kPa for P_1 , $0.11\,m^3$ for V_1 and $0.287\,kJ/kg \cdot K$ for *R*

$$m_1 = \frac{150 \times 0.11}{0.287 \times 295}$$
$$= 0.1948\,kg$$

Find the mass of air at final state by using ideal gas equation,

$$P_2 V_2 = m_2 R T_2$$
$$m_2 = \frac{P_2 V_2}{R T_2}$$

Here, T_2 is final temperature V_2 is final volume of the cylinder and *R* is gas constant.

Substitute 600 kPa for P_2 , $0.22\,m^3$ for V_1 and $0.287\,kJ/kg \cdot K$ for *R*

$$m_2 = \frac{600 \times 0.22}{0.287 \times T_2}$$
$$= \frac{459.93}{T_2}$$

Step 4 of 6

Find the boundary work due to the linear spring by using the following equation:

$$W_{b,out} = \frac{1}{2}(P_1 + P_2)(V_2 - V_1)$$

Substitute 150 kPa for P_1 , 600 kPa for P_2 , $0.11\,m^3$ for V_1 and $0.22\,m^3$ for V_2

$$W_{b,out} = \frac{1}{2}(150 + 600)(0.22 - 0.11)$$
$$= 41.25\,kJ/kg$$

Step 5 of 6

Substitute $\frac{459.93}{T_2}$ for m_2 , 0.1948 kg for m_1 , 295 K for T_1 , 295 K for T_i , $1.005\,kJ/kg \cdot K$ for c_p , $0.718\,kJ/kg \cdot K$ for c_v and 41.25 kJ/kg for $W_{b,out}$ in equation (2)

$$(m_2 - m_1) c_p T_i = m_2 c_v T_2 - m_1 c_v T_1 + W_{b,out}$$
$$\left(\frac{459.93}{T_2} - 0.1948\right) \times 1.005 \times 295 = \frac{459.93}{T_2} \times 0.718 \times T_2 - 0.1948 \times 0.718 \times 295 + 41.25$$
$$\left(\frac{459.93}{T_2} - 0.1948\right) 296.475 = 330.229 - 0.0105$$
$$T_2 = 351.46\,K$$

Therefore, the final temperature of air in the tank is $\boxed{351.46\,K}$

Step 6 of 6

Find the mass of air entering into the tank by using equation (1),

$$m_i = m_2 - m_1$$
$$= \frac{459.93}{T_2} - m_1$$

Substitute 351.46 K for T_2 and 0.1948 kg for m_1

$$m_i = \frac{459.93}{351.46} - 0.1948$$
$$= 1.1138\,kg$$

Therefore, mass of air entering into the tank is $\boxed{1.1138\,kg}$

Question: 6.171P

A piston-cylinder device initially contains 2 kg of l refrigerant-134a at 800 kPa and 80°C. At this state, the piston is touching on a pair of stops at the top. The mass of the piston is such that a 500-kPa pressure is required to move it. A valve at the bottom of the tank is opened, andR-134a is withdrawn from the cylinder. After a while, the piston is observed to move and the valve is closed when half of the refrigerant is withdrawn from the tank and the temperature in the tank drops to 20°C. Determine (*a*) the j work done and (b) the heat transfer.

Step 1 of 4

From the table, saturated refrigeration table of R-134a at various states obtain the following values

At initial pressure $P_1 = 800\,\text{kPa}$, and initial temperature $T_1 = 80°\text{C}$

At initial state

Specific volume $v_1 = 0.32659\ \text{m}^3/\text{kg}$.

Internal energy $u_1 = 290.84\ \text{kJ/kg}$

Enthalpy $h_1 = 316.97\ \text{kJ/kg}$

At final pressure $P_2 = 500\,\text{kPa}$, and final temperature $T_2 = 20°\text{C}$

At final state

Specific volume $v_2 = 0.042115\ \text{m}^3/\text{kg}$.

Internal energy $u_2 = 242.40\ \text{kJ/kg}$

Enthalpy $h_2 = 263.46\ \text{kJ/kg}$

Step 2 of 4

Find the volume of refrigerant at initial state

$$\begin{aligned} V_1 &= m_1 \times v_1 \\ &= 2\ \text{kg} \times 0.32659\ \text{m}^3/\text{kg} \\ &= 0.06532\,\text{m}^3 \end{aligned}$$

Find the volume of refrigerant at final state

$$\begin{aligned} V_2 &= m_2 \times v_2 \\ &= 0.5m_1 \times v_2 \\ &= 0.5 \times 2\,\text{kg} \times 0.042115\ \text{m}/\text{kg} \\ &= 0.04212\ \text{m}^3 \end{aligned}$$

Find the mass left in the cylinder from the mass balance relation

$$\begin{aligned} m_e &= m_1 - m_2 \\ &= 2 - 1 \\ &= 1\ \text{kg} \end{aligned}$$

Step 3 of 4

Find the enthalpy of the refrigerant withdrawn from the cylinder

$$\begin{aligned} h_e &= \left(\frac{h_1 + h_2}{2}\right) \\ &= \left(\frac{316.97 + 263.46}{2}\right) \\ &= 290.21\ \text{kJ/kg} \end{aligned}$$

Find the boundary work for the piston movement

$$\begin{aligned} W_{b,in} &= P_2(V_1 - V_2) \\ &= (500\text{kPa})(0.06532 - 0.04212)\text{m}^3 \\ &= 11.6\ \text{kJ} \end{aligned}$$

Therefore, the boundary work for the piston movement $\boxed{11.6\ \text{kJ}}$

Step 4 of 4

Find the heat transferred from the system as below

Substitute the $11.6\ \text{kJ}$ for $W_{b,in}$ in the equation (2)

$$W_{b,in} - Q_{out} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$11.6\,\text{kJ} - Q_{out} - (1\,\text{kg})(290.21\,\text{kJ/kg}) = (1\ \text{kg})(242.40\ \text{kJ/kg})\ (2\ \text{kg})(290.84\ \text{kJ/kg})$$

$$Q_{out} = 60.7\,\text{kJ}$$

Therefore, the heat transferred from the system is $\boxed{60.7\,\text{kJ}}$

Question: 6.172P

The pump of a water distribution system is powered by a 6-kW electric motor whose efficiency is 95 percent. The water flow rate through the pump is 18 L/s. The diameters of the inlet and outlet pipes are the same, and the elevation difference across the pump is negligible. If the pressures at the inlet and outlet of the pump are measured to be 100 kPa and 300 kPa (absolute), respectively, determine (*a*) the mechanical efficiency of the pump and (b) the temperature rise of water as it flows through the pump due to the mechanical inefficiency.

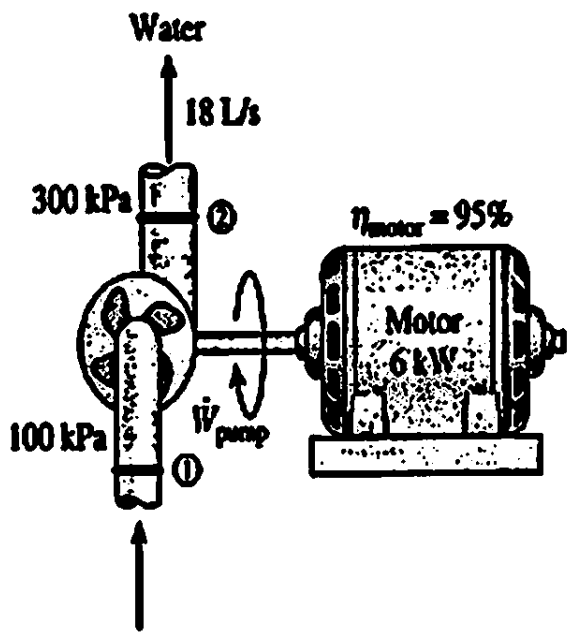


FIGURE P6–172

Step 1 of 3

(a)

Find the mass flow rate of water through the pump by using the equation,

$$\dot{m} = \rho \dot{V}$$

Here, ρ is the density of the water, and $\dot{V}$ is the volume flow rate.

Substitute 1000 kg / m^3 for ρ , and 18 L/s for $\dot{V}$.

$$\begin{aligned}\dot{m} &= 1000\frac{\text{kg}}{\text{m}^3}\times 18\frac{\text{L}}{\text{s}}\times 10^{-3}\frac{\text{m}^3}{\text{L}}\\ &= 18\,\text{kg/s}\end{aligned}$$

Find the mechanical (shaft) power it delivers to the pump is

$$\dot{W}_{\text{pump, shaft}} = \eta_{\text{motor}}\dot{W}_{\text{electric}}$$

Here η_{motor} is the efficiency of the motor, and $\dot{W}_{\text{electric}}$ is the electrical power.

Substitute 0.95 for η_{motor} , and 6 kW for $\dot{W}_{\text{electric}}$

$$\begin{aligned}\dot{W}_{\text{pump, shaft}} &= 0.95\times 6\\ &= 5.7\,\text{kW}\end{aligned}$$

Step 2 of 3

Find the mechanical energy of the fluid when pumping,

$$\begin{aligned}\Delta\dot{E}_{\text{mech,fluid}} &= \dot{E}_{\text{mech, out}} - \dot{E}_{\text{mech, in}}\\ &= \dot{m}\left(\frac{P_2}{\rho}+\frac{V_2^2}{2}+gz_2\right)-\dot{m}\left(\frac{P_1}{\rho}+\frac{V_1^2}{2}+gz_1\right)\end{aligned}$$

Substitute z_1 for z_2 and V_1 for V_2 in the above equation

$$\begin{aligned}\Delta\dot{E}_{\text{mech,fluid}} &= \dot{m}\left(\frac{P_2}{\rho}+\frac{V_1^2}{2}+gz_1\right)-\dot{m}\left(\frac{P_1}{\rho}+\frac{V_1^2}{2}+gz_1\right)\\ &= \dot{m}\left(\frac{P_2-P_1}{\rho}\right)\end{aligned}$$

Here, P_1 is the pressure at state 1 and P_2 is the pressure at state 2.

Substitute $18\,\text{kg/s}$ for $\dot{m}$, $300\,\text{kPa}$ for P_2 , $100\,\text{kPa}$ for P_1, and $1000\,\text{kg/m}^3$ for ρ .

$$\begin{aligned}\Delta\dot{E}_{\text{mech,fluid}} &= 18\left(\frac{300-100}{1000}\right)\times\left(\frac{1\,\text{kJ}}{\text{kPa}\cdot\text{m}^3}\right)\\ &= 3.6\,\text{kW}\end{aligned}$$

Find the mechanical efficiency of the pump by using the equation

$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}}$$

Substitute $3.6\,\text{kW}$ for $\Delta\dot{E}_{\text{mech,fluid}}$, and $5.7\,\text{kW}$ for $\dot{W}_{\text{pump, shaft}}$.

$$\begin{aligned}\eta_{\text{pump}} &= \frac{3.6}{5.7}\\ &= 0.6316\\ &= 63.16\%\end{aligned}$$

Therefore, the mechanical efficiency of the pump $\boxed{63.16\%}$

Step 3 of 3

(b)

Find the energy lost by the pump in the form of heat due to frictional effects,

$$\dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump,shaft}} - \Delta\dot{E}_{\text{mech,fluid}}$$

Substitute $3.6\,\text{kW}$ for $\Delta\dot{E}_{\text{mech,fluid}}$, and $5.7\,\text{kW}$ for $\dot{W}_{\text{pump, shaft}}$ in the equation

$$\begin{aligned}\dot{E}_{\text{mech,loss}} &= 5.7-3.6\\ &= 2.1\,\text{kW}\end{aligned}$$

Find the temperature rise of water,

$$\begin{aligned}\dot{E}_{\text{mech,loss}} &= \dot{m}(u_2-u_1)\\ &= \dot{m}c\Delta T\\ \Delta T &= \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c}\end{aligned}$$

Here, c is the specific heat constant.

Substitute 2.1 kW for $\dot{E}_{\text{mech,loss}}$, 16 kg/s for $\dot{m}$, and $4.18\,\text{kJ/kg}\cdot\text{K}$ for c .

$$\begin{aligned}\Delta T &= \frac{2.1\,\text{kW}}{18\,\text{kg/s}\times 4.18\,\text{kJ/kg}\cdot\text{K}}\\ &= 0.0279°\text{C}\end{aligned}$$

Therefore, the temperature rise of water is $\boxed{0.0279°\text{C}}$

Question: 6.173P pg 1

A pressure cooker is a pot that cooks food much faster than ordinary pots by maintaining a higher pressure and temperature during cooking. The pressure inside the pot is controlled by a pressure regulator (the petcock) that keeps the pressure at a constant level by periodically allowing some steam to escape, thus preventing any excess pressure buildup, pressure cookers, in general, maintain a gage pressure of 2 atm (or 3 atm absolute) inside. Therefore, pressure cookers cook at a temperature of about 133°C instead of 100°C, cutting the cooking time by as much as 70 percent while minimizing the loss of nutrients. The newer pressure cookers use a spring valve with several pressure settings rather than a weight on the cover.

A certain pressure cooker has a volume of 6 L and an operating pressure of 75 kPa gage. Initially, it contains 1 kg of water. Heat is supplied to the pressure cooker at a rate of 500 W for 30 min after the operating pressure is reached. Assuming an atmospheric pressure of 100 kPa, determine (a) the temperature at which cooking takes place and (b)the amount of water left in the pressure cooker at the end of the process.

Step 1 of 8

(a)

Convert the pressure inside the cooker from gage pressure to absolute pressure by using the relation,

$$P_{abs} = P_{gage} + P_{atm}$$

Here, P_{atm} is atmospheric pressure.

Substitute 75 kPa for P_{gage} and 100 kPa for P_{atm}

$$\begin{aligned} P_{abs} &= 75+100 \\ &= 175\,\text{kPa} \end{aligned}$$

Obtain the temperature corresponding to the saturation pressure $P_{abs} = 175\,\text{kPa}$ from saturated water tables,

$$T_{sat} = 116.04°\text{C}$$

Since saturation conditions exist in the cooker, therefore, the cooking temperature is $\boxed{116.04°\text{C}}$

Step 2 of 8

(b)

Apply mass balance equation to the system,

$$m_1 + m_i = m_2 + m_e$$

Here, m_1 is mass of water at initial state, m_2 is mass of water at final state, m_i is mas of fluid entering and m_e is mass of fluid exiting from the system.

Since there is no mass of fluid entering into the system,

$$m_i = 0$$

Then

$$\begin{aligned} m_1 + 0 &= m_2 + m_e \\ m_e &= m_1 - m_2 \end{aligned}$$

Step 3 of 8

Apply energy balance equation to the system,

$$m_1 u_1 + m_i h_i + Q_{in} = m_2 u_2 + m_e h_e$$

Since $m_i = 0$

$$m_1 u_1 + Q_{in} = m_2 u_2 + m_e h_e \quad \text{..... (1)}$$

Here, u indicates specific internal energy, h is specific enthalpy of the fluid and Q_{in} is the amount of heat addition.

Substitute $m_2 - m_1$ for m_e in equation (1)

$$\begin{aligned} m_1 u_1 + Q_{in} &= m_2 u_2 + m_e h_e \\ m_1 u_1 + Q_{in} &= m_2 u_2 + (m_1 - m_2) h_e \end{aligned}$$

Find the amount of heat transfer in kJ by using the equation,

$$Q_{in} = \dot{Q}_{in} \Delta t$$

Here, $\dot{Q}_{in}$ is rate of heat transfer and Δt is the time taken for heat transfer.

Substitute 500 W for $\dot{Q}_{in}$ and $30\,\text{min}$ for Δt

$$\begin{aligned} Q_{in} &= 500\frac{\text{J}}{\text{sec}} \times 30\,\text{min}\frac{60\,\text{sec}}{\text{min}} \\ &= 900000\,\text{J} \\ &= 900\,\text{kJ} \end{aligned}$$

Step 4 of 8

Find the enthalpy of saturated vapour at pressure 175 kPa from saturated water tables,

$$h_e = h_g = 2700.2\,\text{kJ/kg}$$

Calculate the specific volume at state 1

$$v_1 = \frac{V}{m_1}$$

Here, V is volume of pressure cooker and m_1 is mass of water

Substitute $0.006\,\text{m}^3$ for V and 1 kg for m_1

$$\begin{aligned} v_1 &= \frac{0.006}{1} \\ &= 0.006\,\text{m}^3/\text{kg} \end{aligned}$$

Step 5 of 8

Find the quality at state 1 by using the equation,

Step 5 of 8

Find the quality at state 1 by using the equation,

$$v_1 = v_f + x_1\left(v_g - v_f\right)$$

Note down the properties of saturated specific volumes at 175 kPa from saturated water tables,

Specific volume of saturated liquid, $v_f = 0.001057\,\mathrm{m^3/kg}$

Specific volume of saturated vapour, $v_g = 1.0037\,\mathrm{m^3/kg}$

Substitute $0.006\,\mathrm{m^3/kg}$ for v_1 , $0.001057\,\mathrm{m^3/kg}$ for v_f and $1.0037\,\mathrm{m^3/kg}$ for v_g

$$v_1 = v_f + x_1\left(v_g - v_f\right)$$
$$0.006 = 0.001057 + x_1\left(1.0037 - 0.001057\right)$$
$$x_1 = 0.00492$$

Step 6 of 8

Find the specific internal energy at state 1 by using the equation,

$$u_1 = u_{f@175\mathrm{kPa}} + x_1 u_{fg@175\mathrm{kPa}}$$
$$= 486.82 + 0.004929 \times 2037.7$$
$$= 496.863\,\mathrm{kJ/kg}$$

Find the final mass by using the equation,

$$m_2 = \frac{V}{v_2}$$

Find the internal energy at state 2 by using the equation,

$$u_2 = u_{f@175\mathrm{kPa}} + x_2 u_{fg@175\mathrm{kPa}}$$
$$= 486.82 + x_2 \times 2037.7$$

Find the specific volume at state 2 by using the equation,

$$v_2 = v_{f@175\mathrm{kPa}} + x_2\left(v_{g@175\mathrm{kPa}} - v_{f@175\mathrm{kPa}}\right)$$
$$= 0.001 + x_2 \times 1.003$$

Step 7 of 8

Substitute $\frac{V}{v_2}$ for m_2 and $u_f + x_2 u_{fg}$ for u_2 in equation (1)

$$m_1 u_1 + Q_{in} = m_2 u_2 + \left(m_1 - m_2\right) h_e$$
$$m_1 u_1 + Q_{in} = \frac{V}{v_2} u_2 + \left(m_1 - \frac{V}{v_2}\right) h_e$$
$$Q_{in} = \left(m_1 - \frac{V}{v_2}\right) h_e + \frac{V}{v_2} u_2 - m_1 u_1$$

Substitute 1 kg for m_1 , $0.006\,\mathrm{m^3}$ for V, $2700.2\,\mathrm{kJ/kg}$ for h_e , 496.863 kJ/kg for u_1 and 900 kJ for Q_{in}

$$900 = \left[\left(1 - \frac{0.006}{0.001 + x_2\left(1.004 - 0.001\right)}\right)2700.2 + \left(\frac{0.006}{0.001 + x_2\left(1.004 - 0.001\right)} \times \left(486.82 + x_2 2037.7\right)\right) - 1 \times 496.863\right]$$

$$900 = 2700.2 - \frac{16.2012}{0.001 + 1.003x_2} + \frac{0.006 \times \left(486.82 + 2037.7x_2\right)}{0.001 + 1.003x_2} - 496.863$$

Solve the variable x_2

$$x_2 = 0.009$$

Step 8 of 8

Now find the specific volume at state 2 by using the dryness fraction,

$$v_2 = v_{f@175\mathrm{kPa}} + x_2\left(v_{g@175\mathrm{kPa}} - v_{f@175\mathrm{kPa}}\right)$$
$$= 0.001 + x_2 \times 1.003$$
$$= 0.001 + 0.009 \times 1.003$$
$$= 0.010027\,\mathrm{m^3/kg}$$

Find the mass of water at state 2 by using the equation,

$$m_2 = \frac{V}{v_2}$$

Substitute $0.006\,\mathrm{m^3}$ for V and $0.010027\,\mathrm{m^3/kg}$ for v_2

$$m_2 = \frac{0.006}{0.010027}$$
$$= 0.598\,\mathrm{kg}$$

Therefore, mass of water at state 2 is $\boxed{0.598\,\mathrm{kg}}$

Question: 6.174P

A D0 = 10-m-diameter tank is initially filled with water 2 m above the center of a D = 10-cm-diameter valve near the bottom. The tank surface is open to the atmosphere, and the tank drains through a L = 100-m-long pipe connected to the valve. The friction factor of the pipe is given to be f = 0.015, and the discharge velocity is expressed as $V = \sqrt{\frac{2gz}{1.5 + fL/D}}$ where z is the water height above the center of the valve.

Determine (a) the initial discharge velocity from the tank and (b) the time required to empty the tank. The tank can be considered to be empty when the water level drops to the center of the valve.

Step 1 of 8

Discharge velocity is given by below equation $V = \sqrt{\frac{2 \times g \times z}{1.5 + \frac{f \times L}{D}}}$ where g is the acceleration due to gravity, z is the water level in the tank, above the center of the valve, f of the friction factor of the pipe, L is the length of the pipe and D is the diameter of the valve. Substitute $g = 9.81\ m/s^2$, $f = 0.015$, $L =$

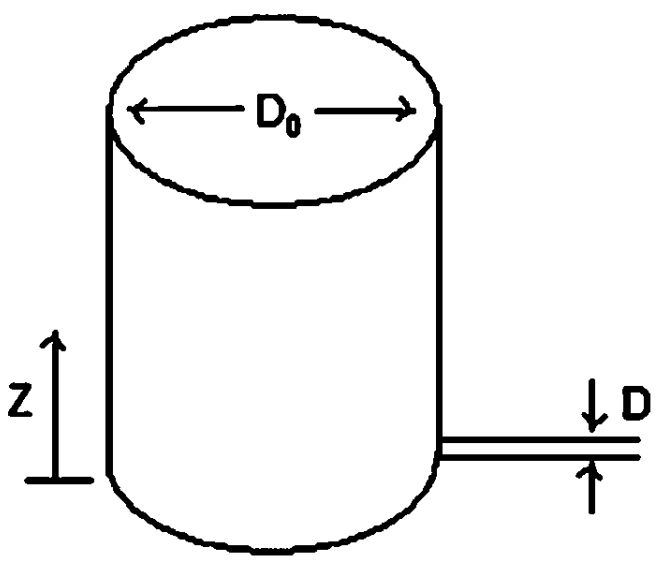
$$V = \sqrt{\frac{2 \times 9.81 \times z}{1.5 + \frac{0.015 \times 100}{0.1}}}$$

$$= \sqrt{\frac{19.62 \times z}{1.5 + \frac{1.5}{0.1}}} \qquad = \sqrt{1.189 \times z}$$

$$= \sqrt{\frac{19.62 \times z}{1.5 + 15}}$$

$$= \sqrt{\frac{19.62 \times z}{16.5}}$$

Step 2 of 8



Step 3 of 8

Substituting the value of z in the above equation we get initial discharge from the tank

$$V_1 = \sqrt{1.189 \times 2}$$

$$= \sqrt{2.378}$$

$$\boxed{V_1 = 1.542\ m/s}$$

Therefore the initial discharge from the tank is $\boxed{V_1 = 1.542\ m/s}$

Step 4 of 8

Volume flow rate of the water from the tank is given by below equation

$$\dot{V} = A_{pipe} \times V_2$$

$$= \pi \times \frac{D^2}{4} \times \sqrt{1.189 \times z}$$

Differentiating the above equation with respect to time we get the below equation

$$d\dot{V} = \pi \times \frac{D^2}{4} \times \sqrt{1.189 \times z} \times dt \quad \text{..... (1)}$$

Step 5 of 8

From conversation of mass, volume flow rate is equal to decrease in volume of the tank

$$d\dot{V} = A_{tank} \times (-dz)$$

$$= A_{tank} \times (-dz)$$

$$= -\pi \times \frac{D_0^2}{4} \times dz \quad \text{..... (2)}$$

where D_0 is the diameter of the tank

Step 6 of 8

From equation (1) and equation (2)

$$\pi \times \frac{D^2}{4} \times \sqrt{1.189 \times z} \times dt = -\pi \times \frac{D_0^2}{4} \times dz$$

$$\frac{D^2}{4} \times \sqrt{1.189 \times z} \times dt = -\times \frac{D_0^2}{4} \times dz$$

$$dt = \frac{-\times \frac{D_0^2}{4} \times dz}{\frac{D^2}{4} \times \sqrt{1.189 \times z}}$$

$$= \frac{-\times \frac{D_0^2}{4} \times dz}{\frac{D^2}{4} \times \sqrt{1.189 \times z}}$$

$$= \frac{-\times D_0^2 \times dz}{D^2 \times \sqrt{1.189} \times \sqrt{z}}$$

$$= \frac{-\times D_0^2 \times dz}{D^2 \times 1.0904} \times (z)^{-\frac{1}{2}}$$

Step 7 of 8

Differentiating the above equation within limit $z = 0$ to $z = z_1$ we get the below equation

$$t = \frac{-\times D_0^2}{D^2 \times 1.0904} \times \left[\frac{z^{\frac{1}{2}+1}}{-\frac{1}{2}+1} \right]_{z_1}^{0}$$

$$= \frac{-2 \times D_0^2}{D^2 \times 1.0904} \times \left[z^{\frac{1}{2}} \right]_{z_1}^{0}$$

$$= \frac{-2 \times D_0^2}{D^2 \times 1.0904} \times \left[0^{\frac{1}{2}} - z_1^{\frac{1}{2}} \right]$$

$$= \frac{2 \times D_0^2}{D^2 \times 1.0904} \times \sqrt{z_1}$$

Step 8 of 8

Time taken to empty the tank is

$$t = \frac{2 \times 10^2}{0.1^2 \times 1.0904} \times \sqrt{2}$$

$$= \frac{2 \times 100}{0.01 \times 1.0904} \times 1.414$$

$$= \frac{200}{0.010904} \times 1.414$$

$$= 18341.893 \times 1.414$$

$$= 25935.437\ s$$

$$\boxed{t = 7.204\ h}$$

Therefore the time taken to empty the tank is $\boxed{t = 7.204\ h}$

CHAPTER 07
SOLUTIONS

Question: 7.001P

Describe an imaginary process that satisfies the first law but violates the second law of thermodynamics.

Step 1 of 2

Assuming in a cyclic process, transferring 15kJ of heat to the system and obtaining 15kJ of work output

If you apply the first law to this cyclic process and according to first law the algebraic sum of work transfers is proportional to the algebraic sum of heat transfers

Therefore

$Q = W$

$15\text{kJ} = 15\text{kJ}$

From the above statement the first law satisfies

Step 2 of 2

Again applying second law to this cyclic process and according to Kelvin Planck statement it is impossible to convert all the heat energy to useful work output. There is some heat rejection from the system.

Here in this cyclic process heat rejection is zero so it violates the second law of thermodynamics

Question: 7.002P

Describe an imaginary process that satisfies the second law but violates the first law of thermodynamics.

Step 1 of 1

First last of thermodynamics, just discusses the changes that a system is undergoing and the processes involved in it. It explains conservation of energy for a system undergoing changes or processes.

Second law of thermodynamics helps in defining the process and also the direction of the processes. It tells about the possibility of a process or the restriction of a process. It states that the entropy of a system always increases.

For this to occur the energy contained by a body has to diminish without converting to work or internal energy. So imagine a machine which works with less than 100% efficiency, this means there are losses but they don't show up anywhere. But the energy is obtained from a higher energy source to lower.

First law is violated since the energy conversion dint apply but the direction of work is applied so second law is satisfied.

Question: 7.003P

Describe an imaginary process that violates both the first and the second laws of thermodynamics.

Step 1 of 5

First last of thermodynamics, just discusses the changes that a system is undergoing. The processes involved in it. It just deals with conservation of energy for a system undergoing changes or processes.

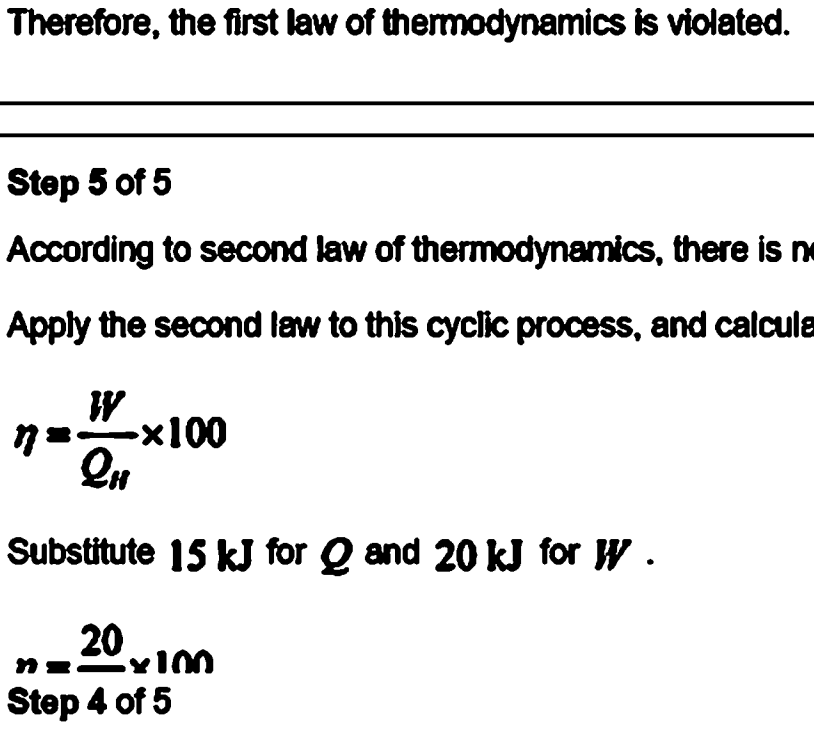
It accounts for the energy balance involving work interactions, heat interactions, etc.

It does not deal, with the possibility of the process, the direction of the process etc.

Second law of thermodynamics helps in defining the process and also the direction of the processes. It tells about the possibility of a process or the restriction of a process. It helps in analyzing the performance of processes, cycles, devices etc.

Step 2 of 5

For example, consider a device taking heat from a reservoir and producing work.



Step 3 of 5

Write an expression for the first law of thermodynamics, upon energy balance for the system.

$$Q = W_{net}$$

Here, Q is the rate of heat transfer, and W_{net} is the net amount of Work done.

It does not deal with the information about the possibility of the process or cycle. It only deals with the energy balance.

Step 4 of 5

Consider a system that transfer $15\ \text{kJ}$ of heat to the air and obtaining $20\ \text{kJ}$ of work, caused by expanding the gas and moving a piston.

Apply the first law to this cyclic process.

$$Q = W$$

Substitute $15\ \text{kJ}$ for Q and $20\ \text{kJ}$ for W .

$$15\,\text{kJ} \neq 20\text{kJ}$$

Observe the above statement, and it indicates that the cyclic process is violating the first law of thermodynamics.

Therefore, the first law of thermodynamics is violated.

Step 5 of 5

According to second law of thermodynamics, there is no heat engine which is 100% efficient.

Apply the second law to this cyclic process, and calculate the efficiency of cyclic process.

$$\eta = \frac{W}{Q_H} \times 100$$

Substitute $15\ \text{kJ}$ for Q and $20\ \text{kJ}$ for W .

$$\eta = \frac{20}{} \times 100$$

Step 4 of 5

Consider a system that transfer $15\ \text{kJ}$ of heat to the air and obtaining $20\ \text{kJ}$ of work, caused by expanding the gas and moving a piston.

Apply the first law to this cyclic process.

$$Q = W$$

Question: 7.004P

An experimentalist claims to have raised the temperature of a small amount of water to 150°C by transferring heat from high-pressure steam at 120°C. Is this a reasonable claim? Why? Assume no refrigerator or heat pump is used in the process.

Step 1 of 1

Here high pressure steam at temperature is at 120°C which is at low temperature and it transfers energy to high temperature and it is at 150°C but as you know that the energy flows from high energy to low energy but not from low energy to high energy.
Therefore transferring of energy from low temperature to high temperature is not possible and this is not a reasonable claim.

Question: 7.005P

What is a thermal energy reservoir? Give some examples.

Step 1 of 1

A thermal reservoir is a hypothetical body with a relatively large thermal energy capacity that can supply or absorb finite amounts of heat without undergoing any change in temperature. Such a body is called a thermal energy reservoir.
Some examples are oceans, lakes, rivers and a large block of copper and so on

Question: 7.006P

Consider the process of baking potatoes in a conventional oven. Can the hot air in the oven be treated as a thermal energy reservoir? Explain.

Step 1 of 1

When the heat is added to the potatoes but the temperature remains constant in the oven and it can be treated as a thermal energy reservoir.

Question: 7.007P

Consider the energy generated by a TV set. What is a suitable choice for a thermal energy reservoir?

Step 1 of 1

The atmospheric air surrounding the TV set acts as the thermal energy reservoir.

Question: 7.008P

Is it possible for a heat engine to operate without rejecting any waste heat to a low-temperature reservoir? Explain.

Step 1 of 1

It is not possible for a heat engine to operate without rejecting any waste heat to a low temperature reservoir.

As per Kelvin-Planck statement of the second law of thermodynamics it states that "It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work."

Question: 7.009P

What are the characteristics of all heat engines?

Step 1 of 1

Heat engines differ considerably from one another, but all can be characterized by the following:

1. They receive heat from a high-temperature source (solar energy, furnace etc.).
2. They convert part of this heat to work.
3. They reject the remaining waste heat to a low-temperature sink (the atmosphere, rivers, etc.).
4. Heat engines are cyclic devices

Question: 7.010P

Baseboard heaters are basically electric resistance heaters and are frequently used in space heating. A home owner claims that her 5-year-old baseboard heaters have a conversion efficiency of 100 percent. Is this claim in violation of any thermodynamic laws? Explain.

Step 1 of 1

According to second law of thermodynamics there is no system which is 100% efficient
so the claim is false i.e. the total amount of work cannot be converted into heat
So it is the violation of second law of thermodynamics

Question: 7.011P

What is the Kelvin-Planck expression of the second law of thermodynamics?

Step 1 of 1

Kelvin-Planck statement of the second law of thermodynamics states that "It is impossible for any device that operates on a cycle to receive heat from a single thermal reservoir and produce a net amount of work."

Question: 7.012P

Does a heat engine that has a thermal efficiency of 100 percent necessarily violate (*a*) the first law and (*b*) the second law of thermodynamics? Explain.

Step 1 of 1

(a) From first law of thermodynamics it states that work and heat are mutually convertible to one another and it is possible to have 100% efficient engine
So it does not violate the first law of thermodynamics
(b) As per second law of thermodynamics it is not possible for a heat engine to have 100% efficient. So it violates the second law of thermodynamics

Question: 7.013P

In the absence of any friction and other irreversibilities, can a heat engine have an efficiency of 100 percent? Explain.

Step 1 of 1

A heat engine having an efficiency of 100 percent violates Kelvin-Planck statement of the second law of thermodynamics which states that:

"It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work."

This means that the operation of a heat engine requires a low-temperature sink as well as a high-temperature source. Therefore, thermal efficiency of any heat engine cannot be 100 percent.

Also, due to the absence of friction or other dissipative effects a heat engine cannot have 100 percent efficiency, which is a limitation that applies to both the idealized and the actual heat engines.

Therefore, it is not possible for a heat engine to have 100 percent efficiency even during the absence of friction and other irreversibilities.

Question: 7.014P

Are the efficiencies of all the work-producing devices, including the hydroelectric power plants, limited by the Kelvin-Planck statement of the second law? Explain.

Step 1 of 1

Kelvin-Planck statement is valid only for heat engines undergoing a cyclic process that transfers heat from high temperature to low temperature and producing some useful work output.

Question: 7.015P

Consider a pan of water being heated (*a*) by placing it on an electric range and (*b*) by placing a heating element in the water. Which method is a more efficient way of heating water? Explain.

Step 1 of 1

Method (b) is desirable by placing a heating element in the water since the electrical heating element will be in direct contact with the water so the heat losses to the surrounding air will be less compared to the other

So the desired effect can be achieved with less power input compared to the method (a)

Question: 7.016P

A heat engine has a heat input of 3 × 104 Btu/h and a thermal efficiency of 40 percent. Calculate the power it will produce, in hp.

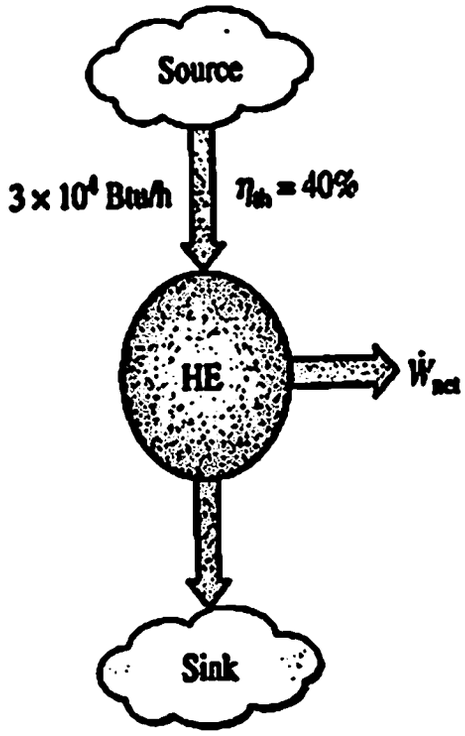


FIGURE P7–16E

Step 1 of 2

Given

The heat input to the heat engine, $\dot{Q}_H = 3\times10^4$ Btu/h

The thermal efficiency of the engine, $\eta_{HE} = 40\%$

Step 2 of 2

The thermal efficiency of a heat engine is defined as $\eta_{HE} = \dfrac{\dot{W}_{net}}{\dot{Q}_H}$

Where $\dot{W}_{net}$ the work net is transferred to the heat engine,

$\dot{Q}_H$ is the heat transfer between the device and high temperature reservoir.

Substituting the values, we get

$$\dot{W}_{net} = \eta_{HE} \times \dot{Q}_H$$
$$= (0.4)\left(3\times10^4\right) \text{ Btu/h}$$
$$= \frac{(0.4)\left(3\times10^4\right)}{2544.5} \text{ hp}$$
$$= 4.716 \text{ hp}$$

Hence the net work done is $\boxed{\dot{W}_{net} = 4.716 \text{ hp}}$

Question: 7.017P

The thermal efficiency of a general heat engine is 40 percent, and it produces 30 hp. At what rate is heat transferred to this engine, in kJ/s?

Step 1 of 2

Given

The power output from the heat engine $\dot{W}_{net} = 30$ hp

The thermal efficiency of the engine, $\eta_{HE} = 40\%$

Step 2 of 2

The thermal efficiency of a heat engine is defined as $\eta_{HE} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

Where $\dot{W}_{net}$ the work net is transferred to the heat engine,

$\dot{Q}_H$ is the heat transfer between the device and high temperature reservoir.

Substituting the values, we get

$$\begin{aligned}\dot{Q}_H &= \frac{\dot{W}_{net}}{\eta_{HE}} \\ &= \frac{30}{0.4}\ \text{hp} \\ &= \frac{30 \times 0.745}{0.4}\ \text{kJ/s} \\ &= 55.875\ \text{kJ/s}\end{aligned}$$

Hence the net work done is $\boxed{\dot{W}_{net} = 55.875\ \text{kJ/s}}$

Question: 7.018P

A 600-MW steam power plant, which is cooled by a nearby river, has a thermal efficiency of 40 percent. Determine the rate of heat transfer to the river water. Will the actual heat transfer rate be higher or lower than this value? Why?

Step 1 of 4

Given

The capacity of the steam power plant is $W_{net} = 600\,\text{MW}$

The thermal efficiency of the plant $\eta_{th} = 40\%$

Step 2 of 4

The thermal efficiency of a heat engine is defined as

$$\eta_{th} = \frac{W_{net}}{Q_H}$$

Where

W_{net} is the net work output

Q_H is the amount of heat transfer from the high temperature reservoir

$$0.4 = \frac{600}{Q_H}$$

$$Q_H = 1500\ \text{MW}$$

Step 3 of 4

By applying first law to the cyclic system

$$\Sigma Q = \Sigma W$$

$$W_{net} = Q_H - Q_L$$

$$Q_L = Q_H - W_{net}$$

$$Q_L = 1500 - 600$$

$$\boxed{Q_L = 900\ \text{MW}}$$

Step 4 of 4

The actual amount of heat rejected to the river water will be less than the obtained value because there will be heat losses in the piping or the heat loss to the surroundings etc.

Question: 7.019P

A heat engine that pumps water out of an underground mine accepts 700 kJ of heat and produces 250 kJ of work. How much heat does it reject, in kJ?

Step 1 of 1

Given

The heat input, $Q_H = 700\ \text{kJ}$

Net work, $W_{net} = 250\ \text{kJ}$

From thermodynamic relation, $W_{net} = Q_H - Q_L$

Therefore the heat rejected is $Q_L = Q_H - W_{net}$

$$Q_L = 700 - 250$$

$$= 450\ \text{kJ}$$

Hence the total heat rejected is $\boxed{Q_L = 450\ \text{kJ}}$

Question:

7.020P

A heat engine with a thermal efficiency of 40 percent rejects 1000 kJ/kg of heat. How much heat does it receive?

Step 1 of 3

Given

Thermal efficiency of heat engine is $\eta = 40\%$

Amount of heat rejection $q_L = 1000\,\text{kJ/kg}$

Step 2 of 3

For heat engines, the desired output is the net work output, and the required input is the amount of heat supplied to the working fluid. Then the thermal efficiency of a heat engine can be expressed as

$$\text{Thermal efficiency} = \frac{\text{Net work output}}{\text{Total heat input}}$$

Question:

A heat engine with a thermal efficiency of 40 percent rejects 1000 kJ/kg of heat. How much heat does it receive?

Step 1 of 3

Given

Thermal efficiency of heat engine is $\eta = 40\%$

Amount of heat rejection $q_L = 1000\,\text{kJ/kg}$

Step 2 of 3

$q_H =$ [illegible] kJ/kg

Question: 7.021P

A steam power plant with a power output of 150 MW consumes coal at a rate of 60 tons/h. If the heating value of the coal is 30,000 kJ/kg, determine the overall efficiency of this plant.

Step 1 of 3

Given

Power output $\dot{W}_{net} = 150\ \text{MW}$
Heating value of the coal $HV = 30,000\ \text{kJ/kg}$

Mass flow rate of coal $\dot{m}_{coal} = 60\ \text{tons/h}$

$$\dot{m}_{coal} = 60 \times 1000 \times \frac{1}{3600}\ \text{kg/s}$$

Step 2 of 3

The rate of heat supplied to the power plant

$$\dot{Q}_{in} = \dot{m}_{coal} \times HV$$

$$\dot{Q}_{in} = 60 \times 1000 \times \frac{1}{3600} \times 30,000$$

$$\dot{Q}_{in} = 500000\ \text{kW}$$

$$\dot{Q}_{in} = 500\ \text{MW}$$

Step 3 of 3

The overall efficiency of the plant is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$\eta_{th} = \frac{150}{500}$$

$$= 0.3$$

$$\boxed{\eta_{th} = 30\%}$$

7.022P

Question:

An automobile engine consumes fuel at a rate of 22 L/h and delivers 55 kW of power to the wheels. If the fuel has a heating value of 44,000 kJ/kg and a density of 0.8 g/cm3, determine the efficiency of this engine.

Step 1 of 3

Given

Power output, $\dot{W}_{net} = 55\ \text{kW}$

Heating value, $HV = 44{,}000\ \text{kJ/kg}$

Volume flow rate of fuel, $\dot{V}_{fuel} = 22\ \text{L/h}$

Density of fuel, $\rho = 0.8\ \text{g/cm}^3$

Step 2 of 3

The mass flow rate of fuel is expressed as

$$\dot{m}_{fuel} = \dot{V}_{fuel} \times \rho$$

Question:

An automobile engine consumes fuel at a rate of 22 L/h and delivers 55 kW of power to the wheels. If the fuel has a heating value of 44,000 kJ/kg and a density of 0.8 g/cm3, determine the efficiency of this engine.

Step 1 of 3

Given

Power output, $\dot{W}_{net} = 55\ \text{kW}$

Heating value, $HV = 44{,}000\ \text{kJ/kg}$

Volume flow rate of fuel, $\dot{V}_{fuel} = 22\ \text{L/h}$

Density of fuel, $\rho = 0.8\ \text{g/cm}^3$

7.023P

Question:

In 2001, the United States produced 51 percent of its electricity in the amount of 1.878 × 1012 kWh from coal-fired power plants. Taking the average thermal efficiency to be 34 percent, determine the amount of thermal energy rejected by the coal-fired power plants in the United States that year.

Step 1 of 3

Given

The amount of power output by means of coal fired power plants is

$\dot{W}_{net} = 1.878 \times 10^{12}\ \text{kWh}$

Average thermal efficiency is $\eta = 34\%$

Step 2 of 3

The overall efficiency of the plant is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{}}$$

Question:

In 2001, the United States produced 51 percent of its electricity in the amount of 1.878 × 1012 kWh from coal-fired power plants. Taking the average thermal efficiency to be 34 percent, determine the amount of thermal energy rejected by the coal-fired power plants in the United States that year.

Step 1 of 3

Given

The amount of power output by means of coal fired power plants is

$\dot{W}_{net} = 1.878 \times 10^{12}\ \text{kWh}$

Question: 7.024P

The Department of Energy projects that between the years 1995 and 2010, the United States will need to build new power plants to generate an additional 150,000 MW of electricity to meet the increasing demand for electric power. One possibility is to build coal-fired power plants, which cost $1300 per kW to construct and have an effi- ciency of 40 percent. Another possibility is to use the clean-burning Integrated Gasification Combined Cycle (IGCC) plants where the coal is subjected to heat and pressure to gasify it while removing sulfur and particulate matter from it. The gaseous coal is then burned in a gas turbine, and part of the waste heat from the exhaust gases is recovered to generate steam for the steam turbine. Currently the construction of IGCC plants costs about $ 1500 per kW but their efficiency is about 48 percent. The average heating value of the coal is about 28,000,000 kJ per ton (that is, 28,000,000 kJ of heat is released when 1 ton of coal is burned). If the IGCC plant is to recover its cost dif- ference from fuel savings in five years, determine what the price of coal should be in $ per ton.

Step 1 of 5

Given

The construction cost for coal power plant is $1300/kW

Efficiency of coal based power plant, $\eta_{coal} = 0.4$

The construction cost for IGCC plant, $1500/kW

Efficiency of IGCC plant, $\eta_{IGCC} = 0.48$

The heating value of coal, $HV = 28{,}000{,}000$ kJ/ton

Step 2 of 5

The amount of electricity generated by a 150,000 MW plant that produces of electricity in 5 years is

$$E_{elec} = (150{,}000\times10^3\ \text{kW})\times(5\times365\times24)$$
$$= 6.57\times10^{12}\ \text{kWh}$$

The efficiency of any power plant is defined as

$$\eta = \frac{E_{elec}}{Q_{in}}$$
$$= \frac{E_{elec}}{\dot{m}_{fuel}\times HV}$$

Hence the mass of fuel is $\dot{m}_{fuel} = \frac{E_{elec}}{\eta\times HV}$

Step 3 of 5

For coal power plant,

$$\dot{m}_{coal} = \frac{E_{elec}}{\eta_{coal}\times HV}$$
$$= \frac{6.57\times10^{12}}{0.4\times\left(28\times10^6\ \text{kJ/ton}\right)}\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right)$$
$$= 2.112\times10^9\ \text{tons}$$

For IGCC power plant,

$$\dot{m}_{IGCC} = \frac{E_{elec}}{\eta_{IGCC}\times HV}$$
$$= \frac{6.57\times10^{12}}{0.48\times\left(28\times10^6\ \text{kJ/ton}\right)}\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right)$$
$$= 1.759\times10^9\ \text{tons}$$

The mass difference between plants is

$$\Delta m = m_{coal} - m_{IGCC}$$
$$= \left(2.112\times10^9\right) - \left(1.759\times10^9\right)$$
$$= 0.352\times10^9\ \text{tons}$$

Step 4 of 5

The construction cost difference between two plants

Construction cost for coal power plant is

$$\text{Cost}_{coal} = (150{,}000\times10^3\ \text{kW})\times\$1300/\text{kW}$$
$$= \$1.95\times10^{11}$$

Construction cost for IGCC power plant is

$$\text{Cost}_{IGCC} = (150{,}000\times10^3\ \text{kW})\times\$1500/\text{kW}$$
$$= \$2.25\times10^{11}$$

Cost difference between two power plants is

$$\Delta\text{cost} = \$(2.25-1.95)\times10^{11}$$
$$= \$0.3\times10^{11}$$

Step 5 of 5

The unit cost of coal per ton used in the power plant is

$$\text{unit cost} = \frac{\Delta\text{cost}}{\Delta m_{coal}}$$
$$= \frac{0.3\times10^{11}}{0.352\times10^9}$$
$$= \$85.2/\text{ton}$$

So IGCC power plant is cost effective if the coal price is $85.2 per ton

Question:

Reconsider Prob. 7–24. Using EES (or other) software, investigate the price of coal for varying simple payback periods, plant construction costs, and operating efficiency.

Step 1 of 7

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 7

Equation window:

```
"Given"
W_dot=15E7 [kW]
Cost_coal=1300 [$/kW]
eta_coal=0.40
Cost_IGCC=1500 [$/kW]
eta_IGCC=0.48
HV_coal=28000 [kJ/kg]
"PaybackYears=5 [yr]"

"Analysis"
time=PaybackYears*Convert(yr, h)
ConstCost_coal=W_dot*Cost_coal
ConstCost_IGCC=W_dot*Cost_IGCC
ConstCostDif=ConstCost_IGCC-ConstCost_coal
W_e=W_dot*time
m_coal_coal=W_e/(eta_coal*HV_coal)*Convert(kWh, kJ)
m_coal_IGCC=W_e/(eta_IGCC*HV_coal)*Convert(kWh, kJ)
DELTAm_coal=m_coal_coal-m_coal_IGCC
UnitCost_coal=ConstCostDif/DELTAm_coal*1000
```

Step 3 of 7

Table for unit cost of coal as a function of payback periods:

1..10	1 PaybackYears [yr]	2 $UnitCost_{coal}$
Run 1	1	426.2
Run 2	2	213.1
Run 3	3	142.1
Run 4	4	106.5
Run 5	5	85.24
Run 6	6	71.03
Run 7	7	60.88
Run 8	8	53.27
Run 9	9	47.35
Run 10	10	42.62

Step 4 of 7

Graph between the unit cost of coal and payback periods:

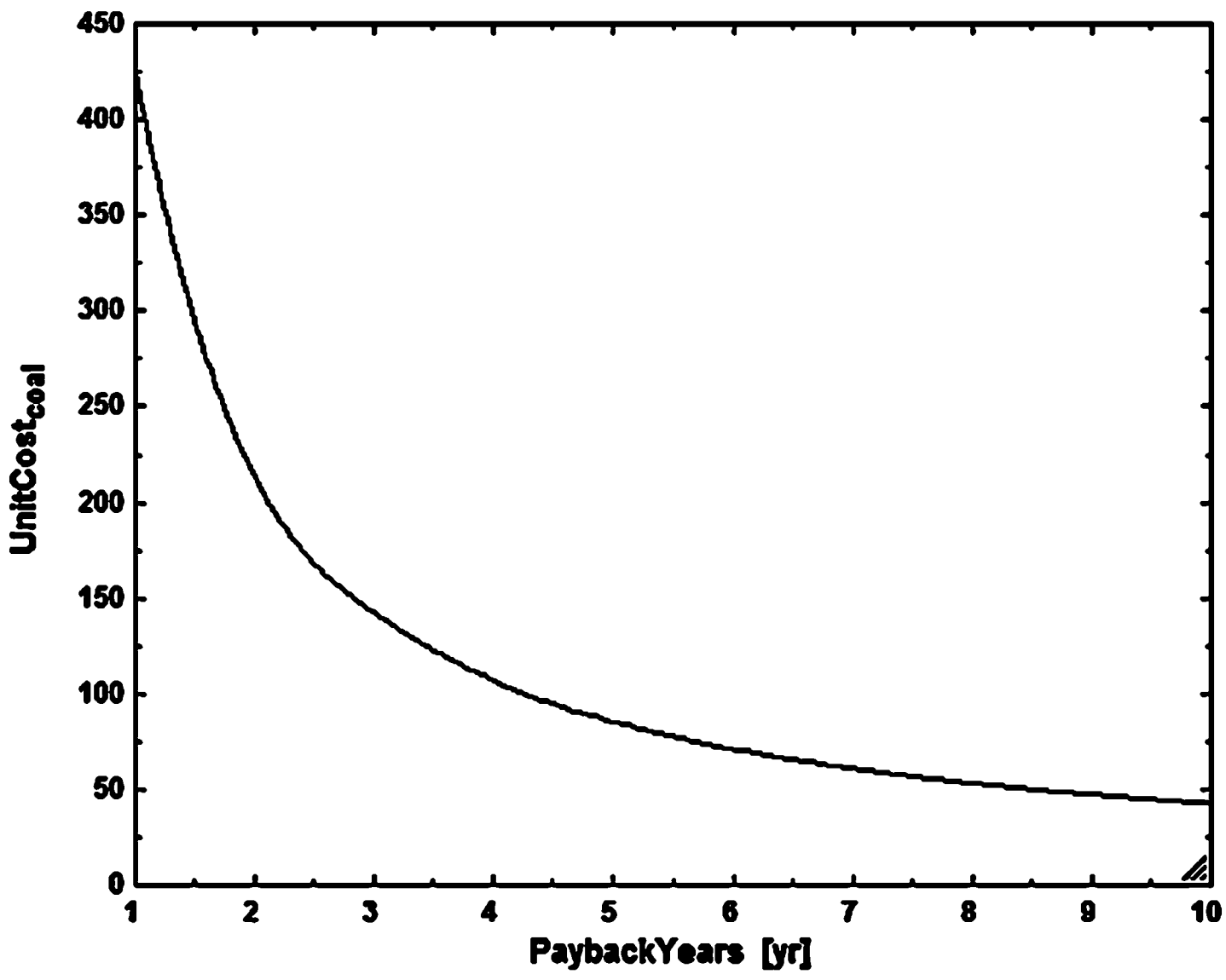


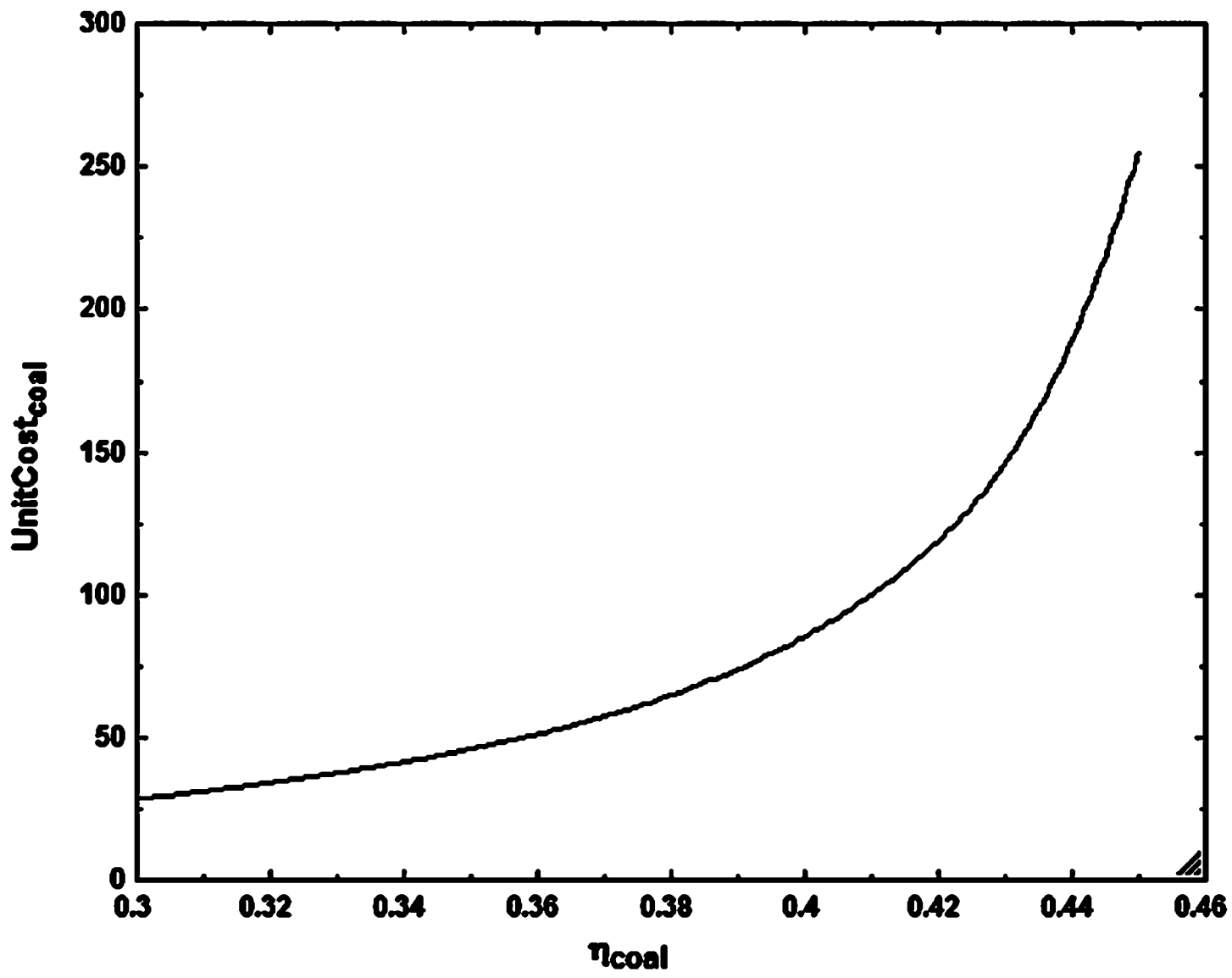
Table for unit cost of coal as a function of efficiency of coal

1..10	1 η_{coal}	2 $UnitCost_{coal}$
Run 1	0.3	28.41
Run 2	0.3167	33.05
Run 3	0.3333	38.74
Run 4	0.35	45.9
Run 5	0.3667	55.15
Run 6	0.3833	67.6
Run 7	0.4	85.24
Run 8	0.4167	112.2

Run 5	0.3667	59.15
Run 6	0.3833	67.6
Run 7	0.4	85.24
Run 8	0.4167	112.2
Run 9	0.4333	158.3
Run 10	0.45	255.7

Step 5 of 7

Graph between the unit cost of coal and efficiency of coal:



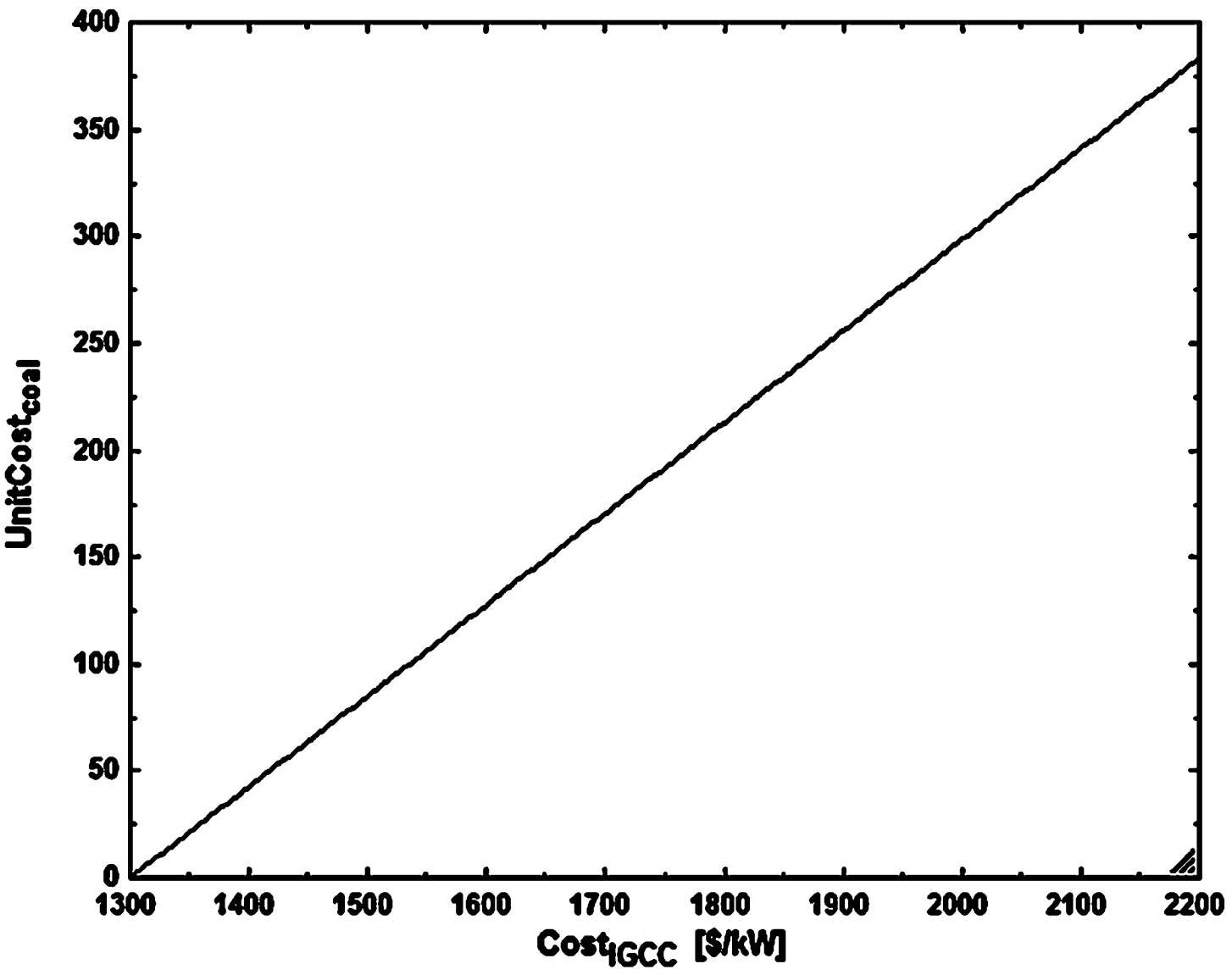
Step 6 of 7

Table for unit cost of coal as a function of constructive coal:

1..10	1 Cost_IGCC [$/kW]	2 UnitCost_coal
Run 1	1300	0
Run 2	1400	42.62
Run 3	1500	85.24
Run 4	1600	127.9
Run 5	1700	170.5
Run 6	1800	213.1
Run 7	1900	255.7
Run 8	2000	298.3
Run 9	2100	340.9
Run 10	2200	383.6

Step 7 of 7

Graph between the unit cost of coal and constrictive cost :



Question: 7.026P

Repeat Prob. 7–24 for a simple payback period of three years instead of five years.

Step 1 of 5

Given
The construction cost for coal power plant is \$1300/kW
Efficiency of coal based power plant, $\eta_{coal} = 0.4$
The construction cost for IGCC plant is \$1500/kW
Efficiency of IGCC plant, $\eta_{IGCC} = 0.48$
The heating value of coal, $HV = 28{,}000{,}000$ kJ/ton

Step 2 of 5

The amount of electricity generated by any of the plant for 3 years in order to produced 150,000 MW of electricity is

$$E_{elec} = (150{,}000 \times 10^3 \text{ kW}) \times (3 \times 365 \times 24)$$
$$= 3.942 \times 10^{12} \text{ kWh}$$

The mass required for this energy is $\dot{m}_{fuel} = \frac{Q_{in}}{HV}$

The efficiency of any power plant is defined as $\eta = \frac{E_{elec}}{Q_{in}}$

Combining both the expression, we get $\dot{m}_{fuel} = \frac{E_{elec}}{\eta \times HV}$

Step 3 of 5

For coal power plant,

$$\dot{m}_{coal} = \frac{E_{elec}}{\eta_{coal} \times HV}$$
$$= \frac{(3.942 \times 10^{12})}{(0.4) \times (28 \times 10^6)} \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}}\right)$$
$$= 1.267 \times 10^9 \text{ tons}$$

For IGCC power plant,

$$\dot{m}_{IGCC} = \frac{E_{elec}}{\eta_{IGCC} \times HV}$$
$$= \frac{(3.942 \times 10^{12})}{(0.48) \times (28 \times 10^6)} \left(\frac{3600 \text{ kJ}}{1 \text{ kWh}}\right)$$
$$= 1.055 \times 10^9 \text{ tons}$$

The mass difference between plants is

$$\Delta m = m_{coal} - m_{IGCC}$$
$$= (1.267 \times 10^9) - (1.055 \times 10^9)$$
$$= 0.211 \times 10^9$$

Step 4 of 5

The construction cost difference between two plants
Construction cost for coal power plant is

$$\text{Cost}_{coal} = (150{,}000 \times 10^3 \text{ kW}) \times (\$1300/\text{kW})$$
$$= \$1.95 \times 10^{11}$$

Construction cost for IGCC power plant is

$$\text{Cost}_{IGCC} = (150{,}000 \times 10^3 \text{ kW}) \times (\$1500/\text{kW})$$
$$= \$2.25 \times 10^{11}$$

Cost difference between two power plants is

$$\Delta\text{cost} = \$(2.25 - 1.95) \times 10^{11}$$
$$= \$0.3 \times 10^{11}$$

Step 5 of 5

The unit cost of coal per ton used in the power plant is

$$\text{unit cost} = \frac{\Delta\text{cost}}{\Delta m}$$
$$= \frac{0.3 \times 10^{11}}{0.211 \times 10^9}$$
$$= \$142.2/\text{ton}$$

So IGCC power plant is cost effective if the coal price is \$142.2/ton

Question: 7.027P

Solar energy stored in large bodies of water, called solar pounds, is being used to generate electricity. If such a solar power plant has an efficiency of 4 percent and a net power output of 350 kW, determine the average value of the required solar energy collection rate, in Btu/h.

Step 1 of 3

Given

Efficiency of solar power plant $\eta = 4\% = 0.04$

Net power output $\dot{W}_{net} = 350\,\text{kW}$

Step 2 of 3

The rate of solar energy collection rate can be determined from the thermal efficiency relation

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$\dot{Q}_{in} = \frac{\dot{W}_{net}}{\eta_{th}}$$

$$\dot{Q}_{in} = \frac{350}{0.04}$$

$$\dot{Q}_{in} = 8750\ \text{kW}$$

Step 3 of 3

We know $1\,\text{Btu} = 1.055\,\text{kJ}$

$$\dot{Q}_{in} = 8750\,\frac{\text{kJ}}{\text{s}}\left(\frac{1\,\text{Btu}}{1.055\,\text{kJ}}\right)\left(\frac{3600\,\text{s}}{1\,\text{h}}\right)$$

$$\boxed{\dot{Q}_{in} = 29.8\times10^{6}\ \text{Btu/h}}$$

Question: 7.028P

A coal-burning steam power plant produces a net power of 300 MW with an overall thermal efficiency of 321 percent. The actual gravimetric air-fuel ratio in the furnace is calculated to be 12 kg air/kg fuel. The heating value of the coal is 28,000 kJ/kg. Determine (*a*) the amount of coal consumed during a 24-hour period and (*b*) the rate of air flowing through the furnace.

Step 1 of 5

Given data:

Net power produced by the plant, $\dot{W}_{net} = 300\ \text{MW}$

Thermal efficiency, $\eta_{th} = 32\%$

The air fuel ratio is, $A/F = 12\ \text{kg air/kg fuel}$

Heating value of coal $HV = 28,000\ \text{kJ/kg}$

Step 2 of 5

(a) The coal consumed in a 24 hour period

The amount of heat input to the power plant is calculated by

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$\dot{Q}_{in} = \frac{300}{0.32}$$

$$\dot{Q}_{in} = 937.5\ \text{MW}$$

$$\dot{Q}_{in} = 937.5 \times 1000\ \text{kW}$$

Step 3 of 5

The rate of heat supplied to the power plant is also expressed as

$$\dot{Q}_{in} = \dot{m}_f \times HV$$

Here the fuel is coal

Therefore, $\dot{m}_{coal} = \frac{\dot{Q}_{in}}{HV}$

$$= \frac{937.5 \times 1000\ \text{kJ/s}}{28000\ \text{kJ/kg}}$$

$$= 33.48\ \text{kg/s}$$

Step 4 of 5

The coal consumed in a 24 hour period

$$m_{coal} = 33.48 \times 24 \times 3600$$

$$= 2.89 \times 10^6\ \text{kg}$$

∴ The amount of coal consumed is, $\boxed{m_{coal} = 2.89 \times 10^6\ \text{kg}}$

Step 5 of 5

(b) The amount of air flowing through the furnace

$$A/F = \frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

$$\dot{m}_{air} = A/F \times \dot{m}_{fuel}$$

$$\dot{m}_{air} = 12 \times 33.48$$

$$\dot{m}_{air} = 401.76\ \text{kg/s}$$

∴ The mass of air flowing, $\boxed{\dot{m}_{air} = 401.76\ \text{kg/s}}$

Question: 7.029P

What is the difference between a refrigerator and a heat pump?

Step 1 of 1

Both the refrigerator and the heat pumps carry the heat from low temperature reservoir or the sink to the high temperature reservoir or source. The main purpose of the refrigerator is to chill or cool or freeze the substance, by maintaining very low temperature in the freezer. It absorbs the heat from the evaporator and throws it to the atmosphere which is at high temperature.

The objective of a heat pump, however, is to maintain a heated space at a high temperature. The main purpose of the heat pump is to heat the room.

Question: 7.030P

What is the difference between a refrigerator and an air conditioner?

Step 1 of 1

Air conditioners are basically refrigerators whose refrigerated space is a room or a building instead of the food compartment. A window air conditioning unit cools a room by absorbing heat from the room air and discharging it to the outside.
A refrigerator cools the space where food or other materials are placed.

Question: 7.031P

In a refrigerator, heat is transferred from a lower-temperature medium (the refrigerated space) to a higher-temperature one (the kitchen air). Is this a violation of the second law of thermodynamics? Explain.

Step 1 of 1

According to clausius statement it is impossible to transfer heat from low temperature to high temperature without an input of work. So in refrigerator heat is transferred from low temperature to high temperature by consuming some work input.
Therefore refrigerator does not violate the second law of thermodynamics.

Question: 7.032P

A heat pump is a device that absorbs energy from the cold outdoor air and transfers it to the warmer indoors. Is this a violation of the second law of thermodynamics? Explain.

Step 1 of 1

According to clausius statement it is impossible to transfer heat from low temperature to high temperature without an input of work. So in heat pump heat is transferred from low temperature to high temperature by consuming some work input.
Therefore heat pump does not violate the second law of thermodynamics.

Question: 7.033P

Define the coefficient of performance of a refrigerator in words. Can it be greater than unity?

Step 1 of 1

The COP (coefficient of performance) of refrigerator is defined as the ratio of amount of heat removed from the refrigerated space to the work input.

$$COP_R = \frac{Q_L}{W_{net,in}}$$

COP of refrigerator is greater than one

Question: 7.034P

Define the coefficient of performance of a heat pump in words. Can it be greater than unity?

Step 1 of 1

The COP (coefficient of performance) of heat pump is defined as the ratio of amount of heat supplied to the work input.

$$COP_{HP} = \frac{Q_H}{W_{net,in}}$$

COP of heat pump is greater than one.

Question: 7.035P

A heat pump that is used to heat a house has a COP of 2.5. That is, the heat pump delivers 2.5 kWh of energy to the house for each 1 kWh of electricity it consumes. Is this a violation of the first law of thermodynamics? Explain.

Step 1 of 2

Given

COP of heat pump $COP_{HP} = 2.5$

Heat supplied to the house $Q_H = 2.5\text{kWh}$

Work input to the heat pump $W_{net,in} = 1\text{kWh}$

Step 2 of 2

COP of heat pump is

$$COP_{HP} = \frac{Q_H}{W_{net,in}}$$

$$COP_{HP} = \frac{2.5}{1}$$

$$COP_{HP} = 2.5$$

Therefore the given COP satisfies

Here heat transfer takes place from low energy to high energy but the energy is neither created nor destroyed. So it does not violate the first law of thermodynamics

Question: 7.036P

A refrigerator has a COP of 1.5. That is, the refrigerator removes 1.5 kWh of energy from the refrigerated space for each 1 kWh of electricity it consumes. Is this a violation of the first law of thermodynamics? Explain.

Step 1 of 3

Given

COP of refrigerator $COP_R = 1.5$

The amount of heat removed from the refrigerated space $\dot{Q}_L = 1.5\text{kWh}$

Work input to the refrigerator $\dot{W}_{net,in} = 1\text{kWh}$

Step 2 of 3

COP of refrigerator is

$$COP_R = \frac{Q_L}{W_{net,in}}$$

$$COP_R = \frac{1.5}{1}$$

$$COP_R = 1.5$$

Therefore the given COP satisfies

Step 3 of 3

Here heat transfer takes place from low energy to high energy but the energy is neither created nor destroyed. So it does not violate the first law of thermodynamics

Question: 7.037P

What is the Clausius expression of the second law of thermodynamics?

Step 1 of 2

The Clausius statement.

It is not possible to construct a thermodynamic system undergoing a cyclic process undergoes the heat transfer from a lower temperature body to a higher temperature body without any work interaction.

Step 2 of 2

The Clausius expression.

$$\oint \frac{\delta Q}{T} \leq 0$$

Here, δQ is the energy flow into the system and T is the absolute temperature.

For a cyclic reversible process.

$$\oint \frac{\delta Q}{T} = 0$$

For a cyclic irreversible process.

$$\oint \frac{\delta Q}{T} < 0$$

Question: 7.038P

Show that the Kelvin-Planck and the Clausius expressions of the second law are equivalent.

Step 1 of 1

The Kelvin-Planck and Clausius expressions of the second law of thermodynamics are equivalent in their consequences which means that if any device violates one statement then consequently the other statement is also violated.

Consider two devices *A* and *B* which are working between a high-temperature reservoir and a low-temperature reservoir. Let device *A* be a heat engine and is assumed to transfer heat $(Q_H)_A$ completely into work $(W_{net})_A$ by which it has 100% efficiency, thus violating Kelvin-Planck statement. Then the amount of heat transfer by device *A* is equal to the net work by the device *A*. Mathematically it is given by the relation as follows:

$$(Q_H)_A = (W_{net})_A$$

Consider device *B* to be a refrigerator which removes heat $(Q_L)_B$ from the low-temperature reservoir and also receives work $(W_{net})_A$ from device *A* and rejects the total energy of $(W_{net})_A$ and $(Q_L)_B$ to the high-temperature reservoir.

Heat rejected by device *C*, which is the combination of devices A and B, to the high-temperature reservoir equals

$$\begin{aligned}\left((Q_L)_B + (W_{net})_A\right) - (Q_H)_A &= \left((Q_L)_B + (W_{net})_A\right) - (W_{net})_A \\ &= (Q_L)_B\end{aligned}$$

Thus device *C* extracts $(Q_L)_B$ amount of heat from the low-temperature reservoir and rejects it to the high-temperature reservoir which results in violation of the Clausius statement. Similarly if any device violates Kelvin-Planck statement Clausius statement is also violated.

Therefore, the Kelvin-Planck and the Clausius expressions of the second law are equivalent.

Question: 7.039P

A residential heat pump has a coefficient of performance of 2.4. How much heating effect, in Btu/h, will result when 5 hp is supplied to this heat pump?

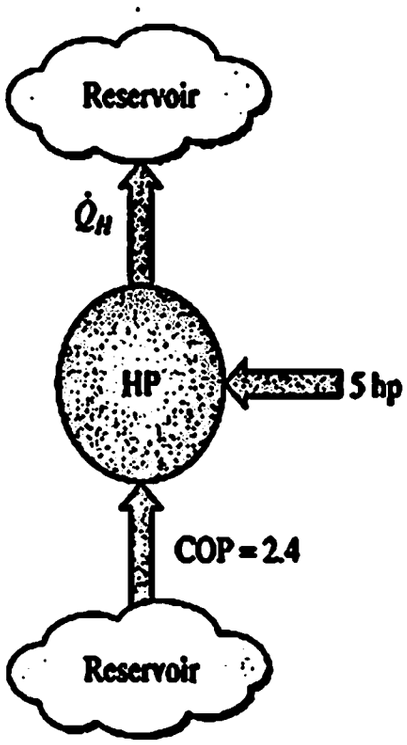


FIGURE P7–39E

Step 1 of 2

Given

The coefficient of performance of the heat pump is COP = 2.4

The power input to the heat pump $\dot{W}_{net} = 5$ hp

Step 2 of 2

The measure of performance of a heat pump is also expressed in terms of the coefficient of performance defined as

$$COP_{HP} = \frac{\text{Desired effect}}{\text{Work input}}$$

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

Therefore the amount of heating effect is

$$\dot{Q}_H = COP_{HP} \times \dot{W}_{net}$$
$$= 2.4 \times 5$$
$$\dot{Q}_H = 12 \text{ hp}$$

We know 1 hp = 2544.5 Btu/h

$$\dot{Q}_H = 12 \times 2544.5 \text{ Btu/h}$$

$$\boxed{\dot{Q}_H = 30534 \text{ Btu/h}}$$

Question: 7.040P

An air conditioner produces a 2-kW cooling effect while rejecting 2.5 kW of heat. What is its COP?

Step 1 of 3

Given

The cooling effect is $\dot{Q}_L = 2\,\text{kW}$

The amount of heat rejected $\dot{Q}_H = 2.5\,\text{kW}$

Step 2 of 3

Applying first law of thermodynamics for this system

$$\dot{W}_{net} = \dot{Q}_H - \dot{Q}_L$$

$$\dot{W}_{net} = 2.5 - 2$$

$$\dot{W}_{net} = 0.5\,\text{kW}$$

Step 3 of 3

The coefficient of performance of refrigerator is defined as desired effect (to remove heat (Q_L) from the refrigerated space) to the work input

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$COP_R = \frac{2}{0.5}$$

$$\boxed{COP_R = 4}$$

Question: 7.041P

A refrigerator used to cool a computer requires 3 kW of electrical power and has a COP of 1.4. Calculate the cooling effect of this refrigerator, in kW.

Step 1 of 2

Given

The COP of refrigerator $COP_R = 1.4$

The amount of power input $\dot{W}_{net} = 3\ \text{kW}$

Step 2 of 2

The coefficient of performance of refrigerator is defined as desired effect (to remove heat (Q_L) from the refrigerated space) to the work input

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

Therefore the cooling effect of the refrigerator is

$$\dot{Q}_L = COP_R \times \dot{W}_{net}$$

$$\dot{Q}_L = 1.4 \times 3$$

$$\boxed{\dot{Q}_L = 4.2\ \text{kW}}$$

Question: 7.042P

A food department is kept at −12°C by a refrigerator in an environment at 30°C. The total heat gain to the food department is estimated to be 3300 kJ/h and the heat rejection in the condenser is 4800 kJ/h. Determine the power input to the compressor, in kW and the COP of the refrigerator.

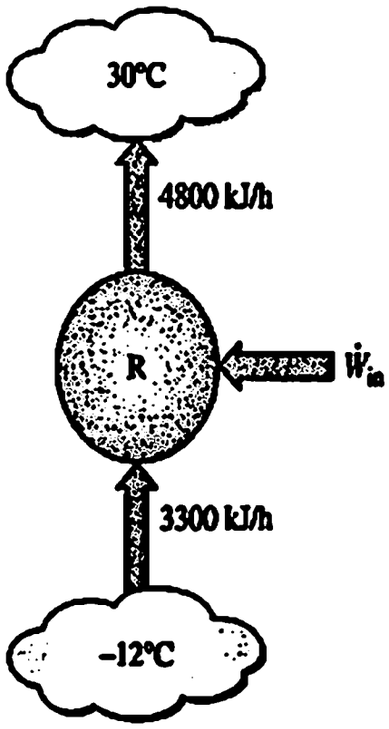


FIGURE P7–42

Step 1 of 3

Given

The cooling effect is $\dot{Q}_L = 3300$ kJ/h

The amount of heat rejected $\dot{Q}_H = 4800$ kJ/h

Step 2 of 3

Applying first law of thermodynamics for this system

$$\dot{W}_{net} = \dot{Q}_H - \dot{Q}_L$$
$$= 4800 - 3300 = 1500 \text{ kJ/h}$$
$$\dot{W}_{net} = 1500 \text{kJ/h}$$

Step 3 of 3

The coefficient of performance of refrigerator is defined as desired effect (to remove heat (Q_L) from the refrigerated space) to the work input

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$COP_R = \frac{3300}{1500}$$

$$\boxed{COP_R = 2.2}$$

Question: 7.043P

A household refrigerator with a COP of 1.2 removes heat from the refrigerated space at a rate of 60 kJ/min. Determine (*a*) the electric power consumed by the refrigerator and (*b*) the rate of heat transfer to the kitchen air.

Step 1 of 3

Given data:

Coefficient of performance $COP = 1.2$

Heat removed from refrigerant $\dot{Q}_L = 60$ kJ/min

Step 2 of 3

a)

The electric power consumed by the refrigerator

The COP of a refrigerator is given by

$$COP = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$1.2 = \frac{60}{\dot{W}_{net}}$$

$$\dot{W}_{net} = \frac{60}{1.2}$$

$$\dot{W}_{net} = 50 \text{ kJ/min}$$

$$\dot{W}_{net} = \frac{50}{60} \text{kJ/s}$$

$$\dot{W}_{net} = 0.833 \text{kJ/s} (\text{or}) \text{kW}$$

∴ The electric power consumed is $\boxed{\dot{W}_{net} = 0.833 \text{kW}}$

Step 3 of 3

b)

The rate of heat transfer to the kitchen is calculated from

$$\dot{W}_{net} = \dot{Q}_H - \dot{Q}_L$$

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{net}$$

$$\dot{Q}_H = (60+50) \text{ kJ/min}$$

$$\dot{Q}_H = 110 \text{ kJ/min}$$

∴ The heat transfer to the kitchen is $\boxed{\dot{Q}_H = 110 \text{ kJ/min}}$

Question: 7.044P

A commercial heat pump removes 10,000 Btu/h from the source, rejects 15,090 Btu/h to the sink, and requires 2 hp of power. What is this heat pump's coefficient of performance?

Step 1 of 2

Given data:

Heat pump absorbs heat at a rate of $\dot{Q}_L = 10000\,\text{Btu/h}$

Heat pump removes heat at a rate of $\dot{Q}_H = 15090\,\text{Btu/h}$

Power input to the heat pump, $W_{net,in} = 2\,\text{hp}$

Conversion factor $1\text{hp} = 2545.458\ \text{Btu/h}$

Step 2 of 2

The measure of performance of a heat pump is also expressed in terms of the coefficient of performance COP_{HP}, defined as

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_{HP} = \frac{\dot{Q}_H}{W_{net,in}}$$

$$COP_{HP} = \frac{15{,}090}{2 \times 2545.458}$$

$$COP_{HP} = 2.964$$

∴ The coefficient of performance of the heat pumps is $\boxed{COP_{HP} = 2.964}$

Question: 7.045P

A refrigerator used for cooling food in a grocery store is to produce a 25,000 kJ/h cooling effect, and it has a coefficient of performance of 1.60. How many kilowatts of power will this refrigerator require to operate?

Step 1 of 2

Given

The COP of refrigerator $COP_R = 1.6$

Cooling effect produced $\dot{Q}_L = 25,000$ kJ/h

Step 2 of 2

The coefficient of performance of refrigerator is defined as desired effect (to remove heat (Q_L) from the refrigerated space) to the work input

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

Therefore the amount of power required to the refrigerator is

$$\dot{W}_{net} = \frac{\dot{Q}_L}{COP_R}$$

$$= \frac{25000}{1.6}$$

$$\dot{W}_{net} = 15625 \text{ kJ/h}$$

$$\dot{W}_{net} = 15625 \times \frac{1}{3600}$$

$$\boxed{\dot{W}_{net} = 4.34 \text{ kW}}$$

Question: 7.046P

A household refrigerator that has a power input of 450 W and a COP of 2.5 is to cool five large watermelons, 10 kg each, to 8°C. If the watermelons are initially at 20°C, determine how long it will take for the refrigerator to cool them. The watermelons can be treated as water whose specific heat is 4.2 kJ/kg·°C. Is your answer realistic or optimistic? Explain.

Step 1 of 5

Given data:

The power input to the refrigerator, $W_{net} = 450\ \text{W}$

The COP of the refrigerator, $COP_R = 2.5$

Low temperature, $T_L = 8°\text{C}$

High temperature, $T_H = 20°\text{C}$

Mass of each watermelon, $m = 10\ \text{kg}$

Specific heat, $c = 4.2\ \text{kJ/kg°C}$

Step 2 of 5

The amount of heat removed from 5 watermelons is

$$Q_L = mc\Delta T\ \text{kJ}$$

$$Q_L = 5\times10\times4.2(20-8)\ \text{kJ}$$

$$Q_L = 2520\ \text{kJ}$$

Step 3 of 5

We know that

$$COP_R = \frac{\dot{Q}_L}{W_{net}}$$

$$2.5 = \frac{\dot{Q}_L}{450}$$

$$\dot{Q}_L = 1125\ \text{W}$$

$$\dot{Q}_L = 1.125\ \text{kW}$$

Step 4 of 5

Therefore, the time required to cool the watermelons is

$$t = \frac{Q_L}{\dot{Q}_L}$$

$$= \frac{2520}{1.125}$$

$$= 2240\ s$$

$$\therefore \boxed{t = 2240\ s}$$

Step 5 of 5

This answer seems to be not realistic since the refrigerated space will gain some heat during this process from the surrounding air, the refrigerator may take even more time to cool the watermelons.

Question: 7.047P

Bananas are to be cooled from 24 to 13°C at a rate of 215 kg/h by a refrigeration system. The power input to the refrigerator is 1.4 kW. Determine the rate of cooling, in kJ/min, and the COP of the refrigerator. The specific heat of banana above freezing is 3.35 kJ/kg·°C.

Step 1 of 3

Calculate the amount of heat removed from banana,

$$\dot{Q}_L = \dot{m}c_p\Delta T$$
$$= \dot{m}c_p(T_2 - T_1)$$

Here,

Initial temperature of banana, T_1

Final temperature of banana, T_2

The mass flow rate of refrigerant air, $\dot{m}$

The specific heat of banana, c_p

Substitute 24°C for T_1, 13°C for T_2, 215 kg/h for $\dot{m}$, 1.4 kW for $\dot{W}_{in}$, 3.35 kJ/kg·°C for c_p

$$\dot{Q}_L = 215\ \text{kg/h}\left(\frac{1\text{h}}{3600\text{s}}\right)\times 3.35\times(24-13)$$
$$= 2.2\ \text{kW}$$

Step 2 of 3

Calculate the rate of cooling in kJ/min,

$$\dot{Q}_L = 2.2\ \text{kW}\times\frac{1\text{J/s}}{1\text{W}}$$
$$= 2.2\ \text{kJ/s}\times\frac{60\text{s}}{\text{min}}$$
$$= 132\ \text{kJ/min}$$

Therefore, the rate of cooling is $\boxed{132\ \text{kJ/min}}$

Step 3 of 3

The coefficient of performance of refrigerator is defined as desired effect (to remove heat ($\dot{Q}_L$) from the refrigerated space) to the work input.

$$(\text{COP})_{\text{Refrigerator}} = \frac{\text{Desired effect}}{\text{work input}}$$
$$= \frac{\dot{Q}_L}{\dot{W}_{in}}$$

Here,

The Power input to the refrigerator, $\dot{W}_{in}$

Substitute 2.2 kW for $\dot{Q}_L$, 1.4 kW for $\dot{W}_{in}$,

$$(\text{COP})_{\text{Refrigerator}} = \frac{2.2}{1.4}$$
$$= 1.57$$

Therefore, the coefficient of performance of refrigerator is $\boxed{1.57}$

Question: 7.048P

A refrigerator is used to cool water from 23 to 5°C in a continuous manner. The heat rejected in the condenser is 570 kJ/min and the power is 2.65 kW. Determine the rate at which water is cooled, in L/min and the COP of the refrigerator. The specific heat of water is 4.18 kJ/kg·°C and its density is 1 kg/L.

Step 1 of 5

Given

Initial temperature of water $T_1 = 23°C$

Final temperature of water $T_2 = 5°C$

The amount of heat rejected is $\dot{Q}_H = 570$ kJ/min

$$\dot{Q}_H = \frac{570}{60} \text{ kJ/sec}$$

The power input to the refrigerator is $\dot{W}_{in} = 2.65$ kW

The specific heat of water is $c_p = 4.18$ kJ/kg.°C

Density of water is $\rho = 1$ kg/L

Step 2 of 5

The cooling effect can be determined from energy balance as

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{in}$$

$$\dot{Q}_L = (570/60) - 2.65$$

$$\dot{Q}_L = 6.85 \text{ kW}$$

Step 3 of 5

The mass flow rate of water can be determined from the amount of cooling effect produced by the water

$$\dot{Q}_L = \dot{m} c_p \Delta T \text{ kJ/s}$$

Therefore the mass flow rate is

$$\dot{m} = \frac{6.85}{(4.18)(23-5)}$$

$$\dot{m} = 0.091 \text{ kJ/s}$$

Step 4 of 5

The volume flow rate in L/min is

$$\dot{V} = \frac{\dot{m}}{\rho} \text{ L/min}$$

$$\dot{V} = \frac{0.091 \text{ kg/s}}{1 \text{ kg/L}} \left(\frac{60 \text{ s}}{1 \text{ min}} \right) \text{ L/min}$$

$$\boxed{\dot{V} = 5.46 \text{ L/min}}$$

Step 5 of 5

The coefficient of performance of refrigerator is defined as desired effect (to remove heat (Q_L) from the refrigerated space) to the work input

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

We know that

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$COP_R = \frac{6.85}{2.65}$$

$$\boxed{COP_R = 2.58}$$

Question: 7.049P

A heat pump is used to maintain a house at a constant temperature of 23°C. The house is losing heat to the outside air through the walls and the windows at a rate of 60,000 kJ/h while the energy generated within the house from people, lights, and appliances amounts to 4000 kJ/h. For a COP of 2.5, determine the required power input to the heat pump.

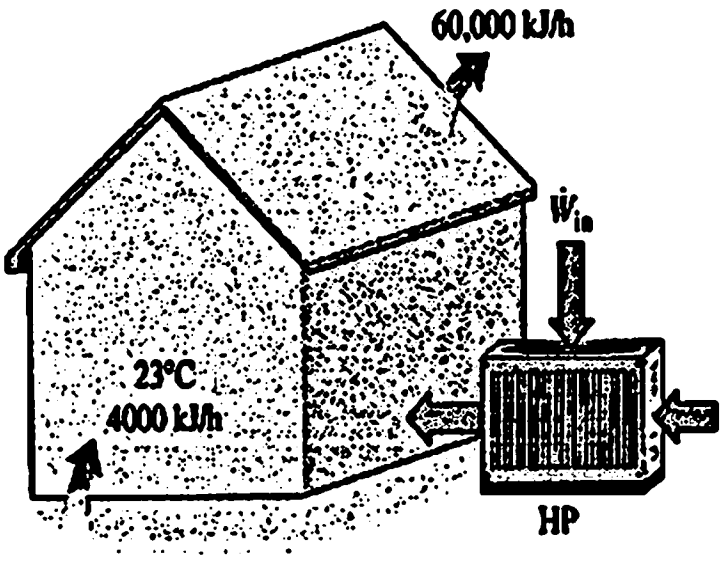


FIGURE P7–49

Step 1 of 3

Given data:

High temperature, $T_H = 23°C$

Heat loss from the house, $\dot{Q}_L = 60,000$ kJ/h

Heat absorbed by appliance, $\dot{Q}_{App} = 4000$ kJ/h

The coefficient of performance of heat pump, $COP_{HP} = 2.5$

Step 2 of 3

The total heat load by the heat pump is the difference between heat loss by the house and heat absorbed by the appliances

$$\dot{Q}_H = \dot{Q}_L - \dot{Q}_{App}$$

$$\dot{Q}_H = 60,000 - 4000$$

$$\dot{Q}_H = 56000 \text{ kJ/h}$$

Step 3 of 3

The measure of performance of a heat pump is also expressed in terms of the coefficient of performance COP_{HP} and it is defined as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

Therefore, the power input to the heat pump is

$$\dot{W}_{net} = \frac{\dot{Q}_H}{COP_{HP}}$$

$$\dot{W}_{net} = \frac{56000}{2.5 \times 3600}$$

$$\dot{W}_{net} = 6.22 \text{ kJ/s}$$

∴ The power input to the heat pump, $\boxed{\dot{W}_{net} = 6.22 \text{ kW}}$

Question: 7.050P

Water enters an ice machine at 55°F and leaves as ice at 25°F. If the COP of the ice machine is 2.4 during this operation, determine the required power input for an ice production rate of 28 lbm/h. (169 Btu of energy needs to be removed from each lbm of water at 55°F to turn it into ice at 25°F.)

Step 1 of 3

Given data:

COP of refrigerator, $COP_R = 2.4$

Low temperature, $T_L = 25°F$

High temperature, $T_H = 55°F$

Mass of ice produced, $\dot{m}_{ice} = 28$ lbm/h

Conversion factor, 1hp = 2545.458 Btu/h

Step 2 of 3

The cooling effect of the ice machine is defined as

$\dot{Q}_L = \dot{m}_{ice} \times 169$

$\dot{Q}_L = 28 \times 169$

$\dot{Q}_L = 4732$ Btu/h

We know that

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}}$$

$$2.4 = \frac{4732}{\dot{W}_{net,in}}$$

$\dot{W}_{net,in} = 1971.67$ Btu/h

Step 3 of 3

The power input for ice production is calculated in hp

$$\dot{W}_{net,in} = \frac{1971.67}{2545.458}$$

$\dot{W}_{net,in} = 0.7745$ hp

∴ The power input is, $\boxed{\dot{W}_{net,in} = 0.7745 \text{ hp}}$

Question: 7.051P

Consider an office room that is being cooled adequately by a 12,000 Btu/h window air conditioner. Now it is decided to convert this room into a computer room by installing several computers, terminals, and printers with a total rated power of 8.4 kW. The facility has several 7000 Btu/h air conditioners in storage that can be installed to meet the additional cooling requirements. Assuming a usage factor of 0.4 (i.e., only 40 percent of the rated power will be consumed at any given time) and additional occupancy of seven people, each generating heat at a rate of 100 W, determine how many of these air conditioners need to be installed to the room.

Step 1 of 5

Given

The cooling effect of window a/c $\dot{Q}_L = 12{,}000$ Btu/h

Power input of the computer and other related parts $\dot{W}_{in} = 8.4\,\text{kW}$

Number of people in the room N=7

Each person is generating heat at a rate of $\dot{Q}_p = 100\,\text{W}$

The cooling capacity of a/c $\dot{Q}_L = 7{,}000$ Btu/h

Usage factor 0.4

Step 2 of 5

The amount of heat dissipated by the computers

$$\dot{Q}_{comp} = \text{usage factor} \times \dot{W}_{in}$$

$$\dot{Q}_{comp} = 0.4 \times 8.4$$

$$\dot{Q}_{comp} = 3.36\,\text{kW}$$

Since we know

$1\,\text{W} = 3.412$ Btu/h

$$\dot{Q}_{comp} = 3.36 \times 1000 \times 3.412$$

$$\dot{Q}_{comp} = 11464.32 \text{ Btu/h}$$

Step 3 of 5

The total amount of heat generated by people in the room'

$$\dot{Q}_{people} = 7 \times 100$$

$$= 700\,\text{W}$$

$$\dot{Q}_{people} = 700 \times 3.412$$

$$\dot{Q}_{people} = 2388.4 \text{ Btu/h}$$

Step 4 of 5

Therefore the total heat generated in the room

$$\dot{Q}_{total} = \dot{Q}_{comp} + \dot{Q}_{people}$$

$$\dot{Q}_{total} = 11464.32 + 2388.4$$

$$\dot{Q}_{total} = 13852.7 \text{ Btu/h}$$

Step 5 of 5

The number of conditioners required for adequate cooling of the room is

$$n = \frac{\text{cooling load}}{\text{cooling capacity}}$$

$$n = \frac{13852.7}{7000}$$

$$n = 1.978 \approx 2$$

Therefore 2 air conditioners are required for the room

Question: 7.052P

Consider a building whose annual air-conditioning load is estimated to be 120,000 kWh in an area where the unit cost of electricity is \$0.10/kWh. Two air conditioners are considered for the building. Air conditioner A has a seasonal average COP of 3.2 and costs \$5500 to purchase and install. Air conditioner B has a seasonal average COP of 5.0 and costs \$7000 to purchase and install. All else being equal, determine which air conditioner is a better buy.

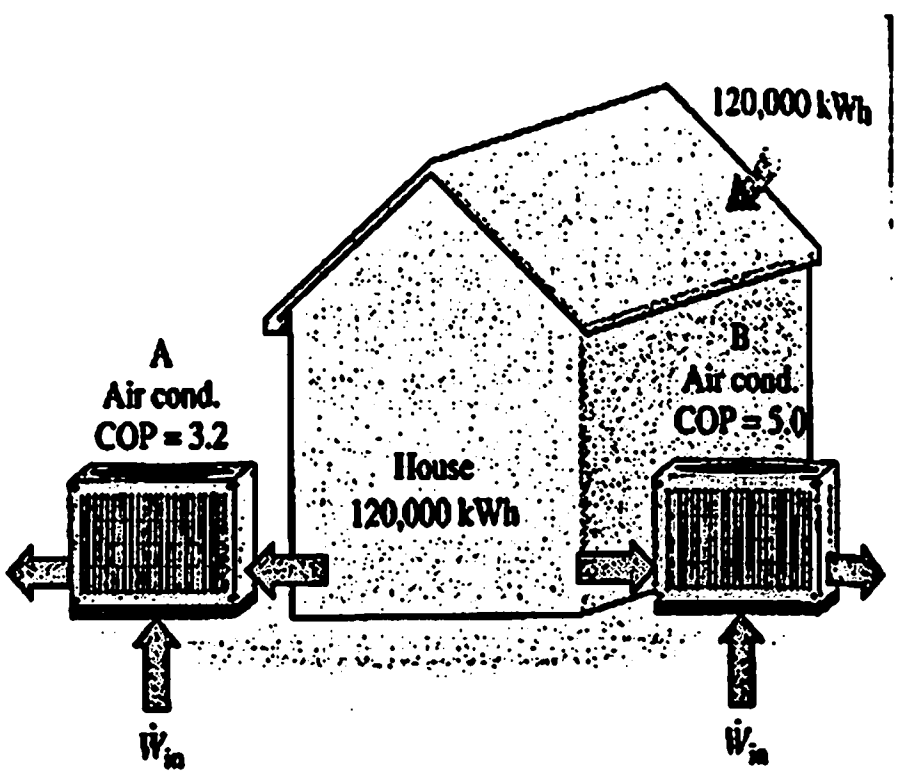


FIGURE P7–52

Step 1 of 4

Calculate the required power input to the Air conditioner *A*.

$$COP = \frac{N}{W}$$

Here, *COP* is the Coefficient Of Performance, *N* is the cooling load, and *W* is the work input.

Substitute 3.2 for COP_A, and $120000\ \text{kWh}$ for *N*.

$$COP_A = \frac{N}{W_A}$$

$$W_A = \frac{120000}{3.2}$$

$$= 37500\ \text{kWh}$$

Calculate the annual cost to the air conditioner *A*.

$$C_A = W_A \times c_u$$

Here, c_u is the unit cost of electricity.

Substitute $\$0.10/\text{kWh}$ for c_u, and $37500\ \text{kWh}$ for W_A.

$$C_A = W_A \times c_u$$

$$= 37500 \times 0.1$$

$$= \$3750$$

Step 2 of 4

Calculate the required power input to the Air conditioner *B*.

$$COP_B = \frac{N}{W_B}$$

Substitute 5 for COP_B, and $120000\ \text{kWh}$ for *N*.

$$COP_B = \frac{N}{W_B}$$

$$W_B = \frac{120000}{5}$$

$$= 24000\ \text{kWh}$$

Calculate the annual cost to the air conditioner *B*.

$$C_B = W_B \times c_u$$

Substitute $\$0.10/\text{kWh}$ for c_u, and $24000\ \text{kWh}$ for W_B.

$$C_B = W_B \times c_u$$

$$= 24000 \times 0.1$$

$$= \$2400$$

Step 3 of 4

Find the total cost of the air conditioner *A* per year:

$$T = C + I$$

Here, *T* is the total cost, *C* is annual service charges, and *I* is installation charges.

Substitute $\$3750$ for C_A, and $\$5500$ for I_A.

$$T_A = C_A + I_A$$

$$= 3750 + 5500$$

$$= \$9250$$

Step 4 of 4

Find the total cost of the air conditioner *B* per year:

$$T = C + I$$

Substitute $\$2400$ for C_B, and $\$7000$ for I_B.

$$T_B = C_B + I_B$$

$$= 2400 + 7000$$

$$= \$9400$$

Since, the total cost for air conditioner *A* is smaller than that of *B*, therefore the air conditioner *A* is better to buy.

Question: 7.053P

A house that was heated by electric resistance heaters consumed 1200 kWh of electric energy in a winter month. If this house were heated instead by a heat pump that has an average COP of 2.4, determine how much money the home owner would have saved that month. Assume a price of 8.50/kWh for electricity.

Step 1 of 3

Given data:

Heat transferred by the electric heater, $\dot{Q}_H = 1200\ \text{kWh}$

Average coefficient of performance, $COP_H = 2.4$

Step 2 of 3

The power consumed by the electrical heater is defined as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$\dot{W}_{net} = \frac{\dot{Q}_H}{COP_{HP}}$$

$$= \frac{1200}{2.4}$$

$$= 500\ \text{kWh}$$

Step 3 of 3

The heat transfer from the low temperature reservoir is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{net}$$

$$\dot{Q}_L = 1200 - 500$$

$$\dot{Q}_L = 700\ \text{kWh}$$

The amount of money saved is $Q_L \times 0.085$

Therefore amount of money saved is

$0.085 \times 700 = 59.5\$$

$\therefore$ The home owner can save $\boxed{59.5\$}$

Question: 7.054P

Refrigerant-134a enters the condenser of a residential heat pump at 800 kPa and 35°C at a rate of 0.018 kg/s and leaves at 800 kPa as a saturated liquid. If the compressor consumes 1.2 kW of power, determine (*a*) the COP of the heat pump and (*b*) the rate of heat absorption from the outside air.

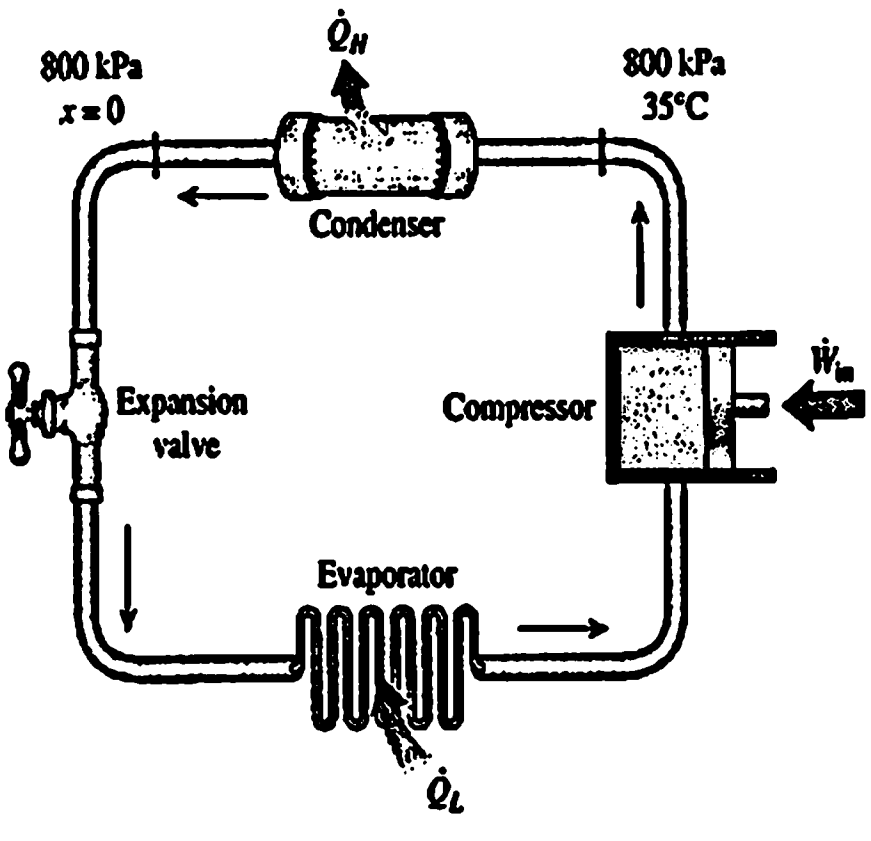


FIGURE P7–54

Step 1 of 6

Given data:

Inlet pressure of the condenser, $P_1 = 800\ \text{kPa}$

Inlet temperature of the condenser, $T_1 = 35°\text{C}$

Flow rate of the refrigerant, $\dot{m} = 0.018\ \text{kg/s}$

Exit pressure, $P_2 = 800\ \text{kPa}$

Quality of refrigerant at exit of condenser, $x_2 = 0$

Power input to the compressor, $\dot{W}_{net} = 1.2\ \text{kW}$

Step 2 of 6

From superheated refrigerant -134a tables

Taking enthalpy at $P_1 = 800\ \text{kPa}$ & $T_1 = 35°\text{C}$,

By interpolating the values

$$h_1 = 267.29 + (276.45 - 267.29)\left[\frac{35 - 31.31}{40 - 31.31}\right]$$

$$h_1 = 271.17\ \text{kJ/kg}$$

Step 3 of 6

From saturated refrigerant -134a tables

Taking enthalpy at $P_2 = 800\ \text{kPa}$ and $x_2 = 0$ is,

By interpolating the values

$$h_2 = h_f = 93.58 + (96.48 - 93.58)\left[\frac{800 - 770.64}{815.89 - 770.64}\right]$$

$$h_2 = h_f = 95.46\ \text{kJ/kg}$$

Step 4 of 6

(a) The COP of heat pump:

The heat rejected in the condenser is expressed as

$$\dot{Q}_H = \dot{m}(h_1 - h_2)\text{kW}$$

$$\dot{Q}_H = 0.018(271.17 - 95.46)\text{kW}$$

$$\dot{Q}_H = 3.16\ \text{kW}$$

Step 5 of 6

The measure of performance of a heat pump is also expressed in terms of the coefficient of performance COP_{HP} and it is defined as

$$\text{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$\text{COP}_{HP} = \frac{3.16}{1.2}$$

$$\text{COP}_{HP} = 2.633$$

∴ The COP of heat pump $\boxed{\text{COP}_{HP} = 2.633}$

Step 6 of 6

(b) The rate of heat absorption from the outside air is expressed as

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{net}$$

$$\dot{Q}_L = 3.16 - 1.2$$

$$\dot{Q}_L = 1.96\ \text{kW}$$

∴ The rate of heat absorption is $\boxed{\dot{Q}_L = 1.96\ \text{kW}}$

Question: 7.055P

Show that processes that use work for mixing are irreversible by considering an adiabatic system whose contents are stirred by turning a paddle wheel inside the system (e.g., stirring a cake mix with an electric mixer).

Step 1 of 1

Adiabatic stirring processes are irreversible .Because of the changing of the temperature which cause the energy stored within the system can not be spontaneously released in a manor to cause the mass of the system to turn the paddle wheel in the opposite direction to do work on the surroundings. So that work for mixing are irreversible process.

Question: 7.056P

Show that processes involving rapid chemical reactions are irreversible by considering the combustion of a natural gas (e.g., methane) and air mixture in a rigid container.

Step 1 of 1

The chemical reactions are almost irreversible.
The reaction which involved combustion of natural gas and air mixture in a rigid container, results in formation of carbon dioxide, water and some releases heat energy.
It is not possible to convert them into back.

Question: 7.057P

A cold canned drink is left in a warmer room where its temperature rises as a result of heat transfer. Is this a reversible process? Explain.

Step 1 of 1

The process is a irreversible process
The temperature gain by the cold canned drink occurs in an infinitesimally temperature difference which is difficult to reverse the process.

The heat transfer through a differential temperature difference dT can be considered to be reversible. There is a smaller heat transfer when there is a small temperature difference between two bodies. Any significant heat transfer through a small temperature difference requires a very large surface area and a very long time. Therefore, even though approaching reversible heat transfer is desirable from a thermodynamic point of view, it is not practical and not economically feasible.

Question: 7.058P

A block slides down an inclined plane with friction and no restraining force. Is this process reversible or irreversible? Justify your answer.

Step 1 of 1

When a block slides down an inclined plane with friction and no restraining force then the process is irreversible because in a reversible process both the system and the surroundings are returned to their initial states at the end of the reverse process. And in this case the process is irreversible we can explain it into two ways

(a)

Due to friction, the block and plane gets warm up. In order to attain equilibrium temperature, the heat is transferred to the surroundings. This causes change in state of surroundings which is permanent and cannot be restored to its initial state. Therefore, the process is irreversible.

(b)

The potential energy of the block gradually decreases and stored in some form of energy in the surroundings. The potential energy can be restored by lifting the block to its original state however, the state of surroundings cannot be changed. Hence, the process is irreversible.

Question: 7.059P

Why are engineers interested in reversible processes even though they can never be achieved?

Step 1 of 1

Because reversible processes can be approached in reality, and they form the limiting cases. Engineers are interested in reversible processes because Work producing devices such as turbines that operate on reversible processes deliver the most work, and work consuming devices such as compressors that operate on reversible processes consume the least work.

We know the second law efficiency is the degree of approximation to the actual processes to the corresponding reversible processes. This enables us to compare the performance of different devices that are designed to do the same task on the basis of their efficiencies. The better the design, the lower the irreversibilities and the higher the second-law efficiency

Question: 7.060P

Why does a nonquasi-equilibrium compression process require a larger work input than the corresponding quasi-equilibrium one?

Step 1 of 1

We know quasi equilibrium process is a reversible process, and the compressors consume least work or in non quasi equilibrium compression process it consumes larger work input

When the compression process is non-quasi equilibrium, the molecules on the piston face cannot escape fast enough, forming a high pressure region in front of the piston. It requires more work to move the piston against this high pressure region.

Question: 7.061P

Why does a nonquasi-equilibrium expansion process deliver less work than the corresponding quasi-equilibrium one?

Step 1 of 1

We know quasi equilibrium process is a reversible process, and work producing devices such as turbines deliver the most work or in non quasi equilibrium expansion process delivers less work output

And when an expansion process is non-quasi equilibrium, the molecules before the piston face cannot follow the piston fast enough, forming a low pressure region behind the piston. The lower pressure that pushes the piston produces less work.

Question: 7.062P

How do you distinguish between internal and external irreversibilities?

Step 1 of 1

A process is called internally irreversible if irreversibilities occur within the boundaries of the system during the process. During an internally reversible process, a system proceeds through a series of equilibrium states, and when the process is reversed, the system passes through exactly the same equilibrium states while returning to its initial state. That is, the paths of the forward and reverse processes coincide for an internally reversible process. The quasi-equilibrium process is an example of an internally reversible process.

A process is called externally irreversible if irreversibilities occur outside the system boundaries during the process. Heat transfer between a reservoir and a system is an externally reversible process if the outer surface of the system is at the temperature of the reservoir.

Question: 7.063P

Is a reversible expansion or compression process necessarily quasi-equilibrium? Is a quasi-equilibrium expansion or compression process necessarily reversible? Explain.

Step 1 of 2

A reversible process is a process that can be reversed without leaving any trace on the surroundings. That is, both the system and the surroundings are returned to their initial states at the end of the reverse process. This is possible only if the net heat and net work exchange between the system and the surroundings is zero Reversible processes actually do not occur in nature. They are merely idealizations of actual processes. So a reversible expansion or compression process cannot involve unrestrained expansion or sudden compression, and thus it is quasi-equilibrium.

Step 2 of 2

Quasi-equilibrium is the quasi-balanced state of a thermodynamic system near to thermodynamic equilibrium in some sense or degree. A process is called quasi-static when it follows a succession of equilibrium states; the surroundings may be irreversibly altered during the process so that after a return path, the system ends up in a final state which differs from its initial state.

A quasi-equilibrium expansion or compression process, on the other hand, may involve external irreversibility's (such as heat transfer through a finite temperature difference), and thus is not necessarily reversible.

Question: 7.064P

What are the four processes that make up the Carnot cycle?

Step 1 of 1

The four reversible processes that make up the Carnot cycle are as follows:

(1) Reversible Isothermal expansion. (Process 1-2, T_H = constant)

(2) Reversible Adiabatic expansion. (Process 2-3)

(3) Reversible Isothermal compression. (Process 3-4, T_L = constant)

(4) Reversible Adiabatic compression. (Process 4-1)

Question: 7.065P

What are the two statements known as the Carnot principles?

Step 1 of 1

Two conclusions pertain to the thermal efficiency of reversible and irreversible (i.e., actual) heat engines and they are known as the Carnot principles
Expressed as follows:
1. The efficiency of an irreversible heat engine is always less than the efficiency of a reversible one operating between the same two reservoirs.
2. The efficiencies of all reversible heat engines operating between the same two reservoirs are the same.

Question: 7.066P

Somebody claims to have developed a new reversible heat-engine cycle that has a higher theoretical efficiency than the Carnot cycle operating between the same temperature limits. How do you evaluate this claim?

Step 1 of 1

According to Carnot principle the efficiencies of all reversible heat engines operating between the same two reservoirs are the same so, the claim is false, since the Carnot theorem says that no heat engine cycle can have a higher thermal efficiency than the Carnot heat engine operating between the same temperature limits.

Question: 7.067P

Somebody claims to have developed a new reversible heat-engine cycle that has the same theoretical efficiency as the Carnot cycle operating between the same temperature limits. Is this a reasonable claim?

Step 1 of 1

According to Carnot principle the efficiencies of all reversible heat engines operating between the same two reservoirs are the same so, the claim is true

Question: 7.068P

Is it possible to develop (*a*) an actual and (*b*) a reversible heat-engine cycle that is more efficient than a Carnot cycle operating between the same temperature limits? Explain.

Step 1 of 2

(a) Impossible

No actual heat engine operating at same temperatures limits produces more efficiency than Carnot heat engine.

Step 2 of 2

(b) Impossible

No reversible heat engine operating at same temperatures limits produces more efficiency than Carnot heat engine.

A reversible heat engine operating at same temperatures limits produces efficiency which is equal to Carnot engine.

Question: 7.069P

Is there any way to increase the efficiency of a Carnot heat engine other than by increasing *TH* or decreasing *TL*?

Step 1 of 1

No, since the thermal efficiency of actual heat engines can be maximized by supplying heat to the engine at the highest possible temperature and rejecting heat from the engine at the lowest possible temperature so, the efficiency of Carnot heat engine is function of T_L and T_H

That is $\eta_{th-rev} = 1 - \frac{T_L}{T_H}$

Question: 7.070P

Consider two actual power plants operating with solar energy. Energy is supplied to one plant from a solar pond at 80°C and to the other from concentrating collectors that raise the water temperature to 600°C. Which of these power plants will have a higher efficiency? Explain.

Step 1 of 1

From the definition of thermal efficiency of a power plant, the thermal efficiency of actual heat engines cycles can be maximized by supplying heat to the engine at the highest possible temperature and rejecting heat from the engine at the lowest possible temperature so, the one which higher source temperature will have higher efficiency

$$\eta_{th-rev} = 1 - \frac{T_L}{T_H}$$

So the concentrating collectors which have 600°C source temperature have higher thermal efficiency.

Question: 7.071P

A heat engine operates between two thermal-energy reservoirs at 1260 R and 510 R. What is the maximum work the engine can produce per unit of heat that the engine receives from the source?

Step 1 of 1

The thermal efficiency η of a heat engine is equal to the ratio between maximum work, W_{max} obtained from the engine to the amount of heat supplied Q_H to the engine.

$$\eta = \frac{W_{max}}{Q_H} \qquad \text{..... (1)}$$

The maximum amount of work is obtained from the heat engine when the cycle is in reversible cycle. The thermal efficiency of reversible cycle is given by the following expression:

$$\eta = \frac{T_1 - T_2}{T_1} \qquad \text{..... (2)}$$

Here, T_1 is temperature of high temperature reservoir and T_2 is temperature of low temperature reservoir.

Equate equations (1) and (2).

$$\eta = \frac{W_{max}}{Q_H} = \frac{T_1 - T_2}{T_1}$$

$$\frac{W_{max}}{Q_H} = \frac{T_1 - T_2}{T_1}$$

Substitute 1260 R for T_1 and 510 R for T_2.

$$\frac{W_{max}}{Q_H} = \frac{1260 - 510}{1260}$$

$$= 0.595$$

Therefore, the maximum work obtained from the heat engine to the amount of heat supplied is $\boxed{0.595}$.

Question: 7.072P

From a work-production perspective, which is more valuable: (*a*) thermal energy reservoirs at 675 K and 325 K or (*b*) thermal energy reservoirs at 625 K and 275 K?

Step 1 of 2

a)

Source temperature, $T_H = 675\ \text{K}$

Sink temperature, $T_L = 325\ \text{K}$

The Carnot efficiency is given by

$$\eta_{th-rev} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th-rev} = 1 - \frac{325}{675}$$

$$\eta_{th-rev} = 0.518$$

$$\eta_{th-rev} = 51.8\%$$

Step 2 of 2

(b)

Source temperature, $T_H = 625\ \text{K}$

Sink temperature, $T_L = 275\ \text{K}$

The Carnot efficiency is given by

$$\eta_{th-rev} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th-rev} = 1 - \frac{275}{625}$$

$$\eta_{th-rev} = 0.56$$

$$\eta_{th-rev} = 56\%$$

The second produces more power, since it has higher thermal efficiency.

Question: 7.073P



A heat engine operates between a source at 477°C and a sink at 25°C. If heat is supplied to the heat engine at a steady rate of 65,000 kJ/min, determine the maximum power output of this heat engine.

Step 1 of 4

Given

Temperature of source $T_H = 477°C$

Temperature of sink $T_L = 25°C$

Heat supplied to the heat engine is $\dot{Q}_H = 65000 \ \frac{kJ}{min}$

Step 2 of 4

Temperature of source

$T_H = 477°C$

$= 477 + 273$

$= 750K$

Temperature of sink

$T_L = 25°C$

$= 25 + 273$

$= 298K$

Step 3 of 4

The maximum efficiency of any plant is the Carnot efficiency,

The Carnot efficiency $\eta_{th-rev} = 1 - \frac{T_L}{T_H}$

$$\eta_{th-rev} = 1 - \frac{298}{750}$$

$$= 0.602$$

$$\eta_{carnot} = 60\%$$

Step 4 of 4

Therefore maximum power output determined from the definition of thermal efficiency to be

$$\dot{W}_{net} = \eta_{carnot} \dot{Q}_H$$

$$\dot{W}_{net} = (0.60)\left(\frac{65000}{60}\right) \frac{kJ}{s}$$

$$\dot{W}_{net} = 650 \text{ kW}$$

$$\boxed{\dot{W}_{net} = 650 \text{ kW}}$$

Question: 7.074P

Reconsider Prob. 7–73. Using EES (or other) software, study the effects of the temperatures of the heat source and the heat sink on the power produced and the cycle thermal efficiency. Let the source temperature vary from 300 to 1000°C, and the sink temperature to vary from 0 to 50°C. Plot the power produced and the cycle efficiency against the source temperature for sink temperatures of 0°C, 25°C, and 50°C, and discuss the results.

Step 1 of 5

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 5

Equation window:

```
"T_H = 477 [C] "
T_L =50 [C]   [ Parameter to be varied 0,25 and 50 C]
Q_dot_H = 65000 [kJ/min]

"First Law applied to the heat engine"
Q_dot_H - Q_dot_L- W_dot_net = 0
W_dot_net_KW=W_dot_net*convert(kJ/min,kW)
"Cycle Thermal Efficiency - Temperatures must be absolute"
eta_th = 1 - (T_L + 273)/(T_H + 273)
"Definition of cycle efficiency"
eta_th=W_dot_net / Q_dot_H
```

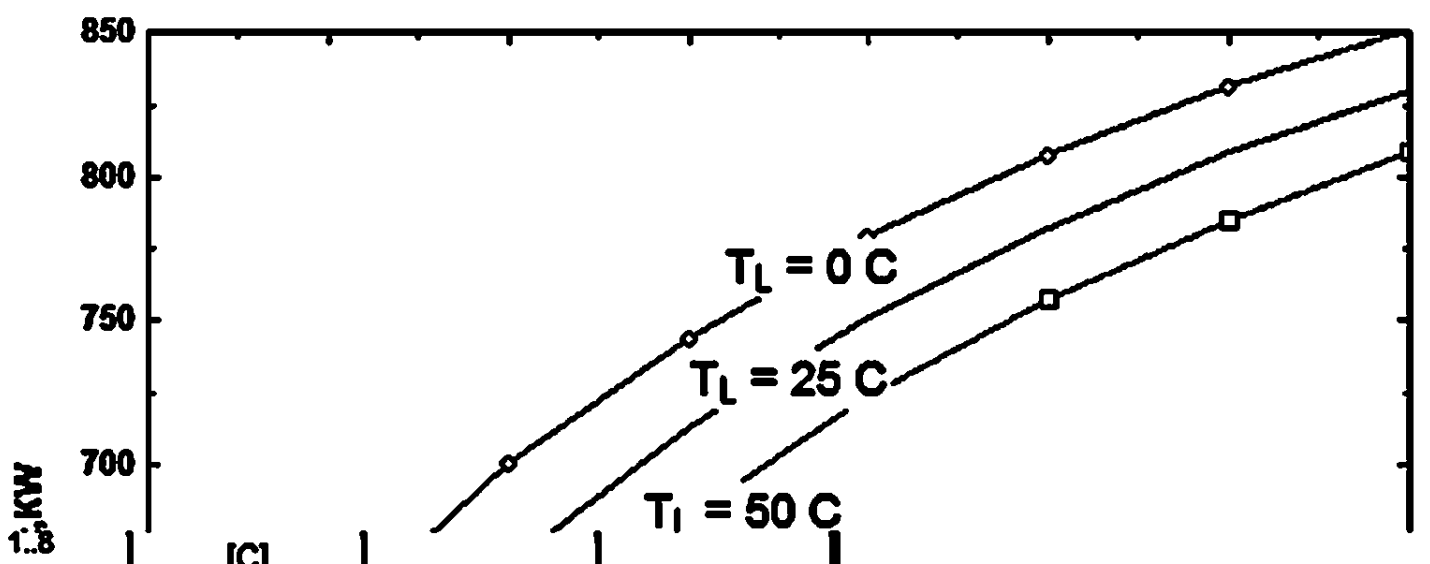
Step 3 of 5

Table for varies parameter as function of source temperature::

$\triangleright_{1..8}$	1 T_H [C]	2 $\dot{W}_{netKW}$	3 η_{th}
Run 1	300	472.7	0.4363
Run 2	400	563.4	0.5201
Run 3	500	630.7	0.5821
Run 4	600	682.5	0.63
Run 5	700	723.7	0.668
Run 6	800	757.2	0.699
Run 7	900	785	0.7246
Run 8	1000	808.5	0.7463

Step 4 of 5

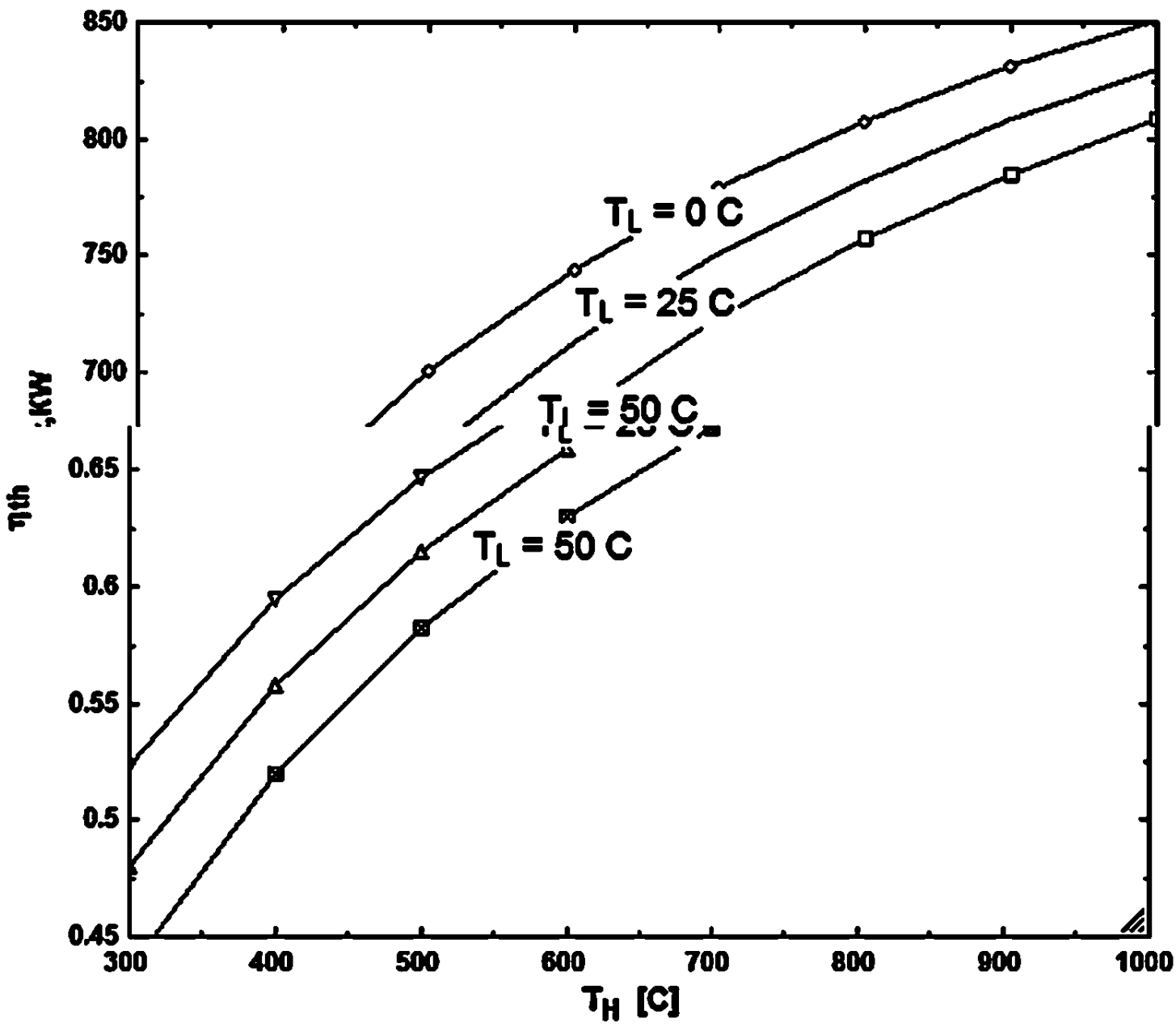
Graph between the Source temperature and Work input::



	[C]		
Run 1	300	472.7	0.4363
Run 2	400	563.4	0.5201
Run 3	500	630.7	0.5821
Run 4	600	682.5	0.63
Run 5	700	723.7	0.668
Run 6	800	757.2	0.699
Run 7	900	785	0.7246
Run 8	1000	808.5	0.7463

Step 4 of 5

Graph between the Source temperature and Work input::



Question: 7.075P

A heat engine is operating on a Carnot cycle and has a thermal efficiency of 75 percent. The waste heat from this engine is rejected to a nearby lake at 60°F at a rate of 800 Btu/min. Determine (*a*) the power output of the engine and (*b*) the temperature of the source.

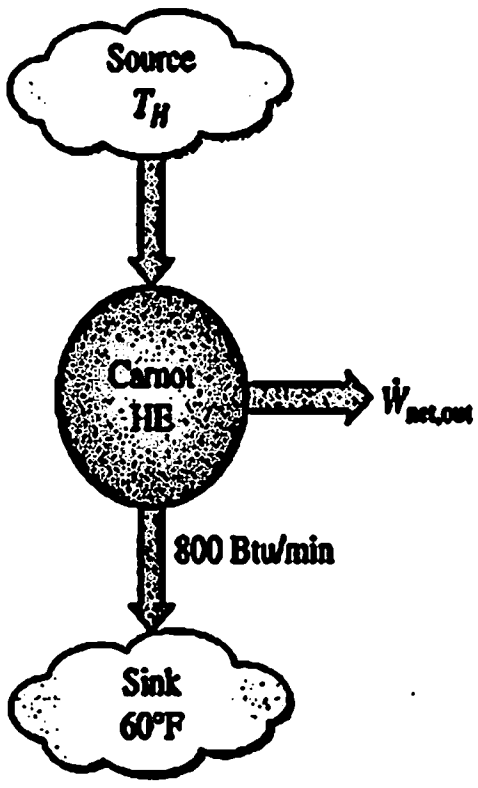


FIGURE P7–75E

Step 1 of 4

Given

Thermal efficiency of Carnot cycle heat engine $\eta_{th} = 75\%$

Temperature of sink $T_L = 60°F$

Heat rejected to lake at a rate of $\dot{Q}_L = 800$ Btu/min

Step 2 of 4

(a)

The power output of the engine

The maximum efficiency of any heat engine is the Carnot cycle engine efficiency

Carnot efficiency

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_H}$$

$$\eta = \frac{\dot{Q}_H - \dot{Q}_L}{\dot{Q}_H}$$

$$\eta = 1 - \frac{\dot{Q}_L}{\dot{Q}_H}$$

$$0.75 = 1 - \frac{800}{\dot{Q}_H}$$

$$\dot{Q}_H = \frac{800}{1 - 0.75}$$

$$\dot{Q}_H = 3200 \text{ Btu/min}$$

Step 3 of 4

Therefore maximum power output determined from the definition of thermal efficiency

$$\dot{W}_{net} = \eta_{carnot} \dot{Q}_H$$

$$\dot{W}_{net} = (0.75)(3200)$$

$$\dot{W}_{net} = 2400 \text{ Btu/min} \qquad (1 \text{ Btu/min} = 0.02356 \text{ hp})$$

$$\dot{W}_{net} = 2400 \times 0.02356$$

$$\boxed{\dot{W}_{net} = 56.544 \text{ hp}}$$

Step 4 of 4

(b) The temperature of the source

We know the Carnot efficiency of the heat engine cycle is

$$\eta = 1 - \frac{T_L}{T_H}$$

$$0.75 = 1 - \frac{60 + 460}{T_H}$$

$$\frac{520}{T_H} = 0.25$$

$$T_H = 2080\text{R}$$

Temperature of the source

$$\boxed{T_H = 2080\text{R}}$$

Question: 7.076P

An inventor claims to have devised a cyclical engine for use in space vehicles that operates with a nuclear-fuel-generated energy source whose temperature is 1000 R and a sink at 550 R that radiates waste heat to deep space. He also claims that this engine produces 5 hp while rejecting heat at a rate of 15,000 Btu/h. Is this claim valid?

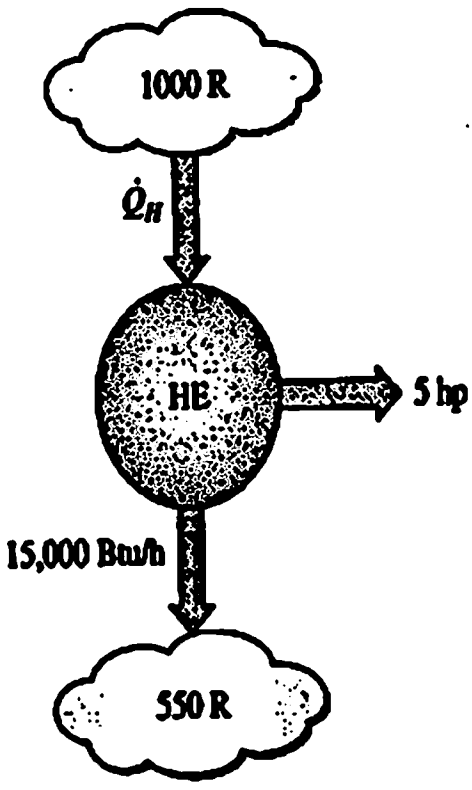


FIGURE P7–76E

Step 1 of 5

Given

Temperature of source $T_H = 1000\text{R}$

Temperature of sink $T_L = 550\text{R}$

Power produced $\dot{W}_{net} = 5\text{ hp}$

Heat rejected from the engine $\dot{Q}_L = 15,000\text{ Btu/h}$

Step 2 of 5

The thermal efficiency of reversible engine is

$$\eta_{rev} = 1 - \frac{T_L}{T_H}$$

$$\eta_{rev} = 1 - \frac{550}{1000}$$

$$= 0.45$$

$$\eta_{rev} = 45\%$$

Step 3 of 5

From first law of thermodynamics

Heat rejected is expressed as

$$\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$$

$$\dot{Q}_H = 5 \times 2545 + 15,000 \qquad \text{(Since 1 hp} = 2545\text{ Btu/h)}$$

$$\dot{Q}_H = 27725\text{ Btu/h}$$

Step 4 of 5

Thermal efficiency of any heat engine is expressed as

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_H}$$

$$\eta = \frac{5 \times 2545}{27725}$$

$$\eta = 0.4589$$

$$\eta = 45.89\%$$

Step 5 of 5

We know efficiency of Carnot cycle engine should be more than efficiency of any heat engine working under same thermal energy reservoirs

And the efficiency of given heat engine is greater than the Carnot efficiency of the given heat engine cycle. So the claim is invalid.

Question: 7.077P

A heat engine receives heat from a heat source at 1200°C and has a thermal efficiency of 40 percent. The heat engine does maximum work equal to 500 kJ. Determine the heat supplied to the heat engine by the heat source, the heat rejected to the heat sink, and the temperature of the heat sink.

Step 1 of 4

Given

Temperature of source $T_H = 1200°C$

Work done by heat engine $\dot{W}_{net} = 500$ kJ

The thermal efficiency of the heat engine $\eta_{th} = 0.4$

Step 2 of 4

Temperature of source

$T_H = 1200°C$

$= 1200 + 273$

$= 1473K$

The maximum efficiency of any plant is the Carnot efficiency,

The Carnot efficiency $\eta_{th-rev} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H}$

$$= \frac{\dot{W}_{net}}{\dot{Q}_H}$$

The heat supplied to heat engine is obtained as

$$\dot{Q}_H = \frac{\dot{W}_{net}}{\eta_{th}}$$

$$\dot{Q}_H = \frac{500}{0.4}$$

$$\boxed{\dot{Q}_H = 1250 \text{ kJ}}$$

Step 3 of 4

From first law of thermodynamics

Heat rejected is expressed as

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{net}$$

$$\dot{Q}_L = 1250 - 500$$

$$\boxed{\dot{Q}_L = 750 \text{ kJ}}$$

Step 4 of 4

The heat rejected to heat temperature can be determined from thermal energy relation

The thermal efficiency $\eta_{th} = 1 - \frac{T_L}{T_H}$

$$0.4 = 1 - \frac{T_L}{1473}$$

$$T_L = 883.8K$$

$$\boxed{T_L = 883.8K}$$

Question: 7.078P

A heat engine operating on the Carnot cycle has a measured work output of 900 kJ and heat rejection of 150 kJ to a heat reservoir at 27°C. Determine the heat supplied to the heat engine by the heat source, in kJ, and the temperature of the heat source, in °C.

Step 1 of 3

Given

Temperature of sink $T_L = 27°C$

Work done by heat engine $\dot{W}_{net} = 900$ kJ

The amount of heat rejected is $\dot{Q}_L = 150$ kJ

Step 2 of 3

Temperature of sink $T_L = 27°C$

$= 27 + 273$

$= 300K$

The maximum efficiency of any plant is the Carnot efficiency,

The Carnot efficiency $\eta_{th-rev} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

From first law of thermodynamics

Heat rejected is expressed as

$$\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$$

$$\dot{Q}_H = 900 + 150$$

$$\boxed{\dot{Q}_H = 1050 \text{ kJ}}$$

Step 3 of 3

The heat rejected to heat temperature can be determined from thermal energy relation

The thermal efficiency $\eta_{th-rev} = 1 - \frac{T_L}{T_H} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

$$1 - \frac{300}{T_H} = \frac{900}{1050}$$

$$T_H = 2100K$$

$$= 2100 - 273$$

$$T_H = 1827°C$$

$$\boxed{T_H = 1827°C}$$

Question: 7.079P

A heat engine is operating on a Carnot cycle and has a thermal efficiency of 75 percent. The waste heat from this engine is rejected to a nearby lake at 15°C at a rate of 14 kW. Determine the power output of the engine and the temperature of the source, in °C.

Step 1 of 4

Given

Temperature of sink $T_L = 15°C = 15+273 = 288K$

The amount of heat rejected is $\dot{Q}_L = 14$ kW

The thermal efficiency of the heat engine $\eta_{th} = 0.75$

Step 2 of 4

Temperature of sink $T_L = 15°C$

$= 15 + 273$

$= 288K$

The maximum efficiency of any plant is the Carnot efficiency,

The Carnot efficiency $\eta_{th-rev} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

The heat supplied to heat engine is obtained as

$$0.75 = 1 - \frac{14}{\dot{Q}_H}$$

$$\dot{Q}_H = 56 \text{ kW}$$

Step 3 of 4

Therefore the power output delivered by the heat engine is

$$\dot{W}_{net} = \eta_{th} \dot{Q}_H$$

$$\dot{W}_{net} = (0.75)(56)$$

$$\boxed{\dot{W}_{net} = 42 \text{ kW}}$$

Step 4 of 4

The heat rejected to heat temperature can be determined from thermal energy relation

The thermal efficiency $\eta_{th-rev} = 1 - \frac{T_L}{T_H} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

$$0.75 = 1 - \frac{288}{T_H}$$

$$T_H = 1152K$$

$$= 1152 - 273$$

$$T_H = 879°C$$

$$\boxed{T_H = 879°C}$$

Question: 7.080P

A geothermal power plant uses geothermal water extracted at 150°C at a rate of 210 kg/s as the heat source and produces 8000 kW of net power. The geothermal water leaves the plant at 90°C. If the environment temperature is 25 °C, determine (*a*) the actual thermal efficiency, (*b*) the maximum possible thermal efficiency, and (*c*) the actual rate of heat rejection from this power plant.

Step 1 of 5

Given

Mass flow rate of geothermal water $\dot{m}_w = 210\ \frac{\text{kg}}{\text{s}}$

Temperature of source $T_H = 150°\text{C}$

Power produces in the thermal plant $\dot{W}_{net} = 8000\ \text{kW}$

The exit temperature of the plant $T_{H_2} = 90°\text{C}$

Temperature of sink $T_L = 25°\text{C}$

Step 2 of 5

(a) The actual thermal efficiency

Using saturated liquid properties of water

At high temperature $T_H = 150°\text{C}$

$$h_1 = h_f = 632.18\ \frac{\text{kJ}}{\text{kg}}$$

At source temperature $T_{H_2} = 90°\text{C}$

$$h_{water} = h_f = 377.04\ \frac{\text{kJ}}{\text{kg}}$$

At low temperature $T_L = 25°\text{C}$

$$h_{sink} = h_f = 104.83\ \frac{\text{kJ}}{\text{kg}}$$

The rate of heat input to the plant is expressed as

$$\dot{Q}_{in} = \dot{m}_w (h_1 - h_2)\ \text{kW}$$

$$\dot{Q}_{in} = 210(632.18 - 377.04)\ \text{kW}$$

$$\dot{Q}_{in} = 53,580\ \text{kW}$$

Step 3 of 5

The thermal efficiency $\eta_{th} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = \frac{\dot{W}_{net}}{\dot{Q}_H}$

$$\eta_{th} = 1 - \frac{8000}{53,580}$$

$$\eta_{th} = 0.149$$

$$\boxed{\eta_{th} = 14.9\%}$$

Step 4 of 5

(b) The maximum possible thermal efficiency

The Carnot efficiency $\eta_{th-rev} = 1 - \frac{T_L}{T_H}$

$$\eta_{th-rev} = 1 - \frac{25+273}{150+273}$$

$$\eta_{th-rev} = 0.296$$

$$\boxed{\eta_{max} = 29.6\%}$$

Step 5 of 5

(c) The actual heat rejected to the system

$$\dot{Q}_{out} = \dot{Q}_{in} - \dot{W}_{net}$$

$$\dot{Q}_{out} = 53,580 - 8000\ \text{kW}$$

$$\boxed{\dot{Q}_{out} = 45,580\ \text{kW}}$$

Question: 7.081P

It is claimed that the efficiency of a completely reversible heat engine can be doubled by doubling the temperature of the energy source. Justify the validity of this claim.

Step 1 of 2

Let

Temperature of the heat source is T_H

Temperature of the heat sink is T_L

The thermal efficiency of completely reversible engine is expressed as

The Carnot efficiency

$$\eta_{rev} = \frac{T_H - T_L}{T_H}$$

$$\eta_{rev} = 1 - \frac{T_L}{T_H}$$

Step 2 of 2

Now the temperature of the source is doubled then temperature of heat source is $2T_H$

$$\eta' = \frac{2T_H - T_L}{2T_H}$$

$$\eta' = 1 - \frac{T_L}{2T_H}$$

We know efficiency of Carnot cycle engine should be more than efficiency of any heat engine working under same thermal energy reservoirs

Since thermal efficiency is less than the thermal efficiency of reversible heat engine

So the claim is not valid

Question: 7.082P

How can we increase the COP of a Carnot refrigerator?

Step 1 of 1

A refrigerator or a heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator.

The coefficient of performance of any refrigerator is expressed as

$$\mathrm{COP}_{\mathrm{Ref}} = \frac{1}{Q_H/Q_L - 1}$$

$$\mathrm{COP}_{\mathrm{Ref}} = \frac{1}{T_H/T_L - 1}$$

$$\mathrm{COP}_{\mathrm{Ref}} = \frac{T_L}{T_H - T_L}$$

From the above relation by reducing the value of T_H, or by increasing the value of T_L, the COP of a Carnot refrigerator can be increased.

Question: 7.083P

A homeowner buys a new refrigerator and a new air conditioner. Which one of these devices would you expect to have a higher COP? Why?

Step 1 of 1

The air conditioner operates at lower temperatures differences than a refrigerator, the smaller the difference between the temperatures limits the higher the COP will, so the air conditioner should have higher COP.

The power input to a refrigerator/air conditioner will be a minimum when the refrigerator/air conditioner operates in a reversible manner. The coefficient of performance of a reversible refrigerator/air conditioner depends on the temperature limits in the cycle only

i.e.

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_R = \frac{T_L}{T_H - T_L}$$

Question: 7.084P

A homeowner buys a new refrigerator with no freezer compartment and a deep freezer for the new kitchen.

Which of these devices would you expect to have a lower COP? Why?

Step 1 of 1

The power input to a refrigerator/air conditioner will be a minimum when the refrigerator/air conditioner operates in a reversible manner. The coefficient of performance of a reversible refrigerator/air conditioner depends on the temperature limits in the cycle only

i.e.
$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_R = \frac{T_L}{T_H - T_L}$$

Here the freezer operates at higher temperature differences, so the COP of the freezer will be lower compared to refrigerator.

Question: 7.085P

In an effort to conserve energy in a heat-engine cycle, somebody suggests incorporating a refrigerator that will absorb some of the waste energy *QL* and transfer it to the energy source of the heat engine. Is this a smart idea? Explain.

Step 1 of 1

No it is a wrong idea. In actual cycle due to irreversibilities the work consumed by the refrigerator will always be greater than work produced by the heat engine, which leads to decrease in the thermal efficiency of the power plant.
For a reversible cycle the power consumed by the refrigerator will be almost equal to power produced.

Question: 7.086P

It is well established that the thermal efficiency of a heat engine increases as the temperature T_L at which heat is rejected from the heat engine decreases. In an effort to increase the efficiency of a power plant, somebody suggests refrigerating the cooling water before it enters the condenser, where heat rejection takes place. Would you be in favor of this idea? Why?

Step 1 of 1

No it is a wrong idea. Here maintaining the refrigerator will be more cost than sending normal water in to the condenser. In actual cycle the work consumed by the refrigerator will always be greater than work produced by the heat engine due to irreversibilities which is leads to decrease in the thermal efficiency of the power plant.

Question: 7.087P

It is well known that the thermal efficiency of heat engines increases as the temperature of the energy source increases. In an attempt to improve the efficiency of a power plant, somebody suggests transferring heat from the available energy source to a higher-temperature medium by a heat pump before energy is supplied to the power plant. What do you think of this suggestion? Explain.

Step 1 of 1

No it is a wrong idea. Here maintaining a heat pump will be a costly thing, better to use a normal boiler for heating the higher temperature body.
In actual cycle due to irreversibilities the work consumed by the heat pump will always be greater than work produced by the heat engine which is leads to decrease in the thermal efficiency of the power plant.

Question: 7.088P

Determine the minimum work per unit of heat transfer from the source that is required to drive a refrigerator with thermal-energy reservoirs at 273 K and 303 K.

Step 1 of 3

Given data:

Temperature of source, $T_H = 303\text{K}$

Temperature of sink, $T_L = 273\text{K}$

Step 2 of 3

The minimum work per unit heat transfer of a refrigerator is expressed as

$$W_{net} = Q_H - Q_L$$

The maximum coefficient of performance of the reversible refrigerator is given by

$$\text{COP} = \frac{T_L}{T_H - T_L}$$

The COP is also given by

$$\text{COP} = \frac{Q_L}{Q_H - Q_L} = \frac{Q_L}{W_{net}}$$

Step 3 of 3

Therefore we can write,

$$\frac{Q_L}{W_{net}} = \frac{T_L}{T_H - T_L}$$

The minimum work per unit of heat transfer from the source that is

$$\frac{W_{net}}{Q_L} = \frac{T_H - T_L}{T_L}$$

$$\frac{W_{net}}{Q_L} = \frac{303 - 273}{273}$$

$$\frac{W_{net}}{Q_L} = 0.1098$$

$\therefore$ The minimum work required to drive a refrigerator is $\boxed{\frac{W_{net}}{Q_L} = 0.1098}$

Question: 7.089P

An inventor claims to have developed a heat pump that produces a 200-kW heating effect for a 293 K heated zone while only using 75 kW of power and a heat source at 273 K. Justify the validity of this claim.

Step 1 of 3

Given data:

The power produced by the heat pump is, $\dot{W}_{net} = 75\,\text{kW}$

The heat effect of the heat pump is, $\dot{Q}_H = 200\,\text{kW}$

The temperature of sink, $T_L = 273\text{K}$

The temperature of source, $T_H = 293\text{K}$

Step 2 of 3

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\begin{aligned} COP_{HP} &= \frac{T_H}{T_H - T_L} \\ &= \frac{\dot{Q}_H}{\dot{W}_{net}} \\ &= \frac{200}{75} \\ &= 2.667 \end{aligned}$$

The actual COP of heat pump is $COP_{HP} = 2.667$

Step 3 of 3

The maximum COP for a reversible heat pump

$$\begin{aligned} COP_{HP_rev} &= \frac{T_H}{T_H - T_L} \\ &= \frac{293}{293 - 273} \\ &= 14.65 \end{aligned}$$

The reversible COP of heat pump is $COP_{HP} = 14.65$

Therefore the actual COP is less than the maximum COP of the heat pump so the claim is valid

Question: 7.090P

A heat pump operates on a Carnot heat pump cycle with a COP of 8.7. It keeps a space at 26°C by consuming 4.25 kW of power. Determine the temperature of the reservoir from which the heat is absorbed and the heating load provided by the heat pump.

Step 1 of 3

Given data:

The COP of the heat pump is $COP = 8.7$

The power input to the heat pump $\dot{W}_{net} = 4.25\ \text{kW}$

Temperature of the source $T_H = 26°\text{C}$

$= 26 + 273$

$= 299\text{K}$

Step 2 of 3

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

From the above relation the temperature of the low temperature reservoir is

$$8.7 = \frac{299}{299 - T_L}$$

$$299 - T_L = \frac{299}{8.7}$$

$$T_L = 299 - 34.367$$

$$T_L = 264.6\text{K}$$

∴ The low temperature is $\boxed{T_L = 264.6\text{K}}$

Step 3 of 3

The heating load effect of the heat pump

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$8.7 = \frac{\dot{Q}_H}{4.25}$$

$$\dot{Q}_H = 36.975\ \text{kW}$$

∴ The required heating load is $\boxed{\dot{Q}_H = 36.975\ \text{kW}}$

Question: 7.091P

A refrigerator is to remove heat from the cooled space at a rate of 300 kJ/min to maintain its temperature at −8°C. If the air surrounding the refrigerator is at 25°C, determine the minimum power input required for this refrigerator.

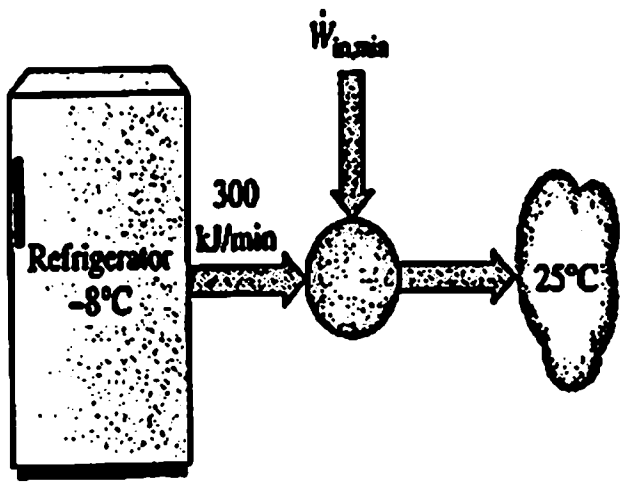


FIGURE P7-91

Step 1 of 4

Given

Surrounding air temperature $T_H = 25°C$

$= 25+273$

$T_H = 298K$

Refrigerated space temperature $T_L = -8°C$

$= -8+273$

$T_L = 265K$

Rate of cooling effect $\dot{Q}_L = 300$ kJ/min

$\dot{Q}_L = 5$ kW

Step 2 of 4

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only

$$COP_{Ref} = \frac{T_L}{T_H - T_L}$$

$$= \frac{265}{298-265}$$

$$COP_{Ref} = 8.0303$$

Step 3 of 4

The coefficient of performance of any refrigerator is also expressed as

$$COP_{Ref} = \frac{Q_L}{Q_H - Q_L}$$

$$8.0303 = \frac{5}{\dot{Q}_H - 5}$$

$$\dot{Q}_H - 5 = 0.6226$$

$$\dot{Q}_H = 5.6226\text{kW}$$

Step 4 of 4

Power input required for this refrigerator is

$$\dot{W}_{net} = \dot{Q}_H - \dot{Q}_L$$

$$= 5.6226 - 5$$

$$\boxed{\dot{W}_{net} = 0.6226\text{kW}}$$

Question: 7.092P

An inventor claims to have developed a refrigeration system that removes heat from the closed region at −12°C and transfers it to the surrounding air at 25°C while maintaining a COP of 6.5. Is this claim reasonable? Why?

Step 1 of 3

Given

Temperature of source $T_H = 25°C$

$= 25+273$

$T_L = 298K$

Temperature of sink $T_L = -12°C$

$= -12+273$

$T_L = 261K$

COP of refrigeration system is $COP = 6.5$

Step 2 of 3

The maximum coefficient of performance of any refrigerator is expressed as

$$COP_{rev} = \frac{T_L}{T_H - T_L}$$

$$COP_{rev} = \frac{261}{298-261}$$

$$COP_{rev} = 7.05$$

Step 3 of 3

We know that reversible refrigeration system is having maximum COP and in the problem COP claimed by the inventor is less than the maximum COP.

So the inventor claim is valid.

Question: 7.093P

An air-conditioning system is used to maintain a house at 70°F when the temperature outside is 100°F. The house is gaining heat through the walls and the windows at a rate of 800 Btu/min, and the heat generation rate within the house from people, lights, and appliances amounts to 100 Btu/min. Determine the minimum power input required for this air-conditioning system.

Step 1 of 3

Conversion factor $(\text{Fahrenheit})°\text{F}$ to $(\text{Rankine})\text{R}$

$$1°\text{F} = 460\,\text{R}$$

Convert the source temperature of the air-conditioner $(\text{Fahrenheit})°\text{F}$ to $(\text{Rankine})\text{R}$ by using the above conversion factor

$$\begin{aligned} T_H &= 100°\text{F} \\ &= 100 + 460 \\ &= 560\,\text{R} \end{aligned}$$

Convert the sink temperature of the air-conditioner $(\text{Fahrenheit})°\text{F}$ to $(\text{Rankine})\text{R}$ by using the above conversion factor

$$\begin{aligned} T_L &= 70°\text{F} \\ &= 70 + 460 \\ &= 530\,\text{R} \end{aligned}$$

Step 2 of 3

Consider the equation for coefficient of performance of refrigerator

$$\begin{aligned} \text{COP}_{\text{Ref}} &= \frac{\text{Desired output}}{\text{Required input}} \\ &= \frac{\dot{Q}_{out}}{\dot{W}_{net}} \qquad \text{..... (1)} \\ &= \frac{\dot{Q}_L + \dot{Q}_W}{\dot{W}_{net}} \end{aligned}$$

Here, the rate of heat gained through the wall is $\dot{Q}_W$ and the heat generation from people and lights and appliances is $\dot{Q}_L$

Consider the equation for coefficient of performance of refrigerator in terms of temperatures

$$\text{COP}_{\text{Ref}} = \frac{T_L}{T_H - T_L}$$

Substitute $530\,\text{R}$ for T_L and $560\,\text{R}$ for T_H in the equation

$$\begin{aligned} \text{COP}_{\text{Ref}} &= \frac{530}{560 - 530} \\ &= 17.66 \end{aligned}$$

Step 3 of 3

Substitute 17.66 for COP_{Ref}, $800\ \text{Btu/min}$ for $\dot{Q}_W$ and $100\ \text{Btu/min}$ for $\dot{Q}_L$ in the equation (1)

$$17.66 = \frac{100 + 800}{\dot{W}_{net}}$$

$$\dot{W}_{net} = 50.94\ \text{Btu/min}$$

Conversion factor for power Btu/min to 1hp

$$1\text{hp} = 42.53\ \text{Btu/min}$$

Convert the units of power Btu/min to 1hp by using the above conversion factor

$$\begin{aligned} \dot{W}_{net} &= \frac{50.94\ \text{Btu/min}}{42.53} \times \frac{\text{hp}}{\text{Btu/min}} \\ &= 1.197\ \text{hp} \end{aligned}$$

Therefore, the minimum power input is $\boxed{1.197\ \text{hp}}$

Question: 7.094P

A heat pump is used to maintain a house at 22°C by extracting heat from the outside air on a day when the outside air temperature is 2°C. The house is estimated to lose heat at a rate of 110,000 kJ/h, and the heat pump consumes 5 kW of electric power when running. Is this heat pump powerful enough to do the job?

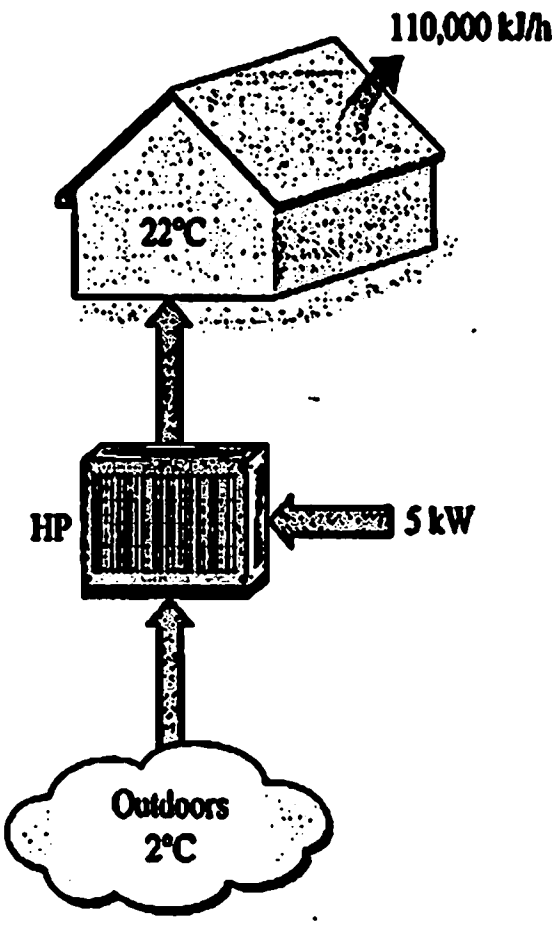


FIGURE P7–94

Step 1 of 3

Given data:

Temperature of source, $T_H = 22°C$

$= 22+273$

$= 295K$

Temperature of sink, $T_L = 2°C$

$= 2+273$

$= 275K$

The rate at which heat is lost is, $\dot{Q}_H = 110,000$ kJ/h

The power consumed by the heat pump, $\dot{W}_{net} = 5$ kW

Step 2 of 3

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP} = \frac{295}{295-275}$$

$$COP_{HP} = 14.75$$

Step 3 of 3

The minimum work per unit heat transfer of a heat pump is expressed as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$14.75 = \frac{110,000}{\dot{W}_{net}}$$

$$\dot{W}_{net} = \frac{110000}{3600 \times 14.75}$$

$$\dot{W}_{net} = 2.07 \text{ kW}$$

The power consumed is 5 kW which is more than $\dot{W}_{net} = 2.07$ kW so this heat pump powerful enough to do the job.

Question: 7.095P

A completely reversible refrigerator operates between thermal energy reservoirs at 450 R and 540 R. How many kilowatts of power are required for this device to produce a 15,000-Btu/h cooling effect?

Step 1 of 3

Given data:

Temperature of source, $T_H = 540\text{R}$

Temperature of sink, $T_L = 450\text{R}$

The cooling effect, $\dot{Q}_L = 15{,}000\ \text{Btu/h}$

Step 2 of 3

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only

$$\text{COP}_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

Step 3 of 3

A refrigerator or a heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator.

We know that network output, $W_{net} = Q_H - Q_L$

The coefficient of performance of any refrigerator is expressed as

$$\text{COP}_{Ref} = \frac{Q_L}{Q_H - Q_L}$$

$$\text{COP}_{Ref} = \frac{1}{Q_H / Q_L - 1}$$

Where $\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$

$$\text{COP}_{Ref} = \frac{1}{T_H / T_L - 1}$$

$$\frac{450}{540 - 450} = \frac{15000}{\dot{W}_{net}}$$

$$\dot{W}_{net} = 3000\ \text{Btu/h}$$

We know that, $1\ \text{kW} = 3412\ \text{Btu/h}$

$$\dot{W}_{net} = 0.879\ \text{kW}$$

Therefore the minimum power input is $\boxed{\dot{W}_{net} = 0.879\ \text{kW}}$

Question: 7.096P

A refrigerator operating on the reversed Carnot cycle has a measured work input of 200 kW and heat rejection of 2000 kW to a heat reservoir at 27°C. Determine the cooling load supplied to the refrigerator, in kW, and the temperature of the heat source, in °C.

Step 1 of 4

Given data:

The source temperature of the refrigerator is $T_H = 27°C$

$= 27 + 273$

$= 300K$

The work input to the refrigerator is, $\dot{W}_{net} = 200\ kW$

The hear supplied to the refrigerator is, $\dot{Q}_H = 2000\ kW$

Step 2 of 4

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$COP_R = \frac{1}{T_H / T_L - 1}$$

Step 3 of 4

A refrigerator or a heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator.

We know that $\dot{W}_{net} = \dot{Q}_H - \dot{Q}_L$

Therefore the cooling effect of the refrigerator is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{net}$$

$$\dot{Q}_L = 2000 - 200$$

$$\dot{Q}_L = 1800\ kW$$

∴ The cooling effect of the refrigerator is $\boxed{\dot{Q}_L = 1800\ kW}$

Step 4 of 4

The temperature of heat sink is determined by from the definition of COP

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$COP_R = \frac{1}{T_H / T_L - 1}$$

$$\frac{1800}{200} = \frac{T_L}{300 - T_L}$$

$$T_L = 270K$$

∴ The temperature is $\boxed{T_L = -3°C}$

Question: 7.097P

An 18,000 Btu/h split air conditioner running at full load is used to keep a room at 22°C in an environment at 33°C. The power input to the compressor is 3.4 kW. Determine (*a*) the rate of heat rejected in the condenser, in kJ/h, (*b*) the COP of the air conditioner, and (*c*) the rate of cooling, in Btu/h, if the air conditioner operated as a Carnot refrigerator for the same power input.

Step 1 of 4

Given data:

The cooling load to the refrigerator is, $\dot{Q}_L = 18000$ Btu/h

The power input to the refrigerator is, $\dot{W}_{in} = 3.4$ kW

The source temperature of the refrigerator is, $T_H = 33+273$
$= 306$K

The sink temperature of the refrigerator is, $T_L = 22+273$
$= 295$K

Conversion factor, 1 Btu = 1.055 kJ

Step 2 of 4

(a) The rate of heat rejected

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$\dot{Q}_H = \dot{W}_{in} + \dot{Q}_L$$

Therefore the rate of heat rejected is

$$\dot{Q}_H = 3.4 \text{ kW}\left(\frac{3600 \text{ kJ/h}}{1 \text{ kW}}\right) + 18000 \text{ Btu/h}\left(\frac{1.055 \text{ kJ}}{\text{Btu}}\right)$$

$$\dot{Q}_H = 31,230 \text{ kJ/h}$$

∴ The rate of heat rejected is $\boxed{\dot{Q}_H = 31,230 \text{ kJ/h}}$

Step 3 of 4

(b) The COP is expressed as

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$COP_R = \frac{18000 \text{ Btu/h}}{3.4 \text{ Btu/h}\,(3600/1.055)}$$

$$COP_R = 1.55$$

∴ The COP of refrigerator is, $\boxed{COP_R = 1.55}$

Step 4 of 4

(c) The rate of cooling of the refrigerator if the cycle operates in as Carnot refrigerator

$$COP_{Carnot} = \frac{T_L}{T_H - T_L}$$

$$COP_{Carnot} = \frac{295}{306-295}$$

$$COP_{Carnot} = 26.818$$

Therefore the cooling effect

$$\dot{Q}_L = COP_{Carnot} \times \dot{W}_{net}$$

$$\dot{Q}_L = 26.818 \times \left(3.4\,(3600/1.055)\right) \text{ Btu/h}$$

$$\dot{Q}_L = 311,140 \text{ Btu/h}$$

∴ The cooling effect is $\boxed{\dot{Q}_L = 311,140 \text{ Btu/h}}$

Question: 7.098P

A Carnot refrigerator absorbs heat from a space at 15°C at a rate of 16,000 kJ/h and rejects heat to a reservoir at 36°C. Determine the COP of the refrigerator, the power input, in kW, and the rate of heat rejected to high-temperature reservoir, in kJ/h.

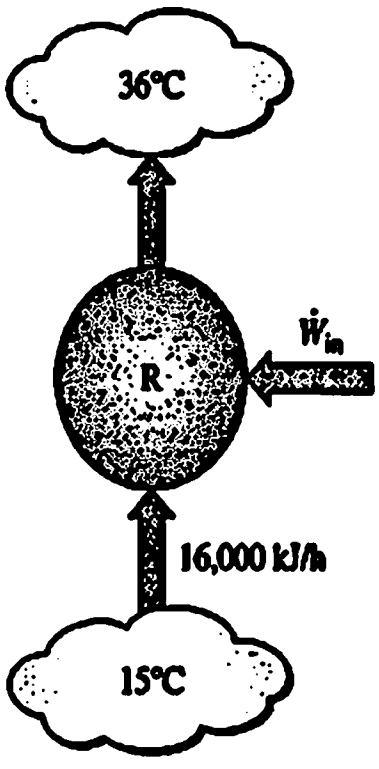


FIGURE P7–98

Step 1 of 4

Given data:

The source temperature of the refrigerator, $T_H = 36 + 273$

$= 309\text{K}$

The sink temperature of the refrigerator, $T_L = 15 + 273$

$= 288\text{K}$

The cooling effect of the refrigerator, $\dot{Q}_L = 16{,}000\ \text{kJ/h}$

Step 2 of 4

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_{Carnot} = \frac{T_L}{T_H - T_L}$$

$$COP_{Carnot} = \frac{288}{309 - 288}$$

$$COP_{Carnot} = 13.71$$

Step 3 of 4

The power input to the refrigerator is

$$COP_{R_net} = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

$$\dot{W}_{net} = \frac{16000}{13.71}$$

$$\dot{W}_{net} = 1167\left(\frac{1\text{h}}{3600}\right)\text{kJ/s}$$

$$\dot{W}_{net} = 0.324\ \text{kW}$$

∴ The power input is $\boxed{\dot{W}_{net} = 0.324\ \text{kW}}$

Step 4 of 4

The rate of heat rejected to the high temperature reservoir is

We know that $\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$

Therefore the rate of heat rejected is

$$\dot{Q}_H = 16000\ \text{kJ/h} + 1167\ \text{kJ/h}$$

$$\dot{Q}_H = 17{,}167\ \text{kJ/h}$$

∴ The rate of heat rejected $\boxed{\dot{Q}_H = 17{,}167\ \text{kJ/h}}$

Question: 7.099P

The performance of a heat pump degrades (i.e., its COP decreases) as the temperature of the heat source decreases. This makes using heat pumps at locations with severe weather conditions unattractive. Consider a house that is heated and maintained at 20°C by a heat pump during the winter. What is the maximum COP for this heat pump if heat is extracted from the outdoor air at (*a*) 10°C, (*b*) -5°C, and (*c*) -30°C?

Step 1 of 4

Given data:

Temperature of source, $T_H = 20°C$

$= 20 + 273$

$= 293K$

Step 2 of 4

(a) Temperature of the heat extracted is, $T_L = 10°C$

$= 10 + 273$

$= 283K$

The maximum COP is calculated from the reversible Carnot COP

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP} = \frac{293}{293 - 283}$$

$$COP_{HP} = 29.3$$

∴ The maximum COP of the heat pump is $\boxed{COP_{HP} = 29.3}$

Step 3 of 4

(b) Temperature of the heat extracted is, $T_L = -5°C$

$= -5 + 273$

$= 268K$

The maximum COP is calculated from the reversible Carnot COP

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP} = \frac{293}{293 - 268}$$

∴ The maximum COP is $\boxed{COP_{HP} = 11.72}$

Step 4 of 4

(c) Temperature of the heat extracted is, $T_L = -30°C$

$= -30 + 273$

$= 243K$

The maximum COP is calculated from the reversible Carnot COP

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{293}{293 - 243}$$

$$COP_{HP} = 5.86$$

∴ The maximum COP is $\boxed{COP_{HP} = 5.86}$

Question: 7.100P

A heat pump is to be used for heating a house in the winter. The house is to be maintained at 78°F at all times. When the temperature outdoors drops to 25°F, the heat losses from the house are estimated to be 55,000 Btu/h. Determine the minimum power required to run this heat pump if heat is extracted from *(a)*the outdoor air at 25°F and *(b)*the well water at 50°F.

Step 1 of 5

Given data:

Temperature of source, $T_H = 78°F$

$= 78+460$

$= 538R$

The rate at which heat is lost is $\dot{Q}_H = 55,000\ \text{Btu/h}$

Step 2 of 5

a)

Temperature of the heat extracted is, $T_L = 25°F$

$= 25+460$

$= 485R$

The maximum COP is calculated from the reversible Carnot COP

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{HP} = \frac{T_H}{T_H - T_L}$$

$$\text{COP}_{HP} = \frac{538}{538-485}$$

$$\text{COP}_{HP} = 10.15$$

Step 3 of 5

The minimum work per unit heat transfer of a heat pump is expressed as

$$\text{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$\frac{55,000}{\dot{W}_{net}} = 10.15$$

$$\dot{W}_{net} = \frac{5418.2}{2545\ \text{Btu/h}}(1\text{hp}) \qquad (\because 1\ \text{hp} = 2545\ \text{Btu/h})$$

∴ The minimum power required is $\boxed{\dot{W}_{net} = 2.12\ \text{hp}}$

Step 4 of 5

(b) Temperature of the heat extracted is, $T_L = 50°F$

$= 50+460$

$= 510R$

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{HP} = \frac{T_H}{T_H - T_L}$$

$$\text{COP}_{HP} = \frac{538}{538-510}$$

$$\text{COP}_{HP} = 19.21$$

Step 5 of 5

The minimum work per unit heat transfer of a heat pump is expressed as

$$\text{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$\frac{55,000}{\dot{W}_{net}} = 19.21$$

$$\dot{W}_{net} = 2863\ \text{Btu/h}$$

$$\dot{W}_{net} = \frac{2863}{2545\ \text{Btu/h}}(1\text{hp}) \qquad (\because 1\ \text{hp} = 2545\ \text{Btu/h})$$

∴ The minimum power required is $\boxed{\dot{W}_{net} = 1.124\ \text{hp}}$

Question:

7.101P

A Carnot heat pump is to be used to heat a house and maintain it at 25°C in winter. On a day when the average outdoor temperature remains at about 2°C, the house is estimated to lose heat at a rate of 55,000 kJ/h. If the heat pump consumes 6.6 kW of power while operating, determine (*a*) how long the heat pump ran on that day; (*b*) the total heating costs, assuming an average price of 8.5e7kWh for electricity; and (*c*) the heating cost for the same day if resistance heating is used instead of a heat pump.

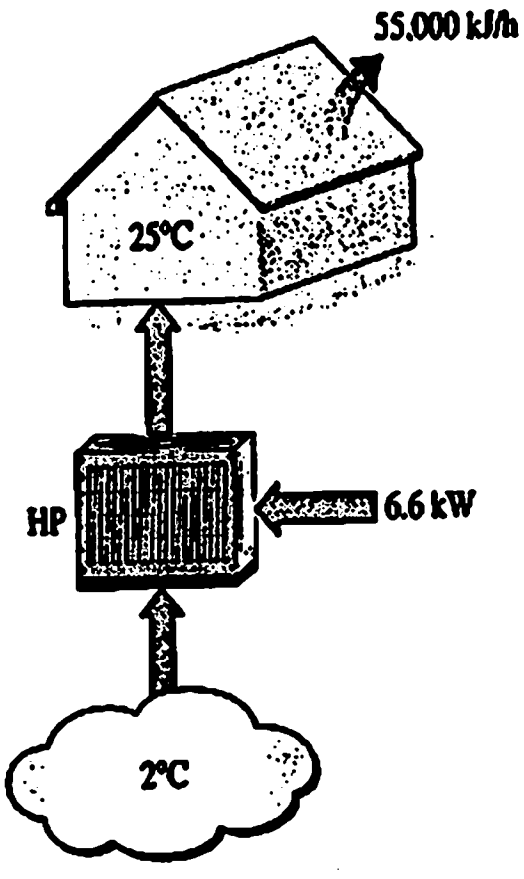


FIGURE P7–101

Step 1 of 6

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Step 2 of 6

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP_rev} = \frac{298}{298-275}$$

$$COP_{HP} = 12.96$$

Step 3 of 6

Amount of heat loss in a day is

$$Q_H = \dot{Q}_H \times 24$$
$$= 55000 \times 24$$
$$= 1{,}320{,}000 \text{ kJ}$$

The minimum work per unit heat transfer of a heat pump is expressed as

$$COP_{HP} = \frac{Q_H}{W_{net}}$$

$$\frac{1{,}320{,}000}{W_{net}} = 12.96$$

$$W_{net} = 101{,}852 \text{ kJ}$$

Step 4 of 6

(a) The period of time that the heat pump ran a day is

$$t = \frac{W_{net}}{\dot{W}_{net}}$$
$$= \frac{101{,}852}{6.6}$$
$$= 15{,}432 \text{ sec}$$
$$= \frac{15{,}432}{3600} h$$

∴ The time required is, $\boxed{t = 4.29 \text{ h}}$

Step 5 of 6

(b) The total heating cost per day

$$\text{Cost} = W \times 8.5\text{cent} \times t$$
$$= 6.6 \times 0.085 \times 4.29$$
$$= \$2.41$$

Therefore the total heating cost is, $\boxed{\text{Cost} = \$2.41}$

Step 6 of 6

(c) The heat cost of the same day

$$Q_H \times 0.085\$/\text{kwh} = 1{,}320{,}000 \times 0.085 \frac{1\text{h}}{3600\text{s}}$$
$$= \$31.2$$

Therefore the heating cost is $\boxed{\$31.2}$

Question:

7.102P

A Carnot heat engine receives heat from a reservoir at 900°C at a rate of 800 kJ/min and rejects the waste heat to the ambient air at 27°C. The entire work output of the heat engine is used to drive a refrigerator that removes heat from the refrigerated space at −5°C and transfers it to the same ambient air at 27°C. Determine (*a*) the maximum rate of heat removal from the refrigerated space and (*b*) the total rate of heat rejection to the ambient air.

Step 1 of 8

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Step 2 of 8

(a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency,

The thermal efficiency, $\eta_{th} = 1 - \frac{T_L}{T_H}$

$$\eta_{th} = 1 - \frac{300}{1173}$$

$$\eta_{th} = 0.744$$

Step 3 of 8

The maximum power output of a heat engine is expressed as

$$\dot{W}_{net} = \eta_{th}\dot{Q}_H$$

$$= 0.744(800)$$

$$= 595.4 \text{ kJ/min}$$

The power output from the heat engine will be power input to the refrigerant

Step 4 of 8

The rate at which heat rejected in the heat engine is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{net}$$

$$= 800 - 595.4$$

$$= 204.6 \text{ kJ/min}$$

Step 5 of 8

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends upon the temperature limits in the cycle only

$$\text{COP}_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$\text{COP}_{Ref_max} = \frac{1}{T_H/T_L - 1}$$

$$\text{COP}_{Ref_max} = \frac{268}{300 - 268}$$

$$\text{COP}_{Ref_max} = 8.375$$

Step 6 of 8

Therefore the rate of heat removal from the refrigerant is expressed as

$$\dot{Q}_{ref} = \text{COP} \times \dot{W}_{net}$$

$$= 8.375 \times 595.4$$

$$= 4986.4 \text{ kJ/min}$$

∴ The rate of heat removal, $\boxed{\dot{Q}_{ref} = 4986.4 \text{ kJ/min}}$

Step 7 of 8

(b) The total heat rejected to ambient

For refrigerator, heat rejected is expressed as

$$\dot{Q}_{H,ref} = \dot{W}_{net} + \dot{Q}_L$$

$$\dot{Q}_{H,ref} = 595.4 + 4986.4$$

$$\dot{Q}_{H,ref} = 5581.8 \text{ kJ/min}$$

Step 8 of 8

Therefore the total heat rejected to ambient is

$$\dot{Q}_{total} = \dot{Q}_L + \dot{Q}_{H,ref}$$

$$\dot{Q}_{total} = 5581.8 + 204.6$$

$$\dot{Q}_{total} = 5786.4 \text{ kJ/min}$$

∴ The total heat rejection is, $\boxed{Q_{total} = 5786.4 \text{ kJ/min}}$

Question: 7.103P

A Carnot heat engine receives heat from a reservoir at 1700°F at a rate of 700 Btu/min and rejects the waste heat to the ambient air at 80°F. The entire work output of the heat engine is used to drive a refrigerator that removes heat from the refrigerated space at 20°F and transfers it to the same ambient air at 80°F. Determine (*a*) the maximum rate of heat removal from the refrigerated space and (*b*) the total rate of heat rejection to the ambient air.

Step 1 of 4

Given data:-

The temperature of source

$$T_H = 1700^\circ F$$
$$= 1700 + 460$$
$$= 2160R$$

The ambient air temperature

$$T_L = 80^\circ F + 460$$
$$= 540R$$

The amount of heat received $\dot{Q}_H = 700$ Btu/min

The amount of heat removed for the refrigerator

$$T_L = 20^\circ F$$
$$= 20 + 460$$
$$= 480R$$

Step 2 of 4

(a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency,

The thermal efficiency

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$
$$= 1 - \frac{540}{2160}$$
$$= 0.75$$

The maximum power output of a heat engine is expressed as

$$\dot{W}_{net} = \eta_{th} \dot{Q}_H$$
$$= 0.75(700)$$
$$= 525 \text{ Btu/min}$$

The power output from the heat engine will be power input to the refrigerant

Step 3 of 4

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends upon the temperature limits in the cycle only

i.e.

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_{Ref} = \frac{1}{T_H / T_L - 1}$$

$$COP_{Ref_max} = \frac{480}{540 - 480}$$

$$COP_{Ref_max} = 8$$

Therefore the rate of heat removal from the refrigerant is expressed as

$$\dot{Q}_{ref} = COP \dot{W}_{net}$$
$$= 8 \times 525$$
$$= 4200 \text{ Btu/min}$$

$$\boxed{\dot{Q}_{ref} = 4200 \text{ Btu/min}}$$

Step 4 of 4

(b) The total heat rejected to ambient

For refrigerator, heat rejected is expressed as

$$\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$$
$$= 525 + 4200$$

$$\dot{Q}_{H_ref} = 4725 \text{ Btu/min}$$

For heat engine

$$\dot{Q}_L = Q_H - \dot{W}_{net}$$
$$= 700 - 525$$
$$= 175 \text{ Btu/min}$$

Therefore the total heat rejected to ambient is

$$Q_{total} = Q_H + Q_{Href}$$

$$Q_{total} = 4725 + 175$$

$$\boxed{Q_{total} = 4900 \text{ Btu/min}}$$

Question: 7.104P

The structure of a house is such that it loses heat at a rate of 3800 kJ/h per °C difference between the indoors and outdoors. A heat pump that requires a power input of 4 kW is used to maintain this house at 24°C. Determine the lowest outdoor temperature for which the heat pump can meet the heating requirements of this house.

Step 1 of 3

Find the total amount of heat lost by the house.

$$Q_L = \dot{Q}_L (T_H - T_L)$$

Here, $\dot{Q}_L$ is the heat loss per unit degree, T_H is the temperature inside the house (higher in value), and T_L is the temperature outside the house (lower in value).

Substitute, $3800\ \text{kJ/h°K}$ for Q_L, and $(24°C+273)°K$ for T_H.

$$\begin{aligned} Q_L &= 3800\ \frac{\text{kJ}}{\text{h°K}}\left[(24°C+273)°K - T_L\right] \\ &= 3800\ \frac{\text{kJ}}{3600\text{s-°K}}\left[297 - T_L\right]°K \\ &= 1.0556\left[297 - T_L\right]\text{kW} \end{aligned}$$

Step 2 of 3

Find the coefficient of performance of the heat pump based on the heat and work.

$$COP_{HP} = \frac{Q_L}{W_{in}}$$

Here, W_{in} is the work input to the heat pump.

Substitute, $1.0556(297 - T_L)\text{kW}$ for Q_L, and 4 kW for W_{in}.

$$\begin{aligned} COP_{HP} &= \frac{1.0556(297 - T_L)\text{kW}}{4\ \text{kW}} \\ &= 0.26388(297 - T_L) \end{aligned}$$

Step 3 of 3

Find the coefficient of performance of the heat pump based on the temperature.

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

Substitute, $(24°C+273)°K$ for T_H, and $0.26388(297 - T_L)$ for COP_{HP}.

$$0.26388(297 - T_L) = \frac{(24°C+273)°K}{(24°C+273)°K - T_L}$$

$$0.26388(297 - T_L) = \frac{297}{(297 - T_L)}$$

$$(297 - T_L)^2 = \frac{297}{0.26388}$$

$$(297 - T_L) = \sqrt{\frac{297}{0.26388}}$$

$$\begin{aligned} T_L &= 297 - \sqrt{\frac{297}{0.26388}} \\ &= 297 - 33.548 \\ &= 263.452°K \\ &= -9.548°C \end{aligned}$$

Therefore, the lowest outdoor temperature for which the heat pump can meet the heating requirements of this house is at $\boxed{-9.548°C}$.

Question:

7.105P

An air-conditioner with refrigerant-134a as the working fluid is used to keep a room at 23°C by rejecting the waste heat to the outdoor air at 37°C. The room gains heat through the walls and the windows at a rate of 250 kJ/min while the heat generated by the computer, TV, and lights amounts to 900 W. The refrigerant enters the compressor at 400 kPa as a saturated vapor at a rate of 100 L/min and leaves at 1200 kPa and 70°C. Determine (*a*) the actual COP, (*b*) the maximum COP, and (*c*) the minimum volume flow rate of the refrigerant at the compressor inlet for the same compressor inlet and exit conditions.

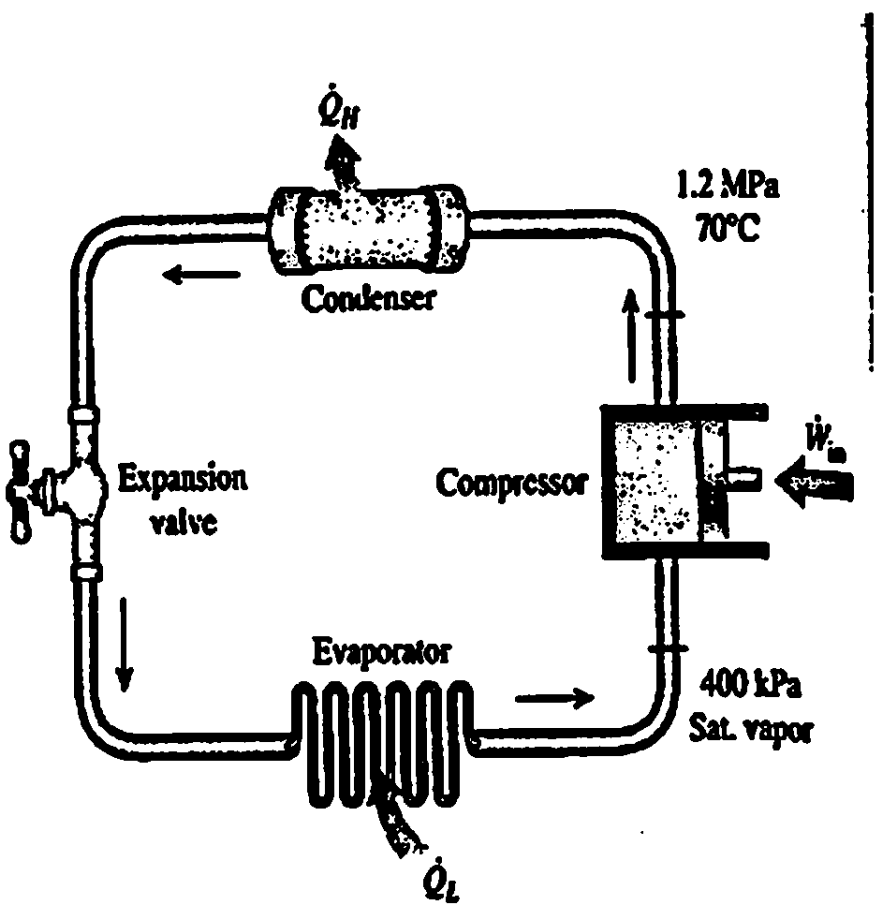


FIGURE P7–105

Step 1 of 8

Loading solution. Please wait ...

Step 2 of 8

From the saturated refrigerant R-134a temperature tables

The enthalpy at $P_1 = 400\ \text{kPa}, x_1 = 1$ is $h_1 = h_g = 255.55\ \text{kJ/kg}$

Specific volume, $v_1 = 0.051201\ \text{m}^3/\text{kg}$

From the superheated refrigerant R-134a tables

At $P_2 = 1.2\ \text{MPa}$ and $T_1 = 70°\text{C}$

Taking enthalpy is, $h_2 = 300.61\ \text{kJ/kg}$

Step 3 of 8

a) The mass flow rate of refrigerant

$$\dot{m}_R = \frac{\dot{V}_1}{v_1}\ \text{kg/s}$$

$$\dot{m}_R = \frac{100\ \text{L/min}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right)\left(\frac{1\ \text{min}}{60\text{s}}\right)}{0.0512}\ \text{kg/s}$$

$$\dot{m}_R = 0.03255\ \text{kg/s}$$

Step 4 of 8

The power consumption of the compressor is

$$\dot{W}_{in} = \dot{m}_R(h_2 - h_1)\text{kW}$$

$$\dot{W}_{in} = (0.03255)(300.61 - 255.55)\text{kW}$$

$$\dot{W}_{in} = 1.4667\text{kW}$$

Step 5 of 8

The heating load by the air conditioner

$$\dot{Q}_L = \dot{Q}_{room} + \dot{Q}_{eq}$$

$$\dot{Q}_L = 250\left(\frac{1\text{min}}{60s}\right) + 0.9$$

$$\dot{Q}_L = 5.067\ \text{kW}$$

We know that

$$COP_R = \frac{\dot{Q}_L}{W_{net}}$$

$$COP_R = \frac{5.067}{1.4667}$$

$$COP_R = 3.455$$

Therefore the COP of the refrigerator is $\boxed{COP_R = 3.455}$.

Step 6 of 8

b) The COP of the reversible refrigerator is expressed as

$$COP_{Rev} = \frac{T_L}{T_H - T_L}$$

$$COP_R = \frac{23+273}{37-23}$$

$$COP_{Rev} = 21.143$$

Therefore the COP of a reversible engine is $\boxed{COP_{Rev} = 21.143}$.

Step 7 of 8

c) The minimum volume flow rate of the compressor is expressed as

$$\dot{V}_{min} = \dot{m}_{min} \times v_1\ \text{m}^3/s$$

The minimum mass flow rate is obtained by

$$\dot{m}_r = \frac{\dot{W}_{min}}{(h_2 - h_1)}\ \text{kg/s}$$

Where

$$\dot{W}_{min} = \frac{Q_L}{COP_{Rev}}$$

$$\dot{W}_{min} = \frac{5.067}{21.14}$$

$$\dot{W}_{min} = 0.24\ \text{kW}$$

Step 8 of 8

Therefore

$$\dot{m}_{min} = \frac{0.24}{(300.61 - 255.55)}\ \text{kg/s}$$

$$\dot{m}_{min} = 0.00532\ \text{kg/s}$$

The minimum volume flow rate is

$$\dot{V}_{min} = 0.00532 \times 0.0512\ \text{m}^3/s$$

$$\dot{V}_{min} = 2.723 \times 10^{-4} \times 60 \times 1000\ \text{L/min}$$

$$\dot{V}_{min} = 16.36\ \text{L/min}$$

Therefore the minimum volume flow rate $\boxed{\dot{V}_{min} = 16.36\ \text{L/min}}$.

Question: 7.106P

Calculate and plot the COP of a completely reversible refrigerator as a function of the temperature of the sink up to 500 K with the temperature of the source fixed at 250 K.

Step 1 of 4

Given

Temperature of source T_H varies from 300K to 500K

Temperature of sink fixed at $T_L = 250K$

Step 2 of 4

The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends upon the temperature limits in the cycle only

$$COP_R = \frac{\text{Desired output}}{\text{work input}}$$

$$= \frac{T_L}{T_H - T_L}$$

$$COP_R = \frac{250}{T_H - 250}$$

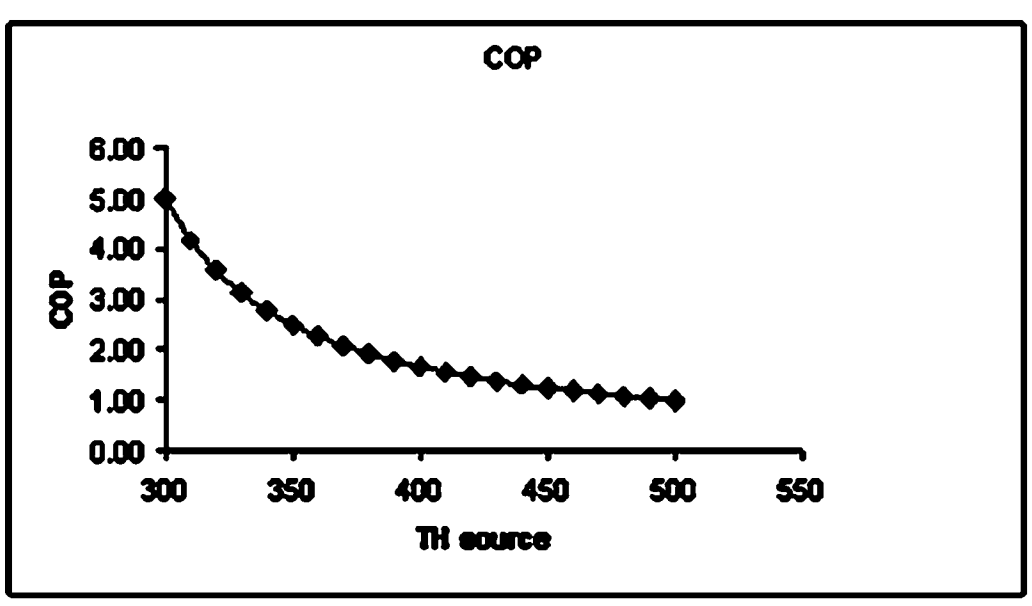
Step 3 of 4

Calculating COP of the reversible refrigerator by varying the temperature of the source (T_H) between 300K to 500K tabulated as follows

TH	COP
300	5.00
310	4.17
320	3.57
330	3.13
340	2.78
350	2.50
360	2.27
370	2.08
380	1.92
390	1.79
400	1.67
410	1.56
420	1.47
430	1.39
440	1.32
450	1.25
460	1.19
470	1.14
480	1.09
490	1.04
500	1

Step 4 of 4

Plotting the graph between COP of the refrigerator and source temperature



Question: 7.107P

Derive an expression for the COP of a completely I reversible refrigerator in terms of the thermal energy reservoir temperatures, *TL* and *TH*.

Step 1 of 2

Let the thermal energy reservoirs is T_L and T_H

The efficiency of a refrigerator is expressed in terms of the coefficient of performance (COP), denoted by COP_R. A refrigerator or a heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator.

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

$$COP_R = \frac{\dot{Q}_L}{W_{net}}$$

Step 2 of 2

The objective of a refrigerator is to remove heat (Q_L) from the refrigerated space. To accomplish this objective, it requires a work input of W_{net}

$$W_{net} = Q_H - Q_L$$

The coefficient of performance of any refrigerator is expressed as

$$COP_R = \frac{Q_L}{Q_H - Q_L}$$

$$COP_R = \frac{1}{Q_H/Q_L - 1}$$

For a reversible cycle the $\frac{Q_H}{Q_L}$ ratio can be replaced by $\frac{T_H}{T_L}$

$$COP_R = \frac{1}{T_H/T_L - 1}$$

From the above relation, by reducing the value of T_H, or by increasing the value of T_L, the COP of a Carnot refrigerator can be increased.

Question: 7.108P

A heat engine operates with a source at 1280 K and a sink at 290 K. What is the maximum work per unit of heat that the engine can remove from the source?

Step 1 of 3

Given

Source temperature $T_H = 1280\text{K}$

Sink temperature $T_L = 290\text{K}$

Step 2 of 3

The Carnot cycle efficiency is the highest possible efficiency of a heat engine

So the efficiency of Carnot cycle is

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th} = 1 - \frac{290}{1280}$$

$$\eta_{th} = 0.773$$

Step 3 of 3

We know thermal efficiency of an engine is

$$\eta_{th} = \frac{W}{Q_H}$$

Where W is the maximum work output from the engine

Q_H is the heat removed from the source

Therefore the work per unit of heat that the engine can remove from the source is

$$\frac{W}{Q_H} = \eta_{th} = 0.773$$

$$\boxed{\frac{W}{Q_H} = 0.773}$$

Question: 7.109P

A heat engine receives heat from a heat source at 1200°C and rejects heat to a heat sink at 50°C. The heat engine does maximum work equal to 500 kJ. Determine the heat supplied to the heat engine by the heat source and the heat rejected to the heat sink.

Step 1 of 4

Given

Temperature of source $T_H = 1200^0\text{C}$

$= 1200 + 273$

$T_H = 1473\text{K}$

Temperature of sink $T_L = 50^0\text{C}$

$= 50 + 273$

$T_L = 323\text{K}$

The maximum work output from the heat engine is $W_{max} = 500\text{ kJ}$

Step 2 of 4

We know the Carnot efficiency is the highest possible efficiency of a heat engine

Efficiency of Carnot heat engine is

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th} = 1 - \frac{323}{1473}$$

$$\eta_{th} = 0.7807$$

Step 3 of 4

Thermal efficiency of Carnot heat engine can also be written as

$$\eta_{th} = \frac{W_{max}}{Q_H}$$

The amount of heat supplied to the heat engine by the heat source

$$Q_H = \frac{W_{max}}{\eta_{th}}$$

$$Q_H = \frac{500}{0.7807}$$

$$\boxed{Q_H = 640.45\text{ kJ}}$$

Step 4 of 4

Similarly the amount of heat rejected to the heat sink can be calculated by applying first law of thermodynamics to the heat engine

$$Q_H - Q_L = W_{max}$$

$$Q_L = Q_H - W_{max}$$

$$= 640.45 - 500$$

$$\boxed{Q_L = 140.45\text{ kJ}}$$

Question: 7.110P

A heat pump creates a heating effect of 32,000 Btu/h for a space maintained at 530 R while using 1.8 kW of electrical power. What is the minimum temperature of the source that satisfies the second law of thermodynamics?

Step 1 of 4

Given

Temperature of source $T_H = 530\text{R}$

The heating effect is $\dot{Q}_H = 32{,}000\ \text{Btu/h}$

The minimum power required is $\dot{W}_{net} = 1.8\ \text{kW}$

Step 2 of 4

The minimum power required is

$\dot{W}_{net} = 1.8\ \text{kW}$

Since $1\ \text{kW} = 3412.14\ \text{Btu/h}$

$\dot{W}_{net} = 1.8 \times 3412.14$

$\dot{W}_{net} = 6141.852\ \text{Btu/h}$

Step 3 of 4

The COP of a reversible heat pump can be calculated as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$= \frac{32000}{6141.852}$$

$$COP_{HP} = 5.21$$

Step 4 of 4

The COP of a reversible heat pump can also be calculated as

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$5.21 = \frac{530}{530 - T_L}$$

$$T_L = 428.272\text{R}$$

Therefore the minimum temperature of the heat source is

$$\boxed{T_L = 428.272\text{R}}$$

Question: 7.111P

A heat pump with a COP of 2.4 is used to heat a house. When running, the heat pump consumes 8 kW of electric power. If the house is losing heat to the outside at an average rate of 40,000 kJ/h and the temperature of the house is 3°C when the heat pump is turned on, determine how long it will take for the temperature in the house to rise to 22°C. Assume the house is well sealed (i.e., no air leaks) and take the entire mass within the house (air, furniture, etc.) to be equivalent to 2000 kg of air.

Step 1 of 2

Calculate the amount of heat lost in the room.

$$\mathrm{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$

$$\dot{Q}_H = \mathrm{COP}_{HP} \times \dot{W}_{in}$$

Here, COP_{HP} is the Coefficient of performance of heat pump, and $\dot{W}_{in}$ is the work input to the compressor.

Substitute 2.4 for COP_{HP} and 8 kW for $\dot{W}_{in}$.

$$\dot{Q}_H = 2.4(8)$$
$$= 19.2 \text{ kW}$$

Convert heat loss to surroundings in kW.

$$\dot{Q}_{loss} = 40000 \text{ kJ/h}$$
$$= 400000 \times \frac{1}{3600} \text{ kW}$$
$$= 11.11 \text{ kW}$$

Step 2 of 2

Considering the house as a system.

Apply the energy balance of the system.

$$E_{in} - E_{out} = \Delta E_{system}$$

Neglect the kinetic and potential energy changes.

Write the equation for internal energy.

$$\Delta U = m(u_2 - u_1)$$
$$= mc_v(T_2 - T_1)$$

Therefore,

$$\left(\dot{Q}_H - \dot{Q}_{loss}\right)\text{kJ/s} = mc_v(T_2 - T_1)\text{ kJ}$$
$$\left(\dot{Q}_H - \dot{Q}_{loss}\right)t = mc_v(T_2 - T_1)$$

Substitute 19.2 kW for $\dot{Q}_H$, 11.11 kW for $\dot{Q}_{loss}$, 2000 kg for m, 0.718 kJ/kg·K for c_v , 22°C for T_2 , and 3°C for T_1 .

$$\left(19.2 - \frac{40,000}{3600}\right)t = 2000(0.718)(22-3)$$
$$8.09t = 27284$$
$$t = \frac{27284}{8.09}$$
$$\Delta t = 3372.55 \text{ s}$$
$$= 3372.5 \times \frac{1}{3600}$$
$$= 0.936 \text{ h}$$

Therefore, the time taken to rise the temperature is $\boxed{\Delta t = 0.936 \text{ h}}$.

Question: 7.112P

A refrigeration system uses water-cooled condenser for rejecting the waste heat. The system absorbs heat from a space at 25°F at a rate of 24,000 Btu/h. Water enters the condenser at 58°F at a rate of 1.45 lbm/s. The COP of the system is estimated to be 1.77. Determine (*a*) the power input to the system, in kW, (*b*) the temperature of the water at the exit of the condenser, in °F and (*c*) the maximum possible COP of the system. The specific heat of water is 1.0 Btu/bm·°F.

Step 1 of 4

Given data:

The cooling load of the refrigerator is $\dot{Q}_L = 24{,}000 \text{ Btu/h}$.

The COP of the Refrigerator is $COP = 1.77$.

The temperature of sink $T_L = 25°\text{F}$

$= 25 + 460$

$T_L = 485\text{R}$.

Temperature of the water entering into the condenser $T_1 = 58°\text{F}$

$= 58 + 460$

$T_1 = 518\text{R}$.

The mass flow rate of water $\dot{m} = 1.45 \text{ lbm/s}$.

Specific heat of water $c_p = 1 \text{ Btu/lbm.°F}$.

Step 2 of 4

(a) The power input to the system

The efficiency of a refrigerator is expressed in terms of the coefficient of performance (COP), denoted by COP_R. The objective of a refrigerator is to remove heat (Q_L) from the refrigerated space. To accomplish this objective, it requires a work input of $\dot{W}_{net}$

$$COP_R = \frac{\text{Desired effect}}{\text{work input}}$$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net}}$$

Therefore power input is

$$\dot{W}_{net} = \frac{\dot{Q}_L}{COP_R}$$
$$= \frac{24000}{1.77}$$
$$= 13559.3 \text{ Btu/h}$$
$$= \frac{13559.3 \text{ Btu}}{3600 \text{ s}} \frac{1.055 \text{ kJ}}{\text{Btu}}$$

$$\boxed{\dot{W}_{net} = 3.974 \text{ kW}}$$

Therefore power input to the system is $\dot{W}_{net} = 3.974 \text{ kW}$

Step 3 of 4

(b) The exit temperature of water from the condenser

The heat rejected in the condenser

$$\dot{Q}_H = \dot{m} c_p (T_2 - T_1)$$

By applying first law of thermodynamics to the system

$$\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$$

Therefore the rate of heat rejection is

$$\dot{Q}_H = 13559.3 + 24000$$

$$\dot{Q}_H = 37559.3 \text{ Btu/h}$$

By substituting the known values in the equation

$$\dot{Q}_H = \dot{m} c_p (T_2 - T_1)$$
$$\frac{37559.3}{3600} = 1.45 \times (1) \times (T_2 - 58)$$
$$T_2 = 65.195°\text{F}$$

Therefore the exit temperature of water in the condenser is

$$\boxed{T_2 = 65.2°\text{F}}$$

Step 4 of 4

(c) The maximum possible COP of the refrigerator

$$COP_{rev} = \frac{T_L}{T_H - T_L}$$

Taking the average temperature of water as source temperature

$$T_H = \frac{T_1 + T_2}{2}$$
$$= \frac{58 + 65.2}{2}$$
$$T_H = 61.6°\text{F}$$

Therefore the maximum COP of the refrigerator

$$COP_{rev} = \frac{485}{61.6 - 25}$$
$$COP_{rev} = 13.25$$

Therefore the maximum possible COP of the system is $\boxed{COP_{rev} = 13.25}$

$$= \frac{58 + 65.2}{2}$$
$$T_H = 61.6°\text{F}$$

Therefore the maximum COP of the refrigerator

$$COP_{rev} = \frac{485}{61.6 - 25}$$
$$COP_{rev} = 13.25$$

Therefore the maximum possible COP of the system is $\boxed{COP_{rev} = 13.25}$

7.113P

Question:

Consider a Carnot heat-engine cycle executed in a closed system using 0.01 kg of refrigerant-134a as the working fluid. The cycle has a thermal efficiency of 15 percent, and the refrigerant-134a changes from saturated liquid to saturated vapor at 50°C during the heat addition process. Determine the net work output of this engine per cycle.

Step 1 of 3

Given data:

Mass of refrigerator $m = 0.01\,\text{kg}$

The thermal efficiency of cycle, $\eta_{th} = 15\%$

The saturated temperature is $T_{sat} = 50°C$

Refrigerant R-134a

Step 2 of 3

From saturated refrigerant R-134a table at $T_{sat} = 50°C$

Question:

Consider a Carnot heat-engine cycle executed in a closed system using 0.01 kg of refrigerant-134a as the working fluid. The cycle has a thermal efficiency of 15 percent, and the refrigerant-134a changes from saturated liquid to saturated vapor at 50°C during the heat addition process. Determine the net work output of this engine per cycle.

Step 1 of 3

Given data:

Mass of refrigerator $m = 0.01\,\text{kg}$

The thermal efficiency of cycle, $\eta_{th} = 15\%$

7.114P

Question:

A heat pump with a COP of 2.8 is used to heat an air-tight house. When running, the heat pump consumes 5 kW of power. If the temperature in the house is 7°C when the heat pump is turned on, how long will it take for the heat pump to raise the temperature of the house to 22°C? Is this answer realistic or optimistic? Explain. Assume the entire mass within the house (air, furniture, etc.) is equivalent to 1500 kg of air.

Step 1 of 4

Given data:

COP of heat pump, $\text{COP} = 2.8$

The power consumed, $\dot{W}_{net} = 5\ \text{kW}$

The initial temperature of house, $T_1 = 7°\text{C}$

The final temperature of house, $T_2 = 22°\text{C}$

The total mass in the house, $m = 1500\ \text{kg}$

Step 2 of 4

The specific heat at constant volume for air $c_v = 0.718\ \text{kJ/kg}\cdot°\text{C}$

The amount of heat lost in the room is

$$\text{COP}_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$

$$\dot{Q}_H = 2.8(5)$$

$$\dot{Q}_H = 14\ \text{kW}$$

Step 3 of 4

The total amount of heat to be supplied to the house

$$Q_{in} = mc_v\Delta T\ \text{kJ}$$

$$Q_{in} = 1500(0.718)(22-7)\ \text{kJ}$$

$$Q_{in} = 16{,}155\ \text{kJ}$$

Step 4 of 4

The heat pump supplies 14 kW of heat into the house per second.

The time required to supply that heat is

$$\Delta t = \frac{16{,}155}{14}$$

$$= 1153.928\ s$$

$$= \frac{1153.928}{60}\text{min}$$

$$= 19.23\ \text{min}$$

$\therefore$ Time taken to heat the pump, $\boxed{\Delta t = 19.23\ \text{min}}$

Question: 7.115P

Consider a Carnot heat-engine cycle executed in a closed system using 0.025 kg of steam as the working fluid. It is known that the maximum absolute temperature in the cycle is twice the minimum absolute temperature, and the net work output of the cycle is 60 kJ. If the steam changes from saturated vapor to saturated liquid during heat rejection, determine the temperature of the steam during the heat rejection process.

Step 1 of 2

The maximum absolute temperature (T_H) in the cycle is twice the minimum absolute temperature (T_L) in the cycle $T_H = 0.5T_L$.

Find the efficiency of the Carnot cycle heat engine.

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th} = 1 - \frac{T_L}{0.5T_L}$$

$$\eta_{th} = 0.5$$

Here, T_H is the maximum absolute temperature and T_L is the minimum absolute temperature.

The efficiency of the Carnot heat engine can also be written as:

$$\eta_{th} = \frac{W_{net}}{Q_H}$$

$$0.5 = \frac{60}{Q_H}$$

$$Q_H = 120\,\text{kJ}$$

Here, W_{net} is the net work done and Q_H is the heat supplied.

Apply the first law of thermodynamics to the system

$$W_{net} = Q_H - Q_L$$

$$60 = 120 - Q_L$$

$$Q_L = 60\,\text{kJ}$$

Step 2 of 2

The amount of heat rejection per kg of steam is

$$q_L = \frac{Q_L}{m}$$

$$= \frac{60}{0.025}$$

$$q_L = 2400\ \text{kJ/kg}$$

For 1 kg of conversion of saturated vapor to saturated liquid it takes 2400 kJ/kg of heat

i.e. enthalpy of vaporization

$$q_L = h_{fg} = 2400\ \text{kJ/kg}$$

At $h_{fg} = 2400\ \text{kJ/kg}$ from saturated water tables

$$T_L = 40 + (45 - 40)\left(\frac{2400 - 2406.0}{2394.0 - 2406.0}\right)$$

$$T_L = 42.5^\circ\text{C}$$

Therefore the temperature of the steam during the heat rejection process is $\boxed{T_L = 42.5^\circ\text{C}}$

Question: 7.116P

Reconsider Prob. 7–115. Using EES (or other) software, investigate the effect of the net work output on the required temperature of the steam during the heat rejection process. Let the work output vary from 40 to 60 kJ.

Step 1 of 4

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 4

Equation window:

```
"Given"
m=0.025 [kg]
RatioT=0.5 "RatioT=T_L/T_H"
"W_net_out=60 [kJ]"

"Properties"
FluidS= 'steam_iapws'
h_f=enthalpy(FluidS, T=T_L, x=0)
h_g=enthalpy(FluidS, T=T_L, x=1)
h_fg=h_g-h_f

"Analysis"
eta_th=1-RatioT
eta_th=W_net_out/Q_H
Q_L=Q_H-W_net_out
Q_L=m*h_fg
```

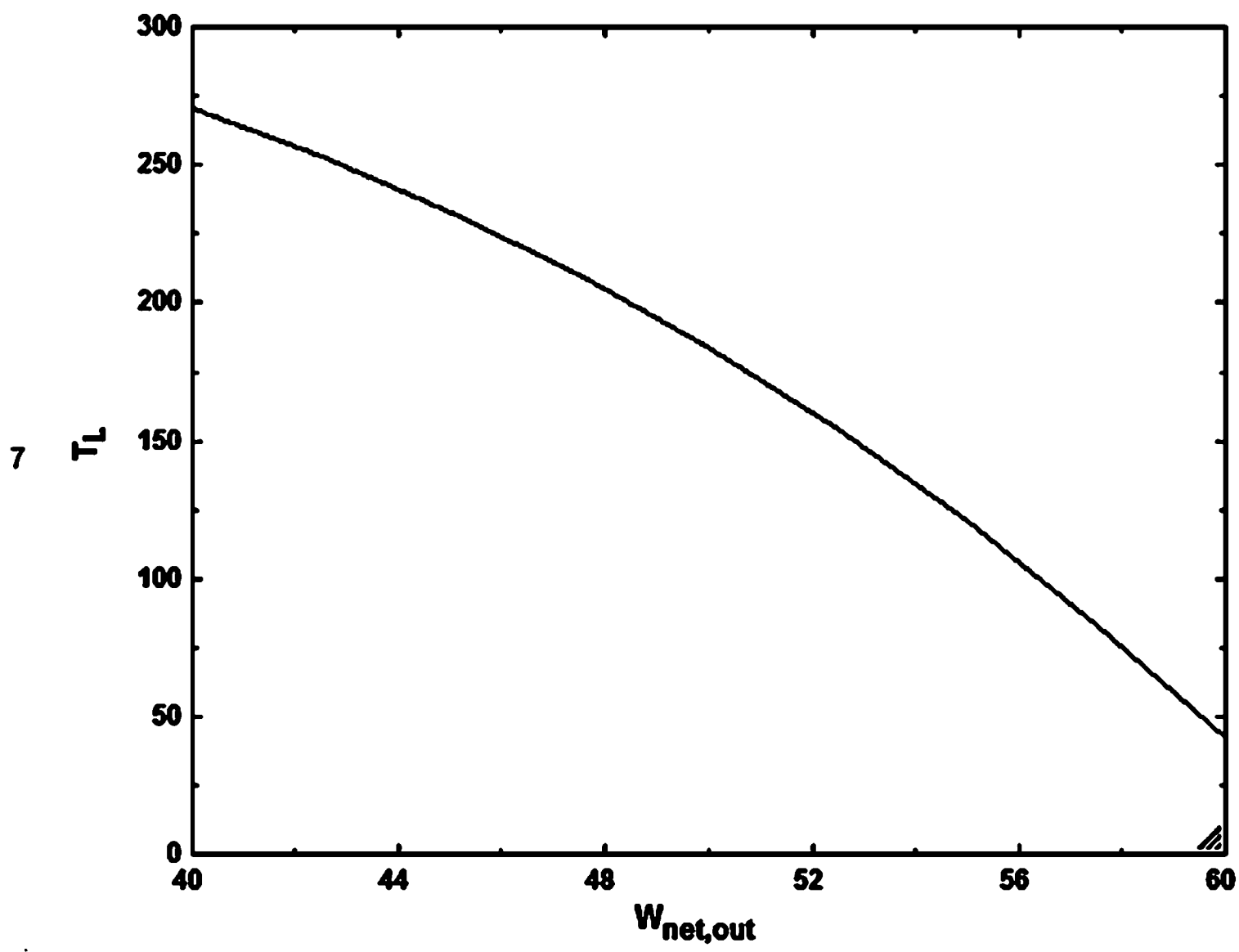
Step 3 of 4

Table for sink temperature as a function of Work input:

1..9	1 $W_{net,out}$	2 T_L
Run 1	40	270.8
Run 2	42.5	252.9
Run 3	45	232.8
Run 4	47.5	209.9
Run 5	50	184
Run 6	52.5	154.5
Run 7	55	120.9
Run 8	57.5	83.18
Run 9	60	42.49

Step 4 of 4

Graph between Sink temperatures Vs. Work input:



Question: 7.117P

Consider a Carnot refrigeration cycle executed in a closed system in the saturated liquid–vapor mixture region using 0.96 kg of refrigerant-134a as the working fluid. It is known that the maximum absolute temperature in the cycle is 1.2 times the minimum absolute temperature, and the net work input to the cycle is 22 kJ. If the refrigerant changes from saturated vapor to saturated liquid during the heat rejection process, determine the minimum pressure in the cycle.

Step 1 of 6

Given data:

The mass of refrigerant R-134a, $m = 0.96\ \text{kg}$

The maximum temperature, $T_H = 1.2T_L$

The net work input to the cycle, $\dot{W}_{net} = 22\ \text{kJ}$

Step 2 of 6

The maximum coefficient of performance of any refrigerator is expressed as

$$\text{COP}_{Ref} = \frac{T_L}{T_H - T_L}$$

The we can also written as $\text{COP}_{Ref} = \frac{\dot{Q}_L}{\dot{W}_{net}}$

$$\text{Here COP}_{Ref} = \frac{T_L}{1.2T_L - T_L} = \frac{\dot{Q}_L}{22}$$

$$\frac{T_L}{1.2T_L - T_L} = \frac{\dot{Q}_L}{22}$$

$$\frac{1}{0.2} = \frac{\dot{Q}_L}{22}$$

$$\dot{Q}_L = 110\ \text{kJ}$$

Step 3 of 6

The refrigerant effect is, $\dot{Q}_L = 110\ \text{kJ}$

The heat rejected in the conversion is

$$\dot{Q}_H = \dot{W}_{net} + \dot{Q}_L$$

$$Q_H = 22 + 110$$

$$Q_H = 132\ \text{kJ}$$

Step 4 of 6

Per unit kg of steam

Heat lost in this process,

$$q_{out} = \frac{Q_H}{m}$$

$$= \frac{132}{0.96}$$

$$= 137.5\ \text{kJ/kg}$$

For 1 kg of conversion of saturated vapor to saturated liquid it takes 137.5 kJ/kg of heat

i.e. Enthalpy of vaporization $h_{fg} = 137.5\ \text{kJ/kg}$

Step 5 of 6

The temperature corresponds to $h_{fg} = 137.5\ \text{kJ/kg}$ from saturated refrigerant R-134a tables

Using method of interpolation

$$T_H = 60 + (65 - 60)\left[\frac{137.5 - 139.10}{132.02 - 139.10}\right]$$

$$T_H = 61.129°\text{C}$$

Therefore the lower temperature reservoir is

$$T_L = \frac{T_H}{1.2}$$

$$= \frac{61.129 + 273}{1.2}$$

$$= 278.44\text{K} - 273$$

$$= 5.44°\text{C}$$

Step 6 of 6

The saturated pressure at minimum temperature is

At $T_L = 5.44°\text{C}$ the minimum pressure is lays between

$T_1 = 4°\text{C}$, the $P_{sat_1} = 337.9\ \text{kPa}$

$T_2 = 6°\text{C}$, the $P_{sat_2} = 362.23\ \text{kPa}$

By interpolation method

$$\frac{P_{sat} - P_1}{P_2 - P_1} = \frac{T_{sat} - T_1}{T_2 - T_1}$$

$$\frac{P_{sat} - 337.9}{363.23 - 337.9} = \frac{5.44 - 4}{6 - 4}$$

$$P_{sat} = 355.4\ \text{kPa}$$

∴ The minimum pressure is $\boxed{P_{sat} = 355.4\ \text{kPa}}$

Question: 7.118P

Reconsider Prob. 7–117. Using EES (or other) software, investigate the effect of the net work input on the minimum pressure. Let the work input vary from 10 to 30 kJ. Plot the minimum pressure in the refrigeration cycle as a function of net work input, and discuss the results.

Step 1 of 4

Solve the solution by entering the following code in 'Equations window' of EES software.

EES Limited Academic Version: C:\EES (LIMITED ACADEMIC)\HELLO.TXT - [Equations Window]

File Edit Search Options Calculate Tables Plots Windows Help Thermal-Fluid Sciences

```
m_R134a=0.96 "Here, m_R134a is the mass of R134a"
THtoTLRatio=1.2 "Here, TH is the higher temperature and TL is the lower temperature"
COP_R=1/(THtoTLRatio-1) "Here, COP_R is the COP of regrigerator"
Q_L=W_in*COP_R "Here, Q_L is the heat rejected and W_in is the work input"
Q_L+W_in=Q_H "Here, Q_H is the heat added"
Q_H=(m_R134a)*(h_fg) "Here, h_fg is the specific enthalpy of R134a at wet region"
h_fg=Enthalpy(R134a,T=T_H,x=1)-Enthalpy(R134a,T=T_H,x=0) "Here, x is the dryness fraction"
T_H=(THtoTLRatio)*T_L
P_min=Pressure(R134a,T=T_L,x=0)
```

Step 2 of 4

Obtain the parametric table for the code entered in the Equations window' of EES software.

Parametric Table

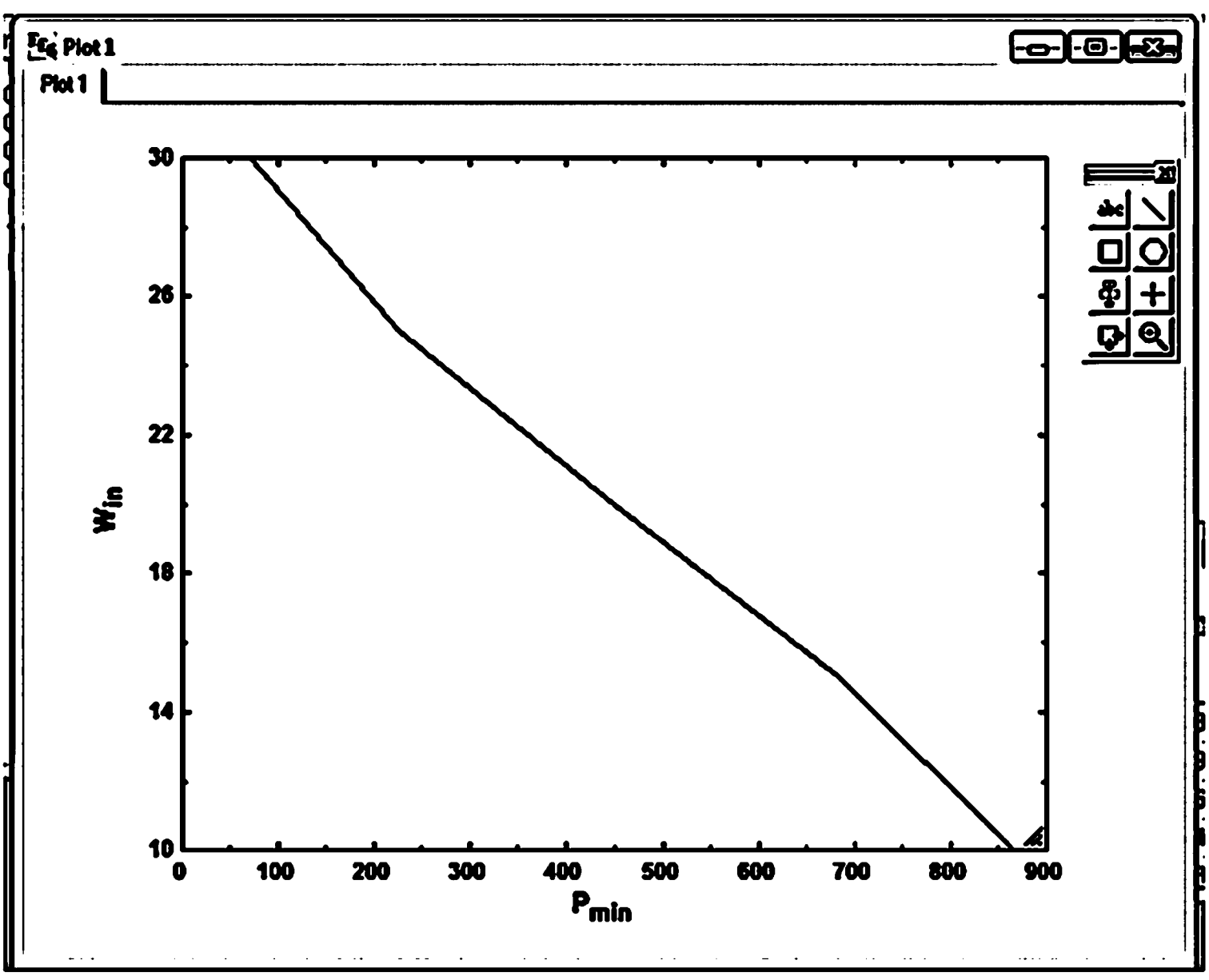
Table 1

1..5	1 P_{min}	2 T_H	3 T_L	4 W_{in}
Run 1	866	368.7	307.3	10
Run 2	683.4	358.8	299	15
Run 3	450.3	342.7	285.6	20
Run 4	225.3	319.3	266.1	25
Run 5	70.01	287.1	239.3	30

Here, T_H values must be kept in limits at "Variable info" for execution of the EES source file.

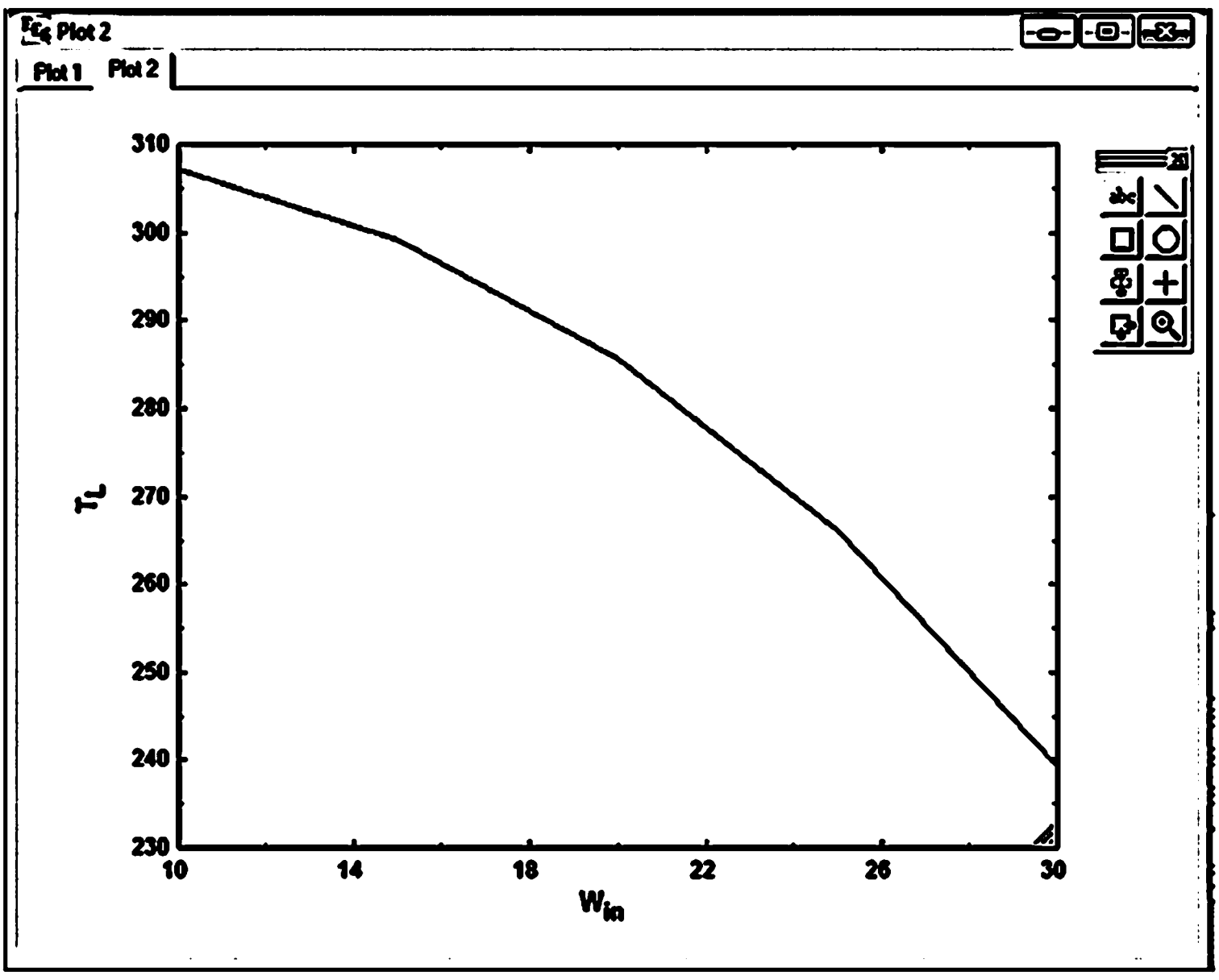
Step 3 of 4

Obtain the plot for minimum pressure and work input from the EES software.



Step 4 of 4

Obtain the plot for lower temperature and work input from the EES software.



Question: 7.119P

Consider two Carnot heat engines operating in series. The first engine receives heat from the reservoir at 1800 K and rejects the waste heat to another reservoir at temperature *T*. The second engine receives this energy rejected by the first one, converts some of it to work, and rejects the rest to a reservoir at 300 K. If the thermal efficiencies of both engines are the same, determine the temperature *T*.

Step 1 of 3

Given data:

HE-1:

Source temperature, $T_H = 1800\,\text{K}$

Sink temperature, $T_L = T$

HE-2:

Source temperature, $T_H = T$

Sink temperature, $T_L = 300\,\text{K}$

Step 2 of 3

The thermal efficiency of Carnot heat engine is given by

$$\eta_{th} = \frac{T_H - T_L}{T_H}$$

For HE-1:

$$\eta_{th_1} = \frac{T_H - T}{T_H}$$

For HE-2:

$$\eta_{th_2} = \frac{T - T_L}{T}$$

Step 3 of 3

Given that the both heat engines having same thermal efficiency

$$\eta_{th_1} = \eta_{th_2}$$

$$\frac{T_H - T}{T_H} = \frac{T - T_L}{T}$$

$$T_H \times T - T^2 = T_H \times T - T_L T_H$$

$$T^2 = T_L T_H$$

$$T = \sqrt{T_L T_H}$$

$$T = \sqrt{1800 \times 300}$$

$$T = 734.84\,\text{K}$$

The intermittent temperature is $\boxed{T = 734.84\,\text{K}}$

Question: 7.120P

Calculate and plot the thermal efficiency of a completely reversible heat engine as a function of the source temperature up to 2000 R with the sink temperature fixed at 500 R.

Step 1 of 3

Given data:

The temperature of source, $T_H = 0 - 2000\text{R}$

The temperature of sink, $T_L = 500\text{R}$

Step 2 of 3

The thermal efficiency of Carnot engine is expressed as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

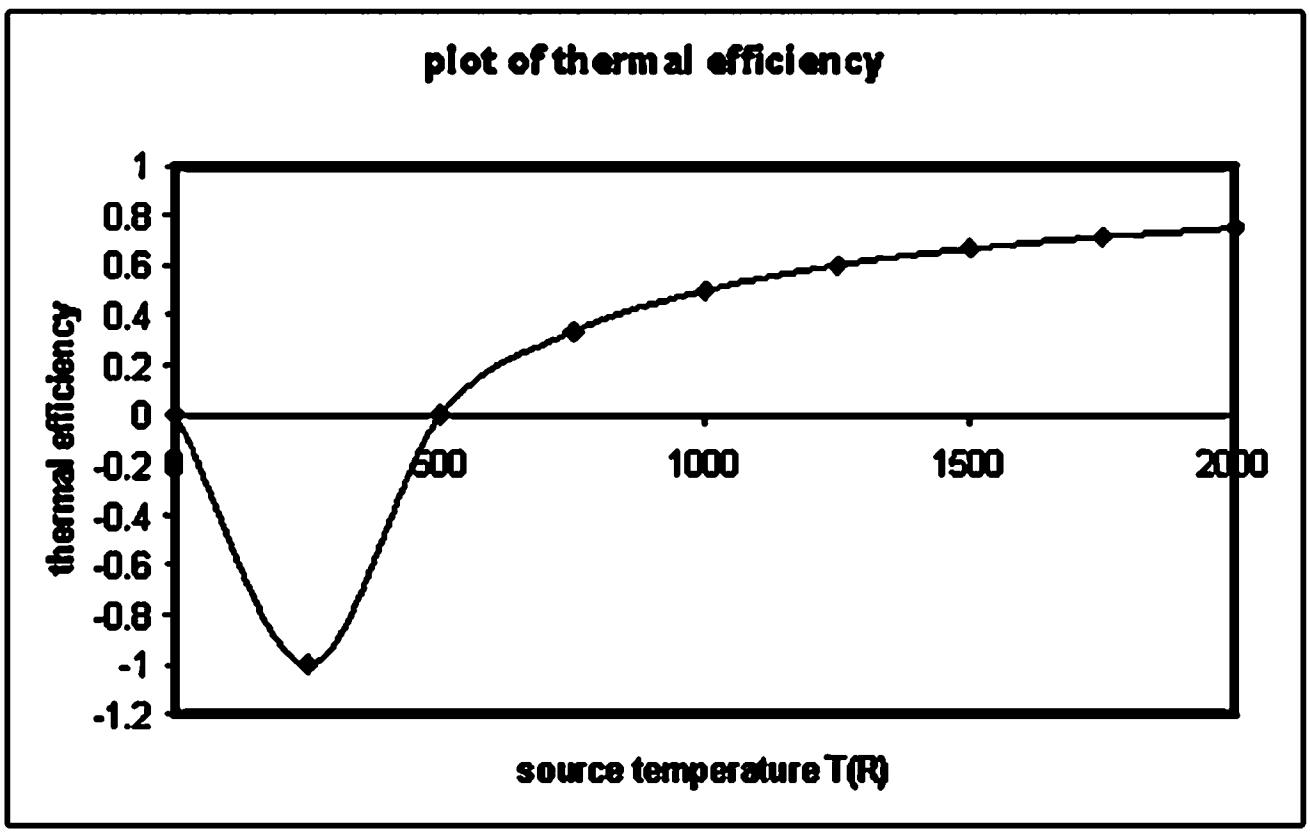
$$\eta_{th} = 1 - \frac{500}{T_H}$$

Finding efficiency of heat engine with the increment of source temperature is 250R

Source T(R)	Thermal efficiency
0	#DIV/0!
250	-1.00
500	0.00
750	0.33
1000	0.50
1250	0.60
1500	0.67
1750	0.71
2000	0.75

Step 3 of 3

Graph between source temperature and thermal efficiency



Question:

7.121P

A Carnot heat engine receives heat at 900 K and rejects the waste heat to the environment at 300 K. The entire work output of the heat engine is used to drive a Carnot refrigerator that removes heat from the cooled space at −15°C at a rate of 250 kJ/min and rejects it to the same environment at 300 K. Determine (*a*) the rate of heat supplied to the heat engine and (*b*) the total rate of heat rejection to the environment.

Step 1 of 6

Given data:

For the heat engine

Temperature of source $T_H = 900\text{K}$

Temperature of sink $T_L = 300\text{K}$

For the refrigerator

Temperature of source $T_{H'} = 300\text{K}$

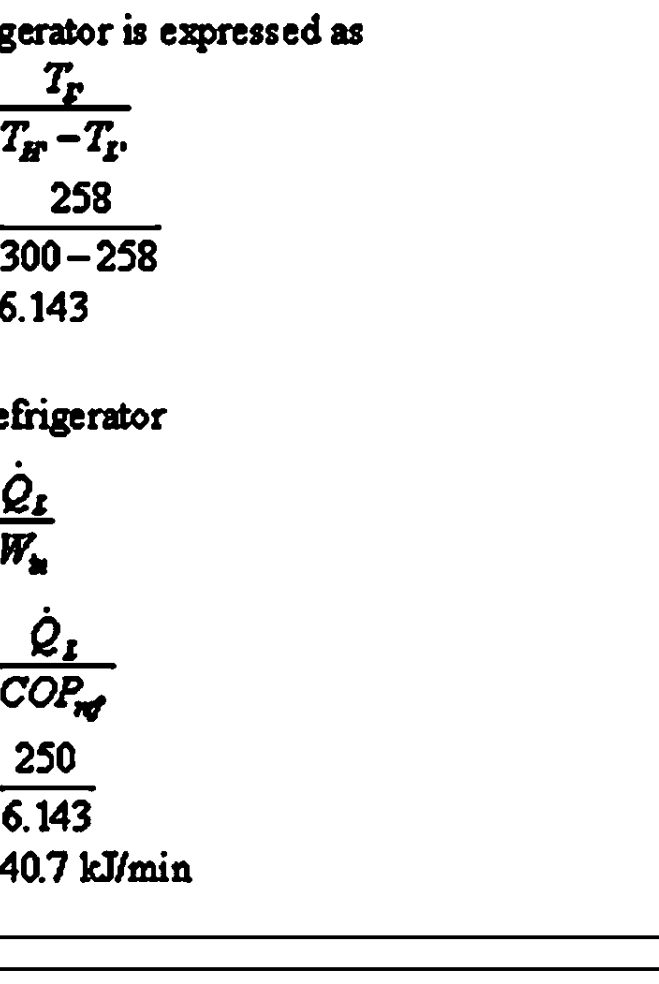
Temperature of sink

$T_{L'} = -15^0\text{C}$

$T_{L'} = 258\text{K}$

The heat removed from the refrigerator $\dot{Q}_{L'} = 250\ \text{kJ/min}$

Step 2 of 6



Where

Q_H Is the heat absorbed from the source to the heat engine

Q_L is the heat rejected from the heat engine to the sink

$Q_{H'}$ Is the heat absorbed from the source to the refrigerator

$Q_{L'}$ Is the heat rejected from the refrigerator to the sink

W_{in} Is the power input to the refrigerator

W_{out} Is the power output from the heat engine

Step 3 of 6

(a)

The COP of Carnot refrigerator is expressed as

$$COP_{ref} = \frac{T_{L'}}{T_{H'} - T_{L'}}$$

$$= \frac{258}{300 - 258}$$

$$COP_{ref} = 6.143$$

The power input to the refrigerator

$$COP_{ref} = \frac{\dot{Q}_L}{W_{in}}$$

$$W_{in} = \frac{\dot{Q}_L}{COP_{ref}}$$

$$= \frac{250}{6.143}$$

$$W_{in} = 40.7\ \text{kJ/min}$$

Step 4 of 6

The power input to the refrigerator is equal to power output from the heat engine

$$W_{in} = W_{out}$$

The thermal efficiency of the Carnot heat engine is expressed as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$= 1 - \frac{300}{900}$$

$$\eta_{th} = 0.667$$

The rate of heat supplied to the heat engine is

$$\eta_{th} = \frac{W_{out}}{\dot{Q}_H}$$

$$0.667 = \frac{40.7}{\dot{Q}_H}$$

$$\dot{Q}_H = 61.1\ \text{kJ/min}$$

Therefore heat supplied to the heat engine is

$$\boxed{\dot{Q}_H = 61.1\ \text{kJ/min}}$$

Step 5 of 6

(b)

The total heat rejection will be sum of the heat rejected by the heat engine and heat rejected by the refrigerant

$$\dot{Q}_{total} = \dot{Q}_L + \dot{Q}_{H'}$$

For heat engine the heat rejection is

$$\dot{Q}_L = \dot{Q}_H - W_{out}$$

$$\dot{Q}_L = 61.1 - 40.7$$

$$\dot{Q}_L = 20.4\ \text{kJ/min}$$

Step 6 of 6

For refrigerator the heat rejection is

$$\dot{Q}_{L'} - \dot{Q}_{H'} = -W_{in}$$

$$\dot{Q}_{H'} = W_{in} + \dot{Q}_{L'}$$

$$= 250 + 40.7$$

$$\dot{Q}_{H'} = 290.7\ \text{kJ/min}$$

The total heat rejection is

$$\dot{Q}_{total} = \dot{Q}_L + \dot{Q}_{H'}$$

$$\dot{Q}_{total} = 290.7 + 20.4$$

$$\boxed{\dot{Q}_{total} = 311.1\ \text{kJ/min}}$$

Therefore the total heat rejection to the environment is $\dot{Q}_{total} = 311.1\ \text{kJ/min}$

Question:

Reconsider Prob. 7–121. Using EES (or other) software, investigate the effects of the heat engine source temperature, the environment temperature, and the cooled space temperature on the required heat supply to the heat engine and the total rate of heat rejection to the environment. Let the source temperature vary from 500 to 1000 K, the environment temperature vary from 275 to 325 K, and the cooled space temperature vary from −20 to 0°C. Plot the required heat supply against the source temperature for the cooled space temperature of −15°C and environment temperatures of 275, 300, and 325 K, and discuss the results.

Step 1 of 7

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 7

Equation window:

```
Q_dot_L_R = 250 [kJ/min]
T_surr = 325 [K]
"T_H = 900 [K]"
T_L_C = -15 [C]
T_L =T_L_C+ 273

"Coefficient of performance of the Carnot refrigerator:"
T_H_R = T_surr
COP_R = 1/(T_H_R/T_L-1)
"Power input to the refrigerator:"
W_dot_in_R = Q_dot_L_R/COP_R
"Power output from heat engine must be:"
W_dot_out_HE =  W_dot_in_R
"The efficiency of the heat engine is:"
T_L_HE = T_surr
eta_HE = 1 - T_L_HE/T_H
"The rate of heat input to the heat engine is:"
Q_dot_H_HE = W_dot_out_HE/eta_HE
"First law applied to the heat engine and refrigerator:"
Q_dot_L_HE = Q_dot_H_HE - W_dot_out_HE
Q_dot_H_R = Q_dot_L_R + W_dot_in_R
Q_dot_surr = Q_dot_L_HE + Q_dot_H_R
```

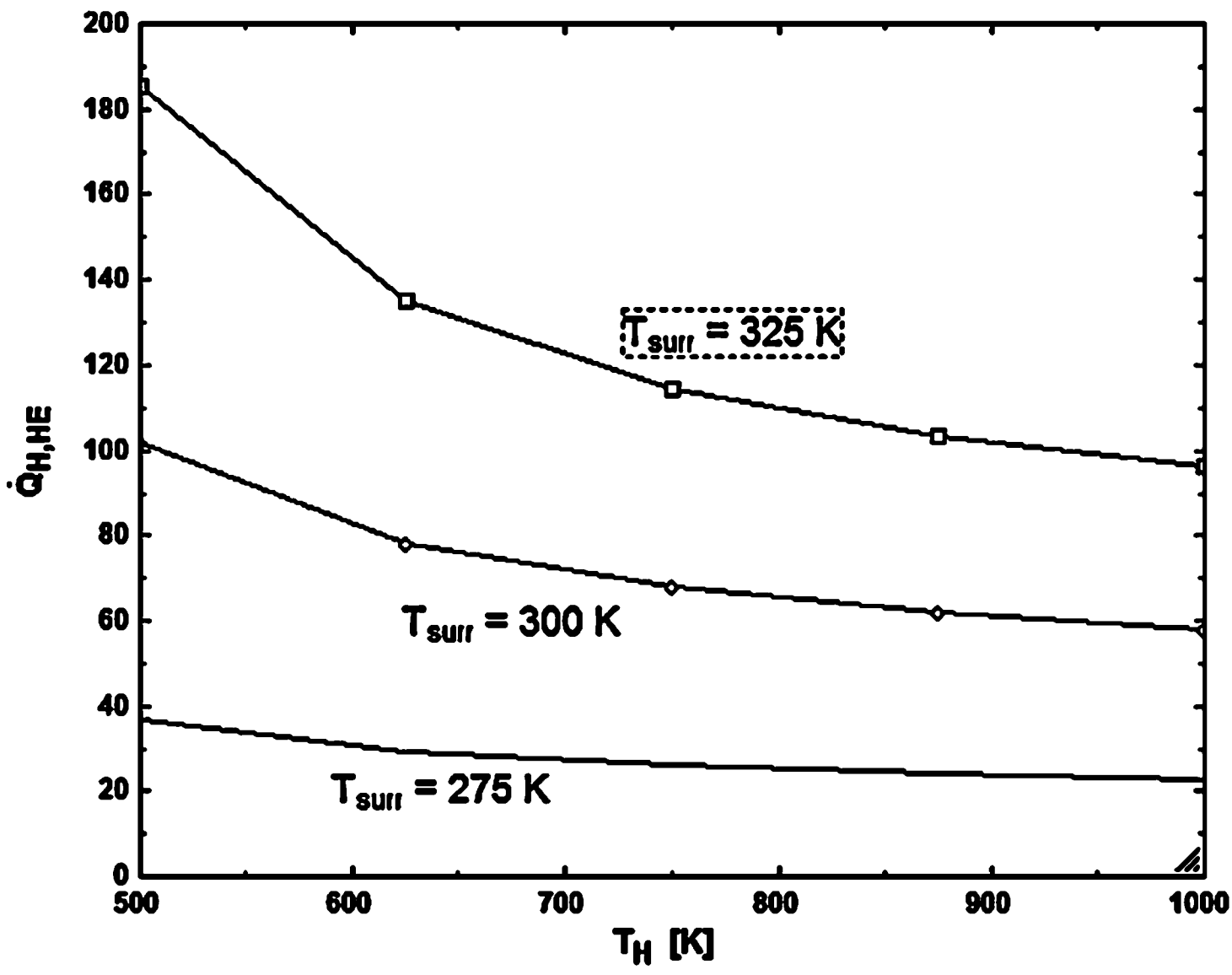
Step 3 of 7

Table for varies parameter as function of source temperature::

$\triangleright_{1..5}$	1 $\dot{Q}_{HHE}$	2 $\dot{Q}_{surr}$	3 T_H [K]
Run 1	185.5	435.5	500
Run 2	135.3	385.3	625
Run 3	114.6	364.6	750
Run 4	103.3	353.3	875
Run 5	96.18	346.2	1000

Step 4 of 7

Graph between the Source temperature and Work input::



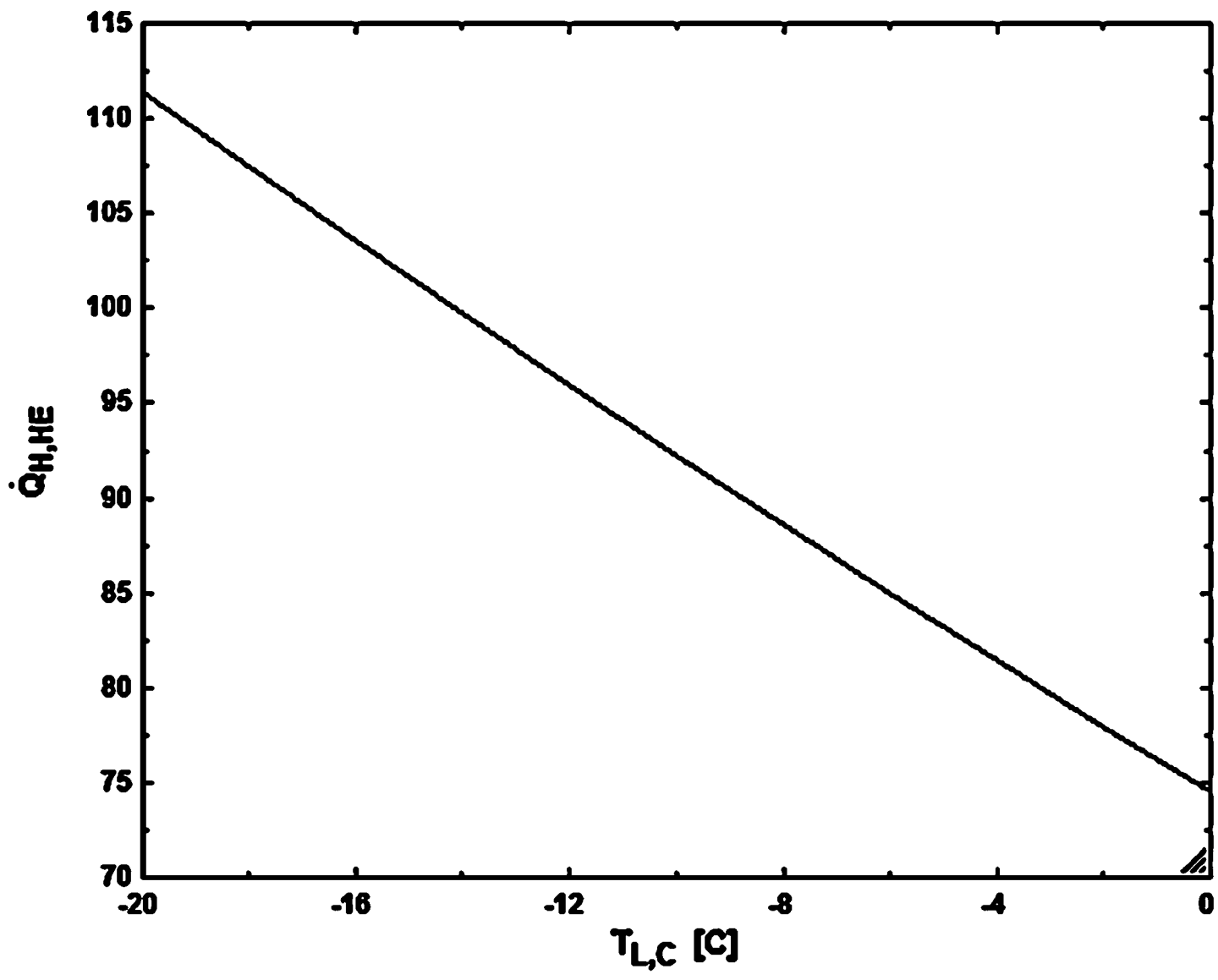
Step 5 of 7

: Table for varies parameter as function of sink temperature

$\triangleright_{1..11}$	1 $T_{L,C}$ [C]	2 $\dot{Q}_{HHE}$	3 $\dot{Q}_{surr}$
Run 1	-20	111.4	361.4
Run 2	-18	107.4	357.4
Run 3	-16	103.5	353.5
Run 4	-14	99.71	349.7
Run 5	-12	96.06	346

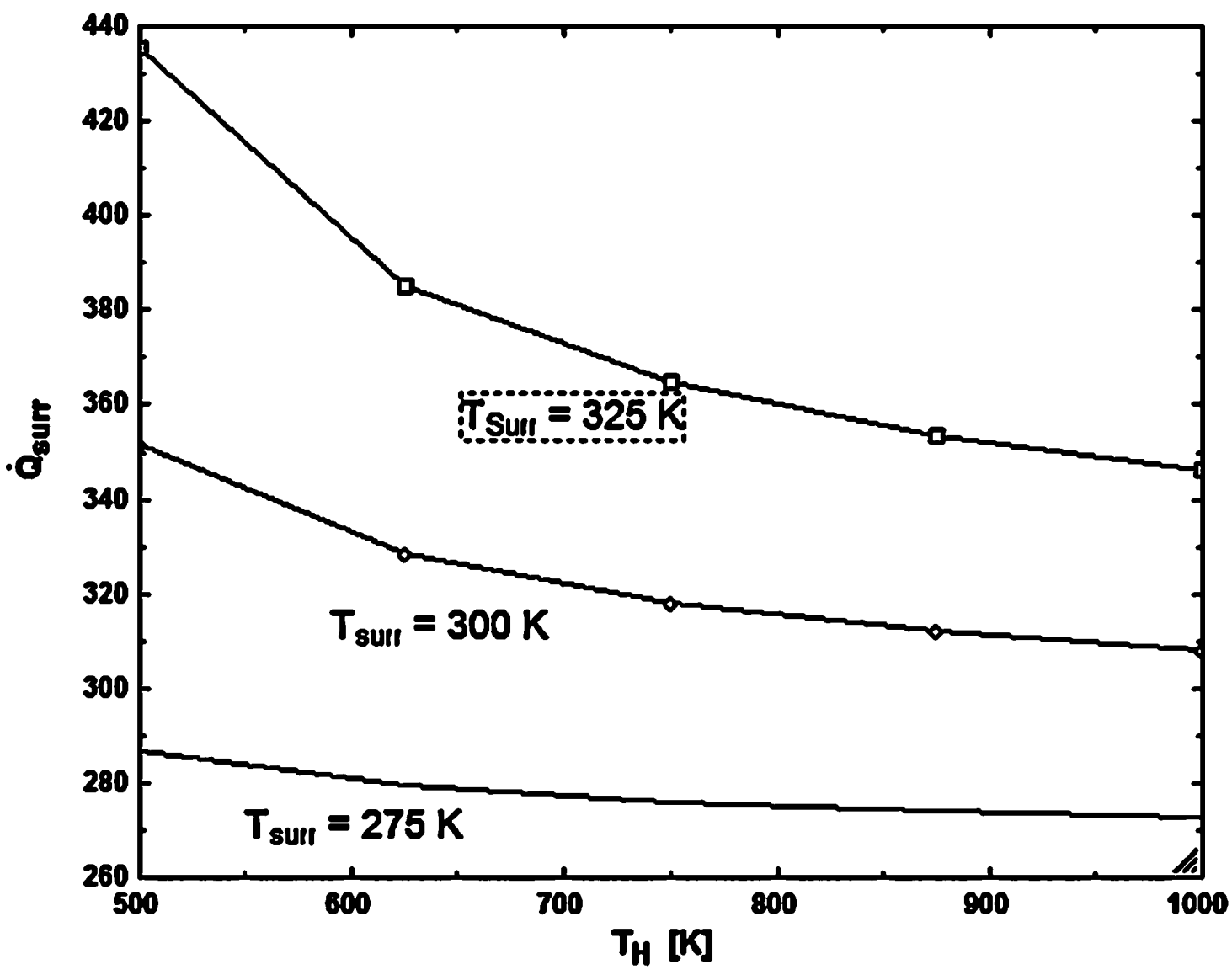
Run 1	-20	711.4122P 3[illegible]g	2
Run 2	-18	107.4	357.4
Run 3	-16	103.5	353.5
Run 4	-14	99.71	349.7
Run 5	-12	95.95	346
Run 6	-10	92.25	342.2
Run 7	-8	88.6	338.6
Run 8	-6	85	335
Run 9	-4	81.46	331.5
Run 10	-2	77.97	328
Run 11	0	74.53	324.5

Graph between the Heat engine and sink temperature :



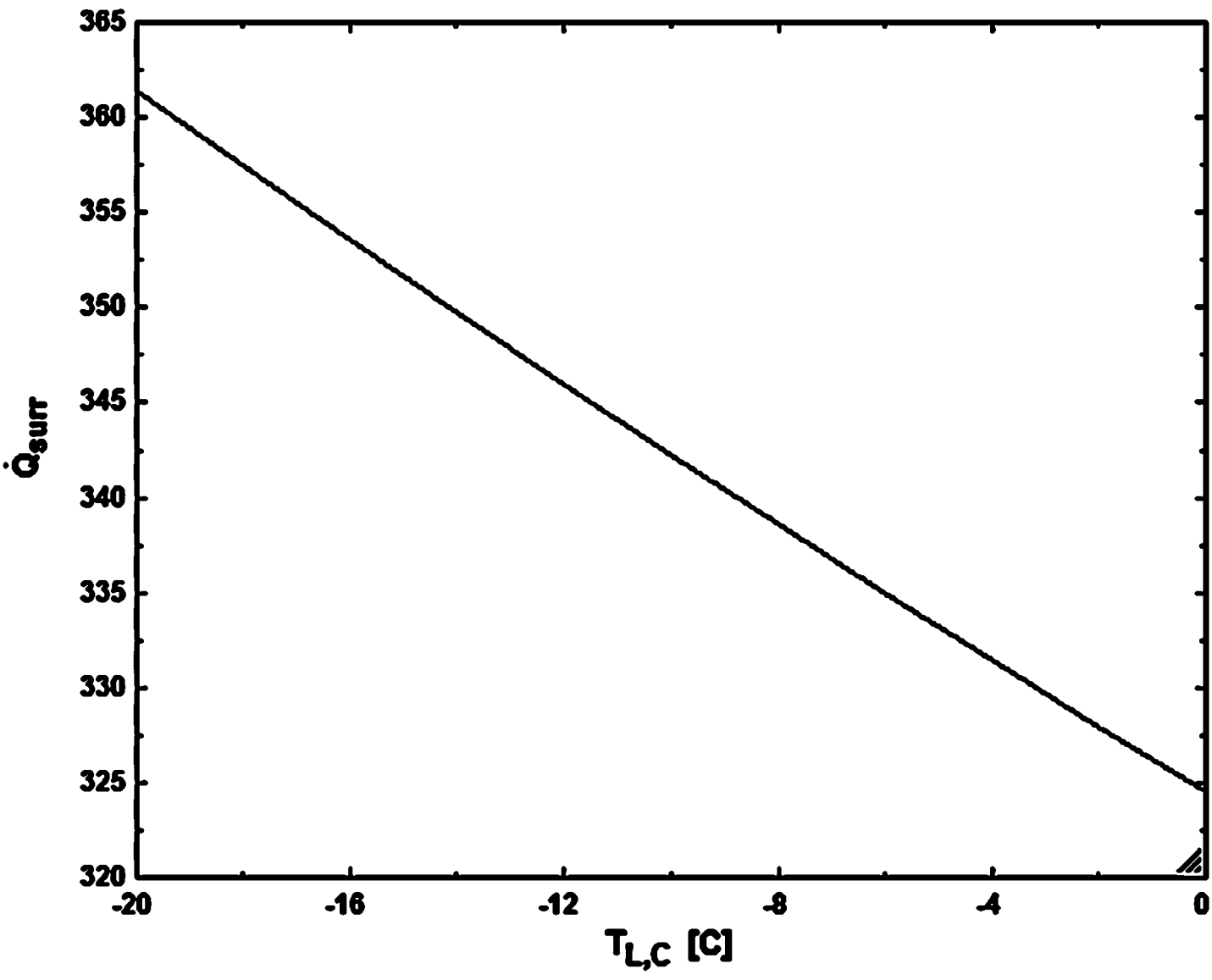
Step 6 of 7

Graph between the sink temperature and Work input :



Step 7 of 7

Graph between the heat surrounding and sink temperature :



Question:

7.123P

A heat engine operates between two reservoirs at 800 and 20°C. One-half of the work output of the heat engine is used to drive a Carnot heat pump that removes heat from the cold surroundings at 2°C and transfers it to a house maintained at 22°C. If the house is losing heat at a rate of 62,000 kJ/h, determine the minimum rate of heat supply to the heat engine required to keep the house at 22°C.

Step 1 of 6

Given data:

For heat engine:

Temperature of source, $T_H = 800°C$

$= 800 + 273$

$= 1073K$

Temperature of sink, $T_L = 20°C$

$= 20 + 273$

$= 293K$

For a heat pump:

Temperature of source, $T_H = 22°C$

$= 22 + 273$

$= 295K$

Temperature of sink, $T_L = 2°C$

$= 20 + 273$

$= 275K$

The work output of a heat engine is used to drive heat pump, $\frac{1}{2}W_{net_HE} = W_{in_HP}$

The heat rejected by the house, $\dot{Q}_H = 62{,}000$ kJ/h

Step 2 of 6

The COP of heat pump is expressed as

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP} = \frac{295}{295 - 275}$$

$$COP_{HP} = 14.75$$

Step 3 of 6

The power input to the heat pump

$$COP_{HP} = \frac{\dot{Q}_H}{W_{net}}$$

$$W_{in} = \frac{62000}{14.75}$$

$$W_{in} = 4203.38 \text{ kJ/h}$$

Step 4 of 6

The work output of a heat engine is used to drive heat pump, $\frac{1}{2}W_{net_HE} = W_{in_HP}$

Therefore the power output from the heat engine

$W_{net} = 2 \times W_{in}$

$= 2 \times 4203.38$

$= 8406.76$ kJ/h

Step 5 of 6

The thermal efficiency of a heat engine is expressed as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$\eta_{th} = 1 - \frac{293}{1073}$$

$$\eta_{th} = 0.7269$$

Step 6 of 6

The minimum rate of heat supplied to the heat engine is expressed as

$$\eta_{th} = \frac{W_{net}}{\dot{Q}_{HP}}$$

$$0.7269 = \frac{8406.76}{\dot{Q}_{HP}}$$

$$\dot{Q}_{HP} = 11{,}565.2 \text{ kJ/h}$$

∴ The minimum rate of heat supplied is $\boxed{\dot{Q}_{HP} = 11{,}565.2 \text{ kJ/h}}$

Question: 7.124P

An inventor claims to have developed a refrigerator that maintains the refrigerated space at 35°F while operating in a room where the temperature is 75°F and that has a COP of 13.5. Is this claim reasonable?

Step 1 of 4

Given

Temperature of source $T_H = 75°C$

Temperature of sink $T_L = 35°C$

COP of the refrigerator is 13.5

Step 2 of 4

Temperature of source

$T_H = 75°C$

$= 75 + 460$

$= 535K$

Temperature of sink

$T_L = 35°C$

$= 35 + 460$

$= 495K$

Step 3 of 4

The efficiency of a refrigerator is expressed in terms of the coefficient of performance (COP), denoted by COP_R. The objective of a refrigerator is to remove heat (Q_L) from the refrigerated space. To accomplish this objective, it requires a work input of W_{net}

$$COP_R = \frac{\text{Desired output}}{\text{Required input}}$$

Step 4 of 4

For a reversible refrigerator has the maximum COP, which is expressed as

$$COP_R = \frac{T_L}{T_H - T_L}$$

$$COP_R = \frac{495}{535 - 495}$$

$$COP_R = 12.375$$

Here

$$COP_R = 12.375 < 13.4$$

The maximum possible COP of this temperature limits is 12.375, but the inventor claims 13.4, So his claim is not reasonable.

Question: 7.125P

Consider a Carnot heat-pump cycle executed in a steady-flow system in the saturated liquid–vapor mixture region using refrigerant-134a flowing at a rate of 0.22 kg/s as the working fluid. It is known that the maximum absolute temperature in the cycle is 1.2 times the minimum absolute temperature, and the net power input to the cycle is 5 kW. If the refrigerant changes from saturated vapor to saturated liquid during the heat rejection process, determine the ratio of the maximum to minimum pressures in the cycle.

Step 1 of 8

Given

Mass flow rate of refrigerant $\dot{m} = 0.22$ kg/s.

The temperature $T_H = 1.2T_L$.

The net power input to the cycle $P = 5$ kW.

Step 2 of 8

The COP of the heat pump is

$$COP_{HP} = \frac{T_H}{T_H - T_L}$$

$$COP_{HP} = \frac{T_H}{T_H - \frac{1}{1.2}T_H}$$

$$= 6$$

Step 3 of 8

The heat rejected to the heat pump

$$COP_{HP} = \frac{\dot{Q}_H}{W_{net}}$$

$$\dot{Q}_H = 6 \times 5$$

$$= 30 \text{ kW}$$

Step 4 of 8

Per unit kg of mass

Heat lost in this process

$$q_H = \frac{\dot{Q}_H}{\dot{m}}$$

$$= \frac{30}{0.22}$$

$$= 136.4 \text{ kJ/kg}$$

Step 5 of 8

For 1 kg of conversion of saturated vapor to saturated liquid it takes 136.4 kJ/kg of heat

I.e. Enthalpy of vaporization $h_{fg} = 136.4$ kJ/kg

The temperature corresponds to $h_{fg} = 136.4$ kJ/kg

From saturated refrigerant 134a Temperature table

At $h_{fg} = 136.4$ kJ/kg

By using the method of interpolation

$$T_H = 60 + \frac{(136.4 - 139.10)(65 - 60)}{((132.02 - 139.10))}$$

$$T_H = 61.903^\circ C$$

$$= 62^\circ C$$

Step 6 of 8

From saturated refrigerant 134a Temperature table

Pressure at $T_H = 62^\circ C$

By interpolation

$$\frac{P_{sat} - P_1}{P_2 - P_1} = \frac{T_{sat} - T_1}{T_2 - T_1}$$

$$\frac{P_{sat} - 1682.8}{1891 - 1682.8} = \frac{62 - 60}{65 - 60}$$

$$P_H = 1821.6 \text{ kPa}$$

Step 7 of 8

Therefore the lower temperature reservoir is

$$T_L = \frac{T_H}{1.2}$$

$$= \frac{62 + 273}{1.2}$$

$$= 279.167\text{K} - 273$$

$$= 6.167^\circ C$$

Step 8 of 8

From saturated refrigerant 134a Temperature table

At $T_L = 6.167^\circ C$

The minimum pressure is

The saturated pressure at minimum temperature is

$$P_L = 363 \text{ kPa}$$

Therefore the ratio of maximum to minimum pressure is

$$\frac{P_H}{P_L} = \frac{1821.6}{363}$$

$$= 5.02$$

$$\boxed{\frac{P_H}{P_L} = 5.02}$$

Question: 7.126P

Replacing incandescent lights with energy-efficient fluorescent lights can reduce the lighting energy consumption to one-fourth of what it was before. The energy consumed by the lamps is eventually converted to heat, and thus switching to energy-efficient lighting also reduces the cooling load in summer but increases the heating load in winter. Consider a building that is heated by a natural gas furnace with an efficiency of 80 percent and cooled by an air conditioner with a COP of 3.5. If electricity costs \$0.08/kWh and natural gas costs \$1.40/therm (1 therm = 105,500 kJ), determine if efficient lighting will increase or decrease the total energy cost of the building (*a*) in summer and (*b*) in winter.

Step 1 of 4

Given data:
The thermal efficiency of furnace, $\eta = 0.8$
The COP of refrigerant, COP = 3.5
Unit cost of electricity for refrigerant, \$0.08/kWh
Unit cost of natural gas for furnace, \$1.40/therm
1 therm = 105,500 kJ

Step 2 of 4

(a) In summer the usage of lightening in the house will be lesser than the winter, so therefore total cost of energy used of the household will be decreased.

Step 3 of 4

(b) In winter the usage of lightening will be higher based on cost of electricity.
For a natural gas furnace

$$\text{Unit cost} = \frac{\$1.40}{\eta_{th}}\left(\frac{1\text{ therm}}{105{,}500}\right)$$

$$= \frac{\$1.40}{0.8}\left(\frac{1\text{ therm}}{105{,}500}\right)$$

$$= \$1.66\times10^{-5}/\text{kJ}$$

For 1 kWh heat supplied the cost for furnace is $= \$1.66\times10^{-5}/\text{kJ}\times3600$

$= \$\,0.06/\text{kWh}$

Step 4 of 4

So for heat supplied for natural gas furnace is \$ 0.06/kWh which is less than \$ 0.08/kWh cost of electricity for refrigerator.
Thus the energy efficient lighting will reduce the total energy cost of this building both in summer and in winter.

Question: 7.127P

A heat pump supplies heat energy to a house at the rate of 140,000 kJ/h when the house is maintained at 25°C. Over a period of one month, the heat pump operates for 100 hours to transfer energy from a heat source outside the house to inside the house. Consider a heat pump receiving heat from two different outside energy sources. In one application the heat pump receives heat from the outside air at 0°C. In a second application the heat pump receives heat from a lake having a water temperature of 10°C. If electricity costs \$0.085/kWh, determine the maximum money saved by using the lake water rather than the outside air as the outside energy source.

Step 1 of 9

Given

The temperature of house $T_H = 25°C$.

The temperature of the outside air $T_L = 0°C$.

The heat energy supplied $\dot{Q}_H = 140{,}000$ kJ/h .

Operating hours of heat pump $t = 100$ hrs .

The cost of electricity is \$0.085/kWh.

In second case heat pump receives from water at $T_{L_B} = 10°C$.

Step 2 of 9

The temperature of house

$$T_H = 25°C$$
$$= 25 + 273$$
$$= 298K$$

The temperature of the outside air

$$T_L = 0°C$$
$$= 273K$$

In second case heat pump receives from water at

$$T_{L_B} = 10°C$$
$$= 10 + 273$$
$$= 283K$$

Step 3 of 9

CASE A:

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP_A} = \frac{T_H}{T_H - T_L}$$
$$COP_{HP_A} = \frac{298}{298 - 273}$$
$$COP_{HP_A} = 11.92$$

Step 4 of 9

The minimum work per unit heat transfer of a heat pump is expressed as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$
$$\dot{W}_{net} = \frac{\dot{Q}_H}{COP_{HP}}$$
$$\dot{W}_{net} = \frac{\left(\frac{140{,}000}{3600}\right)}{11.92}$$
$$\dot{W}_{net} = 3.262 \text{ kW}$$

Step 5 of 9

The cost for electricity if the heat pump takes outside air

$$Cost = \dot{W}_{net} \times 100 \text{ h} \times \$0.085 / \text{kWh}$$
$$Cost = 3.262 \times 100 \text{ h} \times \$0.085 / \text{kWh}$$
$$Cost = \$27.73$$

Step 6 of 9

CASE B:

The heat pump receives heat from lake water

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP_B} = \frac{T_H}{T_H - T_L}$$
$$COP_{HP_B} = \frac{298}{298 - 283}$$
$$COP_{HP_B} = 19.867$$

Step 7 of 9

The minimum work per unit heat transfer of a heat pump is expressed as

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net}}$$
$$\dot{W}_{net} = \frac{\dot{Q}_H}{COP_{HP}}$$
$$\dot{W}_{net} = \frac{\left(\frac{140{,}000}{3600}\right)}{19.867}$$
$$\dot{W}_{net} = 1.957 \text{ kW}$$

Step 8 of 9

The cost for electricity if the heat pump takes outside air

$$Cost = \dot{W}_{net} \times 100 \text{ h} \times \$0.085 / \text{kWh}$$
$$Cost = 1.957 \times 100 \text{ h} \times \$0.085 / \text{kWh}$$
$$Cost_B = \$16.64$$

Step 9 of 9

Therefore money saved is

$$\$ = Cost_A - Cost_B$$
$$= \$27.73 - \$16.64$$
$$= \$11.1$$
$$\boxed{\$ = \$11.1}$$

If the heat pump uses lake water it can save \$11.1 for a month.

Question: 7.128P

The cargo space of a refrigerated truck whose inner dimensions are 12 m × 2.3 m × 3.5 m is to be precooled from 25°C to an average temperature of 5°C. The construction of the truck is such that a transmission heat gain occurs at a rate of 80 W/°C. If the ambient temperature is 25°C, determine how long it will take for a system with a refrigeration capacity of 8 kW to precool this truck.

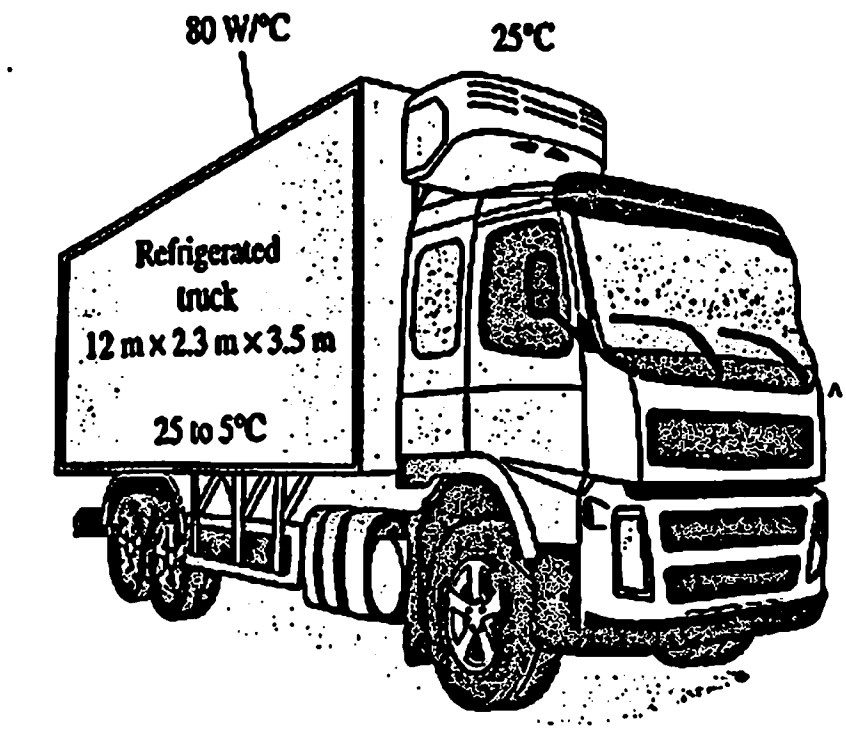


FIGURE P7–128

Step 1 of 6

Given data:

The total volume of the truck, $V = 12 \times 2.3 \times 3.5\ \text{m}^3$

$= 96.6\ \text{m}^3$

The temperature of the ambient air is, $T_1 = 25°\text{C}$

The cooled temperature of the air, $T_2 = 5°\text{C}$

The transmission heat gain into the refrigerator is, 80 W/°C

The cooling capacity of refrigerator is, $\dot{Q}_{ref} = 8\ \text{kW}$

Step 2 of 6

From the properties of air tables at 25^0C is

Specific heat, $c_p = 1.005\ \text{kJ/kg}$

Density of air, $\rho = 1.2\ \text{kg/m}^3$

Step 3 of 6

The amount of heat removed from the truck as the air gets cooled is

$Q_{air} = mc_p\Delta T\ \text{kJ}$

Where m is mass of air in kg

$m = \rho V_{air}\ \text{kg}$

$m = (1.2)(96.6)\ \text{kg}$

$m = 115.9\ \text{kg}$

Step 4 of 6

Therefore the heat removed is

$Q_{air} = mc_p\Delta T\ \text{kJ}$

$Q_{air} = (115.9)(1.005)(25-5)\ \text{kJ}$

$Q_{air} = 2329.99\ \text{kJ}$

Step 5 of 6

The average rate of heat transmission in the truck due to pre-cooling is

$\dot{Q}_{tran} = UA\Delta T\ \text{kW}$

Where U is heat transmission rate 80 W/°C,

Average temperature is $\frac{25+5}{2} = 15°\text{C}$

$\dot{Q}_{tran} = 80(25-15)\ \text{W}$

$\dot{Q}_{tran} = 800\ \text{W}$

$\dot{Q}_{tran} = 0.8\ \text{kW}$

Step 6 of 6

The time taken for refrigerator to cool this truck is

$\dot{Q}_{ref} = \dot{Q}_{tran} + \frac{Q_{air}}{\Delta t}$

$\Delta t = \frac{Q_{air}}{\dot{Q}_{ref} - \dot{Q}_{tran}}\ \text{sec}$

$\Delta t = \frac{2329.99}{8-0.8}$

$\Delta t = 323.6\ \text{s}$

$\Delta t = \frac{323.6}{60}\ \text{min}$

$\Delta t = 5.39\ \text{min}$

Therefore the time taken is $\boxed{\Delta t = 5.39\ \text{min}}$

Question: 7.129P

A refrigeration system is to cool bread loaves with an average mass of 350 g from 30 to −10°C at a rate of 1200 loaves per hour by refrigerated air at −30°C. Taking the average specific and latent heats of bread to be 2.93 kJ/kg·°C and 109.3 kJ/kg, respectively, determine (*a*) the rate of heat removal from the breads, in kJ/h; (*b*) the required volume flow rate of air, in m3/h, if the temperature rise of air is not to exceed 8°C; and (*c*) the size of the compressor of the refrigeration system, in kW, for a COP of 1.2 for the refrigeration system.

Step 1 of 8

Given

The average mass of refrigerant $m = 350\ \text{g} = 0.35\ \text{kg}$.

The higher temperature of the refrigerator $T_2 = 30°\text{C}$.

The lower temperature of the refrigerator $T_1 = -10°\text{C}$.

The temperature of the refrigerant air $T_{air} = -30°\text{C}$.

1200 loves of bread is cooled per hour.

The average specific heat of bread $c = 2.93\ \text{kJ/kg°C}$.

Latent heat $h = 109.3\ \text{kJ/kg}$.

Step 2 of 8

From the ideal gas specific heat of various common gases

The average specific heat of air $c_{p_air} = 1.005\ \text{kJ/kg°C}$

(a) The rate of heat removal from the bread

The mass flow rate of bread

$$\dot{m}_{bread} = \text{no of bread loafs cooled} \times (0.45\ \text{kg/bread})$$

$$\dot{m}_{bread} = 1200 \times (0.35\ \text{kg/bread})$$

$$= 420\ \text{kg/h}$$

$$= \frac{420}{3600}$$

$$= 0.1166\ \text{kg/s}$$

Step 3 of 8

The amount of heat removed from bread is

$$\dot{Q}_{bread} = \dot{m}_{bread} c_p (T_2 - T_1)\ \text{kg/h}$$

$$\dot{Q}_{bread} = 420(2.93)(30 - (-10))\ \text{kg/h}$$

$$\dot{Q}_{bread} = 49,224\ \text{kg/h}$$

Step 4 of 8

The heat produced due to freezing is

$$\dot{Q}_{freezing} = \dot{m}_{bread} h_{latent}\ \text{kJ/h}$$

$$\dot{Q}_{freezing} = (420)(109.3)\ \text{kJ/h}$$

$$\dot{Q}_{freezing} = 45,906\ \text{kJ/h}$$

Step 5 of 8

The total heat removed from the bread

$$\dot{Q}_{total} = \dot{Q}_{bread} + \dot{Q}_{freeze}$$

$$\dot{Q}_{total} = 49224 + 45,906\ \text{kJ/h}$$

$$\boxed{\dot{Q}_{total} = 95,130\ \text{kJ/h}}$$

Step 6 of 8

(b) The volume flow rate, if exceeding 8°C

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}}\ \text{m}^3\text{/h}$$

The density of air can be determined by ideal gas equation

$$\rho_{air} = \frac{P}{RT}$$

$$= \frac{101.325}{(0.287)(-30 + 273)}$$

$$= 1.45\ \text{kg/m}^3$$

Step 7 of 8

The mass flow rate of air is determined by

$$\dot{Q}_{air} = \dot{m}_{air} c_p \Delta T\ \text{kg/h}$$

The heat released by bread will be absorbed by the refrigerant air

$$95,130 = \dot{m}_{air}(1.005)(8)\ \text{kg/h}$$

$$\dot{m}_{air} = 11,832\ \text{kg/h}$$

The volume flow rate is

$$\dot{V}_{air} = \frac{11,832}{1.45}\ \text{m}^3\text{/h}$$

$$\boxed{\dot{V}_{air} = 8160\ \text{m}^3\text{/h}}$$

Step 8 of 8

(c) The size of the compressor is nothing but the power capacity of the compressor

For a COP of 1.2

We know that

$$COP = \frac{\dot{Q}_{ref}}{\dot{W}_{ref}}$$

$$\dot{W}_{ref} = \frac{\dot{Q}_{ref}}{COP}$$

$$= \frac{95,130}{1.2}$$

$$\dot{W}_{ref} = \frac{79275}{3600}\left(\frac{\text{kJ}}{s}\right)$$

$$\boxed{\dot{W}_{ref} = 22.02\ \text{kW}}$$

Question:

7.130P

The drinking water needs of a production facility with 20 employees is to be met by a bubbler type water fountain. The refrigerated water fountain is to cool water from 22 to 8°C and supply cold water at a rate of 0.4 L per hour per person. Heat is transferred to the reservoir from the surroundings at 25°C at a rate of 45 W. If the COP of the refrigeration system is 2.9, determine the size of the compressor, in W, that will be suitable for the refrigeration system of this water cooler.

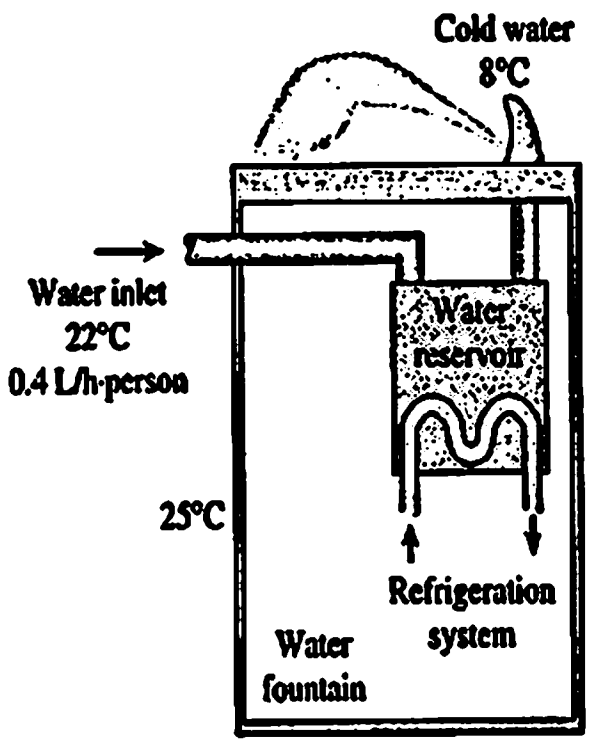


FIGURE P7–130

Step 1 of 5

Given data:
The number of employees in the room = 20
The initial temperature 0f the water, $T_1 = 22°C$
The final temperature of the water, $T_2 = 8°C$
The volume flow rate of the cold water, $\dot{V} = 0.4$ L/h
The amount of heat transferred from the surroundings, $\dot{Q}_{surr} = 45$ W
The COP of the refrigerator system is 2.9

Step 2 of 5

From properties of water tables,
Taking the specific heat of water $c = 4.18$ kJ/kg°C,
The density of water $\rho = 1000$ kg/m^3
The amount of heat removed from water
$\dot{Q}_{water} = \dot{m}c(T_2 - T_1)$ kW

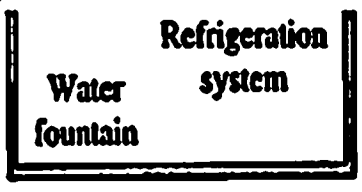


FIGURE P7–130

Step 1 of 5

Given data:
The number of employees in the room = 20
The initial temperature 0f the water, $T_1 = 22°C$
The final temperature of the water, $T_2 = 8°C$
The volume flow rate of the cold water, $\dot{V} = 0.4$ L/h
The amount of heat transferred from the surroundings, $\dot{Q}_{surr} = 45$ W
The COP of the refrigerator system is 2.9

Step 2 of 5

From properties of water tables,
Taking the specific heat of water $c = 4.18$ kJ/kg°C,
The density of water $\rho = 1000$ kg/m^3
The amount of heat removed from water
$\dot{Q}_{water} = \dot{m}c(T_2 - T_1)$ kW

For a COP of 2.9
We know that

$$COP = \frac{\dot{Q}_{ref}}{\dot{W}_{ref}}$$

$$\dot{W}_{ref} = \frac{\dot{Q}_{total}}{COP}$$
$$= \frac{0.175}{2.9}$$
$$= 0.0603 \text{ kW}$$
$$= 60.3 \text{ W}$$

$$\boxed{\dot{W}_{ref} = 60.3 \text{ W}}$$

The power needed to the run the compressor may be greater than 60.3 W to meet the requirement

Question:

7.131P

The "Energy Guide" label on a washing machine indicates that the washer will use $33 worth of hot water if the water is heated by a gas water heater at a natural gas rate , of $1.21/therm. If the water is heated from 60 to 130°F, determine how many gallons of hot water an average family uses per week. Disregard the electricity consumed by the washer, and take the efficiency of the gas water heater to be 58 percent.

Step 1 of 5

Given data:

The thermal efficiency of the gas heater $\eta_{th} = 58\%$

The amount of electricity consumed in a year is $ 33

The amount of natural gas consumed in a year is $1.21/therm

The initial temperature of the water, $T_1 = 60°F$

The final temperature of the water, $T_2 = 130°F$

Step 2 of 5

From the properties of common liquids, solids, and foods tables taking properties of water

The density of water at room temperatures is, $\rho = 62.11 \text{lbm/ft}^3$

Specific heat, $c = 1 \text{ Btu/lbm} \cdot °F$

The amount of electricity consumed to heat water in a year is

$$= \frac{\text{Total Cost}}{\text{Unit Cost}}$$

$$= \frac{\$\ 33/\text{year}}{\$\ 1.21/\text{therm}}$$

$$= 27.2727 \text{ Therm/year}$$

Step 3 of 5

The energy transferred to water is

$$\dot{E}_{in} = \eta \times \text{Total energy}$$

$$\dot{E}_{in} = 0.58(27.2727) \text{ therm/year}$$

For a week

$$\dot{E}_{in} = 0.58(27.2727) \text{ therm/year}\left(\frac{100{,}000 \text{ Btu}}{1 \text{ therm}}\right)\left(\frac{1 \text{ Year}}{52 \text{ weeks}}\right)$$

$$\dot{E}_{in} = 30{,}419.58 \text{Btu/week}$$

Step 4 of 5

The energy transferred is expressed as

$$\dot{E}_{in} = \dot{m}c(T_2 - T_1)\text{Btu}$$

Where m is the mass flow rate of hot water supplied to the heater

Therefore

$$\dot{m} = \frac{\dot{E}_{in}}{c(T_2 - T_1)}\text{Btu}$$

$$\dot{m} = \frac{30{,}419.58}{1 \times (130 - 60)}\text{Btu/week}$$

$$\dot{m} = 434.56 \text{ Btu/week}$$

Step 5 of 5

The amount of flow supplied is the volume flow rate of hot water

$$\dot{V}_{hot} = \frac{\dot{m}\ \text{lbm/week}}{\rho\ \text{lbm/ft}^3} \quad (7.48 \text{ gallons} = 1 \text{ ft}^3)$$

$$\dot{V}_{hot} = \frac{434.56}{62.1}\left(\frac{7.48 \text{ gal}}{1 \text{ ft}^3}\right)$$

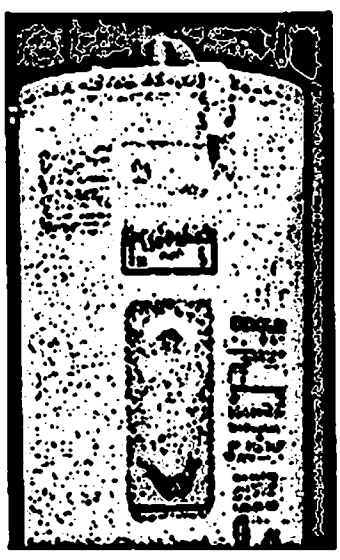
$$\boxed{\dot{V}_{hot} = 52.34 \text{ gal/week}}$$

On an average 52 gallons of hot water is used for washing cloths in a week

Question:

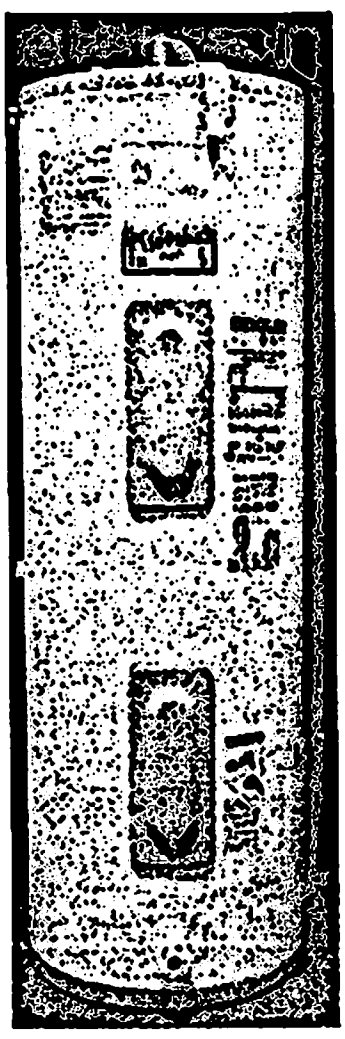
7.132P

A typical electric water heater has an efficiency of 95 percent and costs $250 a year to operate at a unit cost of electricity of $0.08/kWh. A typical heat pump-powered water heater has a COP of 3.3 but costs about $800 more to install. Determine how many years it will take for the heat pump water heater to pay for its cost differential from the energy it saves.



Question:

A typical electric water heater has an efficiency of 95 percent and costs $250 a year to operate at a unit cost of electricity of $0.08/kWh. A typical heat pump-powered water heater has a COP of 3.3 but costs about $800 more to install. Determine how many years it will take for the heat pump water heater to pay for its cost differential from the energy it saves.



Water heater

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Jill Braaten, photographer.

FIGURE P7–132

Step 1 of 6

Loading solution. Please wait ...

Step 2 of 6

From the properties of common liquids, solids and foods
For water

Specific heat of water $c = 4.18 \frac{kJ}{kg \cdot {}^{\circ}C}$,

The density of water $\rho = 1000 \frac{kg}{m^3}$

The amount of electricity consumed by the water heater in a year is

$$= \frac{\text{Total Cost}}{\text{Unit Cost}}$$

$$= \frac{\$\,250}{\$\,0.08}$$

$$= 3125 \frac{kWh}{year}$$

Step 3 of 6

The energy transferred to water is

$$\dot{E}_{in} = \eta \times \text{Total energy}$$

$$\dot{E}_{in} = 0.95 \times 3125 \frac{kWh}{year}$$

$$\dot{E}_{in} = 2968.75 \frac{kWh}{year}$$

Step 4 of 6

The energy consumed by the heat pump in a year is expressed as

$$= \frac{\text{Energy observed by water}}{\text{COP}}$$

$$= \frac{2968.75}{3.3}$$

So if we use a heat pump, it will take approx 7.7 years to repay the amount.

Question: 7.133P



Reconsider Prob. 7–132. Using EES (or other) software, investigate the effect of the heat pump COP on the yearly operation costs and the number of years required to break even. Let the COP vary from 2 to 5. Plot the payback period against the COP and discuss the results.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 7.134P

A homeowner is trying to decide between a high-efficiency natural gas furnace with an efficiency of 97 percent and a ground-source heat pump with a COP of 3.5. The unit costs of electricity and natural gas are \$0.092/kWh and \$1.42/therm (1 therm = 105,500 kJ). Determine which system will have a lower energy cost.

Step 1 of 3

Given data:
The thermal efficiency of furnace, $\eta = 97\%$
The heat pump, COP = 3.5
Unit cost of electricity for heat pump, \$0.092/kWh
Unit cost of natural gas for furnace, \$1.42/therm
1 therm = 105,500 kJ

Step 2 of 3

For a natural gas furnace

$$\text{Unit cost} = \frac{\$1.42}{0.97}\left(\frac{1\,\text{therm}}{105{,}500}\right)$$
$$= \$1.3876\times10^{-5}/\text{kJ}$$

Step 3 of 3

For heat pump

$$\text{Unit cost} = \frac{\$0.092}{3.5}\left(\frac{1\,\text{kWh}}{3600}\right)$$
$$= \$0.73\times10^{-5}/\text{kJ}$$

So the system have a lower energy cost is the heat pump

Question: 7.135P

Cold water at 10°C enters a water heater at the rate of 0.02 m3/min and leaves the water heater at 50°C. The water heater receives heat from a heat pump that receives heat from a heat source at 0°C.

(a) Assuming the water to be an incompressible liquid that does not change phase during heat addition, determine the rate of heat supplied to the water, in kJ/s.

(b) Assuming the water heater acts as a heat sink having an average temperature of 30°C, determine the minimum power supplied to the heat pump, in kW.

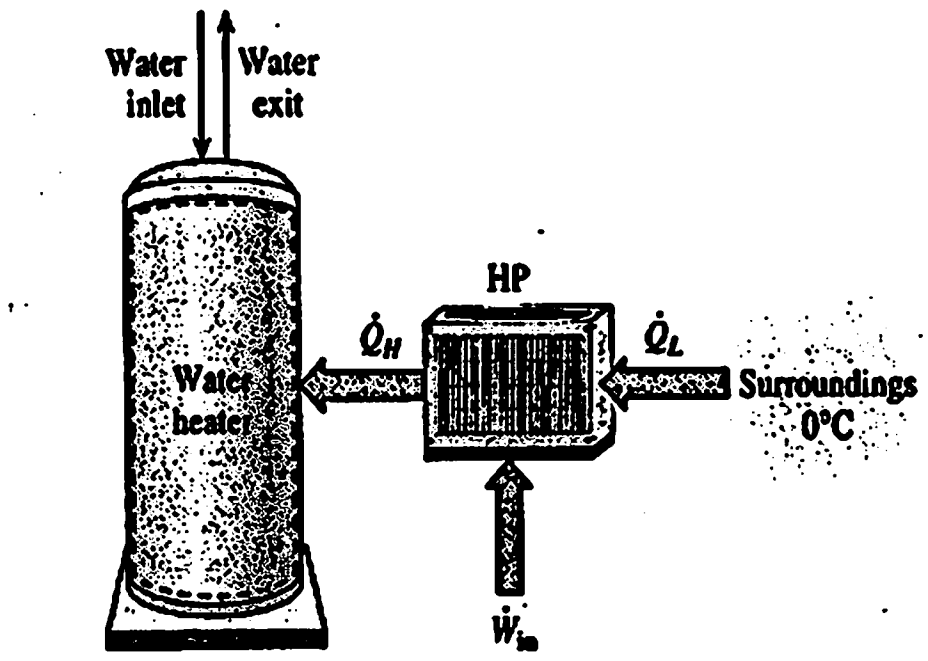


FIGURE P7–135

Step 1 of 5

Given data:

Initial temperature of water, $T_1 = 10°C$

$= 10 + 273$

$= 283K$

Final temperature of water, $T_2 = 50°C$

$= 50 + 273$

$= 323K$

The volume flow rate of cold water, $\dot{V} = 0.02\ m^3/min$

The temperature of the sink for heat pump is $T_L = 0°C$

$= 0 + 273$

$= 273K$

Step 2 of 5

(a) The amount of heat supplied to the water is determined from the energy balance

$\dot{Q}_H = \dot{m} c_p (T_2 - T_1)$ kW

Taking properties of water from properties of common liquids tables

The specific heat of water, $c = 4.18$ kJ/kg°C

The density of water, $\rho = 1000\ kg/m^3$

Where the mass of cold water is expressed as

$\dot{m}_{water} = \rho \dot{V}$ kg/s

$$\dot{m}_{water} = \left(1000 kg/m^3\right)\left(0.02 m^3/min\right)\left(\frac{1}{60\ s}\right) kg/s$$

$\dot{m}_{water} = 0.334$ kg/s

Step 3 of 5

Therefore the heat supplied to water is

$\dot{Q}_H = (0.334)(4.18)(323 - 283)$ kJ/s

$\dot{Q}_H = 55.84$ kJ/s

∴ The rate of heat supplied $\boxed{\dot{Q}_H = 55.84 \text{ kJ/s}}$

Step 4 of 5

(b) The power supplied to heat pump

The temperature of heat source of a heat pump $T_H = 30°C$

$= 30 + 273$

$= 303K$

The COP of a reversible heat pump is expressed as

$$COP_{max} = \frac{T_H}{T_H - T_L}$$

$$COP_{max} = \frac{303}{303 - 273}$$

$COP_{max} = 10.1$

Step 5 of 5

The minimum power input to the heat pump

$$COP_{max} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$

$$\dot{W}_{in} = \frac{55.84}{10.1}$$

$\dot{W}_{in} = 5.528$ kW

∴ The power input is $\boxed{\dot{W}_{in} = 5.528 \text{ kW}}$

Question: 7.136P

A heat pump receives heat from a lake that has an average winter time temperature of 6°C and supplies heat into a house having an average temperature of 27°C.

(*a*) If the house loses heat to the atmosphere at the rate of 64,000 kJ/h, determine the minimum power supplied to the heat pump, in kW.

(*b*) A heat exchanger is used to transfer the energy from the lake water to the heat pump. If the lake water temperature decreases by 5°C as it flows through the lake water-to-heat pump heat exchanger, determine the minimum mass flow rate of lake water, in kg/s. Neglect the effect of the lake water pump.

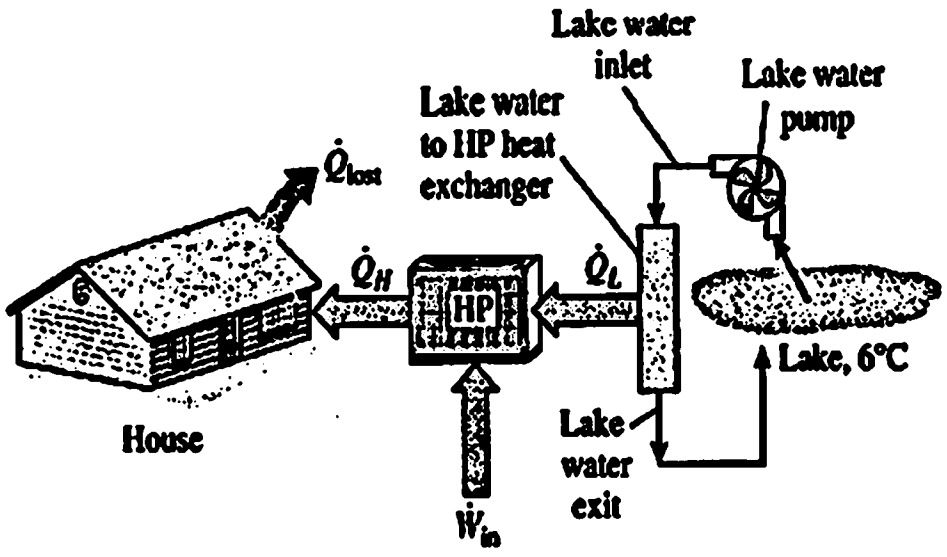


FIGURE P7–136

Step 1 of 6

Given data:

The temperature of heat source of a heat pump $T_H = 27°C$

$= 27 + 273$

The temperature of sink of a heat pump is, $T_L = 6°C$

$= 6 + 273$

$= 279K$

Step 2 of 6

(a) The power supplied to heat pump

The heat lost to the atmosphere, $\dot{Q}_H = 64{,}000$ kJ/h

The COP of a reversible heat pump is expressed as

$$COP_{max} = \frac{T_H}{T_H - T_L}$$

$$COP_{max} = \frac{300}{300 - 279}$$

$$COP_{max} = 14.3$$

Step 3 of 6

The minimum power input to the heat pump

$$COP_{max} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$

$$\dot{W}_{in} = \frac{64000/3600}{14.3}$$

$$\dot{W}_{in} = 1.243 \text{ kW}$$

∴ The minimum power supplied, $\boxed{\dot{W}_{in} = 1.243 \text{ kW}}$

Step 4 of 6

(b)The minimum mass flow rate of lake

The rate of heat absorbed by the lake is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{in}$$

$$\dot{Q}_L = 64000 \text{ kJ/h} - 1.243 \text{ kJ/s}$$

$$\dot{Q}_L = 64000\left(\frac{1 \text{ h}}{3600 \text{ s}}\right) \text{kJ/s} - 1.243 \text{ kJ/s}$$

$$\dot{Q}_L = 16.53 \text{ kJ/s}$$

Step 5 of 6

The amount of heat supplied to the water is determined from the energy balance

$$\dot{Q}_L = \dot{m}c(T_2 - T_1) \text{ kW}$$

From the properties of common liquids, solids, and foods tables taking properties of water

The specific heat of water, $c = 4.18$ kJ/kg°C

The density of water, $\rho = 1000$ kg/m^3

Given the change in temperature of the lake water is 5°C

Step 6 of 6

Where the $\dot{m}$ is the mass flow rate of lake water

$$16.53 = \dot{m}(4.18)(5) \text{ kW}$$

$$\dot{m} = \frac{16.53}{(4.18)(5)} \text{ kg/s}$$

$$\dot{m} = 0.791 \text{ kg/s}$$

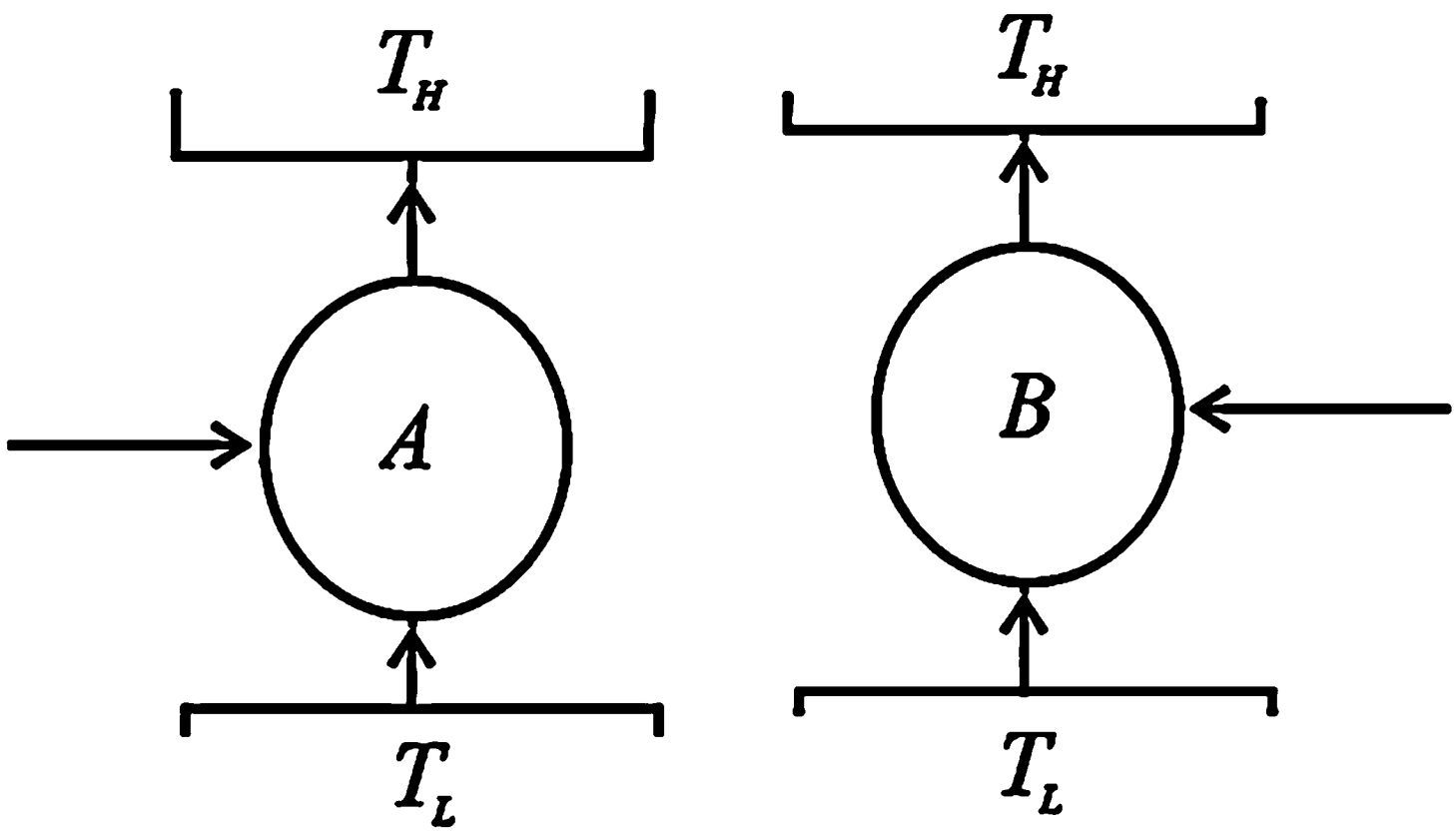
∴ The minimum mass flow rate of lake water, $\boxed{\dot{m} = 0.791 \text{ kg/s}}$

Question: 7.137P

Prove that a refrigerator's COP cannot exceed that of a completely reversible refrigerator that shares the same thermal energy reservoirs.

Step 1 of 4

Consider two refrigerators A and B with completely reversible and irreversible characteristics respectively.



Assume the COP of the general refrigerator B is greater than that of a completely reversible refrigerator A.
i.e. $COP_B > COP_A$

Step 2 of 4

From the definition of COP of a refrigerator, we get

$$\frac{Q_L}{W_B} > \frac{Q_L}{W_A}$$

$$W_B < W_A$$

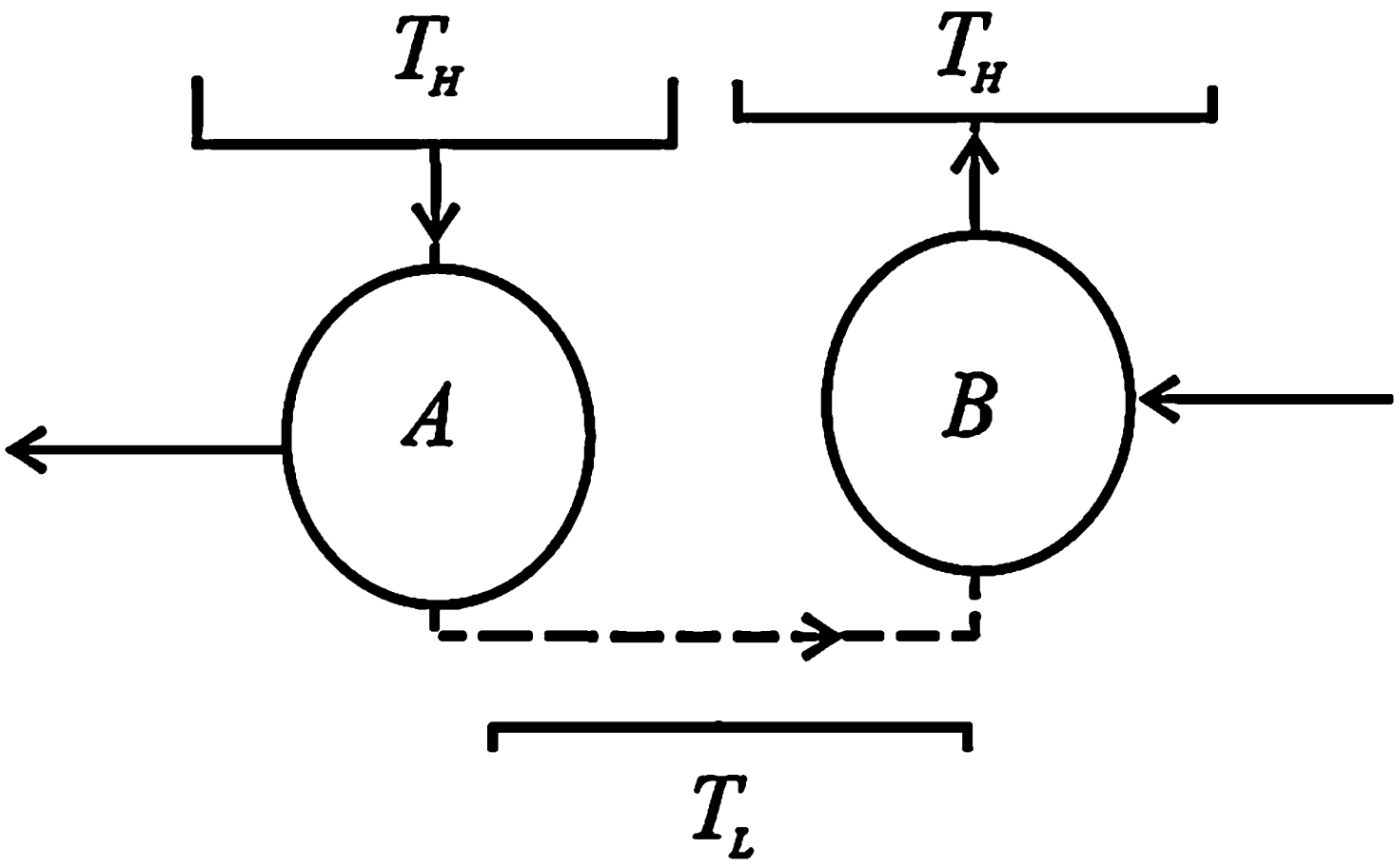
Hence, the work required to drive refrigerator B is less than needed to drive reversible refrigerator A.

Also note that from the first law, assuming the same cooling effect Q_L, the heat rejected by the refrigerators is equal to

$$Q_{H_B} < Q_{H_A}$$

Step 3 of 4

Since A is a completely reversible refrigerator, we can reverse it without changing the magnitude of the heat and work transfers. The heat Q_L, which is rejected by the reversed refrigerator A can now be routed directly to refrigerator B as shown below



Thus, no heat is to be exchanged with the T_L reservoir. The magnitude of the heat supplied to the reversed refrigerator A, Q_{H_A} has been shown to be larger than that rejected by refrigerator B.

Step 4 of 4

Upon observation, there is then a net heat transfer from the T_H reservoir to the combined device whose magnitude is given by $Q_{H_B} - Q_{H_A}$. Similarly, there is a net work production by the combined device whose magnitude is given by $W_B - W_A$

The combined cyclic device then exchanges heat with a reservoir at a single temperature and produces work which is clearly a violation of the Kelvin-Planck statement of the second law.

Therefore the assumption of $COP_B > COP_A$ is wrong. A refrigerator's COP cannot exceed the COP of a completely reversible refrigerator when operating between the same temperature limits.

Question: 7.138P

Prove that the COP of all completely reversible refrigerators must be the same when the reservoir temperatures are the same.

Step 1 of 2

Initially consider the COP of the general refrigerator *A* is lesser than that of the refrigerator *B*.

$$\mathrm{COP}_A < \mathrm{COP}_B \quad \text{..... (1)}$$

Write the relation for COP of refrigerator *A*.

$$\mathrm{COP}_A = \frac{Q_L}{W_A}$$

Here, Q_L is the heat rejected and W_A is the work done by the refrigerator *A*.

Write the relation for COP of refrigerator *B*.

$$\mathrm{COP}_B = \frac{Q_L}{W_B}$$

Here, W_B is the work done by the refrigerator *B*.

Substitute COP's of refrigerators *A* and *B* in equation (1).

$$\frac{Q_L}{W_A} < \frac{Q_L}{W_B}$$

$$W_A > W_B$$

Thus, the work required to drive refrigerator *B* needs more to drive reversible refrigerator *A*.

Step 2 of 2

Obtain the relation by using the first law of thermodynamics to both the refrigerators.

$$Q_{H_A} > Q_{H_B}$$

The magnitude of heat transfer and the magnitude of work transfer because refrigerator *A* is a completely reversible refrigerator.

The heat rejected by the reversed refrigerator *A*, Q_L is routed directly to refrigerator *B*. In this process no heat is exchanged with the reservoir at temperature T_L is the net effect and the magnitude of the heat supplied to the reversed refrigerator *A*, Q_{H_A} is larger than heat rejected by refrigerator *B* which results in a net heat transfer of magnitude $Q_{H_A} - Q_{H_B}$ from the reservoir at a temperature T_H to the combined device.

Similarly, net-work of magnitude $W_A - W_B$ is produced by the combined device. Now the combined cyclic device exchanges heat with a reservoir at a single temperature and produces work violating the Kelvin-Planck statement of the second law.

This clearly states that the assumption of $\mathrm{COP}_A < \mathrm{COP}_B$ is wrong and the assumption is correct only when the reservoir temperatures are the same.

Therefore, the COP of all completely reversible refrigerators will be the same.

Question: 7.139P

Derive an expression for the COP of a completely reversible heat pump in terms of the thermal energy reservoir temperatures, *TL*and *TH*.

Step 1 of 4

Let the higher thermal reservoir is T_H

The lower thermal reservoir is T_L

Step 2 of 4

The COP of a heat pump is expressed as

$$\text{COP}_{HP} = \frac{\text{Desired output}}{\text{input}}$$

$$\text{COP}_{HP} = \frac{Q_H}{W_{net,in}}$$

From first law of thermodynamics we know that

$$W_{net} = Q_H - Q_L$$

Substituting in above relation

$$\text{COP}_{HP} = \frac{Q_H}{Q_H - Q_L}$$

$$\text{COP}_{HP} = \frac{1}{1 - Q_L / Q_H}$$

Step 3 of 4

From seconding of thermodynamics, for a reversible heat pump the temperatures are expressed as

$$\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$$

Substituting in the COP expression

$$\text{COP}_{HP} = \frac{1}{1 - T_L / T_H}$$

$$\text{COP}_{HP} = \frac{T_H}{T_H - T_L}$$

Step 4 of 4

The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{HP} = \frac{T_H}{T_H - T_L}$$

∴ The expression for COP $\boxed{\text{COP}_{HP} = \frac{T_H}{T_H - T_L}}$

Question:

7.140P

A Carnot heat engine is operating between a source at *TH* and a sink at *TL*. If it is desired to double the thermal efficiency of this engine, what should the new source temperature be? Assume the sink temperature is held constant.

Step 1 of 1

The Carnot heat engine operating between source temperature T_H and sink T_L has thermal efficiency

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

The thermal efficiency gets doubled with sink temperature remaining constant

$$\eta_{th_new} = 1 - \frac{T_L}{T_{H_new}}$$

$$\eta_{th_new} = 2\eta_{th}$$

$$1 - \frac{T_L}{T_{H_new}} = 2\left(1 - \frac{T_L}{T_H}\right)$$

$$1 - \frac{T_L}{T_{H_new}} = 2 - 2\frac{T_L}{T_H}$$

$$2\frac{T_L}{T_H} - \frac{T_L}{T_{H_new}} = 1$$

$$\frac{2T_L\left(T_{H_new}\right) - T_L T_H}{T_H T_{H_new}} = 1$$

$$\frac{T_{H_new}\left(2T_L - \dfrac{T_L T_H}{T_{H_new}}\right)}{T_H T_{H_new}} = 1$$

$$\frac{\left(2T_L - \dfrac{T_L T_H}{T_{H_new}}\right)}{T_H} = 1$$

$$2T_L - T_H = \frac{T_L T_H}{T_{H_new}}$$

$$T_{H_new} = \frac{T_L T_H}{2T_L - T_H}$$

∴ The new source temperature is $\boxed{T_{H_new} = \dfrac{T_L T_H}{2T_L - T_H}}$

CHAPTER 08

SOLUTIONS

Question: 8.001P

Does the cyclic integral of work have to be zero (i.e., does a system have to produce as much work as it consumes to complete a cycle)? Explain.

Step 1 of 1

No, the cyclic integral of work need not be zero. For a cycle the system can produce less or more amount of work than it receives. The difference is the network output.

So, the cyclic integral of *work* is not zero. Otherwise, heat engines that work on a cycle such as steam power plants would produce zero net-work.

Question: 8.002P

A system undergoes a process between two fixed states first in a reversible manner and then in an irreversible manner. For which case is the entropy change greater? Why?

Step 1 of 1

Entropy is a property. Like all other properties it has fixed values at fixed states. Therefore, the Entropy change between two specified states is same no matter what path is. It can be reversible or irreversible. So, in this case the entropy change will be same in both the cases.

Question: 8.003P

Is the value of the integral $\int_1^2 \delta Q/T$ the same for all processes between states 1 and 2? Explain.

Step 1 of 1

No.

Reason: Depending upon the process the integral value may vary for an irreversible process. If it is a reversible process the integral value will be same for all the states.

Note that the integral of dQ/T gives us the value of entropy change only if the integration is carried out along an internally reversible path between the two states. The Entropy change between two specified states is same no matter what path is.

The integral of dQ/T along an irreversible path is not a property, and in general, different values will be obtained when the integration is carried out along different irreversible paths.

Question: 8.004P

To determine the entropy change for an irreversible process between states 1 and 2, should the integral $\int_1^2 \delta Q/T$ be performed along the actual process path or an imaginary reversible path? Explain.

Step 1 of 1

The integral value of $\delta Q/T$ gives the value of entropy change if the integration is carried out along an internally reversible path between two states. The integral value of $\delta Q/T$ along an irreversible path is not a property. So, different values will be obtained when integration is carried out on different irreversible paths. Therefore, even for irreversible process entropy change should be determined by carrying out this integration along some convenient imaginary internally reversible path between the specified states.

Question: 8.005P

Is an isothermal process necessarily internally reversible? Explain your answer with an example.

Step 1 of 1

No.

Reason: An isothermal process need not be internally reversible. It can be irreversible also. Consider a cylinder containing water. It is receiving 100KJ of heat and at the same time its losing same amount of heat to the external environment. This is an irreversible isothermal process

Question: 8.006P

How do the values of the integral $\int_1^2 \delta Q/T$ compare for a reversible and irreversible process between the same end states?

Step 1 of 1

For the same states the value of the given integral is always greater for reversible process.

Explanation: The integral of dQ/T gives us the value of entropy change only if the integration is carried out along an internally reversible path between the two states.

The relation can be expressed as $dS \geq \frac{\delta Q}{T}$.

Where, the equality holds for an internally reversible process and the inequality for an irreversible process. We may conclude from these equations that the entropy change of a closed system during an irreversible process is greater than the integral of dQ/T evaluated for that process.

Question: 8.007P

The entropy of a hot baked potato decreases as it cools. Is this a violation of the increase of entropy principle? Explain.

Step 1 of 1

No.

Reason: This is not a violation of the increase of entropy principle. As the potato cools, the air surrounding gains heat and its entropy increase even more. This makes the total entropy change positive.

Question: 8.008P

Is it possible to create entropy? Is it possible to destroy it?

Step 1 of 1

The entropy can be created but it cannot be destroyed.
Explanation: The property entropy is a measure of molecular disorder or randomness of a system, and the second law of thermodynamics states that entropy can be created but it cannot be destroyed.

Question: 8.009P

When a system is adiabatic, what can be said about the entropy change of the substance in the system?

Step 1 of 1

$\Delta S \geq 0$

If the system goes through a reversible process, the entropy of the system remains same as there is not heat transfer. If it is an irreversible process the entropy must increase. In other words, it never decreases. In the absence of any heat transfer, entropy change is due to irreversibility's only, and their effect is always to increase entropy

Question: 8.010P

Work is entropy free, and sometimes the claim is made that work will not change the entropy of a fluid passing through an adiabatic steady-flow system with a single inlet and outlet. Is this a valid claim?

Step 1 of 1

Only those processes that are reversible adiabatic, will keep the entropy unaltered. So the claim that work will not change the entropy of a fluid passing through a reversible adiabatic steady flow system is true. But in real, no process is reversible and hence the entropy of a fluid will always increase.

Question: 8.011P

A piston-cylinder device contains helium gas. During a reversible, isothermal process, the entropy of the helium will (*never* , *sometimes* , *always*) increase.

Step 1 of 1

Entropy of a reversible adiabatic process is constant. But for reversible isothermal process, the entropy may increase sometimes. This is because isothermal process can exchange heat with surroundings.

Question: 8.012P

A piston-cylinder device contains nitrogen gas. During a reversible, adiabatic process, the entropy of the nitrogen will (*never* , *sometimes* , *always*) increase.

Step 1 of 1

As it is a reversible adiabatic process the entropy will never increase.

$\Rightarrow \Delta S = 0$

Question: 8.013P

A piston-cylinder device contains superheated steam. During an actual adiabatic process, the entropy of the steam will (*never* , *sometimes* , *always*) increase.

Step 1 of 1

The entropy always increases as actual processes are never reversible. They are always irreversible.

$\Rightarrow \Delta S > 0$

Question:

8.014P

The entropy of steam will (*increase* , *decrease* , *remain the same*) as it flows through an actual adiabatic turbine.

Step 1 of 1

The entropy of this system always increases as actual processes are never reversible.

$\Rightarrow \Delta S > 0$

Question: 8.015P

The entropy of the working fluid of the ideal Carnot cycle (*increases, decreases, remains the same*) during the isothermal heat addition process.

Step 1 of 1

The heat is adding here but the temperature remains same (isothermal process) so, the entropy increases.

Question: 8.016P

The entropy of the working fluid of the ideal Carnot cycle (*increases, decreases, remains the same*) during the isothermal heat rejection process.

Step 1 of 1

The heat is rejected here but the temperature remains same so, the entropy decreases

Question: 8.017P

During a heat transfer process, the entropy of a system (*always* , *sometimes* , *never*) increases.

Step 1 of 1

The direction of entropy transfer is the same as the direction of heat transfer since the thermodynamic temperature T is always a positive quantity.

The heat transfer into the system increases the entropy, whereas the heat transfer out of the system decreases the entropy.

Hence, the entropy of a system increases some times.

Question: 8.018P

Steam is accelerated as it flows through an actual adiabatic nozzle. The entropy of the steam at the nozzle exit will be (*greater than* , *equal to* , *less than*) the entropy at the nozzle inlet.

Step 1 of 1

As it is irreversible adiabatic process, entropy will increase. So the entropy at the nozzle exit is greater than that of inlet.

Question: 8.019P

Is it possible for the entropy change of a closed system to be zero during an irreversible process? Explain.

Step 1 of 1

Yes

Reason: It can be zero. Consider a system losing heat. The entropy should decrease as its losing heat. But, if the decrease in the entropy is same the increase due to the irreversibility's then the total increase in entropy can be zero.

Question: 8.020P

What three different mechanisms can cause the entropy of a control volume to change?

Step 1 of 1

The three different mechanisms that cause the entropy of a control volume to change are

- Heat transfer
- Irreversibilities
- Entropy transport with mass

Question:

8.021P

A completely reversible heat engine operates with a source at 1500 R and a sink at 500 R. If the entropy of the sink increases by 10 Btu/R, how much will the entropy of the source decrease? How much heat, in Btu, is transferred from the source?

Step 1 of 3

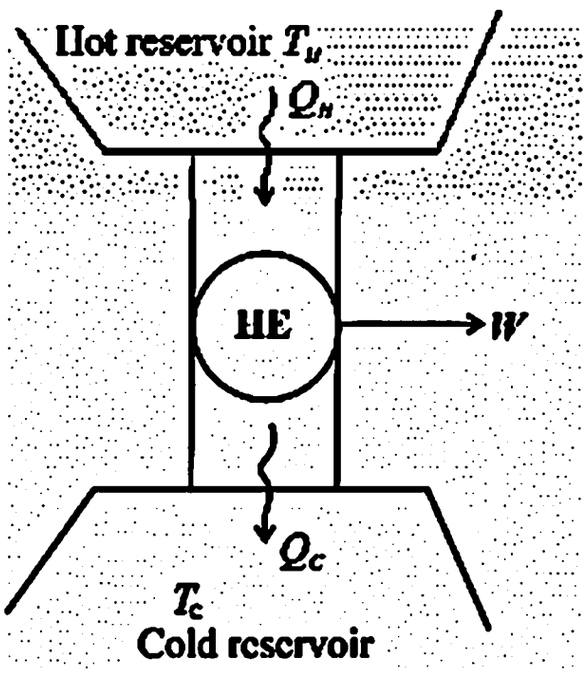
Given

Source temperature is, $T_{source} = T_H = 1500\ \text{R}$

Sink temperature is, $T_{sink} = T_C = 500\ \text{R}$

Entropy of the sink increases by $\Delta S_{sink} = 10\ \text{Btu/R}$

Step 2 of 3



Step 3 of 3

As it is given that the heat engine is reversible, the entropy generation will be zero. So,

$S_{gen} = \Delta S_{source} + \Delta S_{sink}$

$0 = \Delta S_{source} + \Delta S_{sink}$

$\Delta S_{source} = -\ \Delta S_{sink}$

$\Delta S_{source} = -\ 10\ \text{Btu/R}$

The entropy change in the source can be given by

$$\Delta S_{source} = \frac{Q_{source}}{T_{source}}$$

Therefore the heat transfer from the source is given by

$Q_{source} = \Delta S_{source} \times T_{source}$

$Q_{source} = -10 \times 1500$

$\boxed{Q_{source} = -15000\ \text{Btu}}$

Therefore the heat transfer from the source is 15000 Btu

Question: 8.022P

An energy-source at 1000 K transfers heat to a completely reversible heat engine. This engine transfers heat to a sink at 300 K. How much heat must be transferred from the energy-source to increase the entropy of the energy-sink by 20 kJ/K?

Step 1 of 5

Given data:

Source temperature of the heat engine, $T_1 = 1000\ \text{K}$

Sink temperature of the heat engine, $T_2 = 300\ \text{K}$

Increased entropy for sink, $20\ \text{kJ/K}$

Step 2 of 5

Let

The Heat transferred for sink be Q_1

The Heat transferred for sink be Q_2

Change of entropy be Δs

Reversible process, the change of entropy becomes zero, $\Delta s = 0$

For reversible process,

$$\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = \Delta s \quad \text{..... (1)}$$

Since heat transferred for sink be increased by $20\ \text{kJ/K}$, so equation (1), can be re written as

$$\frac{Q_1}{T_1} + \frac{Q_2 + 20}{T_2} = 0 \quad \text{..... (2)}$$

Step 3 of 5

Substituting T_1 and T_2 in equation (2), we get

$$\frac{Q_1}{1000} + \frac{Q_2 + 20}{300} = 0$$

$$300Q_1 + 1000Q_2 + 2000 = 0 \quad \text{..... (3)}$$

As we know that,

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

$$Q_1 = \frac{T_1}{T_2} \times Q_2$$

$$Q_1 = \frac{1000}{300} Q_2$$

$$Q_1 = 3.333 Q_2 \quad \text{..... (4)}$$

Step 4 of 5

Substituting Q_1 in equation (3), we get

$$300 \times 3.33 Q_2 + 1000 Q_2 + 20000 = 0$$

$$1999 Q_2 = -20000$$

$$Q_2 = \frac{-20000}{1999}$$

$$= -10.005\ \text{kJ}$$

Step 5 of 5

Substituting Q_2 value in equation (4), we get

$$Q_1 = 3.333(-10.005)$$

$$= -33.34\ \text{kJ}$$

Here negative sign indicates that heat rejections of the system.

Hence, the energy required to increased entropy, $\boxed{Q_1 = -33.34\ \text{kJ}}$

Question: 8.023P

A heat engine accepts 200,000 Btu of heat from a source at 1500 R and rejects 100,000 Btu of heat to a sink at 600 R. Calculate the entropy change of all the components of this engine and determine if it is completely reversible. How much total work does it produce?

Step 1 of 5

Given

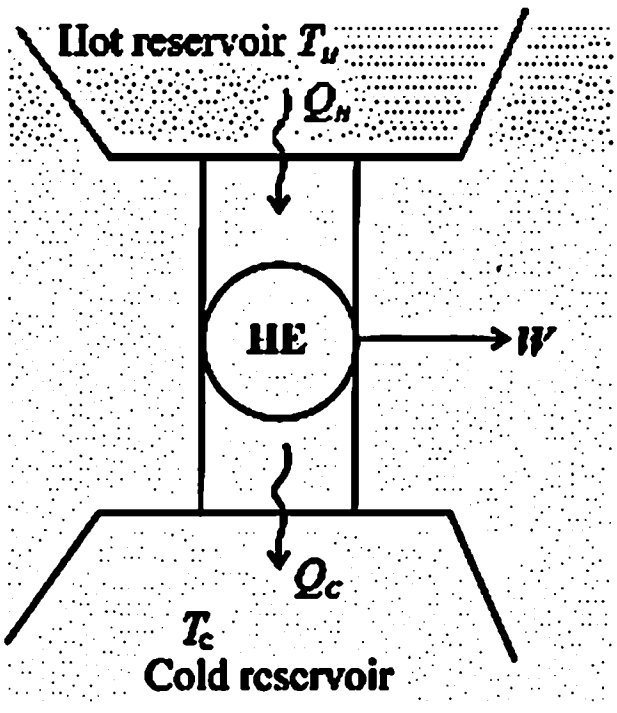
Heat released from the source is, $Q_{source} = 200000\ \text{Btu}$

Source temperature is, $T_{source} = T_H = 1500\ \text{R}$

Sink temperature is, $T_{sink} = T_C = 600\ \text{R}$

Heat rejected to sink from heat engine is, $Q_{sink} = 100000\ \text{Btu}$

Step 2 of 5



Step 3 of 5

Work done by heat engine is given by

$W = Q_{source} - Q_{sink}$

$W = 200000 - 100000$

$\boxed{W = 100000\ \text{Btu}}$

Step 4 of 5

The entropy change of source is given by

$$\Delta S_{source} = \frac{Q_{source}}{T_{source}}$$

As heat is coming out of the source it is taken as negative,

$$\Delta S_{source} = \frac{-200000}{1500}$$

$\Delta S_{source} = -133.33\ \text{Btu/R}$

Similarly, the entropy change of sink is given by

$$\Delta S_{sink} = \frac{Q_{sink}}{T_{sink}}$$

$$\Delta S_{sink} = \frac{100000}{600}$$

$\Delta S_{sink} = 166.67\ \text{Btu/R}$

Step 5 of 5

The entropy generation can be given by

$S_{gen} = \Delta S_{source} + \Delta S_{sink}$

$S_{gen} = -133.33 + 166.67$

$S_{gen} = 33.34\ \text{Btu/R}$

Since $S_{gen} \neq 0$ the heat engine is not reversible.

Therefore the work done by the heat engine is 100000 Btu and the heat engine is not reversible.

Question: 8.024P

Air is compressed by a 30-kW compressor from $P1$, to $P2$. The air temperature is maintained constant at 25°C during this process as a result of heat transfer to the surrounding medium at 17°C. Determine the rate of entropy change of the air. State the assumptions made in solving this problem.

Step 1 of 4

Given

Compressor power input $\dot{W}_{in} = 30\ \text{kW}$

Temperature of air in the compressor $T = 25°\text{C} = 298\ \text{K}$

Temperature of the surroundings $T_{sur} = 17°\text{C} = 290\ \text{K}$

Step 2 of 4

Let us assume,

The process is steady

Kinetic and potential energies are negligible

The process is reversible

Energy balance is given by

$$E_{in} - E_{out} = \frac{dE_{sys}}{dt}$$

As the process is steady, the right hand side in the above equation becomes zero. So,

$E_{in} - E_{out} = 0$

$E_{in} = E_{out}$ ..... (1)

Step 3 of 4

The energy input to the compressor is in terms of work. Therefore,

$E_{in} = \dot{W}_{in}$

$E_{in} = 30\ \text{kW}$

As the process is isothermal, the energy output to the compressor is in terms of heat. Therefore,

$E_{out} = \dot{Q}_{out}$

Substituting the values E_{in} and E_{out} in equation (1) we get,

$30\ \text{kW} = \dot{Q}_{out}$

$\dot{Q}_{out} = 30\ \text{kW}$

Step 4 of 4

As the process is reversible, the change in entropy is given by

$$\Delta\dot{S} = -\frac{\dot{Q}_{out}}{T}$$

$$\Delta\dot{S} = -\frac{30}{298}$$

$$\boxed{\Delta\dot{S} = -\ 0.1007\ \text{kW/K}}$$

Therefore the rate of entropy change of the air decreases by 0.1007 kW/K

Question: 8.025P

Heat in the amount of 100 kJ is transferred directly from a hot reservoir at 1200 K to a cold reservoir at 600 K. Calculate the entropy change of the two reservoirs and determine if the increase of entropy principle is satisfied.

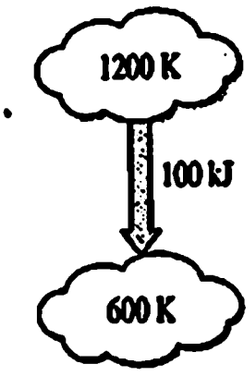


FIGURE P8–25

Step 1 of 4

Given:

Temperature at hot reservoir, $T_H = 1200\ \text{K}$

Temperature at cold reservoir, $T_L = 600\ \text{K}$

Amount of heat transferred, $Q = 100\ \text{kJ}$

Step 2 of 4

Entropy change at hot reservoir, $\Delta S_H = \dfrac{Q_H}{T_H}$

$$\Delta S_H = \frac{-100}{1200}$$

$$\Delta S_H = -0.083333\ \frac{\text{kJ}}{\text{k}}$$

Step 3 of 4

Entropy change at cold reservoir, $\Delta S_L = \dfrac{Q_L}{T_L}$

$$\Rightarrow \Delta S_L = \frac{100}{600}$$

$$\Rightarrow \Delta S_L = 0.166666\ \frac{\text{kJ}}{\text{k}}$$

Step 4 of 4

Total entropy change,

$$\Delta S = \Delta S_H + \Delta S_L$$

$$\Rightarrow \Delta S = -0.083333 + 0.166666$$

$$\Rightarrow \boxed{\Delta S = 0.083333\ \frac{\text{kJ}}{\text{k}}}$$

ΔS is not less than zero, so it satisfies the increase of entropy principle.

Question: 8.-26ASSUME-H

In the previous problem, assume that the heat is transferred from the cold reservoir to the hot reservoir contrary to the Clausius statement of the second law. Prove that this violates the increase of entropy principle—as it must according to Clausius.

Step 1 of 4

Given:

Temperature at hot reservoir, $T_H = 1200\ \text{K}$

Temperature at cold reservoir, $T_L = 600\ \text{K}$

Amount of heat transferred, $Q = 100\ \text{kJ}$

Given that the heat is transferred from the cold reservoir to the hot reservoir.

Step 2 of 4

Entropy change at hot reservoir, $\Delta S_H = \frac{Q_H}{T_H}$

$$\Rightarrow \Delta S_H = \frac{100}{1200}$$

$$\Rightarrow \Delta S_H = 0.083333\ \frac{\text{kJ}}{\text{k}}$$

Step 3 of 4

Entropy change at cold reservoir, $\Delta S_L = \frac{Q_L}{T_L}$

$$\Rightarrow \Delta S_L = \frac{-100}{600}$$

$$\Rightarrow \Delta S_L = -0.166666\ \frac{\text{kJ}}{\text{k}}$$

Step 4 of 4

Total entropy change,

$$\Delta S = \Delta S_H + \Delta S_L$$

$$\Rightarrow \Delta S = 0.083333 - 0.166666$$

$$\Rightarrow \boxed{\Delta S = -0.083333\ \frac{\text{kJ}}{\text{k}}}$$

ΔS is less than zero, so it violates the increase of entropy principle.

Question: 8.027P

A completely reversible heat pump produces heat at rate of 300 kW to warm a house maintained at 24°C. The exterior air, which is at 7°C, serves as the source. Calculate the rate of entropy change of the two reservoirs and determine if this heat pump satisfies the second law according to the increase of entropy principle.

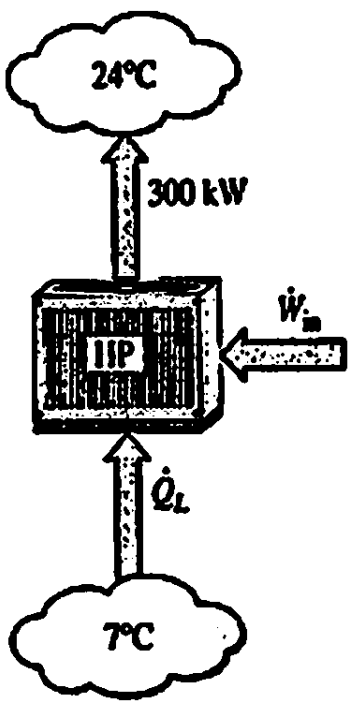


FIGURE P8–27

Step 1 of 5

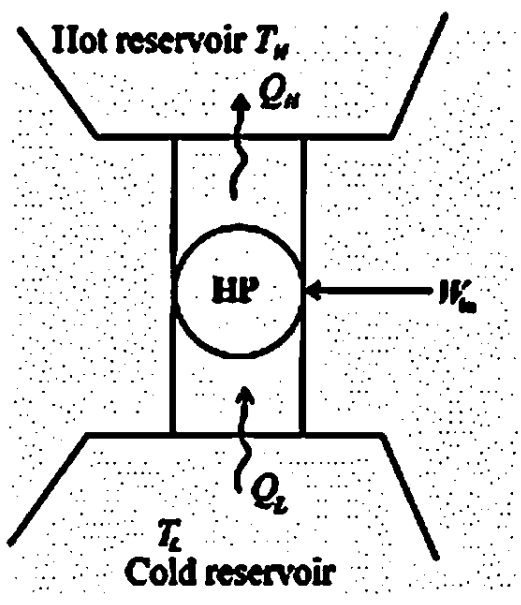
Given

Temperature of warm house (source) is, $T_H = 24°C = 297\ K$

Temperature of exterior air (sink) is, $T_L = 7°C = 280\ K$

Heat produced by heat pump is, $Q_H = 300\ kW$

Step 2 of 5



Step 3 of 5

The coefficient of performance of heat pump is given by

$$(C.O.P)_{HP} = \frac{Q_H}{W_{in}}$$

By applying first law to the heat pump

$$-W_{in} = -Q_H + Q_L$$

$$W_{in} = Q_H - Q_L$$

So C.O.P of heat pump can be written as

$$(C.O.P)_{HP} = \frac{Q_H}{Q_H - Q_L} \quad \ldots\ldots (1)$$

The coefficient of performance of heat pump is also given by

$$(C.O.P)_{HP} = \frac{T_H}{T_H - T_L} \quad \ldots\ldots (2)$$

Equating equations (1) and (2) we get,

$$\frac{Q_H}{Q_H - Q_L} = \frac{T_H}{T_H - T_L}$$

$$\frac{300}{300 - Q_L} = \frac{297}{297 - 280}$$

$$\frac{300}{300 - Q_L} = \frac{297}{17}$$

$$\frac{300 - Q_L}{300} = \frac{17}{297}$$

$$300 - Q_L = 300 \times \frac{17}{297}$$

$$300 - Q_L = \frac{5100}{297}$$

$$300 - Q_L = 17.17$$

$$Q_L = 300 - 17.17$$

$$Q_L = 282.83\ kW$$

Step 4 of 5

The change in entropy of source is given by

$$\Delta S_H = \frac{Q_H}{T_H}$$

$$\Delta S_H = \frac{300}{297}$$

$$\boxed{\Delta S_H = 1.01\ kW/K}$$

The change in entropy of sink is given by

$$\Delta S_L = \frac{Q_L}{T_L}$$

As heat rejected from the sink therefore Q_L is negative

$$\Delta S_L = \frac{-282.83}{280}$$

$$\boxed{\Delta S_L = -1.01\ kW/K}$$

Step 5 of 5

The entropy generation is given by

$$S_{gen} = \Delta S_H + \Delta S_L$$

$$S_{gen} = 1.01 + (-1.01)$$

$$S_{gen} = 1.01 - 1.01$$

$$\boxed{S_{gen} = 0}$$

Therefore the entropy of source is increased by 1.01 kW/K , the entropy of sink is decreased by 1.01 kW/K . As the entropy generation is zero it satisfies the second law according to increase of entropy principle.

Question: 8.028P

During the isothermal heat rejection process of a Carnot cycle, the working fluid experiences an entropy change of -0.7 Btu/R. If the temperature of the heat sink is 95°F, determine (a) the amount of heat transfer, (*b*) the entropy change of the sink, and (c) the total entropy change for this process.

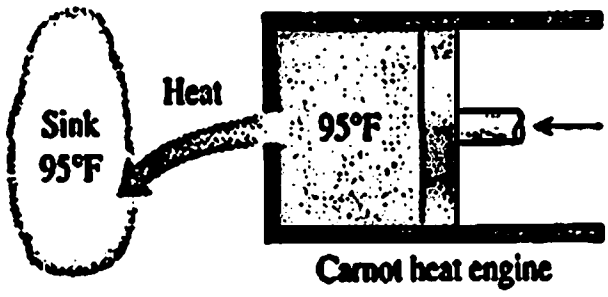


FIGURE P8–28E

Step 1 of 4

Given:

For a carnot cycle, the process is isothermal heat rejection.

Temperature at sink and fluid, $T_{sink} = T_{fluid} = 95°\,\text{F}$

$$\Rightarrow T_{sink} = T_{fluid} = 555\,\text{R}$$

Entropy change at fluid, $\Delta \dot{S}_{fluid} = -0.7\ \text{Btu/R}$

Step 2 of 4

(a) Amount of heat transfer:

Heat transferred from the working fluid,

$$Q_{fluid} = T_{fluid} \times \Delta S_{fluid}$$

$$\Rightarrow Q_{fluid} = 555 \times -0.7$$

$$\Rightarrow Q_{fluid} = -388.5\ \text{Btu}$$

Heat transferred from the working fluid is equal to heat transferred to sink.

So, $Q_{sink} = 388.5\ \text{Btu}$

Amount of heat transfer, $\boxed{Q = 388.5\ \text{Btu}}$

Step 3 of 4

(b) Entropy change of the sink:

Entropy change of the sink,

$$\Delta S_{sink} = \frac{Q_{sink}}{T_{sink}}$$

$$\Rightarrow \Delta S_{sink} = \frac{388.5}{555}$$

$$\Rightarrow \boxed{\Delta S_{sink} = 0.7\ \frac{\text{Btu}}{\text{R}}}$$

Step 4 of 4

(c) Total entropy change during the process:

Total entropy change during the process,

$$\Delta S_{total} = \Delta S_{sink} + \Delta S_{fluid}$$

$$\Rightarrow \Delta S_{total} = 0.7 - 0.7$$

$$\Rightarrow \boxed{\Delta S_{total} = 0\ \frac{\text{Btu}}{\text{R}}}$$

Question: 8.029P

Refrigerant-134a enters the coils of the evaporator of a refrigeration system as a saturated liquid-vapor mixture at a pressure of 160 kPa. The refrigerant absorbs 180 kJ of heat from the cooled space, which is maintained at -5°C, and leaves as saturated vapor at the same pressure. Determine (*a*) the entropy change of the refrigerant, (*b*) the entropy change of the cooled space, and (*c*) the total entropy change for this process.

Step 1 of 5

Given:

For Refrigerant-134a,

Pressure of saturated liquid-vapor mixture, $P = 160\ \text{kPa}$

Temperature of cold space, $T_{space} = -5^{\circ}\ \text{C}$

$$\Rightarrow T_{cold} = 268\ \text{K}$$

Heat absorbed from sold space, $Q_{absorbed} = 180\ \text{kJ}$

Step 2 of 5

From saturated For Refrigerant-134a pressure Table

Temperature of saturated liquid-vapor mixture, $T_{mixture} = -15.60^{\circ}\ \text{C}$

$$\Rightarrow T_{mixture} = 257.4\ \text{K}$$

Step 3 of 5

(a) Entropy change of the refrigerant:

Entropy change of the refrigerant, $\Delta S_{refrigerant} = \dfrac{Q_{absorbed}}{T_{mixture}}$

$$\Rightarrow \Delta S_{refrigerant} = \frac{180}{257.4}$$

$$\Rightarrow \boxed{\Delta S_{refrigerant} = 0.7\ \frac{\text{kJ}}{\text{K}}}$$

Step 4 of 5

(b) Entropy change of the cooled space:

Entropy change of the cooled space, $\Delta S_{space} = \dfrac{Q_{absorbed}}{T_{space}}$

$$\Rightarrow \Delta S_{space} = -\frac{180}{268}$$

$$\Rightarrow \boxed{\Delta S_{space} = -0.6716\ \frac{\text{kJ}}{\text{K}}}$$

Step 5 of 5

(c) Total entropy change for this process:

Total entropy change for this process,

$$\Delta S_{total} = \Delta S_{space} + \Delta S_{refrigerant}$$

$$\Rightarrow \Delta S_{total} = -0.6716 + 0.7$$

$$\Rightarrow \boxed{\Delta S_{total} = 0.0284\ \frac{\text{kJ}}{\text{K}}}$$

Question: 8.030P

Is a process that is internally reversible and adiabatic necessarily isentropic? Explain.

Step 1 of 1

YES.

Reason: The entropy of a fixed mass does not change during a process that is internally reversible and adiabatic. A process during which the entropy remains constant is called an isentropic process.

Question: 8.031P

2-lbm of water at 300 psia fill a weighted piston-cylinder device whose volume is 2.5 ft3 . The water is then heated at constant pressure until the temperature reaches 500°F. Determine the resulting change in the water's total entropy.

Step 1 of 5

Given: For water,
Mass, $m = 2$ lbm
Pressure, $P = 300$ psia
Volume, $V = 2.5\ \text{ft}^3$
Final temperature, $T_2 = 500^\circ\text{F}$

Step 2 of 5

Initial specific volume, $v_1 = \dfrac{V}{m}$

$$\Rightarrow v_1 = \frac{2.5}{2}$$

$$\Rightarrow v_1 = 1.25\ \frac{\text{ft}^3}{\text{lbm}}$$

From saturated water –pressure Table, at $P = 300$ psia

Initial quality, $x_1 = \dfrac{v_1 - v_f}{v_{fg}}$

$$\Rightarrow x_1 = \frac{v_1 - v_f}{v_g - v_f}$$

$$\Rightarrow x_1 = \frac{1.25 - 0.01890}{1.5435 - 0.01890}$$

$$\Rightarrow x_1 = 0.80749$$

Initial entropy, $s_1 = s_f + x_1 s_{fg}$

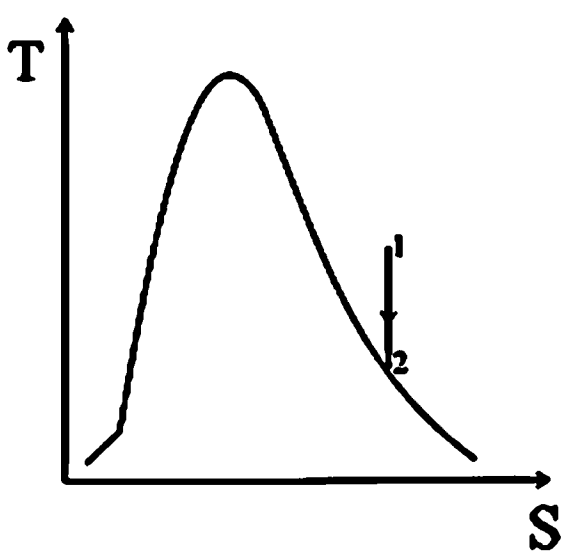
$$\Rightarrow s_1 = 0.58818 + 0.80749 \times 0.92289$$

$$\Rightarrow s_1 = 1.333404\ \frac{\text{Btu}}{\text{lbm} \cdot \text{R}}$$

Step 3 of 5

At final state:
From super heated water tables
The final temperature $T_2 = 500^\circ$ F,
The final pressure $P_2 = 300$ psia
The final entropy $s_2 = 1.5706\ \dfrac{\text{Btu}}{\text{lbm.R}}$

Step 4 of 5



Step 5 of 5

Total entropy change,

$$\Delta S = m(s_2 - s_1)$$

$$\Rightarrow \Delta S = 2 \times (1.5706 - 1.333404)$$

$$\Rightarrow \boxed{\Delta S = 0.4744\ \frac{\text{Btu}}{\text{R}}}$$

Question: 8.032P

A well-insulated rigid tank contains 5 kg of a saturated liquid-vapor mixture of water at 150 kPa. Initially, three-quarters of the mass is in the liquid phase. An electric resistance heater placed in the tank is now turned on and kept on until all the liquid in the tank is vaporized. Determine the entropy change of the steam during this process.

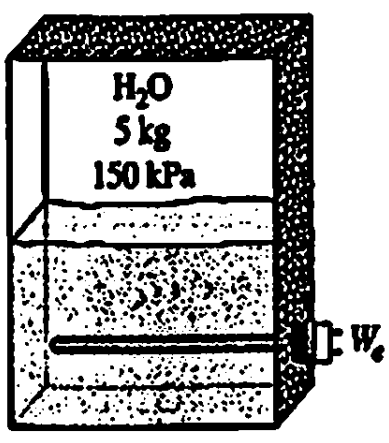


FIGURE P8–32

Step 1 of 8

Given data:

Mass of mixture in the tank is, $m = 5\,kg$.

Pressure in the tank is, $P = 150\,kPa$.

Step 2 of 8

From the saturated water pressure table,

At pressure $P = 150\,kPa$

Entropy of saturated liquid is $s_{f1} = 1.4337\,kJ/kg \cdot K$

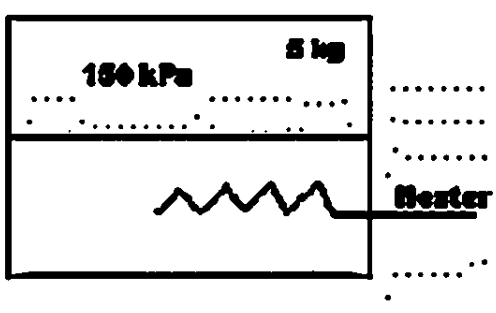
Entropy of evaporation is $s_{fg1} = 5.7894\,kJ/kg \cdot K$

Specific volume of saturated liquid is $v_{f1} = 0.001053\,m^3/kg$

Specific volume of saturated vapor is $v_{g1} = 1.1594\,m^3/kg$

Step 3 of 8

Consider the diagram:



Step 4 of 8

As it is given initially three-fourth of mass is liquid phase, the mass of water is

$$m_w = \frac{3}{4}$$
$$= 0.75$$

Therefore mass of vapor is

$$m_v = 1 - m_w$$
$$= 1 - 0.75$$
$$= 0.25$$

Dryness fraction is the fraction of vapor in the mixture $x_1 = m_v$

So the dryness of the mixture initially is given by $x_1 = 0.25$

Step 5 of 8

Initial specific volume of mixture is given by

$$v_1 = v_{f1} + x_1 \times v_{fg1}$$
$$= v_{f1} + x_1 \times (v_{g1} - v_{f1})$$
$$= 0.001053 + 0.25 \times (1.1594 - 0.001053)$$
$$= 0.001053 + 0.25 \times 1.158347$$
$$= 0.001053 + 0.289587$$
$$= 0.29064\,m^3/kg$$

Step 6 of 8

Initial specific entropy of mixture is given by

$$s_1 = s_{f1} + x_1 \times s_{fg1}$$

As the pressure is constant the equation can be written as

$$s_1 = s_f + x_1 \times s_{fg}$$
$$= 1.4337 + 0.25 \times 5.7894$$
$$= 1.4337 + 1.4474$$
$$= 2.8811\,kJ/kg \cdot K$$

Step 7 of 8

The final state is saturated vapor, and as the tank is closed, the final specific volume of mixture will be same as initial.

$$v_2 = v_1$$

Therefore, $v_2 = 0.29064\,m^3/kg$

From the saturated water pressure table,

At specific volume $v_2 = 0.29064\,m^3/kg$,

The specific entropy is interpolated as

$$\frac{0.29260 - v_2}{0.29260 - 0.27278} = \frac{6.7322 - s_2}{6.7322 - 6.7071}$$
$$\frac{0.29260 - 0.29064}{0.29260 - 0.27278} = \frac{6.7322 - s_2}{6.7322 - 6.7071}$$
$$\frac{0.00196}{0.01982} = \frac{6.7322 - s_2}{0.0251}$$
$$0.0989 = \frac{6.7322 - s_2}{0.0251}$$
$$s_2 = 6.7322 - (0.0989 \times 0.0251)$$
$$s_2 = 6.7322 - 0.00248$$
$$s_2 = 6.7322 - 0.00248$$
$$s_2 = 6.72972\,kJ/kg°K$$

Step 8 of 8

The change in entropy is given by

$$\Delta s = m \times (s_2 - s_1)$$
$$= 5 \times (6.72972 - 2.8811)$$
$$= 5 \times 3.84862$$
$$\boxed{\Delta s = 19.2431\,kJ/K}$$

Therefore change in entropy water mixture is 19.2431 kJ/K .

Question: 8.033P

A rigid tank is divided into two equal parts by a partition. One part of the tank contains 2.5 kg of compressed liquid water at 400 kPa and 60°C while the other part is evacuated. The partition is now removed, and the water expands to fill the entire tank. Determine the entropy change of water during this process, if the final pressure in the tank is 40 kPa.

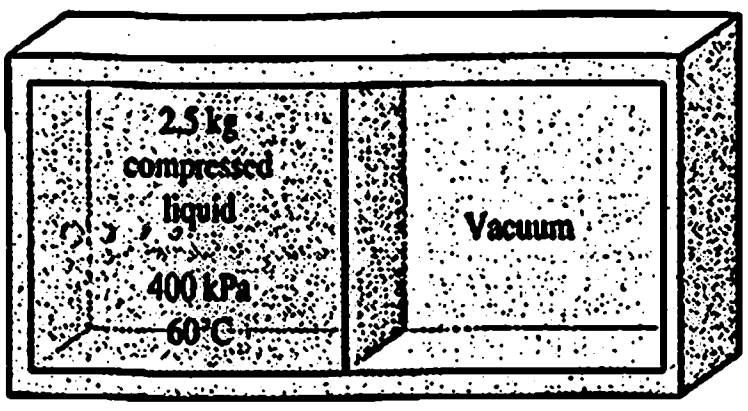


FIGURE P8–33

Step 1 of 6

Given data:

Mass of water in the tank is, $m = 2.5$ kg

Initial pressure of liquid is, $P_1 = 400$ kPa

Initial temperature of liquid is, $T_1 = 60\ ^\circ\text{C} = 333$ K

Final pressure of liquid is, $P_2 = 40$ kPa

Step 2 of 6

As initial temperature of liquid is lesser than the saturated temperature at pressure $P_1 = 400$ kPa,

We take the properties at saturated liquid at temperature $T_1 = 60\ ^\circ\text{C}$

From the table of 'saturated water –temperature table',

At temperature $T_1 = 60\ ^\circ\text{C}$ and saturated liquid condition,

Properties of water are

$v_1 = v_f = 0.001017\ \text{m}^3/\text{kg}$

$s_1 = s_f = 0.8313\ \text{kJ/kg°K}$

From the table of 'saturated water –pressure table',

At pressure $P_2 = 40$ kPa, properties of water are

$v_{f2} = 0.001026\ \text{m}^3/\text{kg}$

$v_{fg2} = 3.9933\ \text{m}^3/\text{kg}$

$s_{f2} = 1.0261\ \text{kJ/kg°K}$

$s_{fg2} = 6.6430\ \text{kJ/kg°K}$

Step 3 of 6

As both the parts of the tank have equal volume, after expansion volume will be doubled.

Final pressure of adiabatic process is given by

$$v_2 = 2 \times v_1$$
$$= 2 \times 0.001017$$
$$= 0.002034\ \text{m}^3/\text{kg}$$

Step 4 of 6

Final volume is also given by

$$v_2 = v_{f2} + x_2 v_{fg2}$$
$$0.002034 = 0.001026 + x_2 \times 3.9933$$
$$x_2 = \frac{0.002034 - 0.001026}{3.9933}$$
$$= \frac{0.001008}{3.9933}$$
$$= 0.0002524$$

Step 5 of 6

Final entropy is given by

$$s_2 = s_{f2} + x_2 \times s_{fg2}$$
$$= 1.0261 + 0.0002524 \times 6.6430$$
$$= 1.0261 + 0.00168$$
$$= 1.02778\ \text{kJ/kg°K}$$

Step 6 of 6

Entropy change in the process is given by

$$\Delta S = m \times (s_2 - s_1)$$
$$= 2.5 \times (1.02778 - 0.8313)$$
$$= 2.5 \times 0.19648$$
$$= 0.4912\ \text{kJ/°K}$$

$\boxed{\Delta S = 0.4912\ \text{kJ/°K}}$

Therefore the entropy change in the process is 0.4912 kJ/°K.

Question: 8.034P

Reconsider Prob. 8-33. Using EES (or other) software, evaluate and plot the entropy generated as a function of surrounding temperature, and determine the values of the surrounding temperatures that are valid for this problem. Let the surrounding temperature vary from 0 to 100°C. Discuss your results.

Step 1 of 8

Given data:

Mass of water in the tank is, $m = 2.5$ kg

Initial pressure of liquid is, $P_1 = 400$ kPa

Initial temperature of liquid is, $T_1 = 60\ ^\circ\text{C} = 333$ K

Final pressure of liquid is, $P_2 = 40$ kPa

Step 2 of 8

Give all the inputs in the EES software as given below

T1 =60 [C]

P1 =400 [kPa]

m = 2.5 [kg]

P2 =40 [kPa]

Step 3 of 8

For calculating other parameters we should give other equations as below

V1 =m *sv1

sv1 =volume(steam_iapws,P=P1,T=T1)

u1 =intenergy(steam_iapws,P=P1,T=T1)

s1 =entropy(steam_iapws,P=P1,T=T1)

V2 =2*V1

sv2 =V2/ m

u2 =intenergy(steam_iapws,P=P2,v=sv2)

s2 =entropy(steam_iapws,v=sv2,P=P2)

For energy balance we give

Ein-Eout =Ediff

Ediff =m *(u2-u1)

Ein =0

Eout =Qout

For entropy generation we give

Sgen =Ssys+Qout/(T0+273)

Ssys =m*(s2-s1)

Step 4 of 8

After writing these sets of equations, we should create a parametric table from

Tables → new parametric

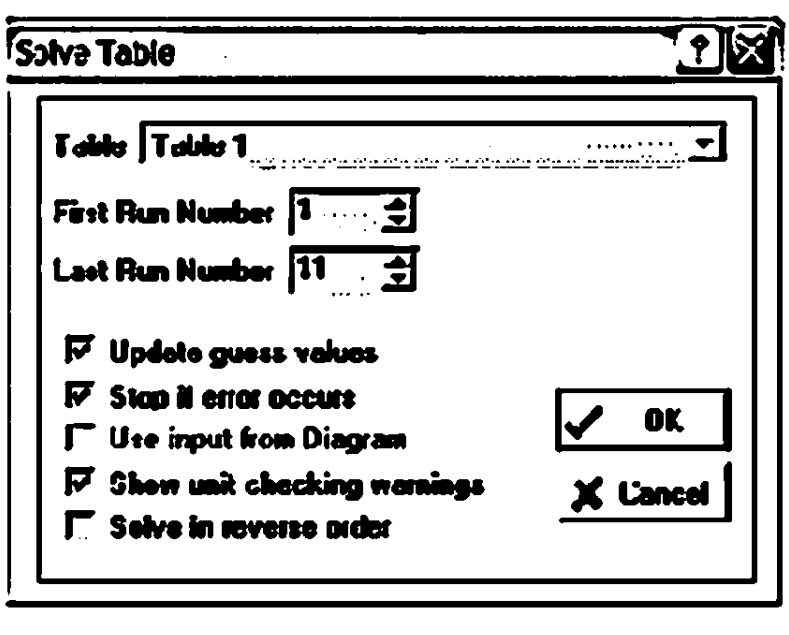
Fill all the values of T0.

Table 1

	1 T0 [C]	2 Sgen [kJ/K]
1..11		
Run 1	0	
Run 2	10	
Run 3	20	
Run 4	30	
Run 5	40	
Run 6	50	
Run 7	60	
Run 8	70	
Run 9	80	
Run 10	90	
Run 11	100	

Step 5 of 8

Now solve the table for respective Sgen values by pressing F3.



Step 6 of 8

Then we get

Table 1

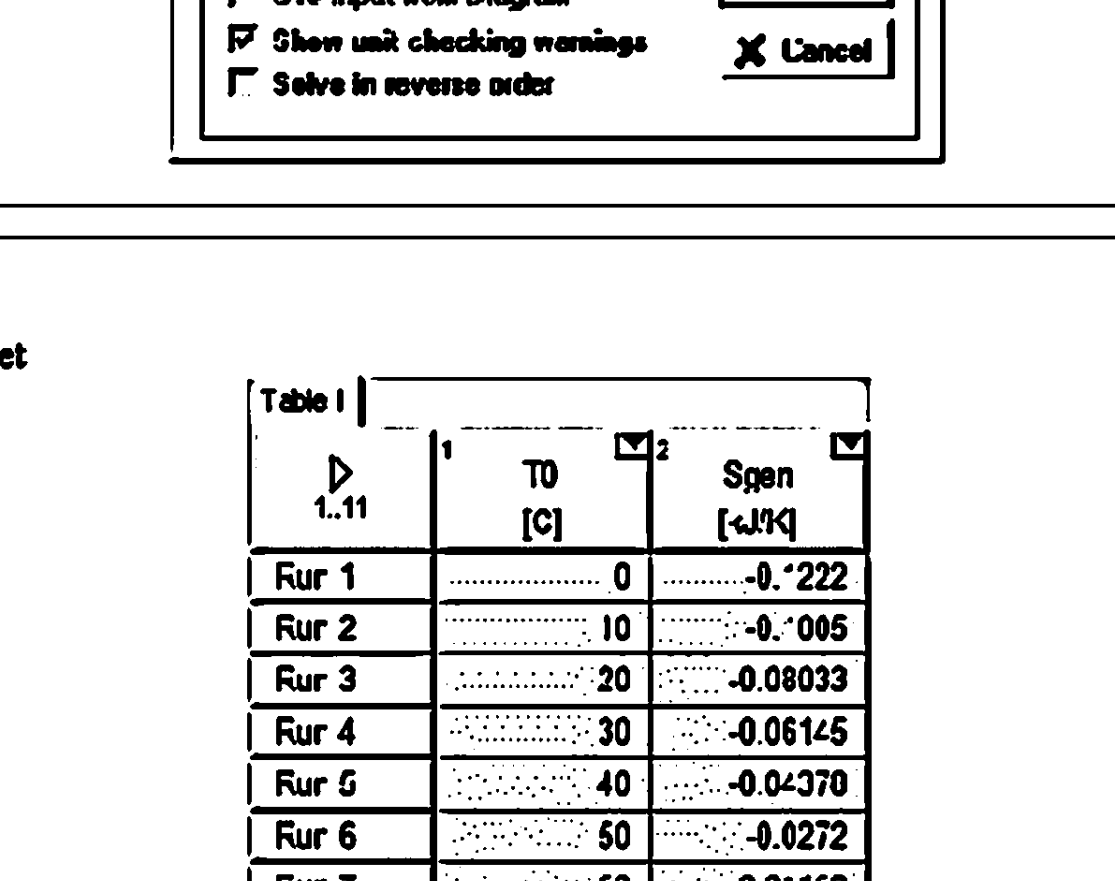
	1 T0 [C]	2 Sgen [kJ/K]
1..11		
Run 1	0	-0.0222
Run 2	10	-0.0105
Run 3	20	-0.08033
Run 4	30	-0.06145
Run 5	40	-0.04370
Run 6	50	-0.0272
Run 7	60	-0.0162
Run 8	70	0.003079
Run 9	80	0.01689
Run 10	90	0.02997
Run 11	100	0.04235

Step 7 of 8

For getting the plot we should go for

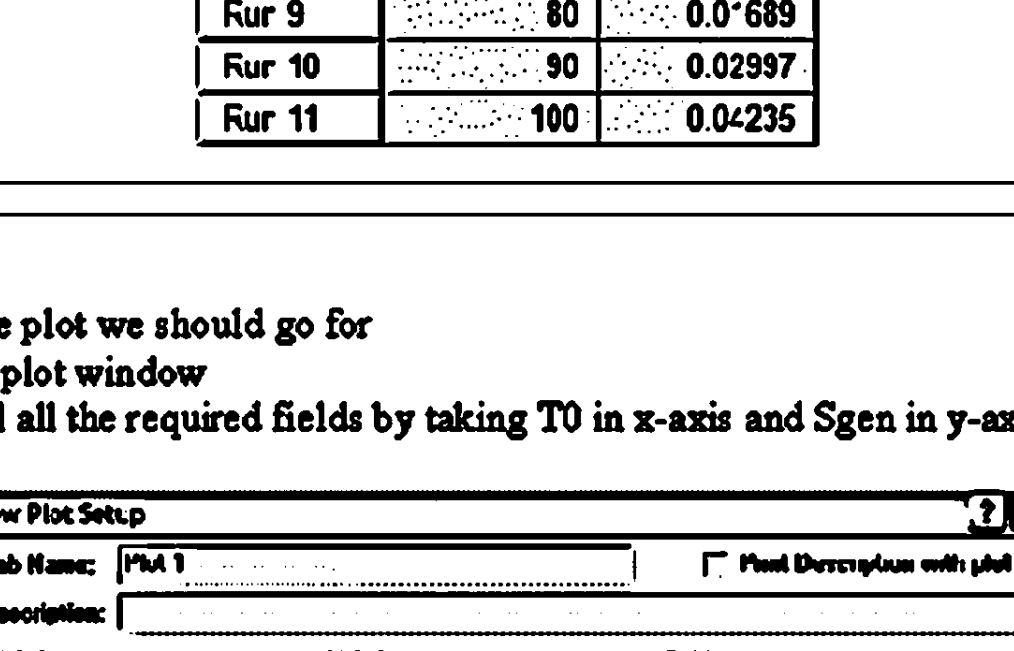
Plots → New plot window

We should fill all the required fields by taking T0 in x-axis and Sgen in y-axis.



Step 8 of 8

After that we get the plot as



Question: 8.035P

A piston-cylinder device contains 2 lbm of refrigerant-134a at 120 psia and 100°F. The refrigerant is now cooled at constant pressure until it exists as a liquid at 50°F. Determine the entropy change of the refrigerant during this process.

Step 1 of 4

Given: For R-134a,

Mass, $m = 2$ lbm

Initial pressure, $P_1 = 120$ psia

Initial temperature, $T_1 = 100°\text{F}$

Final temperature, $T_2 = 50°\text{F}$

The refrigerant is cooled at constant pressure.

Step 2 of 4

Super heated refrigerant -134aTables,

At $T_1 = 100°\text{F}$, $P_1 = 120$ psia

Initial Entropy, $s_1 = 0.22362 \dfrac{\text{Btu}}{\text{lbm.R}}$

Step 3 of 4

saturated refrigerant -134a temperature Tables

At $T_1 = 50°\text{F}$, satuarated liquid

Final Entropy, $s_2 = 0.06039 \dfrac{\text{Btu}}{\text{lbm.R}}$

Step 4 of 4

Entropy change,

$$\Delta S = m \times (s_2 - s_1)$$

$$\Rightarrow \Delta S = 2 \times (0.06039 - 0.22362)$$

$$\Rightarrow \boxed{\Delta S = -0.32646 \text{ Btu/R}}$$

Question: 8.036P

An insulated piston-cylinder device contains 5 L of saturated liquid water at a constant pressure of 150 kPa. An electric resistance heater inside the cylinder is now turned on, and 2200 kJ of energy is transferred to the steam. Determine the entropy change of the water during this process.

Step 1 of 6

Given data

Energy transferred to the steam $E_{in} = 2200\text{kJ}$

Pressure of Saturated liquid water $P_1 = 150\text{kPa}$

Volume of saturated liquid water $V = 5\,\text{L}$

$$= 5\times10^{-3}\text{m}^3$$

Step 2 of 6

From the saturated water-Pressure

At $P_1 = 150\text{kPa}$

Specific volume $v_1 = v_f$

$$= 0.001053\text{m}^3/\text{kg}$$

Enthalpy $h_1 = h_f$

$$= 467.13\text{kJ/kg}$$

Entropy $s_1 = s_f$

$$= 1.4337\text{kJ/kg}\cdot\text{K}$$

Mass of sat liquid

$$m = \frac{V}{v_1}$$

$$m = \frac{0.005\text{m}^3}{0.001053\text{m}^3/\text{kg}}$$

$$= 4.75\text{kg}$$

Step 3 of 6

From first law of thermodynamics, energy balance for this closed system at constant pressure

$$E_{in} + mh_1 = mh_2$$

$$h_2 = h_1 + \frac{E_{in}}{m}$$

$$= 467.13\left(\frac{\text{kJ}}{\text{kg}}\right) + \frac{2200\text{kJ}}{4.75\text{kg}}$$

$$= 930.29\text{kJ/kg}$$

Step 4 of 6

From the saturated water-Pressure

At $P_1 = 150\text{kPa}$

Enthalpy of evaporation $h_{fg} = 2226\text{kJ/kg}$

Enthalpy of evaporation $s_{fg} = 5.7894\text{kJ/kg}\cdot\text{K}$

We know that

$$h_2 = h_f + x_2\left(h_{fg}\right)$$

Quality at stage 2

$$x_2 = \frac{h_2 - hf}{h_{fg}}$$

$$x_2 = \frac{(930.29 - 467.13)}{2226}$$

$$= 0.208$$

Step 5 of 6

Entropy at state (2)

$$s_2 = s_f + x_2 s_{fg}$$

$$= 1.4337 + (0.208)(5.7894)$$

$$= 2.6382\text{kJ/kg}$$

Step 6 of 6

Entropy change of water during this process

$$\Delta s = m(s_2 - s_1)$$

$$= (4.75\text{kg})(2.6382 - 1.4337)\text{kJ/kg}\cdot\text{K}$$

$$= 5.72\text{kJ/K}$$

$$\boxed{\Delta s = 5.72\text{kJ/K}}$$

Question: 8.037P

Calculate the change in the specific entropy of water as it is cooled at a constant pressure of 300 kPa from a saturated vapor to asaturated liquid by using aGibbs equation ($Tds = dh - vdP$) . Use the steam tables to verify your result.

Step 1 of 2

Obtain the properties of water at pressure $P = 300$ kPa from saturated water tables:

Saturation temperature, $T = T_{sat} = 133.52°C = 406.67$ K

Specific enthalpy of saturated liquid, $h_f = 561.43$ kJ/kg

Specific enthalpy of saturated vapor, $h_g = 2724.9$kJ/kg

Write the Gibbs equation.

$$T \times ds = dh - v \times dP$$

$$ds = \frac{1}{T}(dh - v \times dP)$$

Here, T is temperature, dP is change in pressure, dh is change in enthalpy, and ds is change in entropy.

As pressure is constant the change in pressure is zero.

$$dP = 0$$

Substitute this value in Gibb's equation.

$$ds = \frac{1}{T}(dh - v \times 0)$$

$$= \frac{1}{T}dh$$

$$= \frac{1}{T} \times (h_2 - h_1)$$

Here, h_2 is enthalpy of saturated vapor and h_1 is specific enthalpy of saturated liquid.

Substitute the values and calculate ds.

$$ds = \frac{1}{T} \times (h_g - h_f)$$

$$= \frac{1}{406.7} \times (2724.9 - 561.43)$$

$$= 5.319 \text{kJ/kg} \cdot \text{K}$$

Therefore, the change in entropy by using Gibb's equation is $\boxed{5.319 \text{ kJ/kg} \cdot \text{K}}$.

Step 2 of 2

Obtain the properties of water at pressure $P = 300$ kPa from saturated water tables.

Specific entropy of saturated liquid, $s_f = 1.6717 \text{kJ/kg} \cdot \text{K}$

As pressure is constant the change in pressure is zero.

$$dP = 0$$

Substitute this value in Gibb's equation.

$$ds = \frac{1}{T}(dh - v \times 0)$$

$$= \frac{1}{T}dh$$

$$= \frac{1}{T} \times (h_2 - h_1)$$

Here, h_2 is enthalpy of saturated vapor and h_1 is specific enthalpy of saturated liquid.

Substitute the values and calculate ds.

$$ds = \frac{1}{T} \times (h_g - h_f)$$

Question: 8.038P

Saturated R-134a vapor enters a compressor at 0°F. At compressor exit, the specific entropy is the same as that at the inlet, and the pressure is 60 psia. Determine the R-l34a exit temperature and the change in the enthalpy of R-134a.

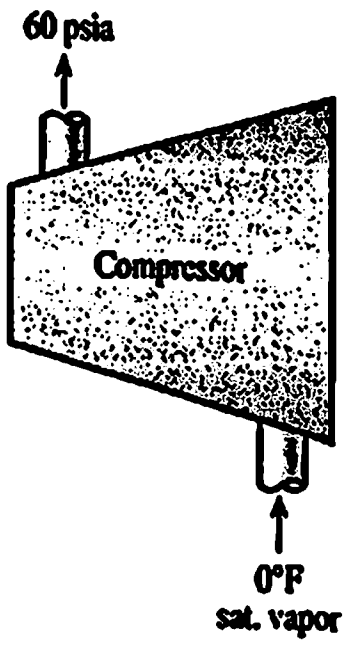


FIGURE P8–38E

Step 1 of 6

Given data:

Temperature at compressor inlet is, $T_1 = 0\ ^\circ F$.

Pressure at compressor exit is, $P_2 = 60$ psia.

Step 2 of 6

From the Saturated refrigerant 134a-Temperature table

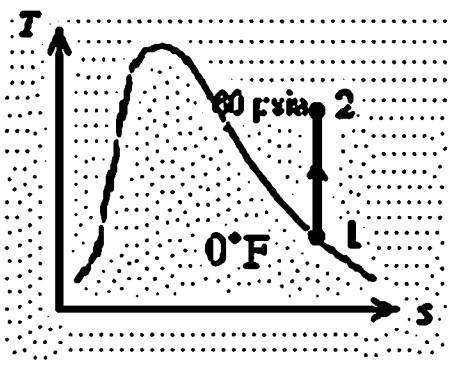
At temperature $T_1 = 0\ ^\circ F$ and saturated vapor state,

$h_1 = h_g = 103.08$ Btu/lbm

$s_1 = s_g = 0.22539$ Btu/lbm · R

Step 3 of 6

Let us draw the T-s diagram:



Step 4 of 6

Given the final state is superheated vapor and the final specific entropy of steam is constant. So,

$s_2 = s_1$

Therefore, $s_2 = 0.22539$ Btu/lbm.R

From the superheated refrigerant 134a-Temperature table

At pressure $P_2 = 60$ psia and specific entropy $s_2 = 0.22539$ Btu/lbm.R ,

The temperature is interpolated as

$$\frac{s_2 - 0.22127}{0.22570 - 0.22127} = \frac{T_2 - 49.84}{60 - 49.84}$$

$$\frac{0.22539 - 0.22127}{0.22570 - 0.22127} = \frac{T_2 - 49.84}{60 - 49.84}$$

$$\frac{0.00412}{0.00443} = \frac{T_2 - 49.84}{10.16}$$

$$0.93 = \frac{T_2 - 49.84}{10.16}$$

$T_2 = (10.16 \times 0.93) + 49.84$

$T_2 = 9.4488 + 49.84$

Therefore the exit temperature is $\boxed{T_2 = 59.2888\ ^\circ F}$

Step 5 of 6

From the superheated refrigerant 134a-Temperature table,

At pressure $P_2 = 60$ psia and specific entropy $s_2 = 0.22539$ Btu/lbm.R ,

The enthalpy is interpolated as

$$\frac{s_2 - 0.22127}{0.22570 - 0.22127} = \frac{h_2 - 110.11}{112.39 - 110.11}$$

$$\frac{0.22539 - 0.22127}{0.22570 - 0.22127} = \frac{h_2 - 110.11}{112.39 - 110.11}$$

$$\frac{0.00412}{0.00443} = \frac{h_2 - 110.11}{2.28}$$

$$0.93 = \frac{h_2 - 110.11}{2.28}$$

$h_2 = (2.28 \times 0.93) + 110.11$

$h_2 = 2.1204 + 110.11$

$h_2 = 112.2304$ Btu/lbm

Step 6 of 6

The change in enthalpy is given by

$\Delta h = h_2 - h_1$

$= 112.2304 - 103.08$

$= 9.1504$ Btu/lbm

Therefore exit temperature is $\boxed{59.2888\ ^\circ F}$ and change in enthalpy is $\boxed{\Delta h = 9.1504\ \text{Btu/lbm}}$.

Question: 8.039P

Water vapor enters a turbine at 6 MPa and 400°C, and leaves the turbine at 100 kPa with the same specific entropy as that at the inlet. Calculate the difference between the specific enthalpy of the water at the turbine inlet and exit.

Step 1 of 5

Given

Inlet pressure of turbine is, $P_1 = 6\ \text{MPa} = 6000\ \text{kPa}$

Inlet temperature of turbine is, $T_1 = 400\ ^\circ\text{C} = 673\ \text{K}$

Exit pressure of turbine is, $P_2 = 100\ \text{kPa}$

Specific entropy at inlet and exit of turbine is same.

Step 2 of 5

From superheated water tables at pressure $P_1 = 6\ \text{MPa}$ and temperature $T_1 = 400\ ^\circ\text{C}$ the enthalpy and entropy values respectively are

$h_1 = 3178.3\ \text{kJ/kg}$ And

$s_1 = 6.5432\ \text{kJ/kg.K}$

From saturated water tables at pressure $P_2 = 100\ \text{kPa}$ enthalpy and entropy of saturated liquid and enthalpy and entropy of vaporization respectively are

$h_{f2} = 417.51\ \text{kJ/kg}$

$h_{fg2} = 2257.5\ \text{kJ/kg}$

$s_{f2} = 1.3028\ \text{kJ/kg.K}$

$s_{fg2} = 6.0562\ \text{kJ/kg.K}$

Step 3 of 5

As it is given that specific entropy at inlet and exit of turbine is same we can write it as

$s_1 = s_2$

$s_1 = s_{f2} + x_2 \times s_{fg2}$

$x_2 \times s_{fg2} = s_1 - s_{f2}$

$$x_2 = \frac{s_1 - s_{f2}}{s_{fg2}}$$

$$x_2 = \frac{6.5432 - 1.3028}{6.0562}$$

$$x_2 = \frac{5.2404}{6.0562}$$

$x_2 = 0.8653$

Step 4 of 5

The enthalpy at turbine exit is given by

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$h_2 = 417.51 + 0.8653 \times 2257.5$

$h_2 = 417.51 + 1953.415$

$h_2 = 2370.925\ \text{kJ/kg}$

Step 5 of 5

The difference in specific enthalpies of water at turbine inlet and exit is given by

$dh = h_1 - h_2$

$dh = 3178.3 - 2370.925$

$\boxed{dh = 807.375\ \text{kJ/kg}}$

Therefore the difference in specific enthalpies of water at turbine inlet and exit is

$dh = 807.375\ \text{kJ/kg}$

Question: 8.040P

1-kg of R-134a initially at 600 kPa and 25°C undergoes a process during which the entropy is kept constant until the pressure drops to 100 kPa. Determine the final temperature of the R-134a and the final specific internal energy.

Step 1 of 3

Obtain the entropy at state 1 from superheated refrigerant-134a tables at initial pressure $P_1 = 600\ \text{kPa}$ and initial temperature $T_1 = 25°\text{C}$ by using interpolation.

$$\frac{s_1 - 0.9218}{0.9499 - 0.9218} = \frac{25 - 21.55}{30 - 21.55}$$

$$\frac{s_1 - 0.9218}{0.0281} = \frac{3.45}{8.45}$$

$$\frac{s_1 - 0.9218}{0.0281} = 0.4083$$

$$s_1 = (0.4083 \times 0.0281) + 0.9218$$

$$= 0.01147 + 0.9218$$

$$= 0.93327\ \text{kJ/kg·K}$$

Step 2 of 3

Entropy remains constant during the process 1-2. So, the final entropy is equal to the initial entropy.

$$s_2 = s_1$$

$$s_2 = 0.93327\ \text{kJ/kg·K}$$

At the final pressure , $P_2 = 100\ \text{kPa}$, from saturated R-134a tables, the following values are obtained:

Specific entropy of saturated liquid $s_{f2} = 0.07188\ \text{kJ/kg·K}$

Specific entropy of liquid vapor mixture $s_{fg2} = 0.87995\ \text{kJ/kg·K}$

Specific entropy of saturated vapor $s_{g2} = 0.95183\ \text{kJ/kg·K}$

Specific internal energy of saturated liquid $u_{f2} = 17.21\ \text{kJ/kg}$

Specific internal energy of liquid vapor mixture $u_{fg2} = 197.98\ \text{kJ/kg}$

Saturation temperature $T_{sat} = -26.37°\text{C}$

The final entropy is in between s_f and s_g.

$$s_f < s_2 < s_g$$

Therefore, the final state is liquid-vapor mixture.

Then the final temperature will be same as the saturated temperature.

$$T_2 = T_{sat@100\text{kPa}} = -26.37°\text{C}$$

Therefore, the final temperature of R-134a is $\boxed{-26.37°\text{C}}$.

Step 3 of 3

Find the dryness fraction at final state 2 by using the entropy expression.

$$s_2 = s_{f2} + x_2 \times s_{fg2}$$

$$x_2 = \frac{s_2 - s_{f2}}{s_{fg2}}$$

$$= \frac{0.93327 - 0.07188}{0.87995}$$

$$= \frac{0.86139}{0.87995}$$

$$= 0.9789$$

Find the final internal energy by using the following expression:

$$u_2 = u_{f2} + x_2 \times u_{fg2}$$

$$= 17.21 + (0.9789 \times 197.98)$$

$$= 17.21 + 193.803$$

$$= 211.013\ \text{kJ/kg}$$

Therefore, the final internal energy u_2 is $\boxed{211.013\ \text{kJ/kg}}$.

Question: 8.041P

Refrigerant-134a is expanded isentropically from 800 kPa and 60°C at the inlet of a steady-flow turbine to 100 kPa at the outlet. The outlet area is 1 m2 , and the inlet area is 0.5 m2 . Calculate the inlet and outlet velocities when the mass flow rate is 0.5 kg/s.

Step 1 of 6

Given data:

Mass of R-134a used is, $m = 1\ kg$

Inlet pressure is, $P_1 = 800\ kPa$

Inlet temperature is, $T_1 = 60\ °C = 333\ K$

Exit Pressure is, $P_2 = 100\ kPa$

Outlet area is, $A_2 = 1\ m^2$

Inlet area is, $A_1 = 0.5\ m^2$

Mass flow rate is, $m = 0.5\ kg/s$

Step 2 of 6

From the superheated refrigerant-134a

At pressure $P_1 = 800\ kPa$ and temperature $T_1 = 60\ °C$,

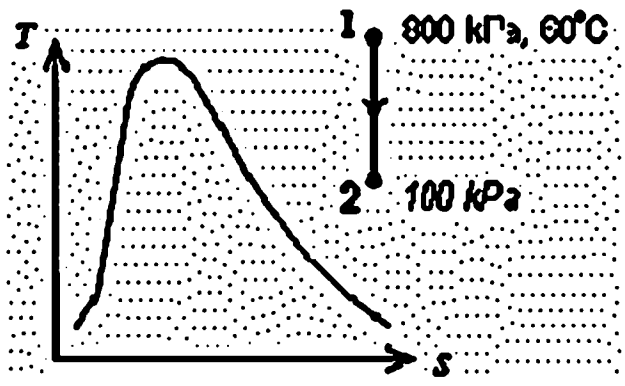
Specific volume and entropy of refrigerant respectively are

$v_1 = 0.029973\ m^3/kg$

$s_1 = 1.011\ kJ/kg \cdot K$

Step 3 of 6

Let us draw the T-s Diagram:



Step 4 of 6

Given entropy is constant. So, the final entropy is given by

$s_2 = s_1$

Therefore, $s_2 = 1.011\ kkJ/kg \cdot K$

From the superheated refrigerant-134a

At pressure $P_2 = 100\ kPa$ and entropy $s_2 = 1.011\ kJ/kg°K$,

The specific volume is interpolated as

$$\frac{v_2 - 0.20743}{0.2163 - 0.20743} = \frac{1.011 - 1.003}{1.0332 - 1.003}$$

$$\frac{v_2 - 0.20743}{0.00887} = \frac{0.008}{0.0302}$$

$$\frac{v_2 - 0.20743}{0.00887} = 0.2649$$

$$v_2 = 0.20743 + (0.2649 \times 0.00887)$$

$$= 0.20743 + 0.00235$$

$$= 0.20978\ m^3/kg$$

Step 5 of 6

The inlet velocity is given by

$$c_1 = \frac{m \times v_1}{A_1}$$

$$= \frac{0.5 \times 0.029973}{0.5}$$

$$= \frac{0.0149865}{0.5}$$

$$= 0.029973\ m/s$$

$\boxed{c_1 = 0.029973\ m/s}$

Step 6 of 6

The exit velocity is given by

$$c_2 = \frac{m \times v_2}{A_2}$$

$$= \frac{0.5 \times 0.20978}{1.0}$$

$$= \frac{0.10489}{1.0}$$

$$= 0.10489\ m/s$$

$\boxed{c_2 = 0.10489\ m/s}$

Therefore inlet velocity is $\boxed{c_1 = 0.029973\ m/s}$ and exit velocity is $\boxed{c_2 = 0.10489\ m/s}$.

Question: 8.042P

A heavily insulated piston-cylinder device contains 0.02 m3 of steam at 300 kPa and 200°C. Steam is now compressed in a reversible manner to a pressure of 1.2 MPa. Determine the work done on the steam during this process.

Step 1 of 5

Given data:

Volume of steam is, $v = 0.02\ m^3$.

Initial pressure is, $P_1 = 300\ kPa$.

Initial temperature is, $T_1 = 200\ ^\circ C = 473\ K$.

Final pressure is, $P_2 = 1.2\ MPa = 1200\ kPa$.

Step 2 of 5

From the superheated water,

At pressure $P_1 = 300\ kPa$ and temperature $T_1 = 200\ ^\circ C$,

Specific volume, internal energy and entropy of steam respectively are

$v_1 = 0.71643\ m^3/kg$

$u_1 = 2651\ kJ/kg$

$s_1 = 7.3132\ kJ/kg \cdot K$

Step 3 of 5

As the cylinder is insulated, the process is isentropic. So, the final entropy is given by

$s_2 = s_1$

Therefore, $s_2 = 7.3132\ kJ/kg \cdot K$

From the superheated water

At pressure $P_2 = 1.2\ MPa$ and entropy $s_2 = 7.3132\ kJ/kg \cdot K$,

The internal energy is interpolated as

$$\frac{u_2 - 2872.7}{2955.5 - 2872.7} = \frac{7.3132 - 7.2139}{7.3793 - 7.2139}$$

$$\frac{u_2 - 2872.7}{82.8} = \frac{0.0993}{0.1654}$$

$$u_2 = 2872.7 + (0.6004 \times 82.8)$$

$$= 2872.7 + 49.7131$$

$$= 2922.4131\ kJ/kg$$

Step 4 of 5

The mass of steam is given by

$$m = \frac{v}{v_1}$$

$$= \frac{0.02}{0.71643}$$

$$= 0.0279\ kg$$

Step 5 of 5

As the cylinder is closed, we will consider internal energy for calculating the work done.

The work done is given by

$$W = m \times (u_2 - u_1)$$

$$= 0.0279 \times (2922.4131 - 2651)$$

$$= 0.0279 \times 271.4131$$

$$= 7.5724\ kJ$$

$\boxed{W = 7.5724\ kJ}$

Therefore the work done on the steam is $\boxed{W = 7.5724\ kJ}$.

Question: 8.043P

Reconsider Prob. 8-42. Using EES (or other) software, evaluate and plot the work done on the steam as a function of final pressure as the pressure varies from 300 kPato 1.2 MPa.

Step 1 of 8

Given data:

Volume of steam is, $v = 0.02\ m^3$

Initial pressure is, $P_1 = 300\ kPa$

Initial temperature is, $T_1 = 200\ °C = 473\ K$

Final pressure varies, $P_2 = 300$ to $1200\ kPa$

Step 2 of 8

Give all the inputs in the EES software as given below

T1 =200 [C]

P1 =300 [kPa]

Vsys =0.02 [m^3]

Step 3 of 8

For calculating other parameters we should give other equations as

u1 =intenergy(steam_iapws,P=P1,T=T1)

v1 =volume(steam_iapws,P=P1,T=T1)

s1 =entropy(steam_iapws,P=P1,T=T1)

Vsys=m_sys*v1

s2 =s1

u2 =intenergy(steam_iapws,P=P2,s=s2)

Ein − Eout = Esys

Esys =m_sys*(u2−u1)

Ein =Win

Eout =0

Step 4 of 8

After writing these sets of equations, we should create a parametric table from

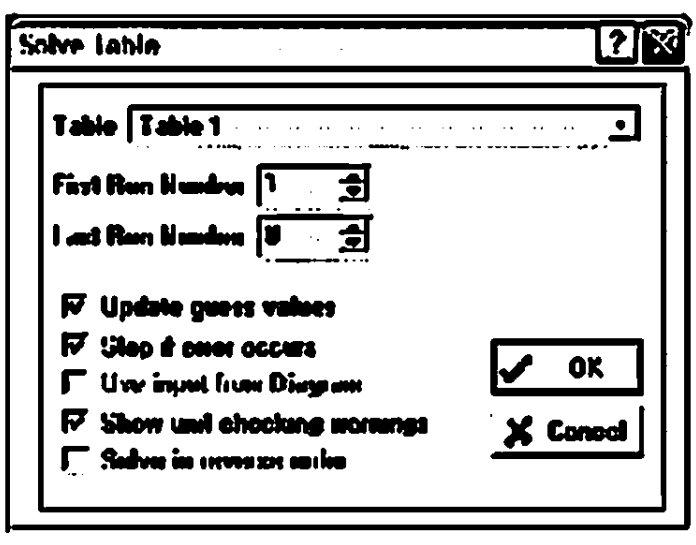
Tables → new parametric

Fill all the values of P2.

	1	2
	P2 [kPa]	Win [kJ]
Run 1	300	
Run 2	400	
Run 3	500	
Run 4	600	
Run 5	700	
Run 6	800	
Run 7	900	
Run 8	1000	

Step 5 of 8

Now solve the table for respective Win values by pressing F3.



Step 6 of 8

Then we get

Table 1

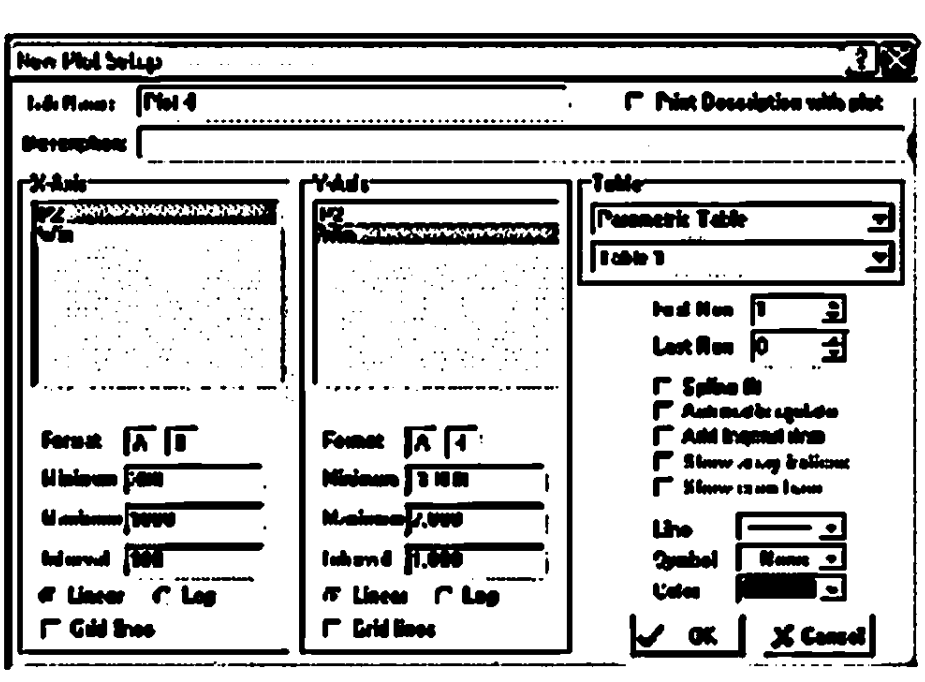
	1	2
	P2 [kPa]	Win [kJ]
Run 1	300	0
Run 2	400	1.363
Run 3	500	2.494
Run 4	600	3.462
Run 5	700	4.314
Run 6	800	5.073
Run 7	900	5.773
Run 8	1000	6.411

Step 7 of 8

For getting the plot we should go for

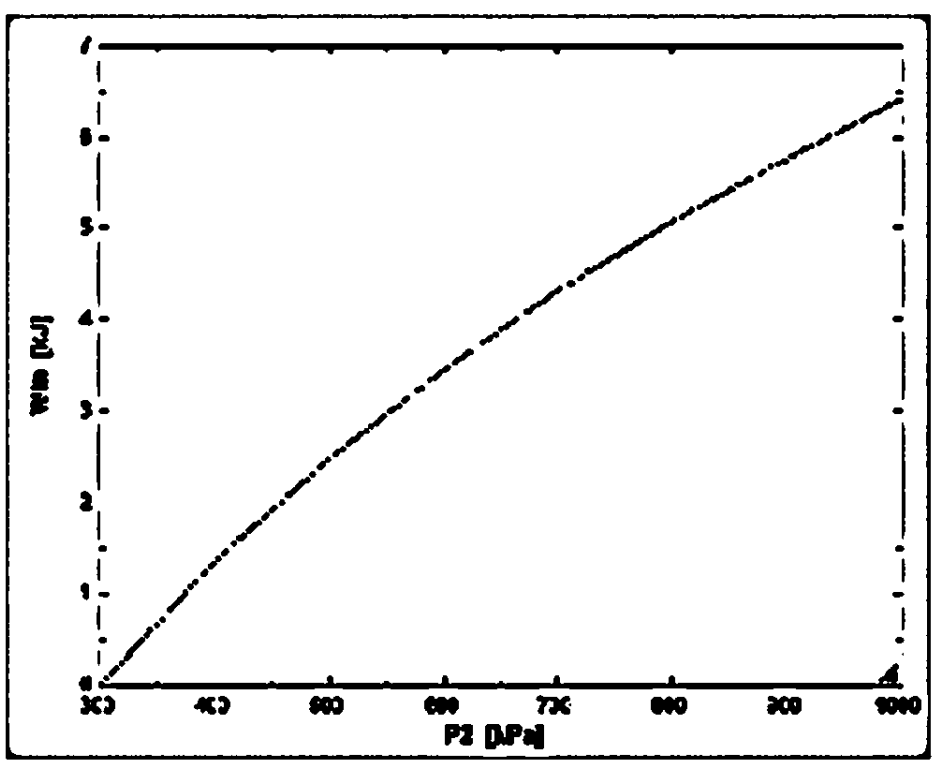
Plots → New plot window

We should fill all the required fields by taking P2 in x-axis and Win in y-axis.



Step 8 of 8

After that we get the plot as



Question: 8.044P

A piston-cylinder device contains 1.2 kg of saturated water vapor at 200°C. Heat is now transferred to steam, and steam expands reversibly and isothermally to a final pressure of 800 kPa. Determine the heat transferred and the work done during this process.

Step 1 of 4

Given data

Mass of saturate water vapor $m = 1.2\text{kg}$

Temperature of saturated vapor $T_1 = 200°\text{C}$

Final pressure $P = 800\text{kPa}$

Step 2 of 4

From the Saturated water –Temperature

At $T_1 = 200°\text{C}$

Saturated water vapor

Internal energy of saturated vapor $u_1 = u_g$

$= 2594.2\text{kJ/kg}$

Entropy of saturated vapor $s_1 = s_g$

$= 6.4302\text{kJ/kg}\cdot\text{K}$

Since process isothermally,

At $T_2 = 200°\text{C}$

$P_2 = 800\text{kPa}$

Internal energy of steam $u_2 = 2631.1\text{kJ/kg}$

Entropy of steam $s_2 = 6.8177\text{kJ/kg}\cdot\text{K}$

Step 3 of 4

Heat transfer for this reversible isothermal process

$$Q = T\Delta s$$
$$= Tm(s_2 - s_1)$$
$$= (200+273)(\text{K})(1.2(\text{kg}))(6.8177 - 6.4302)(\text{kJ/kg}\cdot\text{K})$$
$$= 219.94\text{kJ}$$

$$\boxed{Q = 219.94\text{kJ}}$$

Step 4 of 4

From energy balance for closed system

$$E_{in} - E_{out} = \Delta E_{system}$$
$$Q_{in} - w_{out} = \Delta U$$
$$w_{out} = Q_{in} - \Delta U$$
$$= Q_{in} - m(u_2 - u_1)$$

Work done during this process

$$w_{out} = (219.94) - (1.2)(\text{kg})(2631 - 2599)(\text{kJ/kg})$$
$$= 175.54\text{kJ}$$

$$\boxed{w_{out} = 175.54\text{kJ}}$$

Question: 8.045P

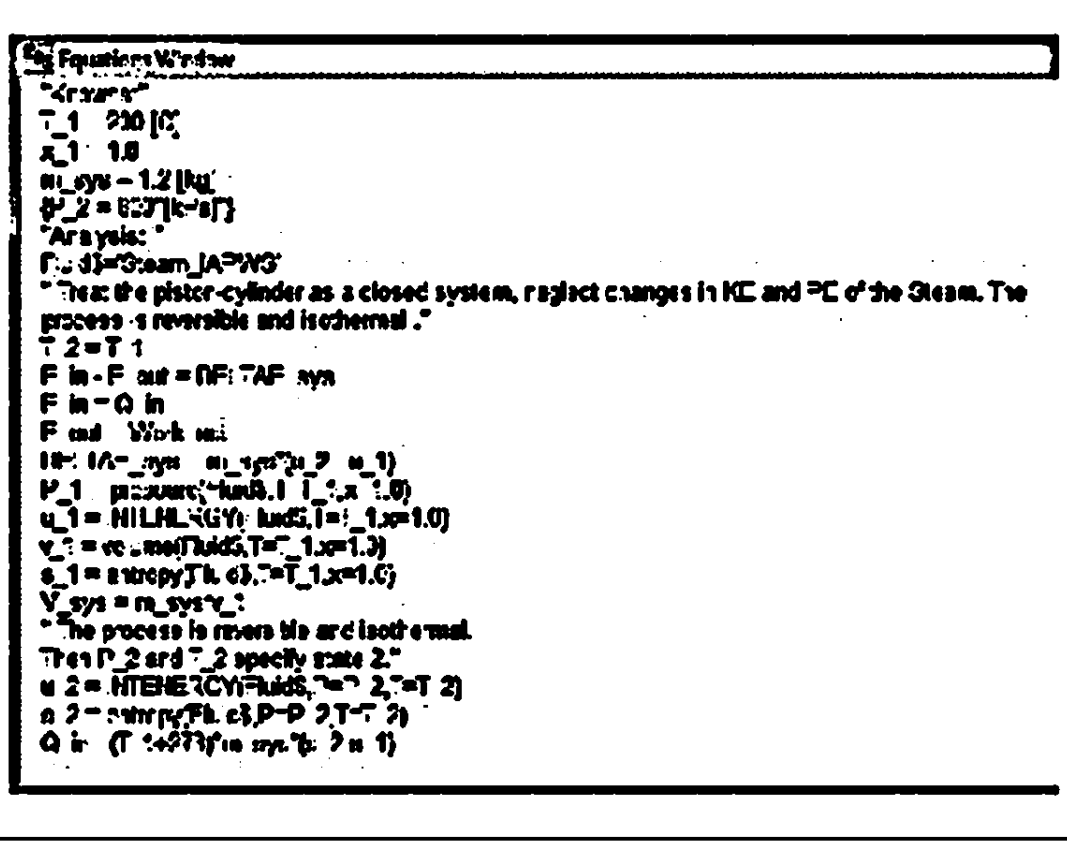
Reconsider Prob. 8-44. Using EES (or other) software, evaluate and plot the heat transferred to the steam and the work done as a function of final pressure as the pressure varies from the initial value to the final value of 800 kPa.

Step 1 of 6

The procedure for developing solution by using EES software:
Open the new equations window EES software and type all the information which is used to solve the problem
For developing table:
1) Click on table followed by new parametric table and add the variables, which are varying variable and required answer variable, then one table will form with that variables
2) Click on the variable which is varying and enter initial value and final value.
3) For getting the required answer variable values corresponding to the varying variable values click on the green color button and then the table will form with required answers
For developing graph:
1) Click on plot followed by new plot window followed by x-y plot.
2) Select the x-axis variable and y-axis variable and then press ok, then the graph will form with the given values.

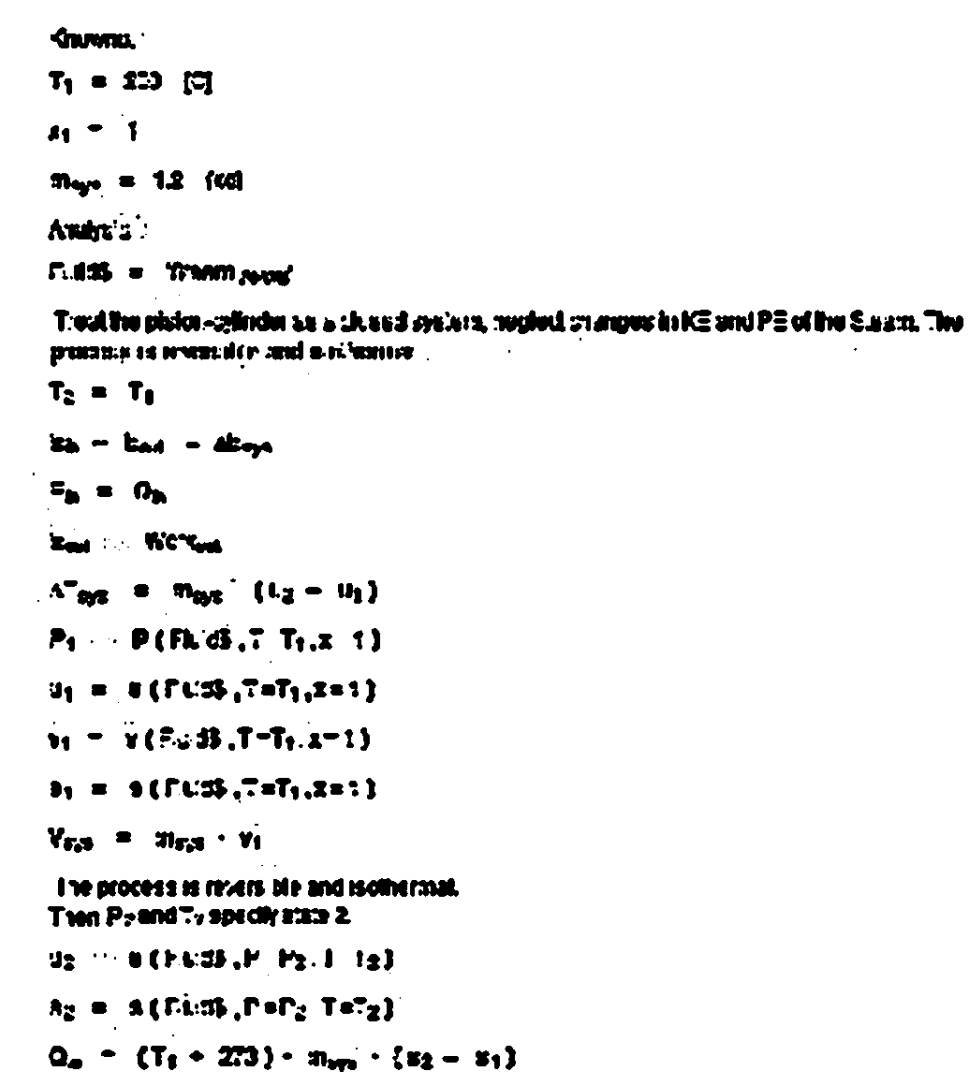
Step 2 of 6

The equations which are using in EES software for developing the solution are given in equations window is as shown below



Step 3 of 6

The EES software will convert the equations in equations window to equations in formatting window is as shown below



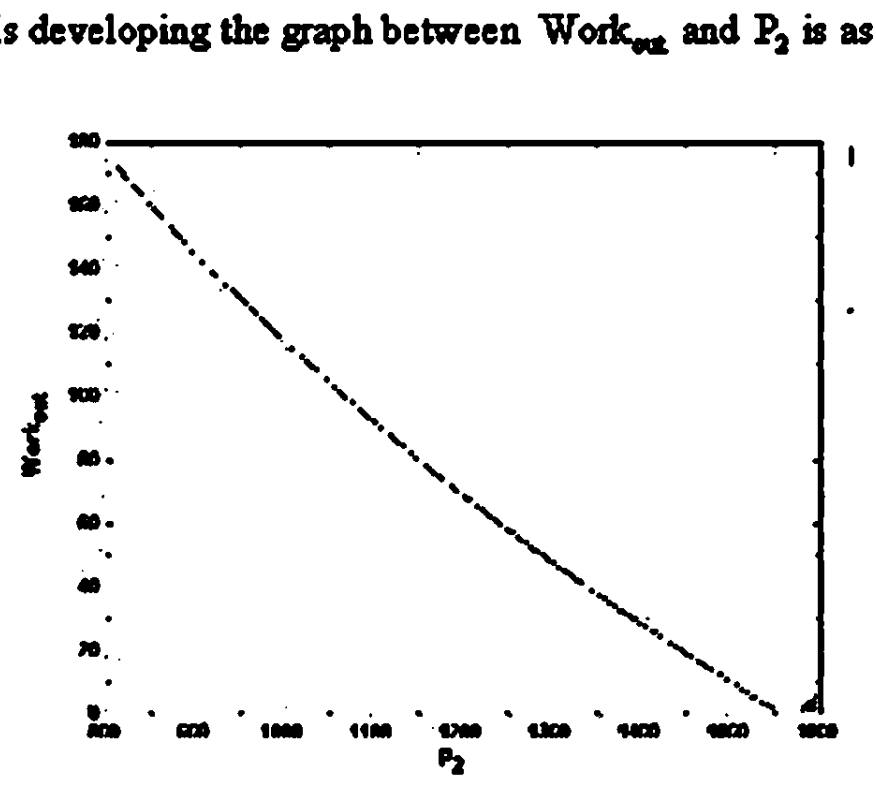
Step 4 of 6

The EES software is developing the table for Q_{in} and $Work_{out}$ corresponding to the P_2 values is as shown below

▷ 1..9	1 P_2	2 Q_{in}	3 $Work_{out}$
Run 1	800	219.9	175.7
Run 2	894.1	185.8	146.5
Run 3	988.3	154.4	120.1
Run 4	1082	125.2	96.11
Run 5	1177	97.83	74.08
Run 6	1271	71.91	53.71
Run 7	1365	47.2	34.75
Run 8	1459	23.45	17.02
Run 9	1553	0.4645	0.3319

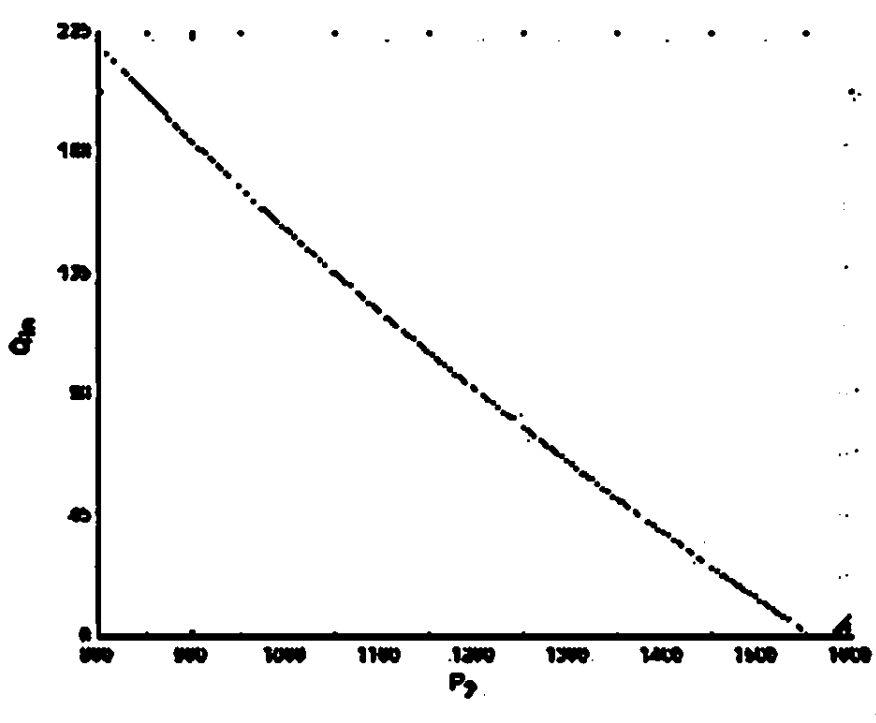
Step 5 of 6

The EES software is developing the graph between $Work_{out}$ and P_2 is as shown below



Step 6 of 6

The EES software is developing the graph between Q_{in} and P_2 is as shown below



Question: 8.046P

Refrigerant-134a at 240 kPa and 20°C undergoes an isothermal process in a closed system until its quality is 20 percent. On per unit mass basis, determine how much work and heat transfer are required.

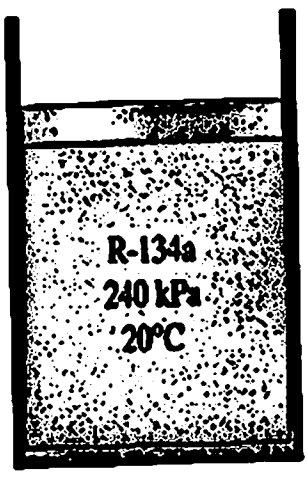


FIGURE P8–46

Step 1 of 5

Given data:

Inlet pressure of the refrigerant-143a, $P_1 = 240$ kPa

Inlet temperature of the refrigerant-143a,

$T_1 = 20°C$

$= 20 + 273$ K

$= 293$ K

Quality of refrigerant-143a at state 2,

$x_2 = 20\%$

$= 0.2$

Step 2 of 5

From the refrigerant-143a table,

At temperature, $T_1 = 20°C$ and pressure, $P_1 = 240$ kPa, we have

Internal energy at state 1, $u_1 = 246.74$ kJ/kg

Entropy at state 1, $s_1 = 1.0134$ kJ/kg·K

Since the process is an isothermal, so temperature remains constant, $T_1 = T_2 = 20°C$

From the refrigerant-143a table,

At temperature, $T_2 = 20°C$ and $x_2 = 0.2$, we have

Internal energy at saturated liquid, $u_f = 78.86$ kJ/kg

Internal energy during evaporation, $u_{fg} = 162.16$ kJ/kg

Entropy at saturated liquid, $s_f = 0.30063$ kJ/kg·K

Entropy during evaporation, $s_{fg} = 0.62172$ kJ/kg·K

Step 3 of 5

Internal energy at state 2,

$u_2 = u_f + x_2 u_{fg}$

$= 78.86 + (0.2 \times 162.16)$

$= 111.292$ kJ/kg

Entropy at state 2,

$s_2 = s_f + x_2 s_{fg}$

$= 0.30063 + (0.2 \times 0.62172)$

$= 0.4249$ kJ/kg·K

Step 4 of 5

Heat transferred into the system, we have

$Q_{in} = T_1(s_2 - s_1)$

$= 293(0.4249 - 1.0134)$

$= -172.43$ kJ/kg

Here the negative sign indicates that heat flow out to the system, Q_{out}

$Q_{out} = 172.43$ kJ/kg

Hence heat transferred from the system, $\boxed{Q_{out} = 172.43 \text{ kJ/kg}}$

Step 5 of 5

From the energy balance equation, we have

$E_{in} = E_{out}$

The equation can be rewritten as

$w_{in} = Q_{out} + u_2 - u_1$

$= 172.43 + (111.292 - 246.74)$

$= 36.98$ kJ/kg

Hence, work required for the system is $\boxed{w_{in} = 36.98 \text{ kJ/kg}}$

Question: 8.047P

Determine the heat transfer, in kJ/kg, for the reversible process 1-3 shown in Fig. P8-47.

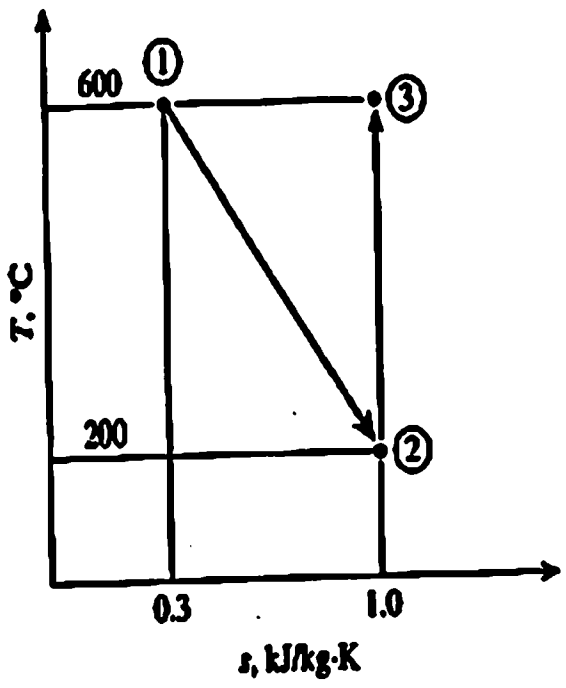


FIGURE P8–47

Step 1 of 5

Given

Temperature at point 1 is, $T_1 = 600°C = 873\ K$

Temperature at point 2 is, $T_2 = 200°C = 473\ K$

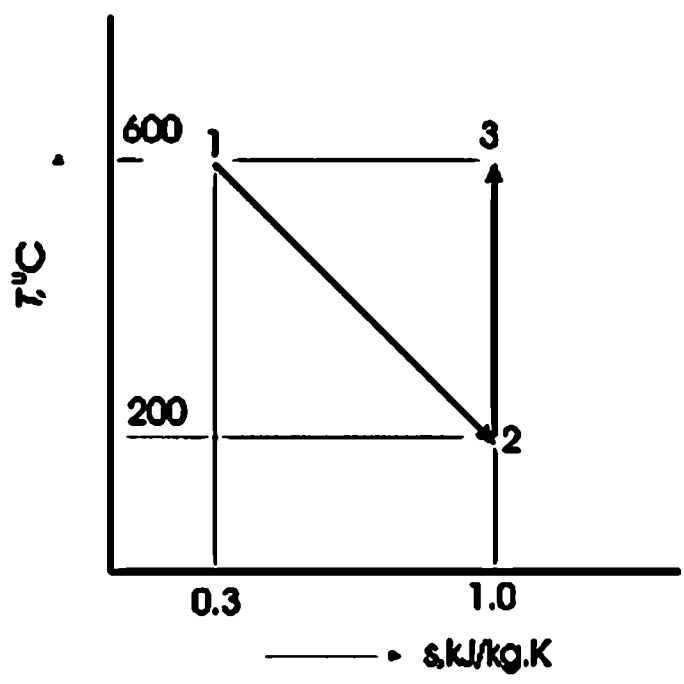
Temperature at point 3 is, $T_3 = 600°C = 873\ K$

Entropy at point 1 is, $s_1 = 0.3\ kJ/kg.K$

Entropy at point 2 is, $s_2 = 1.0\ kJ/kg.K$

Entropy at point 3 is, $s_3 = 1.0\ kJ/kg.K$

Step 2 of 5



Step 3 of 5

The heat transfer in any process is given by the area under the curve in a T-s diagram
So the heat transfer in the process 1-2 is given by

$$q_{1\text{-}2} = \int_1^2 T\,ds$$

$$q_{1\text{-}2} = T(s_2 - s_1)$$

In the process 1-2 the temperature is not constant and assuming that it is linear variation then we can take the temperature as average temperature. Therefore the above equation reduces to

$$q_{1\text{-}2} = \frac{T_1 + T_2}{2} \times (s_2 - s_1)$$

$$q_{1\text{-}2} = \frac{873 + 473}{2} \times (1.0 - 0.3)$$

$$q_{1\text{-}2} = \frac{1346}{2} \times (0.7)$$

$$q_{1\text{-}2} = 673 \times 0.7$$

$$q_{1\text{-}2} = 471.1\ kJ/kg$$

Step 4 of 5

The heat transfer in the process 2-3 is given by

$$q_{2\text{-}3} = \int_2^3 T\,ds$$

$$q_{2\text{-}3} = T(s_3 - s_2)$$

As the process 2-3 is a vertical line in the T-s diagram, there will not be any change in entropy. So, the above equation reduces to

$$q_{2\text{-}3} = T \times 0$$

$$q_{2\text{-}3} = 0$$

Step 5 of 5

The total heat transfer in the process 1-3 is given by

$$q_{1\text{-}3} = q_{1\text{-}2} + q_{2\text{-}3}$$

$$q_{1\text{-}3} = 471.1 + 0$$

$$\boxed{q_{1\text{-}3} = 471.1\ kJ/kg}$$

Therefore heat transfer for the reversible process 1-3 is 471.1 kJ/kg

Question: 8.048P

Calculate the heat transfer, in Btu/lbm, for the reversible process 1-3 shown in Fig. P8–18E.

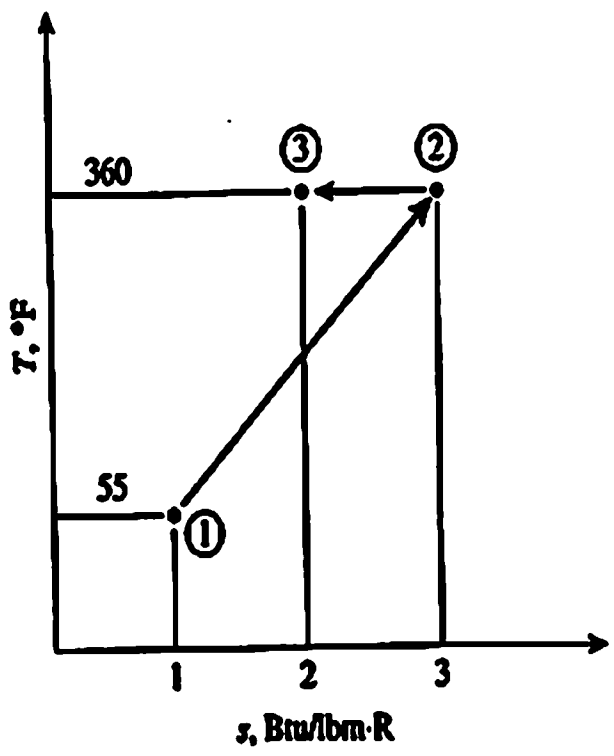


FIGURE P8–48E

Step 1 of 5

Given

Temperature at state 1 is, $T_1 = 55°F = 514.67\ R$

Temperature at state 2 is, $T_2 = 360°F = 819.67\ R$

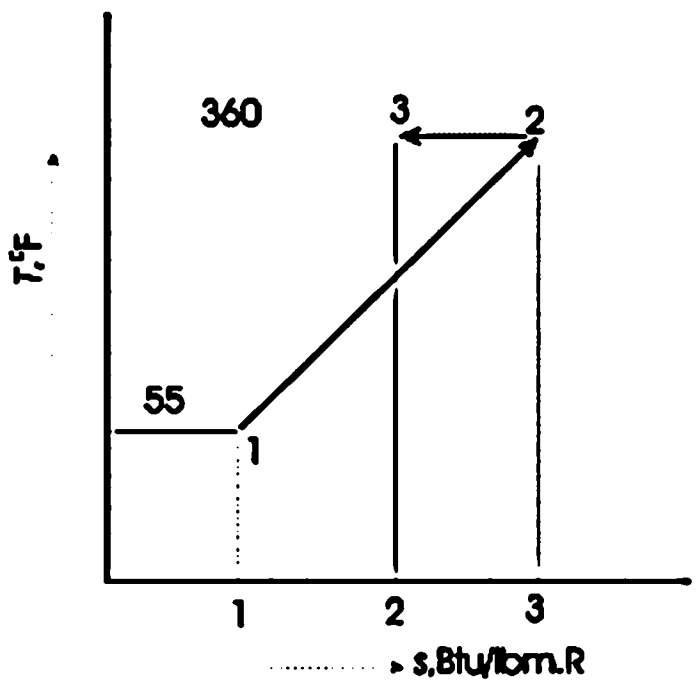
Temperature at state 3 is, $T_3 = 360°F = 819.67\ R$

Entropy at state 1 is, $s_1 = 1\ Btu/lbm\text{-}R$

Entropy at state 2 is, $s_2 = 3\ Btu/lbm\text{-}R$

Entropy at state 3 is, $s_3 = 2\ Btu/lbm\text{-}R$

Step 2 of 5



Step 3 of 5

The heat transfer in a process is given by the area under the curve in a T-s diagram.
So the heat transfer in the process 1-2 is given by

$$q_{1\text{-}2} = \int_1^2 T\,ds$$

$$q_{1\text{-}2} = T(s_2 - s_1)$$

In the process 1-2 the temperature is not constant and assuming that it is linear variation then we can take the temperature as average temperature. Therefore the above equation reduces to

$$q_{1\text{-}2} = \frac{T_1 + T_2}{2} \times (s_2 - s_1)$$

$$q_{1\text{-}2} = \frac{514.67 + 819.67}{2} \times (3 - 1)$$

$$q_{1\text{-}2} = \frac{1334.34}{2} \times 2$$

$$q_{1\text{-}2} = 1334.34\ Btu/lbm$$

Step 4 of 5

The heat transfer in the process 2-3 is given by

$$q_{2\text{-}3} = \int_2^3 T\,ds$$

As the process 2-3 is a horizontal line in the T-s diagram there will not be any change in temperature. So, the above equation reduces to

$$q_{2\text{-}3} = T_2(s_3 - s_2)$$

$$= 819.67 \times (2 - 3)$$

$$= 819.67 \times (-1)$$

$$q_{2\text{-}3} = -819.67\ Btu/lbm$$

Step 5 of 5

The total heat transfer in the process 1-3 is given by

$$q_{1\text{-}3} = q_{1\text{-}2} + q_{2\text{-}3}$$

$$q_{1\text{-}3} = 1334.34 + (-819.67)$$

$$q_{1\text{-}3} = 1334.34 - 819.67$$

$$\boxed{q_{1\text{-}3} = 514.67\ Btu/lbm}$$

Therefore heat transfer in the process 1-3 is 514.67 Btu/lbm

Question: 8.049P

Calculate the heat transfer, in kJ/kg, for the reversible steady-flow process 1-3 shown in Fig. P8-49

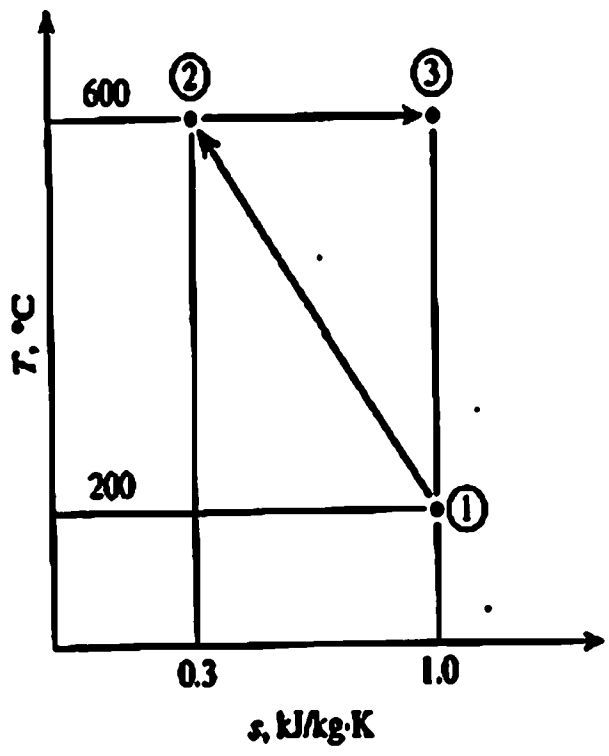


FIGURE P8–49

Step 1 of 5

Given

Temperature at point 1 is, $T_1 = 200^\circ C = 473\,K$

Temperature at point 2 is, $T_2 = 600^\circ C = 873\,K$

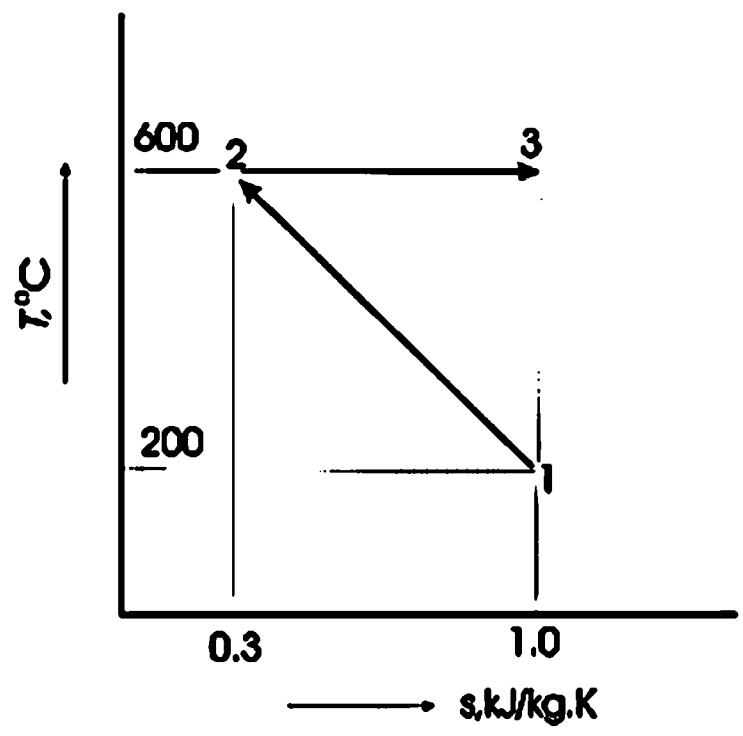
Temperature at point 3 is, $T_3 = 600^\circ C = 873\,K$

Entropy at point 1 is, $s_1 = 1.0\,kJ/kg.K$

Entropy at point 2 is, $s_2 = 0.3\,kJ/kg.K$

Entropy at point 3 is, $s_3 = 1.0\,kJ/kg.K$

Step 2 of 5



Step 3 of 5

The heat transfer in a process is given by the area under the curve in a T-s diagram.
So the heat transfer in the process 1-2 is given by

$$q_{1\text{-}2} = \int_1^2 T\,ds$$

$$q_{1\text{-}2} = T(s_2 - s_1)$$

In the process 1-2 the temperature is not constant and assuming that it is linear variation then we can take the temperature as average temperature. Therefore the above equation reduces to

$$q_{1\text{-}2} = \frac{T_1 + T_2}{2} \times (s_2 - s_1)$$

$$q_{1\text{-}2} = \frac{473+873}{2} \times (0.3 - 1.0)$$

$$q_{1\text{-}2} = \frac{1346}{2} \times (-0.7)$$

$$q_{1\text{-}2} = 673 \times (-0.7)$$

$$q_{1\text{-}2} = -471.1\,kJ/kg$$

Step 4 of 5

The heat transfer in the process 2-3 is given by

$$q_{2\text{-}3} = \int_2^3 T\,ds$$

$$q_{2\text{-}3} = T(s_3 - s_2)$$

As the process 2-3 is a horizontal line in the T-s diagram, there will not be any change in temperature. So, the above equation reduces to

$$q_{2\text{-}3} = T_2(s_3 - s_2)$$

$$q_{2\text{-}3} = 873 \times (1.0 - 0.3)$$

$$q_{2\text{-}3} = 873 \times 0.7$$

$$q_{2\text{-}3} = 611.1\,kJ/kg$$

Step 5 of 5

The total heat transfer in the process 1-3 is given by

$$q_{1\text{-}3} = q_{1\text{-}2} + q_{2\text{-}3}$$

$$q_{1\text{-}3} = -471.1 + 611.1$$

$$\boxed{q_{1\text{-}3} = 140\,kJ/kg}$$

Therefore heat transfer in the process 1-3 is 140 kJ/kg

Question: 8.050P

Steam is expanded in an isentropic turbine with a single inlet and outlet. At the inlet, the steam is at 2 MPa and 360°C The steam pressure at the outlet is 100 kPa. Calculate I the work produced by this turbine, in kJ/kg.

Step 1 of 5

Given

Inlet pressure of the steam is $P_1 = 2\ \text{MPa}$

Inlet temperature of the steam is $T_1 = 360\ \text{C}$

Outlet pressure of the steam is $P_2 = 100\ \text{kPa}$

Step 2 of 5

From Saturated water tables, at $P_1 = 2\ \text{MPa}$

At 350 C, $h = 3137.7\ \text{kJ/kg}$

At 400 C, $h = 3248.4\ \text{kJ/kg}$

Using interpolation technique,

$$h_1 = h_{at\ 350} + \frac{h_{at\ 400} - h_{at\ 350}}{400 - 350}(360 - 350)$$

$$h_1 = 3137.7 + \frac{3248.4 - 3137.7}{400 - 350}(360 - 350)$$

$$h_1 = 3137.7 + \frac{110.7}{50}(10)$$

$$h_1 = 3137.7 + 22.14$$

$$h_1 = 3159.84\ \text{kJ/kg}$$

Step 3 of 5

From Superheated steam tables, at $P_1 = 2\ \text{MPa}$

At 350 C, $s = 6.9583\ \text{kJ/kgK}$

At 400 C, $s = 7.1292\ \text{kJ/kgK}$

Using interpolation technique,

$$s_1 = s_{at\ 350} + \frac{s_{at\ 400} - s_{at\ 350}}{400 - 350}(360 - 350)$$

$$s_1 = 6.9583 + \frac{7.1292 - 6.9583}{400 - 350}(360 - 350)$$

$$s_1 = 6.9583 + \frac{0.1709}{50}(10)$$

$$s_1 = 6.9583 + 0.03418$$

$$s_1 = 6.99248\ \text{kJ/kgK}$$

Step 4 of 5

As the process is isentropic, $s_2 = s_1$

$s_2 = 6.99248\ \text{kJ/kgK}$

Now at pressure $P_2 = 100\ \text{kPa}$ from saturated steam tables

Entropy of saturated liquid $s_f = 1.3028\ \text{kJ/kgK}$

Entropy of evaporation $s_{fg} = 6.0562\ \text{kJ/kgK}$

Enthalpy of saturated liquid $h_f = 417.51\ \text{kJ/kg}$

Enthalpy of evaporation $h_{fg} = 2257.5\ \text{kJ/kg}$

Step 5 of 5

Quality at sate 2, $x_2 = \frac{s_2 - s_f}{s_{fg}}$

$$x_2 = \frac{6.99248 - 1.3028}{6.0562}$$

$$x_2 = 0.9395$$

Enthalpy at sate 2, $h_2 = h_f + x_2 h_{fg}$

$$h_2 = 417.51 + (0.9395)2257.5$$

$$h_2 = 2538.4\ \text{kJ/kg}$$

Work output per unit mass is

$$w_{out} = (h_1 - h_2)$$

$$w_{out} = 3159.84 - 2538.4$$

$$w_{out} = 621.4\ \text{kJ/kg}$$

Work produced by turbine $\boxed{w_{out} = 621.4\ \text{kJ/kg}}$

Question: 8.051P

An isentropic steam turbine processes 5 kg/s of steam at 4 MPa, which is exhausted at 50 kPa and 100°C. Five percent of this flow is diverted for feedwater heating at 700 kPa. Determine the power produced by this turbine, in kW.

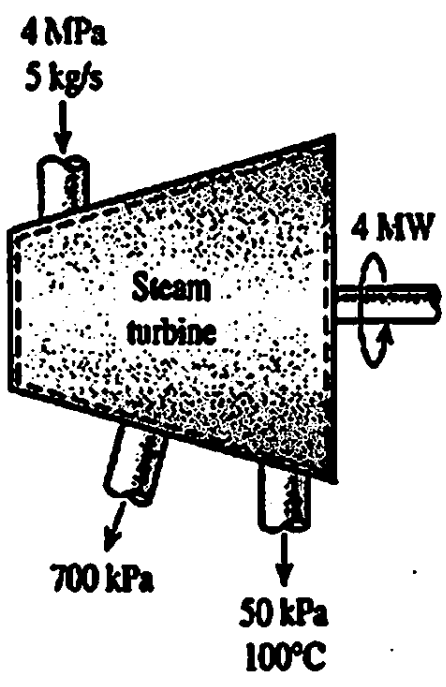
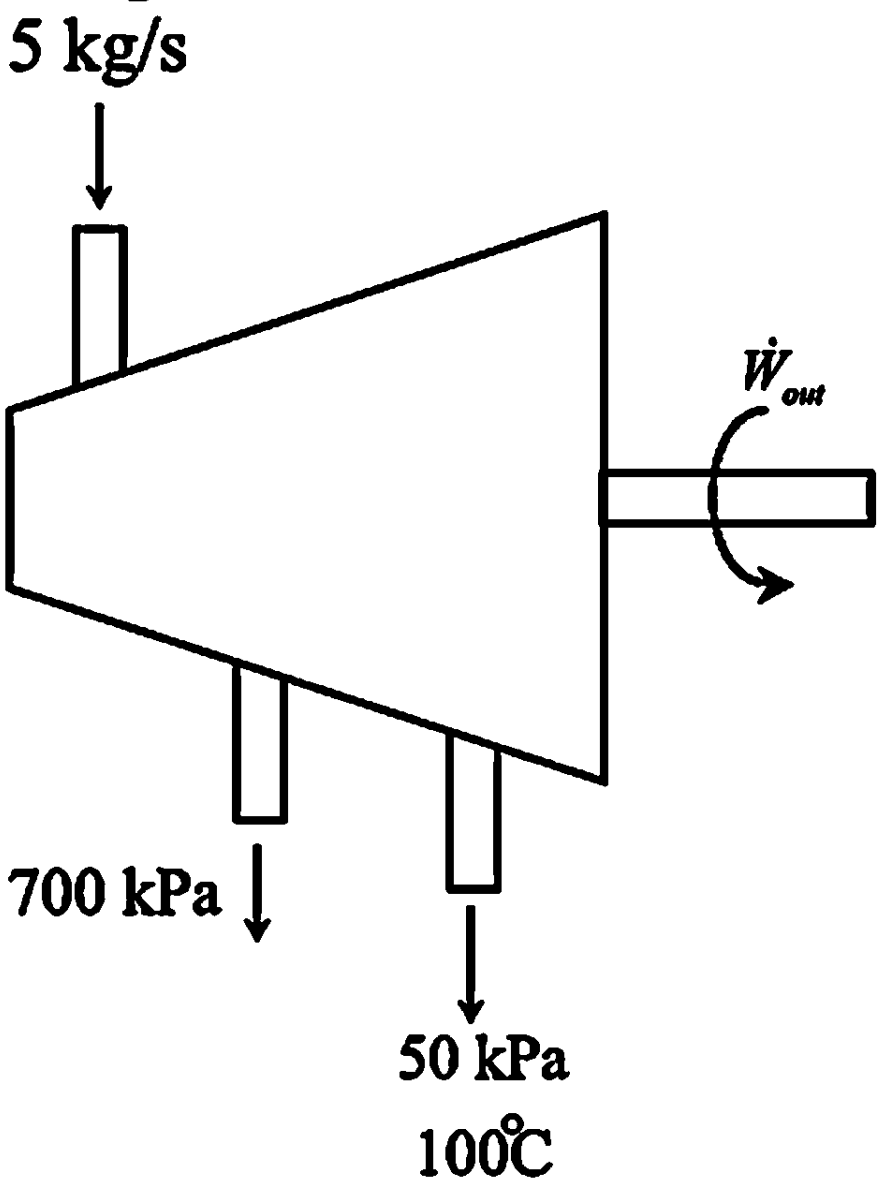


FIGURE P8–51

Step 1 of 7

Draw the schematic of the steam turbine.



Step 2 of 7

Calculate the mass of steam diverted to the feed water heater.

$$\dot{m}_2 = \frac{5}{100} \times \dot{m}_1 \text{ kg/s}$$

Here, $\dot{m}_1$ is the mass flow rate of steam into the turbine and $\dot{m}_2$ is the mass flow rate of diverted steam.

Substitute 5 kg/s for $\dot{m}_1$.

$$\begin{aligned}\dot{m}_2 &= 0.05\,\dot{m}_1 \text{ kg/s} \\ &= 0.05 \times 5 \text{ kg/s} \\ &= 0.25 \text{ kg/s}\end{aligned}$$

Step 3 of 7

Apply conservation of mass of mass to the turbine.

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3 \quad \text{..... (1)}$$

Here, $\dot{m}_3$ is the mass of steam leaving the turbine.

Substitute 5 kg/s for $\dot{m}_1$ and 0.25 kg/s for $\dot{m}_2$.

$$\begin{aligned}5 &= 0.25 + \dot{m}_3 \\ \dot{m}_3 &= 4.75 \text{ kg/s}\end{aligned}$$

Step 4 of 7

Obtain the properties of steam at a temperature of $100°C$ and at pressure 50 kPa from the table A-6 "super-heated water".

The enthalpy, $h_3 = 2682.4 \text{ kJ/kg}$

The entropy, $s_3 = 7.6953 \text{ kJ/kg·K}$

Step 5 of 7

As the expansion is isentropic,

$$s_1 = s_3$$

Obtain the properties of steam at a pressure of 4MPa and entropy of 7.6953 kJ/kg·K , by using the interpolation from the table A-6 "super-heated water".

$$\begin{aligned}h_1 &= 3906.3 + (4142.3 - 3906.3) \times \left(\frac{7.6953 - 7.6214}{7.8523 - 7.6214}\right) \\ &= 3981.8322 \text{ kJ/kg}\end{aligned}$$

Step 6 of 7

Obtain the properties of steam at a pressure of 700 kPa and entropy of 7.6953 kJ/kg·K , by using the interpolation from the table A-6 "super-heated water".

At pressure $P = 0.6 \text{ MPa}$, the enthalpy be is

$$\begin{aligned}(h_2)_1 &= 3166.1 + \frac{(7.6953 - 7.5481)(3270.8 - 3166.1)}{(7.7097 - 7.5481)} \\ &= 3261.47 \text{ kJ/kg}\end{aligned}$$

At pressure $P = 0.8 \text{ MPa}$, the enthalpy is

$$\begin{aligned}(h_2)_2 &= 3267.7 + \frac{(7.6953 - 7.5735)(3481.3 - 3267.7)}{(7.8692 - 7.5735)} \\ &= 3355.6826 \text{ kJ/kg}\end{aligned}$$

Obtain the value of enthalpy at 700 kPa by using double interpolation method.

$$\begin{aligned}h_2 &= 3261.47 + \frac{(0.7 - 0.6)(3355.6826 - 3261.47)}{(0.8 - 0.6)} \\ &= 3308.576 \text{ kJ/kg}\end{aligned}$$

Step 7 of 7

Apply energy balance to the turbine.

$$E_{in} = E_{out}$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{W}_{out}$$

$$\dot{W}_{out} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3$$

Substitute 5 kg/s for $\dot{m}_1$, 0.25 kg/s for $\dot{m}_2$, 4.75 kg/s for $\dot{m}_3$, 3981.8322 kJ/kg for h_1 , 3308.576 kJ/kg for 3308.576 kJ/kg and 2682.4 kJ/kg for h_3 .

$$\begin{aligned}\dot{W}_{out} &= 5 \times (3981.8322) - 0.25 \times 3308.576 - 4.75 \times 2682.4 \\ &= 6340.617 \text{ kW}\end{aligned}$$

Hence, the power produced in the turbine is $\boxed{6340.617 \text{ kW}}$.

Question: 8.052P

Water at 70 kPa and 100°C is compressed is entropically in a closed system to 4 MPa. Determine the final temperature of the water and the work required, in kJ/kg, for this compression.



FIGURE P8–52

Step 1 of 7

Given Data:

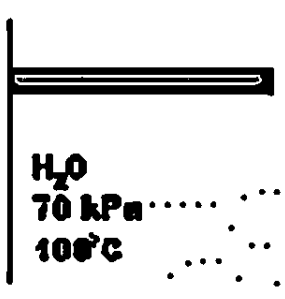
Initial pressure is, $P_1 = 70$ kPa.

Initial temperature is, $T_1 = 100\ °C = 373$ K.

Final pressure is, $P_2 = 4$ MPa $= 4000$ kPa.

Step 2 of 7

Let us draw the diagram:



Step 3 of 7

We should interpolate between 50 kPa and 100 kPa

From the superheated water,

at pressure $P_1 = 70$ kPa and temperature $T_1 = 100\ °C$,

Entropy of steam is interpolated as

$$\frac{s_1 - 7.6953}{7.3611 - 7.6953} = \frac{70 - 50}{100 - 50}$$

$$\frac{s_1 - 7.6953}{-0.3342} = \frac{20}{50}$$

$$\frac{s_1 - 7.6953}{-0.3342} = 0.4$$

$$s_1 = (0.4 \times -0.3342) + 7.6953$$

$$= -0.13368 + 7.6953$$

$$= 7.56162\ \text{kJ/kg} \cdot \text{K}$$

Step 4 of 7

From the superheated water,

At pressure $P_1 = 70$ kPa and temperature $T_1 = 100\ °C$,

internal energy of steam is interpolated as

$$\frac{u_1 - 2511.5}{2506.2 - 2511.5} = \frac{70 - 50}{100 - 50}$$

$$\frac{u_1 - 2511.5}{-5.3} = \frac{20}{50}$$

$$\frac{u_1 - 2511.5}{-5.3} = 0.4$$

$$u_1 = (-5.3 \times 0.4) + 2511.5$$

$$= -2.12 + 2511.5$$

$$= 2509.38\ \text{kJ/kg}$$

Step 5 of 7

As the cylinder is insulated, the process is isentropic. So, the final entropy is given by

$s_2 = s_1$

Therefore, $s_2 = 7.56162\ \text{kJ/kg} \cdot \text{K}$

From the superheated water,

At pressure $P_2 = 4$ MPa and entropy $s_2 = 7.56162$ kJ/kg°K, the temperature is interpolated as

$$\frac{T_2 - 600}{700 - 600} = \frac{7.56162 - 7.3706}{7.6214 - 7.3706}$$

$$\frac{T_2 - 600}{100} = \frac{0.19102}{0.2508}$$

$$\frac{T_2 - 600}{100} = 0.7616$$

$$T_2 = 600 + (0.7616 \times 100)$$

$$T_2 = 600 + 76.16$$

$$= 676.16°C$$

$\boxed{T_2 = 676.16\ °C}$

Step 6 of 7

From the superheated water,

at pressure $P_2 = 4$ MPa and entropy $s_2 = 7.56162\ \text{kJ/kg} \cdot \text{K}$,

The internal energy is interpolated as

$$\frac{u_2 - 3279.4}{3462.4 - 3279.4} = \frac{7.56162 - 7.3706}{7.6214 - 7.3706}$$

$$\frac{u_2 - 3279.4}{183} = \frac{0.19102}{0.2508}$$

$$\frac{u_2 - 3279.4}{183} = 0.7616$$

$$u_2 = 3279.4 + (0.7616 \times 183)$$

$$= 3279.4 + 139.3728$$

$$= 3418.7728\ \text{kJ/kg}$$

Step 7 of 7

As the system is closed system, we will consider internal energy for calculating the work done. The work done is given by

$w = u_2 - u_1$

$w = 3418.7728 - 2509.38$

$\boxed{w = 909.3928\ \text{kJ}}$

Therefore the final temperature is $\boxed{T_2 = 676.16\ °C}$ and the work done is

$\boxed{w = 909.3928\ \text{kJ}}$.

Question: 8.053P

0.5-kg of R-134a is expanded isentropically from 600 kPa and 30°C to 140 kPa. Determine the total heat transfer and work production for this expansion.

Step 1 of 5

Given

Mass of R-134a used is, $m = 0.5\ \text{kg}$.

Initial pressure is, $P_1 = 600\ \text{kPa}$.

Initial temperature is, $T_1 = 30\ ^\circ\text{C} = 303\ \text{K}$.

Final Pressure is, $P_2 = 140\ \text{kPa}$.

Step 2 of 5

From the superheated refrigerant 134a,

At pressure $P_1 = 600\ \text{kPa}$ and temperature $T_1 = 30\ ^\circ\text{C}$,

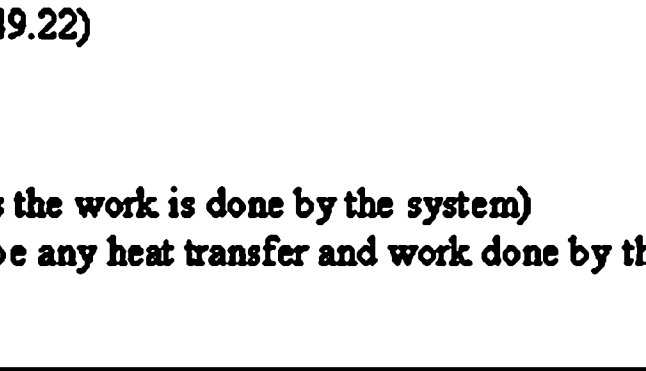
internal energy and entropy of refrigerant respectively are

$u_1 = 249.22\ \text{kJ/kg}$

$s_1 = 0.9499\ \text{kJ/kg} \cdot \text{K}$

Step 3 of 5

Let us draw the T-s diagram:



Step 4 of 5

Given the process is isentropic. Therefore there will not be any heat transfer.

$$\boxed{Q = 0}$$

As the process is isentropic (constant entropy), the final entropy is given by

$s_2 = s_1$

Therefore, $s_2 = 0.9499\ \text{kJ/kg} \cdot \text{K}$

From the superheated refrigerant 134a,

at pressure $P_2 = 140\ \text{kPa}$ and entropy $s_2 = 0.9499\ \text{kJ/kg} \cdot \text{K}$,

the internal energy is interpolated as

$$\frac{u_2 - 219.54}{225.91 - 219.54} = \frac{0.9499 - 0.9446}{0.9724 - 0.9446}$$

$$\frac{u_2 - 219.54}{6.37} = \frac{0.0053}{0.0278}$$

$$\frac{u_2 - 219.54}{6.37} = 0.1906$$

$$u_2 = (6.37 \times 0.1906) + 219.54$$

$$= 1.2141 + 219.54$$

$$= 220.7541\ \text{kJ/kg}$$

Step 5 of 5

The work done on the system is given by

$$W = m \times (u_2 - u_1)$$

$$= 0.5 \times (220.7541 - 249.22)$$

$$= 0.5 \times (-28.4659)$$

$$= -14.233\ \text{kJ}$$

(Negative value indicates the work is done by the system)

Therefore there will not be any heat transfer and work done by the system is

$$\boxed{W = -14.233\ \text{kJ}}$$

Question: 8.054P

Refrigerant-134a enters a steady-flow, adiabatic turbine as a saturated vapor at 1200 kPa and expands to 100 kPa. The power produced by the turbine is found to be 100 kW when the process is also reversible.

(*a*) Sketch the *T*-*s* diagram with respect to the saturation lines for this process.

(*b*) Determine the volume flow rate of the refrigerant-134a at the turbine exit, in m3 /s.

Step 1 of 8

Given Data:

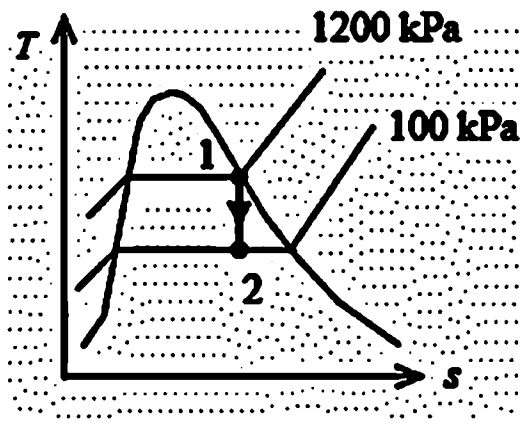
Inlet pressure is, $P_1 = 1200$ kPa.

Exit pressure is, $P_2 = 100$ kPa.

Power produced by the turbine is, $W_{tur} = 100$ kW.

Step 2 of 8

(a) Let us draw the T-s diagram:



Step 3 of 8

(b)

From The Saturated Refrigerant 134a,

At pressure $P_1 = 1200$ kPa and saturated vapor condition,

Enthalpy and entropy of refrigerant respectively are

$h_1 = h_g = 273.87$ kJ/kg

$s_1 = s_g = 0.91303$ kJ/kg · K

Step 4 of 8

From The Saturated Refrigerant 134a,

at pressure $P_2 = 100$ kPa saturated properties of refrigerant are as follows

Entropy of saturated liquid is $s_{f2} = 0.07188$ kJ/kg · K

Entropy of evaporation is $s_{fg2} = 0.87995$ kJ/kg · K

Enthalpy of saturated liquid is $h_{f2} = 17.28$ kJ/kg

Enthalpy of evaporation is $h_{fg2} = 217.16$ kJ/kg

Specific volume of saturated liquid is $v_{f2} = 0.0007259$ m^3/kg

Specific volume of saturated vapor is $v_{g2} = 0.19254$ m^3/kg

Step 5 of 8

Given entropy is constant. So, the final entropy is given by

$s_2 = s_1$

Therefore, $s_2 = 0.91303$ kJ/kg · K

The entropy is also given by

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$0.91303 = 0.07188 + x_2 \times 0.87995$

$$x_2 = \frac{0.91303 - 0.07188}{0.87995}$$

$$= \frac{0.84115}{0.87995}$$

$$= 0.956$$

Step 6 of 8

Exit enthalpy is given by

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$= 17.28 + 0.956 \times 217.16$

$= 17.28 + 207.60496$

$= 224.88496$ kJ/kg

Step 7 of 8

Exit specific volume is given by

$v_2 = v_{f2} + x_2 \times v_{fg2}$

$= v_{f2} + x_2 \times (v_{g2} - v_{f2})$

$= 0.0007259 + 0.956 \times (0.19254 - 0.0007259)$

$= 0.0007259 + 0.956 \times 0.1918141$

$= 0.0007259 + 0.18337428$

$= 0.1841002$ m^3/kg

Step 8 of 8

The turbine work is given by

$W_{tur} = m \times (h_1 - h_2)$

$100 = m \times (273.87 - 224.88496)$

$100 = m \times 48.98504$

$$m = \frac{100}{48.98504}$$

$m = 2.041$ kg/s

The volume flow rate is given by

$Q = m \times v_2$

$= 2.041 \times 0.1841002$

$= 0.3757$ m^3/s

Therefore volume flow rate is $\boxed{Q = 0.3757 \text{ m}^3/\text{s}}$

Question: 8.055P

Two kilograms of saturated water vapor at 600 kPa are contained in a piston-cylinder device. The water expands adiabatically until the pressure is 100 kPa and is said to produce 700 kJ of work output.

(*a*) Determine the entropy change of the water, in kJ/kg-K.

(*b*) Is this process realistic? Using the *T-s* diagram for the process and the concepts of second law, support your answer.

Step 1 of 9

Given

Mass of water in the device is, $m = 2\ kg$.

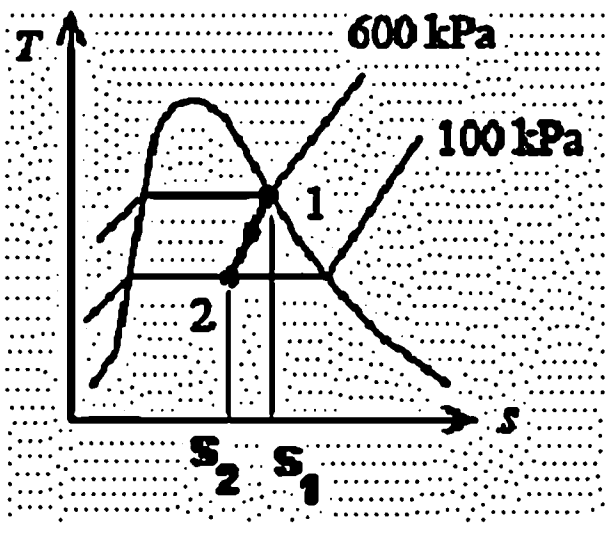
Initial pressure is, $P_1 = 600\ kPa$.

Final pressure is, $P_2 = 100\ kPa$.

Power produced by the device is, $W_{dev} = 700\ kJ$.

Step 2 of 9

Let us draw the T-s diagram:



Step 3 of 9

(a)

From the saturated water pressure table,

At pressure $P_1 = 600\ kPa$ and saturated vapor condition,

Internal energy and entropy of steam respectively are

$u_1 = u_g = 2566.8\ kJ/kg$

$s_1 = s_g = 6.7593\ kJ/kg \cdot K$

Step 4 of 9

Work output is given by

$$W_{dev} = m \times (u_1 - u_2)$$

$$u_2 = u_1 - \frac{W_{dev}}{m}$$

$$u_2 = 2566.8 - \frac{700}{2}$$

$$u_2 = 2566.8 - 350$$

$$u_2 = 2216.8\ kJ/kg$$

Step 5 of 9

From the saturated water pressure table,

At pressure $P_2 = 100\ kPa$ saturated properties of refrigerant are as follows

Entropy of saturated liquid is $s_{f2} = 1.3028 kJ/kg \cdot K$

Entropy of evaporation is $s_{fg2} = 6.0562\ kJ/kg \cdot K$

Internal energy of saturated liquid is $u_{f2} = 417.4\ kJ/kg$

Internal energy of evaporation is $u_{fg2} = 2088.2\ kJ/kg$

Step 6 of 9

The internal energy is also given by

$$u_2 = u_{f2} + x_2 \times u_{fg2}$$

$$2216.8 = 417.4 + x_2 \times 2088.2$$

$$x_2 = \frac{2216.8 - 417.4}{2088.2}$$

$$= \frac{1799.4}{2088.2}$$

$$= 0.8617$$

Step 7 of 9

Exit entropy is given by

$$s_2 = s_{f2} + x_2 \times s_{fg2}$$

$$= 1.3028 + 0.8617 \times 6.0562$$

$$= 1.3028 + 5.2186$$

$$= 6.5214\ kJ/kg \cdot K$$

Step 8 of 9

The entropy change is given by

$$\Delta s = s_2 - s_1$$

$$= 6.5214 - 6.7593$$

$$= -0.2379 kJ/kg \cdot K$$

$$\boxed{\Delta s = -0.2379\ kJ/kg^{\circ}K}$$

(Negative sign indicates that the entropy decreases)

Step 9 of 9

(b) Generally the entropy of an adiabatic process should remain constant. The process is not realistic as the entropy of the steam decreases.

Question: 8.056P

Refrigerant-134a enters a steady-flow adiabatic compressor as a saturated vapor at 320 kPa and is compressed to 1200 kPa. The minimum power supplied to the compressor is found to be 100 kW.

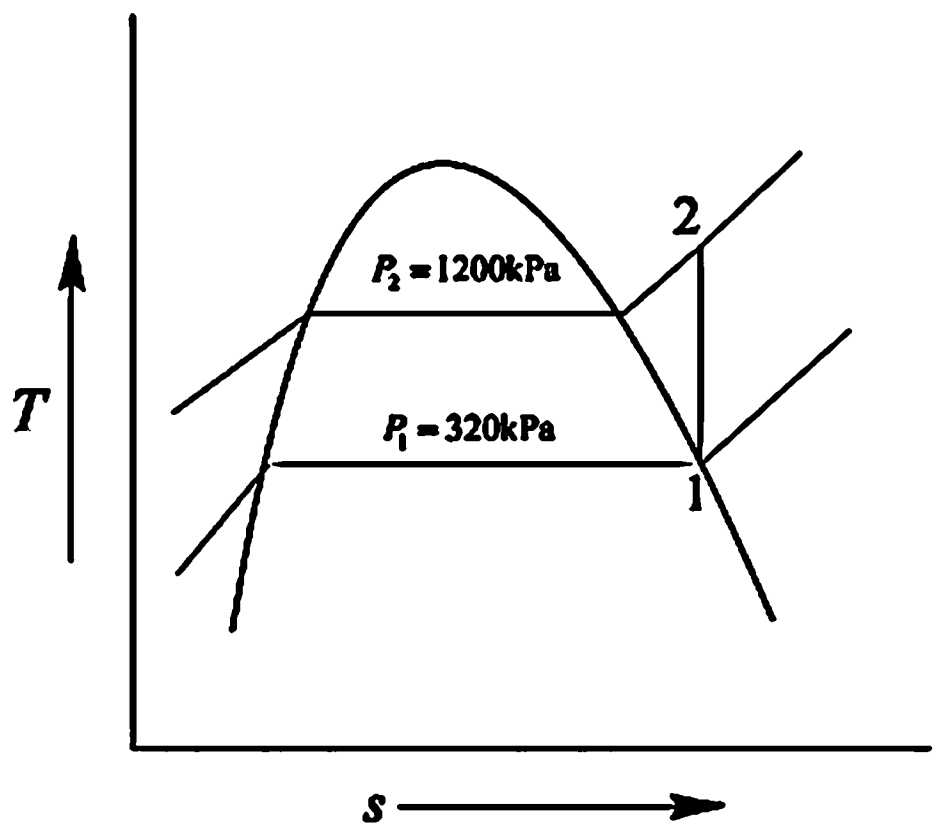
(*a*) Sketch the *T*-*s* diagram with respect to the saturation lines for this process.

(*b*) Determine the volume flow rate of the refrigerant-134a at the compressor inlet, in m3 /s.

Step 1 of 4

(a)

Represent the compression process 1-2 on *T*-*s* diagram as shown below:



Step 2 of 4

(b)

Obtain the properties from saturated refrigerant 134a tables at pressure $P_1 = 320\ \text{kPa}$

Specific enthalpy of saturated vapor $h_1 = h_g = 251.88\ \text{kJ/kg}$

Specific entropy of saturated vapor $s_1 = s_g = 0.93006\ \text{kJ/kg·K}$

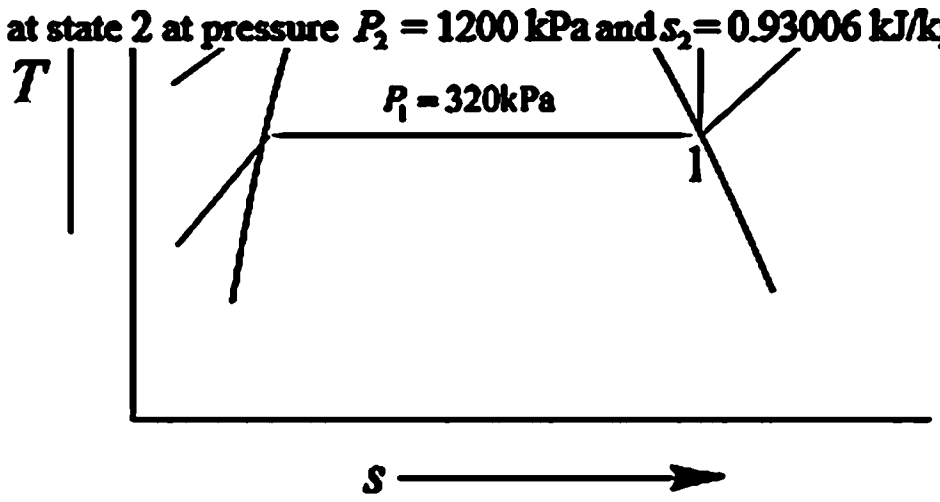
Specific volume of saturated vapor $v_1 = v_g = 0.063604\ \text{m}^3/\text{kg}$

Entropy remains constant during the process 1-2. Find the enthalpy at state 2 by using the condition,

$s_2 = s_1$

$s_2 = 0.93006\ \text{kJ/kg·K}$

Obtain enthalpy at state 2 at pressure $P_2 = 1200\ \text{kPa}$ and $s_2 = 0.93006\ \text{kJ/kg·K}$ from



Step 2 of 4

(b)

Obtain the properties from saturated refrigerant 134a tables at pressure $P_1 = 320\ \text{kPa}$

Specific enthalpy of saturated vapor $h_1 = h_g = 251.88\ \text{kJ/kg}$

Specific entropy of saturated vapor $s_1 = s_g = 0.93006\ \text{kJ/kg·K}$

Specific volume of saturated vapor $v_1 = v_g = 0.063604\ \text{m}^3/\text{kg}$

Entropy remains constant during the process 1-2. Find the enthalpy at state 2 by using the condition,

$s_2 = s_1$

$s_2 = 0.93006\ \text{kJ/kg·K}$

Obtain enthalpy at state 2 at pressure $P_2 = 1200\ \text{kPa}$ and $s_2 = 0.93006\ \text{kJ/kg·K}$ from

$= 0.23137\ \text{m}^3/\text{s}$

Therefore, volume flow rate of the refrigerant at the inlet of the compressor is

$\boxed{0.23137\ \text{m}^3/\text{s}}$

Question: 8.057P

Steam enters a steady-flow adiabatic nozzle with a low inlet velocity as a saturated vapor at 6 MPa and expands to 1.2 MPa.

(*a*) Under the conditions that the exit velocity is to be the maximum possible value, sketch the *T-s* diagram with respect to the saturation lines for this process.

(*b*) Determine the maximum exit velocity of the steam, in m/s.

Step 1 of 5

Given

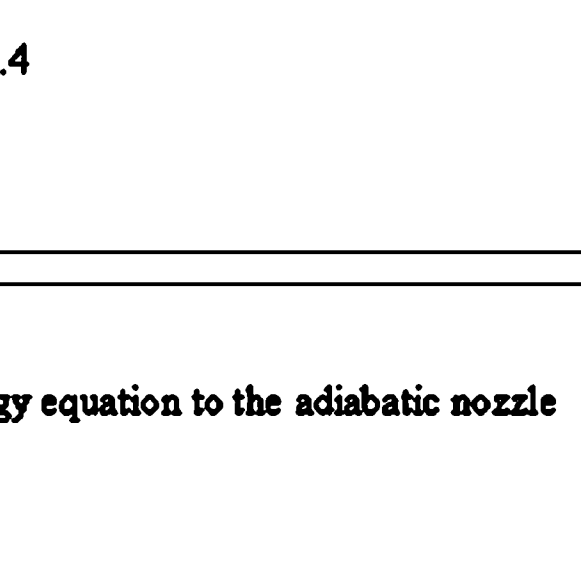
Saturated vapor enters the nozzle at pressure $P_1 = 6\ \text{MPa} = 6000\ \text{kPa}$

Pressure at nozzle exit is, $P_2 = 1.2\ \text{MPa} = 1200\ \text{kPa}$

Step 2 of 5

(a)

For the maximum possible exit velocity $h_1 - h_2$ should be maximum, for which the process to be isentropic.



Step 3 of 5

(b)

From saturated water tables at pressure $P_1 = 6\ \text{MPa}$

$h_1 = h_g = 2784.6\ \text{kJ/kg}$

$s_1 = s_g = 5.8902\ \text{kJ/kg.K}$

From saturated water tables at pressure $P_2 = 1.2\ \text{MPa}$

$h_{f2} = 798.33\ \text{kJ/kg}$

$h_{fg2} = 1985.4\ \text{kJ/kg}$

$s_{f2} = 2.2159\ \text{kJ/kg.K}$

$s_{fg2} = 4.3058\ \text{kJ/kg.K}$

Step 4 of 5

Since the process is isentropic the exit specific entropy can be calculated as

$s_2 = s_1$

$s_{f2} + x_2 \times s_{fg2} = s_1$

$2.2159 + x_2 \times 4.3058 = 5.8902$

$$x_2 = \frac{5.8902 - 2.2159}{4.3058}$$

$x_2 = 0.853$

Specific enthalpy at the exit can be calculated as

$h_2 = h_{f2} + x_2 \times h_{fg2}$

$h_2 = 798.33 + 0.8533 \times 1985.4$

$h_2 = 798.33 + 1694.14$

$h_2 = 2492.47\ \text{kJ/kg}$

Step 5 of 5

Applying steady flow energy equation to the adiabatic nozzle

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$h_1 - h_2 + \frac{V_1^2}{2} = \frac{V_2^2}{2}$$

Since inlet velocity is very low $V_1 = 0$

$$(2784.6 - 2492.47) \times 10^3 + 0 = \frac{V_2^2}{2}$$

$V_2 = 764.369\ \text{m/s}$

Therefore the maximum exit velocity of the steam is $\boxed{V_2 = 764.369\ \text{m/s}}$

Question: 8.058P

A rigid, 20-L steam cooker is arranged with a pressure relief valve set to release vapor and maintain the pressure once the pressure inside the cooker reaches 150 kPa. Initially, this cooker is filled with water at 175 kPa with a quality of 10 percent. Heat is now added until the quality inside the cooker is 40 percent. Determine the minimum entropy change of the thermal energy reservoir supplying this heat.

Step 1 of 7

Obtain the properties of water from table A-15 "properties of saturated water" at pressure of $175\ \text{kPa}$.

Specific volume of saturated liquid $v_{f1} = 0.0010057\ \text{m}^3/\text{kg}$

Specific volume of saturated vapor $v_{g1} = 1.0037\ \text{m}^3/\text{kg}$

Specific entropy of saturated liquid $s_{f1} = 1.4850\ \text{kJ/kg}\cdot\text{K}$

Specific entropy of liquid vapor mixture $s_{fg1} = 5.6865\ \text{kJ/kg}\cdot\text{K}$

Step 2 of 7

Find the initial specific volume at the initial condition.

$$v_1 = v_{f1} + x_1 \times \left(v_{g1} - v_{f1}\right)$$

Substitute $0.0010057\ \text{m}^3/\text{kg}$ for v_{f1}, $1.0037\ \text{m}^3/\text{kg}$ for v_{g1} and 0.1 for x.

$$\begin{aligned} v_1 &= 0.001057 + 0.1 \times (1.0037 - 0.001057) \\ &= 0.10132\ \text{m}^3/\text{kg} \end{aligned}$$

Find the initial specific entropy by using the equation,

$$s_1 = s_{f1} + x_1 \times s_{fg1}$$

Substitute $1.4850\ \text{kJ/kg}\cdot\text{K}$ for s_{f1}, $5.6865\ \text{kJ/kg}\cdot\text{K}$ for s_{fg1}.

$$\begin{aligned} s_1 &= 1.485 + 0.1 \times 5.6865 \\ &= 2.05365\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Step 3 of 7

Obtain the properties of water from table A-15 "properties of saturated water" at pressure of $150\ \text{kPa}$.

Specific volume of saturated liquid $v_{f2} = 0.0010053\ \text{m}^3/\text{kg}$

Specific volume of saturated vapor $v_{g2} = 1.1594\ \text{m}^3/\text{kg}$

Specific entropy of saturated liquid $s_{f2} = 1.4337\ \text{kJ/kg}\cdot\text{K}$

Specific entropy of liquid vapor mixture $s_{fg2} = 5.7894\ \text{kJ/kg}\cdot\text{K}$

Specific entropy of saturated vapor $s_{g2} = 7.2231\ \text{kJ/kg}\cdot\text{K}$

Step 4 of 7

Find the final specific volume at the final condition.

$$v_2 = v_{f2} + x_2 \times \left(v_{g2} - v_{f2}\right)$$

Substitute $0.0010053\ \text{m}^3/\text{kg}$ for v_{f2}, $1.1594\ \text{m}^3/\text{kg}$ for v_{g2} and 0.4 for x as the final quality is 40%.

$$\begin{aligned} v_2 &= 0.001053 + 0.4 \times (1.1594 - 0.001053) \\ &= 0.4643\ \text{m}^3/\text{kg} \end{aligned}$$

Find the final specific entropy by using the equation,

$$s_2 = s_{f2} + x_2 \times s_{fg2}$$

Substitute $1.4337\ \text{kJ/kg}\cdot\text{K}$ for s_{f2}, $5.7894\ \text{kJ/kg}\cdot\text{K}$ for s_{fg2} and 0.4 for x_2.

$$\begin{aligned} s_2 &= 1.4337 + 0.4 \times 5.7894 \\ &= 3.74946\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Step 5 of 7

Find the initial mass of water by using the expression,

$$m_1 = \frac{V}{v_1}$$

Here, V is volume of steam cooker

Substitute $20\times10^{-3}\ \text{m}^3$ for V and $0.10132\ \text{m}^3/\text{kg}$ for v_1.

$$\begin{aligned} m_1 &= \frac{0.02}{0.10132} \\ &= 0.1974\ \text{kg} \end{aligned}$$

Calculate the final mass using the relation,

$$m_2 = \frac{V}{v_2}$$

Substitute $20\times10^{-3}\ \text{m}^3$ for V and $0.4643\ \text{m}^3/\text{kg}$ for v_2.

$$\begin{aligned} m_2 &= \frac{0.02}{0.46439} \\ &= 0.0431\ \text{kg} \end{aligned}$$

Step 6 of 7

Apply conservation of mass equation to the system,

$$\frac{dm}{dt} = \sum_{in} m - \sum_{out} m$$

Since there is no mass of water entering into the system, $\sum_{in} m = 0$

$$\frac{dm}{dt} = 0 - \sum_{out} m$$

$$\frac{dm}{dt} = -m_{out}$$

Apply Entropy balance by using the equation,

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} + s \times m_{out} \geq 0$$

$$\frac{dS_{surr}}{dt} + \frac{d(ms)}{dt} - s\frac{dm}{dt} \geq 0$$

Multiply with dt and integrate the above equation.

$$\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}\left(m_2 - m_1\right) \geq 0$$

Here, s_{out} is specific entropy of saturated vapor coming out from the steam cooker at pressure $p_2 = 150\ \text{kPa}$

Step 7 of 7

Obtain the properties of water from table A-15 "properties of saturated water" at pressure of $150\ \text{kPa}$.

Specific entropy of saturated vapor $s_{g2} = 7.2231\ \text{kJ/kg}\cdot\text{K}$

Substitute $0.0431\ \text{kg}$ for m_2, $0.1974\ \text{kg}$ for m_1, $3.74946\ \text{kJ/kg}\cdot\text{K}$ for s_2 and $7.2231\ \text{kJ/kg}\cdot\text{K}$ for s_{out}.

$$\begin{aligned} &\Delta S_{surr} + m_2 s_2 - m_1 s_1 - s_{out}\left(m_2 - m_1\right) \geq 0 \\ &\Delta S_{surr} + (0.0431\times3.74946) - (0.1974\times2.05365) - 7.2231\times(0.0431 - 0.1974) \geq 0 \\ &\Delta S_{surr} + 0.1616 - 0.4054 - 7.2231\times(-0.1543) \geq 0 \\ &\Delta S_{surr} + 0.1616 - 0.4054 + 1.1145 \geq 0 \\ &\Delta S_{surr} + 0.8707 \geq 0 \\ &\Delta S_{surr} \geq -0.8707\ \text{kJ/K} \end{aligned}$$

Therefore, the minimum entropy change required is $\boxed{-0.8707\ \text{kJ/K}}$

Question: 8.-59WATER-ST

In the preceding problem, the water is stirred at the same time that it is being heated. Determine the minimum entropy change of the heat-supplying source if 100 kJ of work is done on the water as it is being heated.

Step 1 of 11

Given data:

Volume of the cooker is, $V = 20\ \text{L}$

$= 0.02\ \text{m}^3$

Initial pressure in the cooker is, $P_1 = 175\ \text{kPa}$

Initial quality of steam in the cooker is, $x_1 = 10\ \%$

$= 0.1$

Final quality of steam in the cooker is, $x_2 = 40\ \%$

$= 0.4$

Pressure maintained in the cooker is, $P_2 = 150\ \text{kPa}$

Work done on water is, $W = 100\ \text{kJ}$

Step 2 of 11

From Saturated water pressure table

At pressure $P_1 = 175\ \text{kPa}$ the properties of steam are

$v_{f1} = 0.001057\ \text{m}^3/\text{kg}$

$v_{g1} = 1.0037\ \text{m}^3/\text{kg}$

$s_{f1} = 1.485\ \text{kJ/kg°K}$

$s_{fg1} = 5.6865\ \text{kJ/kg°K}$

From Saturated water pressure table

At pressure $P_2 = 150\ \text{kPa}$ the properties of steam are

$v_{f2} = 0.001053\ \text{m}^3/\text{kg}$

$v_{g2} = 1.1594\ \text{m}^3/\text{kg}$

$s_{f2} = 1.4337\ \text{kJ/kg°K}$

$s_{fg2} = 5.7894\ \text{kJ/kg°K}$

Step 3 of 11

Initial specific volume is given by

$v_1 = v_{f1} + x_1 \times v_{fg1}$

$v_1 = v_{f1} + x_1 \times (v_{g1} - v_{f1})$

$v_1 = 0.001057 + 0.1 \times (1.0037 - 0.001057)$

$v_1 = 0.001057 + 0.1 \times 1.002643$

$v_1 = 0.001057 + 0.1002643$

$v_1 = 0.10132\ \text{m}^3/\text{kg}$

Step 4 of 11

Initial specific entropy is given by

$s_1 = s_{f1} + x_1 \times s_{fg1}$

$s_1 = 1.485 + 0.1 \times 5.6865$

$s_1 = 1.485 + 0.56865$

$s_1 = 2.05365\ \text{kJ/kg°K}$

Step 5 of 11

Final specific volume is given by

$v_2 = v_{f2} + x_2 \times v_{fg2}$

$v_2 = v_{f2} + x_2 \times (v_{g2} - v_{f2})$

$v_2 = 0.001053 + 0.4 \times (1.1594 - 0.001053)$

$v_2 = 0.001053 + 0.4 \times 1.158347$

$v_2 = 0.001053 + 0.46334$

$v_2 = 0.46439\ \text{kJ/kg°K}$

Step 6 of 11

Final specific entropy is given by

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$s_2 = 1.4337 + 0.4 \times 5.7894$

$s_2 = 1.4337 + 2.31576$

$s_2 = 3.74946\ \text{kJ/kg°K}$

Step 7 of 11

Initial mass is given by

$$m_1 = \frac{V}{v_1}$$

$$m_1 = \frac{0.02}{0.10132}$$

$m_1 = 0.1974\ \text{kg}$

Final mass is given by

$$m_2 = \frac{V}{v_2}$$

$$m_2 = \frac{0.02}{0.46439}$$

$m_2 = 0.0431\ \text{kg}$

Step 8 of 11

According to conversation of mass principle,

$$\frac{dm}{dt} = \sum_{in} m - \sum_{out} m$$

As no mass is coming in, $\sum_{in} m = 0$

$$\frac{dm}{dt} = 0 - \sum_{out} m$$

$$\frac{dm}{dt} = -m_{out}$$

Step 9 of 11

Work done on water cannot be added to entropy balance as work done has no effect on entropy.

So, entropy balance is given by

$$\frac{dS_{sur}}{dt} + \frac{d(ms)}{dt} + s \times m_{out} \geq 0$$

$$\frac{dS_{sur}}{dt} + \frac{d(ms)}{dt} - s\frac{dm}{dt} \geq 0$$

Multiplying by dt and integrating we get

$\Delta S_{sur} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$

Step 10 of 11

Entropy of water vapor coming out is given by

$s_{out} = s_g$ at final pressure (i.e., 150 kPa)

$s_{out} = 7.2231\ \text{kJ/kg°K}$

Step 11 of 11

$\Delta S_{sur} + m_2 s_2 - m_1 s_1 - s_{out}(m_2 - m_1) \geq 0$

$\Delta S_{sur} + (0.0431 \times 3.74946) - (0.1974 \times 2.05365) - 7.2231 \times (0.0431 - 0.1974) \geq 0$

$\Delta S_{sur} + 0.1616 - 0.4054 - 7.2231 \times (-0.1543) \geq 0$

$\Delta S_{sur} + 0.1616 - 0.4054 + 1.1145 \geq 0$

$\Delta S_{sur} + 0.8707 \geq 0$

$\Delta S_{sur} \geq -0.8707\ \text{kJ/kg}$

Therefore the minimum entropy change required is $\boxed{-0.8707\ \text{kJ/kg}}$

Question: 8.060P

A piston-cylinder device contains 5 kg of steam at 100°C with a quality of 50 percent. This steam undergoes two processes as follows:

1-2 Heat is transferred to the steam in a reversible manner while the temperature is held constant until the steam exists as a saturated vapor.

2-3 The steam expands in an adiabatic, reversible process until the pressure is 15 kPa.

(*a*) Sketch these processes with respect to the saturation lines on a single *T-s* diagram.

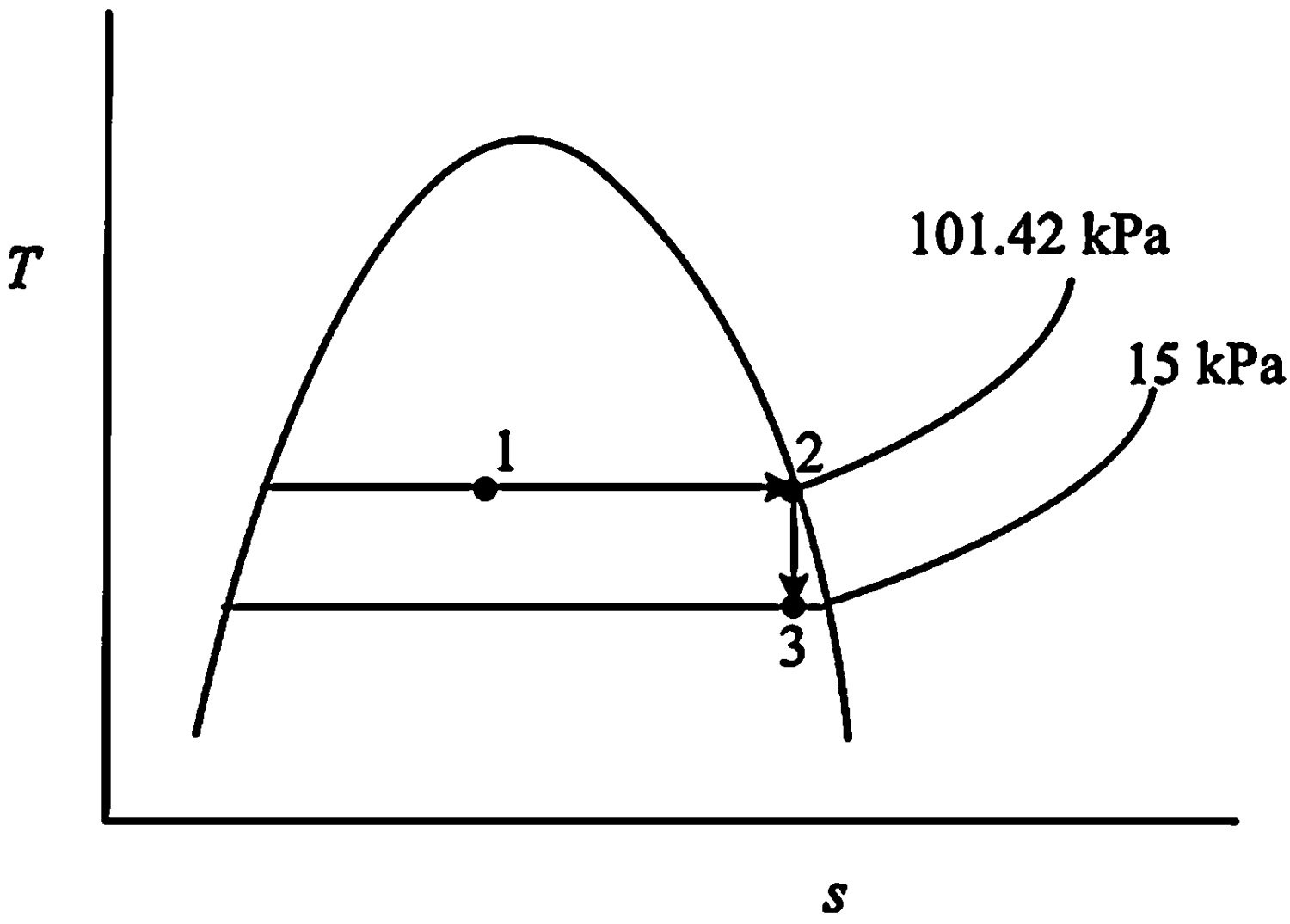
(*b*) Determine the heat transferred to the steam in process 1-2, in kJ.

(*c*) Determine the work done by the steam in process 2-3, in kJ.

Step 1 of 5

(a)

Consider the *T-s* diagram of the processes underwent by the steam.



Step 2 of 5

(b)

Obtain the properties of steam at state 1 from the table A.4, "Saturated water-Temperature table" at a temperature of $100°C$.

Specific enthalpy of saturated liquid, $h_f = 419.17\,kJ/kg$

Specific enthalpy of saturated vapor, $h_g = 2675.6\,kJ/kg$

Determine the enthalpy of the steam at a temperature of $100°C$ and a quality of 0.5.

$$h_1 = h_f + x_1\left(h_g - h_f\right)$$

Substitute $419.17\,kJ/kg$ for h_f , and $2675.6\,kJ/kg$ for h_g .

$$h_1 = 419.17 + 0.5(2675.6 - 419.17)$$
$$= 1547.385\,kJ/kg$$

Step 3 of 5

Obtain the properties of saturated vapor at state 2 from the table A.4, "Saturated water-Temperature table" at a temperature of $100°C$.

Specific enthalpy of saturated vapor, $h_2 = h_g = 2675.6\,kJ/kg$

Internal energy of saturated vapor, $u_2 = u_g = 2506\,kJ/kg$

Specific entropy of saturated vapor, $s_2 = s_g = 7.3542\,kJ/kg \cdot K$

As the process 2-3 is isentropic,

$$s_2 = s_3 = 7.3542\,kJ/kg \cdot K$$

Obtain the properties of the steam at state 3 from the EES software at a pressure of 15 kPa and an entropy of $7.3542\,kJ/kg \cdot K$.

$$u_3 = 2247.9\,kJ/kg$$

Step 4 of 5

Determine the heat transferred to the steam in process 1-2.

$$Q_{12} = m\left(h_2 - h_1\right)$$

Substitute 5 kg for m , $2675.6\,kJ/kg$ for h_2 , and $1547.385\,kJ/kg$ for h_1 .

$$Q_{12} = 5(2675.6 - 1547.385)$$
$$= 5641.075\ kJ$$

Hence the transferred to the steam in process 1-2 is, $\boxed{5641.075\ kJ}$

Step 5 of 5

(c)

Determine the work done by the steam in the process 2-3.

$$W_{23} = m\left(u_2 - u_3\right)$$

Substitute 5 kg for m , $2506\,kJ/kg$ for u_2 , and $2247.9\,kJ/kg$ for u_3 .

$$W_{23} = 5(2506 - 2247.9)$$
$$= 1290.5\ kJ$$

Hence work done by the steam in the process 2-3 is, $\boxed{1290.5\ kJ}$.

Question: 8.061P

A 0.8-ft3 well-insulated rigid can initially contains refrigerant-134a at 140 psia and 50°F. Now a crack develops in the can, and the refrigerant starts to leak out slowly, Assuming the refrigerant remaining in the can has undergone a reversible, adiabatic process, determine the final mass in the can when the pressure drops to 30 psia.

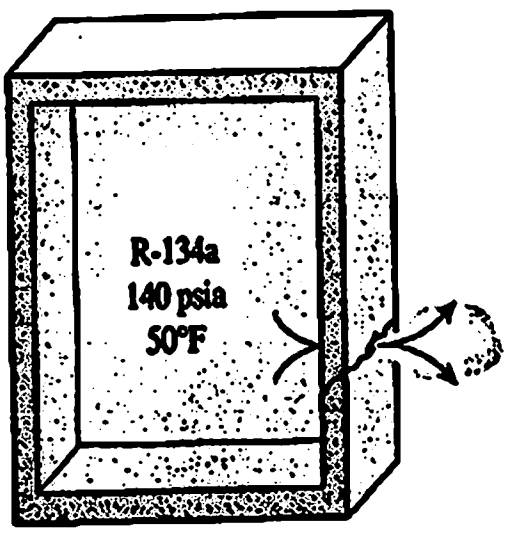


FIGURE P8–61E

Step 1 of 6

Given

Volume of the tank is, $V = 0.8\ ft^3$

Initial pressure of R-134a $P_1 = 140$ psia

Initial temperature $T_1 = 50°F$

Final pressure is, $P_2 = 30$ psia

Step 2 of 6

From saturated R-134a tables at pressure $P_1 = 140$ psia

Saturation temperature is $T_{sat} = 100.51°F$.

Since $T_1 < 100.51°F$

The refrigerant will be in sub cooled region.

We will consider the values at saturated condition at 50°F

From saturated R-134a tables at pressure $T_1 = 50°F$

Entropy of refrigerant is

$s_1 = s_f = 0.06039$ Btu/lbm.R

Step 3 of 6

From saturated R-134a tables at pressure $P_2 = 30$ psia saturated properties of refrigerant are as follows

Entropy of saturated liquid is $s_{f2} = 0.03793$ Btu/lbm.R

Entropy of evaporation is $s_{fg2} = 0.18589$ Btu/lbm.R

Specific volume of saturated liquid is $v_{f2} = 0.01209\ ft^3/lbm$

Specific volume of saturated vapor is $v_{g2} = 1.5492\ ft^3/lbm$

Step 4 of 6

Since the process is reversible adiabatic process entropy remains constant.

$s_2 = s_1$

Therefore, $s_2 = 0.06039$ Btu/lbm.R

The entropy is also given by

$s_2 = s_{f2} + x_2 \times s_{fg2}$

$0.06039 = 0.03793 + x_2 \times 0.18589$

$$x_2 = \frac{0.06039 - 0.03793}{0.18589}$$

$$x_2 = \frac{0.02246}{0.18589}$$

$x_2 = 0.1208$

Step 5 of 6

Final specific volume is given by

$v_2 = v_{f2} + x_2 \times v_{fg2}$

$v_2 = v_{f2} + x_2 \times (v_{g2} - v_{f2})$

$v_2 = 0.01209 + 0.1208 \times (1.5492 - 0.01209)$

$v_2 = 0.01209 + 0.1208 \times .1.53711$

$v_2 = 0.01209 + 0.18568$

$v_2 = 0.19777\ ft^3/lbm$

Step 6 of 6

The final mass is given by

$$m = \frac{V}{v_2}$$

$$m = \frac{0.8}{0.19777}$$

$\boxed{m = 4.045\ lbm}$

Therefore final mass in the can is 4.045 lbm

Question: 8.062P

An electric windshield defroster is used to remove 0.25-in of ice from a windshield. The properties of the ice are *T* sat = 32°F, *uif* = *hif* = 144 Btu/lbm, and *V* = 0.01602 ft3 /lbm. Determine the electrical energy required per square foot of windshield surface area to melt this ice and remove it as liquid water at 32°F. What is the minimum temperature at which the defroster may be operated? Assume that no heat is transferred from the defroster or ice to the surroundings.

Step 1 of 7

Given

Thickness of ice is, $t = 0.25$ in

$= 0.0208$ ft

Saturation temperature is, $T_{sat} = 32°F$

Initial internal energy is, $u_{if} = 144$ Btu/lbm

Specific volume is, $v = 0.01602\ ft^3/lbm$

Temperature of liquid water is, $T_{min} = 32°F$

Step 2 of 7

Mass of ice is given by

$$m_i = \frac{V}{v}$$

Volume is given by product of thickness and area $V = tA$

$$m_i = \frac{tA}{v}$$

Step 3 of 7

According to conversation of mass principle,

$$\frac{dm_{cv}}{dt} = \sum_{in} m - \sum_{out} m$$

As no mass is coming in, $\sum_{in} m = 0$

$$\frac{dm_{cv}}{dt} = 0 - \sum_{out} m$$

$$\frac{dm_{cv}}{dt} = -m_{out} \qquad \ldots\ldots (1)$$

Step 4 of 7

Work done according to first law of thermodynamics is given by

$$W_{out} = -\frac{d(mu)_{cv}}{dt} - m_{out}h_{out}$$

Substituting equation (1) in the above equation we get

$$W_{out} = -\frac{d(mu)_{cv}}{dt} - \left[-\frac{dm_{cv}}{dt}\right]h_{out}$$

$$W_{out} = \left[\frac{dm_{cv}}{dt}\right]h_{out} - \frac{d(mu)_{cv}}{dt}$$

Multiplying above equation by 'dt' and integrating we get

$$W_{out} = h_{out} \times (-m)_i + (mu)_i$$

$$W_{out} = -h_{out} \times m_i + m_i u_i$$

Step 5 of 7

$$W_{out} = -h_{out} \times \frac{tA}{v} + \frac{tA}{v}u_i$$

$$W_{out} = \frac{tA}{v}(u_i - h_{out})$$

$$\frac{W_{out}}{A} = \frac{t}{v}(u_i - h_{out})$$

Step 6 of 7

As non flow process, $h_{out} = u_f$

$$\frac{W_{out}}{A} = \frac{t}{v}(u_i - u_f)$$

$$\frac{W_{out}}{A} = \frac{t}{v} \times (-u_{if})$$

$$\frac{W_{out}}{A} = \frac{0.0208}{0.01602} \times (-144)$$

$$\frac{W_{out}}{A} = 1.298 \times (-144)$$

$$\frac{W_{out}}{A} = -186.91\ Btu/ft^2$$

So the energy input should be

$$\boxed{W_{in} = 186.91\ Btu/ft^2}$$

Step 7 of 7

According to Clausius statement of second law of thermodynamics the temperature of the defroster cannot be lower than the temperature of the ice being melted. So,

$$\boxed{T_{min} = T_{min} = 32°F}$$

Therefore the energy input should be $186.91\ Btu/ft^2$ and the minimum temperature of the defroster should be 32°F .

Question: 8.063P

Consider two solid blocks, one hot and the other cold, brought into contact in an adiabatic container. After a while, thermal equilibrium is established in the container as a result of heat transfer. The first law requires that the amount of energy lost by the hot solid be equal to the amount of energy gained by the cold one. Does the second law require that the decrease in entropy of the hot solid be equal to the increase in entropy of the cold one?

Step 1 of 1

According to first law, amount of energy lost by hot object is the amount of energy gained by cold object and hence it is called as law of conservation of energy.

But there is no principle like conservation of entropy therefore entropy is not at all a conserved property.

Hence the decrease in entropy of hot solid cannot be equal to total entropy change for a process.

Question: 8.064P

A 75-kg copper block initially at 110°C is dropped into an insulated tank that contains 160 L of water at 15°C. Determine the final equilibrium temperature and the total entropy change for this process.

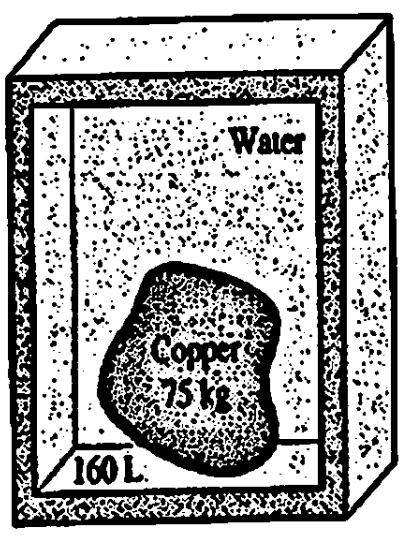


FIGURE P8–64

Step 1 of 6

From properties of common liquids solids table,

The specific heat of water at $25°C$, $c_{pw} = 4.18\ kJ/kg \cdot K$

Density of water at $25°C$, $\rho = 997\ kg/m^3$

The specific heat of copper at $27°C$, $c_{pcu} = 0.386\ kJ/kg \cdot K$

Calculate the mass of water,

$$m_w = \rho \times V$$

Here,

Volume of water in the tank is, V

Density of water in the tank, ρ

Substitute $997\ kg/m^3$ for ρ, $160\ L$ or $0.16\ m^3$ for V,

$$m_w = 997 \times 0.16$$
$$= 159.52\ kg$$

Step 2 of 6

As tank is closed, the internal energy will be constant.

$$\Delta U = 0$$
$$\Delta U_{cu} + \Delta U_w = 0$$
$$m_{cu} \times c_{pcu} \times (T_2 - T_{1cu}) + m_w \times c_{pw} \times (T_2 - T_{1w}) = 0$$

Here,

Mass of copper block, m_{cu}

Initial temperature is, T_{1cu}

Final temperature is, T_{1w}

Final equilibrium temperature, T_2

Step 3 of 6

Substitute $0.386\ kJ/kg \cdot K$ for c_{pcu}, $159.52\ kg$ for m_w, $4.18\ kJ/kg \cdot K$ for c_{pw}, $5\ kg$ for m_{cu},

$110°C$ or $383\ K$ for T_{1cu}, $15°C$ or $288\ K$ for T_{1w}, in the above equation,

$$75 \times 0.386 \times (T_2 - 383) + 159.52 \times 4.18 \times (T_2 - 288) = 0$$
$$28.95 \times (T_2 - 383) + 666.7936 \times (T_2 - 288) = 0$$
$$28.95 \times (T_2 - 383) = 666.7936 \times (288 - T_2)$$
$$28.95\ T_2 - 11087.85 = 192036.5568 - 666.7936\ T_2$$
$$28.95\ T_2 + 666.7936\ T_2 = 192036.5568 + 11087.85$$
$$695.7436\ T_2 = 203124.4068$$
$$T_2 = \frac{203124.4068}{695.7436}$$
$$= 291.95\ K$$
$$= 18.95°C$$

Therefore the final equilibrium temperature is $\boxed{T_2 = 18.95°C}$

Step 4 of 6

Calculate the change in entropy of copper block,

$$\Delta S_{cu} = m_{cu} \times c_{pcu} \times \ln\left(\frac{T_2}{T_{1cu}}\right)$$

Substitute $0.386\ kJ/kg \cdot K$ for c_{pcu}, $5\ kg$ for m_{cu}, $18.95°C$ for T_2, $110°C$ or $383\ K$ for T_{1cu} in the above equation,

$$\Delta S_{cu} = 75 \times 0.386 \times \ln\left(\frac{291.95}{383}\right)$$
$$= 75 \times 0.386 \times \ln(0.7623)$$
$$= 75 \times 0.386 \times (-0.2714)$$
$$= -7.85703\ kJ/K$$

Step 5 of 6

Calculate the change in entropy of water is given by

$$\Delta S_w = m_w \times c_{pw} \times \ln\left(\frac{T_2}{T_{1w}}\right)$$

Substitute $159.52\ kg$ for m_w, $4.18\ kJ/kg \cdot K$ for c_{pw}, $15°C$ or $288\ K$ for T_{1w}, $18.95°C$ for T_2 in the above equation,

$$\Delta S_w = 159.52 \times 4.18 \times \ln\left(\frac{291.95}{288}\right)$$
$$= 159.52 \times 4.18 \times \ln(1.0137)$$
$$= 159.52 \times 4.18 \times 0.0136$$
$$= 9.0684\ kJ/K$$

Step 6 of 6

The total change in entropy is given by

$$\Delta S_{total} = \Delta s_w + \Delta s_{cu}$$

Substitute $9.0684\ kJ/K$ for Δs_w, $-7.85703\ kJ/K$ for Δs_{cu},

$$\Delta S_{total} = 9.0684 - 7.85703$$
$$= 1.21137\ kJ/K$$

Therefore, the total change in entropy is $\boxed{1.21137\ kJ/K}$

Question: 8.065P

Ten grams of computer chips with a specific heat of 0.3 kJ/kg·K are initially at 20°C. These chips are cooled by placement in 5 grams of saturated liquid R-134a at -40°C. Presuming that the pressure remains constant while the chips are being cooled, determine the entropy change of (*a*) the chips, (*b*) the R-134a, and (*c*) the entire system. Is this process possible? Why?

Step 1 of 5

(a)

Write the equation for the change of entropy in the chips.

$$\Delta S_{chips} = m_c c \ln\left(\frac{T_2}{T_1}\right)$$

Here, ΔS_{chips} is the change in entropy in the chips, m_c is the mass of the chips, c is the specific heat of chips, T_2 is temperature at state (2), and T_1 is temperature at state (1).

Substitute $10\text{gm} = 0.01\text{kg}$ for m_c, $0.3\ \text{KJ/kgK}$ for c, $20°\text{C}=293\text{K}$ for T_1, and $-40°\text{C}=233\text{K}$ for T_2.

$$\Delta S_{chips} = 0.01\times 0.3\times \ln\left(\frac{233}{293}\right)$$
$$= -6.874\times 10^{-4}\ \text{kJ/K}$$

Thus, the change of entropy in the chips, $\boxed{\Delta S_{chips} = -6.874\times 10^{-4}\ \text{kJ/K}}$.

Step 2 of 5

(b)

Calculate the heat transferred from the chips to saturated liquid.

$$Q_{chips} = m_c c(T_1 - T_2)$$

Here, Q_{chips} is the heat transferred from the chips to saturated liquid.

Substitute $10\text{gm} = 0.01\text{kg}$ for m_c, $0.3\ \text{KJ/kgK}$ for c, $20°\text{C}=293\text{K}$ for T_1, and $-40°\text{C}=233\text{K}$ for T_2.

$$Q_{chips} = 0.01\times 0.3(293-233)$$
$$= 0.18\ \text{kJ}$$

Step 3 of 5

When the chips are kept in saturated liquid "R-134a," some of the liquid is evaporated due to this heat transfer and remaining liquid is in saturated state.

From the properties of R-134a at temperature $T_2 = -40°\text{C}$

Specific internal energy during evaporation, $u_{fg_2} = 207.37\ \text{kJ/kg}$

Heat transfer during the vaporization, $Q_{\text{R-134a}} = m_{g_2} u_{fg_2}$

Consider heat balance equation and calculate the mass of liquid is evaporated.

$$Q_{\text{R-134a}} = Q_{chips}$$
$$Q_{\text{R-134a}} = 0.18\ \text{kJ}$$
$$0.18 = m_{g_2}\times 207.4$$
$$m_{g_2} = \frac{0.18}{207.4}$$
$$= 8.678\times 10^{-4}\ \text{kg}$$

Mass of the saturated vapor, $m_{g_2} = 8.678\times 10^{-4}\ \text{kg}$

Step 4 of 5

Calculate the mass of the saturated liquid after heat transfer by considering law of conservation of mass.

$$m_{f_2} = m - m_{g_2}$$

Substitute $0.005\ \text{kg}$ for m and $8.678\times 10^{-4}\ \text{kg}$ for m_{g_2}.

$$m_{f_2} = 0.005 - 8.6788\times 10^{-4}$$
$$= 4.132\times 10^{-3}\ \text{kg}$$

Step 5 of 5

Calculate the change of entropy of R-134a,

$$\Delta S_{\text{R-134a}} = \left(m_{g_2} s_{g_2} + m_{f_2} s_{f_2}\right) - m_{f_1} s_{f_1} \quad \text{..... (1)}$$

Obtain the properties from the table "properties of R-134a," recorded at temperature, $T_2 = -40°\text{C}$.

Specific entropy at saturated liquid, $s_{f_2} = s_{f_1} = 0\ \text{kJ/kg·K}$

Specific entropy during evaporation, $s_{fg_2} = 0.96866\ \text{kJ/kg·K}$

Specific entropy at saturated vapor, $s_{g_2} = 0.96866\ \text{kJ/kg·K}$

Specific entropy at saturated liquid, $s_{f_1} = 0\ \text{kJ/kg·K}$

Substitute $4.132\times 10^{-3}\ \text{kg}$ for m_{f_2}, $8.678\times 10^{-4}\ \text{kg}$ for m_{g_2}, and 0.005 for m in equation (1).

$$\Delta S_{\text{R-134a}} = 8.768\times 10^{-4}\times 0.96866 + 4.132\times 10^{-3}\times 0 - 0.005\times 0$$
$$= 8.768\times 10^{-4}\times 0.96866$$
$$= 8.4932\times 10^{-4}\ \text{kJ/kg·K}$$

Thus, the change of entropy of R-134a, $\boxed{\Delta S_{\text{R-134a}} = 8.4932\times 10^{-4}\ \text{kJ/K}}$

(c)

Calculate the total change of entropy,

$$\Delta S = \Delta S_{chips} + \Delta S_{\text{R-134a}}$$

Substitute $8.4932\times 10^{-4}\ \text{kJ/K}$ for $\Delta S_{\text{R-134a}}$ and $-6.874\times 10^{-4}\ \text{kJ/K}$ for ΔS_{chips}.

$$\Delta S = -6.874\times 10^{-4} + 8.4932\times 10^{-4}$$
$$= 1.619\times 10^{-4}\ \text{kJ/kg·K}$$

Thus, total change of entropy is $\boxed{\Delta S = 1.619\times 10^{-4}\ \text{kJ/kg·K}}$.

The process is possible since the total change of entropy value is positive.

Question: 8.066P

A 25-kg iron block initially at 350°C is quenched inan insulated tank that contains 100 kg of water at 18°C. Assuming the water that vaporizes during the process condenses back in the tank, determine the total entropy change j during this process.

Step 1 of 4

Given

Mass of water $m_w = 100$ kg

Temperature of water $T_w = 18°C$

Mass of Iron $m_I = 25$ kg

Temperature of iron $T_I = 350°C$

Step 2 of 4

Specific heat of water at room temperature,

$$c_p = 4.18 \text{ kJ/kg°C}$$

Specific heat of iron at room temperature

$$c_p = 0.45 \text{ kJ/kg°C}$$

From first law of thermodynamics, for closed system

Net energy transfer = change in internal energy.

$$Q - w = \Delta U$$

$$\Delta U = 0$$

$$(\Delta U)_{\text{water}} + (\Delta U)_{\text{Iron}} = 0$$

$$m_w c_p (T_2 - T_w) + m_I c_p (T_2 - T_I) = 0$$

Substituting the values,

$$[100 \times 4.18(T_2 - 18)] + [25 \times 0.45(T_2 - 350)] = 0$$

$$418(T_2 - 18) + (11.25)(T_2 - 350) = 0$$

$$(418 + 11.25)T_2 - (418 \times 18) + (11.25 \times 350) = 0$$

$$419.25 \times T_2 = 11461.5$$

$$T_2 = 26.7°C$$

Step 3 of 4

Entropy change of Iron

$$\Delta s_I = m_I c_{\text{avg}} \ln\left(\frac{T_2}{T_I}\right)$$

$$\Delta s_I = (25 \times 0.45) \times \ln\left(\frac{26.7 + 273}{350 + 273}\right)$$

$$\Delta s_I = -8.23 \text{ kJ/kg}$$

Entropy change of water

$$\Delta s_{\text{water}} = m_w c_{\text{avg}} \ln\left(\frac{T_2}{T_w}\right)$$

$$\Delta s_{\text{water}} = (100 \times 4.18) \times \ln\left(\frac{26.7 + 273}{18 + 273}\right)$$

$$\Delta s_{\text{water}} = 418 \ln\left(\frac{299.7}{291}\right)$$

$$\Delta s_{\text{water}} = 12.31 \text{ kJ/kg}$$

Step 4 of 4

Total entropy change during this process is

$$\Delta s_{\text{total}} = \Delta s_{\text{iron}} + \Delta s_{\text{water}}$$

$$\Delta s_{\text{total}} = -8.232 + 12.314$$

$$\Delta s_{\text{total}} = 4.08 \text{ kJ/kg}$$

Hence total entropy change is $\boxed{\Delta s_{\text{total}} = 4.08 \text{ kJ/kg}}$

Question: 8.067P

A 30-kg aluminum block initially at 140°C is broughtinto contact with a 40-kg block of iron at 60°C in an insulated enclosure. Determine the final equilibrium temperature and the total entropy change for this process.

Step 1 of 3

Obtain the properties of specific heat from "properties of common liquids, solids and foods" at temperature 400 K.

Specific heat of aluminum, $c_{Al} = 0.949\ \text{kJ/kg-K}$

Specific heat of iron, $c_{Fe} = 0.45\ \text{kJ/kg-K}$

Apply energy balance to the given system.

$$E_{in} - E_{out} = \Delta E_{sys}$$

Since there is no amount of energy entering or leaving from the system, $E_{in} = E_{out} = 0$.

$$E_{in} - E_{out} = \Delta E_{sys}$$

$$0 = \Delta U$$

$$\Delta U_{Al} + \Delta U_{Fe} = 0$$

$$\left\{m_{Al} c_{Al} \left(T_{Al,2} - T_{Al,1}\right)\right\} + \left\{m_{Fe} c_{Fe} \left(T_{Fe,2} - T_{Fe,1}\right)\right\} = 0 \qquad \text{..... (1)}$$

Here, m_{Al} is the mass of Aluminum block, m_{Fe} is the mass of iron, $T_{Al,2}$ is the final temperature of aluminum block, $T_{Al,1}$ is the initial temperature of aluminum block, $T_{Fe,2}$ is the final temperature of iron, and $T_{Fe,1}$ is the initial temperature of iron.

Step 2 of 3

The final temperature is same for both the blocks.

$$T_{Al,2} = T_{Fe,2} = T_2$$

Then, rewrite the equation (1) as follows:

$$\left\{m_{Al} c_{Al} \left(T_2 - T_{Al,1}\right)\right\} + \left\{m_{Fe} c_{Fe} \left(T_2 - T_{Fe,1}\right)\right\} = 0$$

Substitute 30 kg for m_{Al}, 40 kg for m_{Fe}, 140°C for $T_{Al,1}$, and 60°C for $T_{Fe,1}$.

$$\left\{30 \times 0.949 \times (T_2 - 140)\right\} + \left\{40 \times 0.45 \times (T_2 - 60)\right\} = 0$$

$$\left\{28.47(T_2 - 140)\right\} + \left\{18(T_2 - 60)\right\} = 0$$

$$46.47 T_2 - 5065.8 = 0$$

$$T_2 = 109.012°\text{C}$$

Therefore, final equilibrium temperature of aluminum and iron block is $\boxed{109.012°\text{C}}$.

Step 3 of 3

Find the entropy change in iron block by using the following equation:

$$\Delta S_{Fe} = m_{Fe} c_{Fe} \ln\left(\frac{T_2}{T_{Fe,1}}\right)$$

$$= 40 \times 0.45 \times \ln\left(\frac{109.012 + 273}{60 + 273}\right)$$

$$= 40.45 \times 0.1373$$

$$= 2.4714\ \text{kJ/K}$$

Similarly, find the entropy change in aluminum block by using the following equation:

$$\Delta S_{Al} = m_{Al} c_{p,Al} \ln\left(\frac{T_2}{T_{Al,1}}\right)$$

$$= 30 \times 0.949 \times \ln\left(\frac{109.012 + 273}{140 + 273}\right)$$

$$= 28.47 \times (-0.0779)$$

$$= -2.2178\ \text{kJ/K}$$

Then find the total entropy change by using the following equation:

$$\Delta S_{total} = \Delta S_{Al} + \Delta S_{Fe}$$

$$= -2.2178 + 2.4714$$

$$= 0.253\ \text{kJ/K}$$

Therefore, the total entropy change during the process is $\boxed{0.253\ \text{kJ/K}}$.

Question: 8.068P pg 1

Reconsider Prob. 8-67. Using EES (or other) software, study the effect of the mass of the iron block on the final equilibrium temperature and the total I entropy change for the process. Let the mass of the iron vary I from 10 to 100 kg. Plot the equilibrium temperature and the total entropy change as a function of iron mass, and discuss 5 the results.

Step 1 of 10

Given data:

Mass of aluminum block is, $m_a = 30\ \text{kg}$

Initial temperature of aluminum block is, $T_{a1} = 140\ ^\circ\text{C} = 413\ \text{K}$

Mass of iron block varies, $m_{fe} = 10$ to $100\ \text{kg}$

Initial temperature of iron block is, $T_{fe1} = 60\ ^\circ\text{C} = 333\ \text{K}$

Step 2 of 10

Give all the inputs in the EES software as given below

T1_fe =60 [C]

T1_al =140 [C]

P1 =140 [kPa]

m_al =30 [kg]

c_al =0.949 [kJ/kg-K]

c_fe =0.45 [kJ/kg-K]

Step 3 of 10

For calculating other parameters we should give other equations as

T2_fe =T2

T2_al =T2

Ein – Eout =Esys

Ein=0

Eout =0

Esys =m_fe*c_fe*(T2_fe – T1_fe)+m_al*c_al*(T2_al – T1_al)

Sdif_fe =m_fe*c_fe*ln((T2_fe+273)/(T1_fe+273))

Sdif_al =m_al*c_al*ln((T2_al+273)/(T1_al+273))

Sdif_tot =Sdif_fe+Sdif_al

Step 4 of 10

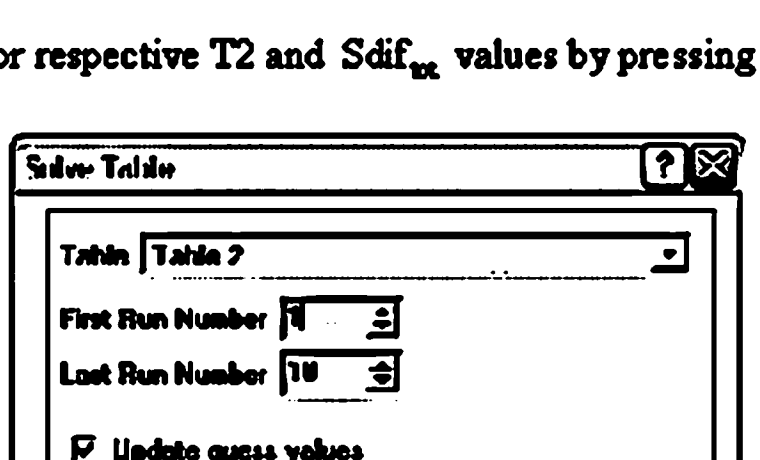
After writing these sets of equations, we should create a parametric table from Tables → new parametric

Fill all the values of m_{fe}.

▷ 1..10	1 m_{fe}	2 I2	3 $Sdif_{tot}$
Run 1	10		
Run 2	20		
Run 3	30		
Run 4	40		
Run 5	50		
Run 6	60		
Run 7	70		
Run 8	80		
Run 9	90		
Run 10	100		

Step 5 of 10

Now solve the table for respective T2 and $Sdif_{tot}$ values by pressing F3.



Step 6 of 10

Then we get

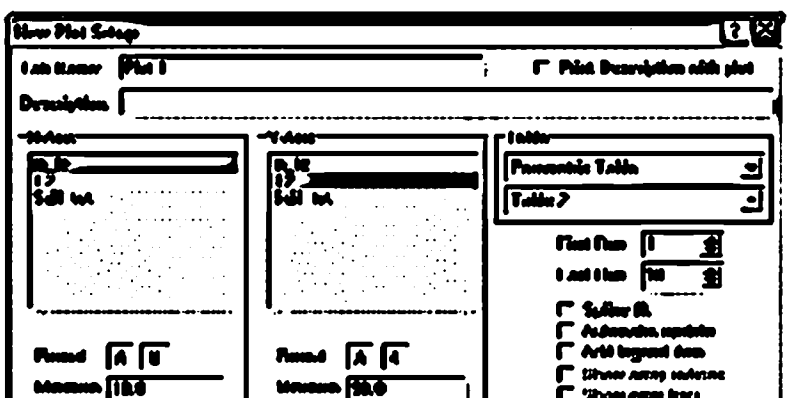
▷ 1..10	1 m_{fe}	2 T2 [C]	3 $Sdif_{tot}$ [kJ/K]
Run 1	10	129.1	0.08547
Run 2	20	120.8	0.1525
Run 3	30	114.3	0.2066
Run 4	40	109	0.2511
Run 5	50	104.7	0.2883
Run 6	60	101.1	0.32
Run 7	70	97.98	0.3472
Run 8	80	95.33	0.3709
Run 9	90	93.02	0.3916
Run 10	100	91	0.41

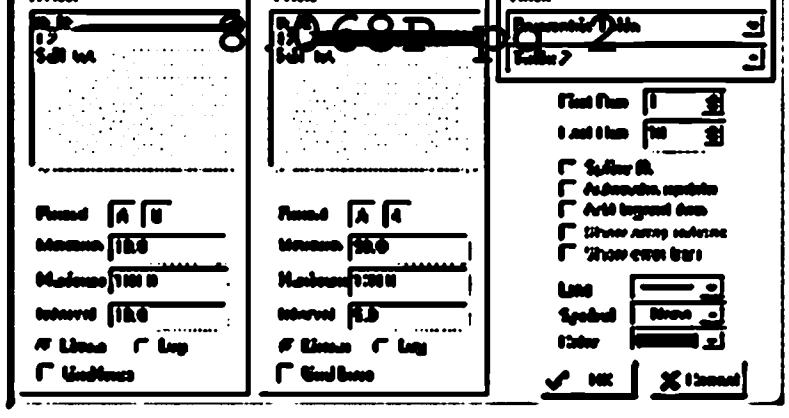
Step 7 of 10

For getting the plot we should go for

Plots → New plot window

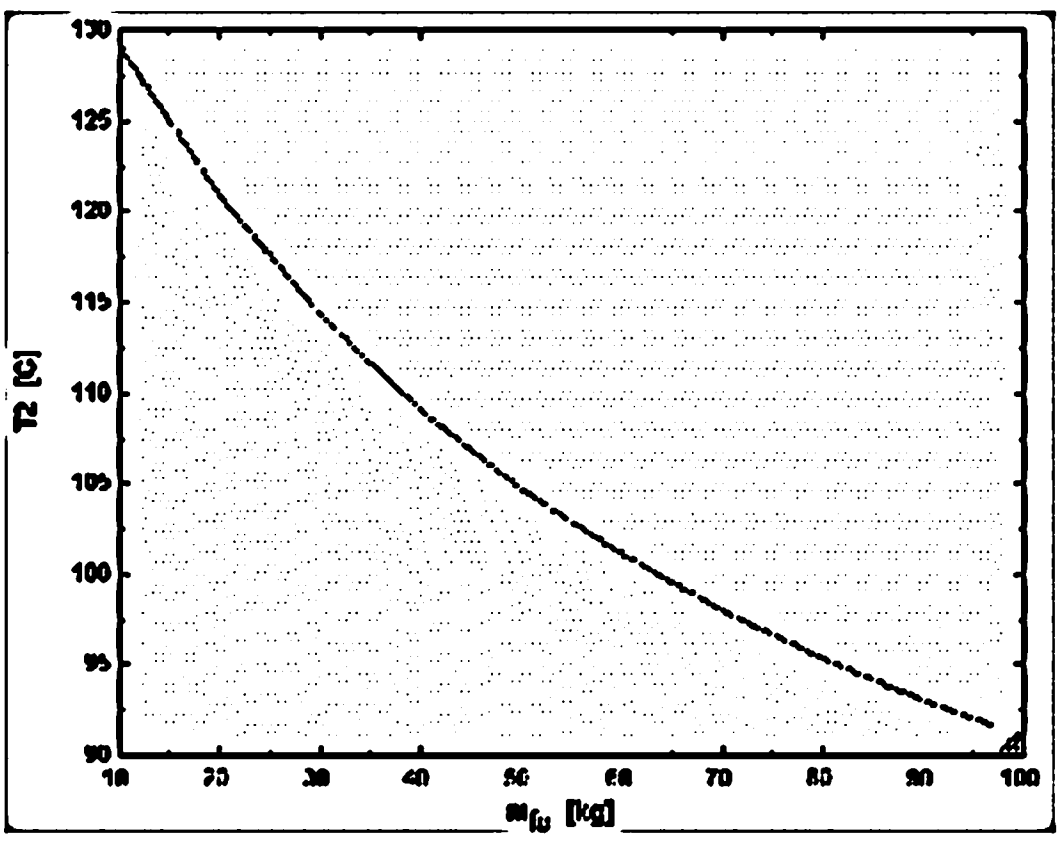
We should fill all the required fields by taking m_{fe} in x-axis and T2 in y-axis.





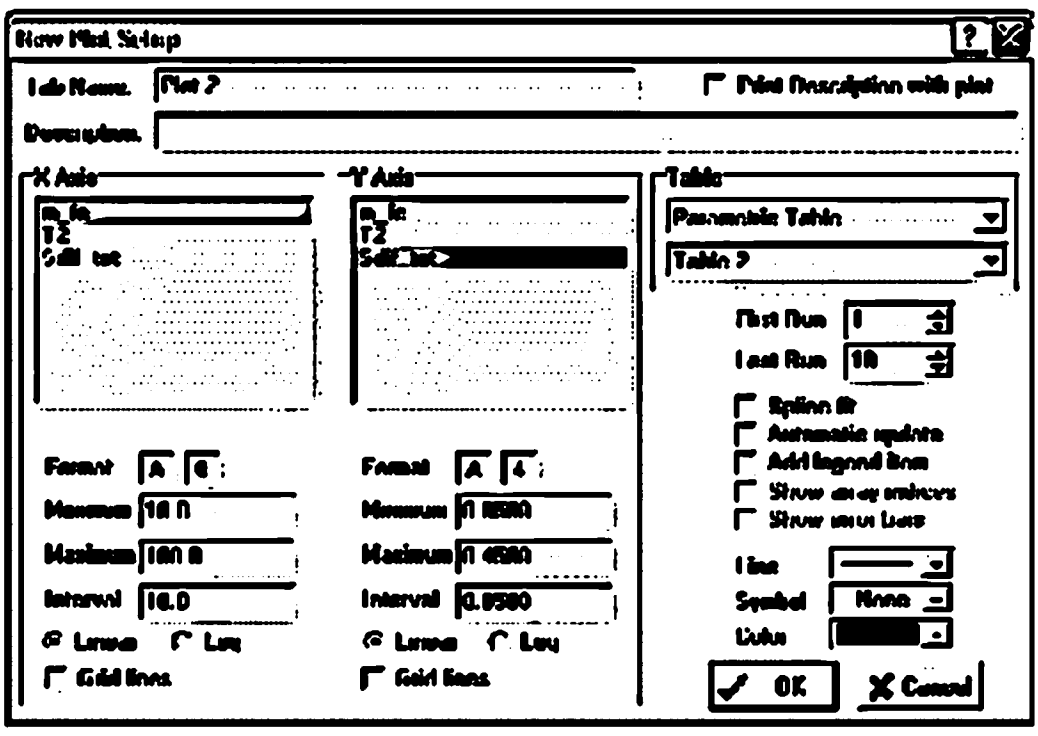
Step 8 of 10

After that we get the plot as



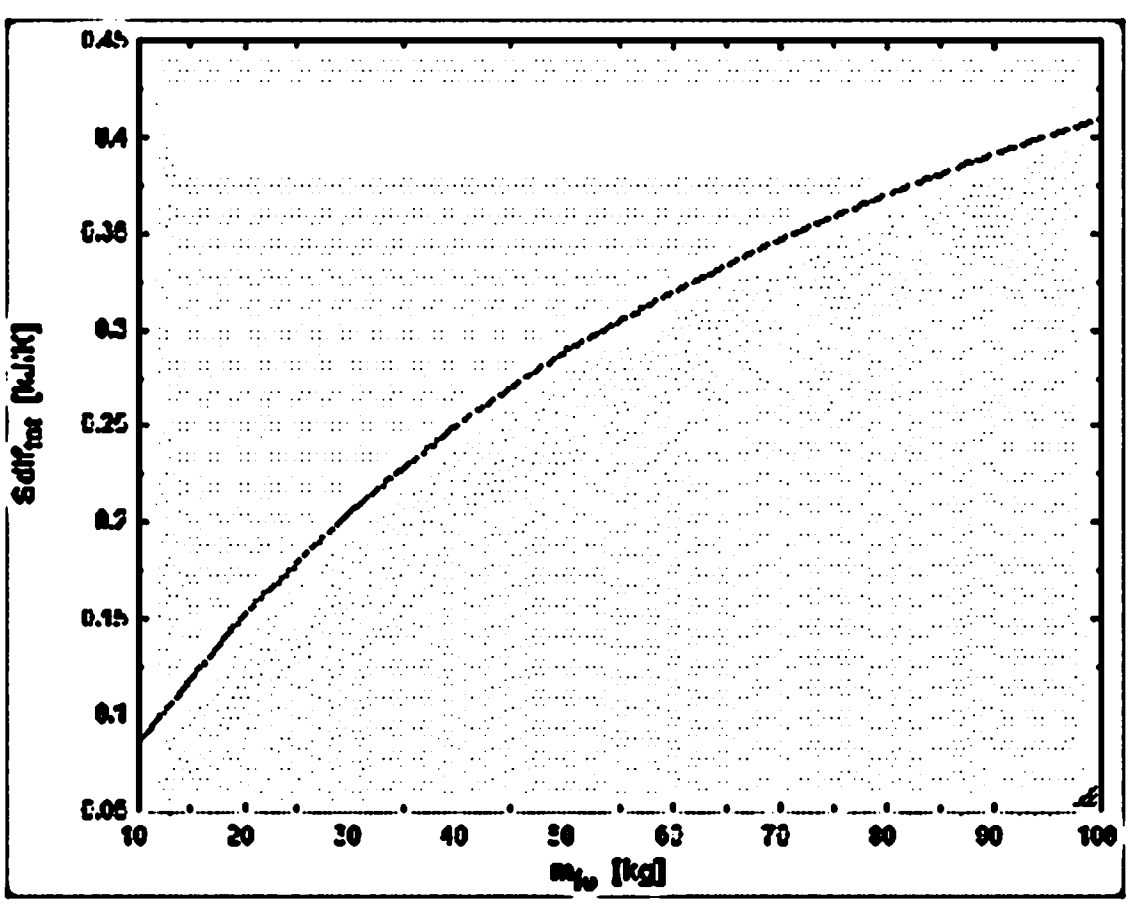
Step 9 of 10

For getting the plot we should go for
Plots → New plot window
We should fill all the required fields by taking m_{fu} in x-axis and $Sdif_{tot}$ in y-axis.



Step 10 of 10

After that we get the plot as



Question: 8.069P

A 50-kg iron block and a 20-kg copper block, both initially at 80°C, are dropped into a large lake at 15°C. Thermal equilibrium is established after a while as a result of heat transfer between the blocks and the lake water. Determine the total entropy change for this process.

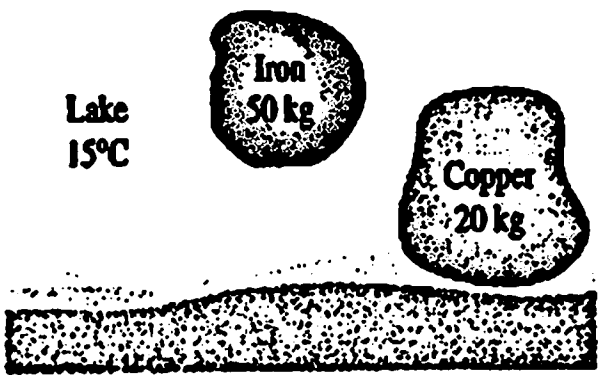


FIGURE P8–69

Step 1 of 4

Given data

Mass of Iron block $m_I = 50$ kg

Mass of copper block $m_{cu} = 20$ kg

Temperature of Iron block $T_I = 80°C$

Temperature of copper block $T_{cu} = 80°C$

Temperature of lake water $T_l = 15°C$

Step 2 of 4

Specific heat of iron at room temperature, $c_p = 0.45$ kJ/kgK

Specific heat of copper at room temperature, $c_p = 0.386$ kJ/kgK

Entropy change of iron during this process

$$\Delta s_{iron} = m_I c_{avg} \times \ln\left(\frac{T_l}{T_I}\right)$$

$$= (50\text{ kg})(0.45\text{kJ/kgK})\ln\left(\frac{15+273}{80+273}\right)$$

$$= -4.579\text{ kJ/K}$$

Entropy change of copper during this process

$$\Delta s_{copper} = m_{cu} c_{avg} \ln\left(\frac{T_l}{T_{cu}}\right)$$

$$= (20\text{ kg})(0.386\text{ kJ/kgK}) \times \ln\left(\frac{15+273}{80+273}\right)$$

$$= -1.571\text{ kJ/K}$$

Step 3 of 4

Energy balance from first law of thermodynamics

Energy in − energy out = change in internal energy

$$Q_{in} - Q_{out} = \Delta U$$

$$0 - Q_{out} = \Delta U$$

$$-Q_{out} = \Delta U_{iron} + \Delta U_{copper}$$

$$Q_{out} = m_I c_{avg}(T_I - T_l) + m_{cu} c_{avg}(T_{cu} - T_l)$$

$$Q_{out} = [50 \times 0.45(80-15)] + [20 \times 0.386 \times (80-15)]$$

$$Q_{out} = 1462.5 + 501.8$$

$$Q_{out} = 1964.3\text{ kJ}$$

Step 4 of 4

Entropy change of lake water, $\Delta s_{lake} = \dfrac{Q_{lake}}{T_{lake}}$

$$\Delta s_{lake} = \frac{Q_{out}}{T_{lake}}$$

$$\Delta s_{lake} = \frac{1964.3}{15+273}$$

$$\Delta s_{lake} = 6.8205\text{ kJ/K}$$

Total entropy change for this process,

$$\Delta s_{total} = \Delta s_{iron} + \Delta s_{copper} + \Delta s_{lake}$$

$$\Delta s_{total} = -4.579 - 1.571 + 6.8205$$

$$\Delta s_{total} = 0.6705\text{ kJ/K}$$

Hence total entropy change for the process is $\boxed{\Delta s_{total} = 0.6705\text{ kJ/K}}$

Question: 8.070P pg 1

An adiabatic pump is to be used to compress saturated liquid water at 10 kPa to a pressure to 15 MPa in a reversible manner. Determine the work input using (*a*) entropy data from the compressed liquid table, (*b*) inlet specific volume and pressure values, (*c*) average specific volume and pressure values. Also, determine the errors involved in parts (*b*) and (*c*).

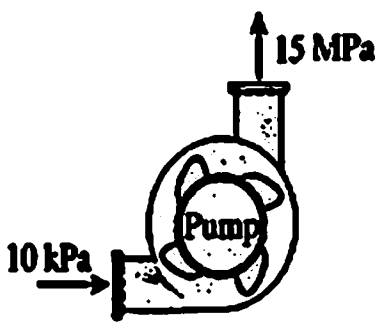


FIGURE P8–70

Step 1 of 8

Obtain the properties of saturated liquid water at pressure 10 kPa from the table A.5, "Saturated water-Pressure table".

Specific enthalpy, $h_1 = 191.8 \text{ kJ/kg}$

Specific entropy, $s_1 = 0.6492 \text{ kJ/kgK}$

Specific volume, $v_1 = 0.00101 \text{ m}^3/\text{kg}$

As the process is isentropic, $s_2 = s_1$

$s_2 = s_1 = 0.649 \text{ kJ/kgK}$

Step 2 of 8

Obtain the Specific enthalpy values from the table A.7 , "Compressed liquid water" at the pressure 15 MPa.

$h = 180.77 \text{ kJ/kg}$ at $s = 0.5666 \text{ kJ/kgK}$

$h = 263.74 \text{ kJ/kg}$ at $s = 0.8234 \text{ kJ/kgK}$

Calculate the enthalpy at state 2 using the interpolation,

$$h_2 = h_{at\ s=0.5666} + \left(\frac{h_{at\ s=0.8234} - h_{at\ s=0.5666}}{0.8234 - 0.5666}\right)(0.649 - 0.5666)$$

$$h_2 = 180.77 + \left(\frac{263.74 - 180.77}{0.8234 - 0.5666}\right)(0.649 - 0.5666)$$

$$h_2 = 180.77 + \left(\frac{82.97}{0.2568}\right)(0.0824)$$

$$h_2 = 207.323 \text{ kJ/kg}$$

Step 3 of 8

Obtain the Specific volume values from the table A.7 , "Compressed liquid water" at the pressure 15 MPa.

$v = 0.0010013 \text{ m}^3/\text{kg}$ for $s = 0.5666 \text{ kJ/kgK}$

$v = 0.0010105 \text{ m}^3/\text{kg}$ for $s = 0.8234 \text{ kJ/kgK}$

Calculate the specific volume at state 2 using the interpolation,

$$v_2 = v_{at\ s=0.5666} + \left(\frac{v_{at\ s=0.8234} - v_{at\ s=0.5666}}{0.8234 - 0.5666}\right)(0.649 - 0.5666)$$

$$v_2 = 0.0010013 + \left(\frac{0.0010105 - 0.0010013}{0.8234 - 0.5666}\right)(0.649 - 0.5666)$$

$$v_2 = 0.0010013 + \left(\frac{9.2 \times 10^{-6}}{0.2568}\right)(0.0824)$$

$$v_2 = 0.0010042 \text{ m}^3/\text{kg}$$

Step 4 of 8

(a)

Determine the work input to the pump using the data from the compressed liquid water tables.

$w_{pump} = h_2 - h_1$

Substitute 207.323 kJ/kg for h_2 , and 191.8 kJ/kg for h_1 .

$w_{pump} = 207.323 - 191.8$

$w_{pump} = 15.523 \text{ kJ/kg}$

Hence work input to the pump is, $\boxed{w_{pump} = 15.513 \text{ kJ/kg}}$.

Step 5 of 8

(b)

Determine the work input to the compressor using inlet specific volume and pressure values.

$w_{pump} = v_1(P_2 - P_1)$

Substitute $0.001010 \text{ m}^3/\text{kg}$ for v_1 , 15000 kPa for P_2 , and 10 kPa for P_1 .

$w_{pump} = 0.00101 \times (15{,}000 - 10)$

$w_{pump} = 15.14 \text{ kJ/kg}$

Hence, the work input to the pump is, $\boxed{15.14 \text{ kJ/kg}}$.

Step 6 of 8

Hence, the work input to the pump is, 15.14 kJ/kg . pg 2

Step 6 of 8

(c)

Calculate the average specific volume.

$$v_{avg} = \frac{v_1 + v_2}{2}$$

Substitute $0.001010\,m^3/kg$ for v_1 , and $0.001004\,m^3/kg$ for v_2 .

$$v_{avg} = \frac{0.001010 + 0.001004}{2}$$
$$= 0.001007\,m^3/kg$$

Step 7 of 8

Determine the work input to the compressor using average specific volume and pressure values.

$$w_{pump} = v_{avg}(P_2 - P_1)$$

Substitute $0.001007\,m^3/kg$ for v_{avg} , 15000 kPa for P_2 , and $10\,kPa$ for P_1 .

$$w_{pump} = 0.001007(15{,}000 - 10)$$

$$w_{pump} = 15.096\ kJ/kg$$

Hence, the work input to the pump is, **15.096 kJ/kg** .

Step 8 of 8

Determine the error involved in part (b).

$$\%Error$$

$$\%Error = \left(\frac{15.14 - 15.513}{15.14}\right) \times 100$$

$$\%Error = 2.13\%$$

Hence, the error involved in part (b) is, **2.13%** .

Determine the error involved in part (c).

$$\%Error = \left(\frac{15.096 - 15.09}{15.096}\right) \times 100$$

$$\%Error = 0.04\%$$

Hence, the error involved in part (b) is, **0.04%** .

Question: 8.071P

Some properties of ideal gases such as internal energy and enthalpy vary with temperature only [that is, $u = u(T)$ and $h = h(T)$]. Is this also the case for entropy?

Step 1 of 1

Unlike internal energy and enthalpy, the entropy of an ideal gas varies with specific volume or pressure as well as the temperature.

The variation of entropy with pressure is accounted for by the last term in the following equation.

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

Question: 8.072P

Can the entropy of an ideal gas change during an isothermal process?

Step 1 of 1

The entropy of an ideal gas changes during isothermal process.
The entropy of an ideal gas depends on pressure as well as the temperature. So during isothermal process, the entropy of ideal varies with pressure as follows.

$$s_2 - s_1 = -R \ln\left(\frac{P_2}{P_1}\right)$$

Question: 8.073P

An ideal gas undergoes a process between two specified temperatures, first at constant pressure and then at constant volume. For which case will the ideal gas experience a larger entropy change? Explain.

Step 1 of 2

At constant pressure $P_1 = P_2$,

Entropy change, $s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_1}{P_1}\right)$

$$\Rightarrow s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) \text{---(1)}$$

At constant volume $v_1 = v_2$,

Entropy change, $s_2 - s_1 = c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{v_1}{v_1}\right)$

$$\Rightarrow s_2 - s_1 = c_v \ln\left(\frac{T_2}{T_1}\right) \text{----(2)}$$

Step 2 of 2

From (1) and (2), Entropy change for these two cases varies with C_p and C_v.

But, C_p is greater than C_v. So, Entropy change will be larger at constant pressure case.

Question: 8.074P

Prove that the two relations for entropy change of ideal gases under the constant-specific-heat assumption (Eqs. 8-33 and 8-34) are equivalent.

Step 1 of 2

Given:

$$s_2 - s_1 = c_{v,avg} \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{v_2}{v_1}\right) \text{--------(1)}$$

$$\Rightarrow s_2 - s_1 = c_{p,avg} \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \text{--------(2)}$$

We need to prove that these two relations for entropy change of ideal under the constant-specific-heat assumptions are equivalent.

Step 2 of 2

For ideal gas, $\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1}$

$$\Rightarrow \frac{v_2}{v_1} = \frac{T_2 P_1}{T_1 P_2} \text{---------(3)}$$

And $c_{p,avg} - c_{v,avg} = R$ --------(4)

From (1), (2), and (3),

$$\Rightarrow s_2 - s_1 = c_{v,avg} \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{T_2 P_1}{T_1 P_2}\right)$$

$$\Rightarrow s_2 - s_1 = c_{v,avg} \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

$$\Rightarrow s_2 - s_1 = \left(c_{v,avg} + R\right) \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

$$\Rightarrow s_2 - s_1 = c_{p,avg} \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$, here we get equation (4)

So, these two relations for entropy change of ideal under the constant-specific-heat assumptions are equivalent.

Question: 8.075P

Starting with the second $T\,ds$ relation (Eq. 8-26), obtain Eq. 8-34 for the entropy change of ideal gases under the constant-specific-heat assumption.

Step 1 of 2

Given:

The differential change of entropy $ds = \frac{dh}{T} - \frac{v\,dP}{T}$ -------(1)

The entropy change under the constant specific heat,

$$\Rightarrow s_2 - s_1 = c_{p,avg} \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \text{-------(2)}$$

Step 2 of 2

For ideal gas, $P \times v = R \times T$

$$\Rightarrow v = \frac{R \times T}{P} \text{-----(3)}$$

And $dh = C_p \times dT$ -------(4)

From (1), (3), and (4),

$$\Rightarrow ds = \frac{dh}{T} - \frac{v\,dP}{T}$$

$$\Rightarrow ds = c_p \frac{dT}{T} - R\frac{dP}{P}$$

On integrating both sides,

$$\Rightarrow [s]_{s_1}^{s_2} = c_p [\ln T]_{T_1}^{T_2} - R[\ln P]_{P_1}^{P_2}$$

$$\Rightarrow \boxed{s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)}$$, here we get equation (2)

Question: 8.076P

Starting with Eq. 8-34, obtain Eq. 8-43.

Step 1 of 2

Given:

The entropy change under the constant specific heat,

$$s_2 - s_1 = c_{p,avg} \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \text{-------(1)}$$

The first isentropic relation for ideal gases

$$\left(\frac{T_2}{T_1}\right)_{s=\text{constant}} = \left(\frac{P_2}{P_1}\right)^{\frac{(k-1)}{k}} \text{--------(2)}$$

Step 2 of 2

For ideal gas, $C_p - C_v = R$ -------(3)

$$\text{And } k = \frac{c_p}{c_v} \text{-----(4)}$$

From (1), (3), and (4), and at constant entropy

$$0 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

$$\Rightarrow \ln\left(\frac{T_2}{T_1}\right) = \frac{R}{c_p}\ln\left(\frac{P_2}{P_1}\right)$$

$$\Rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{R}{c_p}}$$

$$\Rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{c_p - c_v}{c_p}}$$

$$\Rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\Rightarrow \boxed{\left(\frac{T_2}{T_1}\right)_{s=\text{constant}} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}}$$, here we get equation (2)

Question: 8.077P

Which of the two gases—helium or nitrogen—experiences the greatest entropy change as its state is changed from 2000 kPa and 427°C to 200 kPa and 27°C?

Step 1 of 5

Given:

The initial Temperatures, $T_1 = 427°C = 700K$

The final Temperatures, $T_2 = 27°C = 300K$

The initial Pressures, $P_1 = 2000\ kPa$

The final pressure, $P_2 = 200\ kPa$

Step 2 of 5

Average temperature, $T = \frac{T_1 + T_2}{2}$

$$\Rightarrow T = \frac{700 + 300}{2}$$

$$\Rightarrow T = 500K$$

For Helium

From ideal gas specific heats of various common gases at various temperatures table,

Specific Heat at constant pressure, $c_p = 5.1926\ \frac{kJ}{kg \cdot K}$

Gas constant, $R = 2.0769\ \frac{kJ}{kg \cdot K}$

Step 3 of 5

For Nitrogen

From ideal gas specific heats of various common gases at various temperatures table,

Specific Heat at constant pressure at the average temperature, $c_p = 1.056\ \frac{kJ}{kg \cdot K}$

Gas constant, $R = 0.2968\ \frac{kJ}{kg \cdot K}$

Step 4 of 5

For Helium:

Entropy change, $\Delta s_{He} = \left[c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \right]_{He}$

$$\Rightarrow \Delta s_{He} = 5.1926 \times \ln\left(\frac{27+273}{427+273}\right) - 2.0769 \times \ln\left(\frac{200}{2000}\right)$$

$$\Rightarrow \Delta s_{He} = 0.38256\ \frac{kJ}{kg \cdot K} \text{------(1)(greatest entropy change)}$$

Step 5 of 5

For Nitrogen:

Entropy change, $\Delta s_{Nitrogen} = \left[c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \right]_{Nitrogen}$

$$\Rightarrow \Delta s_{Nitrogen} = 1.056 \times \ln\left(\frac{27+273}{427+273}\right) - 0.2968 \times \ln\left(\frac{200}{2000}\right)$$

$$\Rightarrow \Delta s_{Nitrogen} = -0.21134\ \frac{kJ}{kg \cdot K} \text{------(2)}$$

From (1) and (2), Helium experiences the greatest entropy change.

Question: 8.078P

Air is expanded from 2000 kPa and 500°C to 100 kPa and 50°C. Assuming constant specific heats, determine the change in the specific entropy of air.

Step 1 of 2

Given

Initial pressure of air is, $P_1 = 2000\ \text{kPa}$

Initial temperature of air is, $T_1 = 500\ ^\circ\text{C} = 773\ \text{K}$

Final pressure of air is, $P_2 = 100\ \text{kPa}$

Final temperature of air is, $T_2 = 50\ ^\circ\text{C} = 323\ \text{K}$

Step 2 of 2

The change in specific entropies of an ideal gas is given by

$$s_2 - s_1 = c_p \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

From ideal gas specific heats of gases table properties of air are

$(c_p)_{air} = 1.005\ \text{kJ/kg.K}$

$(R)_{air} = 0.287\ \text{kJ/kg.K}$

$$s_2 - s_1 = 1.005 \times \ln\left(\frac{323}{773}\right) - 0.287 \times \ln\left(\frac{100}{2000}\right)$$

$$s_2 - s_1 = 1.005 \times \ln(0.4178) - 0.287 \times \ln(0.05)$$

$$s_2 - s_1 = 1.005 \times (-0.873) - 0.287 \times (-2.996)$$

$$s_2 - s_1 = -0.8774 + 0.8598$$

$$\boxed{s_2 - s_1 = -0.0176\ \text{kJ/kg.K}}$$

Therefore the change in the specific entropy of air is $-0.0176\ \text{kJ/kg.K}$

Question: 8.079P

What is the difference between the entropies of air at 15 psia and 70°F and air at 40 psia and 250°F per unit mass basis.

Step 1 of 2

Given

Initial pressure of air is, $P_1 = 15$ psia

Initial temperature of air is, $T_1 = 70°F = 529.67$ R

Final pressure of air is, $P_2 = 40$ psia

Final temperature of air is, $T_2 = 250°F = 709.67$ R

Step 2 of 2

From ideal gas specific heat of air table

$(c_p)_{air} = 0.24$ Btu/lbm.R

$(R)_{air} = 0.06855$ Btu/lbm.R

As air is an ideal gas the change in specific entropy can be calculated as

$$s_2 - s_1 = c_p \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$s_2 - s_1 = 0.24 \times \ln\left(\frac{709.67}{529.67}\right) - 0.06855 \times \ln\left(\frac{40}{15}\right)$$

$$s_2 - s_1 = 0.24 \times \ln(1.3398) - 0.06855 \times \ln(2.667)$$

$$s_2 - s_1 = 0.24 \times 0.2925 - 0.06855 \times 0.981$$

$$s_2 - s_1 = 0.0702 - 0.0672$$

$$\boxed{s_2 - s_1 = 0.003 \text{ Btu/lbm.R}}$$

Therefore the specific entropy of air increases by 0.003 Btu/lbm.R.

Question: 8.080P

Nitrogen at 900 kPa and 300°C is expanded adiabati-cally in a closed system to 100 kPa. Determine the minimum nitrogen temperature after the expansion.

Step 1 of 3

Given

Initial pressure of Nitrogen is, $P_1 = 900\ \text{kPa}$

Initial temperature Nitrogen is, $T_1 = 300\ ^\circ\text{C} = 573\ \text{K}$

Final pressure of Nitrogen is, $P_2 = 100\ \text{kPa}$

Step 2 of 3

Since the process is adiabatic process the ratio of temperatures is given by

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

Where γ is ratio of specific heats

Assuming the average temperature of 450K and at this temperature from ideal gas specific heats of gases for Nitrogen

$\gamma = 1.395$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{1.395-1}{1.395}}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{0.395}{1.395}}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{0.395}{1.395}}$$

$$T_2 = 573 \times \left(\frac{100}{900}\right)^{\frac{0.395}{1.395}}$$

$$T_2 = 307.579\ \text{K}$$

$$\boxed{T_2 = 34.579^\circ\text{C}}$$

Step 3 of 3

Average temperature

$$T_{avg} = \frac{T_1 + T_2}{2}$$

$$= \frac{573 + 307.579}{2}$$

$$T_{avg} = 440.289\ \text{K}$$

The obtained value of average temperature of Nitrogen is approximately equal to the assumed value so our assumption is correct

Therefore the minimum nitrogen temperature after the expansion is $T_2 = 34.579^\circ\text{C}$

Question: 8.081P

Air at 15 psia and 70°F is compressed adiabatically in a closed system to 200 psia. What is the minimum temperature of the air following this compression?

Step 1 of 2

Given

Initial pressure of closed system is, $P_1 = 15$ psia

Initial temperature of closed system is, $T_1 = 70°F = 529.67$ R

Final pressure of closed system is, $P_2 = 200$ psia

Step 2 of 2

The minimum temperature of the air in the compression process is obtained when the compression process is reversible adiabatic then the final temperature of the air can be calculated by using the relation

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}},$$

Where

γ = ratio of specific heats of air

$\gamma = 1.4$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{1.4-1}{1.4}}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{0.4}{1.4}}$$

$$\frac{T_1}{T_1} = \left(\frac{P_2}{P_1}\right)^{0.2857}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{0.2857}$$

$$T_2 = 529.67 \times \left(\frac{200}{15}\right)^{0.2857}$$

$$T_2 = 529.67 \times (13.33)^{0.2857}$$

$$T_2 = 529.67 \times 2.096$$

$$T_2 = 1110.188 \text{ R}$$

$$T_2 = 650.518°F$$

Therefore the temperature after the compression is 650.518°F.

Question: 8.082P

An insulated piston-cylinder device initially contains 300 L of air at 120 kPa and 17°C. Air is now heated for 15 min by a 200-W resistance heater placed inside the cylinder. The pressure of air is maintained constant during this process. Determine the entropy change of air, assuming (*a*) constant specific heats and (*b*) variable specific heats.

Step 1 of 3

Calculate the Mass of air by using the following equation:

$$m = \frac{P_1 V_1}{R T_1}$$

Here, initial pressure is P_1 ,initial volume is V_1 , gas constant is R, initial temperature is T_1 .

Substitute $120\ \text{kPa}$ for P_1 , $290\ \text{K}$ for T_1 , $0.3\ \text{m}^3$ for V_1 , $0.287\ \text{kJ/kg}\cdot\text{K}$ for R.

$$\begin{aligned} m &= \frac{120\times 0.3}{0.287\times 290} \\ &= \frac{36}{83.23} \\ &= 0.4325\,\text{kg} \end{aligned}$$

Calculate Electrical work done by using the following equation:

$$W = \dot{W} \times \Delta t$$

Here, Power of resistance is $\dot{W}$, time is Δt .

Substitute, $200\ \text{W}$ for $\dot{W}$, $15\ \text{min}$ for Δt :

$$\begin{aligned} W &= 200\times 15\times 60 \\ &= 180\,\text{kJ} \end{aligned}$$

Step 2 of 3

(a)

Assume constant specific heats.

Assume average temperature as room temperature 300 K.

From ideal gas specific heats of various gas table,

For air, the Specific heat at constant pressure is $c_p = 1.005\ \text{kJ/kg}\cdot\text{K}$.

Calculate the final temperature by using the following equation:

$$W = mc_p(T_2 - T_1)$$

$$T_2 = T_1 + \frac{W}{mc_p}$$

Substitute, $1.005\ \text{kJ/kg}\cdot\text{K}$ for c_p , $290\ \text{K}$ for T_1, $0.4325\,\text{kg}$ for m , $180\,\text{kJ}$ for W :

$$\begin{aligned} T_2 &= 290 + \left(\frac{180}{0.4325\times 1.005}\right) \\ &= 290 + 414.114 \\ &= 704.11\,\text{K} \end{aligned}$$

Calculate the Entropy change by using the following equation.

$$\begin{aligned} \Delta s_{sys} &= m\times(s_2 - s_1) \\ &= m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) \end{aligned}$$

Since, pressure is constant, $P_1 = P_2$.

$$\Delta s_{sys} = mc_p \ln\frac{T_2}{T_1}$$

Substitute, $0.4325\,\text{kg}$ for m , $1.029\ \text{kJ/kg}\cdot\text{K}$ for c_p , $290\ \text{K}$ for T_1 and 694.45K for T_2 :

$$\begin{aligned} \Delta s_{sys} &= 0.43254\times 1.029\times \ln\left(\frac{704.11}{290}\right) \\ &= 0.43254\times 1.029\times 0.887 \\ &= 0.3948\,\text{kJ/K} \end{aligned}$$

Therefore, the Entropy change of air for constant specific heat, Δs_{sys} is $\boxed{0.3948\,\text{kJ/K}}$.

(b)

Assume variable specific heats.

Calculate the Work done by using the following equation:

$$W = m(h_2 - h_1)$$

$$h_2 = h_1 + \frac{W}{m}$$

Here, enthalpy at initial state is h_1 , enthalpy at final state is h_2

From ideal gas properties of air table at $T_1 = 290\ \text{K}$,

The enthalpy is $h_1 = 290.16\ \text{kJ/kg}$

The entropy is $s_1^\circ = 1.66802\ \text{kJ/kg}\cdot\text{K}$

Substitute, $290.16\ \text{kJ/kg}$ for h_1 , $180\,\text{kJ}$ for W , $0.4325\,\text{kg}$ for m :

$$\begin{aligned} h_2 &= 290.16 + \frac{180}{0.43254} \\ &= 290.16 + 416.146 \\ &= 706.306\,\text{kJ/kg} \end{aligned}$$

Step 3 of 3

From ideal gas properties of air, calculate entropy at $h_2 = 706.306\,\text{kJ/kg}$ by interpolating the entropy values at $h = 702.52\,\text{kJ/kg}$ and $h = 713.27\,\text{kJ/kg}$:

$$\begin{aligned} s_2^\circ &= s^0_{h=702.52\text{kJ/kg}} + \left(s^\circ_{h=713.27\text{kJ/kg}} - s^0_{h=702.52\text{kJ/kg}}\right)\left(\frac{706.306-702.52}{713.27-702.52}\right) \\ &= 2.55731 + (2.57277 - 2.55731)(0.35355) \\ &= 2.55731 + 0.00546 \\ &= 2.5627\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Calculate the Entropy change by using the following equation:

$$\begin{aligned} \Delta s_{sys} &= m\left(s_2^\circ - s_1^0 + R\ln\frac{P_2}{P_1}\right) \\ &= \Delta s_{sys} = m\left(s_2^0 - s_1^\circ\right) \qquad (P_2 = P_1) \end{aligned}$$

Substitute, $0.4325\,\text{kg}$ for m , $2.5627\ \text{kJ/kg}\cdot\text{K}$ for s_2°, $1.66802\ \text{kJ/kg}\cdot\text{K}$ for s_1° :

$$\begin{aligned} \Delta s_{sys} &= 0.43254\times(2.5628 - 1.66802) \\ &= 0.387\,\text{kJ/K} \end{aligned}$$

Therefore, the Entropy change of air for variable specific heat, Δs_{sys} is $\boxed{0.387\,\text{kJ/K}}$.

Question: 8.083P

A piston-cylinder device contains 0.75 kg of nitrogen gas at 140 kPa and 37°C. The gas is now compressed slowly in a polytropic process during which $PV^{1.3}$ = constant. The process ends when the volume is reduced by one-half. Determine the entropy change of nitrogen during this process.

Step 1 of 3

Given

Mass of nitrogen is, $m = 0.75\,\text{kg}$

Initial pressure of nitrogen is, $P_1 = 140\,\text{kPa}$

Initial temperature of nitrogen is, $T_1 = 37°\text{C} = 310\,\text{K}$

Polytropic index is, $n = 1.3$

Step 2 of 3

The final volume is half of the initial volume. So,

$$v_2 = \frac{v_1}{2}$$

$$v_1 = 2 \times v_2$$

In polytropic process the relation between temperature and volume is

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1}$$

$$T_2 = T_1 \times \left(\frac{v_1}{v_2}\right)^{n-1}$$

$$T_2 = 310 \times \left(\frac{2 \times v_2}{v_2}\right)^{1.3-1}$$

$$T_2 = 310 \times 2^{0.3}$$

$$T_2 = 310 \times 1.2311$$

$$T_2 = 381.641\,\text{K}$$

Step 3 of 3

From properties of ideal gases table the specific heat (at constant volume) and gas constant values of nitrogen are respectively

$c_v = 0.743\,\text{kJ/kg.K}$

$R = 0.2968\,\text{kJ/kg.K}$

The change in entropy is given by

$$\Delta S = m \times \left[c_v \times \ln\left(\frac{T_2}{T_1}\right) + R \times \ln\left(\frac{v_2}{v_1}\right)\right]$$

$$\Delta S = 0.75 \times \left[0.743 \times \ln\left(\frac{381.641}{310}\right) + 0.2968 \times \ln\left(\frac{v_2}{2 \times v_2}\right)\right]$$

$$\Delta S = 0.75 \times \left[0.743 \times \ln(1.2311) + 0.2968 \times \ln(0.5)\right]$$

$$\Delta S = 0.75 \times \left[0.743 \times 0.2079 + 0.2968 \times (-0.693)\right]$$

$$\Delta S = 0.75 \times \left[0.1545 - 0.2057\right]$$

$$\Delta S = 0.75 \times \left[-0.0512\right]$$

$$\boxed{\Delta S = -0.0384\,\text{kJ/K}}$$

Therefore the entropy of nitrogen decreases by 0.0384 kJ/K.

Question: 8.084P

Reconsider Prob. 8-83. Using EES (or other) software, investigate the effect of varying the polytropic exponent from 1 to 1.4 on the entropy change of the nitrogen. Show the processes on a common P- *V* diagram.

Step 1 of 8

Given

Mass of nitrogen is, $m = 0.75\ \text{kg}$

Initial pressure of nitrogen is, $P_1 = 140\ \text{kPa}$

Initial temperature of nitrogen is, $T_1 = 37\ ^\circ\text{C} = 310\ \text{K}$

Polytrophic index varies, $n = 1$ to 1.4

Step 2 of 8

Give all the inputs in the EES software as given below

T1 =310 [K]

P1 =140 [kPa]

m =0.75 [kg]

RatioV =0.5

c_v =0.743 [kJ/kg-K]

R =0.297 [kJ/kg-K]

Step 3 of 8

For calculating other parameters we should give other equations as

T2 =T1*(1/RatioV)^(n-1)

DELTAS =m *(c_v *ln(T2/T1)+R *ln (RatioV))

P1*V1 = m *R *T1

Step 4 of 8

After writing these sets of equations, we should create a parametric table from

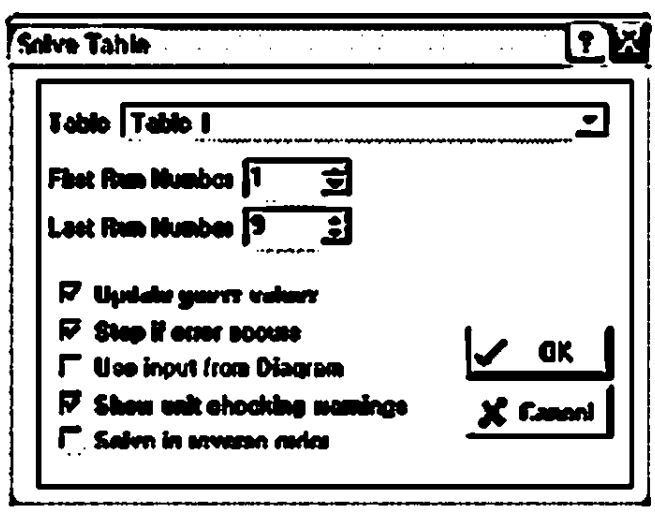
Tables → new parametric

Fill all the values of n.

	n	
Run 1	1	
Run 2	1.05	
Run 3	1.1	
Run 4	1.15	
Run 5	1.2	
Run 6	1.25	
Run 7	1.3	
Run 8	1.35	
Run 9	1.4	

Step 5 of 8

Now solve the table for respective DELTAS values by pressing F3.



Step 6 of 8

Then we get

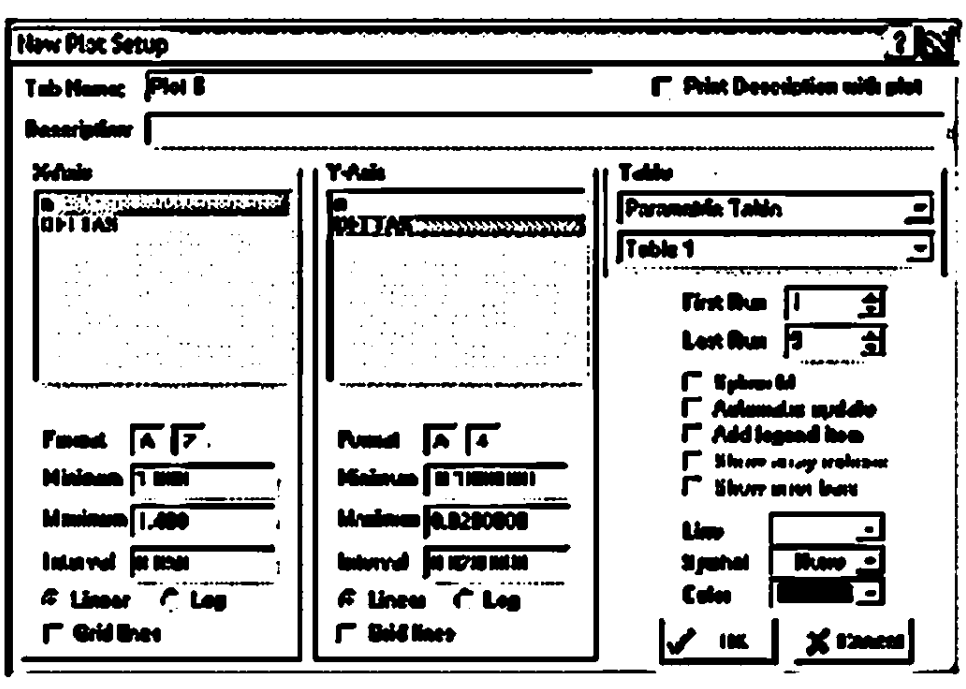
	n	ΔS [kJ/K]
Run 1	1	-0.1544
Run 2	1.05	-0.1361
Run 3	1.1	-0.1159
Run 4	1.15	-0.09646
Run 5	1.2	-0.07715
Run 6	1.25	-0.05783
Run 7	1.3	-0.03852
Run 8	1.35	-0.01921
Run 9	1.4	0.000104

Step 7 of 8

For getting the plot we should go for

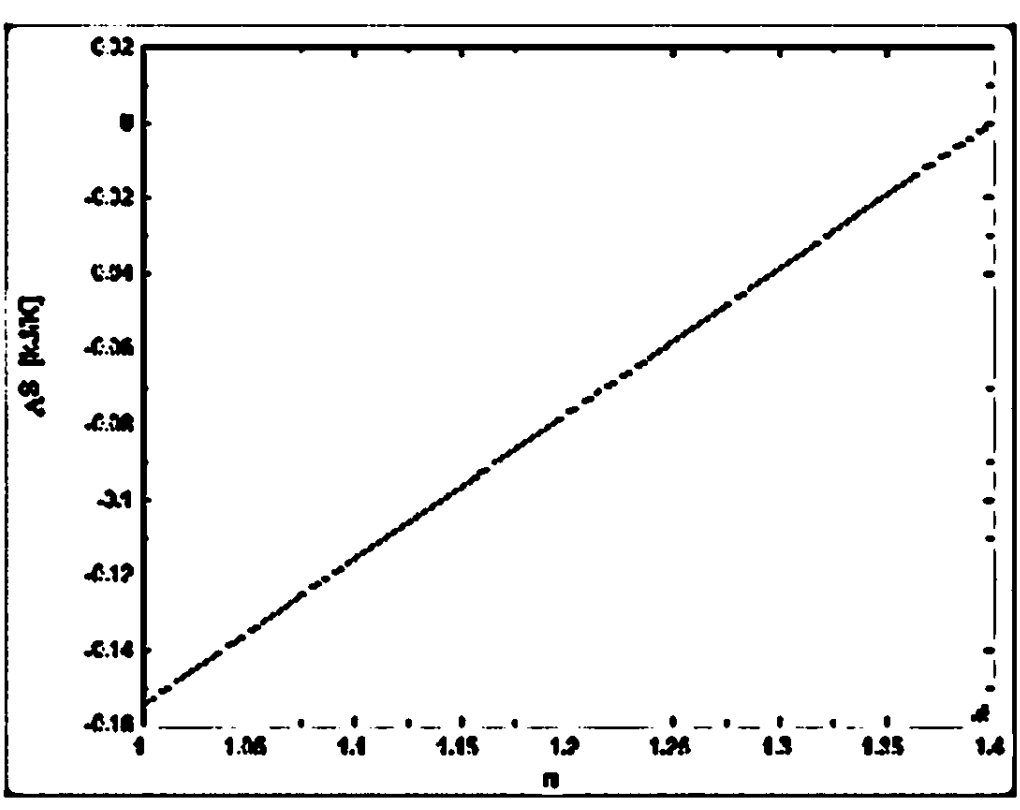
Plots → New plot window

We should fill all the required fields by taking n in x-axis and DELTAS in y-axis.



Step 8 of 8

After that we get the plot as



Question: 8.085P

A mass of 15 lbm of helium undergoes a process from an initial state of 50 ft3 /lbm and 80°F to a final state of 10 ft3 /lbm and 200°F Determine the entropy change of helium during this process, assuming (*a*) the process is reversible and (*b*) the process is irreversible.

Step 1 of 2

Given:

Initial temperature, $T_1 = 80°\text{F}$

$\Rightarrow T_1 = 540\ \text{R}$

Initial volume, $V_1 = 50\ \text{ft}^3/\text{lbm}$

Final temperature, $T_2 = 200°\text{F}$

$\Rightarrow T_2 = 660\ \text{R}$

Final volume, $V_2 = 10\ \text{ft}^3/\text{lbm}$

Mass of helium, $m = 15\ \text{lbm}$

Gas constant of helium, $R = 0.4961\ \text{Btu/lbm.R}$

Constant volume specific heat of helium, $c_v = 0.753\ \text{Btu/lbm.R}$

Step 2 of 2

(a) and (b):

The entropy change will be the same for both reversible and irreversible cases.

Entropy change, $\Delta s_{He} = m \times \left(c_v \ln \frac{T_2}{T_1} + R \ln \frac{V_2}{V_1} \right)$

$$\Rightarrow \Delta s_{He} = 15 \times \left(0.753 \times \ln\left(\frac{660}{540}\right) + 0.4961 \times \ln\left(\frac{10}{50}\right) \right)$$

$$\Rightarrow \boxed{\Delta s_{He} = -9.7101\ \text{Btu/R}}$$

Question: 8.086P

1-kg of air at 200 kPa and 127°C is contained in a piston-cylinder device. Air is now allowed to expand in a reversible, isothermal process until its pressure is 100 kPa. Determine the amount of heat transferred to the air during this expansion.

Step 1 of 3

Given

Mass of air is, $m = 1.0\ \text{kg}$

Initial pressure of air is, $P_1 = 200\ \text{kPa}$

Temperature of air is, $T = 127°\text{C} = 400\ \text{K}$

Final pressure of air is, $P_2 = 100\ \text{kPa}$

Step 2 of 3

From properties of gases gas constant of air is

$R = 0.287\ \text{kJ/kg.K}$

Applying first law of thermodynamics to the piston cylinder device

$dQ = dU + dW$

As work is done by the device, the equation can be written as

$dQ = dU + W_{out}$

$dQ = m(u_2 - u_1) + W_{out}$

As the process is isothermal process, there is no change in internal energy then the first term on right hand side will be zero.

$dQ = 0 + W_{out}$

$dQ = W_{out}$ ..... (1)

Step 3 of 3

The work done in an isothermal process is given by

$$W_{out} = mRT \times \ln\left(\frac{P_1}{P_2}\right)$$

$$W_{out} = 1 \times 0.287 \times 400 \times \ln\left(\frac{200}{100}\right)$$

$W_{out} = 0.287 \times 400 \times \ln 2$

$W_{out} = 0.287 \times 400 \times 0.693$

$W_{out} = 79.5564\ \text{kJ}$ ..... (2)

Substituting value obtained from equation (2) in equation (1) we get

$\boxed{dQ = 79.5564\ \text{kJ}}$

Therefore the heat transferred during expansion of air is 79.5564 kJ

Question: 8.087P

Argon is expanded in an isentropic turbine from 2 MPa and 500°C to 200 kPa. Determine the outlet temperature and the work produced by this turbine per unit mass of argon.

Step 1 of 3

Given

Inlet pressure of argon is, $P_1 = 2\ \text{MPa} = 2000\ \text{kPa}$

Inlet temperature of argon is, $T_1 = 500°\text{C} = 773\ \text{K}$

Exit pressure of argon is, $P_2 = 200\ \text{kPa}$

Step 2 of 3

From ideal gas specific heat of gases specific heat and index values of argon respectively are

$c_p = 0.2081\ \text{kJ/kg.K}$

$k = 1.667$

Since the process is isentropic. The temperature and pressure in an isentropic process can be related as below

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 773 \times \left(\frac{200}{2000}\right)^{\frac{1.667-1}{1.667}}$$

$$T_2 = 773 \times \left(\frac{200}{2000}\right)^{\frac{0.667}{1.667}}$$

$$T_2 = 773 \times (0.1)^{0.4}$$

$$T_2 = 773 \times 0.3981$$

$$T_2 = 307.73\ \text{K}$$

$$\boxed{T_2 = 34.73°\text{C}}$$

Therefore outlet temperature of the argon is $T_2 = 34.73°\text{C}$

Step 3 of 3

The work done by the turbine is given by

$$w_{tur} = h_1 - h_2$$

$$w_{tur} = c_p(T_1 - T_2)$$

$$w_{tur} = 0.2081 \times (773 - 307.73)$$

$$w_{tur} = 0.2081 \times 465.27$$

$$\boxed{w_{tur} = 96.8223\ \text{kJ/kg}}$$

Therefore the work produced by the turbine per unit mass of argon is $96.8223\ \text{kJ/kg}$

Question: 8.088P

Air is compressed in an isentropic compressor from 15 psia and 70°F to 200 psia. Determine the outlet temperature and the work consumed by this compressor per unit mass of air.

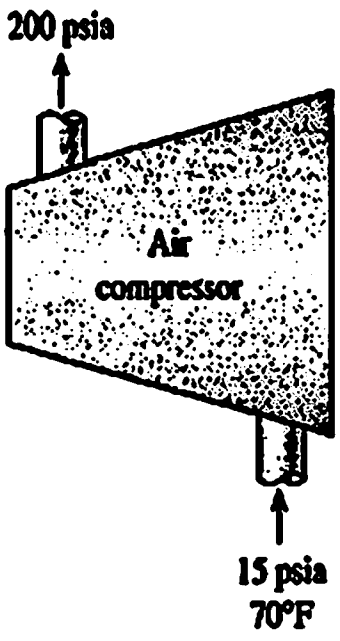


FIGURE P8–88E

Step 1 of 2

Obtain properties of air at an average temperature of $400°F$ from table A-2E, "ideal gas specific heat of gases" in the text book.

Specific heat at constant pressure, $c_p = 0.245$ Btu/lbm.R

Specific heat ratio, $k = 1.389$

Find the temperature at the exit of the compressor by using the following isentropic relation:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Here, P_1 is compressor inlet pressure, P_2 is compressor exit pressure, T_1 is compressor inlet temperature, and T_2 is compressor exit temperature.

Substitute $(70+459.67)$R for T_1, 15 psia for P_1, 200 psia for P_2, and 1.389 for *k*.

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = (70+459.67) \times \left(\frac{200}{15}\right)^{\frac{1.389-1}{1.389}}$$

$$T_2 = 1094.089\,\text{R}$$

Therefore, the outlet temperature of air is $\boxed{1094.089\ \text{R}}$.

Step 2 of 2

Find the work done on the compressor by using the following equation:

$$w_{com} = h_1 - h_2$$
$$w_{com} = c_p(T_1 - T_2)$$

Substitute 0.245 Btu/lbm.R for c_p, $(70+459.67)$R for T_1, and 1094.089 R for T_2.

$$w_{com} = 0.245 \times (529.67 - 1094.089)$$
$$= -138.282\,\text{Btu/lbm}$$

Negative sign indicates work is done on the system.

Therefore, the work consumed by the compressor per unit mass of air is $\boxed{138.282\ \text{Btu/lbm}}$.

Question: 8.089P

An insulated rigid tank is divided into two equal parts by a partition. Initially, one part contains 5 kmol of an ideal gas at 250 kPa and 40°C, and the other side is evacuated. The partition is now removed, and the gas fills the entire tank. Determine the total entropy change during this process.

Step 1 of 3

Given

Number of moles in first part of the tank is, $N = 5\ \text{kmol}$

Initial pressure in first part of the tank is, $P_1 = 250\ \text{kPa}$

Initial temperature in first part of the tank is, $T_1 = 40°C = 313\ \text{K}$

Step 2 of 3

As there is no heat addition and also the tank is insulated, the change in internal energy is zero.

$$\Delta U = 0$$

$$mc_v(T_1 - T_2) = 0$$

$$T_1 - T_2 = 0$$

$$T_1 = T_2$$

As the gas is made to fill the second part of the tank then the final volume will be double (since both the parts are of equal volume) the initial volume.

$$v_2 = 2 \times v_1$$

We know universal gas constant is,

$$\overline{R} = 8.314\ \text{kJ/kmol.K}$$

Step 3 of 3

The entropy change during the process is given by

$$\Delta S = N \times \left[c_v \times \ln\left(\frac{T_2}{T_1}\right) + \overline{R} \times \ln\left(\frac{v_2}{v_1}\right) \right]$$

$$\Delta S = N \times \left[c_v \times \ln\left(\frac{T_1}{T_1}\right) + \overline{R} \times \ln\left(\frac{v_2}{v_1}\right) \right]$$

$$\Delta S = N \times \left[c_v \times \ln(1) + \overline{R} \times \ln\left(\frac{v_2}{v_1}\right) \right]$$

$$\Delta S = N \times \left[c_v \times 0 + \overline{R} \times \ln\left(\frac{v_2}{v_1}\right) \right]$$

$$\Delta S = N \times \left[0 + \overline{R} \times \ln\left(\frac{v_2}{v_1}\right) \right]$$

$$\Delta S = N \times \overline{R} \times \ln\left(\frac{v_2}{v_1}\right)$$

$$\Delta S = 5 \times 8.314 \times \ln\left(\frac{2 \times v_1}{v_1}\right)$$

$$\Delta S = 5 \times 8.314 \times \ln(2)$$

$$\Delta S = 5 \times 8.314 \times 0.693$$

$$\boxed{\Delta S = 28.808\ \text{kJ/K}}$$

Therefore the total entropy change during this process is 28.808 kJ/K

Question: 8.090P

Air is compressed in a piston-cylinder device from 90 kPa and 22°C to 900 kPa in a reversible, adiabatic process. Determine the final temperature and the work done during this process, assuming (*a*) constant specific heats and (*b*) variable specific heats for air.

Step 1 of 6

Given data:

Initial pressure of air is, $P_1 = 90\ \text{kPa}$

Initial temperature of air is, $T_1 = 22°\text{C}$

$= 295\ \text{K}$

Final pressure of air is, $P_2 = 900\ \text{kPa}$

Step 2 of 6

Assuming an average temperature of $T_{avg} = 425\ \text{K}$

From ideal gas specific heats of various common liquids

By taking average of 400 K & 450 K

$$k = \frac{1.395 + 1.391}{2}$$

$$k = \frac{2.786}{2}$$

$$k = 1.393$$

$$c_v = \frac{0.726 + 0.733}{2}$$

$$c_v = \frac{1.459}{2}$$

$$c_v = 0.7295\ \text{kJ/kg}\cdot\text{K}$$

The gas constant, $R = 0.287\ \text{kJ/kg}\cdot\text{K}$

Step 3 of 6

(a) For an isentropic process, the final temperature is given by

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 295\times\left(\frac{900}{90}\right)^{\frac{1.393-1}{1.393}}$$

$$T_2 = 295\times(10)^{\frac{0.393}{1.393}}$$

$$T_2 = 295\times 1.915$$

$$\boxed{T_2 = 564.9\ \text{K}}$$

The final temperature, $T_2 = 564.9\ \text{K}$

Step 4 of 6

Now we know that

Work done by constant specific heat

$$w_{in} = c_v\times(T_2 - T_1)$$

$$w_{in} = 0.7295\times(564.9 - 295)$$

$$w_{in} = 0.7295\times 269.9$$

$$\boxed{w_{in} = 196.89\ \text{kJ/kg}}$$

The work done, $w_{in} = 196.89\ \text{kJ/kg}$

Step 5 of 6

(b) From ideal gas properties of air

At temperature, $T_1 = 295\ \text{K}$

Relative pressure, $P_{r1} = 1.3068$

Internal energy, $u_1 = 210.49\ \text{kJ/kg}$

We know that

Final relative pressure

$$P_{r2} = P_{r1}\left(\frac{P_2}{P_1}\right)$$

$$P_{r2} = 1.3068\times\left(\frac{900}{90}\right)$$

$$P_{r2} = 1.3068\times 10$$

$$P_{r2} = 13.068$$

Now from ideal gas properties of air

Final temperature is interpolated as

$$\left(\frac{T_2 - 560}{570 - 560}\right) = \left(\frac{13.068 - 12.66}{13.50 - 12.66}\right)$$

$$\left(\frac{T_2 - 560}{10}\right) = \left(\frac{0.408}{0.84}\right)$$

$$\left(\frac{T_2 - 560}{10}\right) = 0.4857$$

$$T_2 - 560 = 0.4857\times 10$$

$$T_2 = 4.857 + 560$$

$$\boxed{T_2 = 564.9\ \text{K}}$$

The final temperature, $T_2 = 564.9\ \text{K}$

Step 6 of 6

Now from ideal gas properties of air

Final internal energy is interpolated as

$$\left(\frac{u_2 - 404.42}{411.97 - 404.42}\right) = \left(\frac{13.068 - 12.66}{13.50 - 12.66}\right)$$

$$\left(\frac{u_2 - 404.42}{7.55}\right) = \left(\frac{0.408}{0.84}\right)$$

$$\left(\frac{u_2 - 404.42}{7.55}\right) = 0.4857$$

$$u_2 - 404.42 = 0.4857\times 7.55$$

$$u_2 = 3.667 + 404.42$$

$$u_2 = 408.087\ \text{kJ/kg}$$

Work input by variable specific heats is given by

$$w_{in} = u_2 - u_1$$

$$w_{in} = 408.087 - 210.49$$

$$\boxed{w_{in} = 197.6\ \text{kJ/kg}}$$

The work done, $w_{in} = 197.6\ \text{kJ/kg}$

Question: 8.091P

 Reconsider Prob. 8-90. Using EES (or other) software, evaluate and plot the work done and final temperature during the compression process as functions of the final pressure for the two cases as the final pressure varies from 100 to 1200 kPa.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 8.092P

An insulated rigid tank contains 4 kg of argon gas at 450 kPa and 30°C. A valve is now opened, and argon is allowed to escape until the pressure inside drops to 200 kPa. Assuming the argon remaining inside the tank has undergone a reversible, adiabatic process, determine the final mass in the tank.

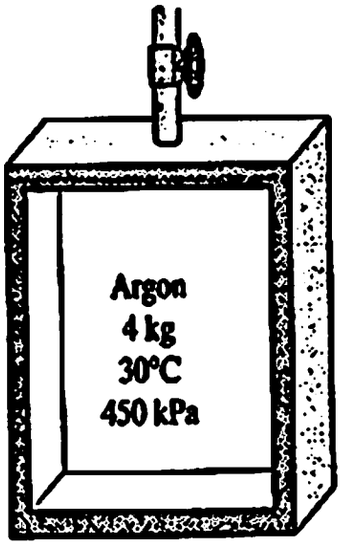


FIGURE P8–92

Step 1 of 2

Given:

Initial temperature, $T_1 = 30°C$

$$\Rightarrow T_1 = 303\ K$$

Initial Pressures, $P_1 = 450\ kPa$

Final pressure, $P_2 = 200\ kPa$

Specific heat ratio of argon, $k = 1.667$

Initial mass, $m = 4\ kg$

Step 2 of 2

Final temperature, $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$

$$\Rightarrow T_2 = 303\times\left(\frac{200}{450}\right)^{(1.667-1)/1.667}$$

$$\Rightarrow T_2 = 219.04K$$

From ideal gas equation, $PV = mRT$

$$\Rightarrow \frac{P_1V}{P_2V} = \frac{m_1RT_1}{m_2RT_2}$$

Final mass, $m_2 = \frac{P_2T_1}{P_1T_2}\times m_1$

$$\Rightarrow m_2 = \frac{200\times 303}{450\times 219}\times 4$$

$$\Rightarrow \boxed{m_2 = 2.46\ kg}$$

Question:

Reconsider Prob. 8-92. Using EES (or other) software, investigate the effect of the final pressure on the final mass in the tank as the pressure varies from 450 to 150 kPa, and plot the results.

Step 1 of 5

Mass of the argon, $m = 4\,\text{kg}$

Initial temperature of the argon gas, $T_1 = 30°\text{C} = 303°\text{F}$

Initial pressure of the argon gas, $P_1 = 450\,\text{kPa}$

Final pressure of the argon gas, $P_2 = 200\,\text{kPa}$

Step 2 of 5

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

Equations Window

```
"Given data"
T_1=30 [C]
P_1=450 [kPa]
"P_2=200 [kPa]"
m_1=4 [kg]

cp_1=Cp(Argon,T=T_1,P=P_1)
cv_1=Cv(Argon,T=T_1,P=P_1)
k=cp_1/cv_1

"For an isentropic process"
T_2/T_1=(P_2/P_1)^((k-1)/k)

"Ideal gas equation P*V=m*R*T"
P_1*V=m_1*R#*T_1
P_2*V=m_2*R#*T_2
```

Step 3 of 5

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below ,

Formatted Equations

Given data

$T_1 = 30 \;\; [C]$

$P_1 = 450 \;\; [kPa]$

$P_2 = 200 \;\; [kPa]$

$m_1 = 4 \;\; [kg]$

$cp_1 = Cp('Argon', T = T_1, P = P_1)$

$cv_1 = Cv('Argon', T = T_1, P = P_1)$

$$k = \frac{cp_1}{cv_1}$$

For an isentropic process

$$\frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{\left[\frac{k-1}{k}\right]}$$

Ideal gas equation P*V=m*R*T

$P_1 \cdot V = m_1 \cdot 8.314 \;\; [kJ/kmol\text{-}K] \cdot T_1$

$P_2 \cdot V = m_2 \cdot 8.314 \;\; [kJ/kmol\text{-}K] \cdot T_2$

Step 4 of 5

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,

New Parametric Table

No. of Runs: 11 Table: Table 1

Variables in equations: cp_1, cv_1, k, m_1, P_1, T_1, T_2, V

Add ->

<- Remove

Variables in table: P_2, m_2

Show Array Variables

OK Cancel

Step 5 of 5

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,

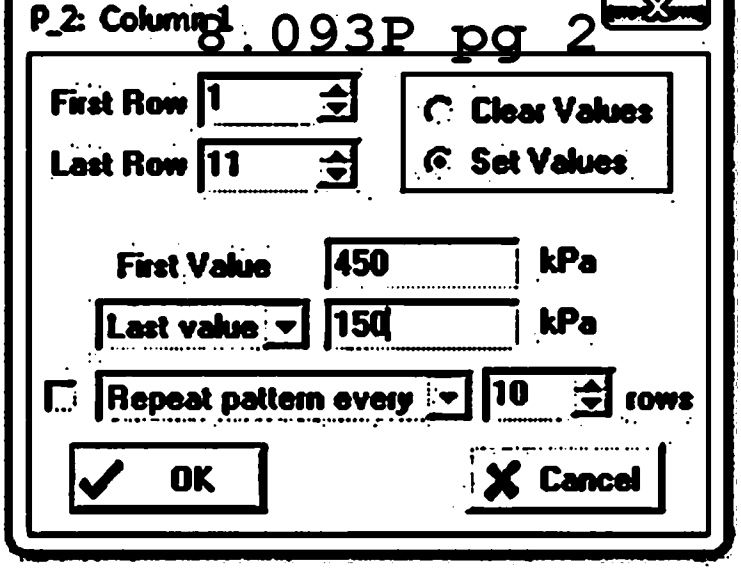
P_2: Column 1

First Row 1

Last Row 11

Clear Values

Set Values



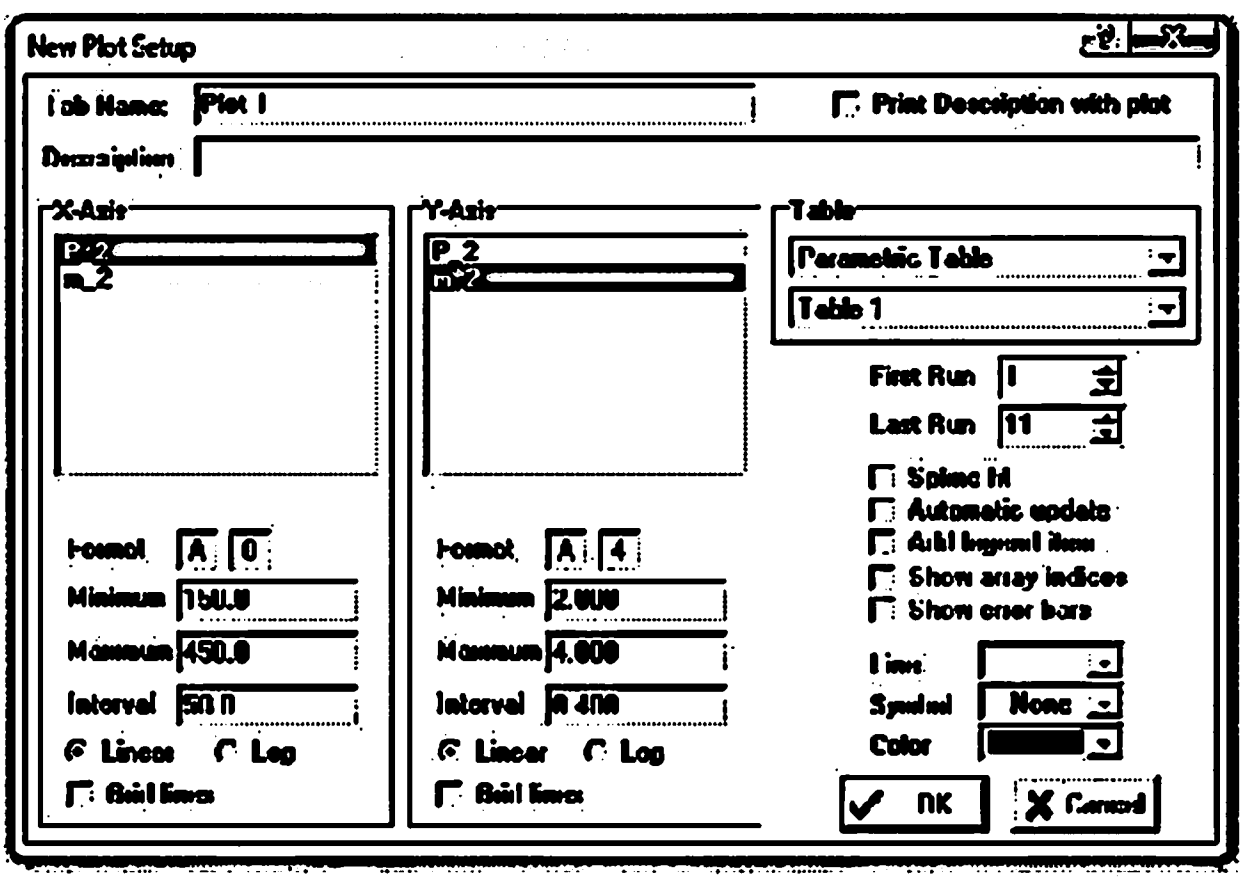
Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

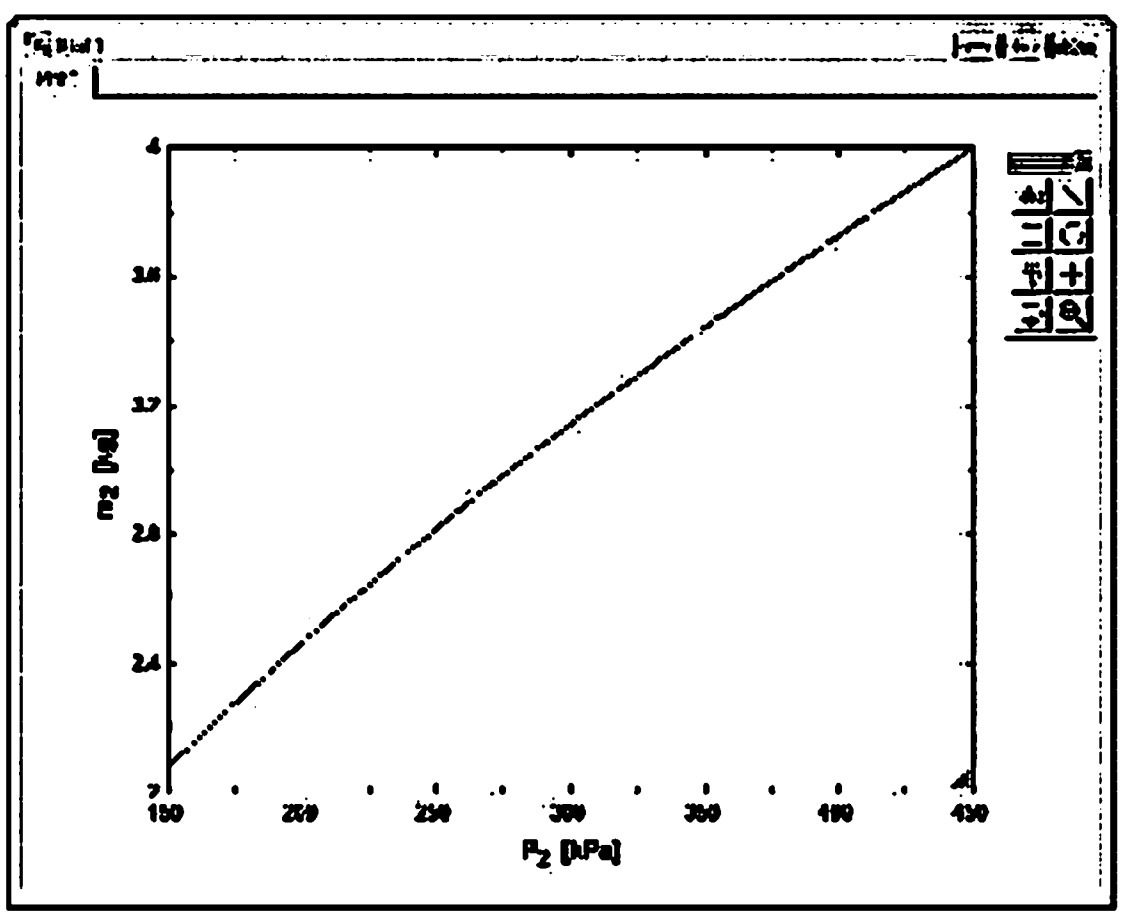
Table 1

1..11	P_2 [kPa]	m_2 [kg]
Run 1	450	4
Run 2	420	3.839
Run 3	390	3.673
Run 4	360	3.502
Run 5	330	3.325
Run 6	300	3.142
Run 7	270	2.951
Run 8	240	2.751
Run 9	210	2.541
Run 10	180	2.318
Run 11	150	2.079

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



By clicking on OK the graph will be generated as shown,



Question: 8.094P

Air enters an adiabatic nozzle at 60 psia, 540°F, and 200 ft/s and exits at 12 psia. Assuming air to be an ideal gas with variable specific heats and disregarding any irreversibilities, determine the exit velocity of the air.

Step 1 of 3

Given:

Initial temperature, $T_1 = 540^\circ F \Rightarrow T_1 = 1000\ \text{R}$

Inlet Pressures, $P_1 = 60\ \text{psia}$

Exit pressure $P_2 = 12\ \text{psia}$

Inlet velocity, $V_1 = 200\ \text{ft/s}$

Initial mass, $m = 4\ \text{kg}$

Step 2 of 3

From ideal gas properties of air table, at $T_1 = 1000\ \text{R}$

Initial relative pressure, $P_{r_1} = 12.3$

Initial enthalpy, $h_1 = 240.98\text{Btu/lbm}$

Initial relative pressure, $P_{r_2} = \dfrac{P_2}{P_1} P_{r_1}$

$$\Rightarrow P_{r_2} = \frac{12}{60} \times 12.3$$

$$\Rightarrow P_{r_2} = 2.46$$

From ideal gas properties of air Table, at $P_{r_2} = 2.46$

Final temperature, $T_2 = 635.9R$

Final enthalpy, $h_2 = 152.11\text{Btu/lbm}$

Step 3 of 3

Apply energy balance for this steady-flow system,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} = 0$$

$$\Rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\Rightarrow \dot{m} \times \left(h_1 + \frac{V_1^2}{2} \right) = \dot{m} \times \left(h_2 + \frac{V_2^2}{2} \right)$$

$$\Rightarrow h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = 0$$

Exit velocity of the air, $V_2 = \sqrt{2(h_1 - h_2) + V_1^2}$

$$\Rightarrow V_2 = \sqrt{2(240.98 - 152.11)\left(\frac{25,037}{1} \right) + 200^2}$$

$$\Rightarrow V_2 = \sqrt{4450076.38 + 40000}$$

$$\Rightarrow \boxed{V_2 = 2118.98\text{ft/s}}$$

Question: 8.095P

Air is expanded in an adiabatic nozzle during a polytropic process with $n = 1.3$. It enters the nozzle at 700 kPa and 100°C with a velocity of 30 m/s and exits at a pressure of 200 kPa. Calculate the air temperature and velocity at the nozzle exit.

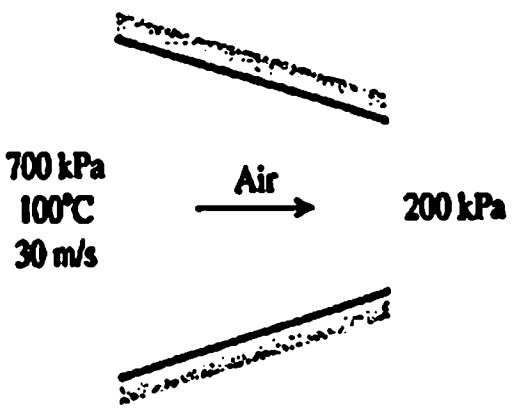


FIGURE P8–95

Step 1 of 2

Calculate the air temperature at the exit of the nozzle.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

Here, the temperature at the inlet is T_1, pressure at the inlet is P_1, temperature at outlet is T_2, pressure at outlet is P_2 and n is polytropic constant.

Substitute $373\ \text{K}$ for T_1, $700\ \text{kPa}$ for P_1, 1.3 for n, and $200\ \text{kPa}$ for P_2.

$$T_2 = (373)\left(\frac{200}{700}\right)^{\frac{1.3-1}{1.3}}$$
$$= 279.35\ \text{K}$$

Therefore, the temperature of air at the exit of the nozzle is, $\boxed{279.35\ \text{K}}$.

Step 2 of 2

Apply the energy balance for the steady flow system.

$$\dot{E}_{in} = \dot{E}_{out}$$
$$(\text{H}+\text{KE})_1 = (\text{H}+\text{KE})_2$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$
$$\frac{V_2^2}{2} = \frac{V_1^2}{2} + (h_1 - h_2)$$
$$V_2 = \sqrt{V_1^2 + 2(h_1 - h_2)}$$
$$V_2 = \sqrt{V_1^2 + 2(c_p T_1 - c_p T_2)}$$

Here, the velocity at the inlet is V_1, mass flow rate is $\dot{m}$, and specific heat at constant pressure is c_p.

Substitute $1.005\ \text{kJ/kg K}$ for c_p, 373 K for T_1, $279.35\ \text{K}$ for T_2, and $30\ \text{m/s}$ for V_1.

$$V_2 = \sqrt{(30)^2 + \left[2\times 1.005\times(373-279.35)\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)\right]}$$
$$= 434.89\ \text{m/s}$$

Therefore, the velocity of air at the exit is, $\boxed{434.89\ \text{m/s}}$

Question: 8.096P

Repeat Prob. 8-95 for the polytropic exponent of $n = 1.1$.

Step 1 of 6

Given

Polytropic index is, $n = 1.1$

Pressure at nozzle inlet is, $P_1 = 700\ \text{kPa}$

Temperature at nozzle inlet is, $T_1 = 100\ ^\circ\text{C}$

$= 373\ \text{K}$

Velocity at nozzle inlet is, $V_1 = 30\ \text{m/s}$

Pressure at nozzle exit is, $P_2 = 200\ \text{kPa}$

Step 2 of 6

From the ideal gas specific heats of various common gases

For specific heat of air is,

$c_p = 1.005\ \text{kJ/kg°K}$

The temperature ratio of polytropic process is given by

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$T_2 = 373\times\left(\frac{200}{700}\right)^{\frac{1.1-1}{1.1}}$$

$$T_2 = 373\times(0.2857)^{\frac{0.1}{1.1}}$$

$$T_2 = 373\times(0.2857)^{0.09}$$

$$T_2 = 373\times 0.8934$$

$$\boxed{T_2 = 333.24\ \text{K}}$$

Step 3 of 6

The energy balance for a steady-flow system is given by

$E_{in} - E_{out} = \Delta E_{sys}$

For a steady flow system the change in energy is zero. $E_{sys} = 0$

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$m_1\left[h_1 + \left(\frac{V_1^2}{2}\right)\right] = m_2\left[h_2 + \left(\frac{V_2^2}{2}\right)\right]$$

Step 4 of 6

As only one inlet and exit are there the mass flow rate will be same $m_1 = m_2 = m$

$$m\left[h_1 + \left(\frac{V_1^2}{2}\right)\right] = m\left[h_2 + \left(\frac{V_2^2}{2}\right)\right]$$

$$h_1 + \left(\frac{V_1^2}{2}\right) = h_2 + \left(\frac{V_2^2}{2}\right)$$

$$\left(\frac{V_2^2}{2}\right) = h_1 - h_2 + \left(\frac{V_1^2}{2}\right)$$

$$V_2^2 = 2\times\left[h_1 - h_2 + \left(\frac{V_1^2}{2}\right)\right]$$

$$V_2^2 = \left[2\times(h_1 - h_2) + V_1^2\right]$$

Step 5 of 6

In order to change enthalpy value from kJ/kg°K to m^2/s^2 we get

$$V_2^2 = \left[2000\times(h_1 - h_2) + V_1^2\right]$$

$$V_2 = \sqrt{2000\times(h_1 - h_2) + V_1^2}$$

$$V_2 = \sqrt{2000\times c_p\times(T_1 - T_2) + V_1^2}$$

Step 6 of 6

$$V_2 = \sqrt{2000\times c_p\times(T_1 - T_2) + V_1^2}$$

$$V_2 = \sqrt{2000\times 1.005\times(373 - 333.24) + 30^2}$$

$$V_2 = \sqrt{2000\times 1.005\times 39.76 + 900}$$

$$V_2 = \sqrt{79917.6 + 900}$$

$$V_2 = \sqrt{80817.6}$$

$$\boxed{V_2 = 284.28\ \text{m/s}}$$

Therefore the exit temperature is 333.24 K and the exit velocity is 284.28 m/s

Question: 8.097P

Air at 427°C and 600 kPa is contained in a piston-cylinder device. The air expands adiabatically until the pressure is 100 kPa. Determine the mass of air needed to produce maximum work of 1000 kJ. Assume air has constant specific heats evaluated at 300 K.

Step 1 of 3

Given

Initial pressure of air is, $P_1 = 600\ \text{kPa}$

Initial temperature of air is, $T_1 = 427°\text{C} = 700\ \text{K}$

Final pressure of air is, $P_2 = 100\ \text{kPa}$

Work produced by the device is $w_{dev} = 1000\ \text{kJ}$

Step 2 of 3

From ideal gas specific heats of gases specific heat and index values of air at temperature 300 K respectively are

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Since the process is isentropic. The temperature and pressure in an isentropic process can be related as below

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 700 \times \left(\frac{100}{600}\right)^{\frac{0.4}{1.4}}$$

$$T_2 = 700 \times (0.1666)^{\frac{0.4}{1.4}}$$

$$T_2 = 700 \times 0.599$$

$$T_2 = 419.3\ \text{K}$$

Step 3 of 3

The work done by the device is given by

$$w_{dev} = m(u_1 - u_2)$$

$$w_{dev} = mc_v(T_1 - T_2)$$

$$m = \frac{w_{dev}}{c_v(T_1 - T_2)}$$

$$m = \frac{1000}{0.718 \times (700 - 419.3)}$$

$$m = \frac{1000}{0.718 \times 280.7}$$

$$m = \frac{1000}{201.5426}$$

$$\boxed{m = 4.9617\ \text{kg}}$$

Therefore the mass of air needed to produce maximum work of 1000 kJ is

$m = 4.9617\ \text{kg}$

Question: 8.098P

Helium enters a steady-flow adiabatic compressor at0.6 kg/s, 100 kPa, and 27°C with a low inlet velocity and compressed to 600 kPa.

(*a*) Determine the exit temperature to make the work input and kinetic energy at the compressor exit have the minimum values.

(*b*) If the rate of work input into the compressor is measured to be a minimum with a value of 1000 kW, determine the compressor exit velocity, in m/s.

Step 1 of 8

Given data:

Mass flow rate is, $\dot{m} = 0.6\,kg/s$

Pressure at compressor inlet is, $P_1 = 100\,kPa$

Temperature at compressor inlet is, $T_1 = 27\,^{\circ}C = 300\,K$

Pressure at compressor exit is, $P_2 = 600\,kPa$

Work input is, $W_{in} = 1000\,kW$

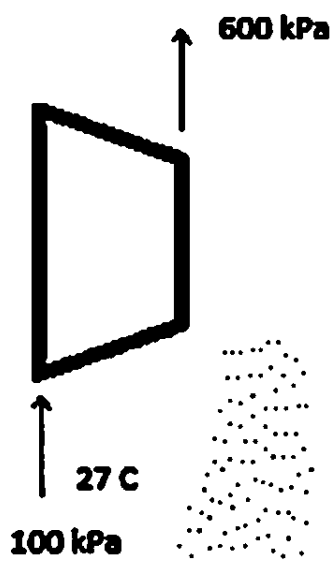
Step 2 of 8

From table A-2(a), properties of helium are,

$c_p = 5.1926\,kJ/kg^{\circ}K$

$k = 1.667$

Step 3 of 8



Step 4 of 8

(a) The temperature ratio of polytropic process is given by

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 300\times\left(\frac{600}{100}\right)^{\frac{1.667-1}{1.667}}$$

$$T_2 = 300\times(6)^{\frac{0.667}{1.667}}$$

$$T_2 = 300\times(6)^{0.4}$$

$$T_2 = 300\times 2.0476$$

$$\boxed{T_2 = 614.3\,K}$$

Step 5 of 8

(b) The energy balance for a steady-flow system is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For a steady flow system the change in energy is zero. $E_{sys} = 0$

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$m_1\left[h_1 + \left(\frac{V_1^2}{2}\right)\right] + W_{in} = m_2\left[h_2 + \left(\frac{V_2^2}{2}\right)\right]$$

Step 6 of 8

As only one inlet and exit are there the mass flow rate will be same $m_1 = m_2 = m$

$$m\left[h_1 + \left(\frac{V_1^2}{2}\right)\right] + W_{in} = m\left[h_2 + \left(\frac{V_2^2}{2}\right)\right]$$

$$h_1 + \left(\frac{V_1^2}{2}\right) + \frac{W_{in}}{m} = h_2 + \left(\frac{V_2^2}{2}\right)$$

$$\left(\frac{V_2^2}{2}\right) = h_1 - h_2 + \left(\frac{V_1^2}{2}\right) + \frac{W_{in}}{m}$$

$$V_2^2 = 2\times\left[h_1 - h_2 + \left(\frac{V_1^2}{2}\right) + \frac{W_{in}}{m}\right]$$

Step 7 of 8

As inlet velocity is very low, it should be negligible,

$$V_2^2 = 2\times\left[h_1 - h_2 + \frac{W_{in}}{m}\right]$$

$$V_2^2 = \left[2\times(h_1 - h_2) + \frac{2\times W_{in}}{m}\right]$$

In order to change enthalpy value from kJ/kg°K to m^2/s^2 we get

$$V_2^2 = 1000\times\left[2\times(h_1 - h_2) + \frac{2\times W_{in}}{m}\right]$$

$$V_2 = \sqrt{1000\times\left[2\times(h_1 - h_2) + \frac{2\times W_{in}}{m}\right]}$$

$$V_2 = \sqrt{1000\times\left[2\times c_p\times(T_1 - T_2) + \frac{2\times W_{in}}{m}\right]}$$

Step 8 of 8

$$V_2 = \sqrt{1000\times\left[2\times 5.1926\times(300 - 614.3) + \frac{2\times 1000}{0.6}\right]}$$

$$V_2 = \sqrt{1000\times\left[2\times 5.1926\times(-314.3) + 3333.33\right]}$$

$$V_2 = \sqrt{1000\times\left[-3264.07 + 3333.33\right]}$$

$$V_2 = \sqrt{1000\times 69.26}$$

$$V_2 = \sqrt{69260}$$

$$\boxed{V_2 = 263.17\,m/s}$$

Therefore the exit temperature is 614.3 K and the exit velocity is 263.17 m/s

Question: 8.099P

5-kg of air at 427°C and 600 kPa are contained in piston-cylinder device. The air expands adiabatically until the pressure is 100 kPa and produces 600 kJ of work output. Assume air has constant specific heats evaluated at 300 K.

(*a*) Determine the entropy change of the air, in kJ/kg-K I

(*b*) Since the process is adiabatic, is the process realistic? Using concepts of the second law, support your answer.

Step 1 of 5

Given

Mass of air is, $m = 5\ \text{kg}$

Initial pressure of air is, $P_1 = 600\ \text{kPa}$

Initial temperature of air is, $T_1 = 427\ ^\circ\text{C} = 700\ \text{K}$

Final pressure of air is, $P_2 = 100\ \text{kPa}$

Work done by the device is, $W_{dev} = 600\ \text{kJ}$

Step 2 of 5

From ideal gas specific heats of air at temperature 300 K are

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

The gas constant of air is

$R = 0.287\ \text{kJ/kg.K}$

Step 3 of 5

(a)

From first law of thermodynamics,

$dQ = dU + dW$

As work is done by the device, the equation can be written as

$dQ = dU + W_{dev}$

$dQ = m(u_2 - u_1) + W_{dev}$

As the process is adiabatic process, first term on right hand side will be zero.

$0 = m(u_2 - u_1) + W_{dev}$

$W_{dev} = m(u_1 - u_2)$

$W_{dev} = mc_v(T_1 - T_2)$

$$T_1 - T_2 = \frac{W_{dev}}{m \times c_v}$$

$$T_2 = T_1 - \frac{W_{dev}}{m \times c_v}$$

$$T_2 = 700 - \frac{600}{5 \times 0.718}$$

$$T_2 = 700 - \frac{600}{3.59}$$

$T_2 = 700 - 167.13$

$T_2 = 532.87\ \text{K}$

Step 4 of 5

The entropy change is given by

$$\Delta s = c_v \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$\Delta s = 1.005 \times \ln\left(\frac{532.87}{700}\right) - 0.287 \times \ln\left(\frac{100}{600}\right)$$

$\Delta s = 1.005 \times \ln(0.761) - 0.287 \times \ln(0.167)$

$\Delta s = 1.005 \times (-0.273) - 0.287 \times (-1.789)$

$\Delta s = -0.274365 - (-0.5134)$

$\Delta s = -0.274365 + 0.5134$

$\boxed{\Delta s = 0.239\ \text{kJ/kg.K}}$

Therefore the entropy change of the air is $\Delta s = 0.239\ \text{kJ/kg.K}$

Step 5 of 5

(b)

As the change in entropy is not zero (positive in this case) the process is irreversible.

Therefore the process is realistic as it is not reversible.

Question: 8.100P

A constant-volume tank contains 5 kg of air at 100 kPa and 327°C. The air is cooled to the surroundings temperature of 27°C. Assume constant specific heats at 300 K. (a) Determine the entropy change of the air in the tank during the process, in kJ/K, (*b*) determine the net entropy change of the universe due to this process, in kJ/K, and (*c*) sketch the processes for the air in the tank and the surroundings on a single *T-s* diagram. Besure to label the initial and final states for both processes.

Step 1 of 5

Given

Mass of air in the tank $m = 5\ \text{kg}$

Temperature of air in the tank $T_1 = 327°\text{C}$

$= 600°\text{K}$

Pressure of air in the tank $p_1 = 100\ \text{kPa}$

Temperature of surroundings $T_2 = 27°\text{C}$

$= 300°\text{K}$

Step 2 of 5

(a)

Entropy change of air in the tank, $\Delta s_{air} = mc_v \ln \frac{T_2}{T_1}$

Specific heat of air, $c_v = 0.718\ \text{kJ/kg K}$

$$\Delta s_{air} = (5\ \text{kg})(0.718\ \text{kJ/kg K})\ln\left(\frac{300\text{K}}{600\text{K}}\right)$$

$$\Delta s_{air} = -2.488\ \text{kJ/K}$$

Entropy change of the air in the tank during the process $\boxed{\Delta s_{air} = -2.488\ \text{kJ/K}}$

Step 3 of 5

(b)

From energy balance

$$Q_{out} = mc_v(T_2 - T_1)$$

$$Q_{out} = (5\ \text{kg})(0.718\ \text{kJ/kg K})(327 - 27)\text{K}$$

$$Q_{out} = 1077\ \text{kJ}$$

Entropy change of the surroundings

$$\Delta s_{surr} = \frac{Q_{out}}{T_{surr}}$$

$$\Delta s_{surr} = \frac{1077\text{kJ}}{300\ \text{K}}$$

$$\Delta s_{surr} = 3.59\ \text{kJ/K}$$

Step 4 of 5

Net entropy change of the universe due to this process

$$\Delta s_{total} = \Delta s_{air} + \Delta s_{surr}$$

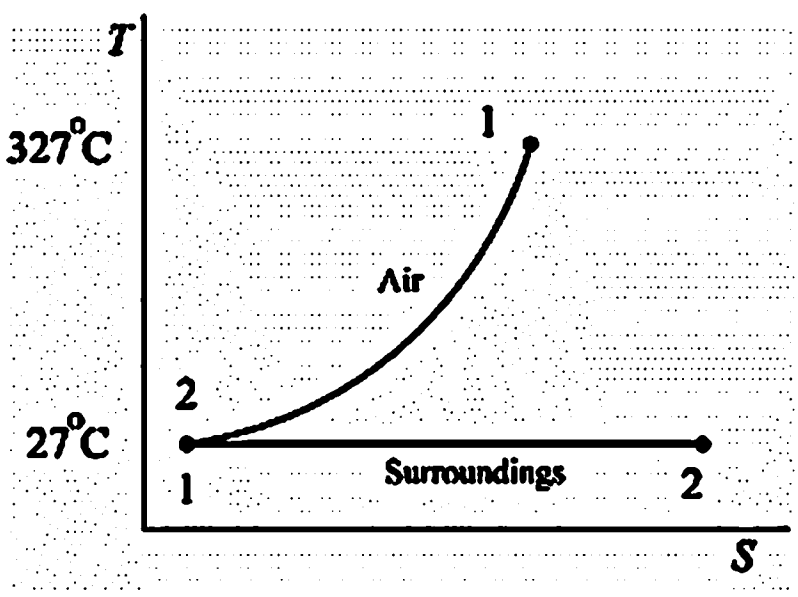
$$\Delta s_{total} = -2.488 + 3.59$$

$$\Delta s_{total} = 1.10\ \text{kJ/K}$$

Entropy change of the universe due to this process, $\boxed{\Delta s_{total} = 1.10\ \text{kJ/K}}$

Step 5 of 5

(c) Air in the tank and surroundings on a single T-s diagram



Question: 8.101P pg 1

A container filled with 45 kg of liquid water at 95°C is placed in a 90-m3 room that is initially at 12°C. Thermal equilibrium is established after a while as a result of heat ? transfer between the water and the air in the room. Using constant specific heats, determine (*a*) the final equilibrium temperature, (*b*) the amount of heat transfer between the water and the air in the room, and (*c*) the entropy generation.Assume the room is well sealed and heavily insulated.

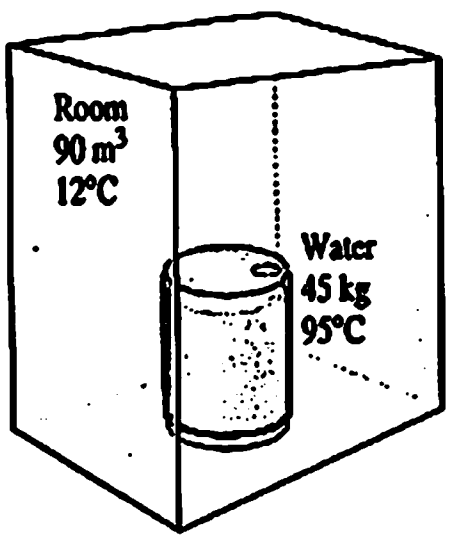


FIGURE P8–101

Step 1 of 7

Obtain the following properties of air from table A-2, "Ideal-gas specific heats of various common gases" in the textbook.

Gas constant for air, $R = 0.287\ \text{kJ/kg.K}$

Specific heat of air at constant pressure, $c_p = 1.005\ \text{kJ/kg.K}$

Specific heat of air at constant volume, $c_v = 0.718\ \text{kJ/kg.K}$

Obtain the specific heat of water from table, "Properties of common liquids, solids, and foods" from the textbook.

$c_{pw} = 4.18\ \text{kJ/kg.K}$

Calculate the mass of air in the room using the following relation.

$$m_{air} = \frac{PV}{RT_1}$$

Here, P is the pressure in the room (atmospheric pressure), V is the volume of the room, R is the gas constant, and T_1 initial room temperature.

Substitute $101.325\,\text{kPa}$ for P, $90\ \text{m}^3$ for V, $0.287\ \text{kJ/kg.K}$ for R, and $12°\text{C}$ for T_1.

$$m_{air} = \frac{(101.325)(90)}{(0.287)(12+273)}$$
$$= 111.5\ \text{kg}$$

Step 2 of 7

(a)

Calculate the final equilibrium temperature by applying energy balance considering the room as the system.

$$E_{in} - E_{out} = \Delta E_{system}$$
$$m_w c_{pw}(T_2 - T_w) + m_{air} c_v (T_2 - T_1) = 0$$

Here, m_w is the mass of water, and T_w is the initial temperature of water.

Substitute 45 kg for m_w, $95°\text{C}$ for T_w, $111.5\ \text{kg}$ for m_{air}, $4.18\ \text{kJ/kg.K}$ for c_{pw}, $0.718\ \text{kJ/kg.K}$ for c_v, and $12°\text{C}$ for T_1.

$$45(4.18)(T_2 - 95) + (111.5)(0.718)(T_2 - 12) = 0$$
$$188.1T_2 - 17869.5 + 80.057T_2 - 960.684 = 0$$
$$268.157T_2 = 18830.184$$
$$T_2 = 70.22°\text{C}$$

Therefore the final equilibrium temperature is $\boxed{T_2 = 70.22°\text{C}}$.

Step 3 of 7

(b)

Calculate the amount of heat transferred from water to air as

$$Q = m_{air} c_v (T_2 - T_1)$$

Substitute $111.5\ \text{kg}$ for m_{air}, $0.718\ \text{kJ/kg.K}$ for c_v, $70.22°\text{C}$ for T_2, and $12°\text{C}$ for T_1.

$$Q = 111.5(0.718)(70.22 - 12)$$
$$= 4660.918\ \text{kJ}$$

Hence, the amount of heat transfer between water and air is $\boxed{Q = 4660.918\ \text{kJ}}$.

Step 4 of 7

(c)

Calculate the entropy change for water.

$$\Delta s_w = m_w c_{pw} \ln \frac{T_2}{T_w}$$

Substitute 45 kg for m_w, $95°\text{C}$ for T_w, $4.18\ \text{kJ/kg.K}$ for c_{pw}, and $70.22°\text{C}$ for T_2.

$$\Delta s_w = 45(4.18) \ln \frac{70.22+273}{95+273}$$
$$\Delta s_w = -13.11\ \text{kJ/K}$$

Step 5 of 7

Calculate the final pressure of air in the room using the ideal gas equation.

$$P_2 = \frac{mRT_2}{}$$

Step 5 of 7

Calculate the final pressure of air in the room using the ideal gas equation.

$$P_2 = \frac{mRT_2}{V}$$

Substitute $90\ \text{m}^3$ for V, $0.287\ \text{kJ/kg.K}$ for R, $70.22°\text{C}$ for T_2, and 111.5 kg for m.

$$P_2 = \frac{111.5 \times 0.287 \times (70.22 + 273)}{90}$$
$$= 122.035\,\text{kPa}$$

Step 6 of 7

Calculate the entropy change for air.

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

Substitute $0.718\ \text{kJ/kg.K}$ for c_v, $70.22°\text{C}$ for T_1, $12°\text{C}$ for T_1, $1.005\ \text{kJ/kg.K}$ for c_p, 101.3 kPa for P_1, and $122.035\,\text{kPa}$ for P_2.

$$\Delta S_{air} = m_a\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$
$$= (111.5)\left((1.005)\ln\frac{70.22+273}{12+273} - (0.287)\ln\frac{122.035}{101.3}\right)$$
$$\Delta S_{air} = 14.87\ \text{kJ/K}$$

Step 7 of 7

Write the entropy balance for whole system (insulated room) considering both water and air.

$$\Delta s_{system} = \int\frac{\delta Q}{T} + s_{gen}$$
$$(\Delta s_{air} + \Delta s_{water}) = \int\frac{\delta Q}{T} + s_{gen}$$

As the room is heavily insulated and well-sealed, there is no heat transfer between the system and surroundings, and the amount of entropy generation is equal to the sum of entropy change of water and air.

$$s_{gen} = (\Delta s_{air} + \Delta s_{water})$$

Substitute $-13.11\ \text{kJ/K}$ for Δs_w, and $14.87\ \text{kJ/K}$ for ΔS_{air}.

$$S_{gen} = -13.1 + 14.88$$
$$S_{gen} = 1.78\ \text{kJ/K}$$

Hence, the amount of entropy generation during the process is $\boxed{S_{gen} = 1.78\ \text{kJ/K}}$.

Question: 8.102P pg 1

An ideal gas at 100 kPa and 15°C enters a steady-flow compressor. The gas is compressed to 600 kPa, and jq percent of the mass that entered the compressor is removed for some other use. The remaining 90 percent of the inlet gas is compressed to 800 kPa before leaving the compressor. The entire compression process is assumed to be reversible and adiabatic. The power supplied to the compressor is measured to be 32 kW. If the ideal gas has constant specific heats such that *cv* = 0.8 kJ/kg-K and *cp* = 1.1 kJ/kg-K, (*a*) sketch the compression process on a *T-s* diagram, (*b*) determine the temperature of the gas at the two compressor exits, in K, and (c) determine the mass flow rate of the gas into the compressor, in kg/s.

Step 1 of 6

Given

Pressure of the gas at compressor inlet is, $P_1 = 100\ \text{kPa}$

Temperature of the gas at compressor inlet is, $T_1 = 15°\text{C} = 288\ \text{K}$

Intermediate pressure is, $P_2 = 600\ \text{kPa}$

Exit pressure is, $P_3 = 800\ \text{kPa}$

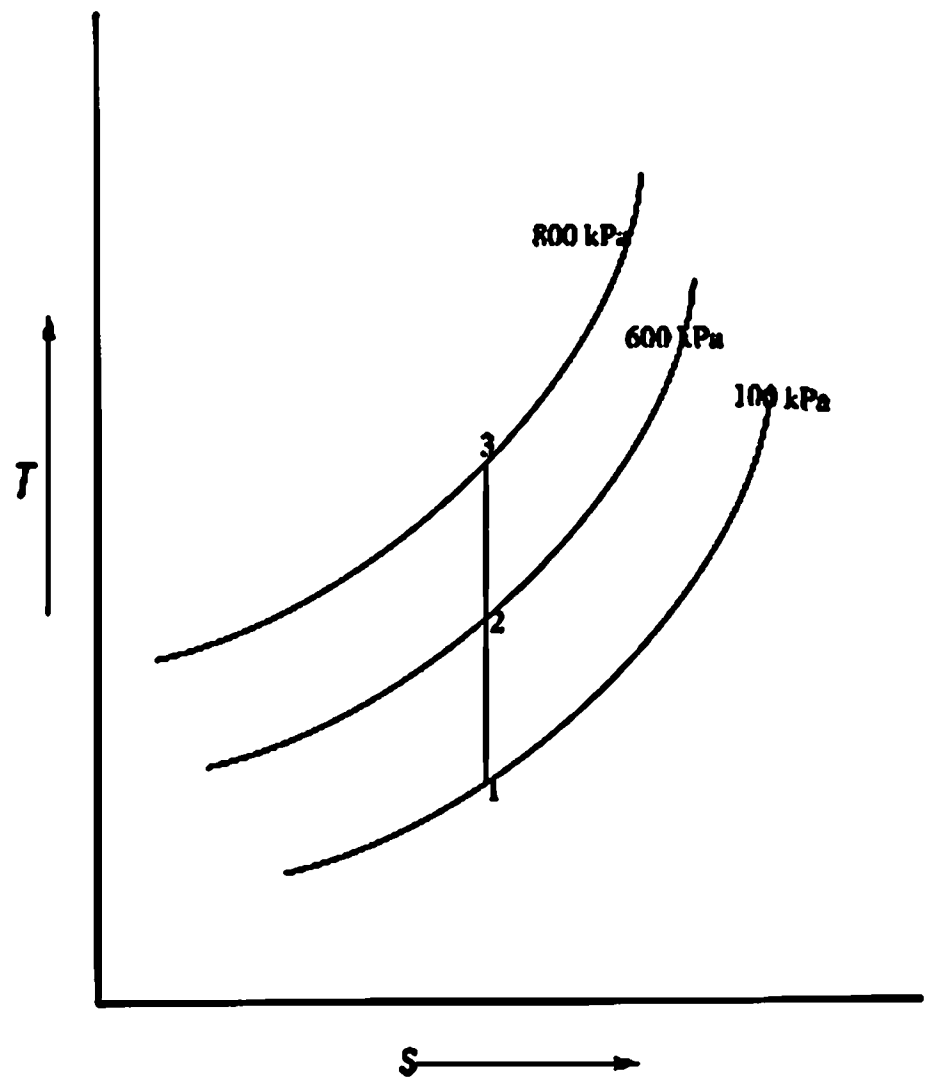
Power supplied to the compressor is, $W_{in} = 32\ \text{kW}$

Specific heat at constant volume is, $c_v = 0.8\ \text{kJ/kg.K}$

Specific heat at constant pressure is, $c_p = 1.1\ \text{kJ/kg.K}$

Step 2 of 6

(a)



Step 3 of 6

(b)

The ratio of specific heats can be calculated as

$$k = \frac{c_p}{c_v}$$

$$k = \frac{1.1}{0.8}$$

$$k = 1.375$$

As given, the process is assumed to be isentropic (reversible and adiabatic). So, the ratio of temperatures of intermediate and inlet is given by

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 288 \times \left(\frac{600}{100}\right)^{\frac{1.375-1}{1.375}}$$

$$T_2 = 288 \times (6)^{\frac{0.375}{1.375}}$$

$$T_2 = 288 \times (6)^{0.2727}$$

$$T_2 = 288 \times 1.63$$

$$\boxed{T_2 = 469.44\ \text{K}}$$

Therefore the intermediate temperatures is given by $T_2 = 469.44\ \text{K}$

Step 4 of 6

As given, the process is assumed to be isentropic (reversible and adiabatic). So, the ratio of temperatures of exit and inlet is given by

$$\frac{T_3}{T_1} = \left(\frac{P_3}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_3 = T_1 \times \left(\frac{P_3}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_3 = 288 \times \left(\frac{800}{100}\right)^{\frac{1.375-1}{1.375}}$$

$$T_3 = 288 \times (8)^{\frac{0.375}{1.375}}$$

$$T_3 = 288 \times (8)^{0.2727}$$

$$T_3 = 288 \times 1.763$$

$$\boxed{T_3 = 507.744\ \text{K}}$$

Therefore the exit temperatures is given by $T_3 = 507.744\ \text{K}$

Step 5 of 6

(c)

Let the mass entered the compressor be, m_1

Mass removed at intermediate stage is,

$m_2 = 10\ \%$ of m_1

$$m_2 = \frac{10}{\quad} \times m_1$$

(c)

Let the mass entered the compressor be, m_1

Mass removed at intermediate stage is,

$m_2 = 10\%$ of m_1

$$m_2 = \frac{10}{100} \times m_1$$

$$m_2 = 0.1 \times m_1$$

The remaining mass entered into the compressor is,

$m_3 = 90\%$ of m_1

$$m_3 = \frac{90}{100} \times m_1$$

$$m_3 = 0.9 \times m_1$$

By applying energy balance to the system

$$E_{in} - E_{out} = \frac{dE_{sys}}{dt}$$

The process is steady and so the right hand side in the above equation becomes zero. So,

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out} \qquad \ldots\ldots (1)$$

Work input is given to the compressor the energy transfer inlet is given by

$$E_{in} = m_1 \times h_1 + W_{in}$$

$$E_{in} = m_1 \times c_p \times T_1 + W_{in} \qquad \ldots\ldots (2)$$

The energy transfer out is given by

$$E_{out} = m_2 \times h_2 + m_3 \times h_3$$

$$E_{out} = m_2 \times c_p \times T_2 + m_3 \times c_p \times T_3 \qquad \ldots\ldots (3)$$

Step 6 of 6

Substituting the equations (2) and (3) in equation (1), we get

$$E_{in} = E_{out}$$

$$m_1 \times c_p \times T_1 + W_{in} = m_2 \times c_p \times T_2 + m_3 \times c_p \times T_3$$

Substituting the values of m_2 and m_3, then the above equation becomes

$$m_1 \times c_p \times T_1 + W_{in} = 0.1\, m_1 \times c_p \times T_2 + 0.9 m_1 \times c_p \times T_3$$

$$W_{in} = 0.1\, m_1 \times c_p \times T_2 + 0.9 m_1 \times c_p \times T_3 - m_1 \times c_p \times T_1$$

$$W_{in} = m_1 \times c_p \times (0.1T_2 + 0.9T_3 - T_1)$$

$$m_1 = \frac{W_{in}}{c_p \times (0.1T_2 + 0.9T_3 - T_1)}$$

Substituting the known values we get,

$$m_1 = \frac{32}{1.1 \times (0.1 \times 469.44 + 0.9 \times 507.744 - 288)}$$

$$m_1 = \frac{32}{1.1 \times (46.944 + 456.969 - 288)}$$

$$m_1 = \frac{32}{1.1 \times 215.913}$$

$$m_1 = \frac{32}{237.504}$$

$$\boxed{m_1 = 0.1347 \text{ kg/s}}$$

Therefore the mass flow rate of the gas into the compressor is $m_1 = 0.1347$ kg/s.

Question: 8.103P

The well-insulated container shown in Fig. P8-103E is initially evacuated. The supply line contains air that is maintained at 200 psia and 100°F. The valve is opened until the pressure in the container is the same as the pressure in the supply line. Determine the minimum temperature in the container when the valve is closed.

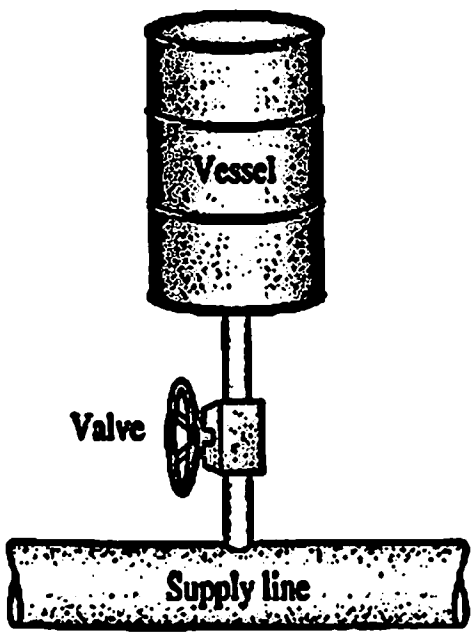


FIGURE P8–103E

Step 1 of 3

Given

Pressure of air in the supply line $p_1 = 200$ psia

Temperature of air in the supply line $T_1 = 100°F$

Pressure in the container be p_2

The valve is opened so $p_2 = p_1$

Step 2 of 3

From the entropy relation, $s_2 - s_1 = c_p \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1}$ ---- (1)

But entropy either increases or remains constant so, $s_2 - s_1 \geq 0$

Hence for minimum temperature, eq (1) becomes

$$c_p \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1} = 0$$

$$c_p \ln \frac{T_2}{T_1} = T \ln \frac{p_2}{p_1}$$

$$c_p \ln \frac{T_2}{T_1} = T \ln \frac{p_1}{p_1}$$

$$c_p \ln \frac{T_2}{T_1} = R \ln (1)$$

$$c_p \ln \frac{T_2}{T_1} = 0$$

Step 3 of 3

But c_p cannot be zero hence

$$\ln \frac{T_2}{T_1} = 0$$

$$\ln \left(\frac{T_2}{T_1} \right) = \ln (1)$$

$$\frac{T_2}{T_1} = 1$$

$$T_2 = T_1$$

$$T_2 = 100°F$$

Minimum temperature in the container is $\boxed{T_2 = 100°F}$

Question: 8.104P

In large compressors, the gas is frequently cooled while being compressed to reduce the power consumed by the compressor. Explain how cooling the gas during a compression process reduces the power consumption.

Step 1 of 4

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Step 2 of 4

The larger the specific volume the larger the reversible work produced or consumed by the steady-flow device. This conclusion is equally valid for actual steady-flow devices. Therefore, every effort should be made to keep the specific volume of a fluid as small as possible during a compression process to minimize the work input.

Step 3 of 4

Cooling a gas during compression will reduce its specific volume. So, cooling the gas during a compression process reduces the power consumption.

Step 4 of 4

Question: 8.105P

The turbines in steam power plants operate essentially under adiabatic conditions. A plant engineer suggests to end this practice. She proposes to run cooling water through the outer surface of the casing to cool the steam as it flows through the turbine. This way, she reasons, the entropy of the steam will decrease, the performance of the turbine will improve, and as a result the work output of the turbine will increase. How would you evaluate this proposal?

Step 1 of 1

The larger the specific volume the larger the reversible work produced or consumed by the steady-flow device. This conclusion is equally valid for actual steady-flow devices. Therefore, every effort should be made to keep the specific volume of a fluid as large as possible during an expansion process to maximize the work output.

Cooling a gas during expansion will reduce its specific volume. So, cooling the gas during an expansion process reduces the work output. So, it is not a good proposal.

Question: 8.106P

It is well known that the power consumed by a compressor can be reduced by cooling the gas during compression. Inspired by this, somebody proposes to cool the liquid as it flows through a pump, in order to reduce the power consumption of the pump. Would you support this proposal? Explain.

Step 1 of 1

The larger the specific volume the larger the reversible work produced or consumed by the steady-flow device. This conclusion is equally valid for actual steady-flow devices. Therefore, every effort should be made to keep the specific volume of a fluid as small as possible during a compression process to minimize the work input.

But, cooling a liquid during compression will not reduce its specific volume significantly. So, cooling a liquid during a compression process will not reduce the work input. So, it is not a good proposal.

Question: 8.107P

Air is compressed isothermally from 13 psia and 90°F to 80 psia in a reversible steady-flow device. Calculate the work required, in Btu/lbm, for this compression.

Step 1 of 2

Given

Inlet pressure is, $P_1 = 13$ psia

Isothermal temperature is, $T = 90°F = 549.67$ R

Exit pressure is, $P_2 = 80$ psia

Process is isothermal compression process

Step 2 of 2

From properties of molar mass, gas constant and critical properties table the gas constant of air is

$R = 0.06855$ Btu/lbm.R

Work done during the isothermal compression process is

$$W_{in} = \int_1^2 v dP$$

$$= \int_1^2 \frac{RT}{P} dP$$

$$= RT \times \int_1^2 \frac{1}{P} dP$$

$$W_{in} = RT \times \int_1^2 \frac{1}{P} dP$$

$$W_{in} = RT \times \left[\ln (P)\right]_1^2$$

$$= RT \times \left[\ln P_2 - \ln P_1\right]$$

$$= RT \times \ln\left(\frac{P_2}{P_1}\right)$$

$$W_{in} = 0.06855 \times 549.67 \times \ln\left(\frac{80}{13}\right)$$

$$= 0.06855 \times 549.67 \times \ln(6.154)$$

$$W_{in} = 0.06855 \times 549.67 \times 1.817$$

$$\boxed{W_{in} = 68.464 \text{ Btu/lbm}}$$

Therefore the required work input for this compression process is 68.464 Btu/lbm

Question: 8.108P

Saturated water vapor at 150°C is compressed in a reversible steady-flow device to 1000 kPa while its specific volume remains constant. Determine the work required, in kJ/kg.

Step 1 of 5

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Step 2 of 5

Given:

Initial conditions:

Temperature, $T_1 = 150^\circ C \Rightarrow T_1 = 423\ \text{K}$

Initially it is saturated vapor

From Table A- 4, at $T_1 = 150^\circ C$ and $x_1 = 1$

Pressure, $P_1 = 476.16\ \text{kPa}$

Specific volume, $V_1 = 0.39248\ \text{m}^3/\text{kg}$

Initial conditions:

Pressure, $P_2 = 1000\ \text{kPa}$

Specific volume remains constant $\Rightarrow V_2 = 0.39248\ \text{m}^3/\text{kg}$

Step 3 of 5

Step 4 of 5

Work required, $W = \int_1^2 V dP$

$\Rightarrow W = V \times (P_2 - P_1)$

$\Rightarrow W = 0.39248 \times (1000 - 476.16)$

$\Rightarrow \boxed{W = 205.597\ kJ/kg}$

Step 5 of 5

Question: 8.109P

Calculate the work produced, in Btu/lbm, for the reversible steady-flow process 1-3 shown in Fig. P8-109E.

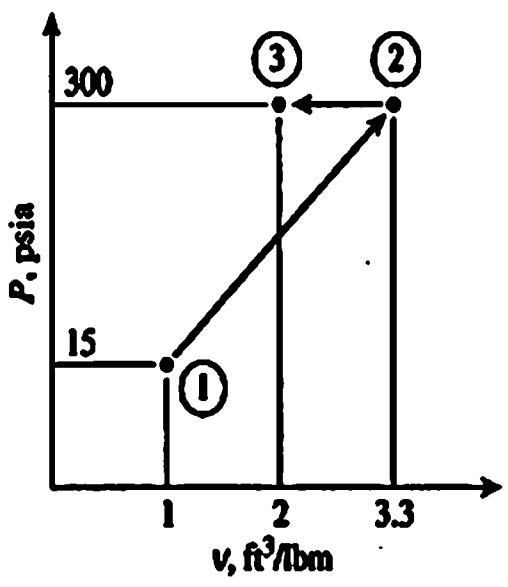


FIGURE P8–109E

Step 1 of 4

Given

Pressure at state 1 is, $P_1 = 15$ psia

Pressure at state 2 is, $P_2 = 300$ psia

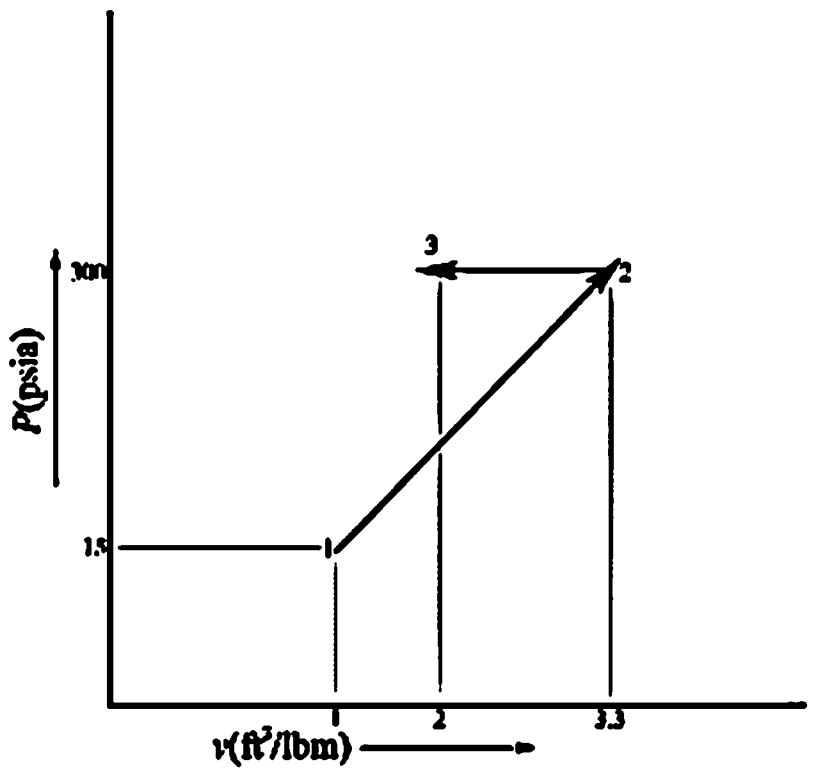
Pressure at state 3 is, $P_3 = 300$ psia

Volume at state 1 is, $v_1 = 1.0\ \text{ft}^3/\text{lbm}$

Volume at state 2 is, $v_2 = 3.3\ \text{ft}^3/\text{lbm}$

Volume at state 3 is, $v_3 = 2.0\ \text{ft}^3/\text{lbm}$

Step 2 of 4



Step 3 of 4

The total work done is given by

$$W_{tot} = W_{1\text{-}2} + W_{2\text{-}3}$$

Work done in a steady flow process is given by

$$W_{sb} = \int_a^b v dP$$

Therefore the equation reduces to

$$W_{tot} = \int_1^2 v dP + \int_2^3 v dP$$

As pressure is constant in process 2-3, change in pressure in that process will be zero

$$W_{tot} = \int_1^2 v dP + \int_2^3 v \times 0$$

$$W_{tot} = \int_1^2 v dP + 0$$

$$W_{tot} = \int_1^2 v dP$$

$$W_{tot} = v \times \int_1^2 dP$$

$$W_{tot} = v \times [P]_1^2$$

$$W_{tot} = v \times [P_2 - P_1]$$

Step 4 of 4

Volume can be taken as average volume. So,

$$W_{tot} = \frac{v_1 + v_2}{2} \times (P_2 - P_1)$$

$$W_{tot} = \frac{1 + 3.3}{2} \times (300 - 15)$$

$$W_{tot} = \frac{4.3}{2} \times 285$$

$$W_{tot} = 2.15 \times 285$$

$$W_{tot} = 612.75\ \text{psia.ft}^3/\text{lbm}$$

We know that

$1\ \text{Btu} = 5.404\ \text{psia.ft}^3$.

So we can write as

$$1\ \text{psia.ft}^3 = \frac{1}{5.404}\ \text{Btu}$$

Then

$$W_{tot} = 612.75 \times \frac{1}{5.404}\ \text{Btu/lbm}$$

$$\boxed{W_{tot} = 113.388\ \text{Btu/lbm}}$$

Therefore the total work done in the reversible steady flow process 1-3 is

$W_{tot} = 113.388\ \text{Btu/lbm}$.

Question:

8.110P

Calculate the work produced, in kJ/kg, for the reversible isothermal, steady-flow process 1-3 shown in Fig. P8-110 when the working fluid is an ideal gas.

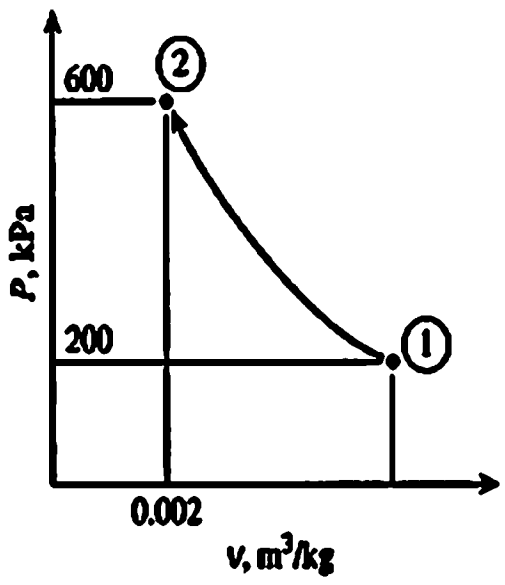


FIGURE P8–110

Question:

Calculate the work produced, in kJ/kg, for the reversible isothermal, steady-flow process 1-3 shown in Fig. P8-110 when the working fluid is an ideal gas.

FIGURE P8–110

Step 2 of 4

The work produced in a process is given by the area under the $P-v$ diagram. So the work produced in the process 1-2 is given by

$$w_{1\text{-}2} = -\int_1^2 v\,dP$$

For an ideal gas, the volume is given by

$$v = \frac{RT}{P}$$

Therefore work produced can be written as

$$w_{1\text{-}2} = -\int_1^2 \frac{RT}{P}\,dP$$

$$w_{1\text{-}2} = -RT\int_1^2 \frac{dP}{P}$$

$$w_{1\text{-}2} = -RT\times\ln\left(\frac{P_2}{P_1}\right)$$

$$w_{1\text{-}2} = -P_2v_2\times\ln\left(\frac{P_2}{P_1}\right)$$

Step 4 of 4

Substituting the known values in the above equation

$$w_{1\text{-}2} = -600\times0.002\times\ln\left(\frac{600}{200}\right)$$

$$w_{1\text{-}2} = -600\times0.002\times\ln(3)$$

$$w_{1\text{-}2} = -1.2\times\ln(3)$$

$$w_{1\text{-}2} = -1.2\times1.0986$$

$$\boxed{w_{1\text{-}2} = -1.31832\ \text{kJ/kg}}$$

The work produced is negative, therefore work is given to the system in the process 1-2 is

$$w_{1\text{-}2} = 1.31832\ \text{kJ/kg}$$

Question: 8.111P

Liquid water enters a 25-kW pump at 100-kPa pressure at a rate of 5 kg/s. Determine the highest pressure the liquid water can have at the exit of the pump. Neglect the kinetic and potential energy changes of water, and take the specific volume of water to be 0.001 m3 /kg.

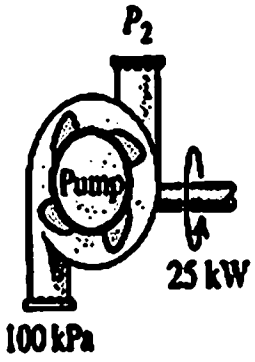


FIGURE P8–111

Step 1 of 2

Given:

Initial Pressure $P_1 = 100$ kPa

Question:

Liquid water enters a 25-kW pump at 100-kPa pressure at a rate of 5 kg/s. Determine the highest pressure the liquid water can have at the exit of the pump. Neglect the kinetic and potential energy changes of water, and take the specific volume of water to be 0.001 m3 /kg.

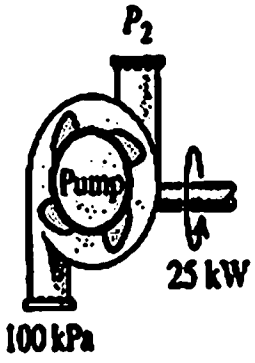


FIGURE P8–111

Question: 8.112P

Consider a steam power plant that operates between the pressure limits of 5 MPa and 10 kPa. Steam enters the pump as saturated liquid and leaves the turbine as saturated vapor. Determine the ratio of the work delivered by the turbine to the work consumed by the pump. Assume the entire cycle to be reversible and the heat losses from the pump and the turbine to be negligible.

Step 1 of 4

Given data:

Lower pressure limit is, $P_1 = P_4 = 10\ \text{kPa}$

Higher pressure limit is, $P_2 = P_3 = 5\ \text{MPa}$

$= 5000\ \text{kPa}$

Inlet to pump is saturated liquid steam

Exit to turbine is saturated vapor steam

Step 2 of 4

From properties of saturated water

At pressure $P_4 = 10\ \text{kPa}$

$$h_4 = h_g = 2583.9\ \text{kJ/kg}$$

$$s_4 = s_g = 8.1488\ \text{kJ/kg}\cdot\text{K}$$

Specific volume, $v_1 = v_f = 0.001010\ \text{m}^3/\text{kg}$

As no heat loss in pump and turbine, compression and expansion are adiabatic process.

As the processes are reversible also, they are isentropic process

Therefore,

$$s_3 = s_4 = 8.1488\ \text{kJ/kg}\cdot\text{K}$$

As $s_3 = 8.1488\ \text{kJ/kg°K}$ is greater than the saturated entropy at pressure $P_3 = 5\ \text{MPa}$, the steam at that condition is superheated state.

From properties of saturated water

At pressure $P_3 = 5\ \text{MPa}$

The enthalpy at that state is interpolated as

$$\left(\frac{h_3 - 4380.2}{4628.3 - 4380.2}\right) = \left(\frac{8.1488 - 7.9619}{8.1648 - 7.9619}\right)$$

$$\left(\frac{h_3 - 4380.2}{248.1}\right) = \left(\frac{0.1869}{0.2029}\right)$$

$$\left(\frac{h_3 - 4380.2}{248.1}\right) = 0.9211$$

$$h_3 = (0.9211 \times 248.1) + 4380.2$$

$$h_3 = 228.53 + 4380.2$$

$$h_3 = 4608.73\ \text{kJ/kg}$$

Step 3 of 4

We know that

Work done by isentropic turbine

$$w_{tur} = (h_3 - h_4)$$

$$w_{tur} = (4608.73 - 2583.9)$$

$$w_{tur} = 2024.83\ \text{kJ/kg}$$

Work done on isentropic pump is given by

$$w_{pump} = \int_1^2 v dP + \Delta KE + \Delta PE$$

As given, the kinetic and potential energies are neglected.

$$w_{pump} = \int_1^2 v dP$$

$$w_{pump} = v\int_1^2 1 \times dP$$

$$w_{pump} = v[P]_1^2$$

$$w_{pump} = v(P_2 - P_2)$$

$$w_{pump} = 0.001010 \times (5000 - 10)$$

$$w_{pump} = 0.001010 \times 4990$$

$$w_{pump} = 5.0399\ \text{kJ/kg}$$

Step 4 of 4

Now we know that

The ratio of turbine work to the pump work

Step 3 of 4

We know that

Work done by isentropic turbine

$$w_{tur} = (h_3 - h_4)$$

$$w_{tur} = (4608.73 - 2583.9)$$

$$w_{tur} = 2024.83\ \text{kJ/kg}$$

Work done on isentropic pump is given by

$$w_{pump} = \int_1^2 v dP + \Delta KE + \Delta PE$$

As given, the kinetic and potential energies are neglected.

Question: 8.113P



Reconsider Prob. 8-112. Using EES (or other) software, investigate the effect of the quality of the steam at the turbine exit on the net work output. Vary the quality from 0.5 to 1.0, and plot the net work output as a function of this quality.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 8.114P

Liquid water at 120 kPa enters a 7-kW pump where its pressure is raised to 5 MPa. If the elevation difference between the exit and the inlet levels is 10 m, determine the highest mass flow rate of liquid water this pump can handle. Neglect the kinetic energy change of water, and take the specific volume of water to be 0.001 m3 /kg.

Step 1 of 2

Given:

Pressures, $P_1 = 120\ \text{kPa}$ and $P_2 = 5\ \text{MPa}$

Power, $\dot{W} = 7\ \text{kW}$

Elevation difference, $\Delta z = 10\ \text{m}$

Mass flow rate, $= \dot{m}$

So, potential energy changes may be significant.

Specific volume of water, $V = 0.001\ \text{m}^3/\text{kg}$

Step 2 of 2

$$\text{Power, } \dot{W} = \int_1^2 \dot{m} \times V \times dP + \dot{m} \times g \times \Delta z$$

$$\Rightarrow \dot{W} = \dot{m} \times V (P_2 - P_1) + \dot{m} \times g \times \Delta z$$

$$\Rightarrow 7 = \dot{m} \times 0.001(5000 - 120) + \dot{m} \times 9.8 \times 10 \times \left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2} \right)$$

$$\Rightarrow 7 = \dot{m} \times 4.88 + \dot{m} \times 0.098$$

$$\Rightarrow 7 = \dot{m} \times 4.978$$

$$\Rightarrow \dot{m} = \frac{7}{4.978}$$

So, Mass flow rate, $\boxed{\dot{m} = 1.4062\ \text{kJ/s}}$

Question: 8.115P

Describe the ideal process for an (*a*) adiabatic turbine, (*b*) adiabatic compressor, and (c) adiabatic nozzle, and define the isentropic efficiency for each device.

Step 1 of 4

There is some heat transfer between these devices and the surrounding medium is unavoidable, many steady-flow devices are intended to operate under adiabatic conditions. Therefore, the ideal process for these devices should be an adiabatic or isentropic process. Furthermore, an ideal process should involve no irreversibilities since the effect of irreversibilities is always to downgrade the performance of engineering devices. Thus, the ideal process that can serve as a suitable model for adiabatic steady-flow devices is the isentropic process.

Step 2 of 4

(a) Isentropic efficiency of adiabatic turbine,

$$\eta_{turbine} = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}}$$

Step 3 of 4

(b) Isentropic efficiency of adiabatic compressor,

$$\eta_{compressor} = \frac{\text{Isentropic compressor work}}{\text{Actual compressor work}}$$

Step 4 of 4

(c) Isentropic efficiency of adiabatic nozzle,

$$\eta_{nozzle} = \frac{\text{Actual Kinetic energy at nozzle exit}}{\text{Isentropic Kinetic energy at nozzle exit}}$$

Question: 8.116P

Is the isentropic process a suitable model for compressors that are cooled intentionally? Explain.

Step 1 of 1

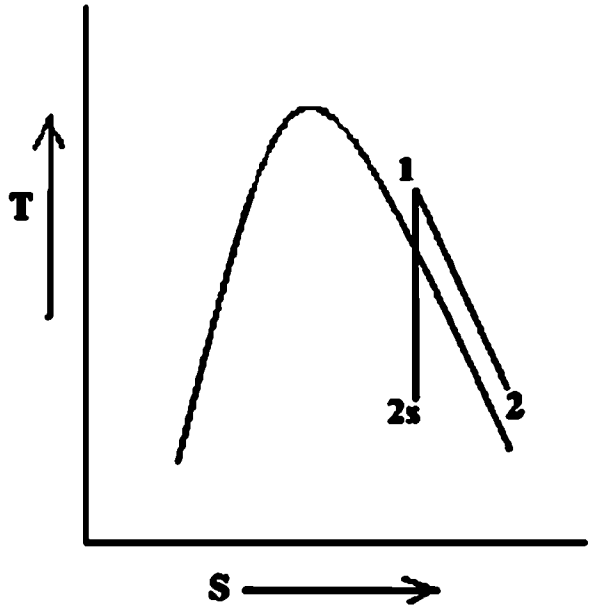
Isentropic process is not the ideal process for compressors that are cooled intentionally
Sometimes, compressors are cooled intentionally by utilizing fins or a water jacket placed around the casing to reduce the work input requirements. In this case, the isentropic process is not suitable as the model process since the device is no longer adiabatic. A realistic model process for compressors that are intentionally cooled during the compression process is the reversible isothermal process.

Question: 8.117P

On a *T-s* diagram, does the actual exit state (state 2) of an adiabatic turbine have to be on the right-hand side of the isentropic exit state (state 2 *s*)? Why?

Step 1 of 1

In actual process the entropy of the exit state increases due to irrversibilities. That is the reason the actual exit state will be the right hand side of the isentropic exit state
In the diagram below process 1-2s is isentropic expansion
Process 1-2 is actual process



Question: 8.118P

Steam at 100 psia and 650°F is expanded adiabatically in a closed system to 10 psia. Determine the work produced, in Btu/lbm, and the final temperature of steam for an isentropic expansion efficiency of 80 percent.

Step 1 of 7

Given

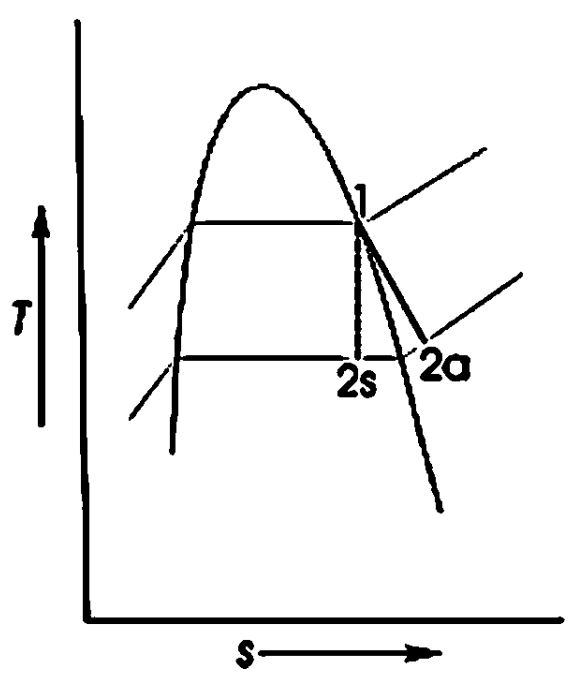
Initial pressure is, $P_1 = 100$ psia

Initial temperature is, $T_1 = 650°F = 1109.67$ R

Final pressure is, $P_2 = 10$ psia

Isentropic efficiency is, $\eta_{isen} = 80\% = 0.8$

Step 2 of 7



Step 3 of 7

As the process is occurring in closed system (non-flow process) we have to take internal energy instead of enthalpy.

From superheated water tables at pressure $P_1 = 100$ psia and $T_1 = 650°F$ by using interpolation

$$u_1 = 1214.4 + (1253.0 - 1214.4)\left(\frac{650 - 600}{700 - 600}\right)$$

$$u_1 = 1233.7 \text{ Btu/lbm}$$

$$s_1 = 1.7586 + (1.8037 - 1.7586)\left(\frac{650 - 600}{700 - 600}\right)$$

$$s_1 = 1.78115 \text{ Btu/lbm.R}$$

Step 4 of 7

From saturated water tables at pressure $P_2 = 10$ psia

Internal energy of saturated liquid is $u_{f2} = 161.22$ Btu/lbm

Internal energy of evaporation is $u_{fg2} = 910.75$ Btu/lbm

Entropy of saturated liquid is $s_{f2} = 0.28362$ Btu/lbm.R

Entropy of evaporation is $s_{fg2} = 1.50391$ Btu/lbm.R

For an isentropic process 1-2s entropy remains constant so,

$s_{2s} = s_1$

We know

$s_{2s} = s_{f2} + x_{2s} \times s_{fg2}$

Then

$s_{f2} + x_{2s} \times s_{fg2} = s_1$

$$x_{2s} = \frac{s_1 - s_{f2}}{s_{fg2}}$$

$$x_{2s} = \frac{1.78115 - 0.28362}{1.50391}$$

$$x_{2s} = \frac{1.49753}{1.50391}$$

$$x_{2s} = 0.99575$$

The internal energy at the end of isentropic process is given by

$u_{2s} = u_{f2} + x_{2s} \times u_{fg2}$

$u_{2s} = 161.22 + 0.99575 \times 910.75$

$u_{2s} = 161.22 + 906.879$

$u_{2s} = 1068.099$ Btu/lbm

Step 5 of 7

The work done in the isentropic process 1-2s is given by

$w_s = u_1 - u_{2s}$

$w_s = 1233.7 - 1068.099$

$w_s = 165.601$ Btu/lbm

The isentropic efficiency is given by

$$\eta_{isen} = \frac{\text{Actual work done by the turbine}}{\text{Isentropic work done by the turbine}}$$

$$\eta_{isen} = \frac{w_a}{w_s}$$

$w_a = \eta_{isen} \times w_s$

$w_a = 0.8 \times 165.601$

$\boxed{w_a = 132.4808 \text{ Btu/lbm}}$

Step 6 of 7

The actual work is also given by

$w_a = u_1 - u_{2a}$

So the actual final internal energy is given by

$u_{2a} = u_1 - w_a$

$u_{2a} = 1233.7 - 132.4808$

$u_{2a} = 1101.2192$ Btu/lbm

Step 7 of 7

At $P_2 = 10$ psia and $u_{2a} = 1101.2192$ Btu/lbm from superheated water tables by using interpolation

$$T_{2a} = 240 + (280 - 240)\left(\frac{1101.2192 - 1089.1}{1103.4 - 1089.1}\right)$$

$\boxed{T_{2a} = 273.89°F}$

Therefore work produced is $w_a = 132.128$ Btu/lbm and the actual final temperature of steam is $T_{2a} = 273.89°F$

Question: 8.119P

Steam at 3 MPa and 400°C is expanded to 30 kpa in an adiabatic turbine with an isentropic efficiency of 92 percent. Determine the power produced by this turbine, in kW, when the mass flow rate is 2 kg/s.

Step 1 of 8

Given:

Mass flow rate, $\dot{m} = 2\ \text{kg/s}$

Initial pressure, $P_1 = 3\ \text{MPa}$

Initial temperature, $T_1 = 400^0\text{C}$

Final pressure, $P_{2s} = 30\ \text{kPa}$

Isentropic efficiency of the turbine, $\eta = 92\%$

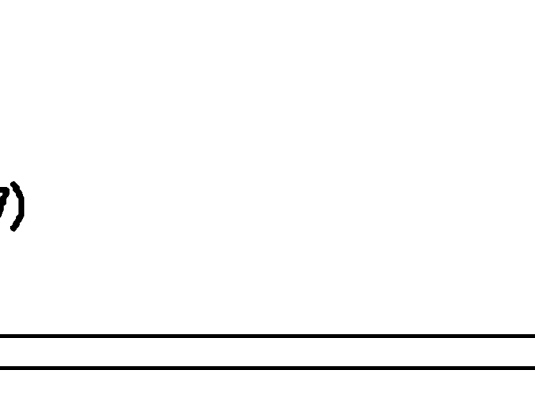
Step 2 of 8

At $P_1 = 3\ \text{MPa}$ and $T_1 = 400^0\text{C}$ from superheated water tables

Entropy, $s_1 = 6.9235\ \text{kJ/kg.K}$

Enthalpy, $h_1 = 3231.7\ \text{kJ/kg}$

Step 3 of 8



Step 4 of 8

At $P_{2s} = 30\ \text{kPa}$ from saturated water tables

$s_f = 0.9441\ \text{kJ/kg.K}$

$s_{fg} = 6.8234\ \text{kJ/kg.K}$

$h_f = 289.27\ \text{kJ/kg}$

$h_{fg} = 2335.3\ \text{kJ/kg}$

Step 5 of 8

Process 1-2s is isentropic expansion

$$s_1 = s_{2s}$$

$$s_{2s} = s_f + x_{2s}s_{fg}$$

$$x_{2s} = \frac{s_{2s} - s_f}{s_{fg}}$$

$$x_{2s} = \frac{6.9235 - 0.9441}{6.8234}$$

$$x_{2s} = 0.87631$$

Step 6 of 8

$$h_{2s} = h_f + x_{2s}h_{fg}$$

$$h_{2s} = 289.27 + 0.87631 \times 2335.3$$

$$h_{2s} = 2335.717\ \text{kJ/kg}$$

Step 7 of 8

Isentropic power output

$$\dot{W}_{ise} = \dot{m}(h_1 - h_{2s})$$

$$\dot{W}_{ise} = 2 \times (3231.7 - 2335.717)$$

$$\dot{W}_{ise} = 1791.966\ \text{kW}$$

Step 8 of 8

Isentropic efficiency of the turbine

$$\eta_T = \frac{\dot{W}_{act}}{\dot{W}_{ise}}$$

$$\dot{W}_{act} = \eta_T \times \dot{W}_{ise}$$

$$\dot{W}_{act} = 0.92 \times 1791.966$$

$$\boxed{\dot{W}_{act} = 1648.608\ \text{kW}}$$

Question: 8.120P

Repeat Prob. 8-119 for a turbine efficiency of 85 percent.

Step 1 of 5

Given:

Mass flow rate, $\dot{m} = 2\ \text{kg/s}$

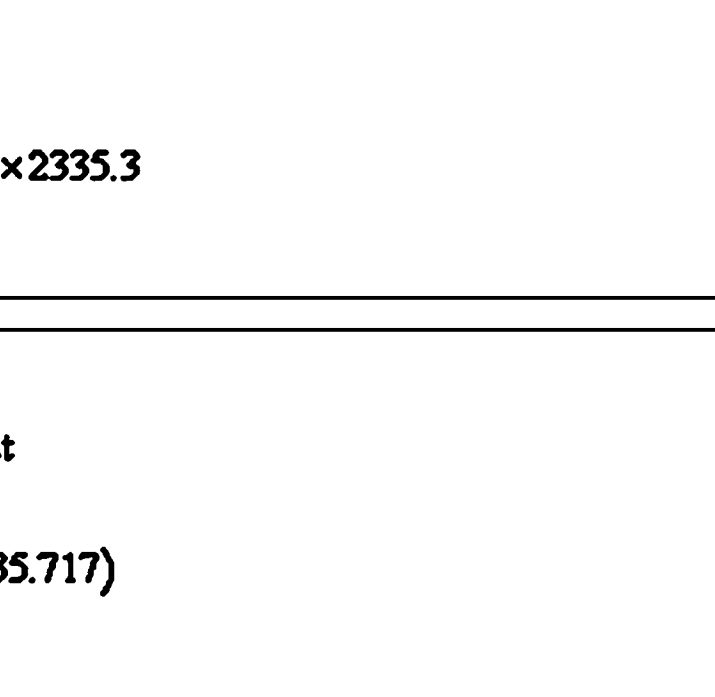
Initial pressure, $P_1 = 3\ \text{MPa}$

Initial temperature, $T_1 = 400^0\text{C}$

Final pressure, $P_{2s} = 30\ \text{kPa}$

Isentropic efficiency of the turbine, $\eta = 85\%$

Step 2 of 5



Step 3 of 5

At $P_1 = 3\ \text{MPa}$ and $T_1 = 400^0\text{C}$ from superheated water tables

Entropy, $s_1 = 6.9235\ \text{kJ/kg.K}$

Enthalpy, $h_1 = 3231.7\ \text{kJ/kg}$

At $P_{2s} = 30\ \text{kPa}$ from saturated water tables

$s_f = 0.9441\ \text{kJ/kg.K}$

$s_{fg} = 6.8234\ \text{kJ/kg.K}$

$h_f = 289.27\ \text{kJ/kg}$

$h_{fg} = 2335.3\ \text{kJ/kg}$

Step 4 of 5

Process 1-2s is isentropic expansion in the turbine

$$s_1 = s_{2s}$$

$$s_{2s} = s_f + x_{2s} s_{fg}$$

$$x_{2s} = \frac{s_{2s} - s_f}{s_{fg}}$$

$$x_{2s} = \frac{6.9235 - 0.9441}{6.8234}$$

$$x_{2s} = 0.87631$$

$$h_{2s} = h_f + x_{2s} h_{fg}$$

$$h_{2s} = 289.27 + 0.87631 \times 2335.3$$

$$h_{2s} = 2335.717\ \text{kJ/kg}$$

Step 5 of 5

Isentropic power output

$$\dot{W}_{ise} = \dot{m}(h_1 - h_{2s})$$

$$= 2 \times (3231.7 - 2335.717)$$

$$\dot{W}_{ise} = 1791.966\ \text{kW}$$

Isentropic efficiency of the turbine

$$\eta_T = \frac{\dot{W}_{act}}{\dot{W}_{ise}}$$

$$\dot{W}_{act} = \eta_T \times \dot{W}_{ise}$$

$$\dot{W}_{act} = 0.85 \times 1791.966$$

$$\boxed{\dot{W}_{act} = 1523.171\ \text{kW}}$$

Therefore power produced by the turbine is $\dot{W}_{act} = 1523.171\ \text{kW}$

Question: 8.121P

Steam enters an adiabatic turbine at 7 MPa. 600°C, and 80 m/s and leaves at 50 kPa, 150°C, and 140 m/s. If the power output of the turbine is 6 MW, determine (*a*) the mass flow rate of the steam flowing through the turbine and (*b*) the isentropic efficiency of the turbine.

Step 1 of 6

Obtain the properties of steam at inlet and out let from "Properties of superheated water" in the textbook.

At pressure, $P_1 = 7\ \text{MPa}$ and temperature, $T_1 = 600°\text{C}$, we have

Enthalpy of water, $h_1 = 3650.6\ \text{kJ/kg}$

Entropy of water, $s_1 = 7.0910\ \text{kJ/kg}\cdot\text{K}$

At pressure, $P_2 = 50\ \text{kPa}$ and temperature, $T_2 = 150°\text{C}$, we have

Actual enthalpy of water, $h_2 = 2780.2\ \text{kJ/kg}$

Step 2 of 6

Apply energy balance to the entire system.

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + W \quad \text{..... (1)}$$

Here, V_1, V_2 are velocities at inlet and exit.

Substitute $3650.6\,\text{kJ/kg}$ for h_1, $2780.2\,\text{kJ/kg}$ for h_2, $80\,\text{m/s}$ for V_1, and $140\,\text{m/s}$ for V_2.

$$3650.6\frac{\text{kJ}}{\text{kg}}\times 1000\frac{\text{J}}{\text{kJ}} + \frac{(80)^2}{2} = 2780.2\frac{\text{kJ}}{\text{kg}}\times 1000\frac{\text{J}}{\text{kJ}} + \frac{(140)^2}{2} + W$$

$$W = 863800\ \text{J/kg}$$

Step 3 of 6

a)

Write the expression for power output of the turbine.

$$\dot{W} = \dot{m}W$$

Here $\dot{m}$ is the mass flow rate of steam.

Substitute $6599\,\text{J/kg}$ for W and $6\,\text{MW}$ for $\dot{W}$.

$$\dot{W} = \dot{m}W$$

$$6\,\text{MW}\times 10^6\frac{\text{W}}{\text{MW}} = \dot{m}\times 863800$$

$$\dot{m} = 6.94\,\text{kg/s}$$

Therefore, the mass flow rate of steam is $\boxed{6.94\,\text{kg/s}}$.

Step 4 of 6

b)

Let Dryness fraction at state 2 be x_{2s}, Enthalpy at state 2 be h_{2s}, The pressure at exit be $P_2 = P_{2s} = 50\ \text{kPa}$.

For Isentropic process entropy remains constant.

Obtain the properties of steam from "Table A-5 Properties of saturated water pressure table" in the textbook.

At $P_2 = P_{2s} = 50\ \text{kPa}$, $s_1 = s_{2s} = 7.0910\ \text{kJ/kg}\cdot\text{K}$, we have

Entropy at saturated liquid, $s_f = 1.0912\ \text{kJ/kg}\cdot\text{K}$

Entropy during evaporation, $s_{fg} = 6.5019\ \text{kJ/kg}\cdot\text{K}$

Enthalpy at saturated liquid, $h_f = 340.54\ \text{kJ/kg}$

Enthalpy during evaporation, $h_{fg} = 2304.7\ \text{kJ/kg}$

Step 5 of 6

Write the relation for entropies.

$$s_{2s} = h_f + x_{2s}h_{fg}$$

$$x_{2s} = \frac{s_{2s} - h_f}{h_{fg}} = \frac{7.0910 - 1.0912}{6.5019} = 0.9227$$

Find the enthalpy at state 2.

$$h_{2s} = h_f + x_{2s}h_{fg}$$

Substitute $340.54\,\text{kJ/kg}$ for h_f, 0.9277 for x_{2s}, and $2304.7\,\text{kJ/kg}$ for h_{fg}.

$$h_{2s} = 340.54 + 0.9227\times 2304.7 = 2467.08\ \text{kJ/kg}$$

Step 6 of 6

Find the isentropic work.

Substitute $3650.6\,\text{kJ/kg}$ for h_1, $2467.08\,\text{kJ/kg}$ for h_2, $80\,\text{m/s}$ for V_1, and $140\,\text{m/s}$ for V_2.

$$3650.6\frac{\text{kJ}}{\text{kg}}\times 1000\frac{\text{J}}{\text{kJ}} + \frac{(80)^2}{2} = 2467.08\frac{\text{kJ}}{\text{kg}}\times 1000\frac{\text{J}}{\text{kJ}} + \frac{(140)^2}{2} + W_{isen}$$

$$W_{isen} = 1176920\ \text{J/kg}$$

Find the isentropic efficiency.

$$\eta = \frac{\dot{W}}{\dot{m}W_{isen}}$$

Substitute $1176920\,\text{J/kg}$ for W_{isen}, $6\,\text{MW}$ for $\dot{W}$, and $6.94\,\text{kg/s}$ for $\dot{m}$.

$$\eta = \frac{6\,\text{MW}\times 10^6\frac{\text{W}}{\text{MW}}}{6.94\times 1176920} = 0.7345 = 73.45\%$$

Therefore, the isentropic efficiency of the turbine is $\boxed{73.45\%}$.

Question: 8.122P

Combustion gases enter an adiabatic gas turbine at 1540°F and 120 psia and leave at 60 psia with a low velocity. Treating the combustion gases as air and assuming an isentropic efficiency of 82 percent, determine the work output of the turbine.

Step 1 of 4

Given

Air temperature at inlet of turbine $T_1 = 1540°F$

$= 2000°R$

Pressure of air at inlet of turbine $P_1 = 120$ psia

Pressure of air at outlet of turbine $P_2 = 60$ psia

Isentropic efficiency of the turbine $\eta_T = 0.82$

Step 2 of 4

Ratio of specific heats for air, $k = 1.4$

For an isentropic process,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{k-1/k}$$

$$\frac{T_2}{(2000\ \text{R})} = \left(\frac{60\ \text{psia}}{120\ \text{psia}}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = (2000\ \text{R}) \times 0.82$$

$$T_2 = 1640°R$$

Step 3 of 4

From air tables,

At 2000 R

Enthalpy of air at inlet, $h_1 = 504.71$ Btu/lbm

At 1640 R

Enthalpy of air at outlet $h_2 = 417.3$ Btu/lbm

Isentropic specific work of the turbine is

$$w_{s,out} = (h_1 - h_2)$$

$$w_{s,out} = (504.71 - 417.3)$$

$$w_{s,out} = 87.41\ \text{Btu/lbm}$$

Step 4 of 4

Work output of the turbine,

$$w_{out} = \eta_T w_{s,out}$$

$$w_{out} = 0.82 \times 87.41\ \text{Btu/lbm}$$

$$w_{out} = 71.7\ \text{Btu/lbm}$$

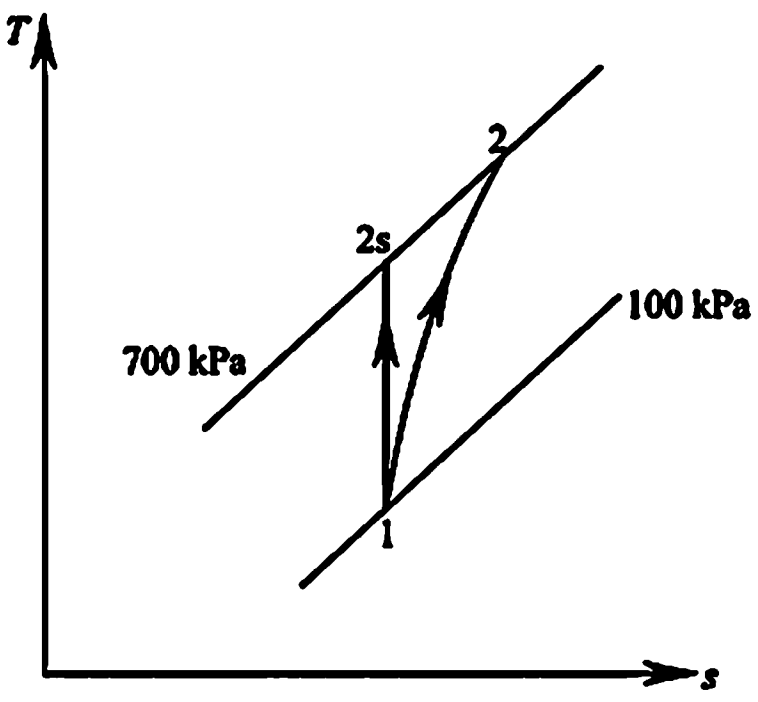
Hence actual work output of the turbine, $\boxed{w_{out} = 71.7\ \text{Btu/lbm}}$

Question: 8.123P

Air at 100 kPa and 20°C is compressed to 700 kPa steadily and adiabatically at a rate of 2 kg/s. Determine the power required to compress this air if the isentropic compression efficiency is 95 percent.

Step 1 of 4

Draw the $T-s$ diagram.



Step 2 of 4

Obtain the specific heat ratio and specific heat constant for air from the "ideal gas specific heats of various common gases table".

At $T_{avg} = 400\,\text{K}$,

$k = 1.395$

$c_p = 1.013\ \text{kJ/kg}\cdot\text{K}$

Find the exit isentropic temperature using the following relation:

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Here, T_1 is the temperature at state 1, P_1 is the pressure at state 1 and P_2 is the pressure at state 2.

Substitute 1.395 for k, 100 kPa for P_1, 700 kPa for P_2 and $293\,\text{K}$ for T_1.

$$\begin{aligned} T_{2s} &= 293\left(\frac{700}{100}\right)^{\frac{1.395-1}{1.395}} \\ &= 293(7)^{\frac{0.395}{1.395}} \\ &= 508.347\ \text{K} \end{aligned}$$

Step 3 of 4

Find the isentropic work input using the following formula:

$$\dot{W}_{s,in} = \dot{m}c_p(T_{2s} - T_1)$$

Here, $\dot{m}$ is the mass flow rate.

Substitute 2 kg/s for $\dot{m}$, $293\,\text{K}$ for T_1, $1.013\ \text{kJ/kg}\cdot\text{K}$ for c_p, and $508.347\ \text{K}$ for T_{2s}.

$$\begin{aligned} \dot{W}_{s,in} &= 2\times 1.013\times(508.347-293) \\ &= 436.293\ \text{kW} \end{aligned}$$

Step 4 of 4

Find the power required to compress the air using the following formula:

$$\dot{W} = \frac{\dot{W}_{s,in}}{\eta_s}$$

Here, η_s is the isentropic efficiency.

Substitute $436.293\ \text{kW}$ for $\dot{W}_{s,in}$ and 0.95 for η_s.

$$\begin{aligned} \dot{W} &= \frac{436.293}{0.95} \\ &= 459.26\,\text{kW} \end{aligned}$$

Therefore, the power required to compress the air is $\boxed{459.26\ \text{kW}}$.

Question: 8.124P

Steam at 4 MPa and 350°C is expanded in an adiabatic turbine to 120 kPa. What is the isentropic efficiency of this turbine if the steam is exhausted as a saturated vapor?

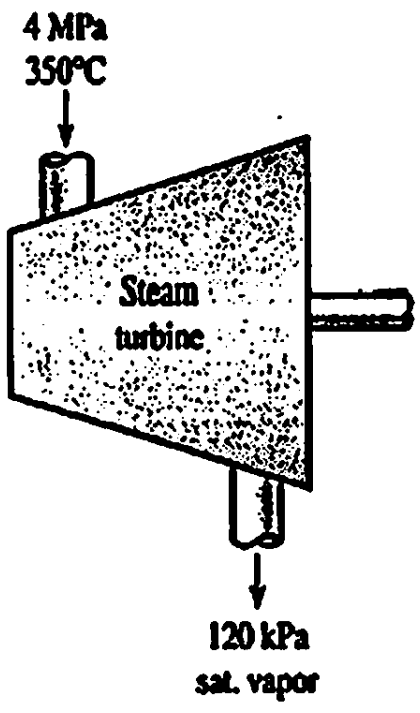


FIGURE P8–124

Step 1 of 6

Given data:

Pressure at turbine inlet is, $P_1 = 4\ \text{MPa} = 4000\ \text{kPa}$.

Temperature at turbine inlet is, $T_1 = 350°\text{C} = 623\ \text{K}$.

Pressure at turbine exit is, $P_2 = 120\ \text{kPa}$.

Exit state of steam is saturated vapor

Step 2 of 6

From superheated water,

At pressure $P_1 = 4\ \text{MPa}$ and temperature $T_1 = 350°\text{C}$

the enthalpy and entropy values are

$h_1 = 3093.3\ \text{kJ/kg}$

$s_1 = 6.5843\ \text{kJ/kg} \cdot \text{K}$

From Saturated water,

At pressure $P_2 = 120\ \text{kPa}$

Saturated properties of steam are as follows

Enthalpy of saturated liquid is $h_{f2} = 439.46\ \text{kJ/kg}$

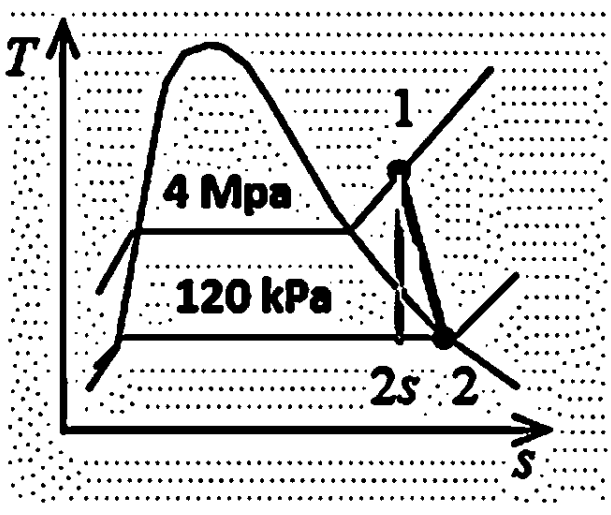
Enthalpy of evaporation is $h_{fg2} = 2243.07\ \text{kJ/kg}$

Entropy of saturated liquid is $s_{f2} = 1.3601\ \text{kJ/kg} \cdot \text{K}$

Entropy of evaporation is $s_{fg2} = 5.936\ \text{kJ/kg} \cdot \text{K}$

Step 3 of 6

Let us draw the T-s diagram:



Step 4 of 6

Actual exit state of steam is saturated vapor.

From Saturated water,

At pressure $P_2 = 120\ \text{kPa}$ and

Saturated vapor condition the actual exit enthalpy is

$h_2 = 2683.1\ \text{kJ/kg}$

If the process is isentropic, the exit specific entropy is given by

$$s_{2s} = s_1$$

$$s_{f2} + x_{2s} \times s_{fg2} = s_1$$

$$1.3601 + x_{2s} \times 5.936 = 6.5843$$

$$x_{2s} \times 5.936 = 6.5843 - 1.3601$$

$$x_{2s} = \frac{6.5843 - 1.3601}{5.936}$$

$$= \frac{5.2242}{5.936}$$

$$= 0.88$$

Therefore isentropic exit dryness fraction is $x_{2s} = 0.88$

Step 5 of 6

The isentropic exit specific enthalpy is given by

$$h_{2s} = h_{f2} + x_{2s} \times h_{fg2}$$

$$= 439.46 + 0.88 \times 2243.07$$

$$= 439.46 + 1973.9$$

$$= 2413.36\ \text{kJ/kg}$$

Step 6 of 6

The isentropic efficiency of turbine is given by

$$\eta_{turb} = \frac{\text{Actual work done by the turbine}}{\text{Isentropic work done by the turbine}}$$

$$= \frac{m \times (h_1 - h_2)}{m \times (h_1 - h_{2s})}$$

$$= \frac{h_1 - h_2}{h_1 - h_{2s}}$$

$$= \frac{3093.3 - 2683.1}{3093.3 - 2413.36}$$

$$= \frac{410.2}{679.94}$$

$$= 0.6033$$

$$= 60.33\%$$

Therefore isentropic efficiency of turbine is $\boxed{\eta_{turb} = 60.33\%}$.

Question: 8.125P

Air is expanded from 2 MPa and 327°C to 100 kPa in an adiabatic turbine. Determine the isentropic efficiency of this turbine if the air is exhausted at 0°C.

Step 1 of 5

Given data:

Inlet pressure to the turbine is, $P_1 = 2\ \text{MPa} = 2000\ \text{kPa}$.

Exit pressure to the turbine is, $P_2 = 100\ \text{kPa}$.

Inlet temperature to the turbine is, $T_1 = 327°\text{C} = 600\ \text{K}$.

Exit temperature the turbine is $T_2 = 0°\text{C} = 273\ \text{K}$.

Step 2 of 5

Let us draw the T-s diagram:

Step 3 of 5

As the process is isentropic, the ratio of temperatures of exit and inlet is given by

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}},$$

From Ideal Gas Specific Heats of Various Common Gases

At temp $= 400\text{K}$

Where $k = 1.395$,

Therefore the isentropic exit temperature of a turbine is given by

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= 600\left(\frac{100}{2000}\right)^{\frac{1.395-1}{1.395}}$$

$$= 600(0.05)^{\frac{0.395}{1.395}}$$

$$= 600(0.05)^{0.28315}$$

$$= 600(0.428)$$

$$= 256.8\ \text{K}$$

Step 4 of 5

The isentropic efficiency of turbine is given by

$$\eta_{isen} = \frac{\text{Actual work done by the turbine}}{\text{Isentropic work done by the turbine}}$$

$$= \frac{m(h_1 - h_2)}{m(h_1 - h_{2s})}$$

$$= \frac{h_1 - h_2}{h_1 - h_{2s}}$$

$$= \frac{c_p \times (T_1 - T_2)}{c_p \times (T_1 - T_{2s})}$$

$$= \frac{T_1 - T_2}{T_1 - T_{2s}}$$

Step 5 of 5

Substituting the values,

$$\eta_{isen} = \frac{600 - 273}{600 - 256.8}$$

$$= \frac{327}{343.2}$$

$$= 0.953$$

$$= 95.3$$

Therefore the isentropic efficiency of turbine is $\boxed{\eta_{isen} = 0.953 = 95.3\,\%}$.

Question: 8.126P

A refrigeration unit compresses saturated R-134a vapor at 20°C to 1000 kPa. How much power is required to compress 0.5 kg/s of R-134a with a compressor efficiency of 85 percent?

Step 1 of 2

Determine the enthalpy at two states to calculate the compression work

From the saturated R-134a tables, at $20°C$

The enthalpy of the saturated vapor is $h_1 = 261.59 \text{ kJ/kg}$

The entropy of the saturated vapor is $s_1 = 0.9223 \text{ kJ/kg} \cdot \text{K}$

Consider isentropic compression

$$\begin{aligned} s_2 &= s_1 \\ &= 0.9223 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

Now find the enthalpy at the exit if the process is isentropic (property of R-134a at pressure of 1 MPa and enthalpy of 0.9223 $\text{kJ/kg} \cdot \text{K}$)

$$h_{2s} = 273.11 \text{ kJ/kg}$$

Step 2 of 2

The power input to the isentropic compression.

$$W_{s,in} = \dot{m}(h_{2s} - h_1)$$

Substitute 0.5 kg/s for $\dot{m}$, 273.11 kJ/kg for h_{2s} , and 261.59 kJ/kg for h_1

$$\begin{aligned} W_{s,in} &= \dot{m}(h_{2s} - h_1) \\ &= 0.5(273.11 - 261.59) \\ &= 0.5 \times 11.52 \\ &= 5.76 \text{ kW} \end{aligned}$$

The work input to the compression process operating with 85 percent efficiency is

$$W_{in} = \frac{W_{s,in}}{\eta}$$

Substitute 0.85 for η , and 5.76 kW for $W_{s,in}$

$$\begin{aligned} W_{in} &= \frac{W_{s,in}}{\eta} \\ &= \frac{5.76}{0.85} \\ &= 6.776 \text{ kW} \end{aligned}$$

Thus the work input to the compression process is $\boxed{6.776 \text{ kW}}$

Question: 8.127P

Refrigerant-134a enters an adiabatic compressor as saturated vapor at 100 kPa at a rate of 0.7 m3 /min and exits at 1-MPa pressure. If the isentropic efficiency of the compressor is 87 percent, determine (*a*) the temperature of the refrigerant at the exit of the compressor and (*b*) the power input, in kW. Also, show the process on a *T-s* diagram with respect to saturation lines.

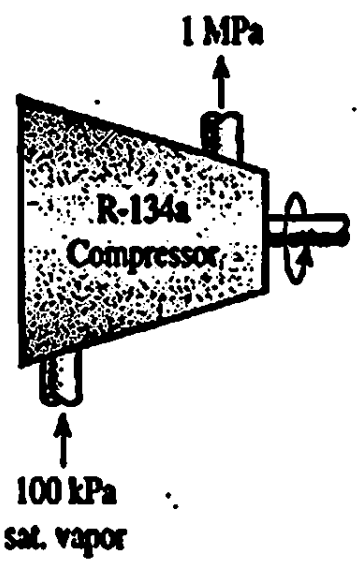


FIGURE P8–127

Step 1 of 6

Obtain the properties of saturated refrigerant 134a from table A-12 (Saturated refrigerant 134a-pressure table) in the text book, at pressure $P_1 = 100\,\text{kPa}$.

Specific volume of saturated vapour is $v_1 = v_g = 0.19254\ \text{m}^3/\text{kg}$.

Specific enthalpy of saturated vapour is $h_1 = h_g = 234.44\ \text{kJ/kg}$.

Specific entropy of saturated vapour is $s_1 = s_g = 0.95183\ \text{kJ/kg-K}$.

Since, the process is an adiabatic process, entropy will be constant. $s_2 = s_1 = 0.95183\ \text{kJ/kg-K}$

Obtain the enthalpy at state-2s from table A-13 (Superheated refrigerant R-134a) in the text book, at pressure $P_2 = 1\,\text{MPa}$ and entropy $s_2 = 0.95183\ \text{kJ/kg-K}$ by using interpolation.

$$\frac{h_{2s}-271.71}{282.74-271.71}=\frac{0.95183-0.9179}{0.9525-0.9179}$$
$$\frac{h_{2s}-271.71}{11.03}=0.9806$$
$$h_{2s}=(0.9806)(11.03)+271.71$$
$$=282.526\ \text{kJ/kg}$$

Step 2 of 6

Calculate the actual enthalpy at state-2 using the isentropic efficiency of compressor.

$$\eta_{isen}=\frac{h_{2s}-h_1}{h_{2a}-h_1}$$

Substitute $282.526\ \text{kJ/kg}$ for h_{2s} , $234.44\ \text{kJ/kg}$ for h_1 , and 0.87 for η_{isen} .

$$0.87=\frac{282.526-234.44}{h_{2a}-234.44}$$
$$0.87=\frac{48.086}{h_{2a}-234.44}$$
$$h_{2a}=\frac{48.086}{0.87}+234.44$$
$$=289.71\ \text{kJ/kg}$$

Step 3 of 6

a)

Obtain the enthalpy at state-2s from table A-13 (Superheated refrigerant R-134a) in the text book, at pressure $P_2 = 1\,\text{MPa}$ and entropy $h_{2a} = 289.71\ \text{kJ/kg}$ by using interpolation.

$$\frac{T_{2a}-50}{60-50}=\frac{289.71-282.74}{293.38-282.74}$$
$$\frac{T_{2a}-50}{10}=0.6551$$
$$T_{2a}=(0.6551)(10)+50$$
$$=56.551°\text{C}$$

Therefore, the temperature of the refrigerant at the exit of the compressor is $\boxed{56.551°\text{C}}$.

Step 4 of 6

b)

Calculate the power input to compressor.

$$W_{act}=\dot{m}(h_{2a}-h_1) \quad \text{..... (1)}$$

Here, $\dot{m}$ is the mass flow rate.

Calculate the mass flow rate.

$$\dot{m}=\frac{V_1}{v_1}$$

Here, V_1 is the volume flow rate at compressor inlet.

Substitute $0.7\ \text{m}^3/\text{min}$ for V_1 and $0.19254\ \text{m}^3/\text{kg}$ for v_1 .

$$\dot{m}=\frac{0.7\,\text{m}^3/\text{min}}{0.19254\,\text{m}^3/\text{kg}}$$
$$=\frac{0.01167\,\text{m}^3/\text{s}}{0.19254}$$
$$=0.0606\ \text{kg/s}$$

Step 5 of 6

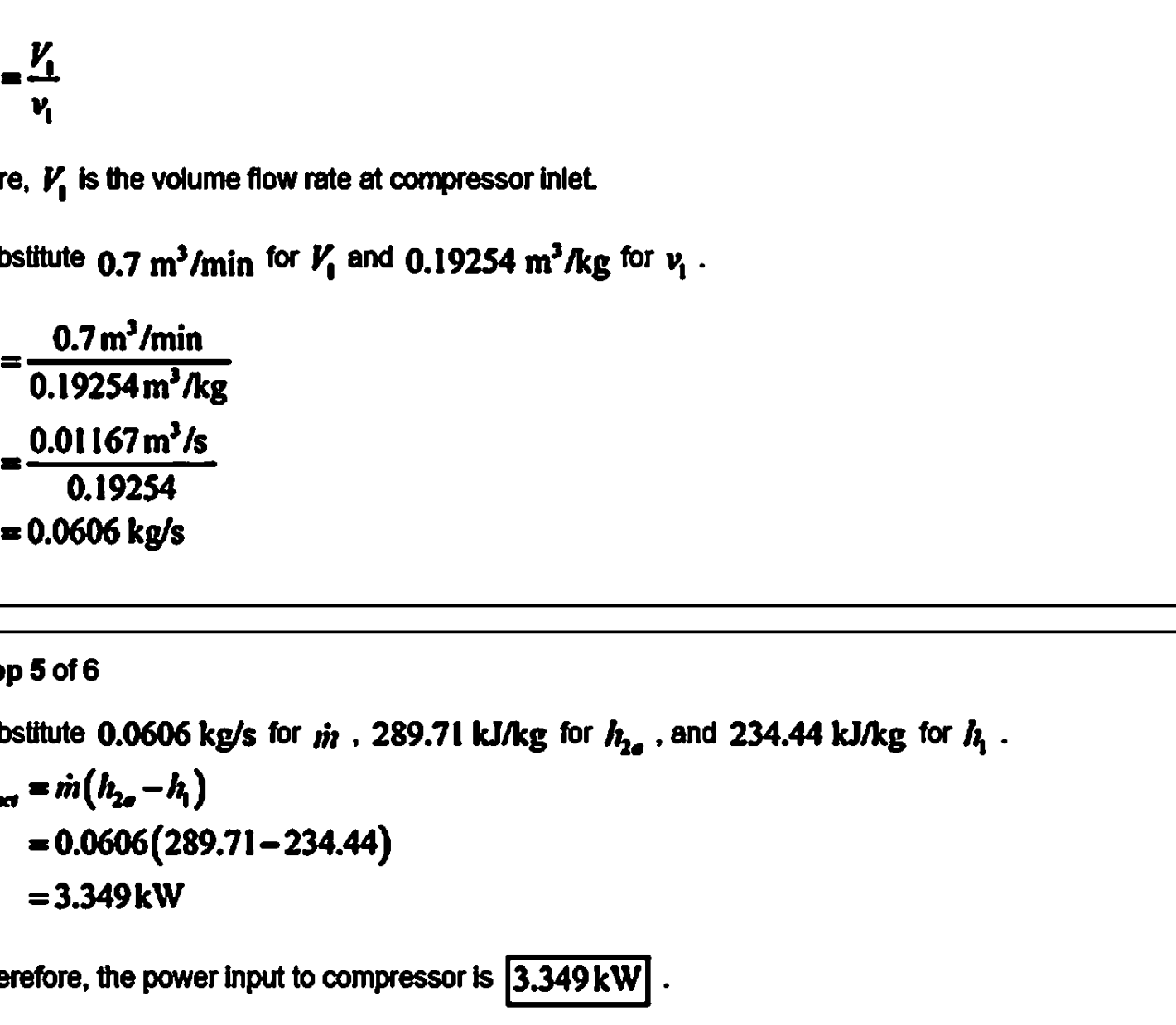
Substitute $0.0606\ \text{kg/s}$ for $\dot{m}$, $289.71\ \text{kJ/kg}$ for h_{2a} , and $234.44\ \text{kJ/kg}$ for h_1 .

$$W_{act}=\dot{m}(h_{2a}-h_1)$$
$$=0.0606(289.71-234.44)$$
$$=3.349\,\text{kW}$$

Therefore, the power input to compressor is $\boxed{3.349\,\text{kW}}$.

Step 6 of 6

Draw the T-s diagram.



Question: 8.128INCLUD-C



Reconsider Prob. 8-127. Using EES (or other) software, redo the problem by including the effects of the kinetic energy of the flow by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

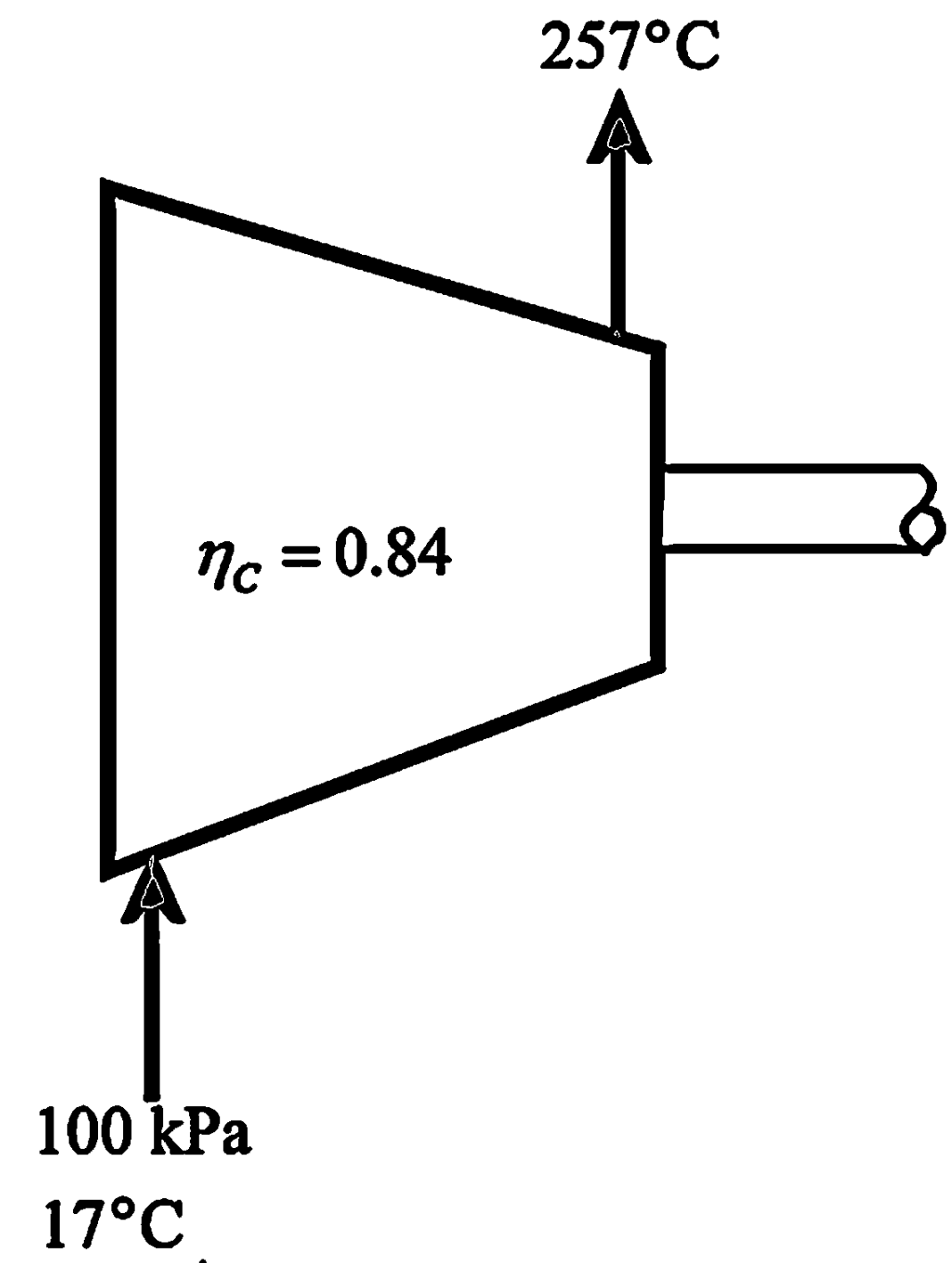
Ask an expert

Question: 8.129P

Air enters an adiabatic compressor at 100 kPa and 17°C at a rate of 2.4 m3 /s, and it exits at 257°C. The compressor has an isentropic efficiency of 84 percent. Neglecting the changes in kinetic and potential energies, determine (a) the exit pressure of air and (*b*) the power required to drive the compressor.

Step 1 of 6

Draw the schematic figure of the compressor.



Step 2 of 6

Convert the units of inlet temperature from Celsius to Kelvin.

$$\begin{aligned} T_1 &= 17°C \\ &= (17°C + 273)\ K \\ &= 290\ K \end{aligned}$$

Convert the units of inlet temperature from Celsius to Kelvin.

$$\begin{aligned} T_2 &= 257°C \\ &= (257°C + 273)\ K \\ &= 530\ K \end{aligned}$$

Obtain the properties of air at the inlet temperature of $290\ K$ from Table A-17, "Ideal-gas properties of air," from the text book.

Enthalpy, $h_1 = 290.16\ kJ/kg$

Reduced pressure, $P_{r1} = 1.2311$

Obtain the properties of air at the exit temperature of $530\ K$ from Table A-17, "Ideal-gas properties of air," from the text book.

Enthalpy, $h_{2s} = 533.98\ kJ/kg$

Step 3 of 6

Write the relation or the isentropic efficiency of the compressor.

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

Here, h_{2s} is the isentropic enthalpy.

Substitute 0.84 for η_C , $290.16\ kJ/kg$ for h_1 , and $533.98\ kJ/kg$ for h_{2a} .

$$\begin{aligned} 0.84 &= \frac{h_{2s} - 290.16}{533.98 - 290.16} \\ h_{2s} &= 494.9688\ kJ/kg \end{aligned}$$

Obtain the reduced pressure corresponding to an enthalpy of $494.9688\ kJ/kg$ from Table A-17, "Ideal-gas properties of air," from the text book by interpolation.

$$\begin{aligned} \frac{503.02 - 494.9688}{503.02 - 492.74} &= \frac{8.411 - P_{r2}}{8.411 - 7.824} \\ P_{r2} &= 7.951 \end{aligned}$$

Step 4 of 6

(a)

Calculate the exit pressure of the air by using the following adiabatic relation:

$$\frac{P_2}{P_1} = \frac{P_{r2}}{P_{r1}}$$

Here, P_1 is the inlet pressure.

Substitute $100\ kPa$ for P_1 , 7.951 for P_{r2} , and 1.2311 for P_{r1} .

$$\begin{aligned} \frac{P_2}{100} &= \frac{7.951}{1.2311} \\ P_2 &= 645.87\ kPa \end{aligned}$$

Therefore, the exit pressure of the air is $\boxed{645.87\ kPa}$.

Step 5 of 6

(b)

Write the energy balance equation for a steady flow system.

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}$$

Here, $\dot{E}_{in} - \dot{E}_{out}$ is the rate of net energy transfer by heat, work, and mass and $\Delta \dot{E}_{system}$ is the rate of change in internal, kinetic, potential, energies.

Neglect the heat, kinetic and potential energies and rewrite the equation.

$$\dot{W}_{a,in} = \dot{m}(h_{2a} - h_1) \quad \text{..... (1)}$$

Here, $\dot{W}_{a,in}$ is the power input to the compressor and $\dot{m}$ is the mass flow rate.

Step 6 of 6

Calculate the mass flow rate by using the following ideal gas equation:

$$\dot{m} = \frac{P_1 \dot{V}_1}{R T_1}$$

Here, $\dot{V}_1$ is the volumetric flow rate and R is the gas constant for air.

Substitute $100\ kPa$ for P_1 , $2.4\ m^3/s$ for $\dot{V}_1$, $0.287\ kPa \cdot m^3/kg$ for R , and $290\ K$ for T_1 .

$$\begin{aligned} \dot{m} &= \frac{100 \times 2.4}{0.287 \times 290} \\ &= 2.884\ kg/s \end{aligned}$$

Substitute $2.884\ kg/s$ for $\dot{m}$, $533.98\ kJ/kg$ for h_{2a} , and $290.16\ kJ/kg$ for h_1 in equation (1) to calculate the power input to the compressor.

$$\begin{aligned} \dot{W}_{a,in} &= \dot{m}(h_{2a} - h_1) \\ &= 2.884(533.98 - 290.16) \\ &= 703.2\ kJ/s \times \left(\frac{1\ kW}{1\ kJ/s}\right) \\ &= 703.2\ kW \end{aligned}$$

Therefore, the power input to the compressor is $\boxed{703.2\ kW}$.

Question: 8.130P

Air is compressed by an adiabatic compressor from 95 kPa and 27°C to 600 kPa and 277°C. Assuming variable specific heats and neglecting the changes in kinetic and potential energies, determine (*a*) the isentropic efficiency of the compressor and (*b*) the exit temperature of air if the process were reversible.

Step 1 of 5

Obtain the properties from ideal-gas properties of air table,

At initial temperature, $T_1 = 300\ \text{K}$

Enthalpy of air at inlet, $h_1 = 300.19\ \text{kJ/kg}$

Relative pressure, $p_{r_1} = 1.386$

At actual temperature, $T_{2a} = 500\ \text{K}$

Actual enthalpy of air at outlet, $h_{2a} = 555.74\ \text{kJ/kg}$

Step 2 of 5

Calculate the enthalpy of air at the end of isentropic compression.

Consider the following isentropic relation as shown below,

$$p_{r_2} = \left(\frac{p_2}{p_1}\right) p_{r_1}$$

Substitute $95\ \text{kPa}$ for p_1 and $600\ \text{kPa}$ for p_2.

$$p_{r_2} = \left(\frac{600}{95}\right)(1.386)$$
$$= 8.754$$

Step 3 of 5

From ideal-gas properties of air table, the value of isentropic enthalpy (h_{2s}) corresponding to relative pressure (p_{r_2}) can be obtained by the use of interpolation technique.

Now, obtain the properties from ideal-gas properties of air table,

At $p_r = 8.411$, $h = 503.02\ \text{kJ/kg}$

At $p_r = 9.031$, $h = 513.32\ \text{kJ/kg}$

The value of h_{2s} lies between the above two values of h

$$h_{2s} = h_{at\ 8.411} + \left(\frac{h_{at\ 9.031} - h_{at\ 8.411}}{9.031 - 8.411}\right)(8.754 - 8.411)$$
$$= 503.02 + \left(\frac{513.32 - 503.02}{0.62}\right)(0.343)$$
$$= 503.02 + 5.7$$
$$= 508.72\ \text{kJ/kg}$$

Therefore, the isentropic enthalpy is, $h_{2s} = 508.72\ \text{kJ/kg}$

Step 4 of 5

(a)

Calculate the isentropic efficiency of the compressor by using the following equation,

$$\eta_c = \frac{\text{Isentropic work input}}{\text{Actual work input}}$$
$$= \frac{h_{2s} - h_1}{h_{2a} - h_1}$$
$$= \frac{508.72 - 300.19}{555.74 - 300.19}$$
$$= 0.816$$

Therefore, the isentropic efficiency of the compressor is, **81.6 percent**

Step 5 of 5

(b)

Calculate the exit temperature of air in the isentropic process.

From the ideal-gas properties of air table, the exit temperature of air (T_{2s}) in an isentropic process corresponding to relative pressure (p_{r_2}) can be obtained by the use of interpolation technique.

Now, obtain the properties from ideal-gas properties of air table,

At $p_r = 8.411$, $T = 500\ \text{K}$

At $p_r = 9.031$, $T = 510\ \text{K}$

$$T_{2s} = T_{at\ 8.411} + \left(\frac{T_{at\ 9.031} - T_{at\ 8.411}}{9.031 - 8.411}\right)(8.754 - 8.411)$$
$$= 500 + \left(\frac{510 - 500}{0.62}\right)(0.343)$$
$$= 500 + 5.53$$
$$= 505.53\ \text{K}$$

Therefore, the exit temperature of air in an isentropic process is, **505.53 K**

Question: 8.131P

Argon gas enters an adiabatic compressor at 20 psia and 90°F with a velocity of 60 ft/s, and it exits at 200 psia and 240 ft/s. If the isentropic efficiency of the compressor is 80 percent, determine (*a*) the exit temperature of the argon and (*b*) the work input to the compressor.

Step 1 of 5

Given data

Pressure of argon at compressor inlet, $P_1 = 20\,\text{psia}$

Temperature of argon at compressor inlet, $T_1 = 90°\text{F}$

$= 550°\text{R}$

Velocity of argon at compressor inlet, $V_1 = 60\,\text{ft/s}$

Pressure of argon at outlet of compressor, $P_2 = 200\,\text{psia}$

Velocity of argon at compressor outlet, $V_2 = 240\,\text{ft/s}$

Isentropic efficiency of compressor $\eta_C = 0.80$

Step 2 of 5

a) Let T_{2a} be the exit temperature of the argon

From the properties of ideal gas specific heats of various common gases

The specific heat of argon, $c_p = 0.1253\,\text{Btu/lbm}\cdot\text{R}$

Ratio of specific heats, $k = 1.667$

For an isentropic process,

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = (550)\left(\frac{200}{20}\right)^{\frac{0.667}{1.667}}$$

$$T_{2s} = 1381.92°\text{R}$$

Step 3 of 5

Now using the change in actual kinetic energy

$$\Delta ke = \frac{V_2^2 - V_1^2}{2}$$

$$\Delta ke = \frac{(240)^2 - (60)^2}{2}$$

$$\Delta ke = 27000\,\text{ft}^2/s^2$$

$$\Delta ke = 27000\left(\frac{\text{ft}^2}{s^2}\right)\left(\frac{1\,\text{Btu/lbm}}{25037\,\text{ft}^2/s^2}\right)$$

$$\Delta ke = 1.078\ \text{Btu/lbm}$$

Step 4 of 5

The isentropic efficiency of a compressor,

$$\eta_C = \frac{(h_{2s} - h_1) + \Delta ke}{(h_{2a} - h_1) + \Delta ke}$$

$$\eta_C = \frac{c_p(T_{2s} - T_1) + \Delta ke}{c_p(T_{2a} - T_1) + \Delta ke}$$

$$0.80 = \frac{0.1253(1381.92 - 550) + 1.078}{0.1253(T_{2a} - 550) + 1.078}$$

$$0.1253(T_{2a} - 550) + 1.078 = 131.65$$

$$(T_{2a} - 550) = 1042.1$$

$$\boxed{T_{2a} = 1592.1°\text{R}}$$

Therefore the exit temperature of the argon, $T_{2a} = 1592.1°\text{R}$

Step 5 of 5

b) Let w_{in} be the work input of the compressor

Energy balance for steady flow system,

$$W_{in} + \dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$W_{in} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) - \dot{m}\left(h_1 + \frac{V_1^2}{2}\right)$$

$$W_{in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$\frac{W_{in}}{\dot{m}} = \left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$w_{in} = \left(c_p T_2 - c_p T_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$w_{in} = \{(0.1253 \times 1592.1) - (0.1253 \times 550) + 1.078\}$$

$$\boxed{w_{in} = 131.65\,\text{Btu/lbm}}$$

Therefore the work input of the compressor $w_{in} = 131.65\,\text{Btu/lbm}$

Question: 8.132P

Air enters an adiabatic nozzle at 45 psia and 940°F with low velocity and exits at 650 ft/s. If the isentropic efficiency of the nozzle is 85 percent, determine the exit temperature and pressure of the air.

Step 1 of 5

Apply mass balance equation to the nozzle

$$m_{in} = m_{out}$$

$$m_1 = m_2$$

Here, m_{in} or m_1 is mass of air entering into the nozzle and m_{out} or m_2 is mass of air exiting from the nozzle.

Apply energy balance to the adiabatic nozzle,

$$m_1\left(h_1 + \frac{V_1^2}{2}\right) = m_2\left(h_2 + \frac{V_2^2}{2}\right)$$

Since $m_1 = m_2$ from the mass balance equation,

$$\left(h_1 + \frac{V_1^2}{2}\right) = \left(h_2 + \frac{V_2^2}{2}\right)$$

Here, V_1 and V_2 are the inlet and exit velocities of the nozzle.

Neglect the velocity of air at inlet, since air enters the adiabatic nozzle at very low velocity, $V_1 = 0$

Then energy equation becomes,

$$(h_1 + 0) = \left(h_2 + \frac{V_2^2}{2}\right)$$

$$h_2 = h_1 - \frac{V_2^2}{2} \qquad \text{..... (1)}$$

Step 2 of 5

Obtain the properties of air, at temperature $T_1 = 940°\text{F} = 1400\text{R}$ from ideal gas properties of air table.

The enthalpy and relative pressure are

$$h_1 = 342.9\ \text{Btu/lbm}$$

$$P_{r1} = 42.88$$

Substitute 650 ft/s for V_2 in equation (1)

$$h_2 = h_1 - \frac{V_2^2}{2}$$

$$= 342.9\text{Btu/lbm} - \frac{650^2\ \text{ft}^2/\text{s}^2}{2} \times \frac{1\text{Btu/lbm}}{25037\ \text{ft}^2/\text{s}^2}$$

$$= 334.462\text{Btu/lbm}$$

Step 3 of 5

Find the exit temperature at $h_2 = 334.462\text{Btu/lbm}$ from ideal gas properties of air table by using interpolation,

$$T_2 = 1360 + (1400 - 1360)\left(\frac{334.462 - 332.48}{342.9 - 332.48}\right)$$

$$= 1360 + 40 \times 0.19$$

$$= 1367.6\text{R}$$

$$T_2 = 1367.6 - 460\text{R}$$

$$= 907.6°\text{F}$$

Therefore, exit temperature of air from the nozzle is $\boxed{907.6°\text{F}}$

Step 4 of 5

Find the isentropic exit enthalpy, h_{2s} by using isentropic efficiency of the nozzle by using the equation,

$$\eta_{isen} = \frac{h_2 - h_1}{h_{2s} - h_1}$$

Substitute 0.85 for η_{isen}

$$0.85 = \frac{334.462 - 342.9}{h_{2s} - 342.9}$$

$$0.85 = \frac{-8.438}{h_{2s} - 342.9}$$

$$h_{2s} - 342.9 = -9.927$$

$$h_{2s} = 332.973\ \text{Btu/lbm}$$

Step 5 of 5

Find the relative exit pressure at $h_{2s} = 332.973\text{Btu/lbm}$ from ideal gas properties of air,

$$P_{r2} = 38.41 + (42.88 - 38.14)\left(\frac{332.973 - 332.48}{342.9 - 332.48}\right)$$

$$= 38.634$$

Find the exit pressure of air by using the relation between relative and normal pressures

$$\frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1}$$

$$P_2 = \frac{P_{r2}}{P_{r1}} \times P_1$$

$$= \frac{38.634}{42.88} \times 45$$

$$= (0.9)(45)$$

$$= 40.54\ \text{psia}$$

Therefore, the exit pressure of air from adiabatic nozzle is $\boxed{40.54\ \text{psia}}$

Question: 8.133P pg 1

Reconsider Prob. 8-132E. Using EES (or other) software, study the effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on both the exit temperature and pressure of the air, and plot the results.

Step 1 of 10

Given data:

Inlet pressure to the nozzle is, $P_1 = 45$ psia

Inlet temperature to the nozzle is, $T_1 = 940°F = 1489.67$ R

Exit velocity is, $C_2 = 650$ ft/s

Nozzle efficiency varies, $\eta_{isen} = 0.8$ to 1

Question:

Reconsider Prob. 8-132E. Using EES (or other) software, study the effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on both the exit temperature and pressure of the air, and plot the results.

Step 1 of 10

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Inlet pressure to the nozzle is, $P_1 = 45$ psia

Inlet temperature to the nozzle is, $T_1 = 940°F = 1489.67$ R

Exit velocity is, $C_2 = 650$ ft/s

Nozzle efficiency varies, $\eta_{isen} = 0.8$ to 1

Step 2 of 10

Give all the inputs in the EES software as given below

T1 =940 [F]

Step 2 of 10

Give all the inputs in the EES software as given below

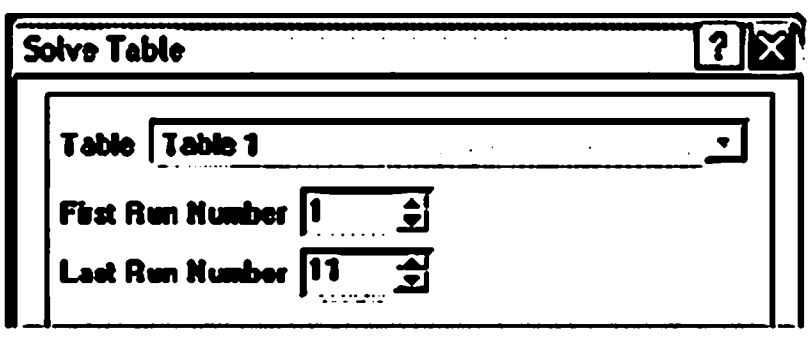
T1 =940 [F]

Fill all the values of η_{isen}.

▷	1 η_{isen}	2 P2 [psia]	3 T2 [F]	4 $T2_s$ [F]
Run 1	0.8			
Run 2	0.82			
Run 3	0.84			
Run 4	0.86			
Run 5	0.88			
Run 6	0.9			
Run 7	0.92			
Run 8	0.94			
Run 9	0.96			
Run 10	0.98			
Run 11	1			

Step 5 of 10

Now solve the table for respective P2 and T2 values by pressing F3.

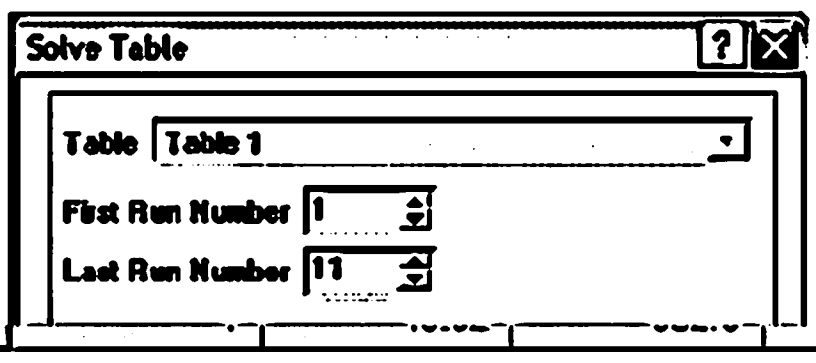


Fill all the values of η_{isen}.

▷	1 η_{isen}	2 P2 [psia]	3 T2 [F]	4 $T2_s$ [F]
Run 1	0.8			
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Run 10	0.98			
Run 11	1			

Step 5 of 10

Now solve the table for respective P2 and T2 values by pressing F3.



Step 7 of 10

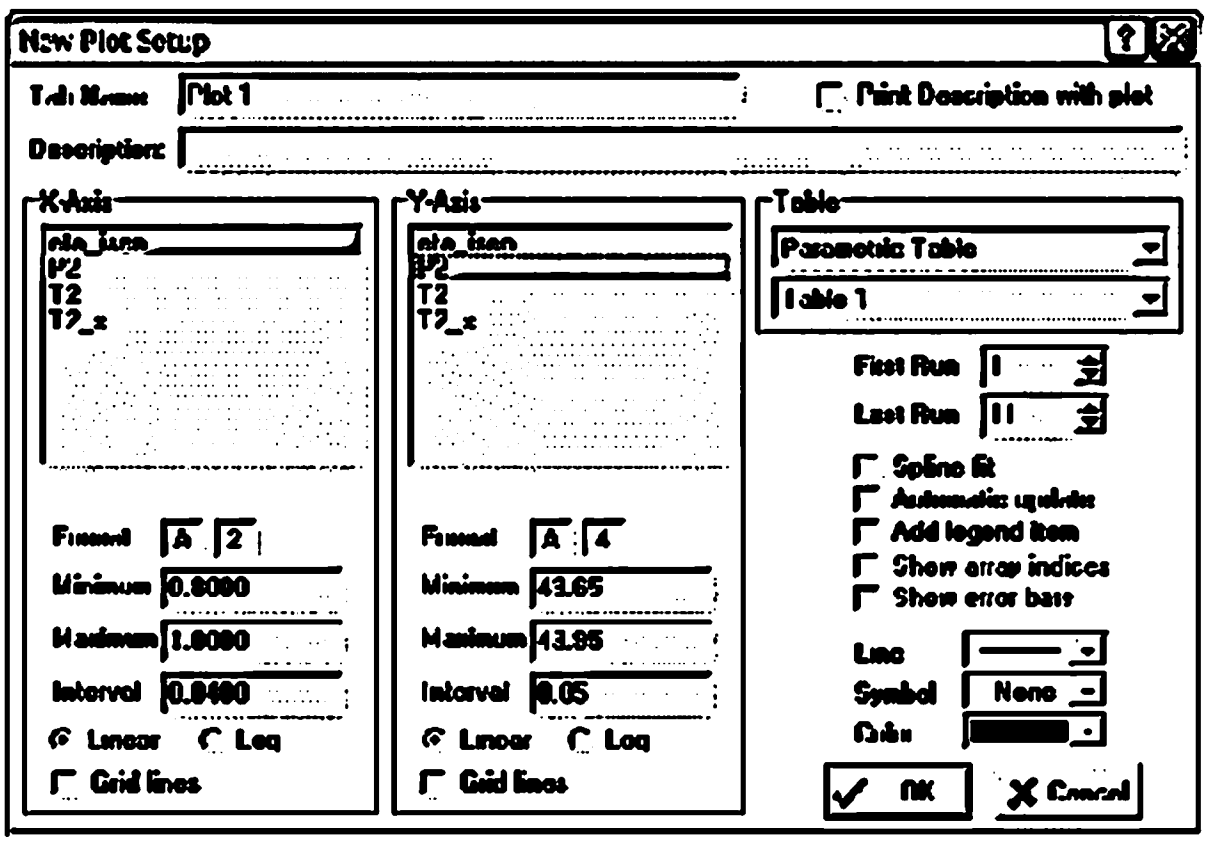
For getting the plot we should go for

Plots → New plot window

We should fill all the required fields by taking η_{isen} in x-axis and P2 in y-axis.

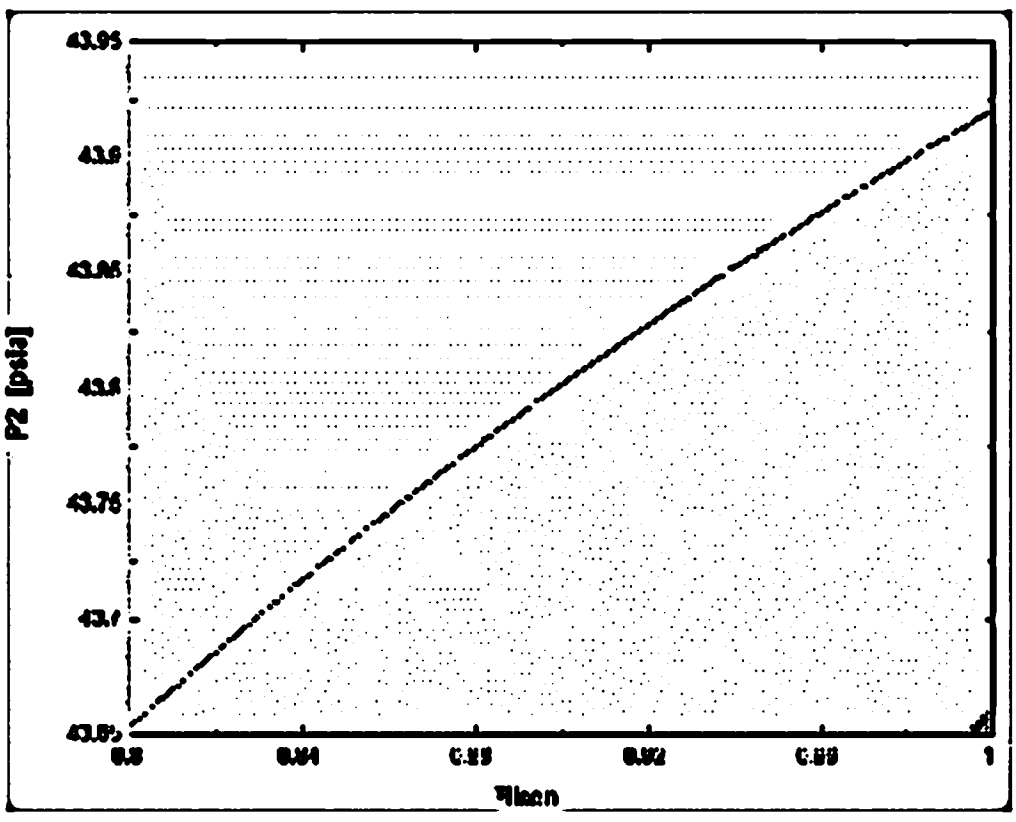
Step 7 of 10

For getting the plot we should go for
Plots → New plot window
We should fill all the required fields by taking η_{ksm} in x-axis and P2 in y-axis.



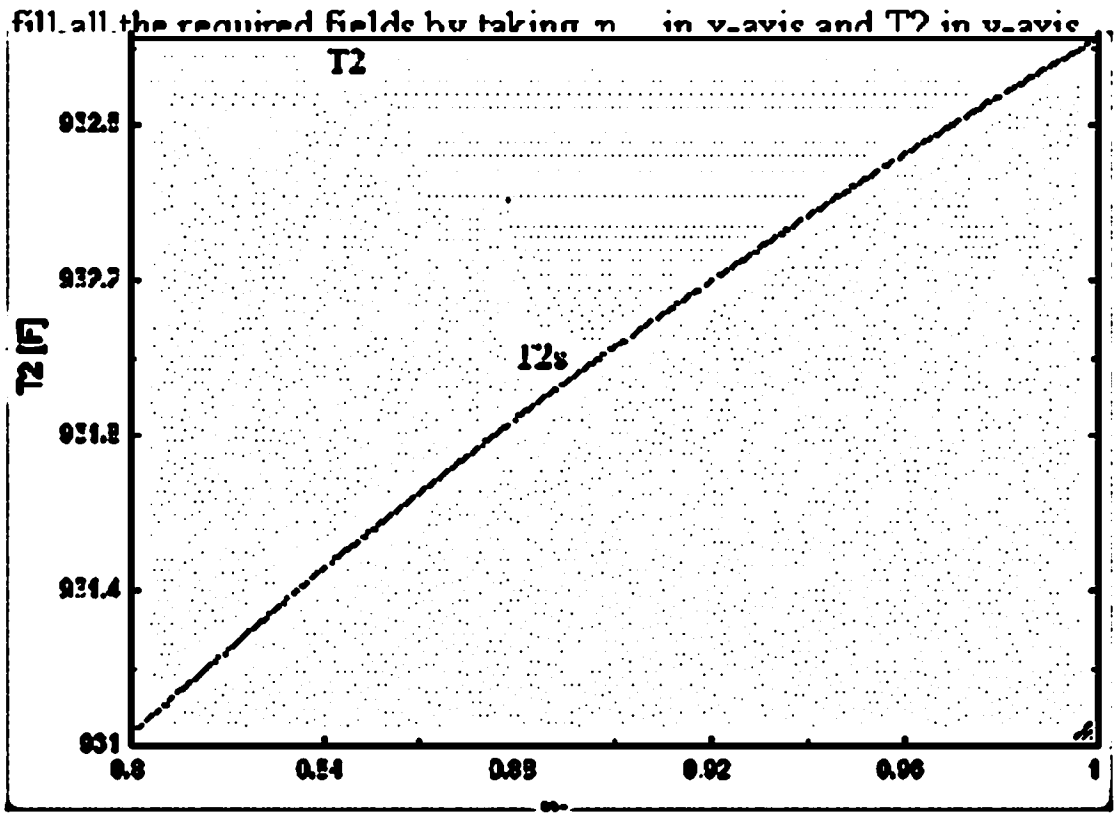
Step 8 of 10

After that we get the plot as



Step 9 of 10

For getting the plot we should go for
Plots → New plot window
We should fill all the required fields by taking η_{ksm} in x-axis and T2 in y-axis



Step 9 of 10

For getting the plot we should go for
Plots → New plot window
We should fill all the required fields by taking η_{ksm} in x-axis and T2 in y-axis

Question:

8.134P

The exhaust nozzle of a jet engine expands air at 300 kPa and 180°C adiabatically to 100 kPa. Determine the air velocity at the exit when the inlet velocity is low and the nozzle isentropic efficiency is 96 percent.

Step 1 of 3

Find the exit temperature of air in case of isentropic process by using isentropic relation during the process 1-2s.

$$\frac{T_{2s}}{T_1}=\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s}=T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Here, T_1 is temperature of air at nozzle inlet, P_1 is pressure of air at nozzle inlet, P_2 is pressure of air at nozzle exit T_{2s} is isentropic temperature of air at nozzle exit and k is isentropic index of air.

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Step 1 of 3

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Here, T_1 is temperature of air at nozzle inlet, P_1 is pressure of air at nozzle inlet, P_2 is pressure of air at nozzle exit T_{2s} is isentropic temperature of air at nozzle exit and k is isentropic index of air.

Find the isentropic kinetic energy change by using the equation,

$$(\Delta kE)_{\text{isentropic}}=c_p(T_1-T_{2s})$$

Here, c_p is specific heat of air at constant pressure.

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p

$$\begin{aligned}(\Delta kE)_{\text{isentropic}}&=(1.005\ \text{kJ/kg}\cdot\text{K})(453-331.221)\\&=122.387\,\text{kJ/kg}\end{aligned}$$

Find the actual kinetic energy change by using the efficiency of the nozzle,

$$\eta_N=\frac{(\Delta kE)_{\text{actual}}}{(\Delta kE)_{\text{isentropic}}}$$

$$(\Delta kE)_{\text{actual}}=\eta_N(\Delta kE)_{\text{isentropic}}$$

Substitute 0.96 for η_N

$$\begin{aligned}(\Delta kE)_{\text{actual}}&=0.96\times122.387\\&=117.491\,\text{kJ/kg}\end{aligned}$$

Find the actual velocity of air at nozzle exit by using the expression,

$$(\Delta kE)_{\text{actual}}=\frac{V_2^2-V_1^2}{2}$$

Find the isentropic kinetic energy change by using the equation,

$$(\Delta kE)_{\text{isentropic}}=c_p(T_1-T_{2s})$$

Here, c_p is specific heat of air at constant pressure.

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p

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$$(\Delta kE)_{\text{actual}}=\eta_N(\Delta kE)_{\text{isentropic}}$$

Substitute 0.96 for η_N

$$\begin{aligned}(\Delta kE)_{\text{actual}}&=0.96\times122.387\\&=117.491\,\text{kJ/kg}\end{aligned}$$

Question: 8.135P

An adiabatic diffuser at the inlet of a jet engine increases the pressure of the air that enters the diffuser at 13 psia and 30°F to 20 psia. What will the air velocity at the diffuser exit be if the diffuser isentropic efficiency defined as the ratio of the actual kinetic energy change to the isentropic kinetic energy change is 82 percent and the diffuser inlet velocity is 1000 ft/s?

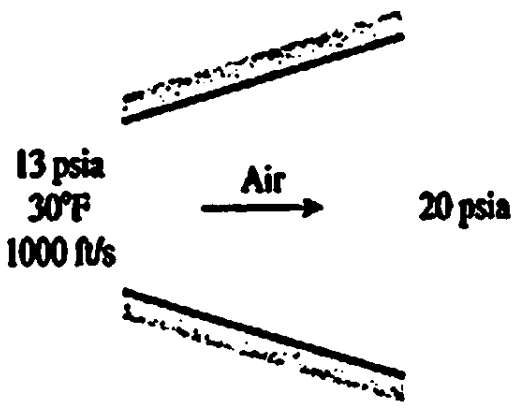


FIGURE P8–135E

Step 1 of 4

Obtain the properties of air from ideal gas properties of gases,

Specific heat ratio $k = 1.4$

Specific heat at constant pressure $c_p = 0.24\ \text{Btu/lbm} \cdot \text{R}$

Find the isentropic exit temperature of the diffuser by using isentropic relation,

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

Here, T_1 is the initial temperature of the air, T_{2s} is the final isentropic temperature, P_1 is the initial pressure of air, P_2 is final pressure of air and k is specific heat ratio of air.

Substitute 13 psia for P_1, 20 psia for P_2 and $\left[30°\text{F} = (460+30)°\text{R}\right]$ for T_1

$$\frac{T_2}{(30+460)} = \left(\frac{20}{13}\right)^{(1.4-1)/1.4}$$

$$\frac{T_2}{490} = 1.1306$$

$$T_{2s} = 553.99\ \text{R}$$

Step 2 of 4

Apply the energy balance for this steady-flow system

Obtain the properties of air from ideal gas properties of gases,

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$$\frac{T_2}{(30+460)} = \left(\frac{20}{13}\right)^{(1.4-1)/1.4}$$

$$\frac{T_2}{490} = 1.1306$$

$$T_{2s} = 553.99\ \text{R}$$

Step 2 of 4

Apply the energy balance for this steady-flow system

Their actual enthalpy drop is

$$c_p(T_1 - T_2) = \eta c_p(T_1 - T_{2s})$$

Step 4 of 4

Substitute enthalpy drop in the energy balance equation,

$$c_p(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2}$$

$$\eta c_p(T_1 - T_{2s}) = \frac{V_2^2 - V_1^2}{2}$$

Here, V_1 is diffuser inlet velocity, V_2 is diffuser exit velocity

Substitute 0.82 for η, 1000 ft/s for V_1

$$0.82(0.24)(490 - 553.99) = \left(\frac{V_2^2 - 1000^2}{2}\right)\frac{\text{ft}^2}{\text{s}^2}\left(\frac{1\ \text{Btu/lbm}}{25037\ \text{ft}^2/\text{s}^2}\right)$$

$$-12.593 \times 25037 = \frac{V_2^2 - 1000^2}{2}$$

$$V_2^2 = 369418.118$$

$$V_2 = 607.79\ \text{ft/s}$$

Therefore, exit velocity of the diffuser is **607.79 ft/s**

Question: 8.136P

Refrigerant-134a is expanded adiabatically from 100 psia and 100°F to a saturated vapor at 10 psia. Determine the entropy generation for this process, in Btu/lbm R.

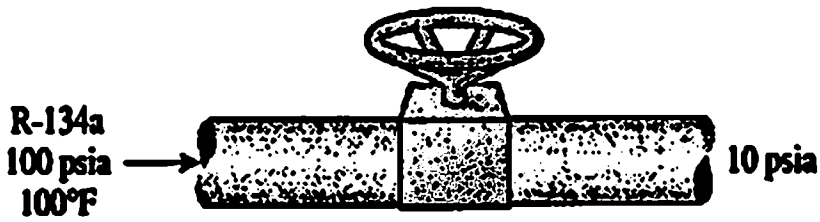


FIGURE P8–136E

Step 1 of 2

Given

Initial pressure in the pipe is, $P_1 = 100$ psia

Initial temperature in the pipe is, $T_1 = 100°F = 559.67$ R

Final pressure in the pipe is, $P_2 = 10$ psia

Step 2 of 2

From superheated R-134a tables at pressure $P_1 = 100$ psia and temperature $T_1 = 100°F$, the superheated entropy is

$s_1 = 0.229$ Btu/lbm.R

At pressure $P_2 = 10$ psia from saturated R-134a tables the entropy of saturated vapor is

$s_2 = s_g = 0.22948$ Btu/lbm.R

By applying entropy balance

$S_{in} - S_{out} + S_{gen} = \Delta S$

Let us assume the process is steady. So, $\Delta S = 0$

$m_1 s_1 - m_2 s_2 + S_{gen} = 0$

$S_{gen} = m_2 s_2 - m_1 s_1$

As there is no flow outside the mass flow rate will be same. $m_2 = m_1$

$S_{gen} = m_1 s_2 - m_1 s_1$

$S_{gen} = m_1 (s_2 - s_1)$

$s_{gen} = s_2 - s_1$

$s_{gen} = 0.22948 - 0.229$

$\boxed{s_{gen} = 0.00048 \text{ Btu/lbm.R}}$

Therefore the entropy generated is 0.00048 Btu/lbm.R .

Question: 8.137P

Oxygen enters an insulated 12-cm-diameter pipe with a velocity of 70 m/s. At the pipe entrance, the oxygen is at 240 kPa and 20°C; and, at the exit, it is at 200 kPa and 18°C. Calculate the rate at which entropy is generated in the pipe.

Step 1 of 4

Given

Diameter of the pipe $D = 12\,\text{cm} = 0.12\,\text{m}$

Velocity at inlet of the pipe $V_1 = 70\,\text{m/s}$

Pressure of O_2 at inlet of the pipe $P_1 = 240\,\text{kPa}$

Temperature of O_2 at inlet $T_1 = 20°\text{C}$

$$= (20 + 273)\,K$$

$$T_1 = 293\,\text{K}$$

Pressure of O_2 at exit of the pipe $P_2 = 200\,\text{kPa}$

Temperature of O_2 at exit of pipe $T_2 = 18°\,\text{C}$

$$= (18 + 273)\,\text{K}$$

$$T_2 = 291\,\text{K}$$

Step 2 of 4

Properties of oxygen are

$R = 0.2598\,\text{kJ/kg K}$

$C_p = 0.918\,\text{kJ/kg K}$

Specific volume of O_2 at inlet

$$v_1 = \frac{RT_1}{P_1}$$

$$v_1 = \frac{(0.2598)\times(293)}{240}$$

$$v_1 = 0.3172\,\text{m}^3/\text{kg}$$

Step 3 of 4

Mass flow rate of oxygen

$$\dot{m} = \frac{A_1 V_1}{v_1}$$

Where area of the pipe

$$A_1 = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.12)^2$$

$$A_1 = 0.0113\,\text{m}^2$$

Then

$$\dot{m} = \frac{A_1 V_1}{v_1}$$

$$= \frac{(0.0113)\times(70)}{(0.3172)}$$

$$\dot{m} = 2.4936\,\text{kg/s}$$

Step 4 of 4

Rate of entropy generation in the pipe

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1)$$

$$\dot{S}_{gen} = \dot{m}\left(C_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$

$$= (2.496)\left[0.918\ln\frac{291}{293} - 0.2598\ln\frac{200}{240}\right]$$

$$= 2.496\times 0.041079$$

$$\boxed{\dot{S}_{gen} = 0.10253\,\text{kW/K}}$$

Question: 8.138P

Nitrogen is compressed by an adiabatic compressor from 100 kPa and 25°C to 800 kPa and 307°C. Calculate the entropy generation for this process, in kJ/kg-K.

Step 1 of 3

Given

Inlet pressure to the compressor is, $P_1 = 100\ \text{kPa}$

Inlet temperature to the compressor is, $T_1 = 25°\text{C} = 298\ \text{K}$

Exit pressure to the compressor is, $P_2 = 800\ \text{kPa}$

Exit temperature to the compressor is, $T_2 = 307°\text{C} = 580\ \text{K}$

Step 2 of 3

The average temperature is,

$$T_{av} = \frac{T_1 + T_2}{2}$$

$$T_{av} = \frac{298 + 580}{2}$$

$$T_{av} = \frac{878}{2}$$

$$T_{av} = 439\ \text{K}$$

From ideal gas specific heats of nitrogen at temperature $T_{av} = 439\ \text{K}$,

The specific heat of nitrogen is

$c_p = 1.048\ \text{kJ/kg.K}$

The gas constant of nitrogen is

$R = 0.2968\ \text{kJ/kg.K}$

Step 3 of 3

The entropy balance is given by

$$S_{in} - S_{out} + S_{gen} = \Delta S$$

Let assume the process is steady. So, $\Delta S = 0$

$$m_1 s_1 - m_2 s_2 + S_{gen} = 0$$

$$S_{gen} = m_2 s_2 - m_1 s_1$$

As there is no flow outside the mass flow rate will be same. $m_2 = m_1$

$$S_{gen} = m_1 s_2 - m_1 s_1$$

$$S_{gen} = m_1 (s_2 - s_1)$$

$$s_{gen} = s_2 - s_1$$

$$s_{gen} = c_p \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$s_{gen} = 1.048 \times \ln\left(\frac{580}{298}\right) - 0.2968 \times \ln\left(\frac{800}{100}\right)$$

$$s_{gen} = 1.048 \times \ln(1.946) - 0.2968 \times \ln(8)$$

$$s_{gen} = 1.048 \times (0.6659) - 0.2968 \times (2.079)$$

$$s_{gen} = 0.6978 - 0.617$$

$$\boxed{s_{gen} = 0.0808\ \text{kJ/kg.K}}$$

Therefore the entropy generation for this process is 0.0808 kJ/kg.K.

Question: 8.139P

Consider a family of four, with each person taking a 5-min shower every morning. The average flow rate through the shower head is 12 L/min. City water at 15°C is heated to 55°C in an electric water heater and tempered to 42°C by cold water at the T-elbow of the shower before being routed to the shower head. Determine the amount of entropy generated by this family per year as a result of taking daily showers.

Step 1 of 5

Members in the family, $n = 4$

Time taking for shower,

Average flow rate through the shower head, $\dot{V} = 12\,\text{L/min}$

Initial temperature f the water, $T_i = 15°\text{C}$

Final temperature of the water, $T_o = 55°\text{C}$

Temperature of the cold water for temper, $T_c = 42°\text{C}$

Step 2 of 5

Calculate the mass flow rate of the water,

$$\dot{m} = \rho\dot{V}$$

Here,

The density of water is ρ

The average flow rate through the shower head is $\dot{V}$

Substitute $12\,\text{L/min}$ for $\dot{V}$ and $1000\,\text{kg/m}^3$ for ρ,

$$\begin{aligned}\dot{m} &= \rho\dot{V}\\ &= 1000\,\text{kg/m}^3 \times 12\,L/\text{min}\\ &= 1000\,\text{kg}/10^3\,\text{L} \times 12\,L/\text{min}\\ &= 12\,\text{kg/min}\end{aligned}$$

Step 3 of 5

From the mass balance equation,

$$\dot{m}_{\text{cold}} + \dot{m}_{\text{hot}} = \dot{m}_{\text{mixed}}$$

The entropy balance equation for the steady floe system,

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + S_{\text{gen}} = 0$$

Therefore,

$$S_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

And $s_1 = s_2$

Step 4 of 5

Calculate the entropy generation,

$$\begin{aligned}S_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2\\ &= \dot{m}_3 s_3 - (\dot{m}_1 + \dot{m}_2) s_1\\ &= (\dot{m}_1 + \dot{m}_2) s_3 - (\dot{m}_1 + \dot{m}_2) s_1\\ &= (\dot{m}_1 + \dot{m}_2)(s_3 - s_1)\\ &= -\dot{m}_3 c_p \ln\left(\frac{T_C}{T_i}\right)\end{aligned}$$

Here,

The mass flow rate of the water is $\dot{m} = \dot{m}_3$

Initial temperature of the water is T_i

The temperature of the cold water is T_C

The specific heat of the water is c_p

Substitute $12\,\text{kg/min}$ for $\dot{m}_3$, $4.18\,\text{kJ/kg}\cdot\text{K}$ for c_p, $278\,\text{K}$ for T_i and $315\,\text{K}$ for T_C, .

$$\begin{aligned}S_{\text{gen}} &= \dot{m}_3 c_p \ln\left(\frac{T_C}{T_i}\right)\\ &= 12 \times 4.18 \times \ln\left(\frac{315}{288}\right)\\ &= 4.495\,\text{kJ/min}\cdot\text{K}\end{aligned}$$

Step 5 of 5

Now calculate the total amount of entropy generated per year,

$$S_{\text{gen}} = \dot{S}_{\text{gen}}(t) n \times N$$

Here,

The number of members in the family is n

The number of days per year is N

Time taking for shower is t

Now substitute $5\,\text{min}$ for t, 4 for n, $365\,\text{days}$ for N and $4.495\,\text{kJ/min}\cdot\text{K}$ for $\dot{S}_{\text{gen}}$,

$$\begin{aligned}S_{\text{gen}} &= \dot{S}_{\text{gen}}(t) n \times N\\ &= 4.495 \times 5 \times 4 \times 365\\ &= 32813.10\ \text{kJ/K/year}\end{aligned}$$

Therefore, the total amount of entropy generated per year is $\boxed{32813.10\ \text{kJ/K/year}}$

Question: 8.140P

Cold water (cp = 4.18 kJ/kg·°C) leading to a shower enters a well-insulated, thin-walled, double-pipe, counter-flow heat exchanger at 10°C at a rate of 0.95 kg/s and is heated to 70°C by hot water (cp = 4.19 kJ/kg·°C) that enters at 85 °C at a rate of 1.6 kg/s. Determine (*a*) the rate of heat transfer and (*b*) the rate of entropy generation in the heat exchanger.

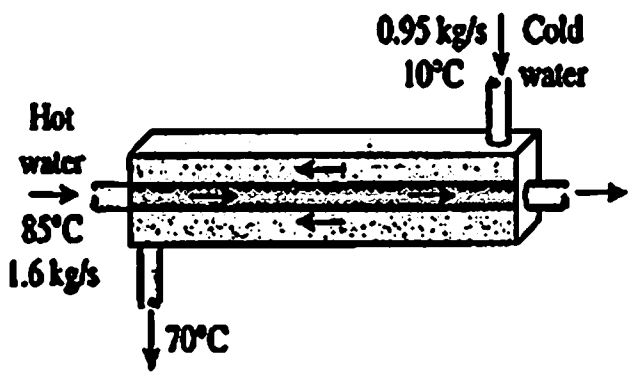
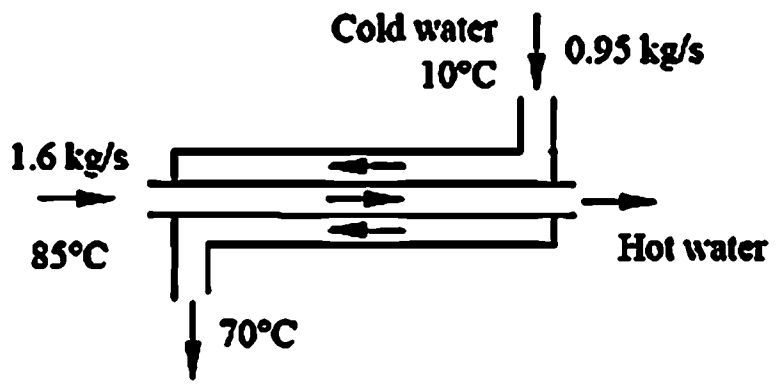


FIGURE P8–140

Step 1 of 5

Draw the double pipe, counter flow heat exchanger of hot water and cold water flowing through pipes as shown below:



Step 2 of 5

(a)

The amount of heat lost from hot water is equal to the amount of heat gained to cold water.

$$\dot{Q}_c = \dot{Q}_h$$

Calculate the amount of heat gained to cold water by using the inlet and exit temperatures of water,

$$\dot{Q}_c = \dot{m}_c \times c_{p,c} \times (T_{c2} - T_{c1})$$

Here, $\dot{m}_c$ is mass flow rate of cold water, $c_{p,c}$ is specific heat of cold water, T_{c2} is exit temperature of cold water and T_{c1} is inlet temperature of cold water.

Substitute 0.95kg/s for $\dot{m}_c$, 4.18kJ/kg·°C for $c_{p,c}$, 70°C for T_{c2} and 10°C for T_{c1}

$$\dot{Q}_c = 0.95 \times 4.18 \times (70 - 10)$$
$$= 238.26\text{kW}$$

Therefore, the rate of heat transfer is $\boxed{238.26\text{ kW}}$

Step 3 of 5

(b)

Find the heat transfer per kg of cold water,

$$q_c = \frac{\dot{Q}_c}{\dot{m}_c}$$
$$= \frac{238.26}{0.95}$$
$$= 250.8\text{kJ/kg}$$

Step 4 of 5

Find the exit temperature of hot water by using the expression,

$$\dot{Q}_h = \dot{m}_h\, c_{p,h} \times (T_{h1} - T_{h2})$$

Here, $\dot{m}_h$ is mass flow rate of hot water, $c_{p,h}$ is specific heat of hot water, T_{h2} is exit temperature of hot water and T_{h1} is inlet temperature of hot water.

Substitute 1.6kg/s for $\dot{m}_h$, 4.19kJ/kg·°C for $c_{p,h}$ and 85°C for T_{h1}

$$238.26 = 1.6 \times 4.19 \times (85 - T_{h2})$$
$$T_{h2} = 49.46°\text{C}$$

Step 5 of 5

Find the entropy generation in the heat exchanger by using the equation,

$$S_{gen} = \dot{m}_c (s_{c2} - s_{c1}) + \dot{m}_h (s_{h2} - s_{h1})$$
$$S_{gen} = \dot{m}_c \times c_{p,c} \times \ln\left(\frac{T_{c2}}{T_{c1}}\right) + \dot{m}_h \times c_{p,h} \times \ln\left(\frac{T_{h2}}{T_{h1}}\right)$$
$$= 0.95 \times 4.18 \times \ln\left(\frac{70+273}{10+273}\right) + 1.6 \times 4.19 \times \ln\left(\frac{49.46+273}{85+273}\right)$$
$$= 0.95 \times 4.18 \times \ln(1.212) + 1.6 \times 4.19 \times \ln(0.9007)$$
$$= 0.95 \times 4.18 \times (0.192) + 1.6 \times 4.19 \times (-0.1045)$$
$$= 0.7624 - 0.7005$$
$$= 0.0619\text{kW/K}$$

Therefore, the entropy generation in the heat exchanger is $\boxed{0.0619\text{ kW/K}}$

Question: 8.141P

Air (*cp* = 1.005 kf/kg·°C) is to be preheated by hot exhaust gases in a cross-flow heat exchanger before it enters the furnace. Air enters the heat exchanger at 95 kPa and 20°C at a rate of 1.6 m3 /s. The combustion gases (*cp* = 1.10 kJ/kg·°C) enter at 180°C at a rate of 2.2 kg/s and leave at 95 °C. Determine (*a*) the rate of heat transfer to the air, (*b*) the outlet temperature of the air, and (*c*) the rate of entropy generation.

Step 1 of 3

(a) Let $\dot{Q}$ be the rate of Heat transfer

Calculate the rate of Heat transfer by using energy balance equation,

$$\dot{Q} = \dot{m}_h \times c_p \times (T_1 - T_2)$$

Here,

The mass flow rate of hot gas is $\dot{m}_h$

The specific heat of gases is c_p

The inlet temperature of hot gas is T_1

The outlet temperature of hot gas is T_2

Substitute $2.2\,\text{kg/s}$ for $\dot{m}_h$, $1.10\ \text{kJ/kg°C}$ for c_p, $453\,\text{K}$ for T_1 and $368\,\text{K}$ for T_2,

$$\begin{aligned}\dot{Q} &= \dot{m}_h \times c_p \times (T_1 - T_2)\\ &= (2.2\times 1.1)\times(180-95)\\ &= 205.7\,\text{kW}\end{aligned}$$

Therefore, the rate of heat transfer is $\boxed{205.7\,\text{kW}}$

Step 2 of 3

b) Let T_{out} be the outlet temperature of the air

Calculate the mass flow rate,

$$P\dot{V} = \dot{m}RT_{in}$$

Here,

The pressure of air is P

The volume flow rate is $\dot{V}$

The temperature of air is T_{in}

Substitute $95\,\text{kPa}$ for P, $1.6\,\text{m}^3/\text{s}$ for $\dot{V}$, $287\,\text{J/kg}\cdot\text{K}$ for R and $293\ \text{K}$ for T_{in},

$$\begin{aligned}P\dot{V} &= \dot{m}RT_{in}\\ (95\times 10^3 \times 1.6) &= \dot{m}(287\times 293)\\ \dot{m} &= 1.808\,\text{kg/s}\end{aligned}$$

Calculate outlet temperature of the air,

Applying for the air

$$\dot{Q} = \dot{m}\times c_p \times (T_{out} - T_{in})$$

Substitute $205.7\,\text{kW}$ for $\dot{Q}$, $1.808\,\text{kg/s}$ for $\dot{m}$ and $20°\text{C}$ for T_{in},

$$\begin{aligned}\dot{Q} &= \dot{m}\times c_p \times (T_{out} - T_{in})\\ 205.7 &= (1.808\times 1.005)\times(T_{out} - 20)\\ (T_{out} - 20) &= 113.21\\ T_{out} &= 133.21°\text{C}\end{aligned}$$

Therefore the outlet temperature of air is $\boxed{133.21°\text{C}}$

Step 3 of 3

c) Let S_{gen} be the entropy generation

Calculate the entropy generation,

$$S_{gen} = \dot{m}_h(s_2 - s_1) + \dot{m}(s_4 - s_3)$$

$$S_{gen} = \dot{m}_h \times c_p \times \ln\left(\frac{T_2}{T_1}\right) + \dot{m}\times c_p \times \ln\left(\frac{T_{out}}{T_{in}}\right)$$

$$S_{gen} = (2.2\times 1.1)\times \ln\left(\frac{368}{453}\right) + (1.808\times 1.005)\times \ln\left(\frac{133.21+273}{293}\right)$$

$$S_{gen} = (-0.503) + 0.594$$

$$S_{gen} = 0.091\,\text{kW/K}$$

Therefore the entropy generation is $\boxed{0.091\,\text{kW/K}}$

Question: 8.142P

A well-insulated, shell-and-tube heat exchanger is used to heat water (cp = 4.18 kJ/kg·°C) in the tubes from 20 to 70°C at a rate of 4.5 kg/s. Heat is supplied by hot oil (cp = 2.30 kJ/kg·°C) that enters the shell side at 170°C at a rate of 10 kg/s. Disregarding any heat loss from the heat exchanger, determine (*a*) the exit temperature of the oil and (*b*) the rate of entropy generation in the heat exchanger.

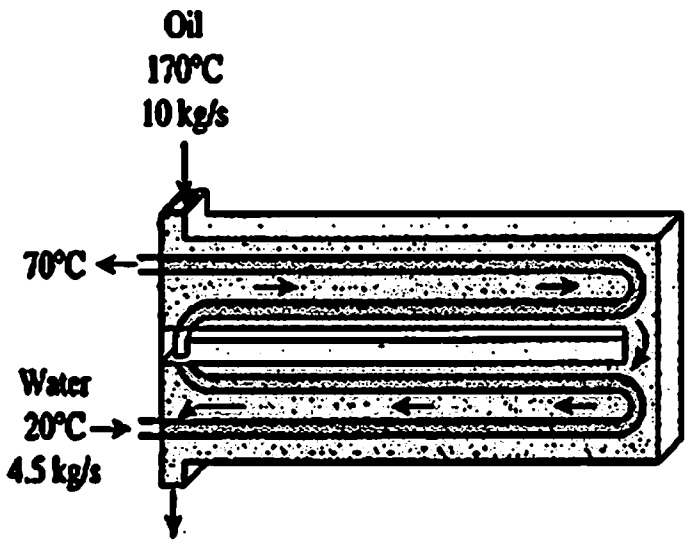


FIGURE P8–142

Step 1 of 3

(a) Let $\dot{Q}$ be the rate of Heat transfer

Calculate the rate of Heat transfer by using energy balance equation,

$$\dot{Q} = \dot{m}_w \times c_p \times (T_1 - T_2)$$

Here,

The mass flow rate of hot gas is $\dot{m}_w$

The specific heat of gases is c_p

The inlet temperature of hot gas is T_{out}

The outlet temperature of hot gas is T_{in}

Substitute $4.5\,\text{kg/s}$ for $\dot{m}_w$, $4.18\ \text{kJ/kg°C}$ for c_p, $70°\text{C}$ for T_{out} and $20°\text{CK}$ for T_{out},

$$\begin{aligned}\dot{Q} &= \dot{m}_w \times c_p \times (T_{out} - T_{in}) \\ &= (4.5 \times 4.18) \times (70 - 20) \\ &= 940.5\,\text{kW}\end{aligned}$$

Therefore, the rate of heat transfer is $\boxed{940.5\,\text{kW}}$

Step 2 of 3

b) Let T_2 be the exit temperature of the oil

Calculate outlet temperature of the air,

Applying for the air

$$\dot{Q} = \dot{m}_{oil} \times c_p \times (T_1 - T_2)$$

Substitute $940.5\,\text{kW}$ for $\dot{Q}$, $10\,\text{kg/s}$ for $\dot{m}_{oil}$ and $170°\text{C}$ for T_1,

$$\begin{aligned}\dot{Q} &= \dot{m}_{oil} \times c_p \times (T_1 - T_2) \\ 940.5 &= (10 \times 2.3) \times (170 - T_2) \\ (170 - T_2) &= 40.89 \\ T_2 &= 129.11°\text{C}\end{aligned}$$

Therefore the exit temperature of air is $\boxed{129.11°\text{C}}$

Step 3 of 3

c) Let S_{gen} be the entropy generation

Calculate the entropy generation,

$$S_{gen} = \dot{m}_w (s_2 - s_1) + \dot{m}_{oil} (s_4 - s_3)$$

$$S_{gen} = \dot{m}_w \times c_p \times \ln\left(\frac{T_{out}}{T_{in}}\right) + \dot{m}_{oil} \times c_p \times \ln\left(\frac{T_2}{T_1}\right)$$

$$S_{gen} = (4.5 \times 4.18) \times \ln\left(\frac{343}{293}\right) + (10 \times 2.3) \times \ln\left(\frac{129.11 + 273}{443}\right)$$

$$S_{gen} = 2.96 + (-2.23)$$

$$S_{gen} = 0.73\,\text{kW/K}$$

Therefore the entropy generation, $\boxed{0.73\,\text{kW/K}}$

Question: 8.143P

Refrigerant-134a is throttled from 1200 kPa and 40°C to 200 kPa. Heat is lost from the refrigerant in the amount of 0.5 kJ/kg to the surroundings at 25°C. Determine (*a*) the exit temperature of the refrigerant and (*b*) the entropy generation during this process.

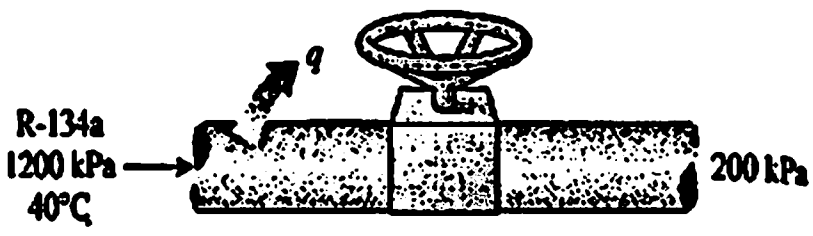


FIGURE P8–143

Step 1 of 6

Given data:

Initial pressure is, $P_1 = 1200\ \text{kPa}$

Initial temperature is, $T_1 = 40\ ^\circ\text{C}$

$= 313\ \text{K}$

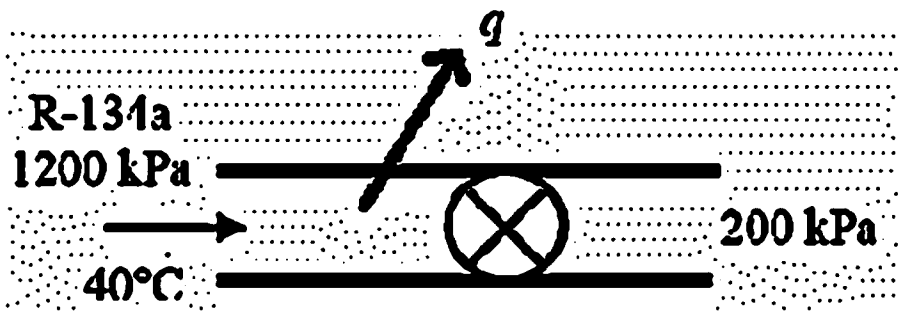
Final pressure is, $P_2 = 200\ \text{kPa}$

Heat lost by refrigerant is, $q_{out} = 0.5\ \text{kJ/kg}$

Surrounding temperature is, $T_{sur} = 25\ ^\circ\text{C}$

$= 298\ \text{K}$

Step 2 of 6



Step 3 of 6

a) As saturation temperature ($T_{sat} = 46.29\ ^\circ\text{C}$) at pressure $P_1 = 1200\ \text{kPa}$ is greater than initial temperature ($T_1 = 40\ ^\circ\text{C}$), we will consider the properties of saturation liquid condition at temperature ($T_1 = 40\ ^\circ\text{C}$)

From the saturated refrigerant 134a temperature table

At temperature $T_1 = 40\ ^\circ\text{C}$ the properties of saturated liquid R-134a are

$h_1 = 108.26\ \text{kJ/kg}$

$s_1 = 0.39486\ \text{kJ/kg°K}$

Step 4 of 6

As the process is throttling the enthalpy does not change.

$h_2 = h_1 = 108.26\ \text{kJ/kg}$

At pressure $P_2 = 200\ \text{kPa}$ saturation enthalpy is $h_{sat} = 244.46\ \text{kJ/kg}$ ($> h_2$) and so the final temperature is same as the saturation temperature at pressure $P_2 = 200\ \text{kPa}$.

The final temperature is $\boxed{T_2 = T_{sat@200kPa} = -10.09\ ^\circ\text{C}}$

Step 5 of 6

b) As the enthalpy is constant, the final entropy can be calculated by

$s_2 = \frac{h_2}{T_2}$ (Here enthalpy is also considered as its units is same as energy unit)

$s_2 = \frac{108.26}{-10.09+273}$

$s_2 = \frac{108.26}{262.91}$

$s_2 = 0.4118\ \text{kJ/kg°K}$

Step 6 of 6

The entropy generation in the process is given by

$s_{gen} = \Delta s_{ref} + \Delta s_{sur}$

$s_{gen} = s_2 - s_1 + \frac{q_{out}}{T_{sur}}$

$s_{gen} = 0.4118 - 0.39486 + \frac{0.5}{298}$

$s_{gen} = 0.4118 - 0.39486 + 0.00168$

$\boxed{s_{gen} = 0.018618\ \text{kJ/kg°K}}$

Therefore the final temperature is $-10.09\ ^\circ\text{C}$ and the entropy generation is $0.018618\ \text{kJ/kg°K}$

Question: 8.144P

In an ice-making plant, water at 0°C is frozen at 1 atmospheric pressure by evaporating saturated R- 134a liquid at -16°C. The refrigerant leaves this evaporator as a saturated vapor, and the plant is sized to produce ice at 0°C at a rate of 2500 kg/h. Determine the rate of entropy generation in § this plant.

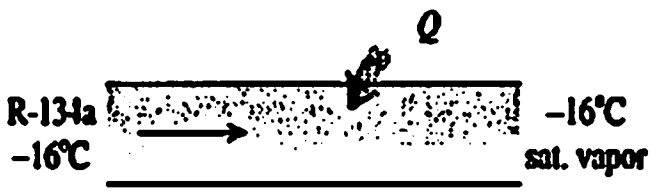


FIGURE P8–144

Step 1 of 3

Given

Inlet and exit temperature of the refrigerant is, $T_1 = T_2 = -16°C = 257\ K$

Temperature of ice is, $T_i = 0°C = 273\ K$

Inlet condition is saturated liquid.

Exit condition is saturated vapor.

Rate of ice produced is, $m_i = 2500\ kg/h = 0.6944\ kg/s$

Step 2 of 3

From saturated R-134a tables at temperature $T_1 = -16°C$

Enthalpy of evaporation is $h_{fg} = 210.18\ kJ/kg$

Entropy of evaporation is $s_{fg} = 0.81729\ kJ/kg°K$

From properties of common liquids, solids table latent heat of fusion of water is $h_f = 333.7\ kJ/kg$

Heat taken out of the plant during freezing (using heat of fusion) is given by

$Q_{out} = m_i \times h_f$

$Q_{out} = 0.69444 \times 333.7$

$Q_{out} = 231.734\ kW$

Mass flow rate of the refrigerant is given by

$$m_R = \frac{Q_{out}}{h_{fg}}$$

$$m_R = \frac{231.734}{210.18}$$

$m_R = 1.1025\ kg/s$

Step 3 of 3

The entropy balance is given by

$S_{in} - S_{out} + S_{gen} = \Delta S$

Let us assume the process is steady. So, $\Delta S = 0$

$$m_1 s_1 - m_2 s_2 + \frac{Q_{out}}{T_i} + S_{gen} = 0$$

$$S_{gen} = m_2 s_2 - m_1 s_1 - \frac{Q_{out}}{T_i}$$

$$S_{gen} = m_R (s_2 - s_1) - \frac{Q_{out}}{T_i}$$

$$S_{gen} = m_R (s_g - s_f) - \frac{Q_{out}}{T_i}$$

$$S_{gen} = m_R \times s_{fg} - \frac{Q_{out}}{T_i}$$

Substituting the known values in the above equation

$$S_{gen} = 1.1025 \times 0.81729 - \frac{231.734}{273}$$

$S_{gen} = 0.90106 - 0.8488$

$\boxed{s_{gen} = 0.05226\ kW/K}$

Therefore the entropy generated is 0.05226 kW/K.

Question: 8.145P

Water at 20 psia and 50° Fenters a mixing chamber at a rate of 300 lbm/min where it is mixed steadily with I steam entering at 20 psia and 240° F.The mixture leaves the chamber at 20 psia and 130°F, and heat is lost to the surrounding air at 70°F at a rate of 180 Btu/min. Neglecting the changes in kinetic and potential energies, determine the rate of entropy generation during this process?

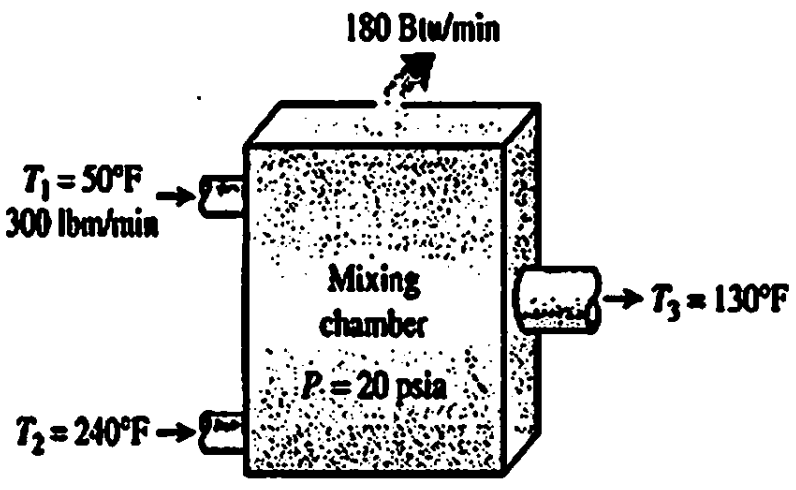


FIGURE P8–145E

Step 1 of 5

Given

Inlet pressure of water is, $P_1 = 20$ psia

Inlet temperature of water is, $T_1 = 50°F$

Mass flow rate of water is, $m_1 = 300$ lbm/min $= 5$ lbm/s

Inlet pressure of steam is, $P_2 = 20$ psia

Inlet temperature of steam is, $T_2 = 240°F$

Exit pressure of mixture is, $P_3 = 20$ psia

Exit temperature of mixture is, $T_3 = 130°F$

Surrounding temperature is, $T_0 = 70°F = 529.67°R$

Rate of heat loss is, $Q_{out} = 180$ Btu/min $= 3$ Btu/s

Step 2 of 5

At $P_1 = 20$ psia from saturated water tables

$T_{sat} = 227.92°F$

The inlet and exit temperatures are less than saturated temperature so we can take the values at saturated liquid condition of that temperature.

From saturated water tables at temperature $T_1 = 50°F$ the enthalpy and entropy of steam are as follows

$h_1 = h_f = 18.07$ Btu/lbm

$s_1 = s_f = 0.03609$ Btu/lbm.R

From saturated water tables at temperature $T_3 = 130°F$, the enthalpy and entropy of steam are as follows

$h_3 = h_f = 97.99$ Btu/lbm

$s_3 = s_f = 0.18174$ Btu/lbm.R

At pressure $P_2 = 20$ psia and temperature $T_2 = 240°F$, the enthalpy and entropy of steam are as follows

$h_2 = 1162.3$ Btu/lbm

$s_2 = 1.7406$ Btu/lbm.R

Step 3 of 5

Applying mass balance

$m_1 + m_2 = m_3$

Applying energy balance

$E_{in} - E_{out} = \Delta E$

As the process is steady, $\Delta E = 0$

$E_{in} - E_{out} = 0$

$E_{in} = E_{out}$

$m_1 h_1 + m_2 h_2 = m_3 h_3 + Q_{out}$

$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3 + Q_{out}$

$m_1 h_1 + m_2 h_2 = m_1 h_3 + m_2 h_3 + Q_{out}$

$m_2 h_2 - m_2 h_3 = m_1 h_3 - m_1 h_1 + Q_{out}$

$m_2 (h_2 - h_3) = m_1 (h_3 - h_1) + Q_{out}$

Step 4 of 5

Mass flow rate of steam is given by

$$m_2 = \frac{m_1 (h_3 - h_1) + Q_{out}}{h_2 - h_3}$$

$$m_2 = \frac{5 \times (97.99 - 18.07) + 3}{1162.3 - 97.99}$$

$$m_2 = \frac{5 \times 79.92 + 3}{1064.31}$$

$$m_2 = \frac{399.6 + 3}{1064.31}$$

$$m_2 = \frac{402.6}{1064.31}$$

$m_2 = 0.3783$ lbm/s

Step 5 of 5

The entropy balance is given by

$S_{in} - S_{out} + S_{gen} = \Delta S$

Let us assume the process is steady. So, $\Delta S = 0$

$S_{in} - S_{out} + S_{gen} = 0$

$$m_1 s_1 + m_2 s_2 - m_3 s_3 - \frac{Q_{out}}{T_0} + S_{gen} = 0$$

$$S_{gen} = m_3 s_3 - m_2 s_2 - m_1 s_1 + \frac{Q_{out}}{T_0}$$

$$S_{gen} = (m_1 + m_2) s_3 - m_2 s_2 - m_1 s_1 + \frac{Q_{out}}{T_0}$$

$$S_{gen} = [(5 + 0.3783) \times 0.18174] - [0.3783 \times 1.7406] - [5 \times 0.03609] + \frac{3}{529.67}$$

$$S_{gen} = [5.3783 \times 0.18174] - [0.3783 \times 1.7406] - [5 \times 0.03609] + \frac{3}{529.67}$$

$$S_{gen} = 0.97745 - 0.6585 - 0.18045 + 0.0057$$

$$\boxed{S_{gen} = 0.1442 \text{ Btu/s.R}}$$

Therefore the rate of entropy generation during this process is 0.1442 Btu/s.R .

Question: 8.146P

Steam is to be condensed on the shell side of a heat exchanger at 120°F. Cooling water enters the tubes at 60°F at a rate of 92 lbm/s and leaves at 73° F.Assuming the heat exchanger to be well-insulated, determine (*a*) the rate of heat transfer in the heat exchanger and (*b*) the rate of entropy generation in the heat exchanger.

Step 1 of 3

(a) Let $\dot{Q}$ be the rate of Heat transfer

From the properties of common liquids, solids and foods

Specific heat of water, $c_p = 1.01\ \text{Btu/lbm} \cdot \text{R}$

From the energy balance equation

Calculate the rate of heat transfer

$$\dot{Q} = \dot{m}_w \times c_p \times (T_1 - T_2)$$

Here,

The mass flow rate of hot gas is $\dot{m}_h$

The specific heat of gases is c_p

The inlet temperature of cooling water is T_1

The outlet temperature of cooling water is T_2

Substitute $92\ \text{lbm/s}$ for $\dot{m}_w$, $1.01\ \text{Btu/lbm} \cdot \text{R}$ for c_p, $520°\text{R}$ for T_1 and $533°\text{R}$ for T_2,

$$\begin{aligned}\dot{Q} &= \dot{m}_w \times c_p \times (T_2 - T_1) \\ &= (92 \times 1.01) \times (73 - 60) \\ &= 1207.96\ \text{Btu/s}\end{aligned}$$

Therefore, the rate of heat transfer, $\boxed{1207.96\ \text{Btu/s}}$

Step 2 of 3

b) Let S_{gen} be the outlet temperature of the air

From the properties of saturated water

At $T_{in} = 120°\text{F}$

The enthalpy, $h_{fg} = 1025.2\ \text{Btu/lbm}$

The entropy, $s_{fg} = 1.76856\ \text{Btu/lbm} \cdot \text{R}$

From the energy balance equation

Applying for the air

$$\dot{Q} = \dot{m}_s (h_{fg})$$

$$1207.96 = \dot{m}_s (1025.2)$$

$$\dot{m}_s = 1.178\ \text{lbm/s}$$

Step 3 of 3

Calculate the entropy generation,

$$\begin{aligned}S_{gen} &= \dot{m}_w (s_2 - s_1) + \dot{m}_s (s_4 - s_3) \\ &= \dot{m}_w \times c_p \times \ln\left(\frac{T_2}{T_1}\right) + \dot{m}_s \times (-s_{fg})\end{aligned}$$

Substitute $92\ \text{lbm/s}$ for $\dot{m}_w$, $1.01\ \text{Btu/lbm} \cdot \text{R}$ for c_p, $520°\text{R}$ for T_1, $533°\text{R}$ for T_2 and $1.178\ \text{lbm/s}$ for $\dot{m}_s$,

$$\begin{aligned}S_{gen} &= \dot{m}_w \times c_p \times \ln\left(\frac{T_2}{T_1}\right) + \dot{m}_s \times (-s_{fg}) \\ &= (92 \times 1.01) \times \ln\left(\frac{533}{520}\right) + (1.178)(-1.76856) \\ &= (92 \times 1.01) \times \ln\left(\frac{533}{520}\right) + (1.178)(-1.76856) \\ &= 2.294 + (-2.083) \\ &= 0.211\ \text{Btu/s} \cdot \text{R}\end{aligned}$$

Therefore the entropy generation, $\boxed{0.211\ \text{Btu/s} \cdot \text{R}}$

Question: 8.147P

In a dairy plant, milk at 4°C is pasteurized continuously at 72°C at a rate of 12 L/s for 24 hours a day and 365 days a year. The milk is heated to the pasteurizing temperature by hot water heated in a natural-gas-fired boiler that has an efficiency of 82 percent. The pasteurized milk is then cooled by cold water at 18°C before it is finally refrigerated back to 4°C. To save energy and money, the plant installs a regenerator that has an effectiveness of 82 percent. If the cost of natural gas is $1.04/therm (1 therm = 105,500 kJ), determine how much energy and money the regenerator will save this company per year and the annual reduction in entropy generation.

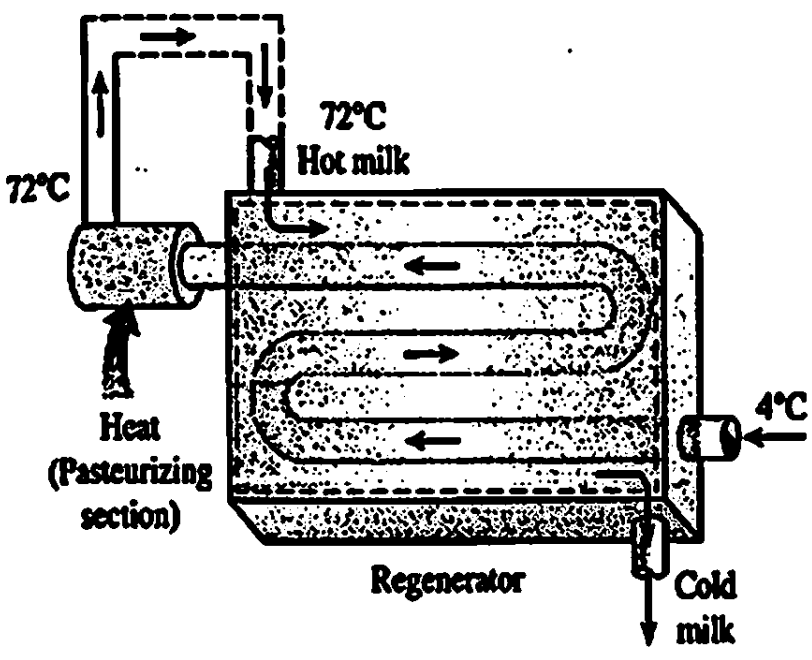


FIGURE P8–147

Step 1 of 4

From the properties of the milk,

Specific heat, $c_p = 3.79\,\text{kJ/kg}\cdot\text{K}$

Consider the density of the milk $\rho = 1\,\text{kg/L}$

Calculate the mass flow rate of the milk,

$$\begin{aligned}\dot{m}_{\text{milk}} &= \rho \dot{V}_{\text{milk}} \\ &= 1\,\text{kg/L} \times 12\,\text{L/sec} \\ &= 12\,\text{kg/sec}\end{aligned}$$

For the pasteurizing section, the energy balance for steady flow system,

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} = 0$$

Therefore,

$$E_{\text{in}} = E_{\text{out}}$$
$$Q_{\text{in}} + \dot{m}h_1 = \dot{m}h_2$$

Calculate the rate of heat transfer,

$$\begin{aligned}Q_{\text{in}} &= \dot{m}(h_2 - h_1) \\ &= \dot{m}c_p(T_2 - T_1) \\ &= 12 \times 3.79 \times (72 - 4) \\ &= 3092.64\,\text{kJ/sec}\end{aligned}$$

Step 2 of 4

Effectiveness of the regenerator, $\varepsilon_{\text{reg}} = 82\%$

Therefore, energy saved,

$$\begin{aligned}\dot{Q}_{\text{saved}} &= \varepsilon \dot{Q}_{\text{in}} \\ &= 0.28 \times 3092.64 \\ &= 2535.965\,\text{kJ/s}\end{aligned}$$

Efficiency of the boiler, $\eta_{\text{eff}} = 82\%$

So, the fuel saving,

$$\begin{aligned}\text{F}_{\text{saved}} &= \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} \\ &= \frac{2535.965}{0.82} \\ &= 3092.64\,\text{kJ/sec} \\ &= \frac{3092.64}{105500} \\ &= 0.02931\,\text{therm/sec}\end{aligned}$$

Step 3 of 4

Annual fuel saved,

$$\begin{aligned}\text{F}_{\text{annual saved}} &= 0.02931 \times (24 \times 3600) \times 365 \\ &= 924450.19\,\text{therm/year}\end{aligned}$$

Therefore, the fuel saved due to the regenerator per year is $\boxed{924450.19\,\text{therm/year}}$

Annual money saved,

$$\begin{aligned}\text{m}_{\text{annual saved}} &= F_{\text{annual saved}} \times C \\ &= 924450.19 \times 1.04 \\ &= \$961400/\text{yr}\end{aligned}$$

Therefore, the money saved due to the regenerator per year is $\boxed{\$961400/\text{yr}}$

Step 4 of 4

From the rate form of the entropy balance on an extended system,

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta \dot{S}_{\text{system}} = 0$$
$$\dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

So,

The reduction in the entropy transfer to water,

$$\begin{aligned}\dot{S}_{\text{gen,reduction}} &= \frac{\dot{Q}_{\text{out}}}{T_{\text{sp}}} \\ &= \frac{\dot{Q}_{\text{saved}}}{T_{\text{sp}}} \\ &= \frac{2536}{18 + 273} \\ &= 8.714\,\text{kW/K}\end{aligned}$$

So, reduction in the entropy transfer to water in a year,

$$\begin{aligned}\dot{S}_{\text{gen,reduction}} &= 8.714\,\text{kW/K} \times (24 \times 3600) \times 365 \\ &= 2.74 \times 10^5\,\text{kW/K/year}\end{aligned}$$

Therefore, the annual reduction is $\boxed{2.74 \times 10^5\,\text{kW/K/year}}$

Question: 8.148P

Stainless-steel ball bearings (ρ = 8085 kg/m3 and cp = 0.480 kJ/kg·°C) having a diameter of 1.8 cm are to be quenched in water at a rate of 1100 per minute. The balls leave the oven at a uniform temperature of 900°C and are exposed to air at 20°C for a while before they are dropped into the water. If the temperature of the balls drops to 850°C prior to quenching, determine (*a*) the rate of heat transfer from the balls to the air and (*b*) the rate of entropy generation due to heat loss from the balls to the air.

Step 1 of 7

Given

Diameter of each ball is, $D = 1.8\ \text{cm}$
$= 0.018\ \text{m}$

No. of balls quenched is, $n_{ball} = 1100\ \text{balls/min}$

Initial temperature of balls is, $T_1 = 900\ ^\circ\text{C}$
$= 1173\ \text{K}$

Final temperature of balls is, $T_2 = 850\ ^\circ\text{C}$
$= 1123\ \text{K}$

Surrounding temperature is, $T_{surr} = 20\ ^\circ\text{C}$
$= 293\ \text{K}$

Density of stainless-steel is, $\rho = 8085\ \text{kg/m}^3$

Specific heat of stainless-steel is, $c_p = 0.48\ \text{kJ/kg°K}$

Step 2 of 7

Mass of balls is given by

$$m = \rho \times V$$

$$m = \rho \times \frac{\pi D^3}{6}$$

$$m = 8085 \times \frac{\pi (0.018)^3}{6}$$

$$m = 8085 \times \frac{3.14 \times 5.832 \times 10^{-6}}{6}$$

$$m = 0.0247\ \text{kg}$$

Step 3 of 7

The of heat transfer per ball is given by

$$Q_{out/ball} = m c_p (T_1 - T_2)$$

$$Q_{out/ball} = 0.0247 \times 0.48 \times (1173 - 1123)$$

$$Q_{out/ball} = 0.0247 \times 0.48 \times 50$$

$$Q_{out/ball} = 0.5928\ \text{kJ}$$

Step 4 of 7

Total heat transfer is given by

$$Q_{out} = Q_{out/ball} \times n_{ball}$$

$$Q_{out} = 0.5928 \times 1100$$

$$\boxed{Q_{out} = 652.08\ \text{kJ/min} = 10.868\ \text{kW}}$$

Step 5 of 7

Entropy change in system is given by

$$\Delta s_{sys} = m \times c_p \times \ln\left(\frac{T_2}{T_1}\right)$$

$$\Delta s_{sys} = 0.0247 \times 0.48 \times \ln\left(\frac{1123}{1173}\right)$$

$$\Delta s_{sys} = 0.0247 \times 0.48 \times \ln(0.9574)$$

$$\Delta s_{sys} = 0.0247 \times 0.48 \times (-0.0435)$$

$$\Delta s_{sys} = -0.00051574\ \text{kJ/K}$$

Step 6 of 7

The entropy generation per ball is given by

$$\delta_{gen/ball} = \frac{Q_{out/ball}}{T_{surr}} + \Delta s_{sys}$$

$$\delta_{gen/ball} = \frac{0.5928}{293} + (-0.00051574)$$

$$\delta_{gen/ball} = 0.002023 - 0.00051574$$

$$\delta_{gen/ball} = 0.00151\ \text{kJ/K}$$

Step 7 of 7

The entropy generation is given by

$$\delta_{gen} = \delta_{gen/ball} \times n_{ball}$$

$$\delta_{gen} = 0.00151 \times 1100$$

$$\boxed{s_{gen} = 1.661\ \text{kJ/min.K} = 0.0277\ \text{kW/K}}$$

The rate of heat transfer from balls to air is 10.868 kW and the entropy generated is 0.0277 kW/K

Question: 8.149P

An ordinary egg can be approximated as a 5.5-cm-diameter sphere. The egg is initially at a uniform temperature of 8°C and is dropped into boiling water at 97°C. Taking the properties of the egg to be ρ = 1020 kg/m3 and cp = 3.32 kJ/kg·°C, determine (*a*) how much heat is transferred to the egg by the time the average temperature of the egg rises to 70°C and (*b*) the amount of entropy generation associated with this heat transfer process.

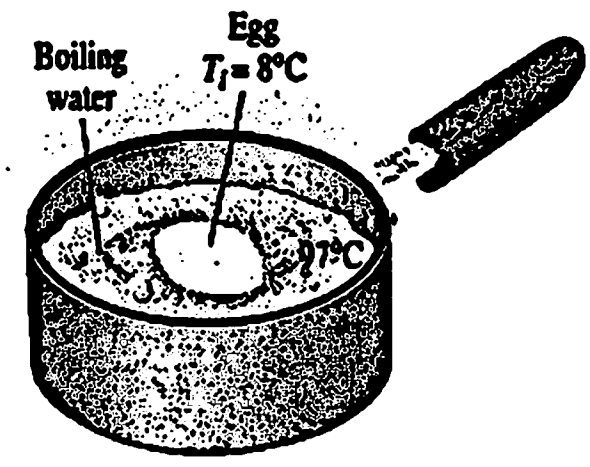


FIGURE P8–149

Step 1 of 4

Given data:

The diameter of egg, $d = 5.5\text{cm}$

The initial temperature of egg, $T_1 = 8°\text{C}$

$= 281\text{K}$

The temperature of boiled water, $T_2 = 97°\text{C}$

$= 370\text{K}$

The density of egg, $\rho = 1020\text{kg/m}^3$

The specific heat of egg, $c_p = 3.32\ \text{kJ/kg°C}$

Step 2 of 4

(a) The average temperature of egg, $T_{avg} = 70°\text{C}$

Let $\dot{Q}$ be the rate of Heat transfer

By using the relation

$m = \rho v$

$$m = \left(1020 \times \frac{\pi d^3}{6}\right)$$

$$m = \left\{1020 \times \frac{\pi \times \left(5.5 \times 10^{-2}\right)^3}{6}\right\}$$

$m = 0.08886\text{kg}$

Step 3 of 4

From the energy balance equation

Applying for the egg

$$Q = m \times c_p \times \left(T_{avg} - T_1\right)$$

$$= (0.08886 \times 3.32) \times (70 - 8)$$

$$= 18.29\text{kJ}$$

Therefore, the heat transfer, $\boxed{Q = 18.29\text{kJ}}$

Step 4 of 4

b) Let S_{gen} be the outlet temperature of the air

Now from the entropy balance equation

$$S_{gen} = (mS_2 - mS_1) - \frac{Q}{T_2}$$

$$S_{gen} = mc_p \ln\left(\frac{T_{avg}}{T_1}\right) - \frac{Q}{T_2}$$

$$S_{gen} = (0.08886 \times 3.32) \ln\left(\frac{70+273}{281}\right) - \left(\frac{18.29}{370}\right)$$

$$S_{gen} = 0.0588 - 0.04945$$

$$\boxed{S_{gen} = 0.00935\text{kJ/K}}$$

Therefore the entropy generation, $S_{gen} = 0.00935\text{kJ/K}$

Question: 8.150P

Long cylindrical steel rods (ρ = 7833 kg/m3 and cp = 0.465 kJ/kg·°C) of 10-cm diameter are heat treated by drawing them at a velocity of 3 m/min through a 7-m-long oven maintained at 900°C. If the rods enter the oven at 30°C and leave at 700°C, determine (*a*) the rate of heat transfer to the rods in the oven and (*b*) the rate of entropy generation associated with this heat transfer process.

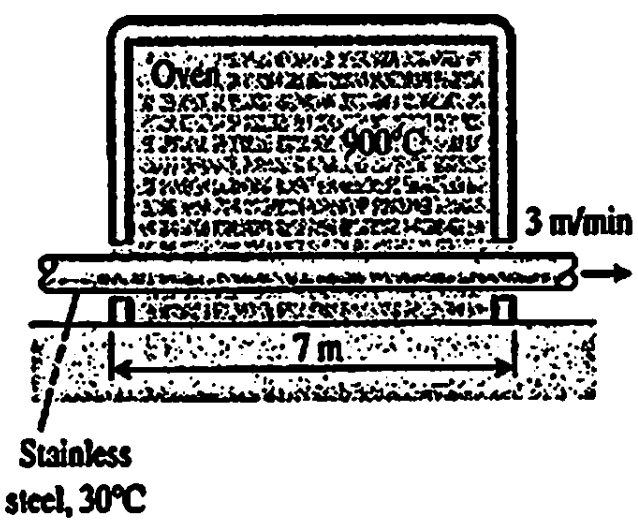


FIGURE P8–150

Step 1 of 4

(a)

Find the volume of the rod using the following equation:

$$V = L\left(\frac{\pi D^2}{4}\right)$$

Here, D is the diameter of the rod and L is the length of the rod.

The rod enters and exits the oven with the same velocity. A 3 m long section is heated in 1 minute.

Substitute $0.1\ \text{m}$ for D and $3\ \text{m}$ for L.

$$V = (3)\left(\frac{\pi(0.1)^2}{4}\right)$$
$$= 0.02356\,\text{m}^3$$

Find the mass of the steel rod (m) using the following equation

$$m = \rho V$$

Here density of the steel rod is ρ

Substitute $0.02356\ \text{m}^3$ for V and $7833\ \text{kg/m}^3$ for ρ.

$$m = \left(7833\ \text{kg/m}^3\right) \times \left(0.02356\ \text{m}^3\right)$$
$$= 184.5455\ \text{kg}$$

Step 2 of 4

Find the heat transfer to the rod in oven.

$$Q_{\text{in}} = \Delta U$$
$$= mc_p(T_2 - T_1)$$

..... (1)

Here, c_p specific heat of steel rod, T_1 is the initial temperature and T_2 is the leaving temperature of the oven.

Substitute $184.5455\ \text{kg}$ for m, $0.465\ \text{kJ/kg°C}$ for c_p, $700°\text{C}$ for T_2, and $30°\text{C}$ for T_1.

$$Q_{\text{in}} = (184.5455\ \text{kg})(0.465\ \text{kJ/kg°C})(700-30)°\text{C}$$
$$= 57495.144\ \text{kJ}$$

Find the rate of heat transfer to the rod in oven.

$$\dot{Q}_{\text{in}} = \frac{Q_{\text{in}}}{t}$$

Here, t is the time.

Substitute $57495.144\ \text{kJ}$ for Q_{in} and 1 min for t.

$$\dot{Q}_{\text{in}} = \frac{57495.144\,\text{kJ}}{1\,\text{min} \times 60\,\frac{\text{s}}{\text{min}}}$$
$$= 958.25\,\text{kJ/s}$$
$$= 958.25\,\text{kW}$$

Therefore, the rate of heat transferred to the rod is $\boxed{958.25\ \text{kW}}$.

Step 3 of 4

(b)

Find the change in net entropy transfer using the following equation:

$$(S_{\text{in}} - S_{\text{out}}) = \frac{Q_{\text{in}}}{T_b}$$

Here, T_b is the temperature of the oven.

Substitute $57495.144\ \text{kJ}$ for Q_{in} and $900°\text{C}$ for T_b.

$$(S_{\text{in}} - S_{\text{out}}) = \frac{57495.144\ \text{kJ}}{(900°\text{C}+273)\,\frac{\text{K}}{°\text{C}}}$$
$$= 49.0155\,\text{kJ/K}$$

Find the entropy change in the system using the following equation:

$$\Delta S_{\text{system}} = mc_p \ln\frac{T_2}{T_1}$$

Substitute $184.5455\ \text{kg}$ for m, $0.465\ \text{kJ/kg°C}$ for c_p, $700°\text{C}$ for T_2, and $30°\text{C}$ for T_1.

$$\Delta S_{\text{system}} = (184.5455)(0.465)\ln\left(\frac{700+273}{30+273}\right)$$
$$= 100.115\,\text{kJ/K}$$

Step 4 of 4

Find the generated entropy change using the entropy balance equation.

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$
$$S_{\text{gen}} = -(S_{\text{in}} - S_{\text{out}}) + \Delta S_{\text{system}}$$

Substitute $49.0155\,\text{kJ/K}$ for $(S_{\text{in}} - S_{\text{out}})$ and $100.115\,\text{kJ/K}$ for ΔS_{system}.

$$S_{\text{gen}} = -49.0155 + 100.115$$
$$= 51.0995\,\text{kJ/K}$$

Find the rate of entropy generation.

$$\dot{s}_{\text{gen}} = \frac{S_{\text{gen}}}{t}$$

Substitute $51.0995\,\text{kJ/K}$ for Q_{in} and 1 min for t.

$$\dot{s}_{\text{gen}} = \frac{51.0995\,\text{kJ/K}}{1\,\text{min} \times 60\,\frac{\text{s}}{\text{min}}}$$
$$= 0.8517\,\text{kW/K}$$

Therefore, the rate of entropy generation is $\boxed{0.8517\ \text{kW/K}}$.

Question: 8.151P

The inner and outer surfaces of a 3-m × 8-m brick wall of thickness 20 cm are maintained at temperatures of 20°C and 2°C, respectively. If the rate of heat transfer through the wall is 1550 W, determine the rate of entropy generation within the wall.

Step 1 of 2

Given data:

Temperature on leading side of wall is, $T_1 = 20°C = 293\ K$.

Temperature on lagging side of wall is, $T_2 = 2°C = 275\ K$.

Heat transfer through the wall is, $Q = 1550\ W$.

Step 2 of 2

The entropy balance is given by

$$S_{in} - S_{out} + S_{gen} = \Delta S$$

Let assume the process is steady. So, $\Delta S = 0$

$$S_{in} - S_{out} + S_{gen} = 0$$

$$\frac{Q_1}{T_1} - \frac{Q_2}{T_2} + S_{gen} = 0$$

As all the heat taken by wall is transferred to the other side, $Q_1 = Q_2 = Q$

$$\frac{Q}{T_1} - \frac{Q}{T_2} + S_{gen} = 0$$

$$S_{gen} = \frac{Q}{T_2} - \frac{Q}{T_1}$$

$$= \frac{1550}{275} - \frac{1550}{293}$$

$$= 5.636 - 5.29$$

$$= 0.346\ W/K$$

Therefore the entropy generated is $\boxed{s_{gen} = 0.346\ W/K}$.

Question: 8.152P

For heat transfer purposes, a standing man can be modeled as a 30-cm-diameter, 170-cm-long vertical cylinder with both the top and bottom surfaces insulated and with the side surface at an average temperature of 34°C. If the rate of heat loss from this man to the environment at 20°C is 336 W, determine the rate of entropy transfer from the body of this person accompanying heat transfer, in W/K.

Step 1 of 2

Given data

Rate of heat loss from man to environment $Q = 336\,\text{W}$

Surface temperature of the man $T = 34^\circ\text{C}$

$$= (34 + 273)\,\text{K}$$

$$T = 307\,\text{K}$$

Step 2 of 2

Rate of entropy transfer from the body of the man accompanying heat transfer is

$$\dot{S} = \frac{Q}{T}$$

$$= \frac{336}{307}$$

$$\dot{S} = 1.0944\,\text{W/K}$$

$$\boxed{\dot{S} = 1.0944\,\text{W/K}}$$

Question: 8.153P

A 1000-W iron is left on the ironing board with its base exposed to the air at 20°C. If the surface temperature is 400°C, determine the rate of entropy generation during this process in steady operation. How much of this entropy generation occurs within the iron?

Step 1 of 3

Given

Heating capacity of iron box $\dot{Q} = 1000\,\text{W}$

Temperature of air $T_{air} = 20°\text{C}$

$= (20+273)\,\text{K}$

$T_{air} = 293\,\text{K}$

Surface temperature of the iron box $T_{iron} = 400°\text{C}$

$= (400+273)\,\text{K}$

$T_{iron} = 673\,\text{K}$

Step 2 of 3

From the entropy balance on the system,

The rate of total entropy generation during this process in steady operation

$$\dot{S}_{gen,total} = \frac{\dot{Q}}{T_{air}}$$

$$= \frac{1000}{293}$$

$$= 3.413\,\text{W/K}$$

$$\boxed{\dot{S}_{gen,total} = 3.413\,\text{W/K}}$$

Step 3 of 3

Rate of entropy generation within the iron

$$\dot{S}_{gen,iron} = \frac{\dot{Q}}{T_{iron}}$$

$$= \frac{1000}{673}$$

$$= 1.486\,\text{W/K}$$

$$\boxed{\dot{S}_{gen,iron} = 1.486\,\text{W/K}}$$

Question: 8.154P

A frictionless piston-cylinder device contains saturated liquid water at 40-psia pressure. Now 600 Btu of heat is transferred to water from a source at 1000°F, and part of the liquid vaporizes at constant pressure. Determine the total entropy generated during this process, in Btu/R.

Step 1 of 5

Given data:

Initial pressure in the device, $P_1 = 40$ psia.

Temperature of source is, $T_{src} = 1000\ ^\circ F = 1459.67$ R.

Heat transfer to the device is, $Q = 600$ Btu.

Step 2 of 5

From the saturated water pressure table,

At pressure $P_1 = 40$ psia

the saturated temperature is

$T_{sat} = T_{dev} = 267.22\ ^\circ F$

$= 726.89$ R

Heat is transferred to the device from the source. So the heat transfer will take as positive in case of device and negative in case of source. Therefore

$Q_{dev} = 600$ Btu And

$Q_{src} = -600$ Btu

Step 3 of 5

The entropy change in the device is given by

$$\Delta S_{dev} = \frac{Q_{dev}}{T_{dev}}$$

$$= \frac{600}{726.89}$$

$$= 0.8254 \text{ Btu/R}$$

Step 4 of 5

The entropy change in the source is given by

$$\Delta S_{src} = \frac{Q_{src}}{T_{src}}$$

$$= \frac{-600}{1459.67}$$

$$= -0.411 \text{ Btu/R}$$

$T_{sat} = T_{dev} = 267.22\ ^\circ F$

$= 726.89$ R

Heat is transferred to the device from the source. So the heat transfer will take as positive in case of device and negative in case of source. Therefore

$Q_{dev} = 600$ Btu And

$Q_{src} = -600$ Btu

Step 3 of 5

The entropy change in the device is given by

$$\Delta S_{dev} = \frac{Q_{dev}}{T_{dev}}$$

$$= \frac{600}{726.89}$$

$$= 0.8254 \text{ Btu/R}$$

Question: 8.155P

Steam enters a diffuser at 20 psia and 240°F with a velocity of 900 ft/s and exits as saturated vapor at 240°F and 100 ft/s. The exit area of the diffuser is 1 ft2 . Determine (*a*) the mass flow rate of the steam and (*b*) the rate of entropy generation during this process. Assume an ambient temperature of 77°F.

Step 1 of 4

Obtain the values for the following:

The pressure at the entrance P_1 of the diffuser is $20\,\text{psia}$.

The temperature at the entrance T_1 of the diffuser is $240°\text{F}$.

From the properties table of super-heated water-temperature table the value for the above points is follows.

For the pressure P_1 and the temperature T_1;

The enthalpy at the entrance of the diffuser h_1 is $1162.3\,\text{Btu/lbm}$.

The entropy at the entrance of the diffuser s_1 is $1.7406\,\text{Btu/lbm}\cdot\text{R}$.

From the properties of saturated water tables,

At $T_2 = 240°\text{F}$;

The enthalpy at the exit h_2 of the diffuser is $1160.5\,\text{Btu/lbm}$.

The entropy at the exit of the diffuser s_2 is $1.7141\,\text{Btu/lbm}\cdot\text{R}$.

The specific volume v_2 at the exit is $16.316\,\text{ft}^3/\text{lbm}$.

Step 2 of 4

a)

Calculate the mass flow rate of the steam through the diffuser:

$$\dot{m} = \frac{A_2 V_2}{v_2}$$

Here, A_2 is the exit area, V_2 is the exit velocity of the steam and v_2 is the specific volume.

Substitute $1\,\text{ft}^2$ for A_2, $100\,\text{ft/sec}$ for V_2 and $16.316\,\text{ft}^3/\text{lbm}$ for v_2.

$$= \frac{1\times 100}{16.316}$$
$$= 6.1289\,\text{lbm/sec}$$

Therefore, mass flow rate $\dot{m}$ of the steam through the diffuser, $\boxed{6.1289\,\text{lbm/sec}}$

Step 3 of 4

b)

Calculate the heat transfer at the outlet:

From the entropy balance equation on an extended system,

$$S_{in} - S_{out} + S_{gen} = \Delta S_{system}$$
$$= 0$$

Hence the above equation becomes as follows.

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_\infty} + \dot{S}_{gen} = 0$$

From the energy balance equation for the steady flow,

$$E_{in} - E_{out} = \Delta E_{system} = 0$$

Therefore, $E_{in} = E_{out}$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) - \dot{Q}_{out} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$= 6.1289\,\text{lbm/sec}$

Therefore, mass flow rate $\dot{m}$ of the steam through the diffuser, $\boxed{6.1289\,\text{lbm/sec}}$

Step 3 of 4

b)

Calculate the heat transfer at the outlet:

From the entropy balance equation on an extended system,

$$S_{in} - S_{out} + S_{gen} = \Delta S_{system}$$
$$= 0$$

Hence the above equation becomes as follows.

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_\infty} + \dot{S}_{gen} = 0$$

From the energy balance equation for the steady flow,

$$E_{in} - E_{out} = \Delta E_{system} = 0$$

Therefore, $E_{in} = E_{out}$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) - \dot{Q}_{out} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$= (6.1289)\left(\text{lbm/sec}\right)\left[(1162.3 - 1160.5)\left(\text{Btu/lbm}\right) + \left(\frac{900^2}{2} - \frac{100^2}{2}\right)\left(\frac{1}{25037}\right)\left(\text{Btu/lbm}\right)\right]$$
$$= 6.1289\left(\text{lbm/sec}\right)\left[(1.8 + 15.976)\left(\text{Btu/lbm}\right)\right]$$
$$= 108.94\,\text{Btu/sec}$$

Calculate the rate of entropy generation during the process:

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_\infty} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_\infty}$$

Here, $\dot{S}_{gen}$ is the rate of entropy generation is the mass flow rate $\dot{m}$, s_2 is the entropy at the inlet, s_1 is the entropy at the outlet, $\dot{Q}_{out}$ is the heat transfer at the outlet and T_∞ is the total temperature.

Substitute $6.1289\,\text{lbm/sec}$ for $\dot{m}$, $1.7406\,\text{Btu/lbm}\cdot\text{R}$ for s_1, $1.7141\,\text{Btu/lbm}\cdot\text{R}$ for s_2 and $108.94\,\text{Btu/sec}$ for $\dot{Q}_{out}$.

$$= 6.1289(1.7141 - 1.7406) + \frac{108.95}{77 + 460}$$
$$= 0.0404\,\text{Btu/sec}\cdot\text{R}$$

Therefore, the rate of entropy generation $\dot{S}_{gen}$ during the process is $\boxed{0.0404\,\text{Btu/sec}\cdot\text{R}}$.

$$= (6.1289)\left(\text{lbm/sec}\right)\left[(1162.3 - 1160.5)\left(\text{Btu/lbm}\right) + \left(\frac{900^2}{2} - \frac{100^2}{2}\right)\left(\frac{1}{25037}\right)\left(\text{Btu/lbm}\right)\right]$$
$$= 6.1289\left(\text{lbm/sec}\right)\left[(1.8 + 15.976)\left(\text{Btu/lbm}\right)\right]$$

Question: 8.156P

Steam enters an adiabatic nozzle at 2.5 MPa and 450°C with a velocity of 55 m/s and exits at 1 MPa and 390 m/s. If the nozzle has an inlet area of 6 cm2 , determine (*a*) the exit temperature and (*b*) the rate of entropy generation for this process.

Step 1 of 5

Given data:

Pressure at nozzle inlet is, $P_1 = 2.5\ \text{MPa}$

$= 2500\ \text{kPa}$

Temperature at nozzle inlet is, $T_1 = 450°\text{C}$

$= 723\ \text{K}$

Velocity at nozzle inlet is, $C_1 = 55\ \text{m/s}$

Pressure at nozzle exit is, $P_2 = 1\ \text{MPa}$

$= 1000\ \text{kPa}$

Velocity at nozzle exit is, $C_2 = 390\ \text{m/s}$

Area at nozzle inlet is, $A_1 = 6\ \text{cm}^2$

Question:

Steam enters an adiabatic nozzle at 2.5 MPa and 450°C with a velocity of 55 m/s and exits at 1 MPa and 390 m/s. If the nozzle has an inlet area of 6 cm2 , determine (*a*) the exit temperature and (*b*) the rate of entropy generation for this process.

Step 1 of 5

Given data:

Pressure at nozzle inlet is, $P_1 = 2.5\ \text{MPa}$

$= 2500\ \text{kPa}$

Temperature at nozzle inlet is, $T_1 = 450°\text{C}$

$= 723\ \text{K}$

Velocity at nozzle inlet is, $C_1 = 55\ \text{m/s}$

Pressure at nozzle exit is, $P_2 = 1\ \text{MPa}$

$= 1000\ \text{kPa}$

Velocity at nozzle exit is, $C_2 = 390\ \text{m/s}$

Area at nozzle inlet is, $A_1 = 6\ \text{cm}^2$

For converting m^2/s^2 to kJ/kg we have to divide by 1000. So,

$$h_2 = h_1 + \left(\frac{C_1^2 - C_2^2}{2000}\right)$$

$$h_2 = 3351.6 + \left(\frac{55^2 - 390^2}{2000}\right)$$

$$h_2 = 3351.6 + \left(\frac{3025 - 152100}{2000}\right)$$

$$h_2 = 3351.6 + \left(\frac{-149075}{2000}\right)$$

$$h_2 = 3351.6 - 74.538$$

$$h_2 = 3277.062\ \text{kJ/kg}$$

Step 3 of 5

a) From properties of superheated steam

At pressure $P_2 = 1\ \text{MPa}$

Final temperature is interpolated as

$$\left(\frac{T_2 - 400}{500 - 400}\right) = \left(\frac{3277.062 - 3264.5}{3479.1 - 3264.5}\right)$$

$$\left(\frac{T_2 - 400}{100}\right) = \left(\frac{12.562}{214.6}\right)$$

$$\left(\frac{T_2 - 400}{100}\right) = 0.0585$$

$$T_2 = (0.0585 \times 100) + 400$$

$$T_2 = 5.85 + 400$$

$$\boxed{T_2 = 405.85°\text{C}}$$

Therefore, the final temperature, $T_2 = 405.85°\text{C}$

Step 4 of 5

b) From properties of superheated steam

At pressure $P_2 = 1\ \text{MPa}$

Final entropy is interpolated as

$$\left(\frac{s_2 - 7.4670}{7.7642 - 7.4670}\right) = \left(\frac{3277.062 - 3264.5}{3479.1 - 3264.5}\right)$$

$$\left(\frac{s_2 - 7.4670}{0.2972}\right) = \left(\frac{12.562}{214.6}\right)$$

$$\left(\frac{s_2 - 7.4670}{0.2972}\right) = 0.0585$$

$$s_2 = (0.0585 \times 0.2972) + 7.4670$$

$$s_2 = 0.0174 + 7.4670$$

$$s_2 = 7.4844\ \text{kJ/kg} \cdot \text{K}$$

$$\left(\frac{T_2 - 400}{100}\right) = 0.0585$$

$$T_2 = (0.0585 \times 100) + 400$$

$$T_2 = 5.85 + 400$$

$$\boxed{T_2 = 405.85°\text{C}}$$

Therefore, the final temperature, $T_2 = 405.85°\text{C}$

Step 4 of 5

b) From properties of superheated steam

At pressure $P_2 = 1\ \text{MPa}$

Final entropy is interpolated as

$$\left(\frac{s_2 - 7.4670}{7.7642 - 7.4670}\right) = \left(\frac{3277.062 - 3264.5}{3479.1 - 3264.5}\right)$$

$$\left(\frac{s_2 - 7.4670}{0.2972}\right) = \left(\frac{12.562}{214.6}\right)$$

$$\left(\frac{s_2 - 7.4670}{0.2972}\right) = 0.0585$$

$$s_2 = (0.0585 \times 0.2972) + 7.4670$$

$$s_2 = 0.0174 + 7.4670$$

$$s_2 = 7.4844\ \text{kJ/kg} \cdot \text{K}$$

Question: 8.157P

Steam expands in a turbine steadily at a rate of 40,000 kg/h, entering at 8 MPa and 500°C and leaving at 40 kPa as saturated vapor. If the power generated by the turbine is 8.2 MW, determine the rate of entropy generation for this process. Assume the surrounding medium is at 25°C.

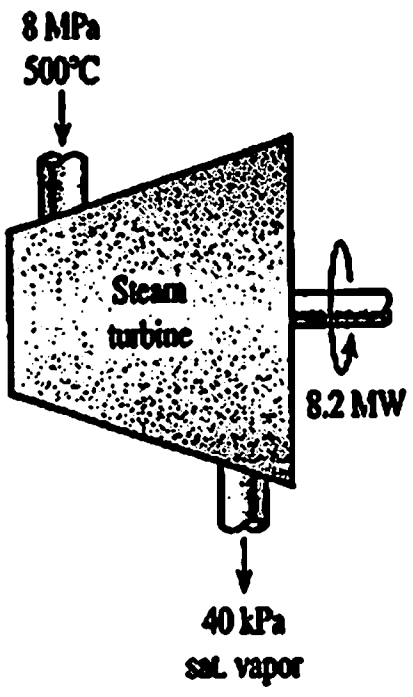


FIGURE P8–157

Step 1 of 5

Given data:

Mass flow rate is

$m = 40{,}000\ kg/h$

$= 40{,}000\ kg/h \times 3600 h/s$

$= 11.11\ kg/s$

Inlet pressure to the turbine is, $P_1 = 8\ MPa = 8000\ kPa$.

Inlet temperature to the turbine is, $T_1 = 500\ °C = 773\ K$.

Exit pressure to the turbine is, $P_2 = 40\ kPa$.

Surrounding temperature is, $T_0 = 25\ °C = 298\ K$.

Power produced by the turbine is, $W_{out} = 8.2\ MW = 8200\ kW$.

Step 2 of 5

From the superheated water

At pressure $P_1 = 8\ MPa$ and temperature $T_1 = 500\ °C$,

the enthalpy and entropy respectively are

$h_1 = 3399.5\ kJ/kg$

$s_1 = 6.7266\ kJ/kg \cdot K$

From the superheated water

At pressure $P_2 = 40\ kPa$ and saturated vapor condition

the enthalpy and entropy respectively are

$h_2 = h_g = 2636.1\ kJ/kg$

$s_2 = s_g = 7.6691\ kJ/kg \cdot K$

Step 3 of 5

The energy balance is given by

$E_{in} - E_{out} = \Delta E_{sys}$

Let assume the process is steady. So, $\Delta E_{sys} = 0$

$E_{in} - E_{out} = 0$

$E_{in} = E_{out}$

$m \times h_1 = Q_{out} + W_{out} + m \times h_2$

$Q_{out} = m \times h_1 - W_{out} - m \times h_2$

Step 4 of 5

The rate of heat transfer is given by

$Q_{out} = m(h_1 - h_2) - W_{out}$

$= 11.11(3399.5 - 2636.1) - 8200$

$= 11.11(763.4) - 8200$

$= 8481.374 - 8200$

$= 281.374\ kW$

Step 5 of 5

The entropy balance is given by

$S_{in} - S_{out} + S_{gen} = \Delta S_{sys}$

Let assume the process is steady. So, $\Delta S_{sys} = 0$

$S_{in} - S_{out} + S_{gen} = 0$

$m \times s_1 - m \times s_2 - \frac{Q_{out}}{T_0} + S_{gen} = 0$

$S_{gen} = m \times s_2 - m \times s_1 + \frac{Q_{out}}{T_0}$

$= m \times (s_2 - s_1) + \frac{Q_{out}}{T_0}$

$= 11.11 \times (7.6691 - 6.7266) + \frac{281.374}{298}$

$= 11.11 \times 0.9425 + 0.9442$

$= 10.4712 + 0.9442$

$= 11.4154 kW/K$

Therefore the entropy generated is $\boxed{s_{gen} = 11.4154\ kW/K}$.

Question: 8.158P

Air enters a compressor at ambient conditions of 15 psia and 60°F with a low velocity and exits at 150 psia, 620°F, and 350 ft/s. The compressor is cooled by the ambient air at 60°F at a rate of 1500 Btu/min. The power input to the compressor is 400 hp. Determine (*a*) the mass flow rate of air and (*b*) the rate of entropy generation.

Step 1 of 7

Given data:

The ambient conditions of air at the entrance,

$p_1 = 15\,\text{psia}$

$T_1 = 60°\text{F}$

The ambient conditions of air at the exit,

$p_2 = 150\,\text{psia}$

$T_2 = 620°\text{F}$

Velocity of ambient air at exit, $V_2 = 350\,\text{ft/s}$

The ambient temperature of air, when the compressor is cooled, $T = 60°\text{F}$

The rate of ambient air, $\dot{Q}_{out} = 1500\,\text{Btu/min}$

Step 2 of 7

(a)Calculation of mass flow rate of air through the compressor:

The mass flow rate of air can be calculated by using energy balance equation as follows:

The energy balance for the given steady flow system can be expressed as,

$\dot{E}_{in} = \dot{E}_{out}$

Since, potential energy changes is zero,

$$\dot{W}_{in} + \dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{Q}_{out} + \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) \qquad \text{..... (1)}$$

Step 3 of 7

From ideal gas properties of air tables,

At $T_1 = 60°\text{F} = 520\,\text{R}$,

The specific enthalpy is, $h_1 = 124.27\,\text{Btu/lbm}$

The specific entropy is, $s_1^0 = 0.59173\,\text{Btu/lbm}\cdot\text{R}$

At $T_2 = 620°\text{F} = 1080\,\text{R}$,

The specific enthalpy is, $h_2 = 260.97\,\text{Btu/lbm}$

The specific entropy is, $s_2^0 = 0.76964\,\text{Btu/lbm}\cdot\text{R}$

Step 4 of 7

By substituting these known values in equation (1), we get

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$(400\,\text{hp})\left(\frac{0.7068\,\text{Btu/s}}{1\,\text{hp}}\right) - \left(\frac{1500\,\text{Btu}}{60\,\text{s}}\right) = \dot{m}\left[\begin{array}{l}(260.97 - 124.27) \\ \quad + \dfrac{(350\,\text{ft/s})^2}{2}\left(\dfrac{1\,\text{Btu/lbm}}{25{,}037\,\text{ft}^2/\text{s}^2}\right)\end{array}\right]$$

$$\dot{m} = \frac{257.72}{136.7}$$

$$= 1.8851\,\text{lbm/s}$$

Therefore, the mass flow rate of air through the compressor is, $\boxed{\dot{m} = 1.885\,\text{lbm/s}}$

Step 5 of 7

(b)Calculation of total rate of entropy generation:

Since, there is no heat transfer to the surroundings.

Therefore, the entropy balance for the given system is,

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_{b,surr}} \qquad \text{..... (2)}$$

Where, $\dot{m}(s_2 - s_1) = \dot{m}\left(s_2^0 - s_1^0 - R\ln\frac{p_2}{p_1}\right)$

Step 6 of 7

The gas constant of air, $R = 0.06855\,\text{Btu/lbm}\cdot\text{R}$

By substituting the known values in the above equation, we get

$$\dot{m}(s_2 - s_1) = 1.852\left(0.76964 - 0.59173 - \left(0.06855\ln\frac{150}{15}\right)\right)$$

$$= 1.852 \times 0.02006$$

$$= 0.0371\,\text{Btu/s}\cdot\text{R}$$

Step 7 of 7

By substituting the above value in equation (2), we get

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_{b,surr}}$$

$$= 0.0372 + \frac{(1500/60)}{520}$$

$$= 0.0372 + 0.048$$

$$= 0.08527\,\text{Btu/s}\cdot\text{R}$$

Therefore, the total rate of entropy generation, $\boxed{\dot{S}_{gen} = 0.08527\,\text{Btu/s}\cdot\text{R}}$

Question: 8.159P

A hot-water stream at 70°C enters an adiabatic mixing chamber with a mass flow rate of 3.6 kg/s, where it is mixed with a stream of cold water at 20°C. If the mixture leaves the chamber at 42°C, determine (*a*) the mass flow rate of the cold water and (*b*) the rate of entropy generation during this adiabatic mixing process. Assume all the streams are at a pressure of 200 kPa.

Step 1 of 5

Given data:

The temperature of hot water stream, $T_1 = 70°C$

The temperature of cold water, $T_2 = 20°C$

Temperature of mixture, $T_3 = 42°C$

Mass flow rate of hot water, $\dot{m}_1 = 3.6\,kg/s$

The pressure for all streams, $p = 200kPa$

Step 2 of 5

(a)Calculation of mass flow rate of cold stream:

The mass flow rate of cold stream can be calculated by using mass and energy balance equations as follow:

The mass balance for the given steady flow system can be expressed as,

$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$ ..... (1)

The energy balance for the given steady flow system can be expressed as,

$\dot{E}_{in} = \dot{E}_{out}$

$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$ ..... (2)

From equations (1), (2) we get,

$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$

$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_1 h_3 + \dot{m}_2 h_3$

$\dot{m}_2 (h_2 - h_3) = \dot{m}_1 (h_3 - h_1)$

$\dot{m}_2 = \dfrac{(h_3 - h_1)}{(h_2 - h_3)} \dot{m}_1$ ..... (3)

Step 3 of 5

From saturated water pressure tables at $p = 200kPa$, the temperature is, $T_{sat} = 120.21°C$

Since, the given temperatures of the three streams are less than the saturated temperature; the three streams can be approximated to saturated liquid state.

From the properties of saturated water temperature tables,

At $T_1 = 70°C$

The specific enthalpy of the saturated liquid, $h_1 = h_f = 293.07\,kJ/kg$

The specific entropy of the saturated liquid, $s_1 = s_f = 0.955kJ/kg \cdot K$

At $T_2 = 20°C$

The specific enthalpy of the saturated liquid, $h_2 = h_{f2} = 83.91kJ/kg$

The specific entropy of the saturated liquid, $s_2 = s_{f2} = 0.2965kJ/kg \cdot K$

At $T_3 = 42°C$

The specific enthalpy of the saturated liquid, $h_3 = h_{f3} = 175.90kJ/kg$

The specific entropy of the saturated liquid, $s_3 = s_{f3} = 0.599\,kJ/kg \cdot K$

Step 4 of 5

By substituting these known values in equation (3), we get

$$\dot{m}_2 = \frac{(h_3 - h_1)}{(h_2 - h_3)} \dot{m}_1$$

$$= \left(\frac{175.90 - 293.07}{83.91 - 175.90}\right) \times 3.6$$

$$= \left(\frac{-117.17}{-91.99}\right) \times 3.6$$

$$= 1.2737 \times 3.6$$

$$= 4.5854\ kg/s$$

And,

$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$

$\dot{m}_3 = 3.6 + 4.586$

$\dot{m}_3 = 8.186 kg/s$

Therefore, the mass flow rate of cold water is, $\boxed{\dot{m}_2 = 4.5854\ kg/s}$

Step 5 of 5

(b)Calculation of total rate of entropy generation:

Since the mixing chamber is adiabatic, there is no heat transfer to the surroundings.

Therefore, the entropy balance for the given system is,

$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{gen} = 0$

$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$

By substituting the known values in the above equation, we get

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

$$= (8.186 \times 0.599) - (3.6 \times 0.955) - (4.586 \times 0.2965)$$

$$= 0.105665 kW/K$$

Therefore, the total rate of entropy generation, $\boxed{\dot{S}_{gen} = 0.105665 kW/K}$

Question: 8.160P

Liquid water at 200 kPa and 20°C is heated in a chamber by mixing it with superheated steam at 200 kPa and 150°C. Liquid water enters the mixing chamber at a rate of 2.5 kg/s, and the chamber is estimated to lose heat to the surrounding air at 25°C at a rate of 1200 kJ/min. If the mixture leaves the mixing chamber at 200 kPa and 60°C, determine (*a*) the mass flow rate of the superheated steam and (*b*) the rate of entropy generation during this mixing process.

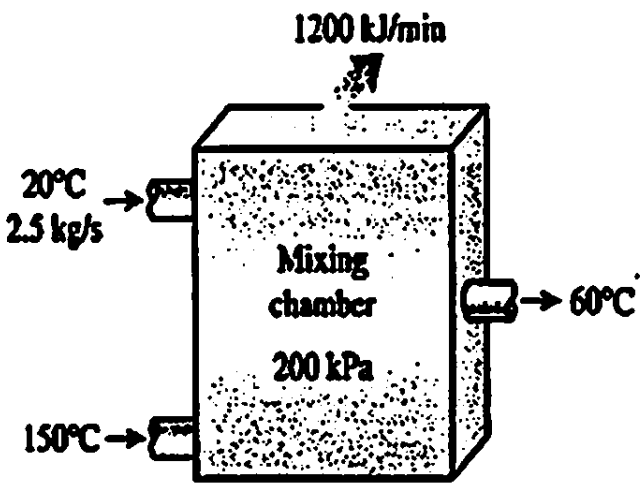


FIGURE P8–160

Step 1 of 6

Given data:

The pressure and temperature of liquid water,

$p_1 = 200\text{kPa}$

$T_1 = 20°\text{C}$

The pressure and temperature of super heated steam,

$p_2 = 200\text{kPa}$

$T_2 = 150°\text{C}$

The pressure and Temperature of mixture after leaving the mixing chamber,

$p_3 = 200\text{kPa}$

$T_3 = 60°\text{C}$

Mass flow rate of liquid water, $\dot{m}_1 = 2.5\text{kg/s}$

The rate of heat transfer, $\dot{Q} = 1200\text{kJ/min}$

The surrounding temperature, $T_{b,surr} = 25°\text{C}$

Step 2 of 6

(a)Calculation of mass flow rate steam:

The mass flow rate of steam can be calculated by using mass and energy balance equations as follow:

The mass balance for the given steady flow system can be expressed as,

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 \qquad \text{..... (1)}$$

The energy balance for the given steady flow system can be expressed as,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{out} + \dot{m}_3 h_3 \qquad \text{..... (2)}$$

From equations (1), (2) we get,

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{out} + (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{Q}_{out} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{Q}_{out} = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$$

$$\dot{m}_2 (h_2 - h_3) = \dot{Q}_{out} - \dot{m}_1 (h_1 - h_3)$$

$$\dot{m}_2 = \frac{\dot{Q}_{out} - \dot{m}_1 (h_1 - h_3)}{(h_2 - h_3)} \qquad \text{..... (3)}$$

Step 3 of 6

From saturated water pressure tables at $p = 200\text{kPa}$, the temperature is, $T_{sat} = 120.21°\text{C}$

Since, the liquid water and exit temperatures of the streams are less than the saturated temperature; this both streams can be approximated to saturated liquid state.

From the properties of saturated water temperature tables,

At $T_1 = 20°\text{C}$

The specific enthalpy of the saturated liquid, $h_1 = h_f = 83.91\text{kJ/kg}$

The specific entropy of the saturated liquid, $s_1 = s_f = 0.2965\text{kJ/kg}\cdot\text{K}$

At $T_3 = 60°\text{C}$

The specific enthalpy of the saturated liquid, $h_3 = h_{f3} = 251.18\text{kJ/kg}$

The specific entropy of the saturated liquid, $s_3 = s_{f3} = 0.8313\text{kJ/kg}\cdot\text{K}$

From super heated water tables at $p_2 = 200\text{kPa}$ and $T_2 = 150°\text{C}$ the properties are

The specific enthalpy of the super heated liquid, $h_2 = 2769.1\text{kJ/kg}$

The specific entropy of the super heated liquid, $s_2 = 7.2810\text{kJ/kg}\cdot\text{K}$

Step 4 of 6

By substituting these known values in equation (3), we get

$$\dot{m}_2 = \frac{\dot{Q}_{out} - \dot{m}_1 (h_1 - h_3)}{(h_2 - h_3)}$$

$$\dot{m}_2 = \frac{(1200/60\text{kJ/s}) - (2.5\text{kg/s})(83.91 - 251.18)\text{kJ/kg}}{(2769.1 - 251.18)\text{kJ/kg}}$$

$$= \frac{438.175}{2517.92}$$

$$= 0.174\text{kg/s}$$

And,

$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$

$\dot{m}_3 = 2.5 + 0.174$

$\dot{m}_3 = 2.674\text{kg/s}$

Therefore, the mass flow rate of steam is, $\boxed{\dot{m}_2 = 0.174\text{ kg/s}}$

Step 5 of 6

(b)Calculation of total rate of entropy generation:

Entropy generation can be calculated by using entropy balance,

Therefore, the entropy balance for the given system is,

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{out}}{T_{b,surr}}$$

Step 6 of 6

By substituting the known values in the above equation, we get

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{out}}{T_{b,surr}}$$

$$= (2.674 \times 0.8313) - (2.5 \times 0.2965) - (0.174 \times 7.2810) + \left(\frac{1200/60}{298}\right)$$

$$= 0.283\text{kW/K}$$

Therefore, the total rate of entropy generation, $\boxed{\dot{S}_{gen} = 0.283\text{kW/K}}$

Question:

A 0.18-m3 rigid tank is filled with saturated liquid water at 120°C. A valve at the bottom of the tank is now opened, and one-half of the total mass is withdrawn from the tank in the liquid form. Heat is transferred to water from asource at 230°C so that the temperature in the tank remains constant. Determine (*a*) the amount of heat transfer and (*b*) the total entropy generation for this process.

Step 1 of 8

Since, the mass inside the rigid tank is crossing the boundaries of the tank. The system is considered as open system.

Apply mass balance equation to the rigid tank.

$$m_i + m_1 = m_e + m_2 \quad \text{..... (1)}$$

Here, m_i is the mass of the water entering the tank, m_1 is the initial mass in the tank, m_e is the mass withdrawn from the tank, and m_2 is the final mass in the tank.

Step 2 of 8

Calculate the initial mass in the tank using the equation,

$$m_1 = \frac{V}{v_1}$$

Here, V is volume of the tank, and v_1 is specific volume of water.

Use saturated water - Temperature tables to determine the specific volume of water at saturated liquid state at a temperature of $120°C$.

$$v_1 = 0.001060 \ \text{m}^3/\text{kg}$$

Substitute $0.001060 \ \text{m}^3/\text{kg}$ for v_1 , and $0.18 \ \text{m}^3$ for V.

$$m_1 = \frac{0.18}{0.001060}$$
$$= 169.81 \ \text{kg}$$

Step 3 of 8

Calculate the mass of water withdrawn from the tank using the equation,

$$m_e = \frac{m_1}{2}$$

Substitute 169.81 kg for m_1 .

$$m_e = \frac{169.81}{2}$$
$$= 84.91 \ \text{kg}$$

Calculate the final mass of water in the tank by substituting 0 for m_i , 169.81 for m_1 , and 84.91 kg for m_e in equation (1).

$$m_i + m_1 = m_e + m_2$$
$$0 + 169.81 = 84.91 + m_2$$
$$m_2 = 84.91 \ \text{kg}$$

Step 4 of 8

Calculate the final specific volume of water in the tank using the equation,

$$v_2 = \frac{V}{m_2}$$

Substitute $0.18 \ \text{m}^3$ for V, and 84.91 kg for m_2 .

$$v_2 = \frac{0.18}{84.91}$$
$$= 2.1199 \times 10^{-3} \ \text{m}^3/\text{kg}$$

Step 5 of 8

Calculate the dryness fraction of water in the tank at the final state using the equation,

$$v_2 = v_f + x(v_g - v_f)$$

Here, x is the dryness fraction, v_f is the specific volume of water at saturated liquid state, and v_g is the specific volume of water at saturated vapor state.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Specific volume at saturated liquid state, $v_f = 0.001060 \ \text{m}^3/\text{kg}$.

Specific volume at saturated vapor state, $v_g = 0.89133 \ \text{m}^3/\text{kg}$.

Substitute $2.1199 \times 10^{-3} \ \text{m}^3/\text{kg}$ for v_2 , $0.001060 \ \text{m}^3/\text{kg}$ for v_f , and $0.89133 \ \text{m}^3/\text{kg}$ for v_g .

$$2.1199 \times 10^{-3} = 0.001060 + x(0.89133 - 0.001060)$$
$$0.89027x = 1.0599 \times 10^{-3}$$
$$x = 0.001191$$

Step 6 of 8

Calculate the internal energy of water in tank at the final stage using the equation,

$$u_2 = u_f + x u_{fg}$$

Here, u_f is the internal energy of water at saturated liquid state, and u_{fg} is the internal energy for evaporate of water.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Internal energy at saturated liquid state, $u_f = 503.60 \ \text{kJ/kg}$.

evaporate of water.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Internal energy at saturated liquid state, $u_f = 503.60 \text{ kJ/kg}$.

Internal energy for evaporate of water, $u_{fg} = 2025.3 \text{ kJ/kg}$.

Substitute 0.001191 for x, 503.60 kJ/kg for u_f , and 2025.3 kJ/kg for u_{fg} .

$$u_2 = 503.60 + 0.001191 \times 2025.3$$
$$= 506.01 \text{ kJ/kg}$$

(a)

Calculate the heat supplied to the tank by applying energy balance equation for the tank.

$$m_i h_i + m_1 u_1 + Q_{in} = m_e h_e + m_2 u_2$$

Here, Q_{in} is the heat supplied to the tank.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Enthalpy at saturated liquid state, $h_e = 503.81 \text{ kJ/kg}$.

Internal energy at saturated liquid state, $u_1 = 503.60 \text{ kJ/kg}$.

Substitute 0 for m_i , 169.81 kg for m_1 , 503.60 kJ/kg for u_1 , 84.91 kg for m_e , 503.81 kJ/kg for h_e , 84.91 kg for m_2 , and 506.01 kJ/kg for u_2 .

$$0 + 169.81 \times 503.60 + Q_{in} = 84.91 \times 503.81 + 84.91 \times 506.01$$
$$Q_{in} = 227.50 \text{ kJ}$$

Therefore, the heat supplied to the tank is $\boxed{227.50 \text{ kJ}}$.

Step 7 of 8

(b)

Calculate the entropy of water in tank at the final stage using the equation,

$$s_2 = s_f + x s_{fg}$$

Here, s_f is the entropy of water at saturated liquid state, and s_{fg} is the entropy for evaporate of water.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Entropy at saturated liquid state, $s_f = 1.5279 \text{ kJ/kg·K}$.

Entropy for evaporate of water, $s_{fg} = 5.6013 \text{ kJ/kg·K}$.

Substitute 0.001191 for x, 1.5279 kJ/kg·K for s_f , and 5.6013 kJ/kg·K for s_{fg} .

$$s_2 = 1.5279 + 0.001191 \times 5.6013$$
$$= 1.5346 \text{ kJ/kg·K}$$

Step 8 of 8

Calculate the entropy generated during the process by applying entropy balance equation.

$$m_i s_i + m_1 s_1 + S_{gen} + \frac{Q_{in}}{T_{source}} = m_e s_e + m_2 s_2$$

Here, T_{source} is the temperature of the source, and S_{gen} is the entropy generated during the process.

Use saturated water - Temperature tables to determine the corresponding values of water at a temperature of $120°C$.

Entropy at saturated liquid state, $s_1 = s_e = 1.5279 \text{ kJ/kg·K}$.

Substitute 0 for m_i , 169.81 kg for m_1 , 1.5279 kJ/kg·K for s_1 , 227.50 kJ for Q_{in} , $(230+273) \text{ K}$ for T_{source} , 84.91 kg for m_e , 1.5279 kJ/kg·K for s_e , 84.91 kg for m_2 , and 1.5346 kJ/kg·K for s_2 .

$$0 + 169.81 \times 1.5279 + S_{gen} + \frac{227.50}{230+273} = 84.91 \times 1.5279 + 84.91 \times 1.5346$$
$$S_{gen} = 0.1319 \text{ kJ/K}$$

Therefore, the entropy generated during the process is $\boxed{0.1319 \text{ kJ/K}}$.

Question: 8.162P

An iron block of unknown mass at 185°F is dropped into an insulated tank that contains 0.8 ft3 of water at 70°F. At the same time, a paddle wheel driven by a 200-W motor is activated to stir the water. Thermal equilibrium is established after 10 min with a final temperature of 75°F. Determine (*a*) the mass of the iron block and (*b*) the entropy generated during this process.

Step 1 of 4

(a) Calculation of mass of the iron block:

The mass of iron block can be calculated by using energy balance equation as follows: The energy balance

$$\dot{E}_{in} - \dot{E}_{out} = \Delta E_{system}$$

for the given steady flow system can be expressed as, $W_{pw,in} = \Delta U$

$$W_{pw,in} = \Delta U_{iron} + \Delta U_{water}$$

$W_{pw,in} = \left[mc(T_2 - T_1)\right]_{iron} + \left[mc(T_2 - T_1)\right]_{water}$ ..... (1) Since, density of water at room temperature,

$$m_{water} = \rho V$$

$\rho = 62.1\,\text{lbm/ft}^3$ Calculate the mass of water, $= 62.1 \times 0.8$

$$= 49.68\,\text{lbm}$$

Step 2 of 4

Calculate the work done driven by the paddle wheel after 10 min,

$$W_{pw} = \dot{W}_{pw}\Delta t$$

$$= (0.2\,\text{kJ/s})\left(\frac{1\text{Btu}}{1.055\text{kJ}}\right)(10 \times 60\text{s})$$

$$= 113.7\,\text{Btu}$$

And, from properties of liquid and solids table,

Specific heat of iron is, $c_{pi} = 0.107\,\text{Btu/lbm °F}$

Specific heat of water is, $c_{pw} = 1.00\,\text{Btu/lbm °F}$

Substitute the known values in equation (1),

$$113.7 = \left[m_{iron} \times 0.107 \times (75 - 185)\right] + \left[49.7 \times 1.00 \times (75 - 70)\right]$$

$$m_{iron} = \frac{113.7 - 248.5}{-11.77}$$

$$m_{iron} = 11.52\,\text{lbm}$$

Therefore, the mass of iron block is, $\boxed{11.52\,\text{lbm}}$

Step 3 of 4

(b) Calculation of total rate of entropy generation:

Entropy generation can be calculated by using entropy balance,

Therefore, the entropy balance for the given system is,

$$S_{gen,total} = \Delta S_{system}$$

$S_{gen,total} = \Delta S_{iron} + \Delta S_{water}$ ..... (2)

Where,

$$\Delta S_{iron} = mc_{pi} \ln\left(\frac{T_2}{T_1}\right)$$

$$= 11.4 \times 0.107 \times \ln\left(\frac{535}{645}\right)$$

$$= -0.2208\,\text{Btu/R}$$

$$\Delta S_{water} = m_w c_{pw} \ln\left(\frac{T_2}{T_1}\right)$$

$$= 49.6 \times 1.0 \times \ln\left(\frac{535}{530}\right)$$

$$= 0.465\,\text{Btu/R}$$

Step 4 of 4

Substitute $-0.2208\,\text{Btu/R}$ for ΔS_{iron} and $0.465\,\text{Btu/R}$ for ΔS_{water} in equation (2)

$$S_{gen,total} = \Delta S_{iron} + \Delta S_{water}$$

$$= -0.2208 + 0.465$$

$$= 0.244\,\text{Btu/R}$$

Therefore, the total rate of entropy generation, $\boxed{0.244\,\text{Btu/R}}$

Question: 8.163P

A rigid tank contains 7.5 kg of saturated water mixture at 400 kPa. A valve at the bottom of the tank is now opened, and liquid is withdrawn from the tank. Heat is transferred to the steam such that the pressure inside the tank remains constant. The valve is closed when no liquid is left in the lank. If it is estimated that a total of 5 kJ of heat is transferred to the tank, determine (*a*) the quality of steam in the tank at the initial state, (*b*) the amount of mass that has escaped, and (*c*) the entropy generation during this process if heat is supplied to the tank from a source at 500T.

Step 1 of 5

Mass of the saturated water in the tank, $m_1 = 7.5\text{kg}$

Pressure of the saturated water, $p = 400\text{kPa}$

Total heat transferred to the tank, $Q = 5\text{kJ}$

Step 2 of 5

For the saturated vapor at pressure $P = 400\text{kPa}$

Properties of the saturated vapor,

Internal energy, $u_g = 2553.1\text{kJ/kg}$

Specific volume, $v_g = 0.46242\text{m}^3/\text{kg}$

Entropy, $s_g = 6.8955\text{kJ/kg}\cdot\text{K}$

For the escaped saturated water at pressure $P_e = 400\text{kPa}$

Properties of the saturated water,

Internal energy, $h_e = 604.66\text{kJ/kg}$

Entropy, $s_e = 1.7765\text{kJ/kg}\cdot\text{K}$

Step 3 of 5

Let the mass of the saturated vapor be m_2

So,

$$m_1 v_1 = m_2 v_2$$

$$m_2 = \frac{m_1 v_1}{v_2}$$

$$= \frac{7.5 v_1}{0.46242}$$

$$m_2 = 16.219 v_1 \quad \text{..... (1)}$$

By the mass balance equation,

Mass of escaped gas, $m_e = m_1 - m_2$

By the energy balance equation,

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1$$

$$5 - (7.5 - 16.219 v_1)(604.66) = 16.219 v_1 (2553.1) - 7.5 u_1 \quad \text{..... (2)}$$

By the entropy balance,

$$\frac{Q_{in}}{T} - m_e s_e + S_{gen} = m_2 s_2 - m_1 s_1$$

Substituting equation (1) in above expression,

$$\frac{Q_{in}}{T} - (7.5 - 16.219 v_1)(1.7765) + S_{gen} = (16.219 v_1) 6.8955 - 7.5 s_1$$

$$\frac{5}{500+273} - (7.5 - 16.219 v_1)(1.7765) + S_{gen} = (16.219 v_1) 6.8955 - 7.5 s_1 \quad \text{..... (3)}$$

Step 4 of 5

Solving the above equations by EES software

$x_1 = 0.866$

Hence, properties of the water at initial state are,

Pressure, $P_1 = 400\text{kPa}$ and $x_1 = 0.866$

Internal energy, $u_1 = 2293.2\text{kJ/kg}$

Specific volume, $v_1 = 0.40089\text{m}^3/\text{kg}$

Entropy, $s_1 = 6.2129\text{kJ/kg}\cdot\text{K}$

Substituting the values in equation (1) and (3)

$$m_e = 7.5 - 16.219(0.40089)$$

$$= 0.998\text{kg}$$

Therefore, amount of mass that has escaped is $\boxed{m_e = 0.998\text{kg}}$

Step 5 of 5

(c) The entropy generation,

From the equation (3)

$$\frac{5}{500+273} - (7.5 - 16.219 v_1)(1.7765) + S_{gen} = (16.219 v_1) 6.8955 - 7.5 s_1$$

$$\frac{5}{500+273} - (7.5 - 16.219(0.40089))(1.7765) + S_{gen} = \left\{ \begin{matrix} (16.219(0.40089))6.8955 \\ -7.5(6.2129) \end{matrix} \right\}$$

$$S_{gen} = -1.761968 + 1.7664$$

$$= 4.43 \times 10^{-3}\text{kJ/K}$$

Therefore, the entropy generation during the process is $\boxed{S_{gen} = 4.43 \times 10^{-3}\text{kJ/K}}$

Question:

8.164P

A heat engine whose thermal efficiency is 40 percent uses a hot reservoir at 1300 R and a cold reservoir at 500 R. Calculate the entropy change of the two reservoirs when 1 Btu of heat is transferred from the hot reservoir to the engine. Does this engine satisfy the increase of entropy principle? If the thermal efficiency of the heat engine is increased to 70 percent, will the increase of entropy principle still be satisfied?

Step 1 of 6

Given data

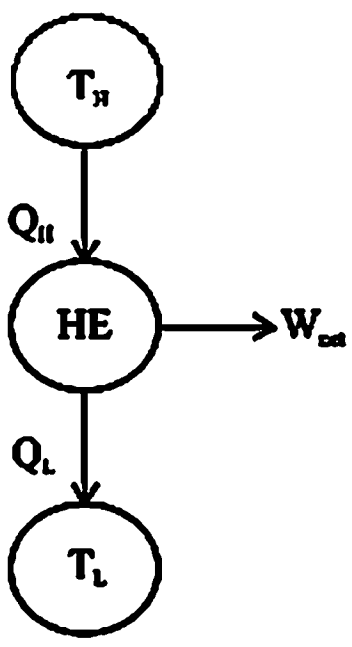
Temperature of hot reservoir $T_H = 1300\text{R}$

Temperature of cold reservoir $T_L = 500\text{R}$

Heat transferred from hot reservoir $Q_H = 1\text{Btu}$

Thermal efficiency $\eta = 0.4$

Step 2 of 6



Step 3 of 6

Thermal efficiency

$$\eta = 1 - \frac{Q_L}{Q_H}$$

$$0.4 = 1 - \frac{Q_L}{1}$$

$$Q_L = 0.6\text{Btu}$$

Heat transferred to cold reservoir

$$Q_L = 0.6\text{Btu}$$

Step 4 of 6

Entropy change of two reservoirs

$$\Delta S_{total} = \Delta S_H + \Delta S_L$$

$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L}$$

$$= \frac{-1}{1300} + \frac{0.6}{500}$$

$$\boxed{\Delta S_{total} = 0.000431\text{Btu/R}}$$

When thermal efficiency of the heat engine is $\eta = 40\%$ total entropy increases

So this engine satisfies the increase of entropy principle

Step 5 of 6

When thermal efficiency of engine is $\eta = 70\%$

$$\eta = 1 - \frac{Q_L}{Q_H}$$

$$0.7 = 1 - \frac{Q_L}{1}$$

$$Q_L = 0.3\text{Btu}$$

Heat transferred to cold reservoir

$$Q_L = 0.3\text{Btu}$$

Step 6 of 6

Total entropy change of two reservoirs

$$\Delta S_{total} = \Delta S_H + \Delta S_L$$

$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L}$$

$$= \frac{-1}{1300} + \frac{0.3}{500}$$

$$= -0.000169\text{Btu/R}$$

$$\boxed{\Delta S_{total} = -0.000169\text{Btu/R}}$$

When thermal efficiency of the heat engine is $\eta = 70\%$ total entropy decreases

So the engine does not satisfies the increase of entropy principle therefore the engine is impossible

Question: 8.165P pg 1

A refrigerator with a coefficient of performance of 4 transfers heat from a cold region at -20°C to a hot region at 30°C. Calculate the total entropy change of the regions when 1 kJ of heat is transferred from the cold region. Is the second law satisfied? Will this refrigerator still satisfy the second law if its coefficient of performance is 6?

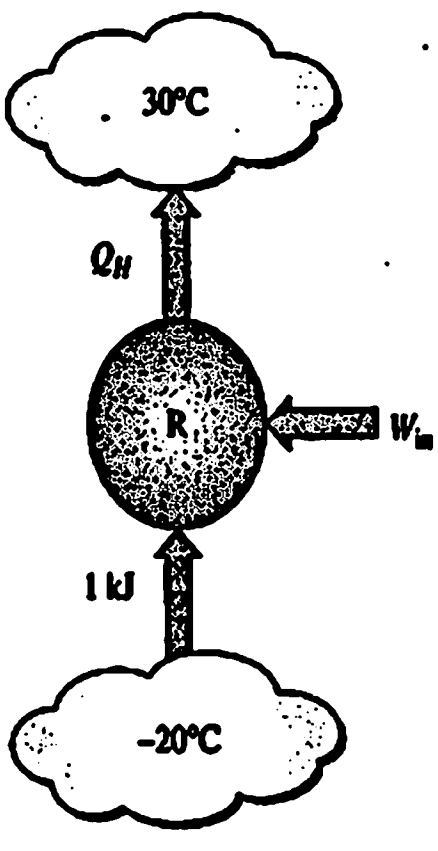


FIGURE P8–165

Step 1 of 6

Given

Coefficient of performance of refrigerator $COP_R = 4$

Temperature of cold region $T_L = -20°C$

$= (-20 + 273)K$

$T_L = 253K$

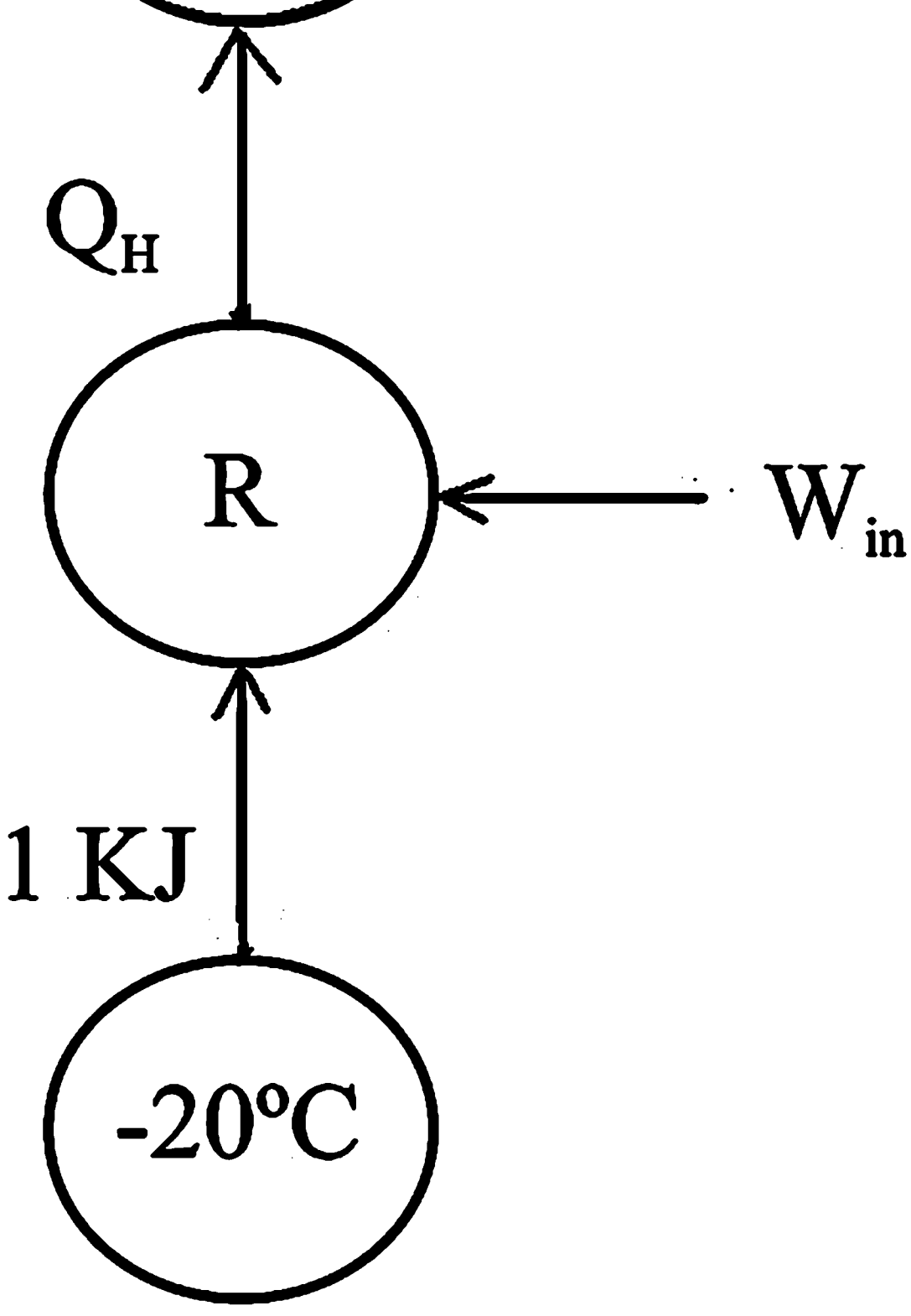
Temperature of hot region $T_H = 30°C$

$= (30 + 273)K$

$T_H = 303K$

Heat transferred from the cold region $Q_L = 1kJ$

Step 2 of 6



Step 3 of 6

Coefficient of the performance of the refrigerator is

$$COP_R = \frac{Q_L}{Q_H - Q_L}$$

$$4 = \frac{1}{Q_H - 1}$$

$$Q_H - 1 = 0.25$$

$$Q_H = 1.25kJ$$

Heat transferred to the hot region

$$Q_H = 1.25kJ$$

$Q_H = 1.25\,\text{kJ}$

Heat transferred to the hot region

$$Q_H = 1.25\,\text{kJ}$$

Step 4 of 6

Entropy change of the process

$$\Delta S_{total} = \Delta S_H + \Delta S_L$$

$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L}$$

$$= \frac{1.25}{303} + \frac{(-1)}{253}$$

$$\boxed{\Delta S_{total} = 0.000173\,\text{kJ/K}}$$

When $COP = 4$ the total entropy increases so the process is possible

Step 5 of 6

When $COP = 6$

$$COP_R = \frac{Q_L}{Q_H - Q_L}$$

$$6 = \frac{1}{Q_H - 1}$$

$$Q_H - 1 = 0.1666$$

$$Q_H = 1.1666\,\text{kJ}$$

Heat transferred to the hot region

$$Q_H = 1.1666\,\text{kJ}$$

Step 6 of 6

Total entropy change

$$\Delta S_{total} = \Delta S_H + \Delta S_L$$

$$= \frac{Q_H}{T_H} + \frac{Q_L}{T_L}$$

$$= \frac{1.1666}{303} + \frac{(-1)}{253}$$

$$= -0.0001024\,\text{kJ/K}$$

$$\boxed{\Delta S_{total} = -0.0001024\,\text{kJ/K}}$$

Since total entropy is negative when $COP = 6$, so this process is impossible.

Question: 8.166P

Calculate the rate of entropy change of all the components of a refrigerator that uses 10 kW of power, rejects 14 kW of heat, and has a high-temperature energy reservoir at 400 K and a low-temperature energy reservoir at 200 K. What is the rate of cooling produced by this refrigerator? Is this refrigerator completely reversible?

Step 1 of 4

Given data:

Power given to the refrigerator is, $W_{in} = 10\,kW$.

Rate of heat rejection at source is $Q_H = 14\,kW$.

Temperature of source is, $T_H = 400\,K$.

Temperature of sink is, $T_L = 200\,K$.

Step 2 of 4

The work input to the refrigerator is given by

$$W_{in} = Q_H - Q_L$$

Therefore the rate of heat addition at sink is given by

$$Q_L = Q_H - W_{in}$$
$$= 14 - 10$$
$$= 4\,kW$$

$$\boxed{Q_L = 4\,kW}$$

The entropy change at sink is given by

$$\Delta S_L = \frac{Q_L}{T_L}$$

$$\Delta S_L = \frac{-4}{200}$$

(as heat is taken out of the sink, negative sign is taken)

$$\Delta S_L = -0.02\,kW/k$$

Step 3 of 4

The entropy change at source is given by

$$\Delta S_H = \frac{Q_H}{T_H}$$

$$\Delta S_H = \frac{14}{400}$$ (as heat is given to the source, positive sign is taken)

$$\Delta S_H = 0.035\,kW/K$$

Step 4 of 4

The total entropy change is given by

$$\Delta S_{tot} = \Delta S_H + \Delta S_L + \Delta S_{dev}$$
$$= 0.035 - 0.02 + 0$$
$$= 0.015\,kW/K$$

Therefore the rate of cooling produced by the refrigerator is 4 kW.

As the entropy is increased, the refrigerator is real and irreversible.

Question: 8.167P

What is the minimum internal energy that R-134a can achieve as it is compressed adiabatically from 200 kPa and 85 percent quality to 800 kPa in a closed system?

Step 1 of 5

Given data:

Inlet pressure of the refrigerant is, $P_1 = 200$ kPa.

Inlet quality of the refrigerant is, $x_1 = 85\% = 0.85$.

Exit pressure of the refrigerant is, $P_2 = 800$ kPa.

Step 2 of 5

From saturated refrigerant 134a Pressure table

At pressure $P_1 = 200$ kPa ,

the saturated properties of steam are

Entropy of saturated liquid is $s_{f1} = 0.15457\ \text{kJ/kg} \cdot \text{K}$

Entropy of evaporation is $s_{fg1} = 0.78316\ \text{kJ/kg} \cdot \text{K}$

From saturated refrigerant 134a Pressure table,

At pressure $P_2 = 800$ kPa

Saturated properties of steam are as follows

Internal energy of saturated liquid is $u_{f2} = 94.79\ \text{kJ/kg}$

Internal energy of evaporation is $u_{fg2} = 152.0\ \text{kJ/kg}$

Entropy of saturated liquid is $s_{f2} = 0.35404\ \text{kJ/kg} \cdot \text{K}$

Entropy of evaporation is $s_{fg2} = 0.56431\ \text{kJ/kg} \cdot \text{K}$

Step 3 of 5

The inlet entropy is given by

$$s_1 = s_{f1} + x_1 \times s_{fg1}$$
$$= 0.15457 + 0.85 \times 0.78316$$
$$= 0.15457 + 0.66569$$
$$= 0.82026\ \text{kJ/kg} \cdot \text{K}$$

Step 4 of 5

As the process is adiabatic, $s_2 = s_1$

Therefore, $s_2 = 0.82026\ \text{kJ/kg} \cdot \text{K}$

The exit entropy is also given by

$$s_2 = s_{f2} + x_2 \times s_{fg2}$$
$$0.82026 = 0.35404 + x_2 \times 0.56431$$
$$x_2 = \frac{0.82026 - 0.35404}{0.56431}$$
$$= \frac{0.46622}{0.56431}$$
$$= 0.8262$$

Step 5 of 5

The exit internal energy is also given by

$$u_2 = u_{f2} + x_2 u_{fg2}$$
$$= 94.79 + (0.8262)(152)$$
$$= 94.79 + 125.5824$$
$$= 220.3724\ \text{kJ/kg}$$

Therefore the exit internal energy is $\boxed{u_2 = 220.3724\ \text{kJ/kg}}$.

Question: 8.168P

Is it possible to cool and condense R-134a to a saturated liquid from 1000 kPa and 180°C in a closed system undergoing an isobaric, reversible process while exchanging heat with an isothermal energy reservoir at 100°C?

Step 1 of 5

Given

Constant pressure is, $P_1 = P_2 = 1000$ kPa

Initial temperature is, $T_1 = 180$ °C

$= 453$ K

Reservoir temperature is, $T_{res} = 100$ °C

$= 373$ K

Step 2 of 5

From the properties of superheated refrigerant 134a,

at pressure $P_1 = 1000$ kPa and temperature $T_1 = 180$ °C properties of R-134a are

$h_1 = 421.36$ kJ/kg

$s_1 = 1.3124$ kJ/kg°K

From the properties of saturated refrigerant 134a,

at pressure $P_2 = 1000$ kPa and saturation liquid condition properties of R-134a are

$h_2 = h_f = 107.32$ kJ/kg

$s_2 = s_f = 0.39189$ kJ/kg°K

Step 3 of 5

The energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

$$-Q_{out} - W_{out} = \Delta U$$

$$-Q_{out} = \Delta U + W_{out}$$

$$-Q_{out} = \Delta H$$

$$-Q_{out} = m(h_2 - h_1)$$

$$Q_{out} = m(h_1 - h_2)$$

$$q_{out} = h_1 - h_2$$

$$q_{out} = 421.36 - 107.32$$

$$q_{out} = 314.04 \text{ kJ/kg}$$

Step 4 of 5

Entropy change of reservoir is given by

$$\Delta s_{res} = \Delta s_{surr} = \frac{q_{out}}{T_{res}}$$

$$\Delta s_{surr} = \frac{314.04}{373}$$

$$\Delta s_{surr} = 0.8419 \text{ kJ/kg°K}$$

Step 5 of 5

The total entropy change is given by

$$\Delta s_{tot} = \Delta s_{R134a} + \Delta s_{surr}$$

$$\Delta s_{tot} = s_2 - s_1 + \Delta s_{surr}$$

$$\Delta s_{tot} = 0.39189 - 1.3124 + 0.8419$$

$$\Delta s_{tot} = -0.0786 \text{ kJ/kg°K}$$

$\boxed{\Delta s_{tot} < 0}$

As the entropy change is negative the process is impossible.

Question: 8.169P

It has been suggested that air at 100 kPa and 25°C can be cooled by first compressing it adiabatically in a closed system to 1000 kPa and then expanding it adiabatically back to 100 kPa. Is this possible?

Step 1 of 4

Given data:

Initial pressure of air is, $P_1 = 100\ kPa$.

Initial temperature of air is, $T_1 = 25\ °C = 298\ K$.

Intermediate pressure of air is, $P_2 = 1000\ kPa$.

Final pressure of air is, $P_3 = 100\ kPa$.

Step 2 of 4

Air is first compressed adiabatically from $P_1 = 100\ kPa$ to $P_2 = 1000\ kPa$

As air is compressed the temperature will be increasing.

$T_2 > T_1$

Air is then expanded adiabatically to the final (same as initial) pressure.

Now it is to be determined if air can be cooled by this process.

Step 3 of 4

The entropy change is given by

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right) - R \times \ln\left(\frac{P_3}{P_1}\right)$$

As $P_3 = P_1 = 100\ kPa$, the equation reduces to

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right) - R \times \ln\left(\frac{P_1}{P_1}\right)$$

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right) - R \times \ln(1)$$

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right) - R \times 0$$

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right) - 0$$

$$\Delta s = c_p \times \ln\left(\frac{T_3}{T_1}\right)$$

Step 4 of 4

As the process is adiabatic, there will not be any heat transfer and so entropy generation will be equal to entropy change.

As entropy generation should not be negative, the entropy change can be

$\Delta s \geq 0$

$$c_p \times \ln\left(\frac{T_3}{T_1}\right) \geq 0$$

$$\ln\left(\frac{T_3}{T_1}\right) \geq \frac{0}{c_p}$$

$$\ln\left(\frac{T_3}{T_1}\right) \geq 0$$

$$\frac{T_3}{T_1} \geq e^0$$

$$\frac{T_3}{T_1} \geq 1$$

$$\boxed{T_3 \geq T_1}$$

By this process the final temperature will be positive or at least equal to the initial temperature and so it is not possible to cool air using this process.

Question: 8.170P

Can 1 lbm of air in a closed system at 20 psia and 100°F be compressed adiabatically to 120 psia and a volume of 3 ft3 ?

Step 1 of 4

Given data:

Mass of air is $m = 1$ lbm

Initial pressure of air is, $P_1 = 20$ psia

Initial temperature of air is, $T_1 = 100\ ^\circ\text{F}$

$= 559.67\ \text{R}$

Final pressure of air is, $P_2 = 120$ psia

Volume of air is, $V = 3\ \text{ft}^3$

Step 2 of 4

From properties of molar mass, gas constant and critical point

For air $R = 0.3704\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$

Similarly, from ideal gas specific heats of various common gases

For air $c_p = 0.240\ \text{Btu/lbm} \cdot \text{R}$ and $k = 1.4$

We know that

Air is an ideal gas.

The ideal gas equation is given by

$$P_1 \times v_1 = m \times R \times T_1$$

$$v_1 = \frac{m \times R \times T_1}{P_1}$$

$$v_1 = \frac{1 \times 0.3704 \times 559.67}{20}$$

$$v_1 = \frac{207.3}{20}$$

$$v_1 = 10.365\ \text{ft}^3/\text{lbm}$$

Step 3 of 4

Now we know that

The ratio of specific volumes in an isentropic process is given by,

$$\frac{v_2}{v_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}$$

$$v_2 = v_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}$$

$$v_2 = 10.365 \times \left(\frac{20}{120}\right)^{\frac{1}{1.4}}$$

$$v_2 = 10.365 \times 0.2781$$

$$v_2 = 2.88\ \text{ft}^3/\text{lbm}$$

Step 4 of 4

Now we know that

The minimum final volume is given by

$$V_2 = m \times v_2$$

$$V_2 = 1 \times 2.8742$$

$$V_2 = 2.8742\ \text{ft}^3/\text{lbm}$$

$$\boxed{V_2 < V}$$

As minimum final volume ($2.8742\ \text{ft}^3/\text{lbm}$) is less than the volume ($3\ \text{ft}^3/\text{lbm}$) given, it is possible to compress the air in the given condition.

Question: 8.171P

A 100-lbm block of a solid material whose specific heat is 0.5 Btu/lbm·R is at 70°R It is heated with 10 lbm of saturated water vapor that has a constant pressure of 14.7 psia. Determine the final temperature of the block and water, and the entropy change of (*a*) the block, (*b*) the water, and (*c*) the entire system. Is this process possible? Why?

Step 1 of 8

Given data:

The mass of solid block, $m = 100\,\text{lbm}$

The specific heat of the solid material, $c = 0.5\,\text{Btu/lbm}\cdot\text{R}$

The initial temperature of the solid block, $T_1 = 70°\text{F}$

The mass of saturated water, $m_w = 10\,\text{lbm}$

The pressure of saturated water vapor, $p = 14.7\,\text{psia}$

Step 2 of 8

Determination of final temperature of the water and solid block:

Since the block is heated, some of the water vapor will be condensed.

From saturated water tables, at $p = 14.7\,\text{psia}$

The saturation temperature is, $T_2 = 212°\text{F}$

Therefore the final temperature of the water and the solid block is, $\boxed{T_2 = 212°\text{F}}$

Step 3 of 8

The heat gained by the block,

$$Q_{block} = mc(T_2 - T_1)$$
$$= 100 \times 0.5 \times (212 - 70)$$
$$= 7100\,\text{Btu}$$

The water properties at initial state can determined as follows:

Since, at initial state the water is in saturated vapor state, the quality of water vapor is,

$x_1 = 1$

From properties of saturated water pressure tables, at $p = 14.7\,\text{psia}$

The specific enthalpy of saturated vapor, $h_g = h_1 = 1150.3\,\text{Btu/lbm}$

The specific entropy of saturated vapor, $s_g = s_1 = 1.7566\,\text{Btu/lbm}\cdot\text{R}$

Step 4 of 8

The water properties at final state can be determined as follows:

The enthalpy is,

$$h_2 = h_1 - \frac{Q_{water}}{m_w}$$
$$= 1150.3 - \left(\frac{7100}{10}\right)$$
$$= 440.3\,\text{Btu/lbm}$$

Now the quality of water at this state is,

$h_2 = h_{f2} + x_2 h_{fg2}$

From properties of saturated water pressure tables, at $p = 14.7\,\text{psia}$

The specific enthalpy of saturated liquid, $h_{f2} = 180.16\,\text{Btu/lbm}$

The specific enthalpy of saturated water mixture, $h_{fg2} = 970.12\,\text{Btu/lbm}$

The specific entropy of saturated liquid, $s_{f2} = 0.31215\,\text{Btu/lbm}\cdot\text{R}$

The specific entropy of saturated water mixture, $s_{fg2} = 1.44441\,\text{Btu/lbm}\cdot\text{R}$

Step 5 of 8

Therefore,

$h_2 = h_{f2} + x_2 h_{fg2}$

$440.3 = 180.16 + (x_2 \times 970.12)$

$$x_2 = \frac{440.3 - 180.16}{970.12}$$

$x_2 = 0.2681$

And,

$$s_2 = s_{f2} + x_2 s_{fg2}$$
$$= (0.31215) + (0.2681 \times 1.44441)$$
$$= 0.31215 + 0.3873$$
$$= 0.69947$$

Step 6 of 8

(a)Calculation of entropy change of water:

The entropy change of water is,

$$\Delta S_{water} = m_w (s_2 - s_1)$$
$$= 10 \times (0.69947 - 1.7566)$$
$$= -10.571\,\text{Btu/R}$$

Therefore the entropy change of water, $\boxed{\Delta S_{water} = -10.571\,\text{Btu/R}}$

Step 7 of 8

(b)Calculation of the entropy change of block:

The entropy change of block is,

$$\Delta S_{block} = mc \ln \frac{T_2}{T_1}$$
$$= 100 \times 0.5 \times \ln\left(\frac{212 + 460}{70 + 460}\right)$$
$$= 11.869\,\text{Btu/R}$$

Therefore, the entropy change of block, $\boxed{\Delta S_{block} = 11.869\,\text{Btu/R}}$

Step 8 of 8

(c) Calculation of entropy change of entire system:

The total entropy change is,

$$\Delta S_{total} = \Delta S_{water} + \Delta S_{block}$$
$$= -10.57 + 11.87$$
$$= 1.299\,\text{Btu/R}$$

Therefore, the entropy change of entire system, $\boxed{\Delta S_{total} = 1.299\,\text{Btu/R}}$

Since, the entropy change for the entire system is positive, this process is possible.

Question: 8.172P

One kilogram of air is in a piston-cylinder apparatus that can only exchange heat with a reservoir at 300 K. Initially this air is at 100 kPa and 27°C. Someone claims that the air can be compressed to 250 kPa and 27°C. Determine if this claim is valid by performing a second-law analysis of the claimed process.

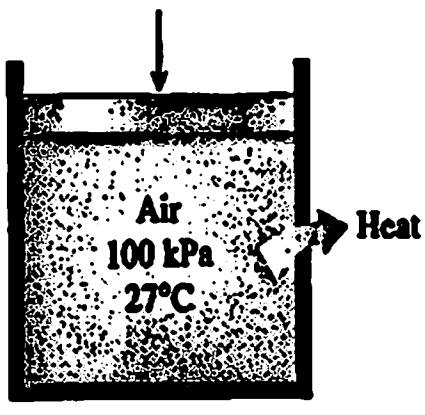


FIGURE P8–172

Step 1 of 5

Given data:

The mass of air, $m = 1\text{kg}$

Temperature of reservoir, $T_R = 300\text{K}$

Initial pressure of air, $p_1 = 100\text{kPa}$

Final pressure of air, $p_2 = 250\text{kPa}$

Initial and final temperature of air is, $T_1 = T_2 = 27°\text{C}$

Step 2 of 5

In order to determine which process is possible, let us calculate the total entropy as follows:

Entropy change of air during isothermal process is,

$$\Delta s_{air} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \qquad \text{..... (1)}$$

Entropy change of reservoir is,

$$\Delta s_R = \frac{q_R}{T_R} \qquad \text{..... (2)}$$

Step 3 of 5

From the properties of ideal gas specific heat properties,

Specific heat of air is, $c_p = 1.005\text{kJ/kg}\cdot\text{K}$

And, we know that, gas constant, $R = 0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$

By substituting these known values in equation (1), we get,

$$\Delta s_{air} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= \left[1.005\times\ln\left(\frac{27}{27}\right)\right] - \left[0.287\times\ln\left(\frac{250}{100}\right)\right]$$

$$= 0 - 0.2629$$

$$= -0.2629\text{kJ/kg}\cdot\text{K}$$

Step 4 of 5

To calculate the rate of heat transfer, let us consider the energy balance,

$$W_{in} - Q_{out} = \Delta U$$

$$W_{in} - Q_{out} = mc_p(T_2 - T_1)$$

$$W_{in} - Q_{out} = 0 \qquad (\because T_1 = T_2)$$

$$W_{in} = Q_{out}$$

$$w_{in} = q_{out}$$

We know that,

$$w_{in} = RT\ln\frac{P_2}{P_1}$$

$$= 0.287\times300\times\ln\left(\frac{250}{100}\right)$$

$$= 78.89\text{kJ/kg}$$

Therefore, $w_{in} = q_{out} = 78.89\text{kJ/kg}$

Step 5 of 5

By substituting this rate of heat transfer value in equation (2), we get

$$\Delta s_R = \frac{q_R}{T_R}$$

$$= \frac{78.89}{300}$$

$$= 0.2629\text{ kJ/kg}\cdot\text{K}$$

Now, the total entropy is,

$$\Delta s_{total} = \Delta s_{air} + \Delta s_R$$

$$= -0.2629 + 0.2629$$

$$= 0\text{kJ/kg}\cdot\text{K}$$

Since, the total entropy is zero; second law of reversible process is possible.

Question: 8.173P

A rigid tank contains 3.2 kg of water at 140°C and 400 kPa. Now 48 kJ of shaft work is done on the system and the final temperature in the tank is 80°C. If the entropy change of water is zero and the surroundings are at 15°C, determine (a) the final pressure in the tank, (*b*) the amount of heat transfer between the tank and the surroundings, and (c) the entropy generation during this process.

Step 1 of 7

Given

Mass of water in the tank is, $m = 3.2\,\text{kg}$

Initial temperature of water is, $T_1 = 140°\text{C}$

Initial pressure of water is, $P_1 = 400\,\text{kPa}$

Work done on the system is, $W_{in} = 48\,\text{kJ}$

Final temperature of water is, $T_2 = 80°\text{C}$

Surrounding temperature is, $T_{sur} = 15°\text{C} = 288\,\text{K}$

Step 2 of 7

At pressure $P_1 = 400\,\text{kPa}$ from saturated water tables

The saturated temperature is $T_{sat} = 143.61°\text{C}$

And $T_{sat} > T_1$

So we will consider saturated liquid properties at initial temperature

At temperature $T_1 = 140°\text{C}$ and saturated liquid condition, properties of water are

$u_1 = u_f = 588.77\,\text{kJ/kg}$

$s_1 = s_f = 1.7392\,\text{kJ/kg}\cdot\text{K}$

Step 3 of 7

From saturated water tables at temperature $T_2 = 80°\text{C}$ properties of water are

$u_{f2} = 334.97\,\text{kJ/kg}$

$u_{fg2} = 2146.6\,\text{kJ/kg}$

$s_{f2} = 1.0756\,\text{kJ/kg}\cdot\text{K}$

$s_{fg2} = 6.5355\,\text{kJ/kg}\cdot\text{K}$

As given there will not be any change in entropy of water.

$s_1 = s_2$

$s_1 = s_{f2} + x_2 \times s_{fg2}$

$1.7392 = 1.0756 + x_2 \times 6.5355$

$$x_2 = \frac{1.7392 - 1.0756}{6.5355}$$

$x_2 = 0.1015$

Step 4 of 7

Therefore the final internal energy is given by

$u_2 = u_{f2} + x_2 \times u_{fg2}$

$u_2 = 334.97 + 0.1015 \times 2146.6$

$u_2 = 334.97 + 217.88$

$u_2 = 552.85\,\text{kJ/kg}$

Step 5 of 7

(a) At state 2 the state is in wet condition the pressure will be the saturation pressure at exit temperature.

From saturated water tables at $T_2 = 80°\text{C}$

$P_2 = P_{sat} = 47.416\,\text{kPa}$

$\boxed{P_2 = 47.416\,\text{kPa}}$

Step 6 of 7

(b) Applying energy balance to the system

$Q_{out} = W_{in} - m(u_2 - u_1)$

$Q_{out} = 48 - 3.2(552.85 - 588.77)$

$Q_{out} = 48 - 3.2(-35.92)$

$Q_{out} = 48 + 114.944$

$\boxed{Q_{out} = 162.944\,\text{kJ}}$

Step 7 of 7

(c) As entropy change of water is zero, the entropy generation will be only due to the change in surroundings entropy

$$S_{gen} = S_{sur} = \frac{Q_{out}}{T_{sur}}$$

$$S_{gen} = \frac{162.944}{288}$$

$\boxed{S_{gen} = 0.5657\,\text{kJ/K}}$

Therefore

The final pressure in the tank is 47.416 kPa

The amount of heat transfer between the tank and the surroundings is 162.944 kJ

The entropy generation is 0.5657 kJ/K

Question: 8.174P pg 1

A horizontal cylinder is separated into two compartments by an adiabatic, frictionless piston. One side contains 0.2 m3 of nitrogen and the other side contains 0.1 kg of helium, both initially at 20°C and 95 kPa. The sides of the cylinder and the helium end are insulated. Now heat is added to the nitrogen side from a reservoir at 500°C until the pressure of the helium rises to 120 kPa. Determine (*a*) the final temperature of the helium, (*b*) the final volume of the nitrogen, (*c*) the heat transferred to the nitrogen, and (*d*) the entropy generation during this process.

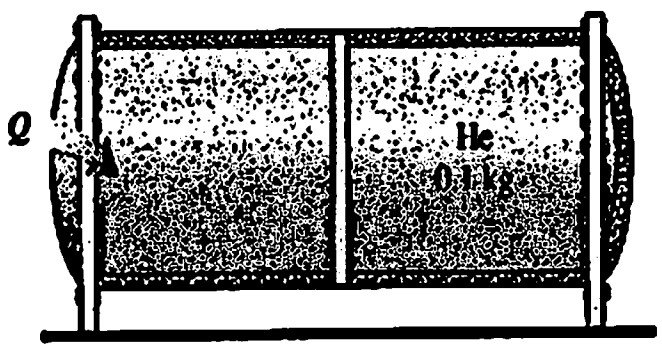


FIGURE P8–174

Step 1 of 6

Obtain the properties of Helium and Nitrogen from "ideal gas specific heats of gases" table,

Gas constant of Helium $R_{He} = 2.0769\,\text{kJ/kg}\cdot\text{K}$

Gas constant of Nitrogen $R_{N_2} = 0.2968\,\text{kJ/kg}\cdot\text{K}$

Specific heat ratio of Helium $k_{He} = 1.667$

Specific heat ratio of Nitrogen $k_{N_2} = 1.400$

Specific heat of Helium at constant volume $c_{v,He} = 3.1156\,\text{kJ/kg}\cdot\text{K}$

Specific heat of Nitrogen at constant volume $c_{v,N_2} = 0.743\,\text{kJ/kg}\cdot\text{K}$

Specific heat of Nitrogen at constant pressure $c_{p,N_2} = 1.039\,\text{kJ/kg}\cdot\text{K}$

Step 2 of 6

(a)

Find the final temperature of helium by applying isentropic relation to the process

$$\frac{T_{2,He}}{T_{1,He}} = \left(\frac{P_{2,He}}{P_{1,He}}\right)^{\frac{k-1}{k}}$$

Here, $T_{1,He}$ and $T_{2,He}$ are the initial and final temperatures of Helium, $P_{1,He}$ and $P_{2,He}$ are the initial and final pressures of Helium

Substitute $(20°\text{C} = 293\,\text{K})$ for $T_{1,He}$, $95\,\text{kPa}$ for $P_{1,He}$ and $120\,\text{kPa}$ for $P_{2,He}$

$$\frac{T_{2,He}}{293} = \left(\frac{120}{95}\right)^{\frac{1.667-1}{1.667}}$$

$$\frac{T_{2,He}}{293} = 1.097$$

$$T_{2,He} = 321.4\,\text{K}$$

Therefore, the final temperature of Helium is $\boxed{321.4\,\text{K}}$

Step 3 of 6

(b)

Find the initial volume of helium by using ideal gas equation

$$P_{1,He}V_{1,He} = m_{1,He}RT_{1,He}$$

Here, $m_{1,He}$ is mass of Helium

Substitute 0.1 kg for m_{He}

$$95 \times V_{1,He} = 0.1 \times 2.0769 \times 293$$

$$V_{1,He} = 0.64\,\text{m}^3$$

Similarly find the final volume of helium by using the equation,

$$V_{2,He} = \frac{m_{He}RT_{2,He}}{P_{2,He}}$$

$$= \frac{0.1 \times 2.0769 \times 321.4}{120}$$

$$= 0.5562\,\text{m}^3$$

Step 4 of 6

Now find the final volume of nitrogen by applying balancing equation,

$$V_{1,N_2} + V_{1,He} = V_{2,N_2} + V_{2,He}$$

$$V_{2,N_2} = V_{1,N_2} + V_{1,He} - V_{2,He}$$

Here, V_{1,N_2} is initial volume of Nitrogen

Substitute $0.2\,\text{m}^3$ for V_{1,N_2}

$$V_{2,N_2} = 0.2 + 0.64 - 0.5562$$

$$= 0.2838\,\text{m}^3$$

Therefore, final volume of Nitrogen is $\boxed{0.2838\,\text{m}^3}$

Step 5 of 6

(c)

Find the Mass of nitrogen by using ideal gas equation,

$$m_{N_2} = \frac{P_{1,N_2}V_{1,N_2}}{R_{N_2}T_{1,N_2}}$$

Find the Mass of nitrogen by using ideal gas equation,

$$\begin{aligned} m_{N_2} &= \frac{P_{1,N_2} V_{1,N_2}}{R_{N_2} T_{1,N_2}} \\ &= \frac{95 \times 0.2}{0.2968 \times 293} \\ &= 0.21848 \text{ kg} \end{aligned}$$

Find final temperature of nitrogen

$$\begin{aligned} T_{2,N_2} &= \frac{P_{2,N_2} V_{2,N_2}}{m_{N_2} R_{N_2}} \\ &= \frac{120 \times 0.2838}{0.21848 \times 0.2968} \\ &= 525.19 \text{ K} \end{aligned}$$

Apply energy balance to the system,

$$\begin{aligned} Q_{in} &= \Delta U_{N_2} + \Delta U_{He} \\ &= \left[mc_v (T_2 - T_1)\right]_{N_2} + \left[mc_v (T_2 - T_1)\right]_{He} \\ &= (0.21848)(0.743)(525.19 - 293) + (0.1)(3.1156)(321.4 - 293) \\ &= 37.69 + 8.848 \\ &= 46.538 \text{ kJ} \end{aligned}$$

Therefore, heat transferred to the Nitrogen is **46.538 kJ**

Step 6 of 6

(d)

Find the entropy generation during this process by using the equation,

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{surr}$$

Here, $\Delta S_{He} = 0$

Since the process is isentropic process in Helium

$$S_{gen} = m_{N_2}\left[c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}\right]_{N_2} + \left[\frac{-Q_{in}}{T_{Surr}}\right]$$

Here, T_{Surr} is temperature of the reservoir from which heat is added to the Nitrogen side.

Substitute $500°C$ for T_{Surr}

$$\begin{aligned} S_{gen} &= (0.21848)\left[(1.039)\ln \frac{525.19}{293} - (0.2968)\ln \frac{120}{95}\right] + \left(\frac{-46.538}{(500+273)}\right) \\ &= 0.21848[0.6063 - 0.06933] - 0.0602 \\ &= 0.05711 \text{ kJ/K} \end{aligned}$$

Therefore, entropy generation during the process is **0.05711 kJ/K**

Question: 8.175P

A 0.8-m3 rigid tank contains carbon dioxide (CO2) gas at 250 K and 100 kPa. A 500-W electric resistance heater placed in the tank is now turned on and kept on for 40 min after which the pressure of CO2 is measured to be 175 kPa. Assuming the surroundings to be at 300 K and using constant specific heats, determine (*a*) the final temperature of CO2 , (*b*) the net amount of heat transfer from the tank, and (*c*) the entropy generation during this process.

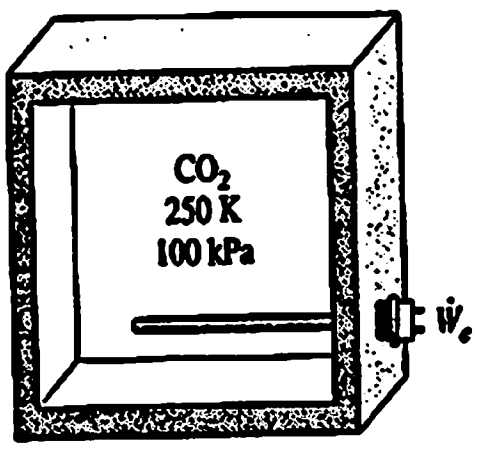


FIGURE P8–175

Step 1 of 5

Given data:

The volume of the tank, $V = 0.8\,\mathrm{m}^3$

The initial temperature of CO_2, $T_1 = 250\,\mathrm{K}$

The initial pressure of CO_2, $P_1 = 100\,\mathrm{kPa}$

The final pressure of CO_2, $P_2 = 175\,\mathrm{kPa}$

Time duration, $\Delta t = 40\,\mathrm{min}$

The power of electrical resistance heater, $\dot{E}_{e,in} = 500\,\mathrm{W}$

The surrounding temperature, $T_\infty = 300\,\mathrm{K}$

Step 2 of 5

(a)Calculation of final temperature of CO_2:

In order to calculate the final temperature, let us calculate mass of CO_2:

The mass of CO_2, $m = \dfrac{P_1 V}{R T_1}$ ..... (1)

Let us assume, the average temperature of CO_2, $T_{avg} = 350\,\mathrm{K}$

From specific heat properties of selected gases table, the properties of CO_2,

Gas constant, $R = 0.1889\,\mathrm{kPa \cdot m^3/kg \cdot K}$

Specific heat of pressure, $c_p = 0.895\,\mathrm{kJ/kg \cdot K}$

Specific heat of temperature, $c_v = 0.706\,\mathrm{kJ/kg \cdot K}$

Step 3 of 5

By substituting these known values in equation (1), we get

$$m = \frac{P_1 V}{R T_1}$$
$$= \frac{(100 \times 0.8)}{(0.1889 \times 250)}$$
$$= 1.694\,\mathrm{kg}$$

Now, the final temperature of CO_2 is,

$$T_2 = \frac{P_2 V}{mR}$$
$$= \frac{(175 \times 0.8)}{(1.694 \times 0.1889)}$$
$$= 437.5\,\mathrm{K}$$

Therefore, the final temperature of CO_2, $\boxed{T_2 = 437.5\,\mathrm{K}}$

Step 4 of 5

(b)Calculation of rate of heat transfer:

The rate of heat transfer can be calculated by using energy balance as follows:

$$Q_{out} = \dot{E}_{e,in}\Delta t - mc_v(T_2 - T_1)$$
$$= [0.5 \times (40 \times 60)] - [1.694 \times 0.706 \times (437.5 - 250)]$$
$$= 1200 - 224.24$$
$$= 975.75\,\mathrm{kJ}$$

Therefore, The rate of heat transfer, $\boxed{Q_{out} = 975.75\,\mathrm{kJ}}$

Step 5 of 5

(c)Calculation of entropy generation:

The entropy generation can be calculated by using,

$$S_{gen} = \Delta S_{CO_2} + \Delta S_{surr}$$
$$= m\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}\right) + \frac{Q_{out}}{T_{surr}}$$

By substituting the known values in the above equation, we get

$$S_{gen} = m\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}\right) + \frac{Q_{out}}{T_{surr}}$$
$$= \left[1.694 \times \left[\left(0.895 \times \ln \frac{437.5}{250}\right) - \left(0.1889 \times \ln \frac{175}{100}\right)\right]\right] + \left(\frac{975.8}{300}\right)$$
$$= [1.694 \times (0.5 - 0.1057)] + 3.25$$
$$= 3.917\,\mathrm{kJ/K}$$

Therefore, the entropy generation, $\boxed{S_{gen} = 3.917\,\mathrm{kJ/K}}$

Question: 8.176P

Helium gas is throttled steadily from 300 kPa and 50°C. Heat is lost from the helium in the amount of 1.75 kJ/kg to the surroundings at 25°C and 100 kPa. If the entropy of the helium increases by 0.25 kJ/kg·K in the valve, determine (*a*) the exit pressure and temperature and (*b*) the entropy generation during this process.

Step 1 of 5

Given data:

Initial pressure is, $P_1 = 300\ kPa$

Initial temperature is, $T_1 = 50\ °C = 323\ K$

Surrounding pressure is, $P_{sur} = 100\ kPa$

Surrounding temperature is, $T_{sur} = 25\ °C = 298\ K$

Heat lost by helium is, $q_{out} = 1.75\ kJ/kg$

Entropy change in helium is, $\Delta s_{He} = 0.25\ kJ/kg°K$

Step 2 of 5

From the ideal gas specific heats of various common gases

For helium

$c_p = 5.1926\ kJ/kg°K$

$R = 2.0769\ kJ/kg°K$

Step 3 of 5

(a) The heat lost by helium is given by

$$q_{out} = c_p(T_1 - T_2)$$

Therefore the final temperature is given by

$$T_2 = T_1 - \frac{q_{out}}{c_p}$$

$$T_2 = 323 - \frac{1.75}{5.1926}$$

$$T_2 = 323 - 0.337$$

$$\boxed{T_2 = 322.63\ K = 49.63\ °C}$$

Step 4 of 5

Entropy change in helium is given by

$$\Delta s_{He} = c_p \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$0.25 = 5.1926 \times \ln\left(\frac{322.63}{323}\right) - 2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$0.25 = 5.1926 \times \ln(0.9989) - 2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$0.25 = 5.1926 \times (-0.0011) - 2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$0.25 = -0.005712 - 2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$0.25 + 0.005712 = -2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$0.255712 = -2.0769 \times \ln\left(\frac{P_2}{300}\right)$$

$$\ln\left(\frac{P_2}{300}\right) = -\frac{0.255712}{2.0769}$$

$$\ln\left(\frac{P_2}{300}\right) = -0.123$$

$$\frac{P_2}{300} = e^{-0.123}$$

$$\frac{P_2}{300} = 0.8843$$

$$P_2 = 0.8843 \times 300$$

$$\boxed{P_2 = 265.29\ kPa}$$

Step 5 of 5

The entropy generation is given by

$$\Delta s_{gen} = \Delta s_{He} + \Delta s_{sur}$$

$$\Delta s_{gen} = \Delta s_{He} + \frac{q_{out}}{T_{sur}}$$

$$\Delta s_{gen} = 0.25 + \frac{1.75}{298}$$

$$\Delta s_{gen} = 0.25 + 0.00587$$

$$\boxed{\Delta s_{gen} = 0.25587\ kJ/kg°K}$$

Therefore the final temperature is 49.63 °C , the final pressure is 265.29 kPa and the entropy generation is 0.25587 kJ/kg°K

Question: 8.177P

Refrigerant-134a enters a compressor as a saturated vapor at 200 kPa at a rate of 0.03 m3 /s and leaves at 700 kPa. The power input to the compressor is 10 kW. If the surroundings at 20°C experience an entropy increase of 0.008 kW/K, determine (*a*) the rate of heat loss from the compressor, (*b*) the exit temperature of the refrigerant, and (c) the rate of entropy generation.

Step 1 of 3

(a)

Calculate the rate of heat loss by using the equation of Entropy increases by surrounding.

$$\Delta S_{surr} = \frac{q_{out}}{T_{surr}}$$

Here, temperature of the surroundings is T_{surr} , Entropy increase by surrounding is ΔS_{surr} , the rate of heat loss is q_{out} .

Substitute $(20°C + 273 = 293\ K)$ for T_{surr} , $0.008\ kW/K$ for ΔS_{surr} .

Calculate the rate of heat loss from the compressor.

$$q_{out} = 293 \times 0.008$$
$$= 2.344\ kW$$

Therefore, rate of heat loss from the compressor is $\boxed{2.344\ kW}$

Step 2 of 3

(b)

From the properties of saturated refrigerant – 134a pressure tables, at $P_1 = 200\ kPa$,

$x_1 = 1(\text{Saturated vapor})$

Specific volume of $134a$, $v_1 = 0.0998\ m^3/kg$

Enthalpy of $R-134a$, $h_1 = 244.46\ kJ/kg$

Entropy of $R-134a$, $s_1 = 0.937\ kJ/kgK$

Calculate the Mass flow rate of refrigerant by using the following equation:

$$\dot{m} = \frac{\dot{V}_1}{v_1}$$

Here, $\dot{V}_1$ is the volume flow rate.

Substitute $0.03\ m^3/s$ for $\dot{V}_1$, $0.0998\ m^3/kg$ for v_1

$$\dot{m} = \frac{0.03}{0.0998}$$
$$= 0.3\ kg/s$$

Step 3 of 3

From the energy balance on compressor, write the following:

$$W_{in} - q_{out} = \dot{m}(h_2 - h_1)$$

Here, the power input to the compressor is W_{in}

Substitute, 10 kW for W_{in} , $0.3\ kg/s$ for $\dot{m}$, $2.344\ kW$ for q_{out} , $269.99\ kJ/kg$ for h_2

$$10 - 2.344 = 0.3 \times (h_2 - 244.46)$$
$$h_2 = 269.99\ kJ/kg$$

From properties of super-heated refrigerant -134a tables, calculate the entropy and temperature at $P_2 = 700\ kPa$, $h_2 = 269.99\ kJ/kg$ by interpolating the entropy and temperature at $h_2 = 268.45\ kJ/kg$ and $h_2 = 278.57\ kJ/kg$.

$$s_2 = s_{h_2=268.45\ kJ/kg} + \left(s_{h_2=278.57\ kJ/kg} - s_{h_2=268.45\ kJ/kg}\right)\left(\frac{269.99-268.45}{278.57-268.45}\right)$$
$$= 0.9313 + (0.9641 - 0.9313)(0.1521)$$
$$= 0.9313 + 0.00451737$$
$$= 0.93581\ kJ/kg \cdot K$$

$$T_2 = T_{h_2=268.45\ kJ/kg} + \left(T_{h_2=278.57\ kJ/kg} - T_{h_2=268.45\ kJ/kg}\right)\left(\frac{269.99-268.45}{278.57-268.45}\right)$$
$$= 30 + (40 - 30)(0.1521)$$
$$= 30 + 1.521$$
$$= 31.521°C$$

Enthalpy, $h_2 = 269.94\ kJ/kg$.

Entropy, $s_2 = 0.936\ kJ/kgK$

Therefore, the Final temperature of refrigerant, T_2 is $\boxed{31.5°C}$.

(c)

Calculate the Rate of entropy generation by using the following equation:

$$\dot{S}_{gen} = \dot{\Delta}S_{R-134a} + \dot{\Delta}S_{surr}$$
$$= \dot{m}(s_2 - s_1) + \dot{\Delta}S_{surr}$$

Substitute, $0.3\ kg/s$ for $\dot{m}$, $0.936\ kJ/kgK$ for s_2 , $0.937\ kJ/kgK$ for s_1 , $0.008\ kW/K$ for ΔS_{surr} .

$$\dot{S}_{gen} = 0.3 \times (0.936 - 0.937) + 0.008$$
$$= 0.0077\ kW/K$$

Therefore, the Entropy generation, $\dot{S}_{gen}$ is $\boxed{0.0075\ kW/K}$.

Question: 8.178P

Air at 500 kPa and 400 K enters an adiabatic nozzle at a velocity of 30 m/s and leaves at 300 kPa and 350 K. Using variable specific heats, determine (*a*) the isentropic efficiency, (*b*) the exit velocity, and (*c*) the entropy generation.

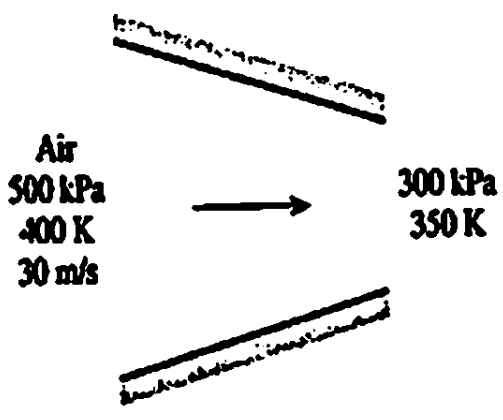
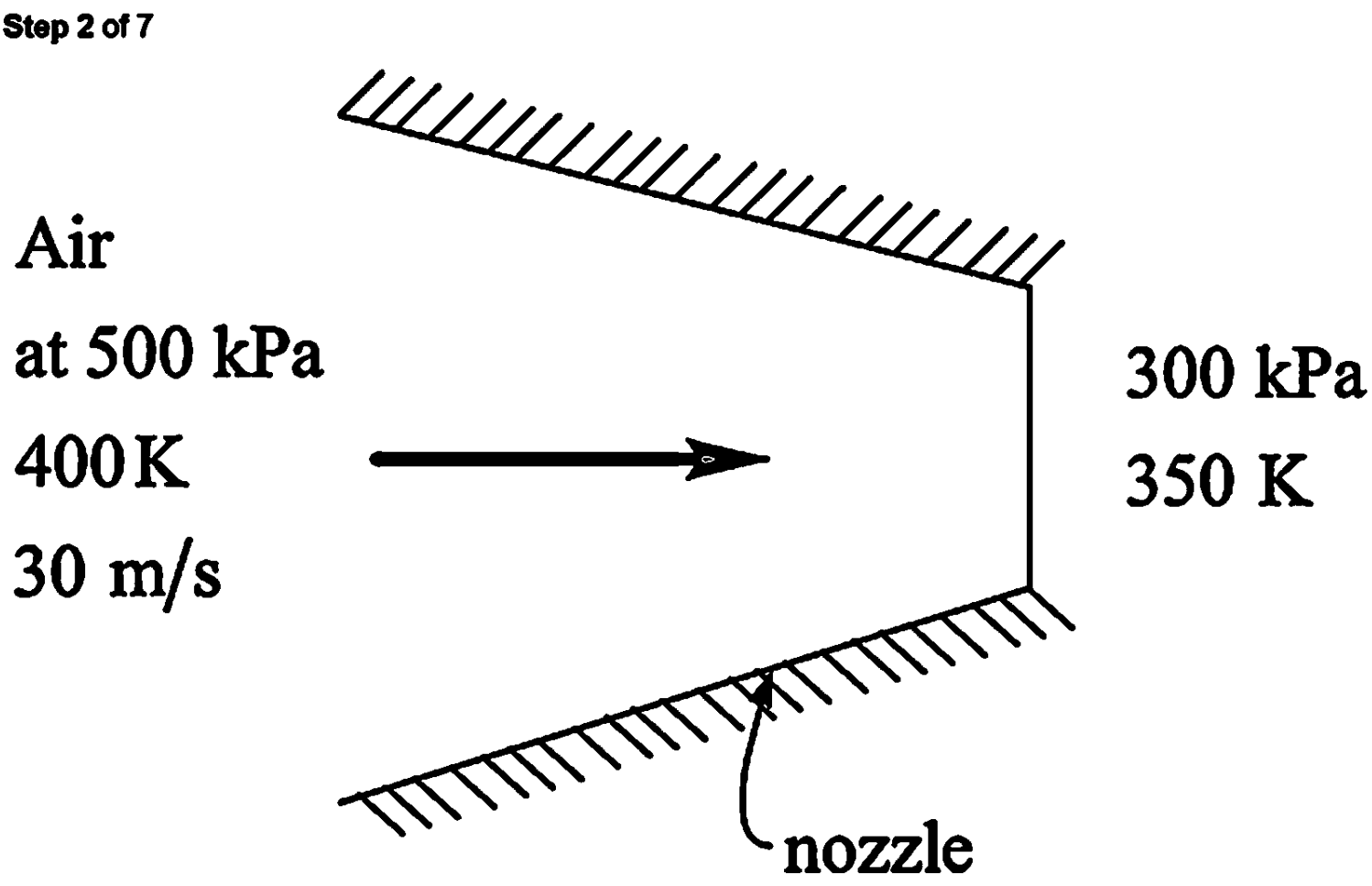


FIGURE P8–178

Step 1 of 7

Flow through the nozzle is as shown below.

Step 2 of 7



Step 3 of 7

The gas constant of air $R = 0.2870\ \text{kJ/kg}\cdot\text{K}$

From the ideal-gas properties of the air table, at inlet temperature $T_1 = 400\ \text{K}$

$h_1 = 400.98\ \text{kJ/kg}$

$s_1^\circ = 1.99194\ \text{kJ/kg}\cdot\text{K}$

$P_{r1} = 3.806$

Similarly From the ideal-gas properties of the air table, at outlet temperature $T_2 = 350\ \text{K}$

$h_2 = 350.49\ \text{kJ/kg}$

$s_2^\circ = 1.85708\ \text{kJ/kg}\cdot\text{K}$

Using the isentropic relation to process 1-2

$$P_{r2} = \frac{P_2}{P_1}(P_{r1})$$
$$= \frac{300}{500}(3.806)$$
$$= 2.2836$$

At $P_{r2} = 2.2836$ from the ideal-gas properties of the air table, at outlet temperature $T_2 = 350\ \text{K}$ the enthalpy is $h_{2s} = 346.31\ \text{kJ/kg}$

Step 4 of 7

From the energy balance equation the actual velocity at the exit is given by

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$400.98(\text{kJ/kg}) + \frac{(30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 350.49(\text{kJ/kg}) + \frac{V_2^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$400.98 + 0.45 = 350.49 + \frac{V_2^2}{2}\left(\frac{1}{1000}\right)$$

$$50.94 = \frac{V_2^2}{2}\left(\frac{1}{1000}\right)$$

$$V_2^2 = 101880$$

$$V_2 = 319.186\ \text{m/s}$$

Step 5 of 7

Using the energy balance equation for the isentropic process the velocity at the exit is given by

$$h_1 + \frac{V_1^2}{2} = h_{2s} + \frac{V_{2s}^2}{2}$$

$$400.98(\text{kJ/kg}) + \frac{(30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 346.31\ (\text{kJ/kg}) + \frac{V_{2s}^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$400.98 + 0.45 = 346.31 + \frac{V_{2s}^2}{2}\left(\frac{1}{1000}\right)$$

$$55.12 = \frac{V_{2s}^2}{2}\left(\frac{1}{1000}\right)$$

$$V_{2s}^2 = 110240$$

$$V_{2s} = 332.024\ \text{m/s}$$

Step 6 of 7

a)

The isentropic efficiency of the nozzle is given by

$$\eta_{nozz} = \frac{V_2^2}{V_{2s}^2}$$
$$= \frac{(319.186)^2}{(332.024)^2}$$
$$= 0.9242$$

Therefore, the isentropic efficiency of the nozzle is $\boxed{\eta = 92.42\%}$

b)

The actual velocity at the exit is $\boxed{V_2 = 319.186\ \text{m/s}}$

Step 7 of 7

c)

The entropy generation for variable specific heat treatment process is given by

$$s_{gen} = s_2^\circ - s_1^0 - R\ln\frac{P_2}{P_1}$$
$$= 1.85708 - 1.99194 - 0.2870\ln\frac{300}{500}$$
$$= 0.0117\ \text{kJ/kg}\cdot\text{K}$$

Therefore, the entropy generation for variable specific heat treatment process is $\boxed{s_{gen} = 0.0117\ \text{kJ/kg}\cdot\text{K}}$

Question: 8.179P pg 1

An insulated tank containing 0.2 m3 of saturated water vapor at 350 kPa is connected to an initially evacuated, insulated piston-cylinder device. The mass of the piston is such that a pressure of 200 kPa is required to raise it. Now the valve is opened slightly, and part of the steam flows to the cylinder, raising the piston. This process continues until the pressure in the tank drops to 200 kPa. Assuming the steam that remains in the tank to have undergone a reversible adiabatic process, determine the final temperature (*a*) in the rigid tank and (*b*) in the cylinder.

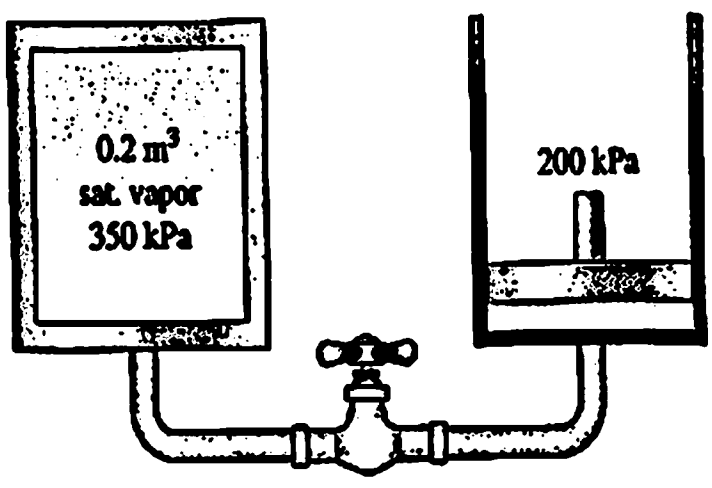


FIGURE P8–179

Step 1 of 8

(a)

The initial pressure inside the tank is 350 kPa.

Use saturated water - Pressure tables to determine the corresponding values of steam at a pressure of 350 kPa.

Specific volume at saturated vapor state, $v_{1,\text{tank}} = 0.52422 \text{ m}^3/\text{kg}$.

Internal energy at saturated vapor state, $u_{1,\text{tank}} = 2548.5 \text{ kJ/kg}$.

Entropy at saturated vapor state, $s_{1,\text{tank}} = 6.9402 \text{ kJ/kg}\cdot\text{K}$.

Step 2 of 8

The final pressure inside the tank is 200 kPa.

Use saturated water - Pressure tables to determine the corresponding values of steam at a pressure of 200 kPa.

Saturated temperature, $T_2 = 120.21°\text{C}$

Specific volume at saturated liquid state, $v_f = 0.001061 \text{ m}^3/\text{kg}$.

Specific volume at saturated vapor state, $v_g = 0.88578 \text{ m}^3/\text{kg}$.

Internal energy at saturated liquid state, $u_f = 504.50 \text{ kJ/kg}$.

Internal energy for evaporate of water, $u_{fg} = 2024.6 \text{ kJ/kg}$.

Entropy at saturated liquid state, $s_f = 1.5302 \text{ kJ/kg}\cdot\text{K}$.

Entropy for evaporate of water, $s_{fg} = 5.5968 \text{ kJ/kg}\cdot\text{K}$.

Therefore, the final temperature of steam inside the rigid tank is $\boxed{120.21°\text{C}}$.

Step 3 of 8

Steam has undergone reversible adiabatic process. Therefore, entropy of steam remains constant.

$$\begin{aligned} s_{1,\text{tank}} &= s_{2,\text{tank}} \\ &= s_f + x s_{fg} \end{aligned}$$

Here, $s_{2,\text{tank}}$ is the entropy of steam inside the rigid tank at state 2, and x is the dryness fraction of steam at state 2.

Substitute $6.9402 \text{ kJ/kg}\cdot\text{K}$ for $s_{1,\text{tank}}$, $1.5302 \text{ kJ/kg}\cdot\text{K}$ for s_f , $5.5968 \text{ kJ/kg}\cdot\text{K}$ and for s_{fg} .

$$\begin{aligned} 6.9402 &= 1.5302 + x \times 5.5968 \\ 5.5968x &= 5.41 \\ x &= 0.9666 \end{aligned}$$

Step 4 of 8

Calculate the initial mass of steam in the rigid tank using the equation,

$$m_{1,\text{tank}} = \frac{V}{v_{1,\text{tank}}}$$

Here, V is the volume of the rigid tank.

Substitute 0.2 m^3 for V, and $0.52422 \text{ m}^3/\text{kg}$ for $v_{1,\text{tank}}$.

$$\begin{aligned} m_{1,\text{tank}} &= \frac{0.2}{0.52422} \\ &= 0.3815 \text{ kg} \end{aligned}$$

Step 5 of 8

Calculate the final specific volume of steam in the rigid tank using the equation,

$$v_{2,\text{tank}} = v_f + x\left(v_g - v_f\right)$$

Substitute $0.001061 \text{ m}^3/\text{kg}$ for v_f , $0.88578 \text{ m}^3/\text{kg}$ for v_g .

$$\begin{aligned} v_{2,\text{tank}} &= 0.001061 + 0.9666\left(0.88578 - 0.001061\right) \\ &= 0.85623 \text{ m}^3/\text{kg} \end{aligned}$$

Step 6 of 8

Calculate the final mass of steam in the rigid tank using the equation,

$$m_{2,\text{tank}} = \frac{V}{v_{2,\text{tank}}}$$

Step 6 of 8

Calculate the final mass of steam in the rigid tank using the equation,

$$m_{2,\,\text{tank}} = \frac{V}{v_{2,\,\text{tank}}}$$

Substitute $0.2\ \text{m}^3$ for V, and $0.85623\ \text{m}^3/\text{kg}$ for $v_{2,\,\text{tank}}$.

$$\begin{aligned} m_{2,\,\text{tank}} &= \frac{0.2}{0.85623} \\ &= 0.2336\ \text{kg} \end{aligned}$$

Step 7 of 8

(b)

Calculate the mass of steam left the rigid tank using the equation,

$$m_{\text{cylinder}} = m_{1,\,\text{tank}} - m_{2,\,\text{tank}}$$

Substitute 0.3815 kg for $m_{1,\,\text{tank}}$, and 0.2336 kg for $m_{2,\,\text{tank}}$.

$$\begin{aligned} m_{\text{cylinder}} &= 0.3815 - 0.2336 \\ &= 0.1479\ \text{kg} \end{aligned}$$

Apply energy balance equation for the whole system (Consider rigid tank, and piston-cylinder device as system).

$$m_{1,\,\text{tank}} u_{1,\,\text{tank}} = m_{2,\,\text{tank}} u_{2,\,\text{tank}} + m_{\text{cylinder}} h_{\text{cylinder}} \quad \cdots\cdots (1)$$

Calculate the final internal energy of steam inside rigid tank using the equation,

$$u_{2,\,\text{tank}} = u_f + x u_{fg}$$

Substitute $504.50\ \text{kJ/kg}$ for u_f, 0.9666 for x, and $2024.6\ \text{kJ/kg}$ for u_{fg}.

$$\begin{aligned} u_{2,\,\text{tank}} &= 504.50 + 0.9666 \times 2024.6 \\ &= 2461.48\ \text{kJ/kg} \end{aligned}$$

Substitute 0.3815 kg for $m_{1,\,\text{tank}}$, $2548.5\ \text{kJ/kg}$ for $u_{1,\,\text{tank}}$, 0.2336 kg for $m_{2,\,\text{tank}}$, $2461.48\ \text{kJ/kg}$ for $u_{2,\,\text{tank}}$, and 0.1479 kg for m_{cylinder} in equation (1).

$$\begin{aligned} m_{1,\,\text{tank}} u_{1,\,\text{tank}} &= m_{2,\,\text{tank}} u_{2,\,\text{tank}} + m_{\text{cylinder}} h_{\text{cylinder}} \\ 0.3815 \times 2548.5 &= 0.2336 \times 2461.48 + 0.1479 \times h_{\text{cylinder}} \\ 0.1479 \times h_{\text{cylinder}} &= 397.2510 \\ h_{\text{cylinder}} &= 2685.94\ \text{kJ/kg} \end{aligned}$$

Step 8 of 8

Use saturated water - Pressure tables to determine the corresponding values of steam at a pressure of 200 kPa.

Enthalpy at saturated liquid state, $h_f = 504.71\ \text{kJ/kg}$.

Enthalpy at saturated vapor state, $h_g = 2706.3\ \text{kJ/kg}$.

Since, $h_f < h_{\text{cylinder}} < h_g$. The steam insider the piston-cylinder device is at a pressure of 200 kPa.

Use saturated water - Pressure tables to determine the corresponding values of steam at a pressure of 200 kPa.

Saturated temperature, $T_{\text{cylinder}} = 120.21°\text{C}$

Therefore, the temperature of steam inside the piston-cylinder device is $\boxed{120.21°\text{C}}$.

Question: 8.180P

3-kg of helium gas at 100 kPa and 27°C are adiabatically compressed to 900 kPa. If the isentropic compression efficiency is 80 percent, determine the required work input *M* and the final temperature of helium.

Step 1 of 4

Given data:

Mass of helium compressed is $m = 3\ \text{kg}$

Initial pressure is, $P_1 = 100\ \text{kPa}$

Initial temperature is, $T_1 = 27°\text{C}$

$= 300\ \text{K}$

Final pressure is, $P_2 = 900\ \text{kPa}$

The isentropic efficiency is, $\eta_{isen} = 80\ \%$

$= 0.8$

Step 2 of 4

From ideal gas specific heats of various common gases

For helium, $c_v = 3.1156\ \text{kJ/kg°K}$ and $k = 1.667$

We know that

The temperature ratio is given by

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = 300 \times \left(\frac{900}{100}\right)^{\frac{1.667-1}{1.667}}$$

$$T_{2s} = 300 \times (9)^{\frac{0.667}{1.667}}$$

$$T_{2s} = 300 \times (9)^{0.4}$$

$$T_{2s} = 300 \times 2.408$$

$$T_{2s} = 722.4\ \text{K}$$

Step 3 of 4

Now we know that

The adiabatic work done is given by

$$W_{in,s} = m \times c_v \times (T_{2s} - T_1)$$

$$W_{in,s} = 3 \times 3.1156 \times (722.4 - 300)$$

$$W_{in,s} = 3 \times 3.1156 \times 422.4$$

$$W_{in,s} = 3948.088\ \text{kJ}$$

Now we know that

The isentropic efficiency of compressor is given by

$$\eta_{isen} = \frac{W_{in,s}}{W_{in,act}}$$

$$W_{in,act} = \frac{W_{in,s}}{\eta_{isen}}$$

$$W_{in,act} = \frac{3948.088}{0.8}$$

$$\boxed{W_{in,act} = 4935.11\ \text{kJ}}$$

Step 4 of 4

We know that

The actual work done is also given by

$$W_{in,act} = m \times c_v \times (T_2 - T_1)$$

$$T_2 - T_1 = \frac{W_{in,act}}{m \times c_v}$$

$$T_2 = T_1 + \frac{W_{in,act}}{m \times c_v}$$

$$T_2 = 300 + \frac{4935.11}{3 \times 3.1156}$$

$$T_2 = 300 + \frac{4935.11}{9.3468}$$

$$T_2 = 300 + 527.99$$

$$T_2 = 827.99\ \text{K}$$

$$\boxed{T_2 = 554.99°\text{C}}$$

Therefore the actual work done, $W_{in,act} = 4935.11\ \text{kJ}$

The actual final temperature, $T_2 = 554.99°\text{C}$

Question: 8.181P

Refrigerant-134a at 600 kPa and 100°C undergoes areversible, isothermal expansion to 200 kPa in a steady-flowdevice with one inlet and one outlet. Determine the powerproduced by this device and the rate of heat transfer when theflow rate through the device is 1 kg/s.

Step 1 of 3

Given data:

Mass of helium compressed is $\dot{m} = 1$ kg/s

Pressure at inlet of the device is, $P_1 = 600$ kPa

Temperature at inlet of the device is, $T_1 = 100°C$

$= 373$ K

Pressure at exit of the device is, $P_2 = 200$ kPa

Step 2 of 3

From properties of superheated refrigerant-134a

At pressure $P_1 = 600$ kPa and temperature $T_1 = 100°C$

$h_1 = 339.47$ kJ/kg

$s_1 = 1.1536$ kJ/kg°K

As the process is isothermal, temperature at inlet and exit will be same.

$T_2 = T_1 = 100°C$

Similarly, from properties of superheated refrigerant-134a

At pressure $P_2 = 200$ kPa and temperature $T_2 = 100°C$

$h_2 = 343.6$ kJ/kg

$s_2 = 1.2512$ kJ/kg°K

We know that

From entropy equation the heat input is given by

$$Q_{in} = m\int_1^2 T\, ds$$

$$Q_{in} = m \times T\ (s_2 - s_1)$$

$$Q_{in} = 1 \times 373 \times (1.2512 - 1.1536)$$

$$Q_{in} = 1 \times 373 \times 0.0976$$

$$\boxed{Q_{in} = 36.405 \text{ kW}}$$

Step 3 of 3

Now we know that

The energy balance given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

As the flow is steady, $\Delta E_{sys} = 0$.

Therefore,

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$Q_{in} + \dot{m}h_1 = W_{out} + \dot{m}h_2$$

$$W_{out} = Q_{in} + \dot{m}h_1 - \dot{m}h_2$$

$$W_{out} = 36.405 + (1 \times 339.47) - (1 \times 343.6)$$

$$W_{out} = 36.405 + 339.47 - 343.6$$

$$\boxed{W_{out} = 32.275 \text{ kW}}$$

Therefore the heat transfer, $Q_{in} = 36.405$ kW

The work done, $W_{out} = 32.275$ kW

Question: 8.182P

An inventor claims to have invented an adiabatic steady-flow device with a single inlet-outlet that produces 100 kW when expanding 1 kg/s of air from 900 kPa and 300°C to 100 kPa. Is this claim valid?

Step 1 of 3

Given data:

Mass of air is $m = 1\ \text{kg/s}$

Work done by the device is, $W_{out,act} = 100\ \text{kW}$

Pressure at inlet of the device is, $P_1 = 900\ \text{kPa}$

Temperature at inlet of the device is, $T_1 = 300\ ^\circ\text{C}$

$= 573\ \text{K}$

Pressure at exit of the device is, $P_2 = 100\ \text{kPa}$

Step 2 of 3

From ideal gas specific heats of various common gases

$T_{avg} = 400\ \text{K}$ (Assumed)

$c_p = 1.013\ \text{kJ/kg°K}$

$k = 1.395$

We know that

The temperature ratio is given by

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = 573 \times \left(\frac{100}{900}\right)^{\frac{1.395-1}{1.395}}$$

$$T_{2s} = 573 \times (0.111)^{\frac{0.395}{1.395}}$$

$$T_{2s} = 573 \times (0.111)^{0.283}$$

$$T_{2s} = 573 \times 0.5368$$

$$T_{2s} = 307.586\ \text{K}$$

Step 3 of 3

Now we that

The energy balance given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

As the flow is steady, $\Delta E_{sys} = 0$. So,

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$mh_1 = W_{out} + mh_2$$

Therefore the isentropic work done is given by

$$W_{out,s} = mh_1 - mh_{2s}$$

$$W_{out,s} = m(h_1 - h_{2s})$$

$$W_{out,s} = m \times c_p \times (T_1 - T_{2s})$$

$$W_{out,s} = 1 \times 1.013 \times (573 - 307.586)$$

$$W_{out,s} = 1 \times 1.013 \times 265.414$$

$$W_{out,s} = 268.864\ \text{kW}$$

$$\boxed{W_{out,s} > W_{out,act}}$$

As actual work done is less than isentropic work done (max, possible work done), the claim is valid

Question: 8.183P

You are to expand a gas adiabatically from 3 MPa and 300°C to 80 kPa in a piston-cylinder device. Which of the two choices - air with an isentropic expansion efficiency of 90 percent or neon with an isentropic expansion efficiency of 80 percent - will produce the most work?

Step 1 of 4

Given

Mass of helium compressed is $m = 3\ \text{kg}$

Initial pressure is, $P_1 = 3\ \text{MPa}$

$= 3000\ \text{kPa}$

Initial temperature is, $T_1 = 300°\text{C}$

$= 573\ \text{K}$

Final pressure is, $P_2 = 80\ \text{kPa}$

The isentropic efficiency while using air is, $\eta_{isen,a} = 0.9$

The isentropic efficiency while using neon is, $\eta_{isen,n} = 0.8$

Step 2 of 4

From ideal gas specific heats of various common gases

For air

$c_{v,a} = 0.718\ \text{kJ/kg°K}$

$k_a = 1.4$

Similarly, from ideal gas specific heats of various common gases

For neon

$c_{v,n} = 0.6179\ \text{kJ/kg°K}$

$k_n = 1.667$

Now we know that

The temperature ratio is given by

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s,a} = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k_a-1}{k_a}}$$

$$T_{2s,a} = 573 \times \left(\frac{80}{3000}\right)^{\frac{1.4-1}{1.4}}$$

$$T_{2s,a} = 573 \times (0.0266)^{\frac{0.4}{1.4}}$$

$$T_{2s,a} = 573 \times (0.0266)^{0.2857}$$

$$T_{2s,a} = 573 \times 0.3548$$

$$T_{2s,a} = 203.3\ \text{K}$$

Step 3 of 4

Now we know that

The adiabatic work done by air is given by

$$w_{s,a} = c_{v,a} \times (T_1 - T_{2s,a})$$

$$w_{s,a} = 0.718 \times (573 - 203.3)$$

$$w_{s,a} = 0.718 \times 369.7$$

$$w_{s,a} = 265.445\ \text{kJ/kg}$$

The actual work done by air is given by

$$w_{act,a} = \eta_{isen,a} \times w_{s,a}$$

$$w_{act,a} = 0.9 \times 265.445$$

$$w_{act,a} = 238.9\ \text{kJ/kg}$$

Similarly, we know that

The adiabatic final temperature of neon is given by

$$T_{2s,n} = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k_n-1}{k_n}}$$

$$T_{2s,n} = 573 \times \left(\frac{80}{3000}\right)^{\frac{1.667-1}{1.667}}$$

$$T_{2s,n} = 573 \times (0.0266)^{\frac{0.667}{1.667}}$$

$$T_{2s,n} = 573 \times (0.0266)^{0.4}$$

$$T_{2s,n} = 573 \times 0.2344$$

$$T_{2s,n} = 134.31\ \text{K}$$

Step 4 of 4

The adiabatic work done by neon is given by

$$w_{s,n} = c_{v,n} \times (T_1 - T_{2s,n})$$

$$w_{s,n} = 0.6179 \times (573 - 134.31)$$

$$w_{s,n} = 0.6179 \times 438.69$$

$$w_{s,n} = 271.067\ \text{kJ/kg}$$

The actual work done by air is neon by

$$w_{act,n} = \eta_{isen,n} \times w_{s,n}$$

$$w_{act,n} = 0.8 \times 271.067$$

$$w_{act,n} = 216.854\ \text{kJ/kg}$$

$$\boxed{w_{act,n} < w_{act,a}}$$

Therefore the actual work done by air is greater than the actual work done by neon.

Question: 8.184P

An adiabatic capillary tube is used in some refrigeration systems to drop the pressure of the refrigerant from the condenser level to the evaporator level. R-134a enters the capillary tube as a saturated liquid at 50°C, and leaves at -12°C. Determine the rate of entropy generation in the capillary tube for a mass flow rate of 0.2 kg/s.



FIGURE P8–184

Step 1 of 4

Apply the entropy balance on the heat exchanger,

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta \dot{S}_{system}$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) \quad \text{..... (1)}$$

Step 2 of 4

From the properties table of the refrigerant R-134a,

Calculate the entropy and enthalpy at $T_1 = 50°C$, $x_1 = 0$ by interpolating the enthalpy and entropy values at $T = 48°C$ and $T = 52°C$

$$h_1 = h_{T=48°C} + (h_{T=52°C} - h_{T=48°C})\left(\frac{50-48}{52-48}\right)$$
$$= 120.39 + (126.59 - 120.39)(0.5)$$
$$= 120.39 + 3.1$$
$$= 123.49 \text{ kJ/kg}$$

$$s_1 = s_{T=48°C}^{\bullet} + (s_{T=52°C}^{\bullet} - s_{T=48°C}^{\bullet})\left(\frac{50-48}{52-48}\right)$$
$$= 0.43242 + (0.45126 - 0.43242)(0.5)$$
$$= 0.43242 + 0.00942$$
$$= 0.44184 \text{ kJ/kg} \cdot \text{K}$$

Step 3 of 4

From the properties table of the refrigerant R-134a,

At $T_1 = -12°C$,

$h_{fg} = 207.38$ kJ/kg.K,

$h_f = 35.92$ kJ/kg.K

$s_f = 0.14504$ kJ/kg·K

$s_{fg} = 0.79406$ kJ/kg·K

A capillary device has a constant enthalpy process $h_1 = h_2$,

$$h_2 = h_1$$
$$= h_f + x_2 h_{fg}$$

From the above equation calculate the Quality at final state as follows:

$$x_2 = \frac{h_2 - h_f}{f_{fg}}$$

Substitute, 123.49 kJ/kg·K for h_1, 0.44184 kJ/kg·K for s_1, 207.38 kJ/kg.K for h_{fg}, 35.92 kJ/kg.K for h_f

$$x_2 = \frac{123.5 - 35.92}{207.38}$$
$$= \frac{87.58}{207.38}$$
$$= 0.4223$$

Step 4 of 4

Calculate Entropy at final state by using the following equation:

$$s_2 = s_f + x_2 s_{fg}$$

Substitute, 0.14504 kJ/kg·K for s_f, 0.79406 kJ/kg·K for s_{fg}

$$s_2 = 0.14504 + (0.4233)(0.794)$$
$$= 0.14504 + 0.3361$$
$$= 0.4811 \text{ kJ/kg} \cdot \text{K}$$

Substitute, in the above entropy relation in equation (1), we have

$$\dot{S}_{gen} = 0.2(0.4811 - 0.44184)$$
$$= 0.2 \times 0.03926$$
$$= 0.00785 \text{ kW/K}$$

Therefore, the rate of entropy generation in capillary tube, $\dot{S}_{gen}$ is $\boxed{0.00785 \text{ kW/K}}$

Question: 8.185P

Air is compressed steadily by a compressor from 100 kPa and 17°C to 700 kPa at a rate of 5 kg/min. Determine the minimum power input required if the process is adiabatic and (*b*) isothermal. Assume air to be an ideal gas with variable specific heats, and neglect the changes in kinetic and potential energies.

Step 1 of 8

Given data:

Pressure at compressor inlet is, $P_1 = 100\ \text{kPa}$

Temperature at compressor inlet is, $T_1 = 17\ ^\circ\text{C} = 290\ \text{K}$

Pressure at compressor exit is, $P_2 = 700\ \text{kPa}$

Mass flow rate of air is, $m = 5\ \text{kg/min} = 0.083\ \text{kg/s}$

Step 2 of 8

From table of ideal gas specific heats of various common gases '

Gas constant of air is

$R = 0.287\ \text{kJ/kg°K}$

From table of 'ideal gas properties of air'

At temperature $T_1 = 290\ \text{K}$ properties of air are

$P_{r1} = 1.2311$

$h_1 = 290.16\ \text{kJ/kg}$

Step 3 of 8

(a) Final pressure of adiabatic process is given by

$$P_{r2} = P_{r1} \times \left(\frac{P_2}{P_1}\right)$$

$$= 1.2311 \times \left(\frac{700}{100}\right)$$

$$= 1.2311 \times 7$$

$$= 8.6177$$

Step 4 of 8

From table of 'ideal gas properties of air'

At pressure ratio $P_{r2} = 8.6177$, final temperature is interpolated as

$$\frac{T_2 - 500}{510 - 500} = \frac{8.6177 - 8.411}{9.031 - 8.411}$$

$$\frac{T_2 - 500}{10} = \frac{0.2067}{0.62}$$

$$\frac{T_2 - 500}{10} = 0.333$$

$$T_2 = 0.333 \times 10 + 500$$

$$= 3.33 + 500$$

$$= 503.3\ \text{K}$$

Step 5 of 8

From table of 'ideal gas properties of air'

At pressure ratio $P_{r2} = 8.6177$, final enthalpy is interpolated as

$$\frac{h_2 - 503.02}{513.32 - 503.02} = \frac{8.6177 - 8.411}{9.031 - 8.411}$$

$$\frac{h_2 - 503.02}{10.3} = \frac{0.2067}{0.62}$$

$$\frac{h_2 - 503.02}{10.3} = 0.333$$

$$h_2 = 0.333 \times 10.3 + 503.02$$

$$= 3.43 + 503.02$$

$$= 506.45\ \text{kJ/kg}$$

Step 6 of 8

Power input is given by

$$W_{in} = m \times (h_2 - h_1)$$

$$= 0.083 \times (506.45 - 290.16)$$

$$= 0.083 \times 216.29$$

$$= 17.952\ \text{kW}$$

$$\boxed{W_{in} = 17.952\ \text{kW}}$$

Step 7 of 8

(b) Energy balance is given by

$$E_{in} - E_{out} = \Delta E_{sys}$$

For steady process, $\Delta E_{sys} = 0$

$$E_{in} - E_{out} = 0$$

$$E_{in} = E_{out}$$

$$W_{in} + mh_1 = Q_{out} + mh_2$$

$$W_{in} = Q_{out} + mh_2 - mh_1$$

$$W_{in} = Q_{out} + m(h_2 - h_1)$$

For ideal gas enthalpy is a function of temperature. As the process is isothermal, change in enthalpy is zero.

$$W_{in} = Q_{out} + 0$$

$$= Q_{out}$$

$$= m \times T \times (s_1 - s_2)$$

$$= -m \times T \times (s_2 - s_1)$$

Step 8 of 8

$$W_{in} = -m \times T \times \left[\left(s_t^0 - s_1^0\right) - R \times \ln\left(\frac{P_2}{P_1}\right)\right]$$

As the process is isothermal, change in entropy is zero.

$$W_{in} = -m \times T \times \left[0 - R \times \ln\left(\frac{P_2}{P_1}\right)\right]$$

$$= -m \times T \times \left[-R \times \ln\left(\frac{P_2}{P_1}\right)\right]$$

$$= m \times T \times R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$= 0.083 \times 290 \times 0.287 \times \ln\left(\frac{700}{100}\right)$$

$$= 0.083 \times 290 \times 0.287 \times \ln 7$$

$$= 0.083 \times 290 \times 0.287 \times 1.946$$

$$= 13.443\text{kW}$$

$$\boxed{W_{in} = 13.443\ \text{kW}}$$

The power input in an adiabatic process is 17.952 kW and the power input in an isothermal process is 13.443 kW

Question: 8.186P

Refrigerant-134a at 100 kPa and - 20°C is compressed by an adiabatic 1.3-kW compressor to an exit state of 800 kPa and 60°C. Neglecting the changes in kinetic and potential energies, determine (*a*) the isentropic efficiency of the compressor, (*b*) the volume flow rate of the refrigerant at the compressor inlet, in L/min, and (*c*) the maximum volume flow rate at the inlet conditions that this adiabatic 0.7-kW compressor can handle without violating the second law.

Step 1 of 8

Given data:

Pressure at compressor inlet is, $P_1 = 100\ \text{kPa}$

Temperature at compressor inlet is, $T_1 = -20\ ^\circ\text{C} = 253\ \text{K}$

Pressure at compressor exit is, $P_2 = 800\ \text{kPa}$

Temperature at compressor exit is, $T_2 = 60\ ^\circ\text{C} = 333\ \text{K}$

Power input to compressor is, $W_{in} = 1.3\ \text{kW}$

Step 2 of 8

From table of superheated refrigerant 134a,

At pressure $P_1 = 100\ \text{kPa}$ and temperature $T_1 = -20\ ^\circ\text{C}$

Properties of refrigerant are

$v_1 = 0.19841\ \text{m}^3/\text{kg}$

$h_1 = 239.5\ \text{kJ/kg}$

$s_1 = 0.9721\ \text{kJ/kg°K}$

From table of superheated refrigerant 134a,

at pressure $P_2 = 800\ \text{kPa}$ and temperature $T_2 = 60\ ^\circ\text{C}$

Properties of refrigerant are

$h_{2a} = 296.81\ \text{kJ/kg}$

Step 3 of 8

From table of superheated refrigerant 134a,

at pressure $P_2 = 800\ \text{kPa}$ and entropy $s_{2s} = s_1 = 0.9721\ \text{kJ/kg°K}$

Enthalpy of refrigerant is interpolated as

$$\frac{h_{2s} - 276.45}{286.69 - 276.45} = \frac{0.9721 - 0.9480}{0.9802 - 0.9480}$$

$$\frac{h_{2s} - 276.45}{10.24} = \frac{0.0241}{0.0322}$$

$$\frac{h_{2s} - 276.45}{10.24} = 0.7484$$

$$h_{2s} = 0.7484 \times 10.24 + 276.45$$

$$= 7.66 + 276.45$$

$$= 284.11\ \text{kJ/kg}$$

Step 4 of 8

(a) Isentropic efficiency of compressor is given by

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$= \frac{284.11 - 239.5}{296.81 - 239.5}$$

$$= \frac{44.61}{57.31}$$

$$= 0.7784$$

$$= 77.84\%$$

$$\boxed{\eta_c = 0.7784 = 77.84\ \%}$$

Step 5 of 8

Actual work input is given by

$$W_{in} = m_a \times (h_{2a} - h_1)$$

Actual mass flow rate of compressor is given by

$$m_a = \frac{W_{in}}{h_{2a} - h_1}$$

$$= \frac{1.3}{296.81 - 239.5}$$

$$= \frac{1.3}{57.31}$$

$$= 0.0227\ \text{kg/s}$$

Step 6 of 8

(b) Volume flow rate at compressor inlet is given by

$$V_1 = m_a \times v_1$$

$$= 0.0227 \times 0.19841$$

$$= 0.0045\ \text{m}^3/\text{s}$$

$$= 270\ \text{L/min}$$

$$\boxed{V_1 = 0.0045\ \text{m}^3/\text{s} = 270\ \text{L/min}}$$

Step 7 of 8

Adiabatic work input is given by

$$W_{in} = m_s \times (h_{2s} - h_1)$$

Adiabatic mass flow rate of compressor is given by

$$m_s = \frac{W_{in}}{h_{2s} - h_1}$$

Given for calculating maximum volume flow rate, $W_{in} = 0.7\ \text{kW}$

$$m_s = \frac{0.7}{284.11 - 239.5}$$

$$= \frac{0.7}{44.61}$$

$$= 0.0157\ \text{kg/s}$$

Step 8 of 8

(c) Maximum volume flow rate at compressor inlet is given by

$$V_{max} = m_s \times v_1$$

$$V_{max} = 0.0157 \times 0.19841$$

$$= 0.00311\ \text{m}^3/\text{s}$$

$$= 186.6\ \text{L/min}$$

$$\boxed{V_{max} = 0.00311\ \text{m}^3/\text{s} = 186.6\ \text{L/min}}$$

The isentropic efficiency of compressor is 77.84 % , the volume flow rate is 270 L/min and the maximum flow rate for the compressor with work input $W_{in} = 0.7\ \text{kW}$ is 186.6 L/min

Question: 8.187P pg 1

An adiabatic air compressor is to be powered by a direct-coupled adiabatic steam turbine that is also driving a generator. Steam enters the turbine at 12.5 MPa and 500°C at a rate of 25 kg/s and exits at 10 kPa and a quality of 0.92. Air enters the compressor at 98 kPa and 295 K at a rate of 10 kg/s and exits at 1 MPa and 620 K. Determine (*a*) the net power delivered to the generator by the turbine and (*b*) the rate of entropy generation within the turbine and the compressor during this process.

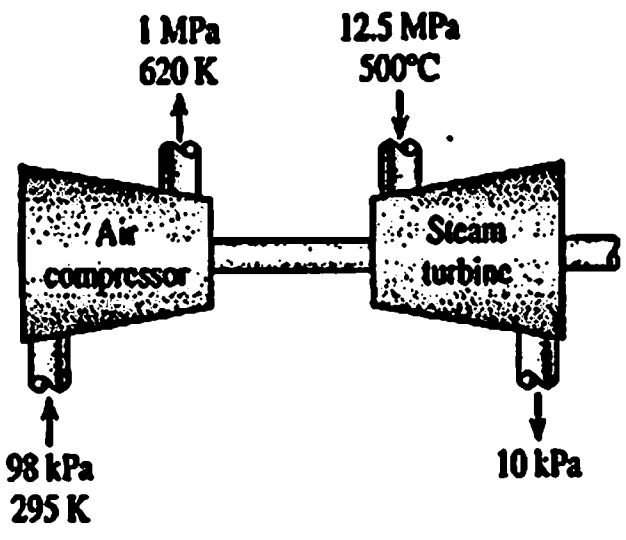


FIGURE P8–187

Step 1 of 9

Given data:

Air entering the compressor at a pressure, $P_1 = 98\ \text{kPa}$

Air entering the compressor at a temperature, $T_1 = 295\ \text{K}$

Mass flow rate of the air in the compressor, $m_1 = 10\ \text{kg/s}$

Compressed air leaving the compressor at a pressure, $P_2 = 1\ \text{MPa}$

Exit temperature of the compressed air, $T_2 = 620\ \text{K}$

Steam entering the turbine at a pressure, $P_3 = 12.5\ \text{MPa}$

Steam entering the turbine at a temperature, $T_3 = 500\ ^\circ\text{C}$

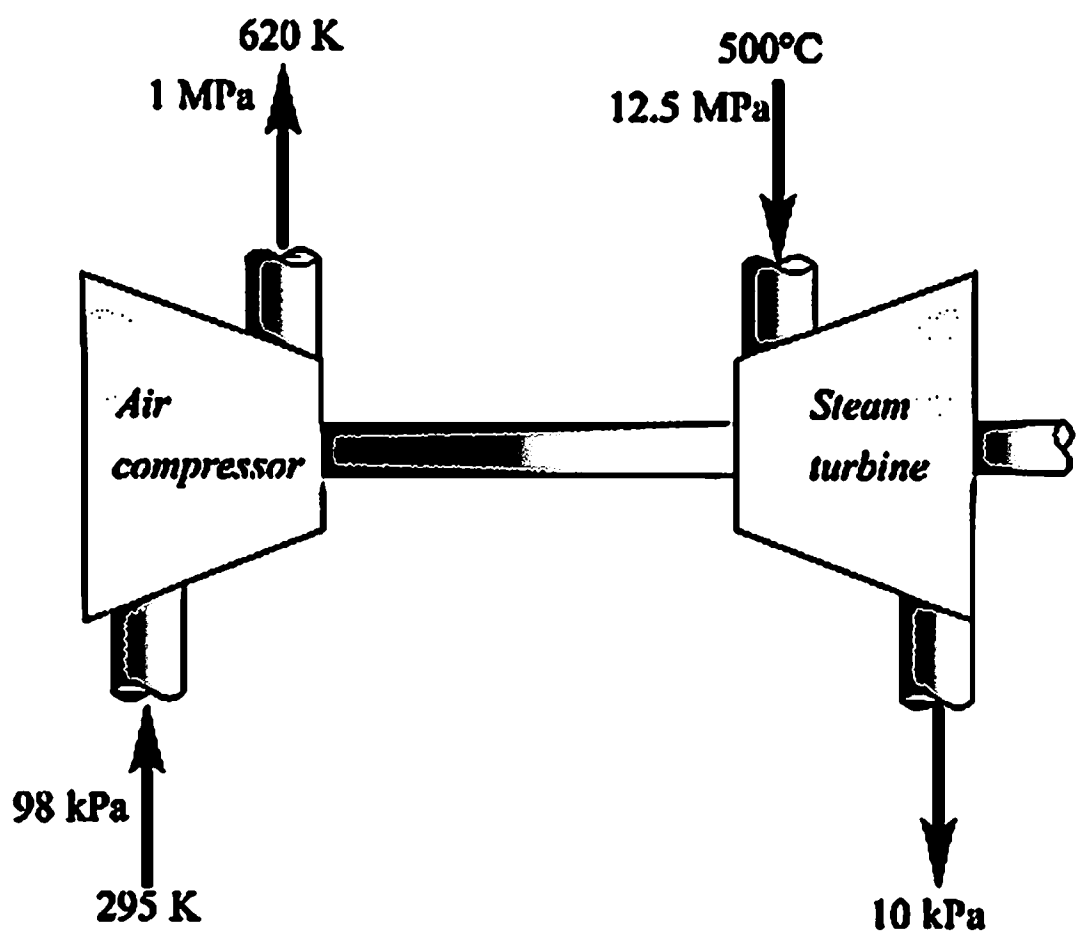
Mass flow rate of the steam, $m_3 = 25\ \text{kg/s}$

Steam exits the turbine at a pressure, $P_4 = 10\ \text{kPa}$

Quality of steam is $x = 0.92$

Step 2 of 9

An adiabatic air compressor double coupled to the turbine is as shown below:



Step 3 of 9

From the ideal-gas properties of air at temperature $T_1 = 295\ \text{K}$, the following values are taken

Enthalpy $h_1 = 295.17\ \text{kJ/kg}$

Entropy $s_1^0 = 1.68515\ \text{kJ/kg}\cdot\text{K}$

Similarly at temperature $T_2 = 620\ \text{K}$, the following values are taken

Enthalpy $h_2 = 628.07\ \text{kJ/kg}$

Entropy $s_2^0 = 2.44356\ \text{kJ/kg}\cdot\text{K}$

In the turbine steam entering at a pressure, $P_3 = 12.5\ \text{MPa}$ maintaining temperature, $T_3 = 500\ ^\circ\text{C}$ the steam is in superheated state at this temperature. Therefore from the superheated water tables at $P_3 = 12.5\ \text{MPa}$ and at temperature $T_3 = 500\ ^\circ\text{C}$ the following values are taken

Enthalpy $h_3 = 3343.6\ \text{kJ/kg}$

Entropy $s_3 = 6.4651\ \text{kJ/kg}\cdot\text{K}$

Step 4 of 9

Similarly at pressure $P_4 = 10\ \text{kPa}$ the steam is in saturated liquid state, therefore from the saturated water tables at $P_4 = 10\ \text{kPa}$ the following values are taken

Enthalpy of Saturated liquid $h_f = 191.81\ \text{kJ/kg}$

Enthalpy of Evaporated liquid $h_{fg} = 2392.1\ \text{kJ/kg}$

Similarly Entropy of Saturated liquid $s_f = 0.6492\ \text{kJ/kg}\cdot\text{K}$

Entropy of Evaporated liquid $s_{fg} = 7.4996\ \text{kJ/kg}\cdot\text{K}$

The enthalpy at state 4 is given by

$$h_4 = h_f + xh_{fg}$$
$$= 191.81 + (0.92)(2392.1)$$
$$= 2392.54\ \text{kJ/kg}$$

The entropy at state 4 is given by

$$s_4 = s_f + xs_{fg}$$
$$= 0.6492 + (0.92)(7.4996)$$
$$= 7.5488\ \text{kJ/kg}\cdot\text{K}$$

Step 5 of 9

Net energy transfer by heat, work and mass is equal to change in the internal, kinetic, potential energies

The energy balance for steady-flow system is

$$E_{in} - E_{out} = \Delta E_{system} = 0\ (steady)$$

Work done on the compressor at the inlet of the system is given by

$$\dot{W}_{comp,in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2$$

energies

The energy balance for steady-flow system is

$$E_{in} - E_{out} = \Delta E_{system} = 0 (steady)$$

Work done on the compressor at the inlet of the system is given by

$$\dot{W}_{comp,in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2$$

$$\dot{W}_{comp,in} = \dot{m}_{air} (h_2 - h_1)$$

$$= 10(628.07 - 295.17)$$

$$= 3328.3 \text{ kW}$$

Step 6 of 9

The work done by the turbine at the outlet of the system is given by

$$E_{in} - E_{out} = 0$$

$$\dot{m}_{steam} h_3 = \dot{W}_{turb,out} + \dot{m}_{steam} h_4$$

$$\dot{W}_{turb,out} = \dot{m}_{steam} (h_3 - h_4)$$

$$= 25(3343.6 - 2392.5)$$

$$= 23777.5 \text{ kW}$$

Step 7 of 9

a)

The net power delivered to the generator by the turbine is given by

$$\dot{W}_{net} = \dot{W}_{turb,out} - \dot{W}_{turb,in}$$

$$= 23777.5 - 3328.3$$

$$= 20449.2 \text{ kW}$$

Therefore, the net power delivered to the generator by the turbine is $\boxed{\dot{W}_{net} = 20449.2 \text{ kW}}$

Step 8 of 9

b)

The rate of entropy generation of air in the compressor is given by

$$\dot{S}_{gen(comp)} = \dot{m}_{air} (s_2 - s_1)$$

$$= \dot{m}_{air} \left(s_2^\circ - s_1^\circ - R \ln \frac{P_2}{P_1} \right)$$

$$= 10 \left(2.44356 - 1.68515 - 0.2871 \ln \frac{1000}{98} \right)$$

$$= 10(2.44356 - 1.68515 - 0.667)$$

$$= 0.92 \text{ kW/K}$$

Step 9 of 9

Similarly the rate of entropy generation of steam in the turbine is

$$\dot{S}_{gen(turb)} = \dot{m}_{steam} (s_4 - s_3)$$

$$= (25)(7.5488 - 6.4651)$$

$$= 27.09 \text{ kW/K}$$

The rate of entropy generation within the turbine and compressor during this process

$$\dot{S}_{gen} = \dot{S}_{gen(comp)} + \dot{S}_{gen(turb)}$$

$$= 0.92 + 27.09$$

$$= 28.012 \text{ kW/K}$$

Therefore, the rate of entropy generation within the turbine and compressor during this process is

$$\boxed{\dot{S}_{gen} = 28.012 \text{ kW/K}}$$

Question: 8.188P

Reconsider Prob. 8-187. Using EES (or other) software, determine the isentropic efficiencies for the compressor and turbine. Then use EES to study how varying the compressor efficiency over the range 0.6 to 0.8 and the turbine efficiency over the range 0.7 to 0.95 affect the net work for the cycle and the entropy generated for the process. Plot the net work as a function of the compressor efficiency for turbine efficiencies of 0.7, 0.8, and 0.9, and discuss your results.

Step 1 of 8

(a)

The initial pressure inside the tank is 500 kPa.

Obtain the properties of saturated water vapor at pressure is 500 kPa from the Table A-5 "saturated water-pressure table" from the text book.

Specific volume at saturated vapor state, $v_{1,\text{tank}} = 0.37483\ \text{m}^3/\text{kg}$.

Internal energy at saturated vapor state, $u_{1,\text{tank}} = 2560.7\ \text{kJ/kg}$.

Entropy at saturated vapor state, $s_{1,\text{tank}} = 6.8207\ \text{kJ/kg}\cdot\text{K}$.

Step 2 of 8

The final pressure inside the tank is 150 kPa.

Obtain the properties of saturated water at pressure is 150 kPa from the Table A-5 "saturated water-pressure table" from the text book.

Saturated temperature, $T_2 = 111.35°\text{C}$

Specific volume at saturated liquid state, $v_f = 0.001053\ \text{m}^3/\text{kg}$.

Specific volume at saturated vapor state, $v_g = 1.1594\ \text{m}^3/\text{kg}$.

Internal energy at saturated liquid state, $u_f = 466.97\ \text{kJ/kg}$.

Internal energy for evaporate of water, $u_{fg} = 2052.3\ \text{kJ/kg}$.

Entropy at saturated liquid state, $s_f = 1.4337\ \text{kJ/kg}\cdot\text{K}$.

Entropy for evaporate of water, $s_{fg} = 5.7894\ \text{kJ/kg}\cdot\text{K}$.

Therefore, the final temperature of the steam inside the rigid tank is, $\boxed{111.35°\text{C}}$.

Step 3 of 8

Steam has undergone reversible adiabatic process. Therefore, entropy of steam remains constant.

$$\begin{aligned} s_{1,\text{tank}} &= s_{2,\text{tank}} \\ &= s_f + x s_{fg} \end{aligned}$$

Here, $s_{2,\text{tank}}$ is the entropy of the steam inside the rigid tank at state 2, and x is the dryness fraction of steam at state 2.

Substitute $6.8207\ \text{kJ/kg}\cdot\text{K}$ for $s_{1,\text{tank}}$, $1.4337\ \text{kJ/kg}\cdot\text{K}$ for s_f , $5.7894\ \text{kJ/kg}\cdot\text{K}$ and for s_{fg} .

$$6.8207 = 1.4337 + x \times 5.7894$$
$$x = 0.9304$$

Step 4 of 8

Calculate the initial mass of steam in the rigid tank using the equation,

$$m_{1,\text{tank}} = \frac{V}{v_{1,\text{tank}}}$$

Here, V is the volume of the rigid tank.

Substitute $0.4\ \text{m}^3$ for V, and $0.37483\ \text{m}^3/\text{kg}$ for $v_{1,\text{tank}}$.

$$\begin{aligned} m_{1,\text{tank}} &= \frac{0.4}{0.37483} \\ &= 1.067\ \text{kg} \end{aligned}$$

Step 5 of 8

Calculate the final specific volume of steam in the rigid tank using the equation,

$$v_{2,\text{tank}} = v_f + x\left(v_g - v_f\right)$$

Substitute $0.001053\ \text{m}^3/\text{kg}$ for v_f , $1.1594\ \text{m}^3/\text{kg}$ for v_g .

$$\begin{aligned} v_{2,\text{tank}} &= 0.001053 + 0.9304(1.1594 - 0.001053) \\ &= 1.079\ \text{m}^3/\text{kg} \end{aligned}$$

Calculate the final mass of steam in the rigid tank using the equation,

$$m_{2,\text{tank}} = \frac{V}{v_{2,\text{tank}}}$$

Substitute $0.4\ \text{m}^3$ for V, and $1.079\ \text{m}^3/\text{kg}$ for $v_{2,\text{tank}}$.

$$\begin{aligned} m_{2,\text{tank}} &= \frac{0.4}{1.079} \\ &= 0.371\ \text{kg} \end{aligned}$$

Step 6 of 8

(b)

Calculate the mass of steam left the rigid tank using the equation,

$$m_{\text{cylinder}} = m_{1,\text{tank}} - m_{2,\text{tank}}$$

Substitute $1.067\ \text{kg}$ for $m_{1,\text{tank}}$, and $0.371\ \text{kg}$ for $m_{2,\text{tank}}$.

$$\begin{aligned} m_{\text{cylinder}} &= 1.067 - 0.371 \\ &= 0.696\ \text{kg} \end{aligned}$$

Apply energy balance equation for the whole system (Consider rigid tank, and piston-cylinder device as system).

$$m_{1,\text{tank}} u_{1,\text{tank}} = m_{2,\text{tank}} u_{2,\text{tank}} + m_{\text{cylinder}} h_{\text{cylinder}} \quad \cdots\cdots (1)$$

Step 7 of 8

Calculate the final internal energy of steam inside rigid tank using the equation,

$$u_{2,\text{tank}} = u_f + x u_{fg}$$

Substitute $466.97\ \text{kJ/kg}$ for u_f , 0.9304 for x, and $2052.3\ \text{kJ/kg}$ for u_{fg} .

$$\begin{aligned} u_{2,\text{tank}} &= 466.97 + 0.9304 \times 2052.3 \\ &= 2376.43\ \text{kJ/kg} \end{aligned}$$

Substitute $1.067\ \text{kg}$ for $m_{1,\text{tank}}$, $2560.7\ \text{kJ/kg}$ for $u_{1,\text{tank}}$, $0.371\ \text{kg}$ for $m_{2,\text{tank}}$, $2376.43\ \text{kJ/kg}$ for $u_{2,\text{tank}}$, and $0.696\ \text{kg}$ for m_{cylinder} in equation (1).

$$m_{1,\text{tank}} u_{1,\text{tank}} = m_{2,\text{tank}} u_{2,\text{tank}} + m_{\text{cylinder}} h_{\text{cylinder}}$$
$$1.067 \times 2560.7 = 0.371 \times 2376.43 + 0.696 \times h_{\text{cylinder}}$$
$$h_{\text{cylinder}} = 2658.924\ \text{kJ/kg}$$

Step 8 of 8

Obtain the properties of saturated water at pressure is 150 kPa from the Table A-5 "saturated water-pressure table" from the text book.

Enthalpy at saturated liquid state, $h_f = 467.13\ \text{kJ/kg}$.

Enthalpy at saturated vapor state, $h_g = 2693.1\ \text{kJ/kg}$.

Since, $h_f < h_{\text{cylinder}} < h_g$. The steam inside the piston-cylinder device is at a pressure of 150 kPa.

Obtain the saturated temperature the water at pressure is 150 kPa from the Table A-5 "saturated water-pressure table" from the text book.

Saturated temperature, $T_{\text{cylinder}} = 111.35°\text{C}$

Therefore, the temperature of the steam inside the piston-cylinder device is, $\boxed{111.35°\text{C}}$.

Question: 8.189P

The explosion of a hot water tank in a school in Spencer, Oklahoma, in 1982 killed 7 people while injuring 33 others. Although the number of such explosions has decreased dramatically since the development of the ASME Pressure Vessel Code, which requires the tanks to be designed to withstand four times the normal operating pressures, they still occur as a result of the failure of the pressure relief valves and thermostats. When a tank filled with a high-pressure and high-temperature liquid ruptures, the sudden drop of the pressure of the liquid to the atmospheric level causes part of the liquid to flash into vapor, and thus to experience a huge rise in its volume. The resulting pressure wave that propagates rapidly can cause considerable damage.

Considering that the pressurized liquid in the tank eventually reaches equilibrium with its surroundings shortly after the explosion, the work that a pressurized liquid would do if allowed to expand reversibly and adiabatically to the pressure of the surroundings can be viewed as the *explosive energy* of the pressurized liquid. Because of the very short time period of the explosion and the apparent calm afterward, the explosion process can be considered to be adiabatic with no changes in kinetic and potential energies and no mixing with the air.

Consider a 80-L hot-water tank that has a working pressure of 0.5 MPa. As a result of some malfunction, the pressure in the tank rises to 2 MPa, at which point the tank explodes. Taking the atmospheric pressure to be 100 kPa and assuming the liquid in the tank to be saturated at the time of explosion, determine the total explosion energy of the tank in terms of the TNT equivalence. (The explosion energy of TNT is about 3250 kJ/kg, and 5 kg of TNT can cause total destruction of unreinforced structures within about a 7-m radius.)

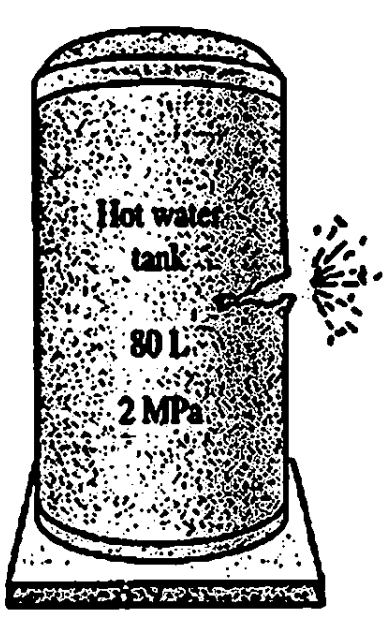


FIGURE P8–189

Step 1 of 6

Given data:

Volume of the water in the tank, $V = 80$ L

Pressure in the hot water tank rises $P_1 = 2$ MPa

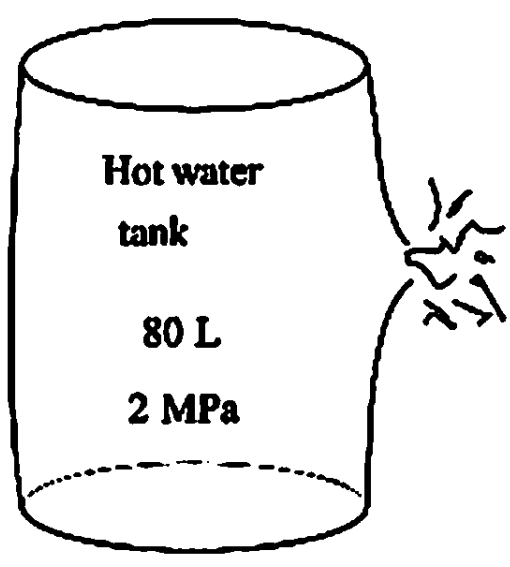
Atmospheric pressure $P_2 = 100$ kPa

The explosion energy of TNT $E_{ex} = 3250$ kJ/kg

Kinetic and potential energy effects can be neglected

Step 2 of 6

The explosion of the hot water tank due to the rise in pressure is as shown below.



Step 3 of 6

From the saturated water-pressure tables, at pressure $P_1 = 2$ MPa the following values are taken

Specific volume, $v_1 = 0.001177\ m^3/kg$

Internal energy, $u_1 = 906.12$ kJ/kg

Entropy, $s_1 = 2.4467$ kJ/kg·K

Similarly at pressure $P_2 = 100$ kPa the following values are taken

Specific volume, $v_f = 0.001043\ m^3/kg$

$u_f = 417.40$ kJ/kg

$u_{fg} = 2088.2$ kJ/kg

$s_f = 1.3028$ kJ/kg·K

$s_{fg} = 6.0562$ kJ/kg·K

Step 4 of 6

Since the system that undergoes a reversible adiabatic process, the relation between the entropies in terms of the dryness fraction of the saturated liquid is given by

$$s_1 = s_2$$

$$s_2 = s_f + x_2 s_{fg}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}}$$

$$= \frac{2.4467 - 1.3028}{6.0562}$$

$$= 0.1889$$

The internal energy at the exit in saturated state is given by

$$u_2 = u_f + x_2 u_{fg}$$

$$= 417.40 + (0.1889)(2088.2)$$

$$= 811.86 \text{ kJ/kg}$$

Step 5 of 6

The work done during the explosion of the tank is given by

$$W_{out} = \text{Energy at TNT} \times \text{mass of TNT}$$

$$= E_{ex} \times m_{TNT}$$

$$= 3250 \times m_{TNT}$$

The mass of the water is given by

$$m = \frac{V}{v_1}$$

$$= \frac{0.08}{0.001177}$$

$$= 67.97 \text{ kg}$$

Step 6 of 6

The explosion process is considered as adiabatic and with negligible kinetic, potential energies. The net energy transfer by work is equal to change in the internal energy.

$$E_{in} - E_{out} = \Delta E_{system}$$

$$-W_{out} = \Delta U$$

$$W_{out} = m(u_1 - u_2)$$

$$= 67.99 \times (906.12 - 811.83)$$

$$E_{ex} m_{TNT} = 6410 \text{ kJ}$$

$$m_{TNT} = \frac{6410 \text{ kJ}}{3250 \text{ kJ/kg}}$$

$$= 1.9723 \text{ kg of TNT}$$

Therefore, the total explosion of the tank in terms of the TNT equivalence is

$$\boxed{m_{TNT} = 1.9723 \text{ kg of TNT}}$$

Question: 8.190P

Using the arguments in the Prob. 8-189, determine the total explosion energy of a 0.35-L canned drink that explodes at a pressure of 1.2 MPa. To how many kg of TNT is this explosion energy equivalent?

Step 1 of 6

Given data:

Volume of the water in the tank, $V = 0.35$ L

Pressure in the hot water tank rises $P_1 = 1.2$ MPa

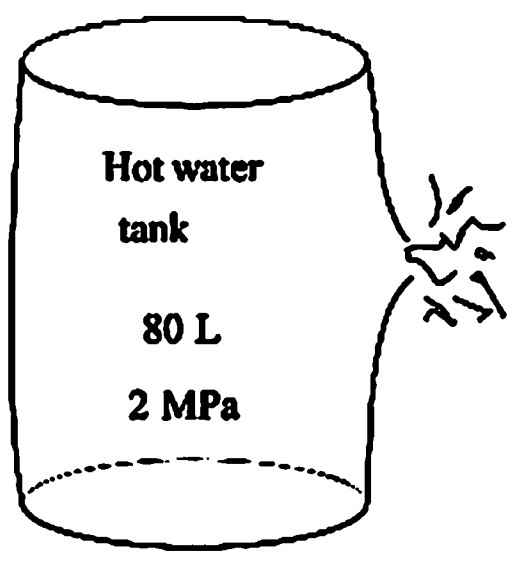
Atmospheric pressure $P_2 = 100$ kPa

The explosion energy of TNT $E_{ex} = 3250$ kJ/kg

Kinetic and potential energy effects can be neglected

Step 2 of 6

The explosion of the hot water tank due to the rise in pressure is as shown below.



Step 3 of 6

From the saturated water-pressure tables, at pressure $P_1 = 1.2$ MPa the following values are taken

Specific volume, $v_1 = 0.001138\ \text{m}^3/\text{kg}$

Internal energy, $u_1 = 796.96$ kJ/kg

Entropy, $s_1 = 2.2159$ kJ/kg·K

Similarly now at pressure $P_2 = 100$ kPa the following values are taken

$v_f = 0.001043\ \text{m}^3/\text{kg}$

$u_f = 417.40$ kJ/kg

$u_{fg} = 2088.2$ kJ/kg

$s_f = 1.3028$ kJ/kg·K

$s_{fg} = 6.0562$ kJ/kg·K

Step 4 of 6

Since the system that undergoes a reversible adiabatic process, the relation between the entropies in terms of the dryness fraction of the saturated liquid is given by

$$s_1 = s_2$$

$$s_2 = s_f + x_2 s_{fg}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}}$$

$$= \frac{2.2159 - 1.3028}{6.0562}$$

$$= 0.15077$$

The internal energy at the exit in saturated state is given by

$$u_2 = u_f + x_2 u_{fg}$$

$$= 417.40 + (0.10577)(2088.2)$$

$$= 732.24\ \text{kJ/kg}$$

Step 5 of 6

The work done during the explosion of the tank is given by

$$W_{out} = \text{Energy at TNT} \times \text{mass of TNT}$$

$$= E_{ex} \times m_{TNT}$$

$$= 3250 \times m_{TNT}$$

The mass of the water is given by

$$m = \frac{V}{v_1}$$

$$= \frac{0.00035}{0.001138}$$

$$= 0.3075\ \text{kg}$$

Step 6 of 6

The explosion process is considered as adiabatic and negligible kinetic and potential energies. The net energy transfer by work is equal to change in the internal energy

$$E_{in} - E_{out} = \Delta E_{system}$$

$$-W_{out} = \Delta U$$

$$W_{out} = m(u_1 - u_2)$$

$$= 0.3075 \times (796.96 - 732.24)$$

$$E_{ex} m_{TNT} = 19.905\ \text{kJ}$$

$$m_{TNT} = \frac{19.905\ \text{kJ}}{3250\ \text{kJ/kg}}$$

$$= 6.125 \times 10^{-3}\ \text{kg of TNT}$$

Therefore, the total explosion of the tank in the terms of the TNT equivalence is

$$\boxed{m_{TNT} = 6.125 \times 10^{-3}\ \text{kg of TNT}}$$

Question: 8.191P

Air is expanded in an adiabatic turbine of 85 percent isentropic efficiency from an inlet state of 2200 kPa and 300°C to an outlet pressure of 200 kPa. Calculate the outlet temperature of air and the work produced by this turbine per unit mass of air.

Step 1 of 6

Given

Isentropic efficiency of the turbine is, $\eta = 85\% = 0.85$

Inlet temperature of air is, $T_1 = 300°C = 573\ K$

Inlet pressure of air is, $P_1 = 2200\ kPa$

Exit pressure of air is, $P_2 = 200\ kPa$

Step 2 of 6

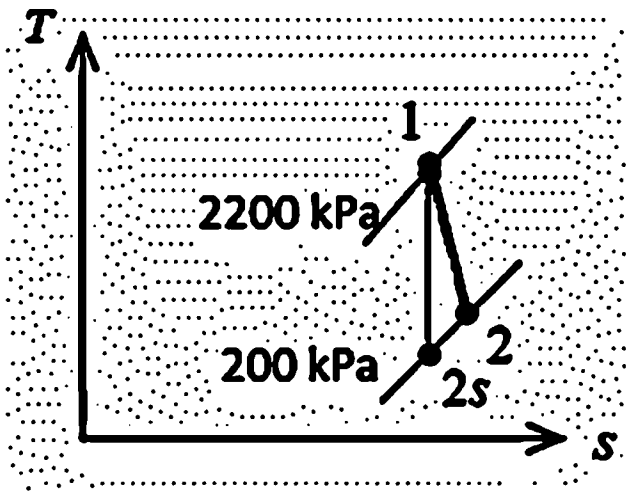
As air is an ideal gas from ideal gas specific heats of gases the properties of air at an average temperature $T_{avg} = 400\ K$ are

$C_p = 1.013\ kJ/kg \cdot K$

$k = 1.395$

Step 3 of 6

The T-s diagram of the expansion in the turbine is as follows



1-2s is isentropic expansion in the turbine
1-2 is actual expansion in the turbine

Step 4 of 6

During the isentropic expansion of the turbine we know

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Therefore the isentropic exit temperature is given by

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = 573 \times \left(\frac{200}{2200}\right)^{\frac{1.395-1}{1.395}}$$

$$T_{2s} = 573 \times (0.09)^{\frac{0.395}{1.395}}$$

$$T_{2s} = 289.76\ K$$

Step 5 of 6

Isentropic efficiency of turbine is given by

$$\eta = \frac{w_a}{w_{is}}$$

Therefore the actual work done is given by

$$w_a = w_s \times \eta$$

$$= \eta \times C_p (T_1 - T_{2s})$$

$$= 0.85 \times 1.013 \times (573 - 289.76)$$

$$\boxed{w_a = 243.883 kJ/kg}$$

Step 6 of 6

The actual work done is also given by

$$w_a = C_p (T_1 - T_2)$$

Therefore the actual exit temperature is given by

$$T_2 = T_1 - \frac{w_a}{C_p}$$

$$= 573 - \frac{243.883}{1.013}$$

$$= 573 - 240.75$$

$$T_{2a} = 332.25\ K$$

$$\boxed{T_{2a} = 59.25°C}$$

Therefore actual work done by the turbine is 243.883 kJ/kg and the actual exit temperature is 59.25°C

Question: 8.192P

Air is expanded in an adiabatic turbine of 90 percent isentropic efficiency from an inlet state of 2200 kPa and 300°C to an outlet pressure of 200 kPa. Calculate the outlet temperature of air, the work produced by this turbine, and the entropy generation.

Step 1 of 8

Given

Isentropic efficiency of turbine is, $\eta = 90\% = 0.9$

Inlet temperature of air is, $T_1 = 300°C = 573\ K$

Inlet pressure of air is, $P_1 = 2200\ kPa$

Exit pressure of air is, $P_2 = 200\ kPa$

Step 2 of 8

As air is an ideal gas from ideal gas specific heats of gases the properties of air at an average temperature $T_{avg} = 400\ K$ are

$C_p = 1.013\ kJ/kg \cdot K$

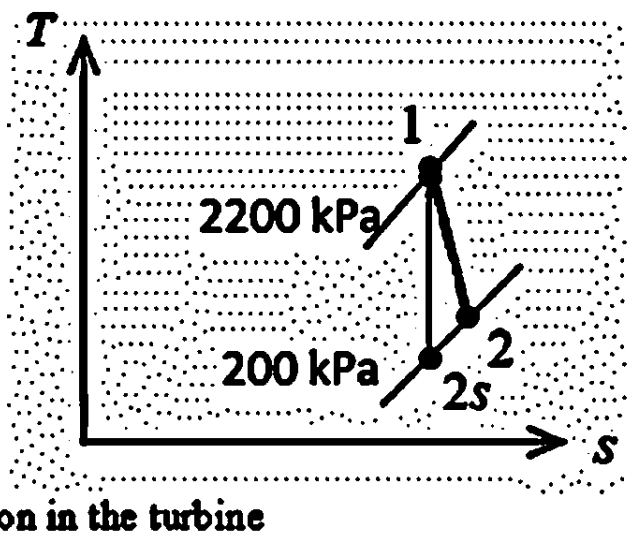
$k = 1.395$

And the gas constant of air is

$R = 0.287\ kJ/kg \cdot K$

Step 3 of 8

The T-s diagram of the expansion in the turbine is as follows



1-2s is isentropic expansion in the turbine

1-2 is actual expansion in the turbine

Step 4 of 8

During the isentropic expansion of the turbine we know

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Therefore the isentropic exit temperature is given by

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_{2s} = 573 \times \left(\frac{200}{2200}\right)^{\frac{1.395-1}{1.395}}$$

$$T_{2s} = 573 \times (0.09)^{\frac{0.395}{1.395}}$$

$$T_{2s} = 289.76\ K$$

Step 5 of 8

Isentropic efficiency of turbine is given by

$$\eta = \frac{w_a}{w_{bs}}$$

Therefore the actual work done is given by

$$w_a = w_s \times \eta$$

$$= \eta \times C_p (T_1 - T_{2s})$$

$$= 0.9 \times 1.013 \times (573 - 289.76)$$

$$\boxed{w_a = 258.229\ kJ/kg}$$

Step 6 of 8

The actual work done is also given by

$$w_a = C_p (T_1 - T_2)$$

Therefore the actual exit temperature is given by

$$T_2 = T_1 - \frac{w_a}{C_p}$$

$$= 573 - \frac{258.229}{1.013}$$

$$= 573 - 254.915$$

$$\boxed{T_2 = 318.085\ K}$$

Step 7 of 8

The entropy balance is given by

$$S_{in} - S_{out} + S_{gen} = \Delta S_{sys}$$

Considering the process to be steady, the entropy of system can be neglected. $\Delta S_{sys} = 0$

$$S_{in} - S_{out} + S_{gen} = 0$$

$$S_{gen} = S_{out} - S_{in}$$

$$s_{gen} = s_{out} - s_{in}$$

$$s_{gen} = s_2 - s_1$$

$$s_{gen} = C_p \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln\left(\frac{P_2}{P_1}\right)$$

$$s_{gen} = 1.013 \times \ln\left(\frac{318.085}{573}\right) - 0.287 \times \ln\left(\frac{200}{2200}\right)$$

$$s_{gen} = -0.596 - (-0.68819)$$

$$\boxed{s_{gen} = 0.09219\ kJ/kg \cdot K}$$

Step 8 of 8

Therefore actual work done by the turbine is 258.229 kJ/kg ,

The actual exit temperature is 318.085K

And the entropy generation is 0.09219 kJ/kg · K

Question: 8.193P pg 1

To control the power output of an isentropic steam turbine, a throttle valve is placed in the steam line supplying the turbine inlet, as shown in the figure. Steam at 6 MPa and 400°C is supplied to the throttle inlet, and the turbine exhaust pressure is set at 70 kPa. Compare the work produced by this steam turbine, in kJ/kg, when the throttle valve is completely open (so that there is no pressure loss) and when it is partially closed so that the pressure at the turbine inlet is 3 MPa.

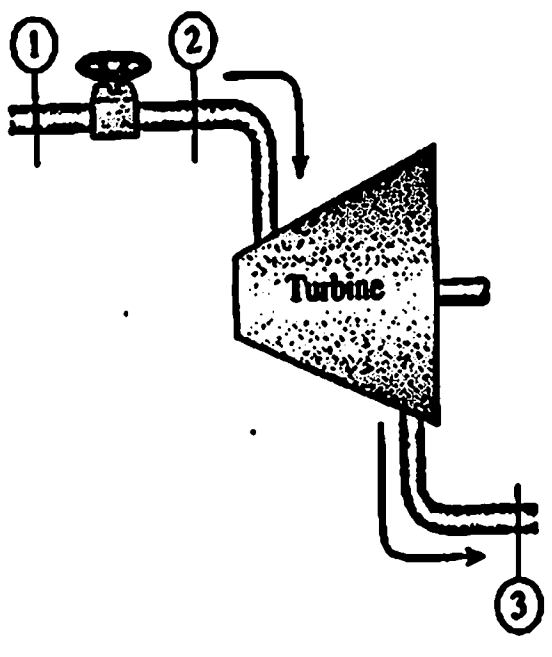


FIGURE P8–193

Step 1 of 13

Given data:

Temperature at throttle inlet is, $T_1 = 400\,^{\circ}\text{C} = 673\,\text{K}$

Pressure at throttle inlet is, $P_1 = 6\,\text{MPa} = 6000\,\text{kPa}$

Pressure at turbine inlet is, $P_2 = 3\,\text{MPa} = 3000\,\text{kPa}$

Pressure at turbine exit is, $P_3 = 70\,\text{kPa}$

Step 2 of 13

From table of 'superheated water',

At pressure $P_1 = 6\,\text{MPa}$ and temperature $T_1 = 400\,^{\circ}\text{C}$ the properties of steam are

$h_1 = 3178.3\,\text{kJ/kg}$

$s_1 = 6.5432\,\text{kJ/kg°K}$

From table of saturated water,

At pressure $P_2 = 3\,\text{MPa}$ the properties of steam are

$h_{f2} = 1008.3\,\text{kJ/kg}$

$h_{fg2} = 1794.9\,\text{kJ/kg}$

$s_{f2} = 2.6454\,\text{kJ/kg°K}$

$s_{fg2} = 3.5402\,\text{kJ/kg°K}$

Step 3 of 13

From table of 'saturated water,

At pressure $P_3 = 70\,\text{kPa}$ the properties of steam are interpolated between

50 kPa & 75 kPa

$$\frac{h_{f3} - 340.54}{384.44 - 340.54} = \frac{70 - 50}{75 - 50}$$

$$\frac{h_{f3} - 340.54}{43.9} = \frac{20}{25}$$

$$\frac{h_{f3} - 340.54}{43.9} = 0.8$$

$$h_{f3} = 0.8 \times 43.9 + 340.54$$

$$= 35.12 + 340.54$$

$$= 375.66\,\text{kJ/kg}$$

Step 4 of 13

$$\frac{h_{fg3} - 2304.7}{2278 - 2304.7} = \frac{70 - 50}{75 - 50}$$

$$\frac{h_{fg3} - 2304.7}{-26.7} = \frac{20}{25}$$

$$\frac{h_{fg3} - 2304.7}{-26.7} = 0.8$$

$$h_{fg3} = -0.8 \times 26.7 + 2304.7$$

$$= -21.36 + 2304.7$$

$$= 2283.34\,\text{kJ/kg}$$

Step 5 of 13

By using the method of interpolation

$$\frac{s_{f3} - 1.0912}{1.2132 - 1.0912} = \frac{70 - 50}{75 - 50}$$

$$\frac{s_{f3} - 1.0912}{0.122} = \frac{20}{25}$$

$$\frac{s_{f3} - 1.0912}{0.122} = 0.8$$

$$s_{f3} = 0.8 \times 0.122 + 1.0912$$

$$= 0.0976 + 1.0912$$

$$= 1.1888\,\text{kJ/kg°K}$$

Step 6 of 13

Similarly,

$$\frac{s_{fg3} - 6.5019}{6.2426 - 6.5019} = \frac{70 - 50}{75 - 50}$$

$$\frac{s_{fg3} - 6.5019}{-0.2593} = \frac{20}{25}$$

$$\frac{s_{fg3} - 6.5019}{-0.2593} = 0.8$$

$$s_{fg3} = -0.8 \times 0.2593 + 6.5019$$

$$= -0.2074 + 6.5019$$

$$= 6.2945\,\text{kJ/kg°K}$$

8.193P pg 2

$$-0.2593$$

$$s_{fg3} = -0.8 \times 0.2593 + 6.5019$$
$$= -0.2074 + 6.5019$$
$$= 6.2945 \text{ kJ/kg°K}$$

Step 7 of 13

For an isentropic process,

$s_3 = s_1$

$s_3 = 6.54323 \text{ kJ/kg°K}$

The entropy at turbine exit is also is given by

$$s_3 = s_{f3} + x_3 \times s_{fg3}$$
$$6.54323 = 1.1888 + x_3 \times 6.2945$$
$$x_3 = \frac{6.54323 - 1.1888}{6.2945}$$
$$= \frac{5.35443}{6.2945}$$
$$= 0.8507$$

Step 8 of 13

The enthalpy at turbine exit is also is given by

$$h_3 = h_{f3} + x_3 \times h_{fg3}$$
$$= 375.66 + 0.8507 \times 2283.34$$
$$= 375.66 + 1942.44$$
$$= 2318.1 \text{ kJ/kg}$$

Step 9 of 13

The work produced when throttle is open fully is given by

$$w_{open} = h_1 - h_3$$
$$w_{open} = 3178.3 - 2318.1$$
$$\boxed{w_{open} = 860.2 \text{ kJ/kg}}$$

Step 10 of 13

As throttling is an isenthalpic process,

$h_2 = h_1$

$h_2 = 3178.3 \text{ kJ/kg}$

From table of superheated water ,

At pressure $P_2 = 3$ MPa and enthalpy $h_2 = 3178.3$ kJ/kg , entropy is interpolated as

$$\frac{s_2 - 6.745}{6.9235 - 6.745} = \frac{3178.3 - 3116.1}{3231.7 - 3116.1}$$
$$\frac{s_2 - 6.745}{0.1785} = \frac{62.2}{115.6}$$
$$\frac{s_2 - 6.745}{0.1785} = 0.538$$
$$= 0.538 \times 0.1785 + 6.745$$
$$= 0.096 + 6.745$$
$$s_2 = 6.841 \text{ kJ/kg°K}$$

Step 11 of 13

For an isentropic process,

$s_3 = s_2$

$s_3 = 6.841 \text{ kJ/kg°K}$

The entropy at turbine exit is also is given by

$$s_3 = s_{f3} + x_3 \times s_{fg3}$$
$$6.841 = 1.1888 + x_3 \times 6.2945$$
$$x_3 = \frac{6.841 - 1.1888}{6.2945}$$
$$= \frac{5.6522}{6.2945}$$
$$= 0.898$$

Step 12 of 13

The enthalpy at turbine exit is also is given by

$$h_3 = h_{f3} + x_3 \times h_{fg3}$$
$$= 375.66 + 0.898 \times 2283.34$$
$$= 375.66 + 2050.44$$
$$= 2426.1 \text{ kJ/kg}$$

Step 13 of 13

The work produced when throttle is partial closed is given by

$$w_{par} = h_1 - h_3$$
$$w_{par} = 3178.3 - 2426.1$$
$$\boxed{w_{par} = 752.2 \text{ kJ/kg}}$$

Therefore actual work produced by turbine when throttle fully open is greater than when throttle partially closed.

Question: 8.194P pg 1

Oxygen tanks are filled by compressing the oxygen gas, as shown in the figure. An initially evacuated 1-m3 tank is to be filled to 13,000 kPa while the temperature of the oxygen in the tank remains constant at 20°C with this system. An isentropic compressor is used. The oxygen enters this compressor with a constant pressure and temperature of 150 kPa and 20°C. Determine the total work required by the compressor and the total heat transfer from the oxygen tank.

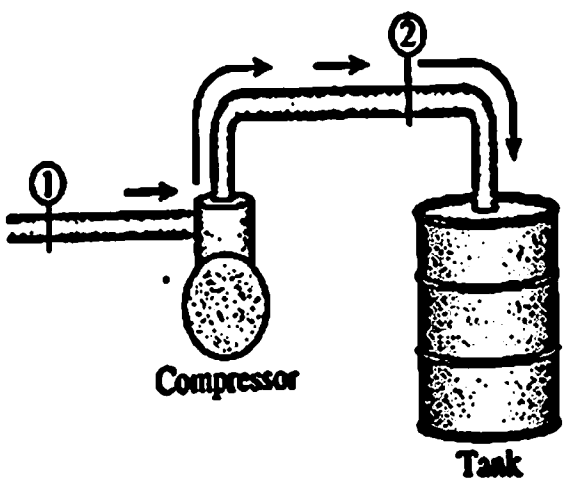


FIGURE P8–194

Step 1 of 9

Given data:

Volume of tank is, $V = 1\ \text{m}^3$

Final pressure in the tank is, $P_2 = 13000\ \text{kPa}$

Temperature in the tank is, $T_2 = 20\ ^\circ\text{C} = 293\ \text{K}$

Initial pressure at compressor inlet is, $P_1 = 150\ \text{kPa K}$

Temperature at compressor inlet is, $T_1 = 20\ ^\circ\text{C} = 293\ \text{K}$

Step 2 of 9

From table of 'ideal gas specific heats of various common gases'

Properties of oxygen are

$R = 0.2598\ \text{kJ/kg°K}$

$c_p = 0.918\ \text{kJ/kg°K}$

$c_v = 0.658\ \text{kJ/kg°K}$

$k = 1.395$

Step 3 of 9

Mass of the control volume is given by

$$m_{cv} = \frac{PV}{RT}$$

Initial mass at of tank is given by

$$m_{in} = \frac{d\left(m_{cv}\right)}{dt}$$

$$= \frac{d\left(\frac{PV}{RT}\right)}{dt}$$

$$= \frac{V}{RT} \times \frac{dP}{dt}$$

Step 4 of 9

Final mass at of tank is given by

$$m_2 = \frac{P_2 \times V}{R \times T_2}$$

$$= \frac{13000 \times 1}{0.2598 \times 293}$$

$$= \frac{13000}{76.1214}$$

$$= 170.78\ \text{kg}$$

Step 5 of 9

According to first law of thermodynamics

$$Q = \frac{d(\text{mu})}{dt} - \text{m}_{\text{in}} \times \text{h}_{\text{in}}$$

$$= \text{c}_\text{v} \times \text{T} \times \frac{\text{dm}}{\text{dt}} - \text{m}_{\text{in}} \times \text{c}_\text{p} \times \text{T}_2$$

For isentropic process, the final temperature is given by $T_2 = T_1\left(\frac{P}{P_1}\right)^{\frac{k-1}{k}}$. So,

$$Q = c_v \times T \times \frac{dm}{dt} - m_{in} \times c_p \times T_1\left(\frac{P}{P_1}\right)^{\frac{k-1}{k}}$$

$$= \left[c_v \times T \times \frac{dm}{dt}\right] - \left[\frac{V}{RT} \times \frac{dP}{dt} \times c_p \times T_1\left(\frac{P}{P_1}\right)^{\frac{k-1}{k}}\right]$$

Step 6 of 9

Integrating from initial to end of compression we get

$$Q_{in} = \left[c_v \times T \times \int_1^2 dm\right] - \left[\frac{V}{RT} \times c_p \times T_1\left(\frac{1}{P_1}\right)^{\frac{k-1}{k}} \times \int_1^2 P^{\frac{k-1}{k}} dP\right]$$

$$Q_{in} = \left[c_v \times T \times (m_2 - m_1)\right] - \left[\frac{V}{RT} \times c_p \times T_1\left(\frac{1}{P_1}\right)^{\frac{k-1}{k}} \times \frac{k}{2k-1} \times P_2^{\frac{k-1}{k}} \times P_2\right]$$

As initial tank is evacuated, $\text{m}_1 = 0$

$$Q_{in} = \left[c_v \times T \times (m_2 - 0)\right] - \left[\frac{V}{RT} \times c_p \times T_1\left(\frac{1}{P_1}\right)^{\frac{k-1}{k}} \times \frac{k}{2k-1} \times P_2^{\frac{k-1}{k}} \times P_2\right]$$

$$Q_{in} = \left[c_v \times T \times m_2\right] - \left[c_p \times T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \times \frac{k}{2k-1} \times \frac{P_2 V}{RT}\right]$$

Step 7 of 9

Step 7 of 9

$$Q_{in} = [c_v \times T \times m_2] - \left[c_p \times T_1 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \times \frac{k}{2k-1} \times m_2\right]$$

$$= [0.658 \times 293 \times 170.78] - \left[0.918 \times 293 \times \left(\frac{13000}{150}\right)^{\frac{1.395-1}{1.395}} \times \frac{1.395}{2 \times 1.395 - 1} \times 170.78\right]$$

$$= [0.658 \times 293 \times 170.78] - \left[0.918 \times 293 \times (86.67)^{\frac{0.395}{1.395}} \times \frac{1.395}{2.79-1} \times 170.78\right]$$

$$= [32925.36] - \left[0.918 \times 293 \times (86.67)^{0.2831} \times \frac{1.395}{1.79} \times 170.78\right]$$

$$= [32925.36] - [0.918 \times 293 \times 3.537 \times 0.7793 \times 170.78]$$

$$= [32925.36] - [126615.55]$$

$$\boxed{Q_{in} = -93690.19 \text{ kJ}}$$

Step 8 of 9

Applying first law to the whole system,

$$Q_{in} + W_{in} = \frac{d(mu)}{dt} - m_{in} \times h_{in}$$

$$Q_{in} + W_{in} = (m_2 u_2 - m_1 u_1) - m_2 \times h_1$$

As initial tank is evacuated, $m_1 = 0$

$$Q_{in} + W_{in} = (m_2 u_2 - 0) - m_2 \times h_1$$

$$Q_{in} + W_{in} = m_2 u_2 - m_2 \times h_1$$

$$W_{in} = -Q_{in} + m_2 u_2 - m_2 \times h_1$$

$$= -Q_{in} + m_2 \times (u_2 - h_1)$$

$$= -Q_{in} + m_2 \times (c_v \times T_2 - c_p \times T_1)$$

Step 9 of 9

As the temperature is constant $T_2 = T_1 = T$

$$W_{in} = -Q_{in} + m_2 \times (c_v \times T - c_p \times T)$$

$$= -Q_{in} + m_2 \times T \times (c_v - c_p)$$

$$= -(-93690.19) + 170.78 \times 293 \times (0.658 - 0.918)$$

$$= 93690.19 + 170.78 \times 293 \times (-0.26)$$

$$= 93690.19 - 13010.02$$

$$\boxed{W_{in} = 80680.17 \text{ kJ}}$$

Therefore total work input is 80680.17 kJ and total heat lost is 93690.19 kJ

Question: 8.195P pg 1

Two rigid tanks are connected by a valve. Tank A is insulated and contains 0.3 m3 of steam at 400 kPa and 60 percent quality. Tank B is uninsulated and contains 2 kg of steam at 200 kPa and 250°C. The valve is now opened, and steam flows from tank A to tank B until the pressure in tank A drops to 200 kPa. During this process 300 kJ of heat is transferred from tank B to the surroundings at 17°C. Assuming the steam remaining inside tank A to have undergone a reversible adiabatic process, determine (a) the final temperature in each tank and (*b*) the entropy generated during this process.

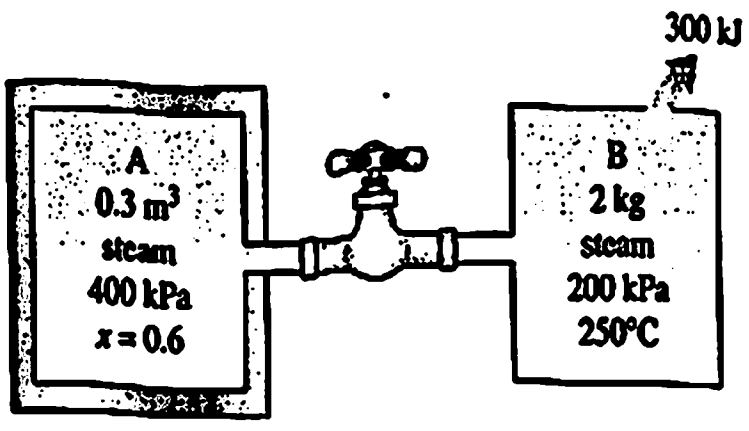
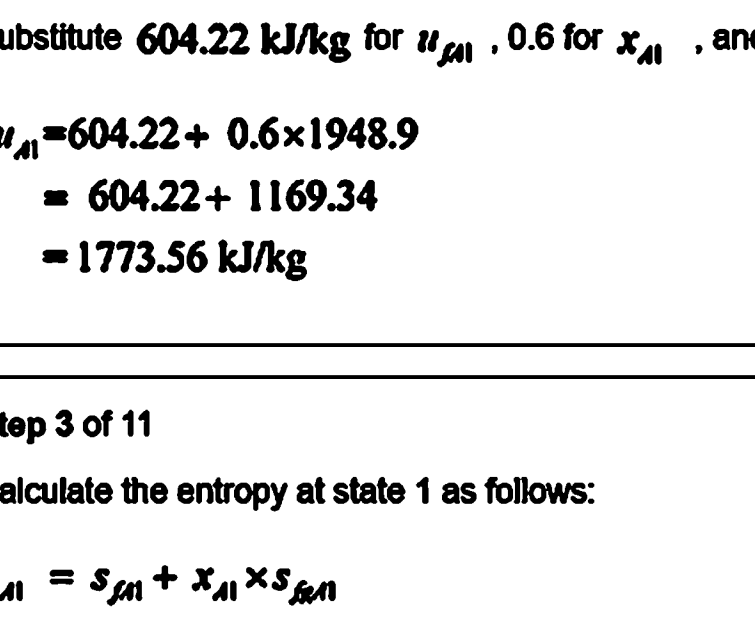


FIGURE P8–195

Step 1 of 11

Draw the schematic diagram of rigid tanks connected system:



Step 2 of 11

Obtain the properties from the table A-5, "Saturated water - pressure table" in textbook.

At pressure $P_{A1} = 400\ \text{kPa}$ and quality $x_{A1} = 0.6$ the properties of steam are

$$u_{fA1} = 604.22\ \text{kJ/kg}$$
$$u_{fgA1} = 1948.9\ \text{kJ/kg}$$
$$s_{fA1} = 1.7765\ \text{kJ/kg}\cdot\text{K}$$
$$s_{fgA1} = 5.1191\ \text{kJ/kg}\cdot\text{K}$$
$$v_{fA1} = 0.001084\ \text{m}^3\text{/kg}$$
$$v_{gA1} = 0.46242\ \text{m}^3\text{/kg}$$

Calculate the internal energy at state 1 as follows:

$$u_{A1} = u_{fA1} + x_{A1} \times u_{fgA1}$$

Substitute $604.22\ \text{kJ/kg}$ for u_{fA1}, 0.6 for x_{A1}, and $1948.9\ \text{kJ/kg}$ for u_{fgA1}.

$$u_{A1} = 604.22 + 0.6 \times 1948.9$$
$$= 604.22 + 1169.34$$
$$= 1773.56\ \text{kJ/kg}$$

Step 3 of 11

Calculate the entropy at state 1 as follows:

$$s_{A1} = s_{fA1} + x_{A1} \times s_{fgA1}$$

Substitute $1.7765\ \text{kJ/kg}\cdot\text{K}$ for s_{fA1} and $5.1191\ \text{kJ/kg}\cdot\text{K}$ for s_{fgA1}.

$$s_{A1} = 1.7765 + 0.6 \times 5.1191$$
$$= 1.7765 + 3.0715$$
$$= 4.848\ \text{kJ/kg°K}$$

Calculate the specific volume at state 1 as follows:

$$v_{A1} = v_{fA1} + x_{A1} \times v_{fgA1}$$
$$= v_{fA1} + x_{A1} \times (v_{gA1} - v_{fA1})$$

Substitute $0.001084\ \text{m}^3\text{/kg}$ for v_{fA1}, 0.6 for x_{A1}, and $0.46242\ \text{m}^3\text{/kg}$ for v_{gA1}.

$$v_{A1} = 0.001084 + 0.6 \times (0.46242 - 0.001084)$$
$$= 0.001084 + 0.6 \times 0.461336$$
$$= 0.001084 + 0.276802$$
$$= 0.277886\ \text{m}^3\text{/kg}$$

Step 4 of 11

For a reversible adiabatic process entropy remains constant.

$$s_{A2} = s_{A1} = 4.848\ \text{kJ/kg°K}$$

As saturated vapor entropy is greater than $s_{A2} = 4.848\ \text{kJ/kg°K}$, final temperature of tank A is given by saturated temperature at pressure $P_{A2} = 200\ \text{kPa}$.

Therefore, from table of saturated water pressure table', final temperature of tank A is

$$\boxed{T_{A2} = 120.21\ °\text{C}}.$$

Final entropy of tank A is also given by

$$s_{A2} = s_{fA2} + x_{A2} \times s_{fgA2}$$
$$4.848 = 1.5302 + x_{A2} \times 5.5968$$
$$x_{A2} = \frac{4.848 - 1.5302}{5.5968}$$
$$= \frac{3.3178}{5.5968}$$
$$= 0.5928$$

Obtain the properties from the table A-5, "Saturated water - pressure table" in textbook.

At pressure $P_{A1} = 200\ \text{kPa}$ and quality $x_{A1} = 0.5928$ the properties of steam are

$= 0.5928$

Obtain the properties from the table A-5, "Saturated water - pressure table" in textbook.

At pressure $P_{A1} = 200\ \text{kPa}$ and quality $x_{A1} = 0.5928$ the properties of steam are

$v_{fA2} = 0.001061\ \text{m}^3/\text{kg}$

$v_{gA2} = 0.88578\ \text{m}^3/\text{kg}$

$u_{fA2} = 504.5\ \text{kJ/kg}$

$u_{fgA2} = 2024.6\ \text{kJ/kg}$

Step 5 of 11

Final volume of tank A is given by

$$\begin{aligned} v_{A2} &= v_{fA2} + x_{A2} \times v_{fgA2} \\ &= v_{fA2} + x_{A2} \times \left(v_{gA2} - v_{fA2}\right) \end{aligned}$$

Substitute $0.001061\ \text{m}^3/\text{kg}$ for v_{fA2}, $0.88578\ \text{m}^3/\text{kg}$ for v_{gA2}, and 0.5928 for x_{A2}.

$$\begin{aligned} v_{A2} &= 0.001061 + 0.5928 \times (0.88578 - 0.001061) \\ &= 0.001061 + 0.5928 \times 0.884719 \\ &= 0.001061 + 0.524461 \\ &= 0.5255\ \text{m}^3/\text{kg} \end{aligned}$$

Step 6 of 11

Final volume of tank A is given by

$$\begin{aligned} u_{A2} &= u_{fA2} + x_{A2} \times u_{fgA2} \\ &= 504.5 + 0.5928 \times 2024.6 \\ &= 504.5 + 1200.18 \\ &= 1704.68\ \text{kJ/kg} \end{aligned}$$

Obtain the properties from the table A-6, "Superheated water" in the textbook.

At pressure $P_{B1} = 200\ \text{kPa}$ and temperature $T_{B1} = 250\ ^\circ\text{C}$.

Properties of steam are

$u_{B1} = 2731.4\ \text{kJ/kg}$

$s_{B1} = 7.7100\ \text{kJ/kg} \cdot \text{K}$

$v_{B1} = 1.1989\ \text{m}^3/\text{kg}$

Calculate the initial mass in tank A as follows:

$$m_{A1} = \frac{V_A}{v_{A1}}$$

Here, V_A is the volume of the tank *A*.

Substitute $0.3\ \text{m}^3$ for V_A and $0.277886\ \text{m}^3/\text{kg}$ for v_{A1}.

$$\begin{aligned} m_{A1} &= \frac{0.3}{0.277886} \\ &= 1.0796\ \text{kg} \end{aligned}$$

Step 7 of 11

Calculate the final mass of the tank as follows:

$$m_{A2} = \frac{V_A}{v_{A2}}$$

Substitute $0.3\ \text{m}^3$ for V_A and $0.5255\ \text{m}^3/\text{kg}$ for v_{A2}.

$$\begin{aligned} m_{A2} &= \frac{0.3}{0.5255} \\ &= 0.5709\ \text{kg} \end{aligned}$$

Calculate the mass of steam out from tank A as follows:

$$m_{Aout} = m_{A1} - m_{A2}$$

Substitute 1.0796 kg for m_{A1} and $0.5709\ \text{kg}$ for m_{A2}.

$$\begin{aligned} m_{Aout} &= 1.0796 - 0.5709 \\ &= 0.5087\ \text{kg} \end{aligned}$$

Final mass of steam in tank B is given by

$$m_{B2} = m_{B1} + m_{Bin}$$

As steam is moved from tank A to tank B, $m_{Aout} = m_{Bin}$

$$\begin{aligned} m_{B2} &= m_{B1} + m_{Aout} \\ &= 2 + 0.5087 \\ &= 2.5087\ \text{kg} \end{aligned}$$

Step 8 of 11

Final specific volume of steam in tank B as follows:

$$\begin{aligned} v_{B2} &= \frac{V_B}{m_{B2}} \\ &= \frac{m_{B1} \times v_{B1}}{m_{B2}} \end{aligned}$$

Here, m_{B1} is the initial mass of tank *B*.

Substitute 2 kg for m_{B1}, $1.1989\ \text{m}^3/\text{kg}$ for v_{B1}, and $2.5087\ \text{kg}$ for m_{B2}.

$$\begin{aligned} v_{B2} &= \frac{2 \times 1.1989}{2.5087} \\ &= \frac{2.3978}{} \end{aligned}$$

Substitute 2 kg for m_{B1}, 1.1989 m^3/kg for v_{B1}, and 2.5087 kg for m_{B2}.

$$v_{B2} = \frac{2 \times 1.1989}{2.5087}$$
$$= \frac{2.3978}{2.5087}$$
$$= 0.9558 \text{ m}^3/\text{kg}$$

Energy balance is given by

$$E_{in} - E_{in} = \Delta E_{sys}$$
$$0 - Q_{out} = \Delta U$$

Step 9 of 11

Re-write the equation as follows:

$$-Q_{out} = \left(m_{A2}u_{A2} - m_{A1}u_{A1}\right) + \left(m_{B2}u_{B2} - m_{B1}u_{B1}\right)$$

Here, Q_{out} is the heat transferred from *B* to surroundings.

Substitute 300 kJ for Q_{out}, 0.5709 kg for m_{A2}, 1704.68 kg for u_{A2}, 1.7096 kg for m_{A1}, 1773.02 kJ/kg for u_{A1}, 2.5087 kg for m_{B2}, 2 kg for m_{B1}, and 2731.4 kJ/kg for u_{B1}.

$$-300 = \left(0.5709 \times 1704.68 - 1.0796 \times 1773.02\right) + \left(2.5087 \times u_{B2} - 2 \times 2731.4\right)$$
$$-300 = -940.95 + 2.5087 \times u_{B2} - 5462.8$$
$$u_{B2} = \frac{940.95 - 300 + 5462.8}{2.5087}$$
$$= \frac{6103.75}{2.5087}$$
$$= 2433.03 \text{ kJ/kg}$$

Step 10 of 11

From table of saturated water temperature table,

At specific volume $v_{B2} = 0.9558 \text{ m}^3/\text{kg}$ and $u_{B2} = 2433.03 \text{ kJ/kg}$ is interpolated as

$$\frac{T_{B2} - 115}{120 - 115} = \frac{0.9558 - 1.0360}{0.89133 - 1.0360}$$
$$\frac{T_{B2} - 115}{5} = \frac{-0.0802}{-0.14467}$$
$$\frac{T_{B2} - 115}{5} = 0.5544$$
$$T_{B2} = 0.5544 \times 5 + 115$$
$$= 2.772 + 115$$

Therefore, the final temperature of tank B is $\boxed{T_{B2} = 117.772\ ^\circ\text{C}}$.

Step 11 of 11

(b)

Final the quality of the steam as follows:

$$u_{B2} = u_{fB2@200kpa} + x_{B2} \times u_{fgB22@200kpa}$$

Substitute 2433.03 kJ/kg for u_{B2}, 504.5 kJ/kg for $u_{fB2@200kpa}$, and 2024.6 kJ/kg for $u_{fgB22@200kpa}$.

$$2433.03 = 504.5 + x_{B2} \times 2024.6$$
$$x_{B2} = \frac{2433.03 - 504.5}{2024.6}$$
$$= \frac{1928.53}{2024.6}$$
$$= 0.9525$$

Find the final entropy as follows:

$$s_{B2} = s_{fB2} + x_{B2} \times s_{fgB2}$$

Substitute 1.5302 kJ/kg·K for s_{fB2}, 0.953 for x_{B2}, and 5.5968 kJ/kg for s_{fgB2}.

$$s_{B2} = 1.5302 + 0.953 \times 5.5968$$
$$= 1.5302 + 5.3310$$
$$= 6.8612 \text{ kJ/kg°K}$$

Write the equation for entropy balance as follows:

$$S_{in} - S_{in} + S_{gen} = \Delta S_{sys}$$
$$-\frac{Q_{out}}{T_{surr}} + S_{gen} = \Delta S_A + \Delta S_B$$
$$S_{gen} = \Delta S_A + \Delta S_B + \frac{Q_{out}}{T_{surr}}$$
$$S_{gen} = \left(m_{A2}s_{A2} - m_{A1}s_{A1}\right) + \left(m_{B2}s_{B2} - m_{B1}s_{B1}\right) + \frac{Q_{out}}{T_{surr}}$$

Substitute 0.5709 kg for m_{A2}, 1.7096 kg for m_{A1}, 4.848 kJ/kg·K for s_{A2}, 4.848 kJ/kg·K for s_{A1}, 6.8612 kJ/kg·K for s_{B2}, 7.71 kJ/kg·K for s_{B1}, 300 kJ for Q_{out}, and 290 K for T_{surr}.

$$S_{gen} = \left(0.5709 \times 4.848 - 1.0796 \times 4.848\right) + \left(2.5087 \times 6.8612 - 2 \times 7.7100\right) + \frac{300}{290}$$
$$S_{gen} = \left(2.768 - 5.234\right) + \left(17.213 - 15.420\right) + 1.034$$
$$S_{gen} = -2.466 + 1.793 + 1.034$$

Therefore, the entropy generated during this process is $\boxed{0.361 \text{ kJ/·K}}$

Question: 8.196P

Heat is transferred steadily to boiling water in the pan through its flat bottom at a rate of 500 W. If the temperatures of the inner and outer surfaces of the bottom of the tank are 104 and 105°C, respectively, determine the rate of entropy generation within bottom of the pan, in W/K.

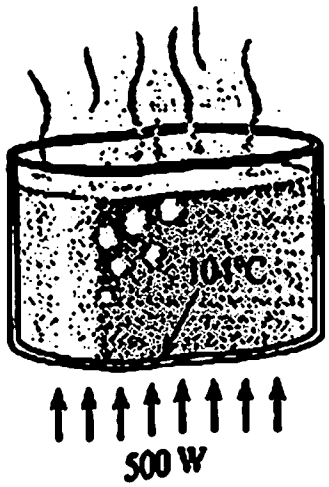


FIGURE P8–196

Step 1 of 2

Given data

Rate of heat transfer to boiling water from the bottom of the pan $Q = 500\,\text{W}$

Temperature of the inner surface of the bottom of the tank $T_{inner} = 104°\text{C}$

Temperature of the outer surface of the bottom of the tank $T_{outer} = 105°\text{C}$

Step 2 of 2

From entropy balance for this steady system is

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in}$$

$$\dot{S}_{gen} = -\frac{Q}{T_{outer}} + \frac{Q}{T_{inner}}$$

$$= -\frac{500}{(105+273)} + \frac{500}{(104+273)}$$

$$= -\frac{500}{378} + \frac{500}{377}$$

$$\dot{S}_{gen} = 0.00351\,\text{W/K}$$

Rate of entropy generation within bottom of the pan is

$$\boxed{\dot{S}_{gen} = 0.00351\,\text{W/K}}$$

Question: 8.197P

A 1200-W electric resistance heating element whose diameter is 0.5 cm is immersed in 40 kg of water initially at 20°C. Assuming the water container is well-insulated, determine how long it will take for this heater to raise the water temperature to 50°C. Also, determine the entropy generated during this process, in kJ/K.

Step 1 of 4

Given data

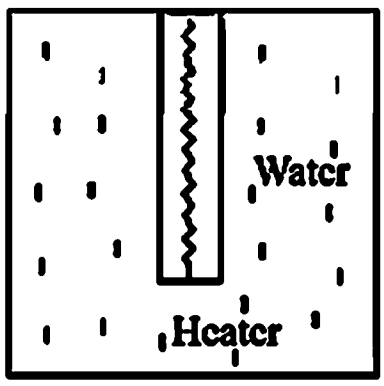
Mass of water $m = 40\,\text{kg}$

Heating capacity of the heater $Q = 1200\,\text{W}$

Initial temperature of water $T_1 = 20°\text{C}$

Final temperature of water $T_2 = 50°\text{C}$

Step 2 of 4



Step 3 of 4

Assume specific heat of water is

$c = 4.18\,\text{kJ/kg}\,°\text{C}$

Energy balance for this closed system

$(Q)t = mc\Delta T$

$Q \cdot t = mc(T_2 - T_1)$

$(1200) \times t = (40) \times 4.18 \times 10^3 \times (50 - 20)$

$(1200) \times t = 5016000$

$t = 4180\,s$

$= \left(\frac{4180}{3600}\right)$

$t = 1.16\,\text{h}$

Time taken to reach the final temperature of 50° C is

$\boxed{t = 1.16\,\text{h}}$

Step 4 of 4

Entropy generated during this process

$\Delta S = mC \ln \frac{T_2}{T_1}$

$= (40)(4.18) \ln \left[\frac{(50+273)}{(20+273)}\right]$

$\Delta S = 16.3\,\text{kJ/K}$

Entropy generated during this process $\boxed{\Delta S = 16.3\,\text{kJ/K}}$

Question: 8.198P

In large steam power plants, the feedwater is frequently heated in closed feedwater heaters, which are basically heat exchangers, by steam extracted from the turbine at some stage. Steam enters the feedwater heater at 1 MPa and 200°C and leaves as saturated liquid at the same pressure. Feedwater enters the heater at 2.5 MPa and 50°C and leaves 10°C below the exit temperature of the steam. Neglecting any heat lossesfrom the outer surfaces of the heater, determine (*a*) the ratio of the mass flow rates of the extracted steam and the feedwater heater and (*b*) the total entropy change for this process per unit mass of the feedwater.

Step 1 of 9

Given data:

Pressure at steam inlet is, $P_1 = 1\,\text{MPa} = 1000\,\text{kPa}$

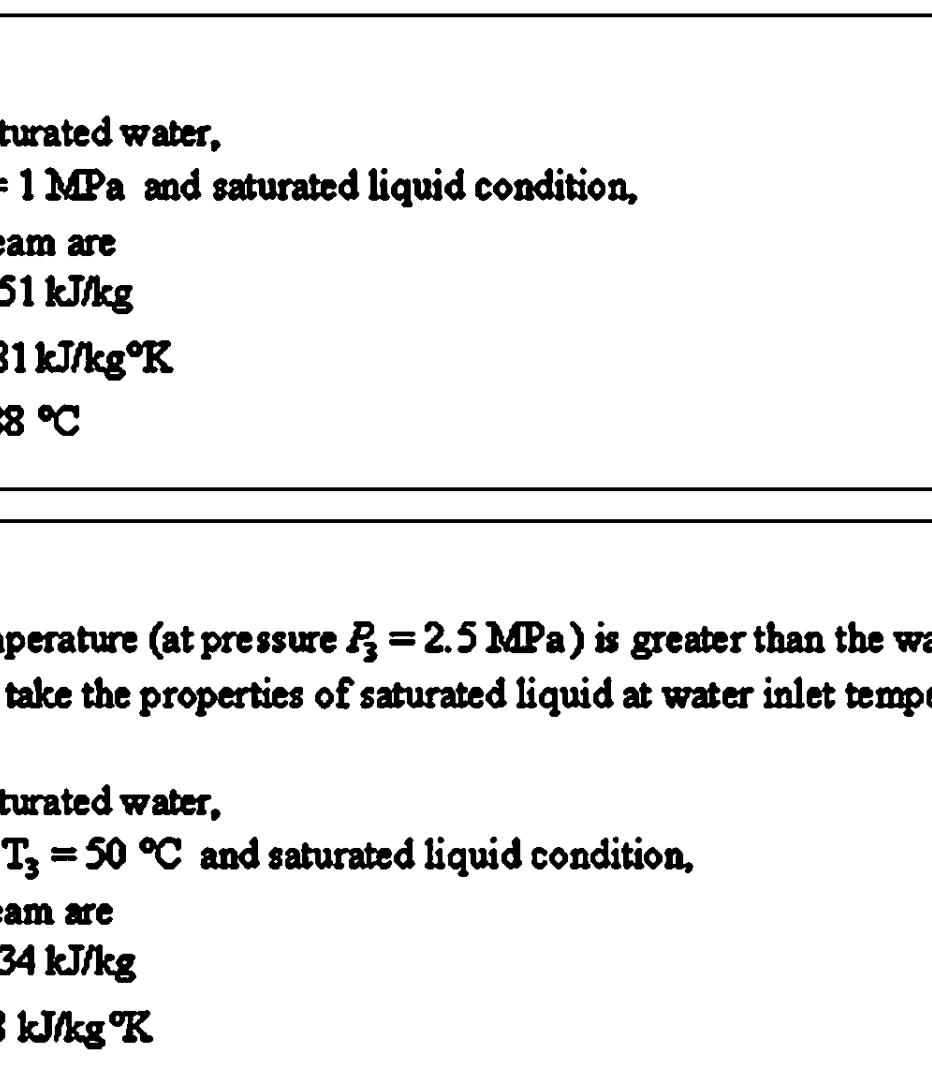
Temperature at steam inlet is, $T_1 = 200\,^\circ\text{C} = 473\,\text{K}$

Pressure at water inlet is, $P_3 = 2.5\,\text{MPa} = 2500\,\text{kPa}$

Temperature at water inlet is, $T_3 = 50\,^\circ\text{C} = 323\,\text{K}$

Temperature at water exit is, $T_4 = T_2 - 10$

Step 2 of 9



Step 3 of 9

From table of superheated water' ,

At pressure $P_1 = 1\,\text{MPa}$ and Temperature $T_1 = 200\,^\circ\text{C}$, properties of steam are

$h_1 = 2828.3\,\text{kJ/kg}$

$s_1 = 6.6956\,\text{kJ/kg°K}$

Step 4 of 9

From table of saturated water,

At pressure $P_2 = 1\,\text{MPa}$ and saturated liquid condition,

Properties of steam are

$h_2 = h_f = 762.51\,\text{kJ/kg}$

$s_2 = s_f = 2.1381\,\text{kJ/kg°K}$

$T_2 = T_{sat} = 179.88\,^\circ\text{C}$

Step 5 of 9

As saturated temperature (at pressure $P_3 = 2.5\,\text{MPa}$) is greater than the water inlet temperature, we take the properties of saturated liquid at water inlet temperature ($T_3 = 50\,^\circ\text{C}$)

From table of saturated water,

At temperature $T_3 = 50\,^\circ\text{C}$ and saturated liquid condition,

properties of steam are

$h_3 = h_f = 209.34\,\text{kJ/kg}$

$s_3 = s_f = 0.7038\,\text{kJ/kg°K}$

Step 6 of 9

Then,

$T_4 = T_2 - 10$

$T_4 = 179.88 - 10$

$= 169.88\,^\circ\text{C}$

As saturated temperature (at pressure $P_4 = 2.5\,\text{MPa}$) is greater than the water inlet temperature, we take the properties of saturated liquid at water inlet temperature ($T_4 = 169.88\,^\circ\text{C}$)

From table of saturated water,

At temperature $T_4 = 169.88\,^\circ\text{C} = 170\,^\circ\text{C}$ and saturated liquid condition,

Properties of steam are

$h_4 = h_f = 719.08\,\text{kJ/kg}$

$s_4 = s_f = 2.0417\,\text{kJ/kg°K}$

Step 7 of 9

As there is no mass loss,

$m_1 = m_2 = m_s$

$m_3 = m_4 = m_w$

Heat loss by steam is same as heat gained by water. So,

$$m_s(h_2 - h_1) = m_w(h_3 - h_4)$$

$$\frac{m_s}{m_w} = \frac{h_3 - h_4}{h_2 - h_1}$$

$$= \frac{209.34 - 719.08}{762.51 - 2828.3}$$

$$= \frac{-509.74}{-2065.79}$$

$$\boxed{\frac{m_s}{m_w} = 0.2468}$$

Step 8 of 9

Entropy balance is given by

$$S_{in} - S_{in} + S_{gen} = \Delta S_{sys}$$

For steady process $\Delta S_{sys} = 0$

$$S_{in} - S_{in} + S_{gen} = \Delta S_{sys}$$

$$m_1 s_1 - m_2 s_2 + m_3 s_3 - m_4 s_4 + S_{gen} = 0$$

$$m_s s_1 - m_s s_2 + m_w s_3 - m_w s_4 + S_{gen} = 0$$

$$m_s(s_1 - s_2) + m_w(s_3 - s_4) + S_{gen} = 0$$

Step 9 of 9

Entropy generation is

$$S_{gen} = m_s(s_2 - s_1) + m_w(s_4 - s_3)$$

$$\frac{S_{gen}}{m_w} = \frac{m_s}{m_w}(s_2 - s_1) + (s_4 - s_3)$$

$$\frac{S_{gen}}{m_w} = 0.2468 \times (2.1381 - 6.6956) + (2.0417 - 0.7038)$$

$$\frac{S_{gen}}{m_w} = 0.2468 \times (-4.5575) + 1.3379$$

$$\frac{S_{gen}}{m_w} = -1.1248 + 1.3379$$

$$\boxed{\frac{S_{gen}}{m_w} = 0.2131\,\text{kJ/°K/kg of feed water}}$$

Therefore ratio of mass flow rates is 0.2468 and entropy generation is 0.2131 kJ/°K/kg of feed water

Question: 8.199P



Reconsider Prob. 8-198. Using EES (or other) software, investigate the effect of the state of the steam at the inlet of the feedwater heater. Assume the entropy of the extraction steam is constant at the value for 1 MPa, 200°C and decrease the extraction steam pressure from 1 MPa to 100 kPa. Plot both the ratio of the mass flow rates of the extracted steam and the feedwater heater and the total entropy change for this process per unit mass of the feedwater as functions of the extraction pressure.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

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Question: 8.200P

A 5-ft3 rigid tank initially contains refrigerant-134a at 80 psia and 100 percent quality. The tank is connected by a valve to a supply line that carries refrigerant-134a at 160 psia and 80°F. The valve is now opened, allowing the refrigerant to enter the tank, and is closed when it is observed that the tank contains only saturated liquid at 120 psia. Determine (*a*) the mass of the refrigerant that entered the tank, (*b*) the amount of heat transfer with the surroundings at 100°F, and (*c*) the entropy generated during this process.

Step 1 of 9

Given data:

Volume of the tank is, $V = 5\ \text{ft}^3$

Initial pressure in the tank is, $P_1 = 80$ psia

Initial quality in the tank is, $x_1 = 100\%$

Final pressure in the tank is, $P_2 = 120$ psia

Final quality in the tank is, $x_2 = 0\%$

Pressure of supply line is, $P_{in} = 160$ psia

Temperature of supply line is, $T_{in} = 80\ ^\circ\text{F} = 540\ \text{R}$

Surrounding temperature is, $T_{surr} = 100\ ^\circ\text{F} = 560\ \text{R}$

Step 2 of 9

From table A-12E, at pressure $P_1 = 80$ psia and saturated vapor condition, properties of R-134a are

$v_1 = v_g = 0.5975\ \text{ft}^3/\text{lbm}$

$u_1 = u_g = 103.35\ \text{Btu/lbm}$

$s_1 = s_g = 0.2204\ \text{Btu/lbm}^\circ\text{R}$

Step 3 of 9

From table A-12E, at pressure $P_2 = 120$ psia and saturated liquid condition, properties of R-134a are

$v_2 = v_f = 0.01360\ \text{ft}^3/\text{lbm}$

$u_2 = u_f = 41.485\ \text{Btu/lbm}$

$s_2 = s_f = 0.08589\ \text{Btu/lbm}^\circ\text{R}$

Step 4 of 9

As saturated temperature (at pressure $P_{in} = 160$ psia) is greater than the water temperature, we take the properties of saturated liquid at water temperature ($T_{in} = 80\ ^\circ\text{F}$)

From table A-11E, at temperature $T_{in} = 80\ ^\circ\text{F}$ and saturated liquid condition, properties of steam are

$h_{in} = h_f = 38.169\ \text{Btu/lbm}$

$s_{in} = s_f = 0.07934\ \text{Btu/lbm}^\circ\text{R}$

Step 5 of 9

Initial mass in the tank is given by

$$m_1 = \frac{V}{v_1}$$

$$= \frac{5}{0.5975}$$

$$= 8.368\ \text{ft}$$

Step 6 of 9

Final mass in the tank is given by

$$m_2 = \frac{V}{v_2}$$

$$= \frac{5}{0.0136}$$

$$= 367.647\ \text{ft}$$

Step 7 of 9

(a) Mass entered in the tank is given by,

$$m_{in} = m_2 - m_1$$

$$= 367.647 - 8.638$$

$$\boxed{m_{in} = 359.009\ \text{ft}}$$

Step 8 of 9

(b) Energy balance is given by

$$E_{in} - E_{out} = \Delta S_{sys}$$

As there is no energy is going out, $E_{out} = 0$

$$E_{in} - 0 = \Delta S_{sys}$$

$$E_{in} = \Delta S_{sys}$$

$$Q_{in} + m_{in} \times h_{in} = m_2 \times u_2 - m_1 \times u_1$$

$$Q_{in} = m_2 \times u_2 - m_1 \times u_1 - m_{in} \times h_{in}$$

$$Q_{in} = 367.647 \times 41.485 - 8.368 \times 103.35 - 359.009 \times 38.169$$

$$Q_{in} = 15251.84 - 864.83 - 13703.01$$

$$\boxed{Q_{in} = 684\ \text{Btu}}$$

Step 9 of 9

(c) Entropy balance is given by

$$S_{in} - S_{out} + S_{gen} = \Delta S_{sys}$$

$$\frac{Q_{in}}{T_{surr}} - m_{in} \times s_{in} + S_{gen} = m_2 \times s_2 - m_1 \times s_1$$

$$S_{gen} = m_2 \times s_2 - m_1 \times s_1 - \frac{Q_{in}}{T_{surr}} + m_{in} \times s_{in}$$

$$S_{gen} = 367.647 \times 0.08589 - 8.368 \times 0.2204 - \frac{684}{560} + 359.009 \times 0.07934$$

$$S_{gen} = 31.58 - 1.84 - 1.22 + 28.28$$

$$\boxed{S_{gen} = 56.8\ \text{Btu/lbm.R}}$$

The mass input to the tank is 359.009 ft , heat transfer to the surroundings is 684 Btu and the entropy generation is 56.8 Btu/lbm.R

Question: 8.201P

A passive solar house that is losing heat to the outdoors at 3°C at an average rate of 50,000 kJ/h is maintained at 22°C at all times during a winter night for 10 h. The house is to be heated by 50 glass containers, each containing 20 L of water that is heated to 80°C during the day by absorbing solar energy. A thermostat controlled 15 kW backup electric resistance heater turns on whenever necessary to keep the house at 22°C. Determine how long the electric heating system was on that night and the amount of entropy generated during the night.

Step 1 of 4

Given data:

The quantity of heat rate losing per hour, $Q_{out} = 50000\ \text{kJ/h}$

Temperature of the water, maintained in the house during all times of winter, $T_2 = 22°\text{C}$

Each volume of the water containing in glass containers, $V = 20\ \text{L}$

Work done by the electric resistance, $\dot{W}_{in} = 15\ \text{kW}$

Water is initially at temperature, $T_1 = 80°\text{C}$

Surrounding temperature of the house,

$T_{surr} = 3°\text{C}$

$= 276\ \text{K}$

Step 2 of 4

From the properties of common liquids, solids, and foods table,

The density of water, $\rho = 1\ \text{kg/L}$

Specific weight of the water at room temperature, $c = 4.18\ \text{kJ/kg}\cdot°\text{C}$

Mass of the water containing in the 50 glass containers is given by

$m = \rho V$

$= 1\ \text{kg/L} \times (50 \times 20\ \text{L})$

$= 1000\ \text{kg}$

Step 3 of 4

The amount of time taken by the electric system is determined from the energy balance equation of the solar house is given by

Net energy transfer by heat, work and mass is equal to change in the internal, kinetic, potential energies

$$E_{in} - E_{out} = \Delta E_{system}$$

$$(\dot{W}_{in})\Delta t - Q_{out} = \Delta U$$

$$= (\Delta U)_{water} + (\Delta U)_{air}$$

$$= (\Delta U)_{water} + 0$$

$$(\dot{W}_{in})\Delta t - Q_{out} = mc(T_2 - T_1)$$

$$(15\text{kJ/s})\Delta t - (50000\ \text{kJ/h})(10\ \text{h}) = 1000\ \text{kg} \times 4.18\ \text{kJ/kg}\cdot°\text{C}(22°\text{C} - 80°\text{C})$$

$$(15\text{kJ/s})\Delta t = -242440 + 500000$$

$$\Delta t = \frac{257560}{15}$$

$$= 17170.67\ \text{s}$$

$$= \frac{17170.67}{3600}$$

$$= 4.769\ \text{h}$$

Therefore the electric heating system was on that night for a period of time is $\boxed{\Delta t = 4.77\ \text{h}}$

Step 4 of 4

The entropy generated during this process is determined by applying the entropy balance on the house during the night at all times of the winter. The entropy generated for the house on the during the night is given by

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta S_{system}$$

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta S_{water} + \Delta S_{air}$$

$$-\frac{Q_{out}}{T_{surr}} + \dot{S}_{gen} = \left(mc\ln\frac{T_2}{T_1}\right)_{water} + 0$$

$$\dot{S}_{gen} = \left(mc\ln\frac{T_2}{T_1}\right)_{water} + \frac{Q_{out}}{T_{surr}}$$

$$= \left(1000\text{kg} \times 4.18\ \text{kJ/kg}\cdot°\text{C} \times \ln\frac{295\ \text{K}}{353\ \text{K}}\right) + \frac{(50000\ \text{kJ/h})(10\ \text{h})}{276\ \text{K}}$$

$$= -750.28 + 1811.59$$

$$= 1061.31\ \text{kJ/K}$$

Therefore, the amount of entropy generated for the house during night is $\boxed{\dot{S}_{gen} = 1061.31\ \text{kJ/K}}$

CHAPTER 09
SOLUTIONS

Question: 9.001P

How does the thermal efficiency of an ideal cycle, in general, compare to that of a Carnot cycle operating between the same temperature limits?

Step 1 of 1

The Carnot cycle have the maximum thermal efficiency of all heat engines operating between the same temperature limits.

An ideal cycle is a cycle that delivers work without the loss of heat.

It is impossible to construct a cycle with 100% efficiency as there are always losses involved as per the second law of thermodynamics. The Carnot cycle is regarded as the ideal cycle in thermodynamics and as such, the efficiencies of an ideal cycle and the Carnot cycle are the same.

Question: 9.002P

What does the area enclosed by the cycle represent on a *P-V* diagram? How about on a *T-s* diagram?

Step 1 of 1

On both P-v and T-s diagram, the area enclosed by the process curve represents the net work of the cycle.

Question: 9.003P

What are the air-standard assumptions?

Step 1 of 1

The air standard assumptions are:

(1) The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.

(2) All the processes that make up the cycle are internally reversible.

(3)The combustion process is replaced by a heat-addition process from an external source.

(4) The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

Question: 9.004P

What is the difference between air-standard assumptions and the cold-air-standard assumptions?

Step 1 of 1

The air standard assumptions are

a) The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
b) All the processes that make up the cycle are internally reversible.
c) The combustion process is replaced by a heat addition process from an external source.
d) The exhaust process is replaced by a heat rejection process that restores the working fluid to its initial state.

Cycles for which the air standard assumptions are applicable are known as air standard cycle

To simplify the analysis even more is that air has constant specific heats whose values are determined at room temperature. When this assumption is utilized, the air standard assumptions are called the cold air standard assumptions.

Question: 9.005P

What is the difference between the clearance volume and the displacement volume of reciprocating engines?

Step 1 of 1

The minimum volume formed in the cylinder when the piston is at TDC is called the clearance volume.

The volume displaced by the piston as it moves between TDC and BDC is called the displacement volume.

Question: 9.006P

Define the compression ratio for reciprocating engines.

Step 1 of 1

The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the compression ratio r of the engine.

$$r = \frac{V_{max}}{V_{min}}$$

$$r = \frac{V_{BDC}}{V_{TDC}}$$

Question: 9.007P

How is the mean effective pressure for reciprocating engines defined?

Step 1 of 1

It is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.

$W_{net} = \text{MEP} \times \text{displacement volume}$

$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}}$$

Question: 9.008P

Can the mean effective pressure of an automobile engine in operation be less than the atmospheric pressure?

Step 1 of 1

2760-9-10P SA: 2981

Yes, the MEP can be less than the atmospheric pressure.

Question: 9.009P

As a car gets older, will its compression ratio change? How about the mean effective pressure?

Step 1 of 2

Assuming there is no accumulation of carbon deposits on piston face the compression ratio remains same. If there is a carbon deposit on the face of the piston, the compression ratio of the engine increases.

Step 2 of 2

The mean effective pressure is mainly a function of swept volume of the cylinder which may increase due to wear and tear in the cylinder and piston. Consider the equation for the mean effective pressure.

$$MEP = \frac{W_{net}}{V_1 - V_2}$$

Here, V_1 is the swept volume and V_2 is the clearance volume.

From the above equation it is clear that the mean effective pressure reduces due to change in volume of the cylinder.

Therefore, as the car gets older, the compression remains same or increases and the mean effective pressure will reduce.

.

Question: 9.010P

What is the difference between spark-ignition and I compression-ignition engines?

Step 1 of 1

In spark ignition engines the fuel air mixture is compressed to a temperature that is below the auto ignition temperature of the fuel, and the combustion takes place by the ignition of fuel and air mixture using a spark plug. SI engines are usually petrol engines.
In a compression ignition engine, the air is compressed to high temperature that is above the auto ignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. CI engines are usually diesel engines.
Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines.

Question: 9.011P

Define the following terms related to reciprocating engines: stroke, bore, top dead center, and clearance volume.

Step 1 of 1

Stroke: The distance between the top dead center (TDC) and the bottom bead center (BDC) is the largest distance that the piston can travel in one direction, and it is called the stroke

Bore: The diameter of the piston is called the bore

Top dead center: The position of the cylinder when it is at the top part near to the valves or ports.

Clearance volume: the minimum volume formed in the cylinder when the piston is at TDC is called the clearance volume.

Question: 9.012P

What is the maximum possible thermal efficiency of a gas power cycle when using thermal energy reservoirs at 1 940°F and 40°F?

Step 1 of 4

Given,

The higher temperature limit, $T_H = 940^0\text{F}$

The lower temperature limit, $T_L = 40^0\text{F}$

The conversion from Degrees Fahrenheit to Degree Kelvin is

$$T(K) = \frac{5}{9}\left[T(^\circ\text{F}) - 32\right] + 273$$

Hence higher temperature limit in K is,

$$T_H = \frac{5}{9}[940 - 32] + 273$$

$$T_H = 504.444 + 273$$

$$T_H = 777.444^\circ\text{C}$$

Hence lower temperature limit,

$$T_H = \frac{5}{9}[40 - 32] + 273$$

$$T_H = 4.444 + 273$$

$$T_H = 277.444^\circ\text{C}$$

Step 2 of 4

The heat input to a cycle is $q_{in} = T_H (ds)_H$

And heat output from a cycle is $q_{out} = T_L (ds)_L$

As the carnot cycle operates with two isentropic processes and two isothermal processes,

The change in entropy remains constant for both heat transfer processes so $(ds)_H = (ds)_L$

Hence $q_{in} = T_H (ds)$ and

$$q_{out} = T_L (ds)$$

Step 3 of 4

Carnot Efficiency is given by

$$\eta_{carnot} = \frac{w_{net}}{q_{in}}$$

$$\eta_{carnot} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$\eta_{carnot} = 1 - \frac{q_{out}}{q_{in}}$$

$$\eta_{carnot} = 1 - \frac{T_L (ds)}{T_H (ds)}$$

Step 4 of 4

Substituting the values of temperatures, we get

$$\eta_{carnot} = \left(1 - \frac{T_L}{T_H}\right) \times 100$$

$$\eta_{carnot} = \left(1 - \frac{277.444}{777.444}\right) \times 100$$

$$\eta_{carnot} = (1 - 0.3568) \times 100$$

$$\eta_{carnot} = 64.3\ \%$$

Hence efficiency of carnot cycle is $\boxed{\eta_{carnot} = 64.3\ \%}$

Question:

An air-standard cycle with constant specific heats is .1 executed in a closed piston-cylinder system and is composed of the following three processes:

1-2 Isentropic compression with a compression ratio $r = 6$, $T1$,=27°C, and $P1 = 100$ kPa

2-3 Constant pressure heat addition

3-1 Constant volume heat rejection

Assume air has constant properties with cv= 0.718 kJ/kg.K, cp = 1.005 kJ/kg·K. R =0.287 kj/kg.K, and k= 1.4.

(*a*) Sketch the *P-v* and *T-s* diagrams for this cycle.

(*b*) Determine the back work ratio for this cycle.

Step 1 of 9

(a)

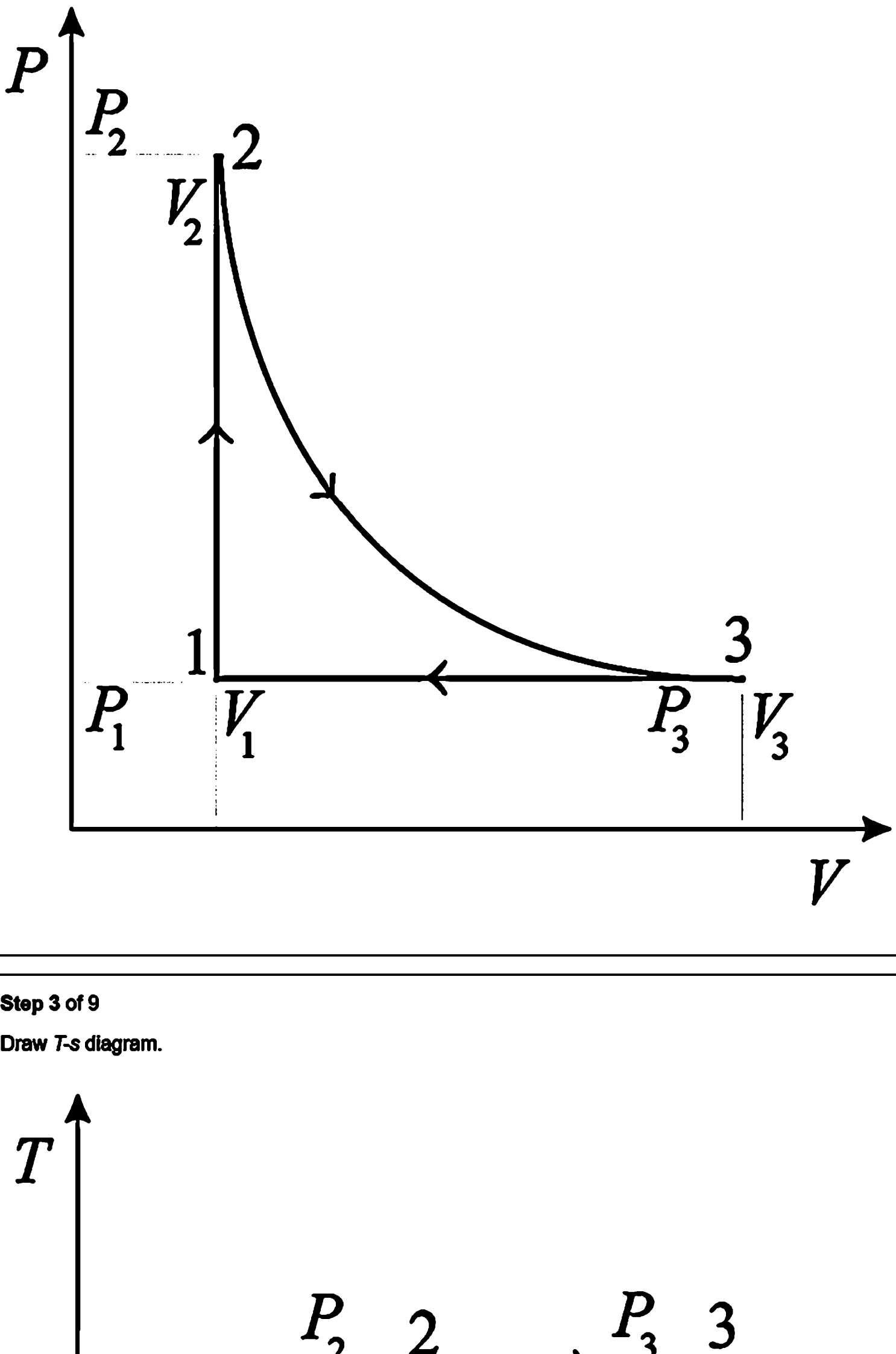
The process 1-2 is of constant volume. Hence the process can be represented as a vertical line in *P-V* diagram and as a curve on *T-s* diagram.

The process 2-3 is a isothermal process. Hence the process can be represented as a parabolic curve on *P-V* diagram and as a horizontal line in *T-s* diagram.

The process 3-1 is of constant pressure. Hence the process can be represented as a horizontal line in *P-V* diagram and as a curve in *T-s* diagram.

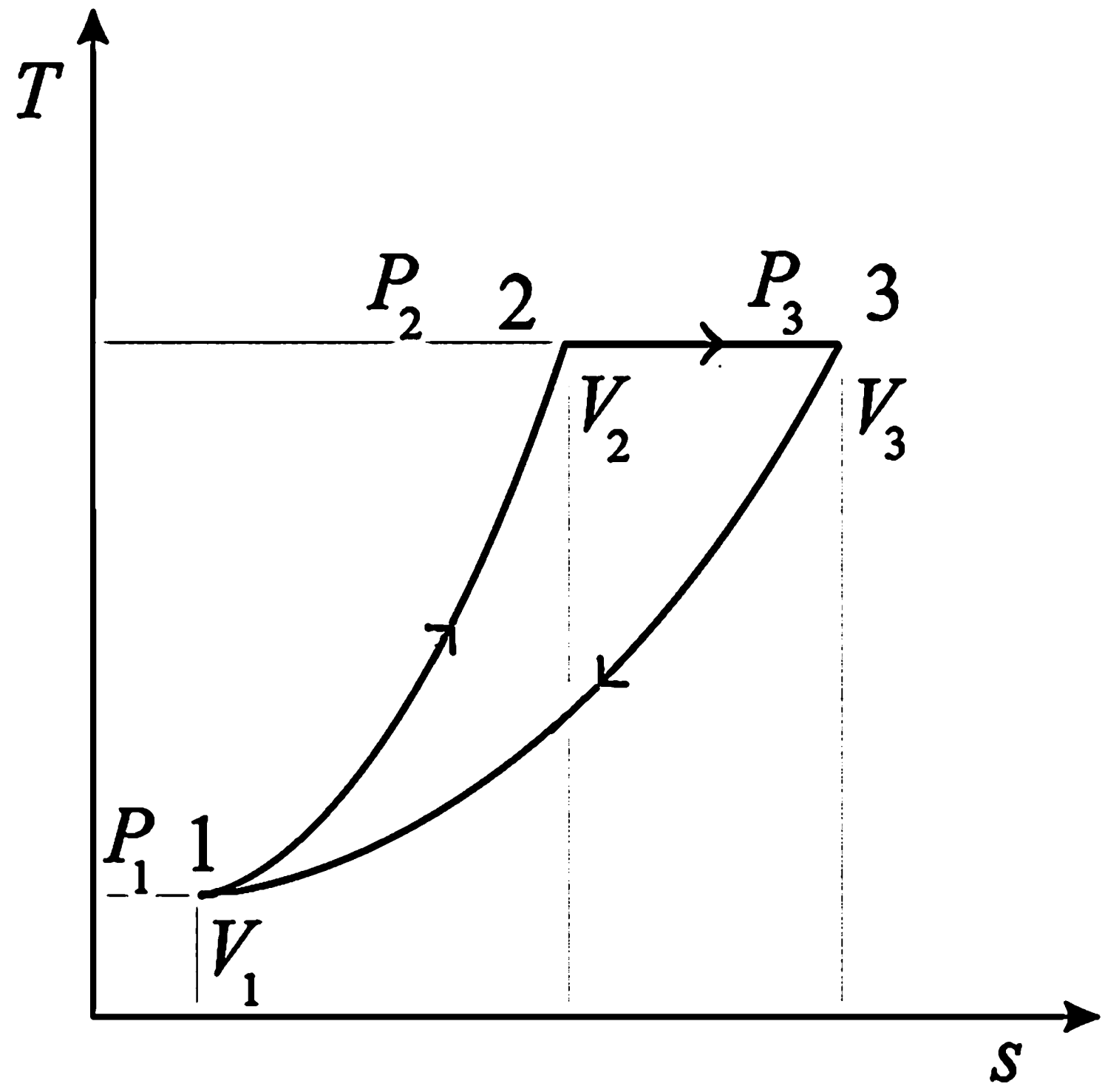
Step 2 of 9

Draw *P-V* diagram.



Step 3 of 9

Draw *T-s* diagram.



Step 4 of 9

(b)

Process 1-2 is Constant volume heat addition.

Find the temperature at state 2 using the ideal gas relations.

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

$$T_2 = T_1 \frac{P_2}{P_1}$$

Find the temperature at state 2 using the ideal gas relations.

$$\frac{P_1}{T_1}=\frac{P_2}{T_2}$$

$$T_2=T_1\frac{P_2}{P_1}$$

Here, T_1 is the temperature at state 1, P_1 is the pressure at state 1 and P_2 is the pressure at state 2.

Substitute 300 K for T_1, 100 kPa for P_1 and 700 kPa for P_2.

$$T_2=300\times\frac{700}{100}$$
$$=2100\ \text{K}$$

Step 5 of 9

For constant volume process, ${}_1W_2=0$.

Find the Heat transfer in 1-2 process using first law of thermodynamics.

$${}_1Q_2={}_1U_2+{}_1W_2$$
$${}_1Q_2=c_v(T_2-T_1)+{}_1W_2$$

Here, c_v is the specific heat constant at constant volume.

Substitute $0.718\,\text{kJ/kg}\cdot\text{K}$ for c_v, 0 for ${}_1W_2$, 300 K for T_1, and $2100\ \text{K}$ for T_2.

$${}_1Q_2=0.718\times(2100-300)$$
$$=1292.4\ \text{kJ/kg}$$

Step 6 of 9

Process 2-3 is Isothermal Expansion.
Find the pressure at state 3 using the ideal gas relation.

$$P_2V_2=P_3V_3$$

$$P_3=\frac{P_2V_2}{V_3}$$

Here, V_2 is the volume at state 2 and V_3 is the volume at state 3.

Substitute $7V_2$ for V_3 and 700 kPa for P_2.

$$P_3=\frac{700\times V_2}{7V_2}$$
$$=100\ \text{kPa}$$

Step 7 of 9

Find the work done in 2-3 process.

$${}_2W_3=P_2V_2\ln\left(\frac{V_3}{V_2}\right)$$
$$=RT_2\ln\left(\frac{V_3}{V_2}\right)$$

Here, R is the universal gas constant.

Substitute $7V_2$ for V_3, $2100\ \text{K}$ for T_2, $0.287\,\text{kJ/kg}\cdot\text{K}$ for R, and 700 kPa for P_2.

$${}_2W_3=0.287\times2100\times\ln\left(\frac{7V_2}{V_2}\right)$$
$$=1172.8\ \text{kJ/kg}$$

For Isothermal Expansion, ${}_1U_2=0$.

Find the Heat transfer in 2-3 process using first law of thermodynamics.

${}_2Q_3={}_2U_3+{}_2W_3$ Substitute 0 for ${}_1U_2$ and $1172.8\ \text{kJ/kg}$ for ${}_2W_3$.

$${}_2Q_3=0+1172.8$$
$$=1172.8\ \text{kJ/kg}$$

Step 8 of 9

Process 3-1 is constant pressure heat rejection.

Find the temperature at state 3 using the ideal gas relations.

$$\frac{T_3}{T_1}=\frac{V_3}{V_1}$$

$$T_3=T_1\left(\frac{V_3}{V_1}\right)$$

Substitute 300 K for T_1 and $7V_1$ for V_3.

$$T_3=300\times\frac{7V_1}{V_1}$$
$$=2100\ \text{K}$$

Find the work done in 3-1 process.

$${}_1W_3=P_3(V_1-V_3)$$
$$=R(T_1-T_3)$$

Substitute 300 K for T_1, $2100\ \text{K}$ for T_3 and $0.287\,\text{kJ/kg}\cdot\text{K}$ for R.

$${}_1W_3=0.287\times(300-2100)$$
$$=-516.6\ \text{kJ/kg}$$

Step 9 of 9

Find the back work ratio.

$$r_{bw}=\frac{\text{Comperssion work}}{}$$

Step 9 of 9

Find the back work ratio.

$$r_{bw} = \frac{\text{Comperssion work}}{\text{Expansion work}}$$
$$= \frac{W_{3-1}}{W_{2-3}}$$

Substitute $516.6\ \text{kJ/kg}$ for W_{3-1} and $1172.8\ \text{kJ/kg}$ for ${}_2W_3$.

$$r_{bw} = \frac{516.6}{1172.8}$$
$$= 0.44$$

Therefore, the back work ratio is $\boxed{0.44}$.

(c)

Find the thermal efficiency.

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$
$$= \frac{{}_2W_3 - {}_3W_1}{{}_1Q_2 + {}_2Q_3}$$

Substitute $1292.4\ \text{kJ/kg}$ for ${}_1Q_2$, $1172.8\ \text{kJ/kg}$ for ${}_2Q_3$, $516.6\ \text{kJ/kg}$ for W_{3-1} and $1172.8\ \text{kJ/kg}$ for ${}_2W_3$.

$$\eta_{th} = \frac{1172.8 - 516.6}{1292.4 + 1172.8}$$
$$= 0.266$$
$$= 26.6\%$$

Therefore, the thermal Efficiency of the cycle is $\boxed{26.6\%}$

Question:

An air-standard cycle is executed within a closed I piston-cylinder system and consists of three processes as follows:

1-2 *V*= *constant* heat addition from 100 kPa and 27°C to 700 kPa

2-3 Isothermal expansion until *V*3 = 7*V*2

3-1 *P* = *constant* heat rejection to the initial state

Assume air has constant properties with *cv*=0.718 kJ/kg·K, *cp*= 1.005 kJ/kg·K, *R* =0.287 kJ/kg·K, and *k* = 1.4.

Sketch the *P-v* and *T-s* diagrams for the cycle.

(b) Determine the ratio of the compression work to the expansion work (the back work ratio).

(c) Determine the cycle thermal efficiency.

Step 1 of 9

Given,

Temperature of air before compression $T_1 = 27^0\,\mathrm{C}$

$$T_1 = 300\,\mathrm{K}$$

Pressure of air before compression $P_1 = 100\ \mathrm{kPa}$

Pressure of air after compression $P_2 = 700\ \mathrm{kPa}$

Volume relation, $V_3 = 7V_2$

Step 2 of 9

(a)

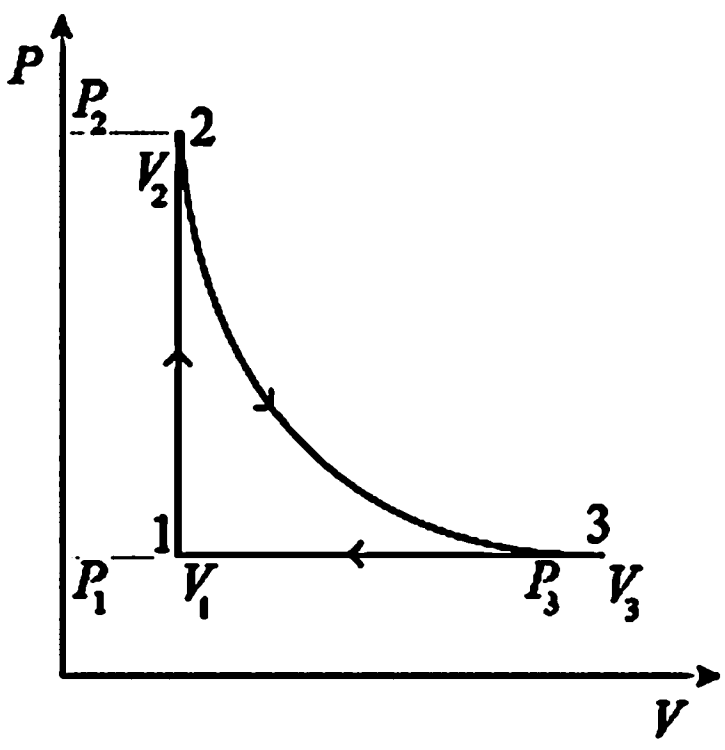
The process from 1-2 is of constant volume. Hence the process can be represented as a vertical line in *P-V* diagram and as a curve on *T-s* diagram.

The process from 2-3 is a isothermal process. Hence the process can be represented as a parabolic curve on *P-V* diagram and as a horizontal line in *T-s* diagram.

The process from 3-1 is of constant pressure. Hence the process can be represented as a horizontal line in *P-V* diagram and as a curve in *T-s* diagram.

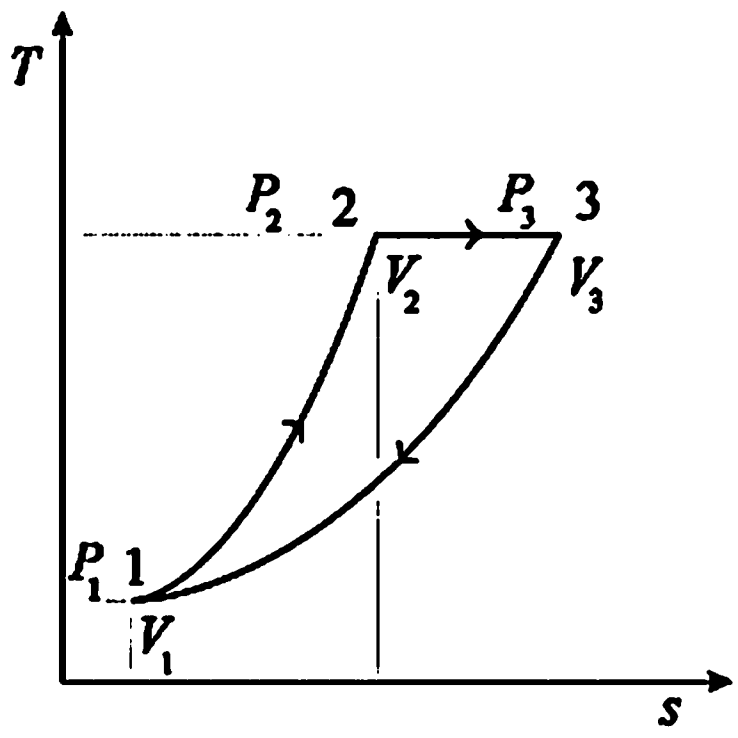
Step 3 of 9

P-V diagram:



Step 4 of 9

T-s diagram:



Step 5 of 9

(b)

Process 1-2 is of constant volume hence it follows, the relation, $\frac{P_1}{T_1} = \frac{P_2}{T_2}$

$$T_2 = T_1\frac{P_2}{P_1}$$

$$T_2 = 300\times\frac{700}{100}$$

$$T_2 = 2100\ \mathrm{K}$$

In a constant volume process, $dV = 0$. So ${}_1W_2 = 0$. From energy balance, ${}_1Q_2 = {}_1U_2 + {}_1W_2$

$${}_1Q_2 = {}_1U_2 + 0$$

$${}_1Q_2 = c_v(dT)$$

$${}_1Q_2 = c_v(T_2 - T_1)$$

$${}_1Q_2 = 0.718\times(2100-300)$$

$${}_1Q_2 = 1292.4\ \mathrm{kJ/kg}$$

Step 6 of 9

Process 2-3 is Isothermal Expansion. Hence it follows, $P_2V_2 = P_3V_3$

$$P_3 = \frac{P_2V_2}{V_3}$$

$$P_3 = \frac{700\times V_2}{7V_2}$$

$$P_3 = 100\ \mathrm{kPa}$$

Work done is ${}_2W_3 = P_2V_2\ln\left(\frac{V_3}{V_2}\right)$

$${}_2W_3 = RT_2\times\ln\left(\frac{7V_2}{V_2}\right)$$

$P_3 = 100$ kPa

Work done is $${}_2W_3 = P_2V_2 \ln\left(\frac{V_3}{V_2}\right)$$

$${}_2W_3 = RT_2 \times \ln\left(\frac{7V_2}{V_2}\right)$$

$${}_2W_3 = 0.287 \times 2100 \times \ln\left(\frac{7V_2}{V_2}\right)$$

Step 7 of 9

$${}_2W_3 = 1172.8 \text{ kJ/kg}$$

In an isothermal process, $dT = 0$. So ${}_1U_2 = 0$. From energy balance, ${}_2Q_3 = {}_2U_3 + {}_2W_3$

$${}_2Q_3 = 0 + {}_2W_3$$

$${}_2Q_3 = {}_2W_3$$

$${}_2Q_3 = 1172.8 \text{ kJ/kg}$$

Process 3-1 is of constant pressure. Hence

$${}_1W_3 = P_3(V_1 - V_3)$$

$${}_1W_3 = R(T_1 - T_3)$$

$${}_1W_3 = 0.287 \times (300 - 2100)$$

$${}_1W_3 = -516.6 \text{ kJ/kg}$$

Step 8 of 9

$$\textit{Back work ratio} = \frac{\textit{Comperssion work}}{\textit{Expansion work}}$$

$$\textit{Back work ratio} = \frac{W_{3\text{-}1}}{W_{2\text{-}3}}$$

$$\textit{Back work ratio} = \frac{516.6}{1172.8}$$

$$\textit{Back work ratio} = 0.44$$

Hence back work ratio is $\boxed{\textit{Back work ratio} = 0.44}$

Step 9 of 9

(c)

Thermal efficiency, $\eta_{th} = \frac{W_{net}}{Q_{in}}$

$$\eta_{th} = \frac{{}_2W_3 - {}_3W_1}{{}_1Q_2 + {}_2Q_3}$$

$$\eta_{th} = \frac{1172.8 - 516.6}{1292.4 + 1172.8}$$

$$\eta_{th} = 0.266$$

$$\eta_{th} = 26.6\%$$

Thermal Efficiency of the cycle is $\boxed{\eta_{th} = 26.6\%}$

Question:

An air-standard cycle with variable specific heats is executed in a closed system and is composed of the following four processes:

1–2 Isentropic compression from 100 kPa and 22°C to 600 kPa

2-3 *V - constant* heat addition to 1500 K

3-4 Isentropic expansion to 100 kPa

4-4 *P=constant* heat rejection to initial state

(a) Show the cycle on *P-V* and *T-s* diagrams.

(b) Calculate the net work output per unit mass.

(c) Determine the thermal efficiency.

Step 1 of 12

Given data:

Temperature before compression of the given system, $T_1 = 22°C$

Pressure before compression of the given system, $p_1 = 100\,kPa$

Pressure after compression of the given system, $P_2 = 600\,kPa$

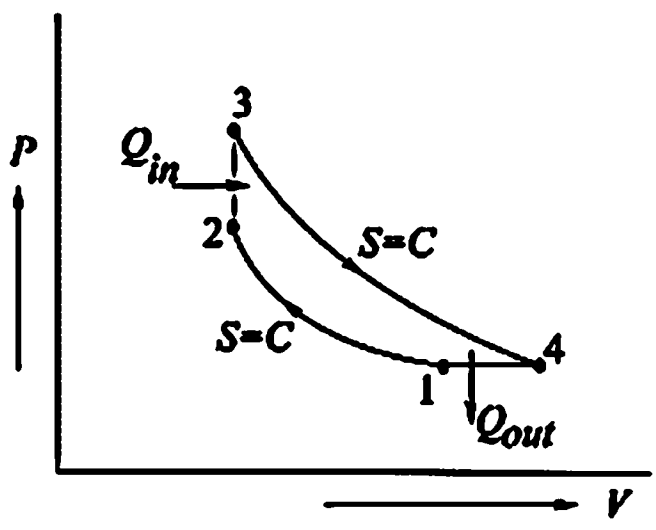
Temperature after heat addition, $T_3 = 1500\,K$

Pressure after expansion of the given system, $p_4 = 100\,kPa$

Step 2 of 12

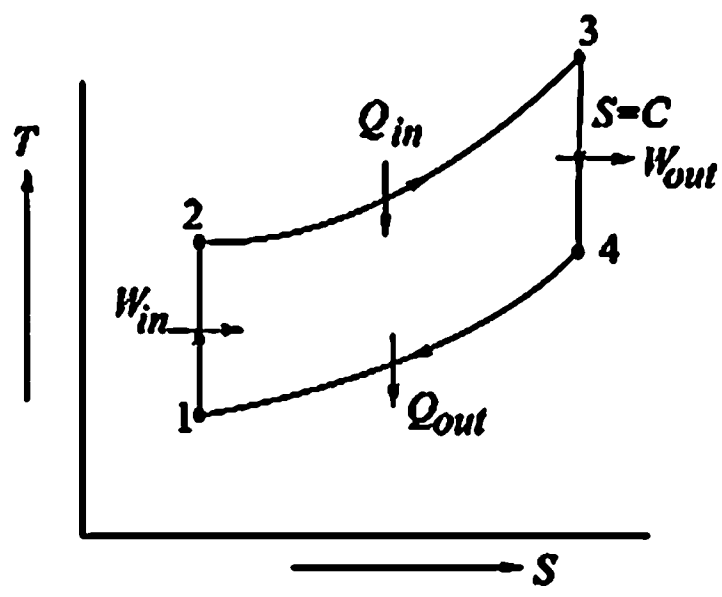
(a)

PV- diagram of the given system:



Step 3 of 12

TS- diagram for the given system:



Step 4 of 12

(b)

Since we are considering variable specific heats the specific heats at constant pressure and constant volume will be different at different temperatures.

At $T_1 = 22°C = 295\,K$ from ideal gas properties of air, we have

$h_1 = 295.17\ kJ/kg$

$u_1 = 210.49\ kJ/kg$

$P_{r1} = 1.3068$

Step 5 of 12

For the isentropic compression process 1-2, we have

$$\frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1}$$

$$\frac{P_{r2}}{1.3068} = \frac{600}{100}$$

$$P_{r2} = 7.8408$$

Step 6 of 12

At $P_{r2} = 7.8408$ $P_{r_1} = 11.088$ from ideal gas properties of air by using interpolation

$$\frac{T_2 - 490}{500 - 490} = \frac{7.8408 - 7.824}{8.411 - 7.824}$$

$$T_2 = 490.286\,K$$

Similarly,

$$\frac{u_2 - 344.7}{352.08 - 344.7} = \frac{7.8408 - 7.824}{8.411 - 7.824}$$

$$u_2 = 344.91\,kJ/kg$$

Step 7 of 12

For the process constant volume heat addition 2-3, we have

$$\frac{P_2}{P_3} = \frac{T_2}{T_3}$$

$$P_3 = P_2 \frac{T_3}{T_2}$$

$$= (600)\frac{1500}{490.286}$$

$$P_3 = 1835.66\ kPa$$

At $T_3 = 1500K$ from ideal gas properties of air, we have

$h_3 = 1635.97\ kJ/kg$

$u_3 = 1205.41\ kJ/kg$

$P_{r3} = 601.9$

$h_3 = 1635.97$ kJ/kg

$u_3 = 1205.41$ kJ/kg

$P_{r3} = 601.9$

Step 8 of 12

For the isentropic expansion process 3-4, we have

$$\frac{P_{r4}}{P_{r3}} = \frac{P_4}{P_3}$$

$$P_{r4} = \frac{P_4}{P_3} P_{r3}$$

$$= \frac{100}{1835.66} \times 601.9$$

$$P_{r4} = 32.789$$

At $P_{r4} = 32.789$ from ideal gas properties of air, we have

$$\frac{h_4 - 734.82}{745.62 - 734.82} = \frac{32.789 - 32.02}{33.72 - 32.02}$$

$h_4 = 739.71$ kJ/kg

Step 9 of 12

Process 2-3 is constant volume heat addition

$$q_{in} = u_3 - u_2$$

$$= 1205.41 - 344.91$$

$$= 860.5$$

$q_{in} = 860.5$ kJ/kg

Step 10 of 12

Process 4-1 is heat rejection at constant pressure

$$q_{out} = h_4 - h_1$$

$$= 739.71 - 295.17$$

$$= 444.54$$

$q_{out} = 444.54$ kJ/kg

Step 11 of 12

Net work output of the given system per unit mass,

$$w_{net} = q_{in} - q_{out}$$

$$= 860.5 - 444.54$$

$$= 415.96 \text{ kJ/kg}$$

Therefore, Net work output of the given system per unit mass, $\boxed{w_{net} = 415.96 \text{ kJ/kg}}$

Step 12 of 12

(c)

Thermal efficiency of the given system,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{415.96}{860.5}$$

$$= 0.4834$$

Therefore, thermal efficiency of the given system, $\boxed{\eta_{th} = 48.34\%}$

Question:

Reconsider Prob. 9–15. Using EES (or other) software, study the effect of varying the temperature after the constant-volume heat addition from 1500 K to 2500 K. Plot the net work output and thermal efficiency as a function of the maximum temperature of the cycle. Plot the *T-s* and *P-V* diagrams for the cycle when the maximum temperature of the cycle is 1500 K.

Step 1 of 7

Given data:

Inlet pressure $P_1 = 100\,\text{kPa}$

Inlet temperature, $T_1 = 22°\text{C}$

Pressure at state 2, $P_2 = 600\,\text{kPa}$

Maximum temperature of the cycle, $T_3 = 1500\,\text{K}$

Step 2 of 7

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.

```
"Input Data"
T[1]=295 [K]
P[1]=100 [kPa]
P[2] = 600 [kPa]
"T[3]=1500 [K]"

"Analysis"
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
P[4] = 100 [kPa]
```

Step 3 of 7

```
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0 "isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
```

Step 4 of 7

```
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0 "constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=0.287*T[4]}

"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
```

Step 5 of 7

```
"Process 4-1 is constant pressure heat rejection"
{P[4]*v[4]/T[4]=P[1]*v[1]/T[1]}
"Conservation of energy for process 4 to 1"
q_41 -w_41 = DELTAu_41
w_41 =P[1]*(v[1]-v[4]) "constant pressure process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
```

Step 6 of 7

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed:

EES Limited Academic Version: C:\EES (LIMITED ACADEMIC)\HELLO.DTT - [Param

File Edit Search Options Calculate Tables Plots Windows Help

Table 1

1.10	1 W_{net}	2 $q_{in,total}$	3 η_{th}	4 T_3 [K]
Run 1	376.6	774.6	48.61	1500
Run 2	429.9	880.3	48.83	1611
Run 3	484	906.9	49.05	1722
Run 4	539.2	1094	49.27	1833
Run 5	594.8	1202	49.47	1944
Run 6	651	1311	49.66	2056
Run 7	707.9	1420	49.85	2167
Run 8	765.4	1530	50.04	2278
Run 9	823.5	1640	50.22	2389
Run 10	882.2	1750	50.4	2500

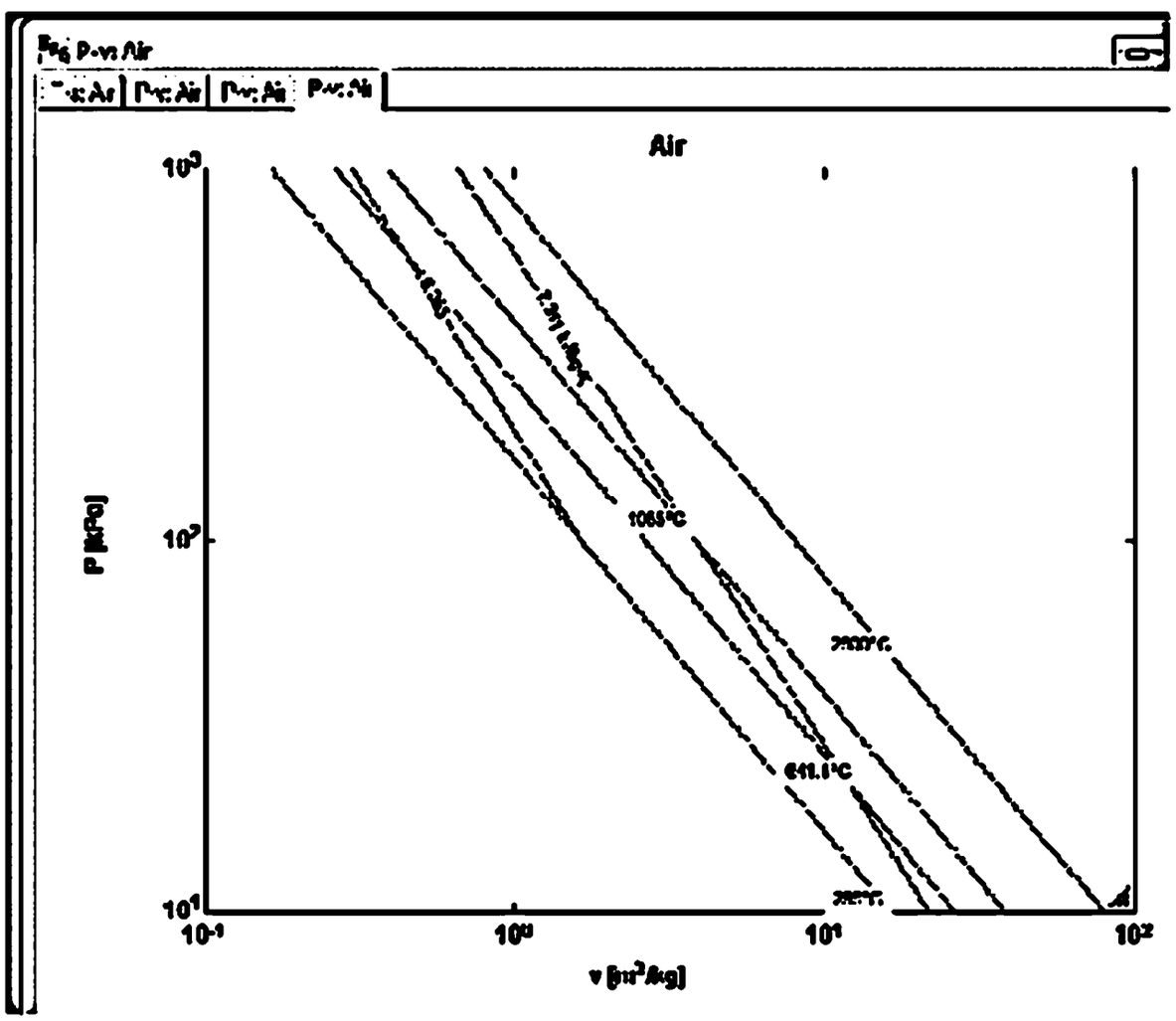
Run 8	765.4	1630	50.01	2276
Run 9	823.5	1640	50.22	2339
Run 10	882.2	1750	50.4	2500

g 016p pg 2

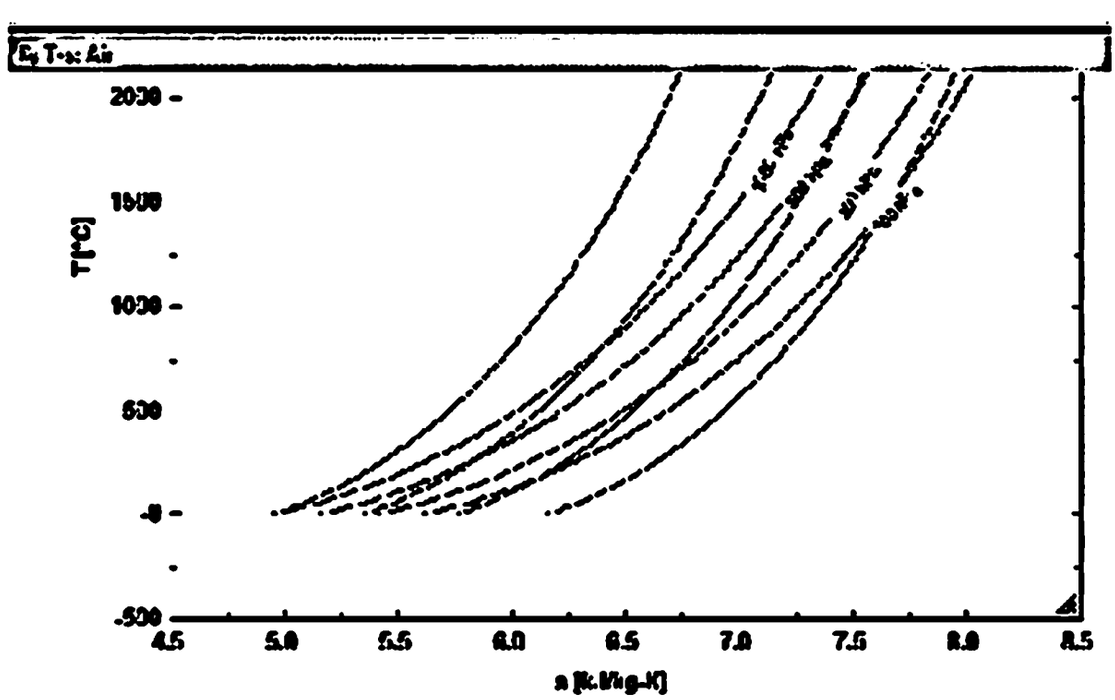
Step 7 of 7

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

By plotting the graph between pressure and volume:



By plotting the graph between temperature and entropy:



Question:

An air-standard cycle is executed in a closed system with 0.004 kg of air and consists of the following three processes:

1-2 Isentropic compression from 100 kPa and 27°C to 1 MPa

2-3 *P - constant* heat addition in the amount of 2.76 kJ

3-3*P* = *c*1*V* + *c*2 heat rejection to initial state (*c*1, and *c*2 are constants)

(a) Show the cycle on *P-v* and *T-s* diagrams.

(b) Calculate the heat rejected.

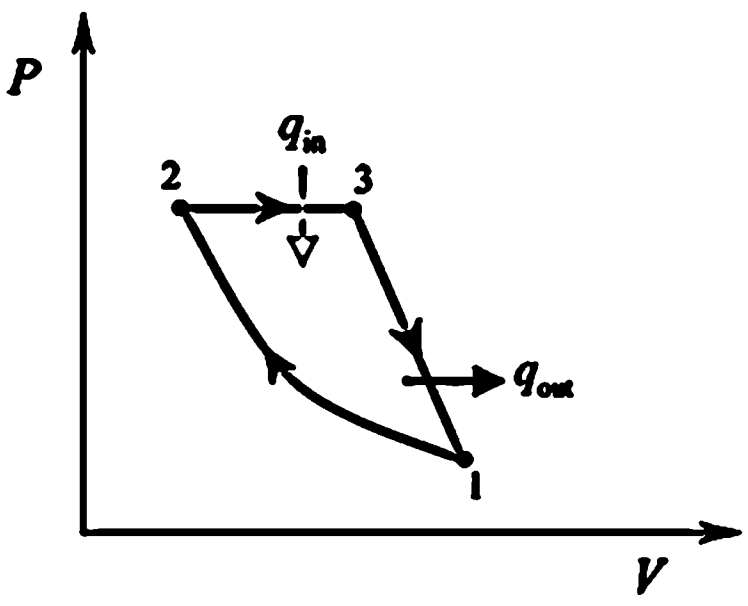
(c) Determine the thermal efficiency.

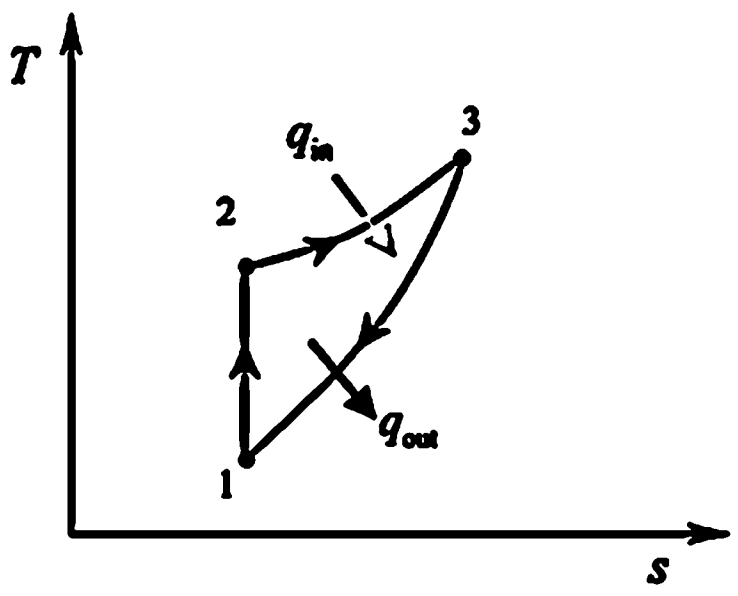
Assume constant specific heats at room temperature.

Step 1 of 8

(a)

Draw $P-V$ and $T-S$ diagrams of an air-standard cycle as follows:





Step 2 of 8

The process $1-2$ is an isentropic compression process.

Hence, the process is represented as a curve on the $P-V$ diagram and as a vertical line on $T-S$ diagram.

The process $2-3$ is of constant pressure heat addition process.

Hence, the process is represented as a horizontal line on the $P-V$ diagram and as a curve on $T-S$ diagram.

The process $3-1$ is a heat rejection process which follows the law, $P=c_1V+c_2$

Hence, the process is represented as a straight line on the $P-V$ diagram and as a curve on $T-S$ diagram.

Step 3 of 8

(b)

Obtain the properties of air from the **Table A-2a**.

At room temperature,

Gas constant, $R=0.287\,\text{kJ/kg}\cdot\text{K}$

Specific heats at constant pressure, $c_p=1.005\,\text{kJ/kg}\cdot\text{K}$

Specific heats at constant volume, $c_v=0.718\,\text{kJ/kg}\cdot\text{K}$

Ratio of specific heats (or) Specific heat ratio, $k=1.4$

Step 4 of 8

Consider the process $1-2\left(\text{Isentropic Compression process}\right)$:

Calculate the temperature at the end of compression in an isentropic compression process by using the following equation,

$$\frac{T_2}{T_1}=\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}$$

$$T_2=T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}$$

Here, the temperature of air before compression is T_1, the volume of air before compression is V_1, the volume of air after compression is V_2, the temperature of air at the end of compression is T_2, and the specific heat ratio is k.

Substitute 300 K for T_1, 1000 kPa for P_2, 100 kPa for P_1, and 1.4 for k.

$$\begin{aligned}T_2&=T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}\\&=300\left(\frac{1000}{100}\right)^{\left(\frac{1.4-1}{1.4}\right)}\\&=579.20\text{ K}\end{aligned}$$

Step 5 of 8

Write an expression for the heat input to the system during the constant pressure heat addition process $2-3$.

$$Q_{in,2\text{-}3}=mc_p\left(T_3-T_2\right)$$

Here, the mass of air is m, the temperature of air at the end of heat addition process is T_3 and the specific heat at constant pressure is c_p.

Substitute 0.004 kg for m, $1.005\text{ kJ/kg}\cdot\text{K}$ for c_p, 2.76 kJ for $Q_{in,2\text{-}3}$, and 579.20 K for T_2.

$$\begin{aligned}2.76&=0.004\times1.005\left(T_3-579.20\right)\\T_3-579.20&=686.56\\T_3&=1265.76\text{ K}\end{aligned}$$

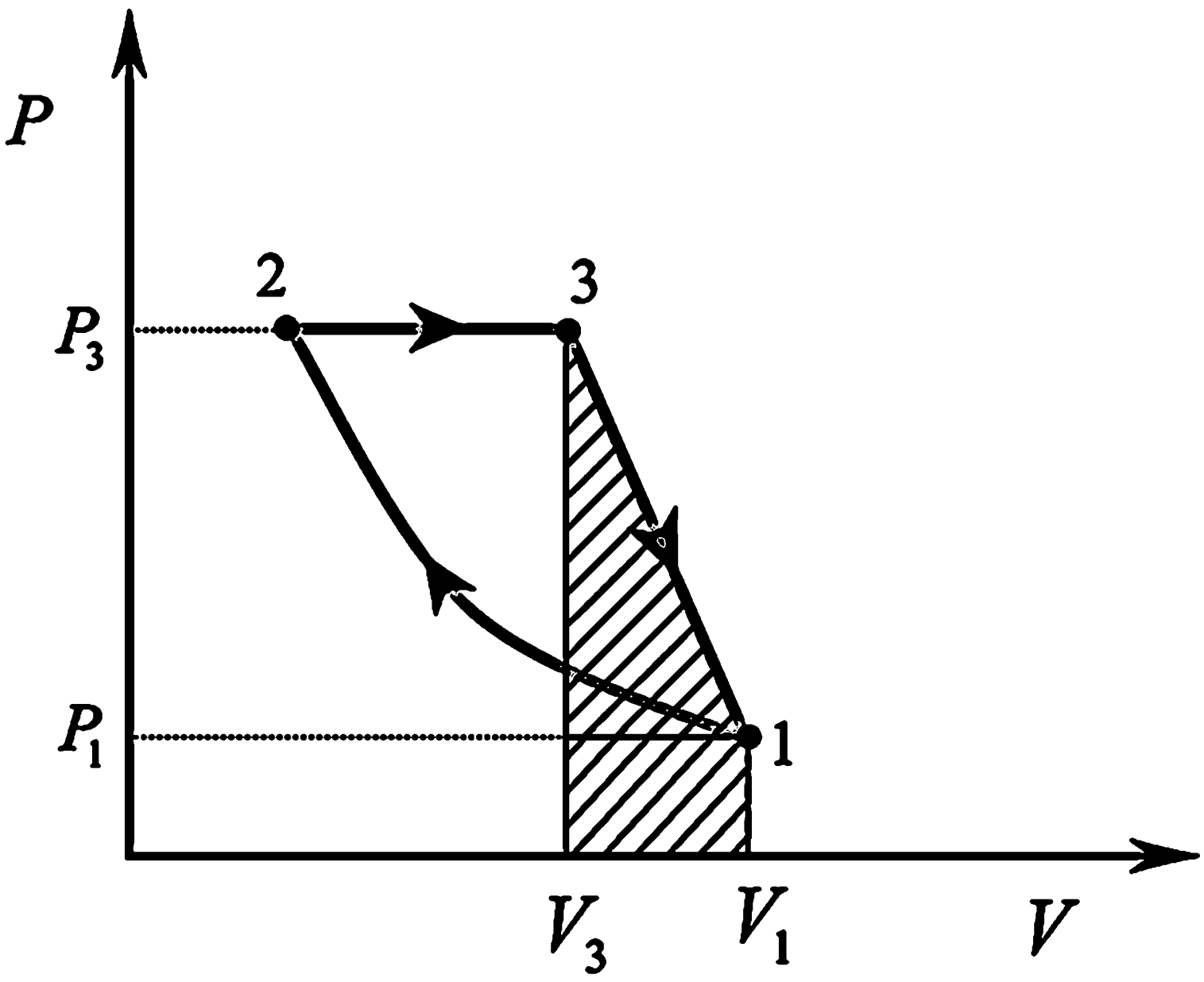
Substitute 0.004 kg for m, 1.005 kJ/kg·K for c_p, 2.76 kJ for $Q_{in,2-3}$, and 579.20 K for T_2.

$$2.76 = 0.004 \times 1.005 (T_3 - 579.20)$$

$$T_3 - 579.20 = 686.56$$

$$T_3 = 1265.76 \text{ K}$$

Draw the shaded region representing the work done during the process 3-1.



The process 3-1 is a straight line on the $P-V$ diagram. Thus, the area under the process curve 3-1 represents the work done during the process.

Step 6 of 8

Calculate the work done during the process 3-1.

$$W_{3-1} = \left[P_1(V_1 - V_3)\right] + \left[\frac{1}{2}(V_1 - V_3)(P_3 - P_1)\right]$$

$$= (V_1 - V_3)\left[P_1 + \left(\frac{P_3 - P_1}{2}\right)\right]$$

$$= (V_1 - V_3)\left[\frac{P_1 + P_3}{2}\right]$$

$$= \left[mR\left(\frac{T_1}{P_1} - \frac{T_3}{P_3}\right)\right]\left[\frac{P_1 + P_3}{2}\right] \qquad (\because PV = mRT)$$

Substitute 0.004 kg for m, 0.287 kJ/kg·K for R, 300 K for T_1, 1265.76 K for T_3, 1000 kPa for P_3, and 100 kPa for P_1.

$$W_{3-1} = \left[0.004 \times 0.287\left(\frac{300}{100} - \frac{1265.76}{1000}\right)\right]\left[\frac{100 + 1000}{2}\right]$$

$$= 1.0949 \text{ kJ}$$

Step 7 of 8

Write the energy balance equation for the process 3-1.

$$E_{in} - E_{out} = \Delta E_{system}$$

$$-Q_{out,3-1} - W_{3-1} = m(u_1 - u_3)$$

$$Q_{out,3-1} = -mc_v(T_1 - T_3) - W_{3-1}$$

Here, the specific heat at constant volume is c_v and the heat rejected during the process $3-1$ is $Q_{out,3-1}$.

Substitute 0.004 kg for m, 0.718 kJ/kg·K for c_v, 300 K for T_1, 1265.76 K for T_3, and 1.0949 kJ for W_{3-1}.

$$Q_{out,3-1} = -\left[0.004 \times 0.718(300 - 1265.76)\right] - 1.0949$$

$$= -\left[-2.7736\right] - 1.0949$$

$$= 1.6787 \text{ kJ}$$

Therefore, the heat rejected during the process $3-1$ is $\boxed{1.6787 \text{ kJ}}$.

Step 8 of 8

(c)

Calculate the thermal efficiency of the system by using the following equation,

$$\eta = \frac{Q_{in,2-3} - Q_{out,3-1}}{Q_{in,2-3}}$$

$$= 1 - \frac{Q_{out,3-1}}{Q_{in,2-3}}$$

Substitute 1.6787 kJ for $Q_{out,3-1}$ and 2.76 kJ for $Q_{in,2-3}$.

$$\eta = 1 - \frac{1.6787}{2.76}$$

$$= 0.3917$$

$$= 39.17\%$$

Therefore, thermal efficiency of the system is $\boxed{39.17\%}$.

Question:

An air-standard cycle with variable specific heats is executed in a closed system and is composed of the following four processes:

1–2 *V*= *constant* heat addition from 14.7 psia and 80°F in

the amount of 300 Btu/lbm

2-3 *P* = *constant* heat addition to 3200 R

3-4 Isentropic expansion to 14.7 psia

4-1 *P* = *constant* heat rejection to initial state

(*a*) Show the cycle on *P-V* and *T-s* diagrams.

(*b*) Calculate the total heat input per unit mass.

(*c*) Determine the thermal efficiency.

Step 1 of 14

Given

Pressure at state 1 $p_1 = 14.7\,\text{psia}$

Temperature at state 1 $T_1 = 80^0\text{F} = 540\text{R}$

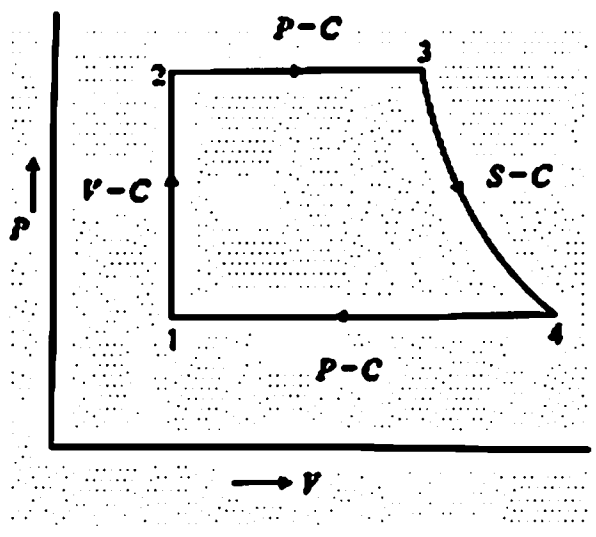
Heat addition during the process 1-2 is $q_{in1\text{-}2} = 300\,\text{Btu/lbm}$

Temperature at state 3 $T_3 = 3200\text{R}$

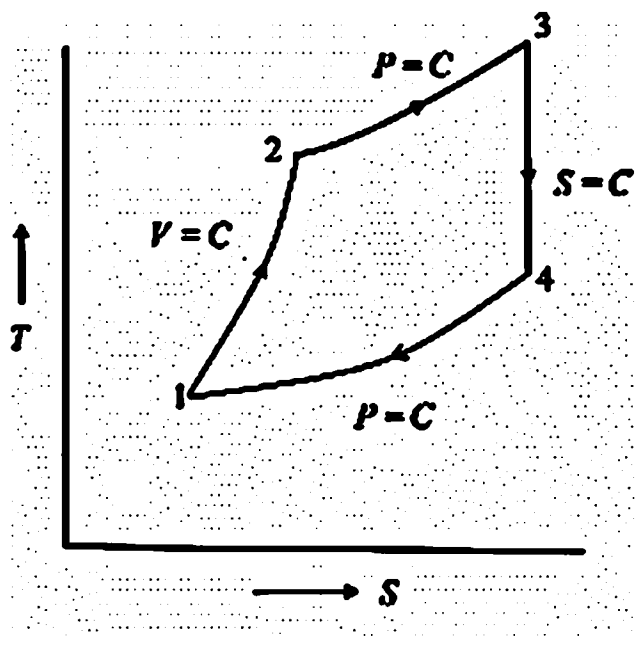
Pressure at state 4 $p_4 = 14.7\,\text{psia}$

Step 2 of 14

(a)



Step 3 of 14



Step 4 of 14

(b)

Since we are considering variable specific heats the specific heat at constant pressure and specific heat at constant volume will be different at different temperatures.

At $T_1 = 540\text{R}$ from ideal gas properties of air

$h_1 = 129.06\,\text{Btu/lbm}$

$u_1 = 92.04\,\text{Btu/lbm}$

$P_{r1} = 1.386$

Step 5 of 14

Process 1-2 is constant volume heat addition

$q_{in1\text{-}2} = u_2 - u_1$

$300 = u_2 - 92.04$

$u_2 = 392.04\,\text{Btu/lbm}$

Step 6 of 14

At $u_2 = 392.04\,\text{Btu/lbm}$ from ideal gas properties of air

$$T_2 = 2100 + (2150 - 2100)\left(\frac{392.04 - 388.60}{399.17 - 388.60}\right)$$

$T_2 = 2116.27\text{R}$

$$h_2 = 532.55 + (546.54 - 532.55)\left(\frac{392.04 - 388.60}{399.17 - 388.60}\right)$$

$h_2 = 537.103\,\text{Btu/lbm}$

Step 7 of 14

Process 1-2 is constant volume process so

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}$$

$$P_2 = P_1 \times \frac{T_2}{T_1}$$

$$P_2 = 14.7 \times \frac{2116.27}{540}$$

$P_2 = 57.609\,\text{psia}$

Step 8 of 14

At $T_3 = 3200\text{R}$ from ideal gas properties of air

$h_3 = 849.48\,\text{Btu/lbm}$

Step 8 of 14

At $T_3 = 3200\text{R}$ from ideal gas properties of air

$h_3 = 849.48\ \text{Btu/lbm}$

$P_{r3} = 1242$

Step 9 of 14

Process 3-4 is isentropic process

$$\frac{P_{r4}}{P_{r3}} = \frac{P_4}{P_3}$$

$$\frac{P_{r4}}{1242} = \frac{14.7}{57.609} \qquad (\text{Since } p_2 = p_3)$$

$$P_{r4} = 0.255 \times 1242$$

$$P_{r4} = 316.71$$

Step 10 of 14

At $P_{r4} = 316.71$ from ideal gas properties of air

$$h_4 = 588.82 + (603.00 - 588.82)\left(\frac{316.71 - 308.1}{336.8 - 308.1}\right)$$

$$h_4 = 593.074\ \text{Btu/lbm}$$

Step 11 of 14

The heat input in process 2-3

$$q_{in2\text{-}3} = h_3 - h_2$$

$$q_{in2\text{-}3} = 849.48 - 537.103$$

$$q_{in2\text{-}3} = 312.377\ \text{Btu/lbm}$$

Step 12 of 14

The total heat input

$$q_{in} = q_{in1\text{-}2} + q_{in2\text{-}3}$$

$$= 300 + 312.377$$

$$\boxed{q_{in} = 612.377\ \text{Btu/lbm}}$$

Step 13 of 14

(c)

The heat output in process 4-1

$$q_{out4\text{-}1} = h_4 - h_1$$

$$= 593.074 - 129.06$$

$$q_{out4\text{-}1} = 464.014\ \text{Btu/lbm}$$

Step 14 of 14

The thermal efficiency of the cycle

$$\eta_{th} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$= \frac{612.377 - 464.014}{612.377}$$

$$= 0.2422$$

$$\boxed{\eta_{th} = 24.22\%}$$

Question: 9.019P

Repeat Prob. 9-18E using constant specific heats at room temperature.

Step 1 of 11

Given

Pressure at state 1 $p_1 = 14.7 \text{psia}$

Temperature at state 1 $T_1 = 80^0\text{F} = 540\text{R}$

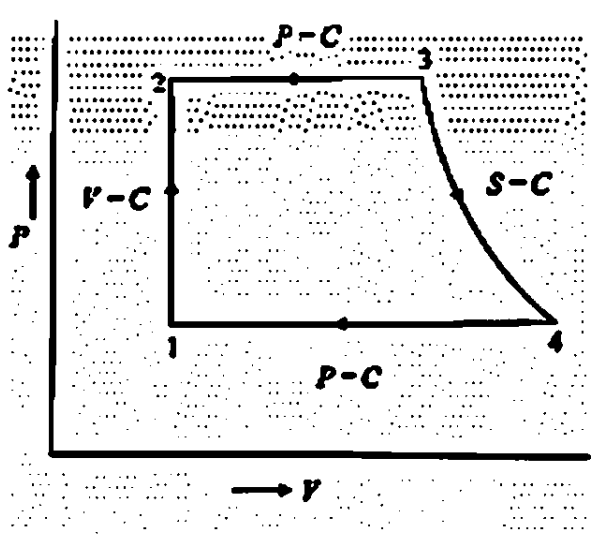
Heat addition during the process 1-2 is $q_{in1-2} = 300 \text{Btu/lbm}$

Temperature at state 3 $T_3 = 3200\text{R}$

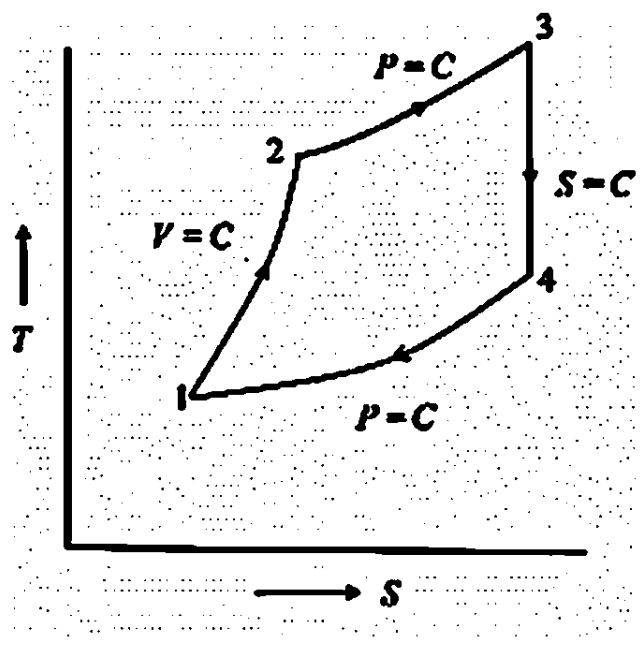
Pressure at state 4 $p_4 = 14.7 \text{psia}$

Step 2 of 11

(a)



Step 3 of 11



Step 4 of 11

(b)

Considering constant specific heats at room temperature

From table ideal gas specific heats of various common gases

$c_p = 0.24 \text{ Btu/lbm.R}$

$c_v = 0.171 \text{ Btu/lbm.R}$

$k = 1.4$

Step 5 of 11

Process 1-2 is constant volume heat addition

$q_{in1-2} = u_2 - u_1$

$q_{in1-2} = c_v (T_2 - T_1)$

$300 = (0.171)(T_2 - 540)$

$T_2 = 2294.38 \text{ R}$

Step 6 of 11

Process 1-2 is constant volume process

By using ideal gas relation

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}$$

$$P_2 = P_1 \times \frac{T_2}{T_1}$$

$$P_2 = 14.7 \times \frac{2294.38}{540}$$

$P_2 = 62.45 \text{ psia}$

Step 7 of 11

Process 2-3 is constant pressure heat addition

$q_{in2-3} = h_3 - h_2$

$q_{in2-3} = c_p (T_3 - T_2)$

$q_{in2-3} = (0.24)(3200\text{-}2294.38)$

$q_{in2-3} = 217.34 \text{ Btu/lbm}$

Step 8 of 11

The total heat input

$q_{in} = q_{in1-2} + q_{in2-3}$

$= 300 + 217.34$

$\boxed{q_{in} = 517.34 \text{ Btu/lbm}}$

Step 9 of 11

(c)

Process 3-4 is isentropic expansion

Therefore

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{3200} = \left(\frac{14.7}{62.45}\right)^{\frac{1.4-1}{1.4}} \qquad (\text{Since } P_4 = P_3)$$

$T_4 = 2116.70\text{R}$

Step 10 of 11

The heat output in process 4-1

$q_{out4-1} = h_4 - h_1$

$= c_p (T_4 - T_1)$

$= (0.24)(2116.70 - 540)$

$q_{out4-1} = 378.408 \text{ Btu/lbm}$

Step 11 of 11

The thermal efficiency

$$\eta_{th} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$= \frac{517.34 - 378.408}{517.34}$$

$= 0.2685$

$\boxed{\eta_{th} = 26.85\%}$

Question: 9.020P

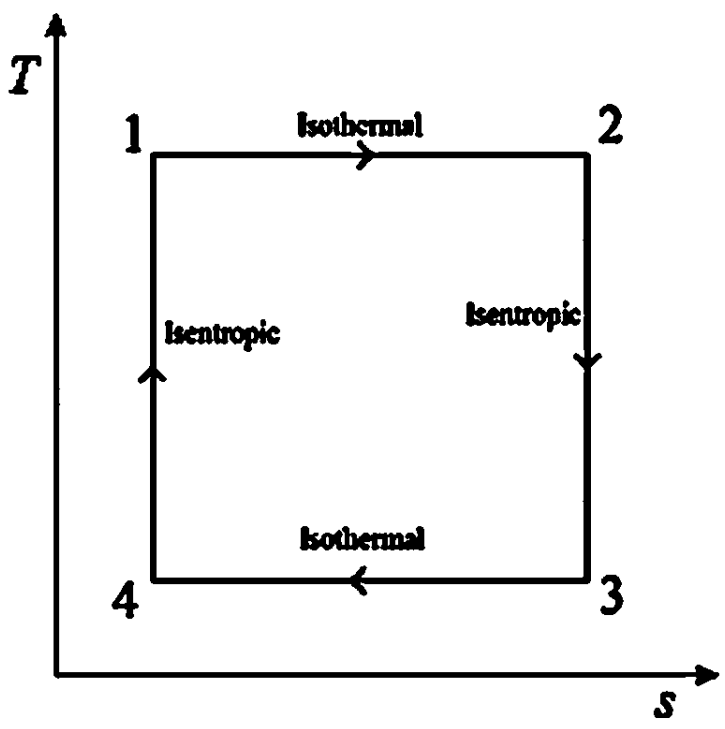
Consider a Carnot cycle executed in a closed system with 0.6 kg of air. The temperature limits of the cycle are 300 and 1100 K, and the minimum and maximum pressures that occur during the cycle are 20 and 3000 kPa. Assuming constant specific heats, determine the net work output per cycle.

Step 1 of 7

All four processes that comprise the Carnot cycle are reversible, and thus the area under each process curve represents the heat transfer for that process. Heat is transferred to the system during 1-2 and rejected during process 3-4 through isothermal processes and expansion, compression of medium is done in isentropic processes. Therefore, the representation of carnot cycle in T-S diagram is a rectangle

Step 2 of 7

***T-s* diagram for Carnot cycle:**



Step 3 of 7

For process 2-3 is isentropic,

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{(k/1-k)}$$

$$P_2 = P_{min}\left(\frac{T_2}{T_3}\right)^{(k/1-k)}$$

Here, P_2 is the pressure at state point 2 and P_3 is the pressure at state point 3 , P_{min} is the pressure at minimum pressure and T_3 and T_2 are the temperature at sate point 2 and 3 respectively.

Substitute 20kPa for P_{min} , 1100K for T_2 and 300K for T_3

$$P_2 = 20\left(\frac{1100}{300}\right)^{(1.4/1-1.4)}$$

$$= 1887.9 \text{ kPa}$$

Step 4 of 7

From energy balance, $dQ = dU + dW$ but $dQ = mTds$ Hence

$$mTds = mc_v dT + mPdV$$

$$ds = \frac{c_v dT}{T} + \frac{PdV}{T}$$

$$ds = \frac{c_v dT}{T} + \frac{RdV}{V}$$

$$ds = c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right)$$

$$ds = c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right)$$

Step 5 of 7

For isothermal process, $T_2 = T_1$

And for isothermal process, $P_1V_1 = P_2V_2$

Write the relation for entropy change,

$$ds = c_v \ln\left(\frac{T_1}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

$$= -R\ln\left(\frac{P_2}{P_1}\right)$$

$$= -0.287 \times \ln\left(\frac{1887.9}{3000}\right)$$

$$= -0.1329 \text{ kJ/kg}\cdot\text{K}$$

Step 6 of 7

Calculate the heat required for evaporation process

$$Q_{sup} = mT_H ds$$

Here, T_H is the source temperature and m is the mass flow rate through a processes

Substitute 0.6kg for m, 1100K for T_H and 0.132 kJ/kg·K for ds

$$Q_{sup} = 0.6 \times 1100 \times 0.132$$

$$= 87.12 \text{ kJ}$$

Calculate the Efficiency of carnot cycle

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

Substitute 300K for T_L

$$\eta_{th} = 1 - \frac{300}{1100}$$

$$= 0.7273$$

Step 7 of 7

Calculate the Net work in cycle,

$$W_{net} = \eta_{th} \times Q_{sup}$$

$$= 0.7273 \times 87.12 \text{ kJ}$$

$$= 63.36 \text{ kJ}$$

Hence net work output of the cycle is $\boxed{63.36 \text{ kJ}}$

Question: 9.021P

Consider a Carnot cycle executed in a closed system with air as the working fluid. The maximum pressure in the cycle is 800 kPa while the maximum temperature is 750 K. If the entropy increase during the isothermal heat rejection process is 0.25 kJ/kg-K and the net work output is 100 kJ/kg, determine (*a*) the minimum pressure in the cycle, (*b*) the heat rejection from the cycle, and (*c*) the thermal efficiency of the cycle, (*d*) If an actual heat engine cycle operates between the same temperature limits and produces 5200 kW of power for an air flow rate of 90 kg/s, determine the second law efficiency of this cycle.

Step 1 of 10

Given

Maximum temperature $T_1 = T_2 = 750K$

Maximum pressure $P_1 = 800$ kPa

Ideal net work output $w_{carnot} = 100$ kJ/kg

Entropy change during the isothermal heat rejection is $\Delta s_{3-4} = 0.25$ kJ/kg

Actual work output $W_{actual} = 5200$kW

Step 2 of 10

Considering constant specific heats

At room temperature from table ideal gas specific heats of various common gases

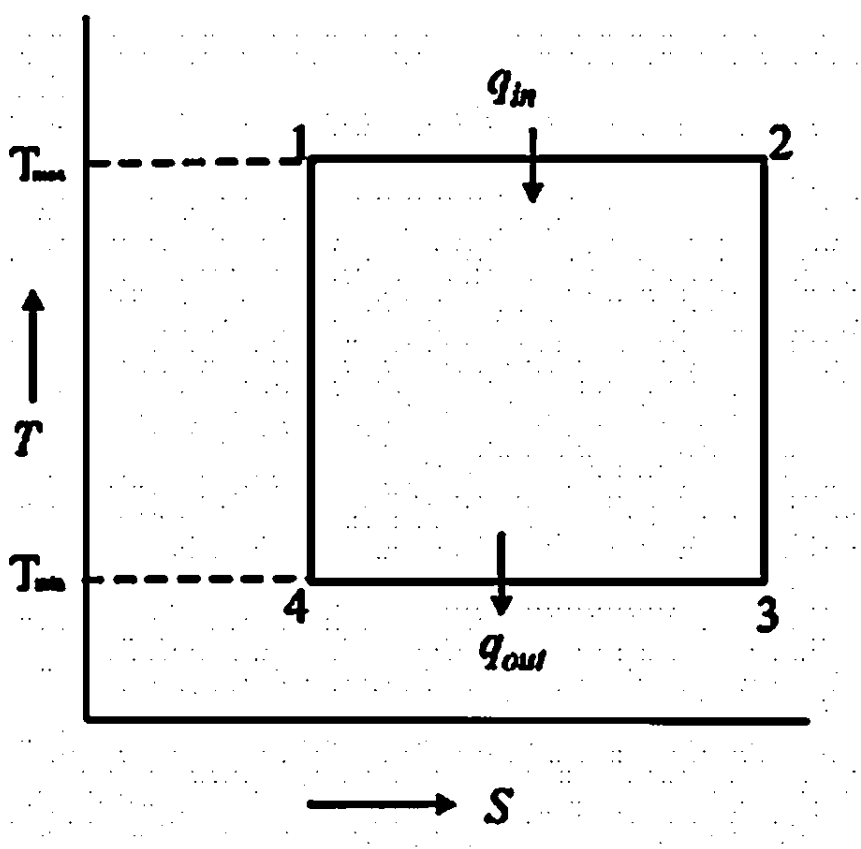
Gas constant $R = 0.287$ kJ/kg.K

$c_p = 1.005$ kJ/kg.K

$c_v = 0.718$ kJ/kg.K

$k = 1.4$

Step 3 of 10



Step 4 of 10

A Carnot cycle consists of 4 processes executing a cycle they are

Process 1-2 is constant temperature heat addition.

Process 2-3 is isentropic expansion.

Process 3-4 is constant temperature heat rejection.

Process 4-1 is isentropic compression.

Step 5 of 10

(a) The minimum pressure in the cycle

The work net is expressed as

$$w_{carnot} = q_{in} - q_{out}$$

$$= T_{max}\Delta s - T_{min}\Delta s$$

$$w_{carnot} = (T_{max} - T_{min})\Delta s$$

$$100 = (750 - T_{min})0.25$$

$$T_{min} = T_3 = T_4 = 350K$$

Step 6 of 10

Change in entropy we can write it as

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$0.25 = 0 - (0.287)\ln\frac{P_2}{800} \qquad (\text{Since } T_2 = T_1)$$

$$-0.871 = \ln\frac{P_2}{800}$$

$$P_2 = 334.8 \text{ kPa}$$

Step 7 of 10

Process 2-3 is isentropic expansion

$$\frac{T_2}{T_3} = \left(\frac{P_2}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{750}{350} = \left(\frac{334.8}{P_3}\right)^{\frac{1.4-1}{1.4}}$$

$$\boxed{P_3 = 23.24 \text{ kPa}}$$

Step 8 of 10

(b) Heat rejection from the cycle is

$$q_{out} = T_{min}\Delta s$$

$$= 350(0.25)$$

$$\boxed{q_{out} = 87.5 \text{ kJ/kg}}$$

Step 9 of 10

(c) The thermal efficiency of a Carnot cycle is defined as

$$\eta = 1 - \frac{T_{min}}{T_{max}}$$

$$= 1 - \frac{350}{750}$$

$$= 0.533333$$

$$\boxed{\eta = 53.34\%}$$

Step 10 of 10

(d) The second law of efficiency is defined as

$$\eta_{II} = \frac{W_{actual}}{W_{carnot}}$$

$$\eta_{II} = \frac{W_{actual}}{\dot{m}w_{carnot}}$$

$$\eta_{II} = \frac{5200}{90(100)}$$

$$= 0.5777$$

$$\boxed{\eta_{II} = 57.77\%}$$

Question: 9.022P

An ideal gas Carnot cycle uses air as the working fluid, receives heat from a heat reservoir at 1027°C, is repeated 1500 times per minute, and has a compression ratio of 12. The compression ratio is defined as the volume ratio during the compression process. Determine the maximum temperature of the low-temperature heat reservoir, the cycle's thermal efficiency, and the amount of heat that must be supplied per cycle if this device is to produce 500 kW of power.

Step 1 of 7

Given

High temperature reservoir $T_H = 1027^0\text{C}$

$$= 1027 + 273$$

$$T_H = 1300\text{K}$$

The compression ratio $r = \frac{v_1}{v_2} = 12$

Power output $\dot{W}_{net} = 500\ \text{kW}$

Number of cycles per minute $N = 1500\ \text{cycles/min}$

Step 2 of 7

Considering constant specific heats at room temperature

Taking properties of air at room temperature from table ideal gas specific heats of various common gases

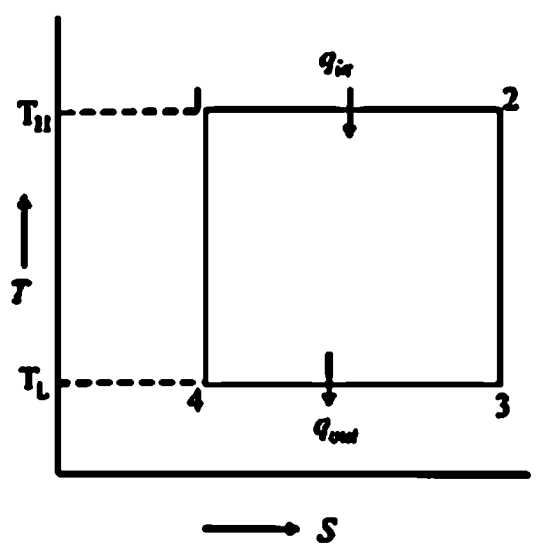
$R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 7



Step 4 of 7

We know that

$$\frac{T_2}{T_1} = \frac{T_H}{T_L} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_H}{T_L} = (r)^{k-1}$$

$$T_L = \frac{T_H}{r^{k-1}}$$

$$T_L = \frac{1300}{12^{1.4-1}}$$

$$\boxed{T_L = 481.139\text{K}}$$

Step 5 of 7

Efficiency of Carnot cycle is

$$\eta = 1 - \frac{T_L}{T_H}$$

$$\eta = 1 - \frac{481.139}{1300}$$

$$= 0.629$$

$$\boxed{\eta = 62.9\%}$$

Step 6 of 7

Work output per cycle

$$W_{net} = \frac{\dot{W}_{net}}{N}$$

$$= 500 \times \frac{1}{1500/60}$$

$$W_{net} = 20\text{kJ/cycle}$$

Step 7 of 7

The amount of heat supplied is

$$\eta = \frac{W_{net}}{Q_{in}}$$

$$Q_{in} = \frac{W_{net}}{\eta}$$

$$= \frac{20}{0.629}$$

$$\boxed{Q_{in} = 31.79\ \text{kJ/cycle}}$$

Question: 9.023P

How is the rpm (revolutions per minute) of an actual four-stroke gasoline engine related to the number of thermodynamic cycles? What would your answer be for a two-stroke engine?

Step 1 of 1

While calculating the performance of the four stroke engine the rpm is taken twice the actual value, since the actual engine gives one power stroke for every two revolutions or four strokes of the piston cylinder arrangement

In two stroke engine it gives one power stroke for one revolution or two strokes of the piston cylinder arrangement. So for two stroke engine rpm is equal to the number of thermodynamic cycles.

Question: 9.024P

How do the efficiencies of the ideal Otto cycle and the Carnot cycle compare for the same temperature limits? Explain.

Step 1 of 1

In Carnot cycle there are two isothermal processes and two isentropic processes whereas in Otto cycle two constant volume processes and two isentropic processes.
Otto cycle is not totally reversible, however, since they involve heat transfer through a finite temperature difference during the non isothermal heat addition and heat rejection processes, which are irreversible. Therefore the thermal efficiency of an Otto cycle will be less than that of a Carnot engine operating between the same temperature limits

$\eta_{Otto} < \eta_{carnot}$

Question: 9.025P

What four processes make up the ideal Otto cycle?

Step 1 of 2

An ideal Otto cycle consists of four internally reversible processes:

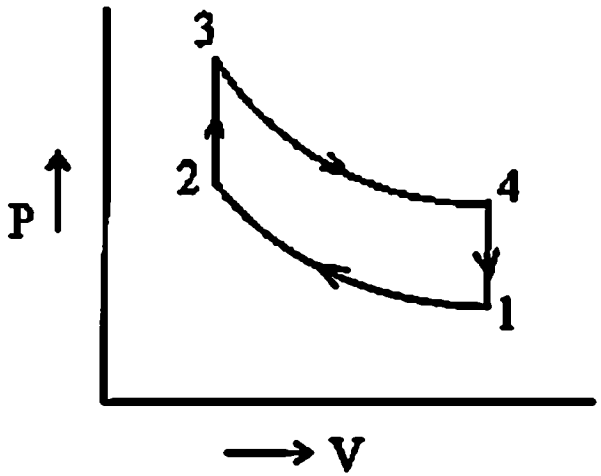
1-2 isentropic compression

2-3 Constant-volume heat addition

3-4 isentropic expansion

4-1 Constant-volume heat rejection

Step 2 of 2



Question: 9.026P

Are the processes that make up the Otto cycle analyzed as closed-system or steady-flow processes? Why?

Step 1 of 1

All the processes in Otto cycle are analyzed as closed system because there is no mass transfer from the system boundary in any one of the processes.

Question: 9.027P

How does the thermal efficiency of an ideal Otto cycle change with the compression ratio of the engine and the specific heat ratio of the working fluid?

Step 1 of 1

The ideal efficiency of an Otto cycle is defined as

$$\eta_{otto} = 1 - \frac{1}{r^{k-1}}$$

Where

r Is compression ratio

k Is specific heat ratio

The efficiency of an Otto cycle depends on the compression ratio and ratio of gas constant. As compression ratio increases efficiency increases and specific heat ratio increases the efficiency of Otto cycle increases

Question: 9.028P

Why are high compression ratios not used in spark-ignition engines?

Step 1 of 1

The increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios. Also when high compression ratios are used, the temperature of the air fuel mixture rises above the auto ignition temperature of the fuel during the combustion process, causing an early and rapid burn of the fuel. This premature ignition of the fuel, called auto ignition, produces an audible noise, which is called engine knock. So high compression ratios are not desirable, since they lead to engine knock

Question: 9.029P

An ideal Otto cycle with a specified compression ratio is executed using (*a*) air. (*b*) argon, and (*c*) ethane as the working fluid. For which case will the thermal efficiency be the highest? Why?

Step 1 of 1

The thermal efficiency of an Otto cycle is defined as

$$\eta_{otto} = 1 - \frac{1}{r^{k-1}}$$

We know as specific heat ratio, k increases thermal efficiency increases

Specific heat for air $k = 1.4$

Specific heat ratio for argon $k = 1.667$

Specific heat ratio for ethane $k = 1.2$

Since specific heat ratio for argon is more compared to the air and ethane the thermal efficiency of Otto cycle using argon is high.

Question: 9.030P

What is the difference between fuel-injected gasoline engines and diesel engines?

Step 1 of 1

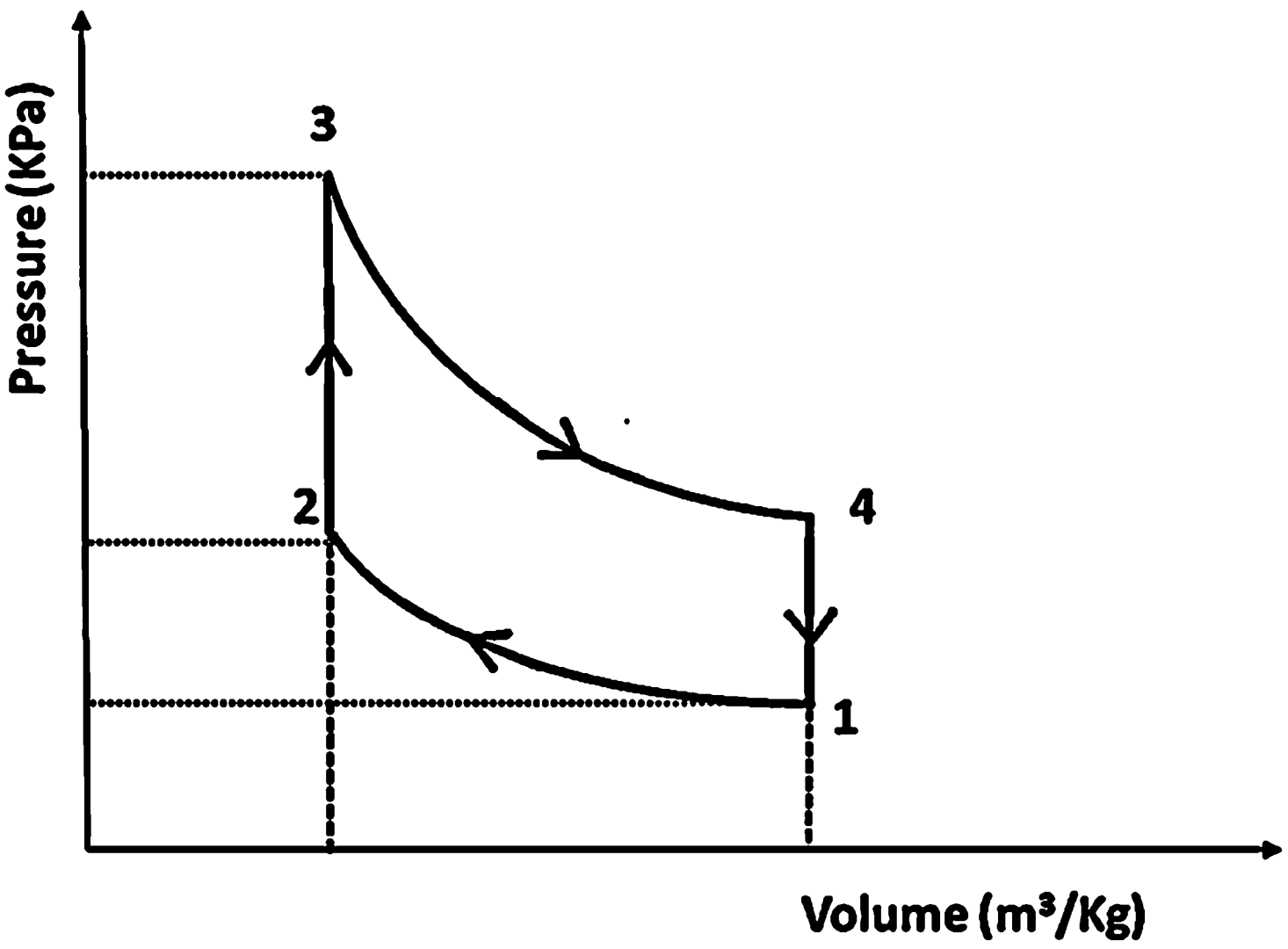
In a fuel-injected gasoline engine the ignition is done by a spark, where as in the diesel engine there in no requirement of spark.

Question: 9.031P

An ideal Otto cycle has a compression ratio of 10.5, takes in air at 90 kPa and 40°C, and is repeated 2500 times per minute. Using constant specific heats at room temperature, determine the thermal efficiency of this cycle and the rate of heat input if the cycle is to produce 90 kW of power.

Step 1 of 4

Draw the *P-V* diagram:



Step 2 of 4

Process 1-2 is an isentropic Compression process, hence $P_2V_2^{\gamma} = P_1V_1^{\gamma}$

From ideal gas equation, $\frac{P_2V_2}{T_2} = \frac{P_1V_1}{T_1}$.

Here, Pressures are P_1 , P_2; temperatures are T_1 , T_2 and volumes are V_1 and V_2 .

Substitute $P_1 = \frac{P_2V_2}{T_2} \times \frac{T_1}{V_1}$ in the equation

$$T_1V_1^{\gamma-1} = T_2V_2^{\gamma-1}$$

$$T_2 = \frac{T_1V_1^{\gamma-1}}{V_2^{\gamma-1}}$$

$$= T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

$$T_2 = T_1(r)^{\gamma-1}$$

Here, Compression ratio of the Otto cycle is $r = 10.5$

Substitute 1.4 for γ and 10.5 for r in the equation

$$T_2 = (313)(10.5)^{1.4-1}$$

$$= 801.715 \text{ K}$$

Step 3 of 4

Thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

Substitute 801.715 K for T_2 and $(313\text{K} = 40°\text{C})$ for T_1 in the equation

$$\eta_{th} = 1 - \frac{313}{801.715}$$

$$= 1 - 0.39$$

$$= 0.61$$

Step 4 of 4

Find the heat added to the cycle by using the following equation

$$W_{net} = \eta_{th} \times Q_{in}$$

$$Q_{in} = \frac{W_{net}}{\eta_{th}}$$

Here, Power output is W_{net}

Substitute 90 kW for W_{net} 0.61 for η_{th} in the equation

$$Q_{in} = \frac{90 \text{ kW}}{0.61}$$

$$= 147.54 \text{ kW}$$

Therefore, the heat added to the cycle is $\boxed{147.54 \text{ kW}}$

Question: 9.032P

Repeat Prob. 9–31 for a compression ratio of 8.5.

Step 1 of 5

Given,

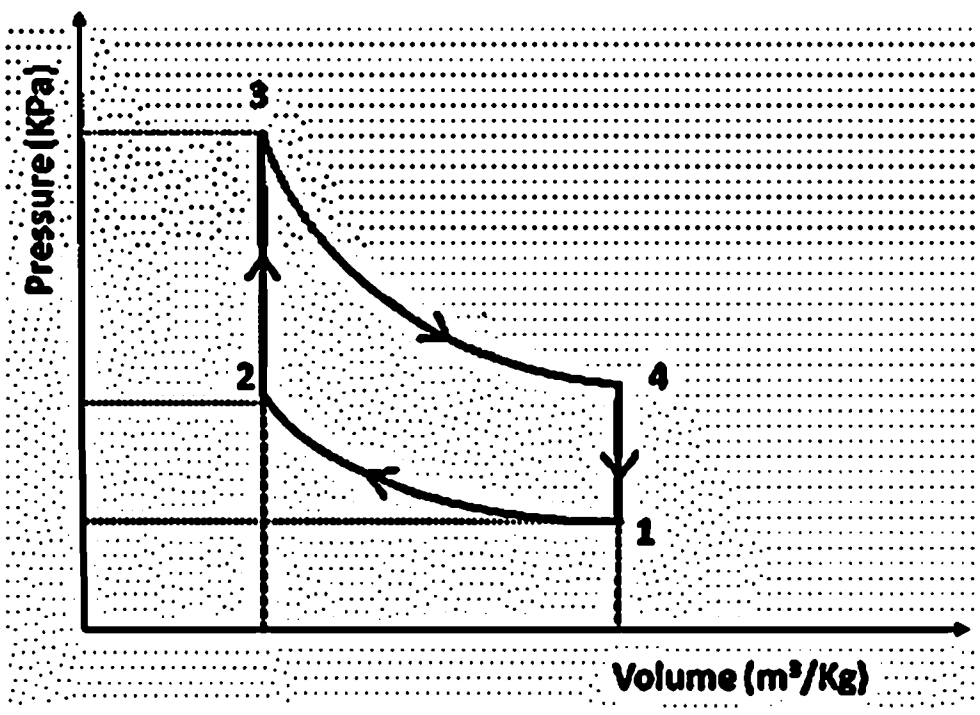
Compression ratio of the Otto cycle is $r = 8.5$

Temperature of air before compression, $T_1 = 40°C$

Pressure of air before compression, $P_1 = 90\ \text{kPa}$

Power output, $W_{net} = 90\ \text{kW}$

Step 2 of 5



Step 3 of 5

Process 1-2 is an isentropic Compression process, hence $P_2V_2^k = P_1V_1^k$

But from ideal gas equation, $\frac{P_2V_2}{T_2} = \frac{P_1V_1}{T_1}$.

$$T_1V_1^{k-1} = T_2V_2^{k-1}$$

$$T_2 = \frac{T_1V_1^{k-1}}{V_2^{k-1}}$$

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1}$$

$$T_2 = T_1(r)^{k-1}$$

$$T_2 = (313)(8.5)^{1.4-1}$$

$$T_2 = 736.736\ \text{K}$$

Step 4 of 5

Thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

$$\eta_{th} = 1 - \frac{313}{736.736}$$

$$\eta_{th} = 1 - 0.425$$

$$\eta_{th} = 0.575$$

Step 5 of 5

Work output of the cycle is

$$W_{net} = \eta_{th} \times Q_{in}$$

$$Q_{in} = \frac{W_{net}}{\eta_{th}}$$

$$Q_{in} = \frac{90\ \text{kW}}{0.575}$$

$$Q_{in} = 156.52\ \text{kW}$$

Hence heat added to the cycle is $\boxed{Q_{in} = 156.52\ \text{kW}}$

Question:

Someone has suggested that the air-standard Otto cycle is more accurate if the two isentropic processes are replaced with polytropic processes with a polytropic exponent n= 1.3. Consider such a cycle when the compression ratio is 8, $P1$ =95 kPa, $T1$, = 15°C, and the maximum cycle temperature is 1200T. Determine the heat transferred to and rejected from this cycle, as well as the cycle's thermal efficiency. Use constant specific heats at room temperature.

Step 1 of 5

From the properties of air (Table A-2a):

Universal gas constant, $R = 0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$

Specific heat constants,

$$c_p = 1.005\,\text{kJ/kg}\cdot\text{K}$$
$$c_v = 0.718\,\text{kJ/kg}\cdot\text{K}$$

Calculate the temperature at stage 2:

The process 1-2 and 3-4 are replaced with polytropic compression and expression processes respectively.

For the polytrophic compression process $1-2$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1}$$
$$= \left(\frac{1}{r}\right)^{n-1}$$

Here r is the Compression ratio, n is the polytrophic index and T_1 , T_2 are temperatures at stage1,stage 2 respectively.

Substitute 8 for r and 288 K for T_1 and 1.3 for n in above relation.

Therefore,

$$\frac{288}{T_2} = \left(\frac{1}{8}\right)^{1.3-1}$$
$$T_2 = 288\times(8)^{1.3-1}$$
$$= 537.43\,\text{K}$$

Therefore, temperature at state 2 is $T_2 = 537.43\,\text{K}$

Step 2 of 5

Calculate the temperature at stage 4:

The process 3-4 is a polytrophic expansion process,

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{n-1}$$
$$T_4 = T_3\left(\frac{V_2}{V_1}\right)^{n-1}$$
$$= T_3\left(\frac{1}{r}\right)^{n-1}$$

Substitute 8 for r and 1473 K for T_3 and 1.3 for n in above relation.

$$T_4 = 1473\left(\frac{1}{8}\right)^{1.3-1}$$
$$= 789.36\,\text{K}$$

Step 3 of 5

Calculate specific work done in the 1-2 process:

$$w_{1-2} = \frac{RT_1}{n-1}\left(\left(\frac{v_1}{v_2}\right)^{n-1} - 1\right)$$
$$= \frac{RT_1}{n-1}\left((r)^{n-1} - 1\right)$$

Substitute $R = 0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$, $T_1 = 288\,\text{K}$, $n = 1.3$ and $r = 8$ in above formula,

$$w_{1-2} = \frac{0.287\times288}{1.3-1}\left((8)^{1.3-1} - 1\right)$$
$$= 238.62\,\text{kJ/kg}$$

Calculate specific work done in the 3-4 process:

$$w_{3-4} = -\frac{RT_3}{n-1}\left(\left(\frac{v_3}{v_4}\right)^{n-1} - 1\right)$$
$$= -\frac{RT_3}{n-1}\left(\left(\frac{1}{r}\right)^{n-1} - 1\right)$$

Substitute $R = 0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$, $T_3 = 1473\,\text{K}$, $n = 1.3$ and $r = 8$ in above formula,

$$w_{3-4} = -\frac{0.287\times1473}{1.3-1}\left(\left(\frac{1}{8}\right)^{1.3-1} - 1\right)$$
$$= 654\,\text{kJ/kg}$$

Step 4 of 5

Calculate specific heat in 1-2 process by using first law of thermodynamics:

$$q_{1-2} = w_{1-2} - c_v(T_2 - T_1)$$

Substitute $238.62\,\text{kJ/kg}$ for w_{1-2} , $0.718\,\text{kJ/kg}\cdot\text{K}$ for c_v ,537.43 K for T_2 and 288 K for T_1 in above relation.

$$q_{1-2} = 238.62 - 0.718(537.4 - 288)$$
$$= 59.55\,\text{kJ/kg}$$

Calculate specific heat in 2-3 process

relation.

$$q_{1-2} = 238.62 - 0.718(537.4 - 288)$$
$$= 59.55\,\text{kJ/kg}$$

Calculate specific heat in 2-3 process,

$$q_{2-3} = c_v(T_3 - T_2)$$
$$= 0.718(1473 - 537.43)$$
$$= 671.74\,\text{kJ/kg}$$

Calculate specific heat in 2-3 process by using first law of thermodynamics:

$$q_{3-4} = w_{3-4} - c_v(T_3 - T_4)$$

Substitute 654 for w_{3-4} , $0.718\,\text{kJ/kg}\cdot\text{K}$ for c_v ,1473 K for T_3 and 789.36 K for T_4 in above relation.

$$q_{3-4} = 654 - 0.718(1473 - 789.36)$$
$$= 163.15\,\text{kJ/kg}$$

Calculate specific heat in 4-1 process,

$$q_{4-1} = c_v(T_4 - T_1)$$
$$= 0.718(789.36 - 288)$$
$$= 359.98\,\text{kJ/kg}$$

Step 5 of 5

Calculate heat addition to the cycle,

$$q_{in} = q_{2-3} + q_{3-4}$$

Substitute 671.74 kJ/kg for q_{2-3} and 163.15 kJ/kg for q_{3-4} in above relation.

$$q_{in} = 671.74 + 163.15$$
$$= 834.89\,\text{kJ/kg}$$

Therefore, heat transfer to cycle is $\boxed{835\,\text{kJ/kg}}$.

Calculate heat rejection from the cycle,

$$q_{out} = q_{1-2} + q_{4-1}$$

Substitute 59.53 kJ/kg for q_{1-2} and 359.98 kJ/kg for q_{4-1} in above relation.

$$q_{out} = 59.53 + 359.98$$
$$= 419.51\,\text{kJ/kg}$$

Therefore, heat rejection from cycle is $\boxed{419.5\,\text{kJ/kg}}$

Calculate efficiency of the cycle is given by

$$\eta_{thermal} = \frac{q_{in} - q_{out}}{q_{in}}$$

Substitute 834.89 kJ/kg for q_{in} and 419.51 kJ/kg for q_{out} in above relation.

$$\eta_{thermal} = \frac{834.89 - 419.51}{834.89}$$
$$= 0.4976$$
$$= 49.76\%$$

Therefore, efficiency of the Otto cycle is $\boxed{49.8\%}$

Question: 9.034P

How do the results of Prob. 9–33 change when isentropic processes are used in place of the polytropic processes?

Step 1 of 5

Given data:

For the Otto cycle

Temperature at state 1, $T_1 = 15°C = 288K$

Pressure at state 1, $P_1 = 95kPa$

Compression ratio, $r = 8$

Maximum temperature in the cycle, $T_3 = 1200°C = 1473K$

Polytropic exponent, $n = 1.3$

Step 2 of 5

Compression ratio, $r = \frac{V_1}{V_2}$

For the isentropic compression process 1-2

Therefore,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{r-1}$$

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{r-1}$$

$$= 288 \times (8)^{1.4-1}$$

$$= 661.65K$$

Therefore, temperature at state 2, $T_2 = 661.65K$

Step 3 of 5

The process 3-4 is an isentropic expansion process.

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{r-1}$$

$$T_4 = T_3\left(\frac{V_2}{V_1}\right)^{r-1}$$

$$= 1473\left(\frac{1}{8}\right)^{1.4-1}$$

$$= 641.16K$$

Step 4 of 5

Specific heat of the air, $c_v = 0.718kJ/kg$

Heat added in the process 2-3,

$$Q_{added} = c_v(T_3 - T_2)$$

$$= 0.718(1473 - 661.65)$$

$$= 582.54kJ/kg$$

Heat added in the cycle, $\boxed{Q_{added} = 582.54kJ/kg}$

Heat rejected in the process 2-3,

$$Q_{rejected} = c_v(T_4 - T_1)$$

$$= 0.718(641.16 - 288)$$

$$= 253.56kJ/kg$$

Therefore, Heat rejected in the cycle is $\boxed{Q_{rejected} = 253.56kJ/kg}$

Step 5 of 5

Efficiency of the cycle,

$$\eta_{thermal} = \frac{Q_{in} - Q_{out}}{Q_{in}}$$

$$= \frac{582.54 - 253.56}{582.54}$$

$$= 0.564$$

$$= 56.47\%$$

Therefore, Efficiency of the Otto cycle is $\boxed{\eta_{thermal} = 56.47\%}$

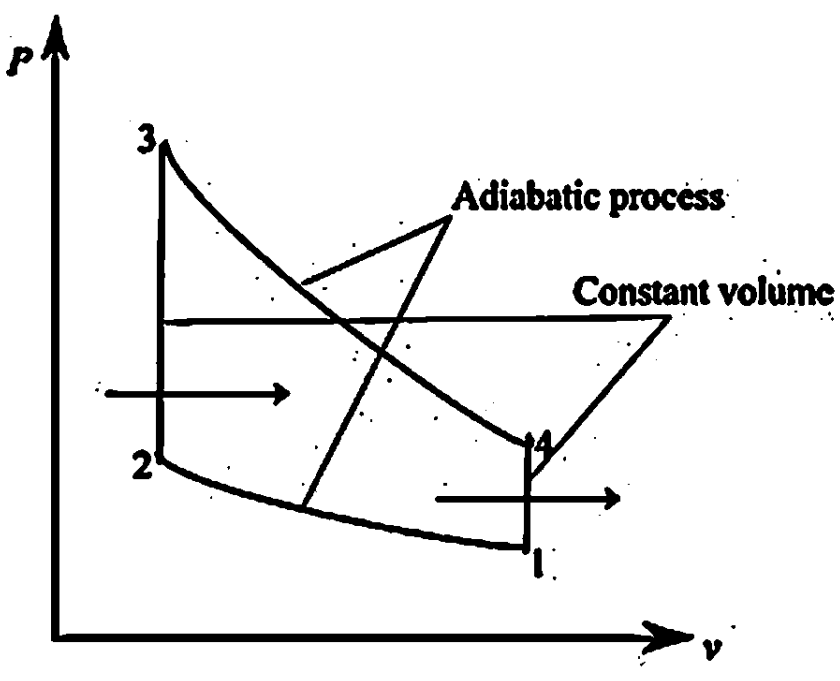
The thermal efficiency of engine will increase with the isentropic processes than the polytropic processes in the cycle.

Question: 9.035P

A six-cylinder, four-stroke, spark-ignition engine, operating on the ideal Otto cycle takes in air at 14 psia and 65°F, and is limited to a maximum cycle temperature of 1600°F. Each cylinder has a bore of 3.5 in. and each piston has a stroke of 3.9 in. The minimum enclosed volume is 14 percent of the maximum enclosed volume. How much power will this engine produce when operated at 2500 rpm? Use constant specific heats at room temperature.

Step 1 of 5

Draw the *P-v* diagram of the Otto-cycle:



Step 2 of 5

Considering constant specific heats at room temperature

Gas constant $R = 0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$

$c_p = 0.240\,\text{Btu/lbm}\cdot\text{R}$

$c_v = 0.171\,\text{Btu/lbm}\cdot\text{R}$

$k = 1.4$

Calculate compression ratio by using,

From the isentropic compression Process 1-2,

$$r = \frac{V_1}{V_2}$$

Here V_1 is the maximum volume and V_2 is the minimum volume

Substitute 14% of V_1 for V_2 in above relation

$$r = \frac{V_1}{0.14V_1}$$
$$= 7.1429$$

Step 3 of 5

Calculate temperature at state 2 by using

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$
$$T_2 = T_1 r^{k-1}$$

Here k is the index and T_1 is the temperature at state 1

Substitute 1.4 for k, $65°\text{F} = 524.67°\text{R}$ for T_1 and 7.1429 for r in above relation

$$T_2 = 524.67\left(7.1429^{1.4-1}\right)$$
$$= 1151.95°\text{R}$$

Calculate temperature at state 4 by using

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1}$$
$$T_4 = T_3\left(\frac{1}{r}\right)^{k-1}$$

Here k is the index and T_3 is the temperature at state 3

Substitute 1.4 for k, $2060°\text{R}$ for T_3 and 7.1429 for r in above relation

$$T_4 = 2060\left(\frac{1}{7.1429}\right)^{1.4-1}$$
$$= 938.26°\text{R}$$

Step 4 of 5

Calculate net work done per mass by using

$$w_{net} = c_v(T_3 - T_4) - c_v(T_2 - T_1)$$

Substitute 0.171 Btu/lbm for c_v, $524.67°\text{R}$ for T_1, $1151.95°\text{R}$ for T_2, $2060°\text{R}$ for T_3 and $938.26°\text{R}$ for T_4 in above relation

$$w_{net} = 0.171(2060 - 938.26) - 0.171(1151.95 - 524.67)$$
$$= 191.81754 - 107.26488$$
$$= 84.55\,\text{Btu/lbm}$$

Calculate total mass m by using ,

$$m = n\rho V$$
$$\dot{m} = n \times \frac{P_1}{RT_1} \times \frac{\pi}{4} d^2 L$$

Here n is the number of the cylinders, P_1 is the initial pressure of the given system, T_1 is the initial temperature of the given system, d is the bore of the cylinder, L is the length of the cylinder and N is the operating speed of the engine.

Substitute 6 for n,14 psia for P_1,0.3704 $\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$ for R, $\frac{3.5}{12}\text{ft}$ for d, $\frac{3.9}{12}\text{ft}$ for L and $524.67°\text{R}$ for T_1 in above relation

$$m = 6 \times \frac{14}{0.3704 \times 524.67} \times \frac{\pi}{4} \times \left(\frac{3.5}{12}\right)^2 \times \frac{3.9}{12}$$
$$= 0.0093856\,\text{lbm}$$

Calculate network produced per cycle by using

$$W_{net} = mw_{net}$$

Substitute 0.0093856 lbm for m and 84.55 Btu/lbm for w_{net} in above relation

$$W_{net} = 0.0093856 \times 84.55$$
$$= 0.7935\,\text{Btu/cycle}$$

Step 5 of 5

Calculate power produced by using

$$\dot{W}_{net} = \frac{W_{net}N}{N_{rev}}$$

Here N_{rev} is the number of revolutions per cycle

Substitute 0.7935 Btu/cycle for W_{net}, $\frac{2500}{60}\text{rps}$ for N and 2 rev/cycle for N_{rev} in above relation

$$\dot{W}_{net} = \frac{0.7935 \times \frac{2500}{60}}{2}$$
$$= 16.53125\,\text{Btu/s}$$

Convert the power units from Btu/s to hp

$$\dot{W}_{net} = 16.53125\,\text{Btu/s} \times \frac{1}{0.7068}\frac{\text{hp}}{\text{Btu/s}}$$
$$= 23.39\,\text{hp}$$

Therefore, total power generated in the given Otto cycle, **23.4 hp**

Question: 9.036P

A spark-ignition engine has a compression ratio of 8, an isentropic compression efficiency of 85 percent, and an isentropic expansion efficiency of 95 percent. At the beginning of the compression, the air in the cylinder is at 13 psia and 60°F. The maximum gas temperature is found to be 2300°F by measurement. Determine the heat supplied per unit mass, the thermal efficiency, and the mean effective pressure of this engine when modeled with the Otto cycle. Use constant specific heats at room temperature.

Step 1 of 12

Given

Compression ratio $r = \frac{v_1}{v_2} = 8$

Isentropic compression efficiency $\eta_c = 85\%$

Isentropic expansion efficiency $\eta_t = 95\%$

Inlet pressure $P = 13$ psia

Inlet temperature $T_1 = 60°F$

$= 60+460$

$T_1 = 520R$

Maximum temperature $T_3 = 2300°F = 2760R$

Step 2 of 12

Considering constant specific heats at room temperature

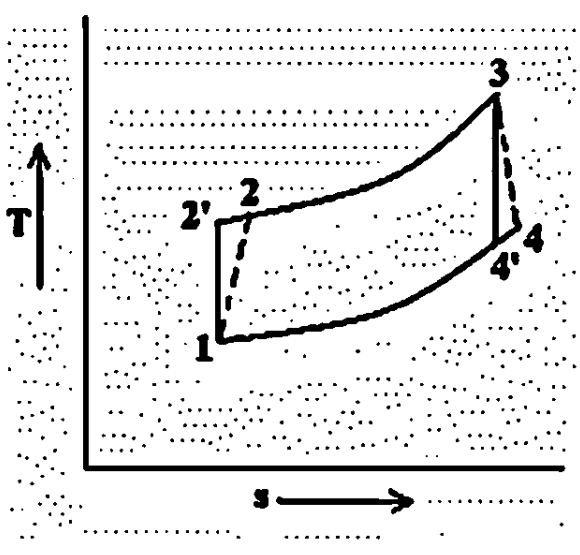
Gas constant $R = 0.3704$ psia.ft^3/lbm.R

$c_p = 0.24$ Btu/lbm.R

$c_v = 0.171$ Btu/lbm.R

$k = 1.4$

Step 3 of 12



Step 4 of 12

Process 1-2' is isentropic compression

$$\frac{T_2'}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2'}{520} = (8)^{1.4-1}$$

$$T_2' = 1194.6R$$

Step 5 of 12

Isentropic compression efficiency is

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$0.85 = \frac{1194.6 - 520}{T_2 - 520}$$

$$T_2 = 1313.647R$$

Step 6 of 12

The heat supplied per unit mass is

$$q_{in} = c_v(T_3 - T_2)$$

$$= (0.171)(2760 - 1313.647)$$

$$\boxed{q_{in} = 247.32 \text{ Btu/lbm}}$$

Step 7 of 12

Process 4'-1 is isentropic expansion

$$\frac{T_4'}{T_3} = \left(\frac{v_3}{v_4'}\right)^{k-1}$$

$$\frac{T_4'}{T_3} = \left(\frac{v_2'}{v_1}\right)^{k-1}$$

$$\frac{T_4'}{2760} = \left(\frac{1}{8}\right)^{1.4-1}$$

$$T_4' = 1201.35R$$

Step 8 of 12

The isentropic expansion efficiency is

$$\eta_{ex} = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$0.95 = \frac{2760 - T_4}{2760 - 1201.35}$$

$$T_4 = 1279.28R$$

Step 9 of 12

Net work of the cycle is

$$w_{net} = w_{3-4} - w_{1-2}$$

$$w_{net} = c_v(T_3 - T_4) - c_v(T_2 - T_1)$$

$$w_{net} = c_v\left[(T_3 - T_4) - (T_2 - T_1)\right]$$

$$= (0.171)\left[(2760 - 1279.28) - (1313.647 - 520)\right]$$

$$w_{net} = 117.48 \text{ Btu/lbm}$$

Step 10 of 12

The thermal efficiency is defined as

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{117.48}{247.32}$$

$$= 0.475$$

$$\boxed{\eta_{th} = 47.5\%}$$

Step 11 of 12

The mean effective pressure of an engine is defined as

$$MEP = \frac{w_{net}}{v_1 - v_2}$$

Where specific volume at inlet is

$$v_1 = \frac{RT_1}{P_1}$$

$$v_1 = \frac{0.3704(520)}{13}$$

$$v_1 = 14.816 \text{ ft}^3/\text{lbm}$$

Step 12 of 12

Compression ratio

$$\frac{v_1}{v_2} = 8$$

$$v_2 = \frac{v_1}{8}$$

$$v_2 = \frac{14.816}{8}$$

$$v_2 = 1.852 \text{ ft}^3/\text{lbm}$$

$$MEP = \frac{w_{net}}{v_1 - v_2}$$

$$MEP = \frac{117.48}{14.816 - 1.852} \times 778 \times \frac{1}{144}$$

$$\boxed{MEP = 48.96 \text{ psia}}$$

Question: 9.037P

The compression ratio of an air-standard Otto cycle is 9.5. Prior to the isentropic compression process, the air is at 100 kPa. 35°C, and 600 cm 3, The temperature at the end of the isentropic expansion process is 800 K. Using specific heat values at room temperature, determine (*a*) the highest temperature and pressure in the cycle; (*b*) the amount of heat transferred in, in kJ; (*c*) the thermal efficiency; and (*d*) the mean effective pressure.

Step 1 of 10

Given

The compression ratio $r = \frac{V_1}{V_2} = 9.5$

Initial pressure $P_1 = 100\ \text{kPa}$

Initial temperature $T_1 = 35°\text{C}$

$= 35 + 273$

$T_1 = 308\text{K}$

Initial volume $V_1 = 600\ \text{cm}^3$

Temperature at the end of expansion process $T_4 = 800\text{K}$

Step 2 of 10

Considering constant specific heats at room temperature

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 10

(a)

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = (r)^{k-1}$$

$$T_2 = 308(9.5)^{1.4-1}$$

$$T_2 = 757.94\text{K}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{k}$$

$$P_2 = 100(9.5)^{1.4}$$

$$P_2 = 2337.83\ \text{kPa}$$

Step 4 of 10

Process 3-4 is an isentropic expansion

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1}$$

$$\frac{T_3}{T_4} = (r)^{k-1}$$

$$T_3 = 800(9.5)^{1.4-1}$$

$$T_3 = 1968.69\text{K}$$

The maximum temperature is

$$\boxed{T_3 = 1968.69\text{K}}$$

Step 5 of 10

Process 2-3 is constant volume process

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$P_3 = P_2 \times \frac{T_3}{T_2}$$

$$P_3 = 2337.83 \times \left(\frac{1968.69}{757.94}\right)$$

$$P_3 = 6072.33\text{kPa}$$

Therefore maximum pressure

$$\boxed{P_3 = 6072.33\ \text{kPa}}$$

Step 6 of 10

(b) The amount of heat transferred

Mass of air can be calculated as by using ideal gas equation

$$P_1V_1 = mRT_1$$

$$m = \frac{P_1V_1}{RT_1}$$

$$m = \frac{100\left(600\times10^{-6}\right)}{0.287(308)}$$

$$m = 6.78\times10^{-4}\ \text{kg}$$

Step 7 of 10

Therefore the amount of heat transferred from the cycle

$$Q_{in} = mc_v(T_3 - T_2)\ \text{kJ}$$

$$= 6.78\times10^{-4}(0.718)(1968.69 - 757.94)$$

$$\boxed{Q_{in} = 0.589\ \text{kJ}}$$

Step 8 of 10

(c) The thermal efficiency

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}}$$

The amount of heat rejected is expressed as

$$Q_{out} = mc_v(T_4 - T_1)\ \text{kJ}$$

$$= 6.78\times10^{-4}(0.718)(800 - 308)$$

$$Q_{out} = 0.2395\ \text{kJ}$$

Step 9 of 10

Therefore thermal efficiency

$$\eta_{th} = 1 - \frac{0.2395}{0.589}$$

$$= 0.5933$$

$$\boxed{\eta_{th} = 59.33\%}$$

Step 10 of 10

(d) The Mean effective pressure

$$MEP = \frac{W_{net}}{V_1 - V_2}$$

$$MEP = \frac{Q_{in} - Q_{out}}{V_1\left(1 - \frac{V_2}{V_1}\right)}$$

$$MEP = \frac{0.589 - 0.2395}{V_1\left(1 - \frac{1}{r}\right)}$$

$$= \frac{0.3495}{6\times10^{-4}\left(1 - \frac{1}{9.5}\right)}$$

$$\boxed{MEP = 651.029\ \text{kPa}}$$

Question: 9.038P

Repeat Prob. 9–37, but replace the isentropic expansion process by a polytropic expansion process with the polytropic exponent n =1.35.

Step 1 of 11

Given

The compression ratio $r = \frac{V_1}{V_2} = 9.5$

Initial pressure $P_1 = 100\ \text{kPa}$

Initial temperature $T_1 = 35°\text{C}$

$= 35 + 273$

$T_1 = 308\text{K}$

Initial volume $V_1 = 600\ \text{cm}^3$

Temperature at the end of expansion process $T_4 = 800\text{K}$

Polytropic exponent $n = 1.35$

Step 2 of 11

Considering constant specific heats at room temperature

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 11

(a) The highest temperature in the cycle

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = (r)^{k-1}$$

$$T_2 = 308(9.5)^{1.4-1}$$

$$T_2 = 757.949\text{K}$$

Step 4 of 11

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k$$

$$\frac{P_2}{P_1} = (r)^k$$

$$P_2 = P_1(r)^k$$

$$P_2 = 100(9.5)^{1.4}$$

$$P_2 = 2337.8\ \text{kPa}$$

Step 5 of 11

Process 3-4 is a polytropic expansion with index $n = 1.35$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{n-1}$$

$$\frac{T_3}{T_4} = \left(\frac{V_1}{V_2}\right)^{n-1}$$

$$\frac{T_3}{T_4} = (r)^{n-1}$$

$$T_3 = 800(9.5)^{1.35-1}$$

$$\boxed{T_3 = 1759.11\text{K}}$$

Step 6 of 11

Process 2-3 is constant volume process

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$$P_3 = 2337.8\left(\frac{1759.1}{757.949}\right)$$

$$\boxed{P_3 = 5425.72\ \text{kPa}}$$

Step 7 of 11

b) The amount of heat transferred

The amount of mass of air can be calculated as

$$P_1V_1 = mRT_1$$

$$m = \frac{P_1V_1}{RT_1}$$

$$m = \frac{100\left(600\times10^{-6}\right)}{0.287(308)}$$

$$m = 6.78\times10^{-4}\ \text{kg}$$

Step 8 of 11

The amount of heat transferred is

$$Q_{in} = mc_v(T_3 - T_2)\ \text{kJ}$$

$$= 6.78\times10^{-4}(0.718)(1759.1 - 757.949)$$

$$\boxed{Q_{in} = 0.4873\ \text{kJ}}$$

Step 9 of 11

(c)

The thermal efficiency

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}}$$

The amount of heat rejected is expressed as

$$Q_{out} = mc_v(T_4 - T_1)\ \text{kJ}$$

$$= 6.78\times10^{-4}(0.718)(800 - 308)$$

$$Q_{out} = 0.2395\ \text{kJ}$$

Step 10 of 11

Therefore thermal efficiency

$$\eta_{th} = 1 - \frac{0.2395}{0.4873}$$

$$= 0.5085$$

$$\boxed{\eta_{th} = 50.85\%}$$

Step 11 of 11

(d) The Mean effective pressure

$$MEP = \frac{W_{net}}{V_1 - V_2}$$

$$MEP = \frac{Q_{in} - Q_{out}}{V_1\left(1 - \frac{V_2}{V_1}\right)}$$

$$MEP = \frac{0.4873 - 0.2395}{6\times10^{-4}\left(1 - \frac{1}{9.5}\right)}$$

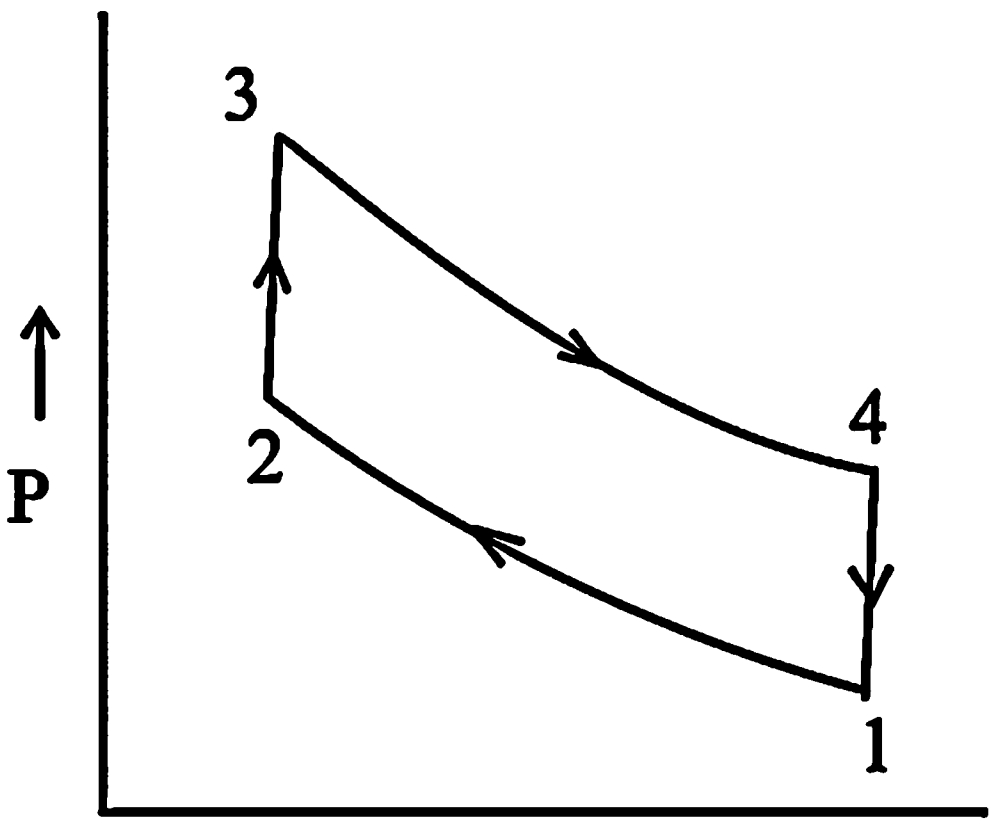
$$\boxed{MEP = 461.588\ \text{kPa}}$$

Question:

An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The minimum and maximum temperatures in the cycle are 540 and 2400 R. Accounting for the variation of specific heats with temperature, determine (*a*) the amount of heat transferred to the air during the heat-addition process, (*b*) the thermal efficiency, and (*c*) the thermal efficiency of a Carnot cycle operating between the same temperature limits.

Step 1 of 8

Draw *P-v* diagram of the ideal Otto cycle.



Here, process 1-2 is isentropic compression, process 2-3 is constant volume heat addition, process 3-4 is isentropic expansion, and process 4-1 is constant volume heat rejection.

Step 2 of 8

Obtain properties of air at $T_1 = 540\text{R}$ from table A-17E, "ideal gas properties of air" in the text book.

Specific internal energy, $u_1 = 92.04\ \text{Btu/lbm}$

Relative volume, $v_{r1} = 144.32$

Similarly, obtain properties of air at $T_3 = 2400\text{R}$ from table A-17E, "ideal gas properties of air" in the text book.

Specific internal energy, $u_3 = 452.70\ \text{Btu/lbm}$

Relative volume, $v_{r3} = 2.419$

Step 3 of 8

Apply isentropic relation to the process 1-2.

$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1}$$

$$v_{r2} = v_{r1} \times \frac{v_2}{v_1}$$

Here, $\frac{v_1}{v_2}$ is compression ratio.

Substitute 8 for $\frac{v_1}{v_2}$

$$v_{r2} = \frac{144.32}{8}$$
$$= 18.04$$

Step 4 of 8

Obtain properties of air at $v_{r2} = 18.04$ from table A-17E, "ideal gas properties of air" in the text book.

Specific internal energy,

$$u_2 = 209.05 + (216.53 - 209.05)\left(\frac{18.04 - 18.51}{16.93 - 18.51}\right)$$
$$= 211.275\ \text{Btu/lbm}$$

Step 5 of 8

Apply isentropic relation to the process 3-4,

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3}$$

$$v_{r4} = \frac{v_4}{v_3} v_{r3}$$

$$v_{r4} = \frac{v_1}{v_2} v_{r3}$$

Substitute 8 for $\frac{v_1}{v_2}$ and 2.419 for v_{r3}

$$v_{r4} = 8(2.419)$$
$$= 19.352$$

Obtain properties of air at $v_{r4} = 19.352$ from table A-17E, "ideal gas properties of air" in the text book.

Specific internal energy,

$$u_4 = 201.63 + (209.05 - 201.63)\left(\frac{19.352 - 20.29}{18.51 - 20.29}\right)$$
$$= 205.54\ \text{Btu/lbm}$$

Step 6 of 8

(a)

$= 205.54$ Btu/lbm

Step 6 of 8

(a)

Find the amount of heat addition during the process 2-3.

$$q_{in} = (u_3 - u_2)$$

Substitute 452.70 Btu/lbm for u_3 and 211.275 Btu/lbm for u_2

$$q_{in} = (452.70 - 211.275)$$
$$= 241.425 \text{ Btu/lbm}$$

Therefore, the amount of heat addition is $\boxed{241.425 \text{ Btu/lbm}}$

Step 7 of 8

(b)
Find the amount of heat rejection in the process 4-1.

$$q_{out} = (u_4 - u_1)$$

Substitute 205.54 Btu/lbm for u_4 and 92.04 Btu/lbm for u_1

$$q_{out} = (205.54 - 92.04)$$
$$= 113.5 \text{ Btu/lbm}$$

Find the thermal efficiency by using the following equation:

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Substitute 241.425 Btu/lbm for q_{in} and 113.5 Btu/lbm for q_{out}

$$\eta_{th} = 1 - \frac{113.5}{241.425}$$
$$= 0.5298$$
$$= 52.98\%$$

Therefore, the thermal efficiency of the cycle is $\boxed{52.98\%}$

Step 8 of 8

(c)

Find the thermal efficiency of a Carnot cycle operating between the same temperature limits by using the following equation:

$$\eta_{carnot} = 1 - \frac{T_L}{T_H}$$
$$= 1 - \frac{T_1}{T_3}$$

Substitute 540 R for T_1 and 2400 R for T_3

$$\eta_{carnot} = 1 - \frac{540}{2400}$$
$$= 0.775$$
$$= 77.5\%$$

Therefore, the efficiency of eth Carnot cycle is $\boxed{77.5\%}$.

Question: 9.040P

Repeat Prob. 9-39E using argon as the working flujj

Step 1 of 9

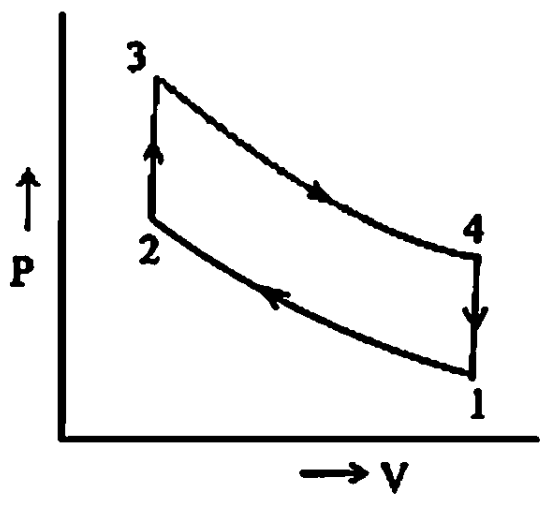
Given

The compression ratio $r = \frac{v_1}{v_2} = 8$

Minimum temperature $T_1 = 540\text{R}$

Maximum temperature $T_3 = 2400\text{R}$

Step 2 of 9



Step 3 of 9

Considering constant specific heats at room temperature

From table ideal gas specific heats of common gases for Argon gas

$c_p = 0.1253$ Btu/lbm.R

$c_v = 0.0756$ Btu/lbm.R

$k = 1.667$

Step 4 of 9

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{540} = (8)^{1.667-1}$$

$$T_2 = 2161.497\text{R}$$

Step 5 of 9

Process 3-4 as isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$T_4 = T_3\left(\frac{1}{8}\right)^{1.667-1}$$

$$T_4 = 2400 \times \left(\frac{1}{8}\right)^{1.667-1}$$

$$T_4 = 599.58\text{R}$$

Step 6 of 9

(a) Process 2-3 is the constant volume heat addition

$$q_{in} = u_3 - u_2$$

$$= c_v(T_3 - T_2)$$

$$= (0.0756)(2400 - 2161.497)$$

$$\boxed{q_{in} = 18.03\ \text{Btu/lbm}}$$

Step 7 of 9

(b) The thermal efficiency is defined as

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

The amount of heat rejected is expressed as

$$q_{out} = (u_4 - u_1)\ \text{Btu/lbm}$$

$$= c_v(T_4 - T_1)$$

$$= (0.0756)(599.58 - 540)$$

$$q_{out} = 4.5042\ \text{Btu/lbm}$$

Step 8 of 9

Therefore thermal efficiency

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

$$\eta_{th} = 1 - \frac{4.5042}{18.03}$$

$$= 0.75018$$

$$\boxed{\eta_{th} = 75.018\%}$$

Step 9 of 9

(c) The thermal efficiency of a Carnot cycle operating between the same temperature limits is

$$\eta_{carnot} = 1 - \frac{T_L}{T_H}$$

$$\eta_{carnot} = 1 - \frac{T_1}{T_3}$$

$$= 1 - \frac{540}{2400}$$

$$= 0.775$$

$$\boxed{\eta_{Carnot} = 77.5\%}$$

Question:

A four-cylinder, four-stroke. 1.6-L gasoline engine operates on the Otto cycle with a compression ratio of 11.The air is at 100 kPa and 37°C at the beginning of the compression process, and the maximum pressure in the cycle is 8 MPa. The compression and expansion processes may be modeled aspolytropic with a polytropic constant of 1.3. Using constant specific heats at 850 K. determine (*a*) the temperature at the end of the expansion process, (*b*) the net work output and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the engine speed for a net power output of 50 kW. and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the network produced. The air-fuel ratio, defined as the amount of air divided by the amount of fuel intake, is 16.

Step 1 of 9

Calculate the temperature at state-2 using the relation.

$$\frac{T_2}{T_1}=\left(\frac{V_1}{V_2}\right)^{n-1}$$

$$T_2=T_1r^{n-1}$$

Here, T_1 is the temperature at state 1, r is the compression ratio, and n is the polytropic index.

Substitute $310\,\mathrm{K}$ for T_1, 11 for r, and 1.3 for n.

$$T_2=310(11)^{1.3-1}$$
$$=636.47\,\mathrm{K}$$

Calculate the pressure at state-2 using the relation.

$$\frac{P_2}{P_1}=\left(\frac{V_1}{V_2}\right)^{n}$$

$$P_2=P_1r^{n}$$

Here, P_1 is the pressure at state 1.

Substitute $100\,\mathrm{kPa}$ for P_1, 11 for r, and 1.3 for n.

$$P_2=100(11)^{1.3}$$
$$=2258.45\,\mathrm{kPa}$$

Step 2 of 9

Consider process 2-3 is constant volume heat addition process.

Calculate the temperature at state-3 using the relation.

$$\frac{T_3}{T_2}=\frac{P_3}{P_2}$$

$$T_3=T_2\left(\frac{P_3}{P_2}\right)$$

Substitute $636.47\,\mathrm{K}$ for T_2, $8000\,\mathrm{kPa}$ for P_3, and $2258.45\,\mathrm{kPa}$ for P_2.

$$T_3=636.47\left(\frac{8\times10^3}{2258.45}\right)$$
$$=2254.53\ \mathrm{K}$$

Calculate the heat addition in process 2-3.

$$Q_{in}=c_v(T_3-T_2)$$

Substitute $0.823\,\mathrm{kJ/kg\text{-}K}$ for c_v, $2254.53\ \mathrm{K}$ for T_3, and $636.47\,\mathrm{K}$ for T_2.

$$Q_{in}=0.823(2254.53-636.47)$$
$$=1331.669\ \mathrm{kJ/kg}$$

Step 3 of 9

(a)

Consider the process 3-4 is polytropic expansion.

Calculate the temperature at the end of the expansion process.

$$\frac{T_4}{T_3}=\left(\frac{V_3}{V_4}\right)^{n-1}$$

$$T_4=T_3\left(\frac{1}{r}\right)^{n-1}$$

Substitute $2254.53\ \mathrm{K}$ for T_3, 11 for r, and 1.3 for n.

$$T_4=2254.53\times\left(\frac{1}{11}\right)^{1.3-1}$$
$$=1098.09\,\mathrm{K}$$

Therefore, the temperature at the end of the expansion process is $\boxed{1098.09\,\mathrm{K}}$.

Step 4 of 9

(b)

Calculate the heat rejected during the process 4-1.

$$Q_{out}=c_v(T_4-T_1)$$

Substitute $0.823\,\mathrm{kJ/kg\text{-}K}$ for c_v, $1098.09\,\mathrm{K}$ for T_4, and $310\,\mathrm{K}$ for T_1.

$$Q_{out}=0.823(1098.09-310)$$
$$=648.59\,\mathrm{kJ/kg}$$

Calculate the net-work output.

$$W_{net}=Q_{in}-Q_{out}$$

Substitute $648.59\,\mathrm{kJ/kg}$ for Q_{out} and $1331.669\ \mathrm{kJ/kg}$ for Q_{in}.

$$W_{net}=1331.669-648.59$$

$W_{net} = Q_{in} - Q_{out}$

Substitute $648.59\,kJ/kg$ for Q_{out} and $1331.669\,kJ/kg$ for Q_{in}.

$$W_{net} = 1331.669 - 648.59$$
$$= 683.08\,kJ/kg$$

Therefore, the net-work output is $\boxed{683.08\,kJ/kg}$.

Step 5 of 9

Calculate the thermal efficiency of the cycle.

$$\eta_{th} = \frac{Q_{in} - Q_{out}}{Q_{in}}$$

Substitute $648.59\,kJ/kg$ for Q_{out} and $1331.669\,kJ/kg$ for Q_{in}.

$$\eta_{th} = \frac{1331.669 - 648.59}{1331.67}$$
$$= 0.513$$
$$= 51.3\%$$

Therefore, the thermal efficiency of the cycle is $\boxed{51.3\%}$.

Step 6 of 9

(c)

Calculate the Mean effective pressure.

$$m.e.p = \frac{W_{net}}{v_1\left(1 - \frac{1}{r}\right)} \quad \text{..... (1)}$$

Here, v_1 is the specific volume at state-1.

Calculate the specific volume at state 1.

$$v_1 = \frac{RT_1}{P_1}$$

Substitute $0.287\,kJ/kg\text{-}K$ for R, $310\,K$ for T_1, and $100\,kPa$ for P_1.

$$v_1 = \frac{0.287 \times 310}{100}$$
$$= 0.8897\,m^3/kg$$

Substitute $683.08\,kJ/kg$ for W_{net}, $0.8897\,m^3/kg$ for v_1, and 11 for r in equation (1).

$$m.e.p = \frac{W_{net}}{v_1\left(1 - \frac{1}{r}\right)}$$
$$= \frac{683.08}{0.8897\left(1 - \frac{1}{11}\right)}$$
$$= 844.54\,kPa$$

Therefore, the mean effective pressure is $\boxed{844.54\,kPa}$.

Step 7 of 9

(d)

Calculate the clearance volume using the compression ratio.

$$r = \frac{v_c + v_s}{v_c}$$

Here, v_s is the swept volume.

Substitute 11 for r and $1.6 \times 10^{-3}\,m^3$ for v_s.

$$11 = \frac{v_c + \left(1.6 \times 10^{-3}\right)}{v_c}$$
$$10v_c = 1.6 \times 10^{-3}$$
$$v_c = 1.6 \times 10^{-4}\,m^3$$

Calculate the total mass of the air in the cylinder.

$$m = \frac{P_1 V_1}{RT_1}$$

Substitute $0.287\,kJ/kg\text{-}K$ for R, $310\,K$ for T_1, $100\,kPa$ for P_1, and $1.6 \times 10^{-4}\,m^3$ for V_1.

$$m = \frac{100 \times 1.6 \times 10^{-4}}{0.287 \times 310}$$
$$= 1.798 \times 10^{-4}\,kg$$

Step 8 of 9

Calculate the engine speed.

$$P = m \times W_{net} \times n$$
$$n = 2\left(\frac{P}{mW_{net}}\right)$$

Substitute $1.798 \times 10^{-4}\,kg$ for m, $683.08\,kJ/kg$ for W_{net}, and $50\,kW$ for P.

$$n = 2 \times \left(\frac{50}{1.798 \times 10^{-4} \times 683.08}\right)$$
$$= 814\,rev/sec$$

Therefore, the speed of the engine is $\boxed{814\,rev/sec}$.

(e)

$= 814\,\text{rev/sec}$

Therefore, the speed of the engine is $\boxed{814\,\text{rev/sec}}$.

(e)

Calculate the specific fuel consumption.

$$sfc = \frac{m_f}{m_1 \times W_{net}} \quad \text{..... (2)}$$

Here, m_f is the mass of the fuel.

Calculate the mass of the fuel using the air-fuel ratio.

$$AF = \frac{m_a}{m_f}$$

$$AF = \frac{m - m_f}{m_f}$$

Substitute 16 for AF and $1.6\times10^{-4}\,\text{kg}$ for m .

$$16 = \frac{1.6\times10^{-4} - m_f}{m_f}$$

$$m_f = \frac{1.6\times10^{-4}}{17}$$

$$= 9.41\times10^{-6}\,\text{kg}$$

Step 9 of 9

Substitute $9.41\times10^{-6}\,\text{kg}$ for m_f , $1.6\times10^{-4}\,\text{kg}$ for m , and $683.08\,\text{kJ/kg}$ for W_{net} in equation (2).

$$sfc = \frac{m_f}{m_1 \times W_{net}}$$

$$= \frac{9.41\times10^{-6}}{1.6\times10^{-4}\times683.08}$$

$$= 8.609\times10^{-5}\,\text{kg/kJ}$$

$$= 309.96\,\text{g/kWh}$$

Therefore, the specific fuel consumption is $\boxed{309.96\,\text{g/kWh}}$.

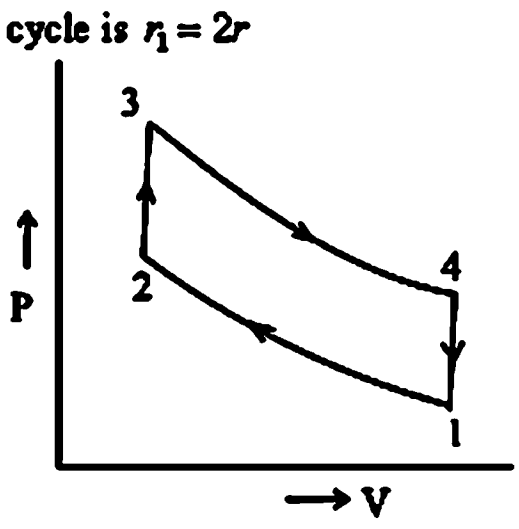
Question: 9.042P

When we double the compression ratio of an ideal Otto cycle, what happens to the maximum gas temperature and pressure when the state of the air at the beginning of the compression and the amount of heat addition remain the same? Use constant specific heats at room temperature.

Step 1 of 8

Given

Compression ratio of Otto cycle is $r_1 = 2r$



Step 2 of 8

An ideal Otto cycle consists of four internally reversible processes:
1-2 Isentropic compression
2-3 Constant-volume heat addition
3-4 Isentropic expansion
4-1 Constant-volume heat rejection

Step 3 of 8

For a compression ratio r,

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = T_1 r^{k-1}$$

If the compression ratio double then

$$T_2' = T_1'(2r)^{k-1}$$

$$T_2' = 1.32 T_1' r^{k-1}$$

Step 4 of 8

Since the amount of heat addition remains constant
The maximum temperature is expressed as

$$q_{in} = c_v (T_3 - T_2) \text{ kJ/kg}$$

$$T_3 = \frac{q_{in}}{c_v} + T_2$$

When the compression ratio is r the maximum temperature will be

$$T_3 = \frac{q_{in}}{c_v} + T_1 r^{k-1} \text{ ---------- (1)}$$

Step 5 of 8

When the compression ratio is 2r the maximum temperature will be

$$T_3' = \frac{q_{in}}{c_v} + T_2'$$

$$T_3' = \frac{q_{in}}{c_v} + 1.32\left(T_1' r^{k-1}\right) \text{ ---------- (2)}$$

Comparing equations 1 and 2 the maximum temperature increases when the compression ratio increases.

Step 6 of 8

The smallest gas specific volume is obtained at state 3

$$\frac{v_2}{v_1} = \frac{1}{r}$$

$$\frac{v_3}{v_1} = \frac{1}{r} \qquad (\text{Since } v_2 = v_3)$$

$$v_3 = \frac{v_1}{r}$$

When the compression ratio double then

$$v_3' = \frac{v_1'}{2r}$$

Step 7 of 8

By using ideal gas

$$P_3 v_3 = R T_3$$

$$P_3 = \frac{R T_3}{v_3}$$

By substituting T_3 and v_3 then

$$P_3 = \frac{R\left[\frac{q_{in}}{c_v} + \left(T_1 r^{k-1}\right)\right]}{v_1 / r}$$

$$P_3 = \frac{Rr\left[\frac{q_{in}}{c_v} + \left(T_1 r^{k-1}\right)\right]}{v_1} \text{ ---------- (3)}$$

Step 8 of 8

When the compression ratio doubles then the maximum pressure will be

$$P_3' = \frac{R\left[\frac{q_{in}}{c_v} + 1.32\left(T_1' r^{k-1}\right)\right]}{v_1' / 2r}$$

$$P_3' = \frac{R \times 2r\left[\frac{q_{in}}{c_v} + 1.32\left(T_1' r^{k-1}\right)\right]}{v_1'} \text{ ---------- (4)}$$

Comparing 3 and 4 equations as compression ratio increases maximum pressure increases

Question: 9.043P

In a spark-ignition engine, some cooling occurs as the gas is expanded. This may be modeled by using a polytropic process in lieu of the isentropic process. Determine if the polytropic exponent used in this model will be greater than or less than the isentropic exponent.

Step 1 of 3

In spark ignition engine, during the expansion process in case of isentropic expansion

$$Pv^k = cons$$

$$\frac{T_{4'}}{T_3} = \left(\frac{P_{4'}}{P_3}\right)^{\frac{k-1}{k}}$$

$$T_{4'} = T_3\left(\frac{P_{4'}}{P_3}\right)^{\frac{k-1}{k}}$$

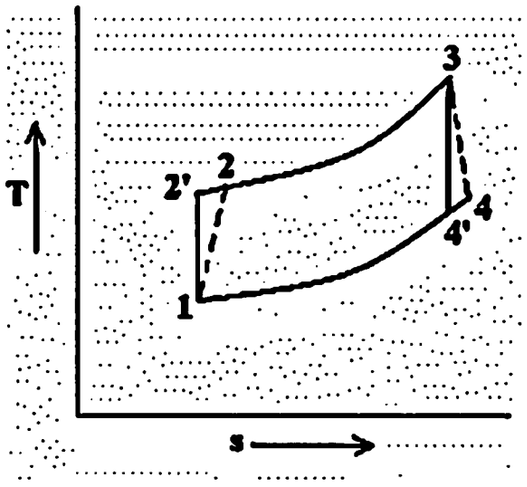
In case of polytropic expansion

$$Pv^n = const$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}}$$

Step 2 of 3



Step 3 of 3

From the above figure the exit temperature during the polytropic process is greater than the exit temperature during the isentropic process

$$T_{4\,poly} > T_{4\,ideal}$$

$$T_3\left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}} > T_3\left(\frac{P_{4'}}{P_3}\right)^{\frac{k-1}{k}} \qquad (\text{Since } P_{4'} = P_4)$$

$$\frac{n-1}{n} > \frac{k-1}{k}$$

$$\frac{1}{n} > \frac{1}{k}$$

From the above statement it is clear that $n \le k$

Question: 9.044P

How does a diesel engine differ from a gasoline engine?

Step 1 of 1

In gasoline engines the air–fuel mixture is compressed to a temperature that is below the auto ignition temperature of the fuel, and the combustion process is initiated by firing a spark plug.

In CI engines (also known as diesel engines), the air is compressed to a temperature that is above the auto ignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines.

Question: 9.045P

How does the ideal Diesel cycle differ from the ideal Otto cycle?

Step 1 of 1

The Diesel cycle differs from the Otto cycle in the heat addition process.
Heat addition takes place at constant volume in the Otto cycle, but in the Diesel cycle heat addition takes place at constant pressure.

Question: 9.046P

For a specified compression ratio, is a diesel or gasoline engine more efficient?

Step 1 of 1

The thermal efficiency of the Otto cycle is given by

$$\eta = 1 - \frac{1}{r^{k-1}}$$

The thermal efficiency of the diesel cycle is given by

$$\eta = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]$$

Where

r is compression ratio

r_c is cutoff ratio

Equation for the diesel cycle differs from equation for the Otto cycle only by the term in brackets, which for $r_c > 1$ is greater than unity. Thus, when the compression ratio is the same, the thermal efficiency of the cold air standard diesel cycle would be less than that of the cold air standard Otto cycle

Question: 9.047P

Do diesel or gasoline engines operate at higher compression ratios? Why?

Step 1 of 1

In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of auto ignition or engine knock.

In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of auto ignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24. Fuels that are less refined can be used in diesel engines.

Question: 9.048P

What is the cutoff ratio? How does it affect the thermal efficiency of a Diesel cycle?

Step 1 of 1

Write the equation for cutoff ratio r_c.

$$r_c = \frac{v_3}{v_2}$$

Write the equation for thermal efficiency of a Diesel engine.

$$\eta_{th} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_c^k - 1}{k(r_c - 1)}\right)$$

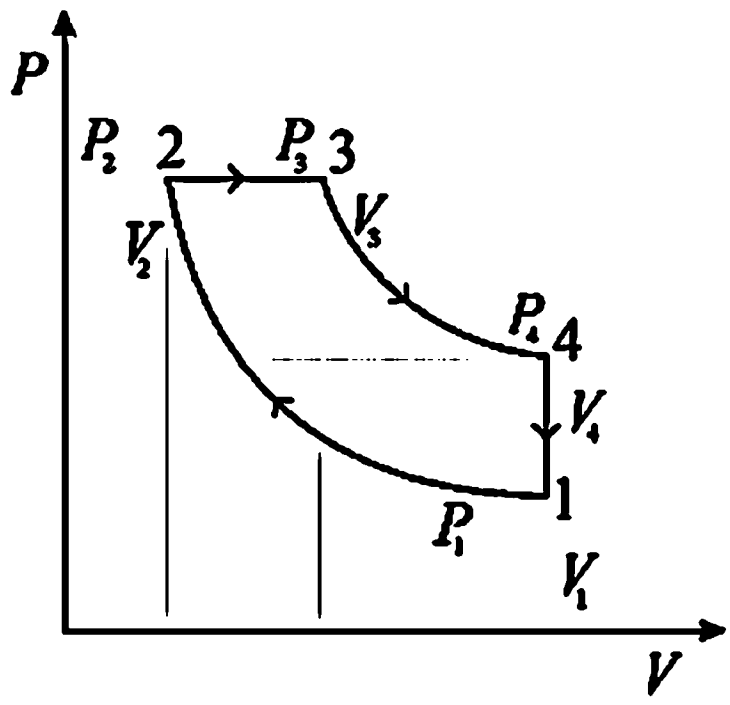
Since, in the equation thermal efficiency of the engine increases with decrease in cutoff ratio. In the limiting case the cutoff ratio is equal to 1 where efficiencies of diesel engine and gasoline engine will be identical.

Question: 9.049P

An ideal Diesel cycle has a compression ratio of 20 and a cutoff ratio of 1.3. Determine the maximum temperature of the air and the rate of heat addition to this cycle when it produces 250 kW of power and the state of the air at the beginning of the compression is 90 kPa and 15°C. Use constant specific heats at room temperature.

Step 1 of 5

The *P-V* diagram of ideal diesel cycle shown in figure:



Step 2 of 5

Process 1-2 is an Isentropic Compression.

$$T_1V_1^{\gamma-1} = T_2V_2^{\gamma-1}$$

$$T_2 = T_1 \times \frac{V_1^{\gamma-1}}{V_2^{\gamma-1}}$$

$$= T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

Here, T_2 and T_1 is the temperature at state points at 1 and 2 respectively and V_1 and V_2 are the volume at sate points 1 and 2 respectively.

Substitute 288K for T_1 and 20 for $\frac{V_1}{V_2}$

$$T_2 = 288(20)^{1.4-1}$$

$$= 954.56 \text{ K}$$

Step 3 of 5

Write the relation for process 2 and 3

$$T_2V_2 = T_3V_3$$

$$T_3 = T_2 \times \frac{V_3}{V_2}$$

$$= T_2\left(\frac{V_3}{V_2}\right)$$

$$= T_2(r_e)$$

Here, r_e is the expansion processes

Substitute 1.3 for r_e

$$T_3 = 954.56 \times 1.3$$

$$T_3 = 1240.93 \text{ K}$$

Therefore, Maximum temperature in the cycle is $\boxed{1240.93 \text{ K}}$

Step 4 of 5

Calculate the Thermal Efficiency of the Diesel cycle,

$$\eta_{th} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_e^{\gamma} - 1}{\gamma(r_e - 1)}\right]$$

$$= 1 - \frac{1}{20^{1.4-1}}\left[\frac{1.3^{1.4} - 1}{1.4(1.3-1)}\right]$$

$$= 1 - \frac{1.056775}{3.31445}$$

$$= 1 - 0.31884$$

$$= 0.681$$

Step 5 of 5

Calculate the Heat transferred

$$Q_{in} = \frac{W_{net}}{\eta_{th}}$$

Here, W_{net} is the net work done through a processes

Substitute 250 kW for W_{net} and 0.681 for η_{th}

$$Q_{in} = \frac{250}{0.681}$$

$$= 367.11 \text{ kW}$$

Hence heat input to the cycle is $\boxed{367.11 \text{ kW}}$

Question: 9.050P

An ideal Diesel cycle has a maximum cycle temperature of 2300°F and a cutoff ratio of 1.4. The state of the air at the beginning of the compression is $P1$ = 14.4 psia and $T1$= 50°F. This cycle is executed in a four-stroke, eight-cylinder engine with a cylinder bore of 4 in and a piston stroke of 4 in. The minimum volume enclosed in the cylinder is 4.5 percent of the maximum cylinder volume. Determine the power produced by this engine when it is operated at 1800 rpm. Use constant specific heats at room temperature.

Step 1 of 10

Given data:

Cutoff ratio of the given diesel cycle, $r_c = 1.4$

Initial pressure of the given cycle, $P_1 = 14.4\,\text{psi}$

Initial temperature of the given cycle, $T_1 = 50°\text{F} = 509.67°\text{R}$

Maximum Temperature of the given cycle, $T_3 = 2300°\text{F} = 2759.67°\text{R}$

Cylinder bore, $d = 4\,\text{in.}$

Piston stroke, $L = 4\,\text{in.}$

$$\left(\begin{matrix}\text{Minimum volume enclosed in the cylinder is}\\ 5\text{ percent of the maximum cylinder volume}\end{matrix}\right) V_2 = 0.045 \times V_1$$

Speed of the operating engine, $N = 1800\,\text{rpm}$

Number of cylinders, $n = 8$

Step 2 of 10

Considering constant specific heats at room temperature

Gas constant $R = 0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$

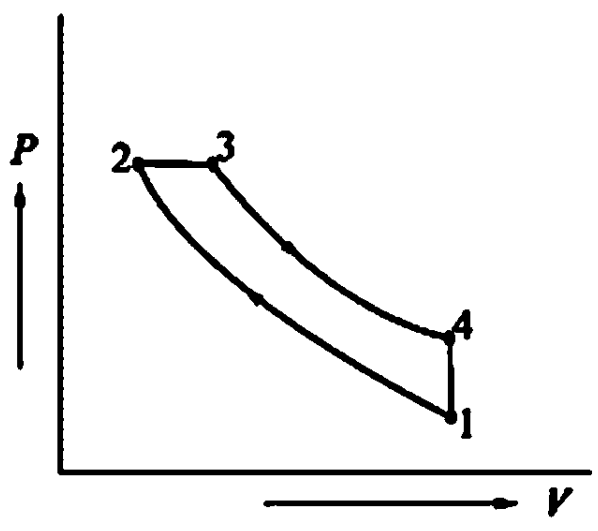
$c_p = 0.240\,\text{Btu/lbm}\cdot\text{R}$

$c_v = 0.171\,\text{Btu/lbm}\cdot\text{R}$

$k = 1.4$

Step 3 of 10

Representation of the diesel cycle in PV-coordinates:



Step 4 of 10

An ideal Diesel cycle consists of four internally reversible processes:

1-2 isentropic compression

2-3 Constant-pressure heat addition

3-4 isentropic expansion

4-1 Constant-volume heat rejection

Step 5 of 10

The compression ratio is defined as volume before compression to volume after compression

$$r = \frac{V_1}{V_2}$$

$$= \frac{1}{0.045}$$

$$r = 22.2$$

Step 6 of 10

From the isentropic compression Process 1-2, we have

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 509.67\left(22.2^{1.4-1}\right)$$

$$T_2 = 1762°\text{R}$$

Step 7 of 10

Thermal efficiency of a Diesel engine,

$$\eta_{th} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_c^k - 1}{k(r_c - 1)}\right)$$

$$= 1 - \frac{1}{22.2^{1.4-1}}\left(\frac{1.4^{1.4} - 1}{1.4(1.4-1)}\right)$$

$$\eta_{th} = 0.689$$

Step 8 of 10

The heat supplied during the process 2-3:

$$Q_{in} = \dot{m}c_p(T_3 - T_2)$$

Where, total mass flow rate of air

$$\dot{m} = n\rho\dot{V}$$

$$\dot{m} = n \times \frac{p_1}{RT_1} \times \frac{\pi}{4}d^2L \times \frac{N}{2}$$

$$= 8 \times \frac{14.4}{0.3704 \times 509.67} \times \frac{\pi}{4} \times \left(\frac{4}{12}\right)^2 \times \frac{4}{12} \times \frac{1800}{2}$$

$$= 15.975\,\text{lb/min}$$

(Since there are two revolutions per cycle in a four stroke engine $N = \frac{N}{2}$)

Step 9 of 10

Heat supplied of the given cycle,

$$Q_{in} = \dot{m}c_p(T_3 - T_2)$$

$$= 15.975 \times 0.24(2759.67 - 1762)$$

$$= 3825.244\,\text{Btu/min}$$

Step 10 of 10

The efficiency of the given cycle,

$$\eta_{th} = \frac{\dot{W}_{net}}{Q_{in}}$$

$$\dot{W}_{net} = Q_{in} \times \eta_{th}$$

$$\dot{W}_{net} = 3825.224 \times 0.689$$

$$= 2635.59\,\text{Btu/min}$$

Therefore, the net work done of the given cycle, $\boxed{\dot{W}_{net} = 2635.59\,\text{Btu/min}}$

Question:

An air-standard dual cycle has a compression ratio of 14 and a cut off ratio of 1.2. The pressure ratio during the constant-volume heat addition process is 1.5. Determine the thermal efficiency, amount of heat added, the maximum gas pressure and temperature when this cycle is operated at 80 kPa and 20°C at the beginning of the compression. Use constant specific heats at room temperature.

Step 1 of 10

Given,

Compression ratio of the Diesel cycle, $r = 14$

Cutoff ratio, $r_c = 1.2$

Pressure ratio, $r_p = 1.5$

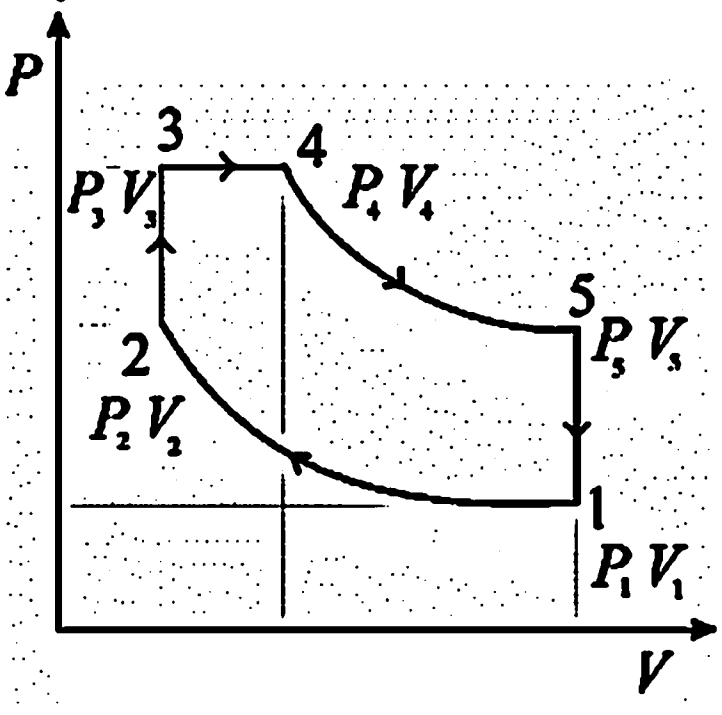
Temperature of air before compression, $T_1 = 20°C$

$T_1 = 293\ K$

Pressure of air before compression, $P_1 = 80\ kPa$

Step 2 of 10

The *P-V* diagram of dual cycle:



Step 3 of 10

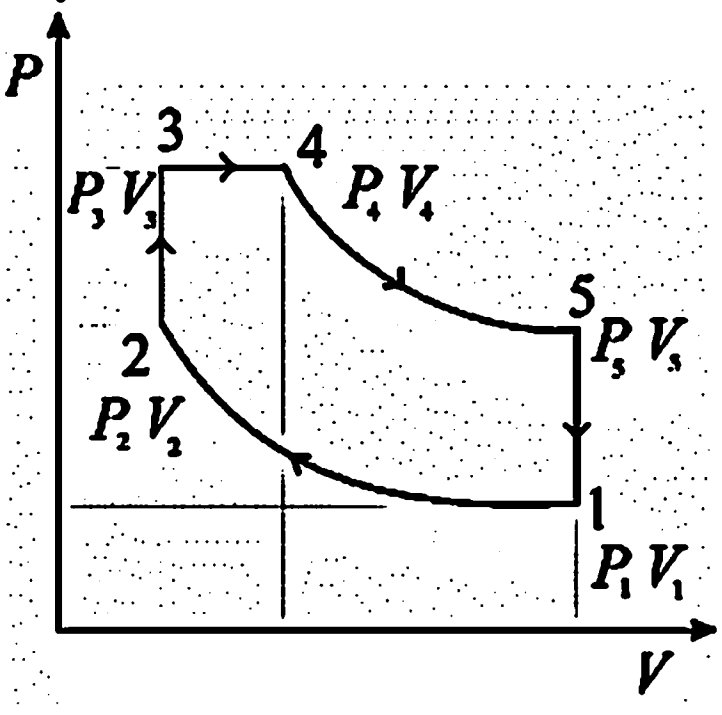
Process 1-2 is an Isentropic Compression. Hence it follows, $T_1 v_1^{\gamma-1} = T_2 v_2^{\gamma-1}$

$$T_2 = T_1 \times \frac{v_1^{\gamma-1}}{v_2^{\gamma-1}}$$

$$T_2 = T_1 \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$

$$T_2 = T_1 (r)^{\gamma-1}$$

The *P-V* diagram of dual cycle:



Step 3 of 10

Process 1-2 is an Isentropic Compression. Hence it follows, $T_1 v_1^{\gamma-1} = T_2 v_2^{\gamma-1}$

$$T_2 = T_1 \times \frac{v_1^{\gamma-1}}{v_2^{\gamma-1}}$$

$$T_2 = T_1 \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$

$$T_2 = T_1 \left(\frac{r}{r_2}\right)^{\gamma-1}$$

$$T_3 = T_2 (r_p)$$

$$T_3 = 842 \times (1.5)$$

$$T_3 = 1263\ K$$

Pressure at state 3 is $P_3 = r_p \times P_2$

$$P_3 = 1.5 \times 3218.616$$

$$P_3 = 4827.924\ kPa$$

Volume at state 3 is $v_3 = v_2$

$$v_3 = 0.075\ m^3/kg$$

Step 6 of 10

Process 3-4 is of constant pressure. Hence it follows, $\frac{v_3}{T_3} = \frac{v_4}{T_4}$

$$T_4 = T_3 \left(\frac{v_4}{v_3}\right)$$

$$T_4 = T_3 (r_c)$$

$$T_4 = 1263 \times (1.2)$$

$$T_4 = 1515.6\ K$$

Hence the maximum temperature in the cycle is $\boxed{T_4 = 1515.6\ K}$

Step 7 of 10

And pressure at state 4 is $P_4 = P_3$

$$P_4 = 4827.924\ kPa$$

Hence maximum pressure in the cycle is $\boxed{P_4 = 4827.924\ kPa}$

Also $v_4 = r_c \times v_3$

$$v_4 = 1.2 \times 0.075$$

9 051P pg 2

Hence maximum pressure in the cycle is $\boxed{P_4 = 4827.924 \text{ kPa}}$

Also $v_4 = r_c \times v_3$

$v_4 = 1.2 \times 0.075$

$v_4 = 0.09 \text{ m}^3/\text{kg}$

Step 8 of 10

Process 4-5 is an isentropic expansion. Hence it follows, $T_4 v_4^{\gamma-1} = T_5 v_5^{\gamma-1}$

$$T_5 = T_4 \times \frac{v_4^{\gamma-1}}{v_5^{\gamma-1}}$$

$$T_5 = T_4 \left(\frac{v_4}{v_1}\right)^{\gamma-1}$$

$$T_5 = T_4 \left(\frac{v_4}{r \times v_2}\right)^{\gamma-1}$$

$$T_5 = 1515.6 \left(\frac{0.09}{1.05}\right)^{1.4-1}$$

$$T_5 = 567.3 \text{ K}$$

Step 9 of 10

Heat added in process 2-3

$$q_{2-3} = c_v (T_3 - T_2)$$

$$q_{2-3} = 0.718 \times (1263 - 842)$$

$$q_{2-3} = 302.278 \text{ kJ/kg}$$

Heat added in process 3-4

$$q_{3-4} = c_p (T_4 - T_3)$$

$$q_{3-4} = 1.005 \times (1515.6 - 1263)$$

$$q_{3-4} = 253.863 \text{ kJ/kg}$$

Total heat input to cycle is $q_{in} = q_{2-3} + q_{3-4}$

$$q_{in} = 302.278 + 253.863$$

$$q_{in} = 556.141 \text{ kJ/kg}$$

Step 10 of 10

Heat rejected in process 5-1

$$q_{5-1} = c_v (T_5 - T_1)$$

$$q_{4-5} = 0.718 \times (567.3 - 293)$$

$$q_{4-5} = 196.95 \text{ kJ/kg}$$

Amount of heat rejected in the cycle is $q_{out} = q_{4-5}$

$$q_{out} = 196.95 \text{ kJ/kg}$$

Step 9 of 10

Heat added in process 2-3

$$q_{2-3} = c_v (T_3 - T_2)$$

$$q_{2-3} = 0.718 \times (1263 - 842)$$

$$q_{2-3} = 302.278 \text{ kJ/kg}$$

Heat added in process 3-4

$$q_{3-4} = c_p (T_4 - T_3)$$

$$q_{3-4} = 1.005 \times (1515.6 - 1263)$$

$$q_{3-4} = 253.863 \text{ kJ/kg}$$

Total heat input to cycle is $q_{in} = q_{2-3} + q_{3-4}$

Question: 9.052P pg 1

Repeat Prob. 9–51 when the state of the air at the beginning of the compression is 80 kPa and –20°C

Step 1 of 10

Given,

Compression ratio of the Diesel cycle, $r = 14$

Cutoff ratio, $r_c = 1.2$

Pressure ratio, $r_p = 1.5$

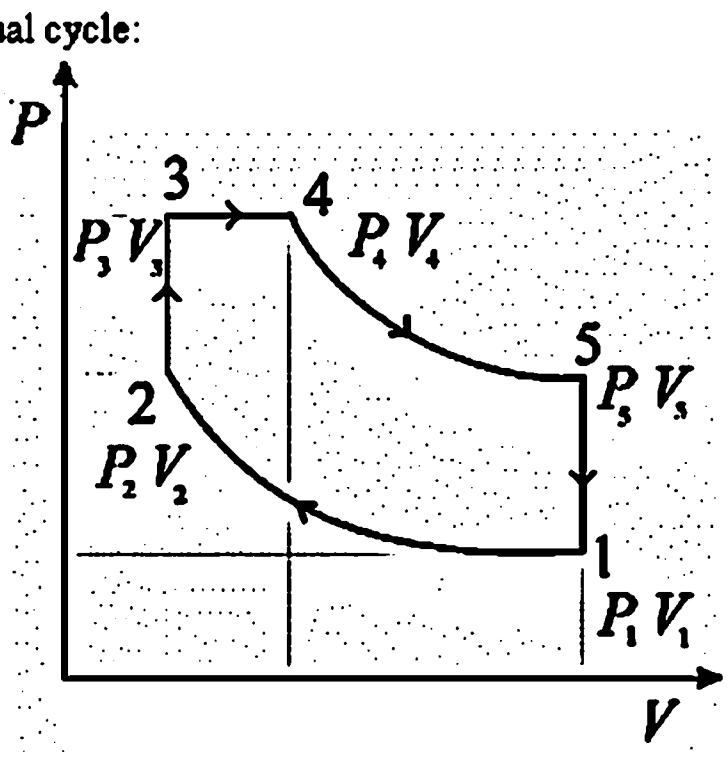
Temperature of air before compression, $T_1 = -20°C$

$T_1 = 253\ K$

Pressure of air before compression, $P_1 = 80\ kPa$

Step 2 of 10

The P-V diagram of dual cycle:



Step 3 of 10

Process 1-2 is an Isentropic Compression. Hence it follows, $T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$

$$T_2 = T_1 \times \frac{V_1^{\gamma-1}}{V_2^{\gamma-1}}$$

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

$$T_2 = T_1 (r)^{\gamma-1}$$

$$T_2 = 253(14)^{1.4-1}$$

$$T_2 = 727\ K$$

Step 4 of 10

Pressure at state 2 is $P_2 = P_1 \left(\frac{v_1}{v_2}\right)^{\gamma}$

$$P_2 = 80(14)^{1.4}$$

$$P_2 = 3218.616\ kPa$$

Volume at state 2 is $v_2 = \frac{v_1}{r}$

$$v_2 = \frac{RT_1}{rP_1}$$

$$v_2 = \frac{0.287 \times 273}{14 \times 80}$$

$$v_2 = 0.07\ m^3/kg$$

Step 5 of 10

Process 2-3 is of constant volume. Hence it follows, $\frac{P_2}{T_2} = \frac{P_3}{T_3}$

$$T_3 = T_2 \left(\frac{P_3}{P_2}\right)$$

$$T_3 = T_2 (r_p)$$

$$T_3 = 727 \times (1.5)$$

$$T_3 = 1090.5\ K$$

Pressure at state 3 is $P_3 = r_p \times P_2$

$$P_3 = 1.5 \times 3218.616$$

$$P_3 = 4827.924\ kPa$$

Volume at state 3 is $v_3 = v_2$

$$v_3 = 0.07\ m^3/kg$$

Step 6 of 10

Process 3-4 is of constant pressure. Hence it follows, $\frac{v_3}{T_3} = \frac{v_4}{T_4}$

$$T_4 = T_3 \left(\frac{v_4}{v_3}\right)$$

$$T_4 = T_3 (r_c)$$

$$T_4 = 1090.5 \times (1.2)$$

$$T_4 = 1308.6\ K$$

Hence the maximum temperature in the cycle is $\boxed{T_4 = 1308.6\ K}$

Step 7 of 10

And pressure at state 4 is $P_4 = P_3$

$$P_4 = 4827.924\ kPa$$

Hence maximum pressure in the cycle is $\boxed{P_4 = 4827.924\ kPa}$

Also $v_4 = r_c \times v_3$

$$v_4 = 1.2 \times 0.07$$

$$v_4 = 0.084\ m^3/kg$$

Also $v_4 = r_c \times v_3$

$v_4 = 1.2 \times 0.07$

$v_4 = 0.084 \text{ m}^3/\text{kg}$

Step 8 of 10

Process 4-5 is an isentropic expansion. Hence it follows, $T_4 v_4^{\gamma-1} = T_5 v_5^{\gamma-1}$

$$T_5 = T_4 \times \frac{v_4^{\gamma-1}}{v_5^{\gamma-1}}$$

$$T_5 = T_4 \left(\frac{v_4}{v_1}\right)^{\gamma-1}$$

$$T_5 = T_4 \left(\frac{v_4}{r \times v_2}\right)^{\gamma-1}$$

$$T_5 = 1308.6 \left(\frac{0.084}{0.98}\right)^{1.4-1}$$

$$T_5 = 489.8 \text{ K}$$

Step 9 of 10

Heat added in process 2-3

$$q_{2-3} = c_v (T_3 - T_2)$$

$$q_{2-3} = 0.718 \times (1090.5 - 727)$$

$$q_{2-3} = 260.993 \text{kJ/kg}$$

Heat added in process 3-4

$$q_{3-4} = c_p (T_4 - T_3)$$

$$q_{3-4} = 1.005 \times (1308.6 - 1090.5)$$

$$q_{3-4} = 219.19 \text{ kJ/kg}$$

Total heat input to cycle is $q_{in} = q_{2-3} + q_{3-4}$

$$q_{in} = 260.993 + 219.19$$

$$q_{in} = 480.183 \text{ kJ/kg}$$

Step 10 of 10

Heat rejected in process 5-1

$$q_{5-1} = c_v (T_5 - T_1)$$

$$q_{4-5} = 0.718 \times (489.8 - 253)$$

$$q_{4-5} = 170.02 \text{ kJ/kg}$$

Amount of heat rejected in the cycle is $q_{out} = q_{4-5}$

$$q_{out} = 170.02 \text{ kJ/kg}$$

Efficiency of the cycle is $\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$

$$\eta_{th} = 1 - \frac{170.02}{480.183}$$

$$\eta_{th} = 0.646$$

Hence efficiency of the cycle is $\boxed{\eta_{th} = 64.6\%}$

Question:

An air-standard Diesel cycle has a compression ratio of 18.2. Air is at 80°F and 14.7 psia at the beginning of the compression process and at 3000 R at the end of the heat-addition process. Accounting for the variation of specific heats with temperature, determine (*a*) the cutoff ratio, (*b*) the heat rejection per unit mass, and (*c*) the thermal efficiency.

Step 1 of 14

Given

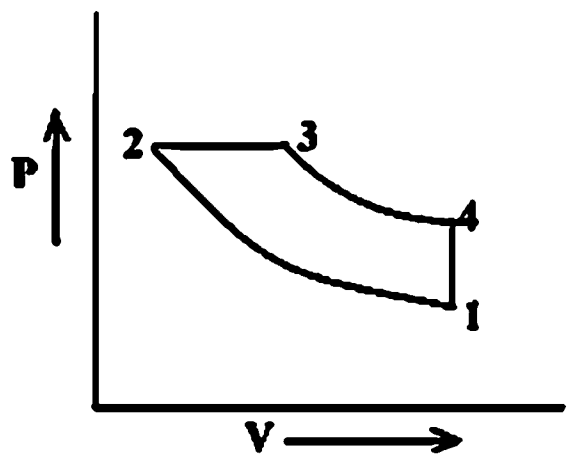
Compression ratio $r = 18.2$

Initial pressure $P_1 = 14.7$ psia

Initial temperature $T_1 = 80°F = 540R$

Maximum temperature is $T_3 = 3000R$

Step 2 of 14



Step 3 of 14

An ideal Diesel cycle consists of four internally reversible processes:

1-2 Isentropic compression

2-3 Constant-pressure heat addition

3-4 Isentropic expansion

4-1 Constant-volume heat rejection

Step 4 of 14

Assuming variation of specific heats with temperature

At $T_1 = 540R$ from ideal gas properties of air

$u_1 = 92.04$ Btu/lbm

$v_{r1} = 144.32$

Step 5 of 14

We know that

$$\frac{v_1}{v_2} = \frac{v_{r1}}{v_{r2}}$$

$$v_{r2} = v_{r1} \times \frac{v_2}{v_1}$$

$$v_{r2} = v_{r1} \times \frac{1}{r}$$

$$v_{r2} = \frac{144.32}{18.2}$$

$$v_{r_2} = 7.929$$

Step 6 of 14

At $v_{r_2} = 7.929$ from ideal gas properties of air

$$h_2 = 395.74 + (409.13 - 395.74)\left(\frac{7.929 - 8.263}{7.556 - 8.263}\right)$$

$$h_2 = 402.065 \text{Btu/lbm}$$

$$T_2 = 1600 + (1650 - 1600)\left(\frac{7.929 - 8.263}{7.556 - 8.263}\right)$$

$$T_2 = 1623.62R$$

Step 7 of 14

At $T_3 = 3000R$ from ideal gas properties of air

$h_3 = 790.68$ Btu/lbm

$v_{r3} = 1.18$

Step 8 of 14

(a)

The cut-off ratio is expressed as

$$r_c = \frac{v_3}{v_2}$$

Process 2-3 is constant pressure heat addition

$$\frac{v_3}{v_2} = \frac{T_3}{T_2}$$

$$r_c = \frac{3000}{1623.62}$$

$$\boxed{r_c = 1.8477}$$

Step 9 of 14

(b)

The heat rejection per unit mass

$q_{out} = u_4 - u_1$ Btu/lbm

Step 10 of 14

Process 3-4 is constant volume heat addition

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3}$$

$$\frac{v_{r4}}{v_{r3}} = \frac{v_1}{v_3}$$

Process 3-4 is constant volume heat addition

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3}$$

$$\frac{v_{r4}}{v_{r3}} = \frac{v_1}{v_3}$$

$$\frac{v_{r4}}{v_{r3}} = \frac{v_1}{v_2} \times \frac{v_2}{v_3}$$

$$\frac{v_{r4}}{v_{r3}} = r \times \frac{1}{r_c}$$

$$v_{r4} = \frac{r}{r_c} v_{r3}$$

$$v_{r4} = \frac{18.2}{1.8477}(1.18)$$

$$v_{r4} = 11.623$$

Step 11 of 14

At $v_{r4} = 11.623$ from ideal gas properties of air

$$u_4 = 246.93 + (254.66 - 246.93)\left(\frac{11.623 - 12.10}{11.17 - 12.10}\right)$$

$$u_4 = 250.894 \text{ Btu/lbm}$$

Step 12 of 14

Therefore the amount of heat rejection is

$$q_{out} = u_4 - u_1 \text{ Btu/lbm}$$

$$= 250.894 - 92.04$$

$$\boxed{q_{out} = 158.85 \text{ Btu/lbm}}$$

Step 13 of 14

(c)

The thermal efficiency

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Where q_{in} heat supplied is expressed as

$$q_{in} = h_3 - h_2 \text{ Btu/lbm}$$

$$= 790.68 - 402.065$$

$$q_{in} = 388.615 \text{ Btu/lbm}$$

Step 14 of 14

Therefore thermal efficiency is

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

$$\eta_{th} = 1 - \frac{158.85}{388.615}$$

$$= 0.5912$$

$$\boxed{\eta_{th} = 59.12\%}$$

Question: 9.054P

Repeat Prob. 9-53E using constant specific heats at room temperature.

Step 1 of 10

Given

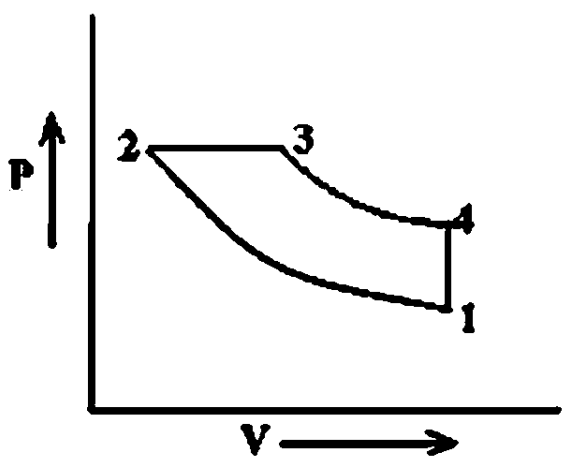
Compression ratio $r = 18.2$

Initial pressure $P_1 = 14.7$ psia

Initial temperature $T_1 = 80°F = 540R$

Maximum temperature is $T_3 = 3000R$

Step 2 of 10



Step 3 of 10

An ideal Diesel cycle consists of four internally reversible processes:

1-2 Isentropic compression

2-3 Constant-pressure heat addition

3-4 Isentropic expansion

4-1 Constant-volume heat rejection

Step 4 of 10

Considering constant specific heats at room temperature

Gas constant $R = 0.3704$ Btu/lbm.R

$c_p = 0.24$ Btu/lbm.R

$c_v = 0.171$ Btu/lbm.R

$k = 1.4$

Step 5 of 10

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 540\left(18.2^{1.4-1}\right)$$

$$T_2 = 1723.5R$$

Step 6 of 10

(a)

The cut-off ratio is expressed as

$$r_c = \frac{v_3}{v_2}$$

Process 2-3 is constant pressure heat addition

$$\frac{v_3}{v_2} = \frac{T_3}{T_2}$$

$$r_c = \frac{3000}{1723.5}$$

$$\boxed{r_c = 1.74}$$

Step 7 of 10

(b)

The heat rejection per unit mass is expressed as

$$q_{out} = u_4 - u_1$$

$$q_{out} = c_v (T_4 - T_1)$$

Process 4-1 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_2} \times \frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(r_c \times \frac{1}{r}\right)^{k-1}$$

$$T_4 = (3000)\left(\frac{1.74}{18.2}\right)^{1.4-1}$$

$$T_4 = 1173.038R$$

Step 8 of 10

Therefore heat rejection is

$$q_{out} = c_v (T_4 - T_1)$$

$$= (0.171)(1173.038 - 540)$$

$$\boxed{q_{out} = 108.24 \text{ Btu/lbm}}$$

Step 9 of 10

(c)

The thermal efficiency

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Heat supplied is expressed as

$$q_{in} = h_3 - h_2$$

$$= c_p (T_3 - T_2)$$

$$= (0.24)(3000\text{-}1723.5)$$

$$q_{in} = 306.36 \text{ Btu/lbm}$$

Step 10 of 10

Therefore the thermal efficiency

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

$$\eta_{th} = 1 - \frac{108.24}{306.36}$$

$$= 0.64669$$

$$\boxed{\eta_{th} = 64.669\%}$$

Question:

An ideal diesel engine has a compression ratio of 20 and uses air as the working fluid. The state of air at the beginning of the compression process is 95 kPa and 20°C. If the maximum temperature in the cycle is not to exceed 2200 K, determine (*a*) the thermal efficiency and (*b*) the mean effective pressure. Assume constant specific heats for air at room temperature.

Step 1 of 12

Given

Compression ratio $r = 20$

Initial pressure $P_1 = 95\ \text{kPa}$

Initial temperature $T_1 = 20°\text{C} = 293\text{K}$

Maximum Temperature $T_3 = 2200\text{K}$

Step 2 of 12

Considering constant specific heats at room temperature

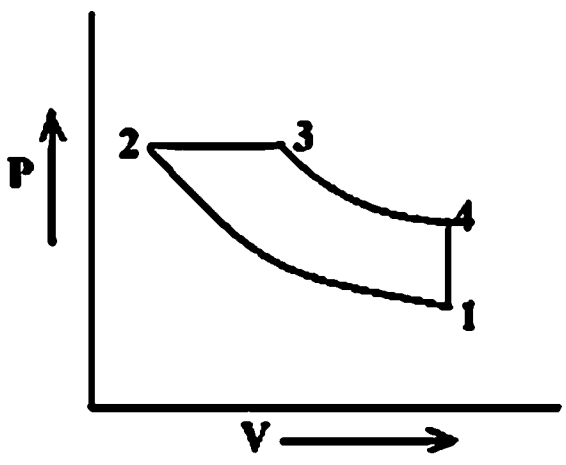
Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 12



Step 4 of 12

An ideal Diesel cycle consists of four internally reversible processes:

1-2 Isentropic compression

2-3 Constant-pressure heat addition

3-4 Isentropic expansion

4-1 Constant-volume heat rejection

Step 5 of 12

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 293\left(20^{1.4-1}\right)$$

$$T_2 = 971.13\text{K}$$

Step 6 of 12

Process 2-3 is constant pressure heat addition

$$\frac{v_3}{v_2} = \frac{T_3}{T_2}$$

$$r_c = \frac{2200}{971.13}$$

$$r_c = 2.265$$

Step 7 of 12

Process 3-4 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_2} \times \frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(r_c \times \frac{1}{r}\right)^{k-1}$$

$$T_4 = (2200)\left(\frac{2.265}{20}\right)^{1.4-1}$$

$$T_4 = 920.53\text{K}$$

Step 8 of 12

Heat supplied is expressed as

$$q_{in} = h_3 - h_2$$

$$= c_p\left(T_3 - T_2\right)$$

$$q_{in} = (1.005)(2200\text{-}971.13)$$

$$q_{in} = 1235.014\ \text{kJ/kg}$$

Step 9 of 12

The heat rejected per unit mass is expressed as

$$q_{out} = u_4 - u_1$$

$$= c_v\left(T_4 - T_1\right)$$

$$= 0.718(920.53 - 293)$$

$$q_{out} = 450.566\ \text{kJ/kg}$$

$$= c_v (T_4 - T_1)$$
$$= 0.718(920.53 - 293)$$
$$q_{out} = 450.566 \text{ kJ/kg}$$

Step 10 of 12

(a)

The thermal efficiency

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{450.566}{1235.014}$$
$$= 0.635$$
$$\boxed{\eta_{th} = 63.5\%}$$

Step 11 of 12

(b)

The Mean effective pressure

$$MEP = \frac{w_{net}}{v_1 - v_2}$$

Work net is expressed as

$$w_{net} = q_{in} - q_{out}$$
$$= 1235.014 - 450.566$$
$$w_{net} = 784.44 \text{ kJ/kg}$$

Specific volume

$$v_1 = \frac{RT_1}{P_1}$$
$$v_1 = \frac{0.287 \times 293}{95}$$
$$v_1 = 0.885 \text{m}^3/\text{kg}$$

Step 12 of 12

Then the mean effective pressure

$$MEP = \frac{w_{net}}{v_1 - v_2}$$
$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{1}{r}\right)}$$
$$= \frac{784.44}{(0.885)\left(1 - \frac{1}{20}\right)}$$
$$\boxed{MEP = 933.02 \text{ kPa}}$$

Question:

Repeat Prob. 9–55, but replace the isentropic expansion process by polytropic expansion process with the polytropic exponent n= 1.35.

Step 1 of 11

Considering constant specific heats at room temperature.

Gas constant of air, $R = 0.287\ \text{kJ/kg}\cdot\text{K}$

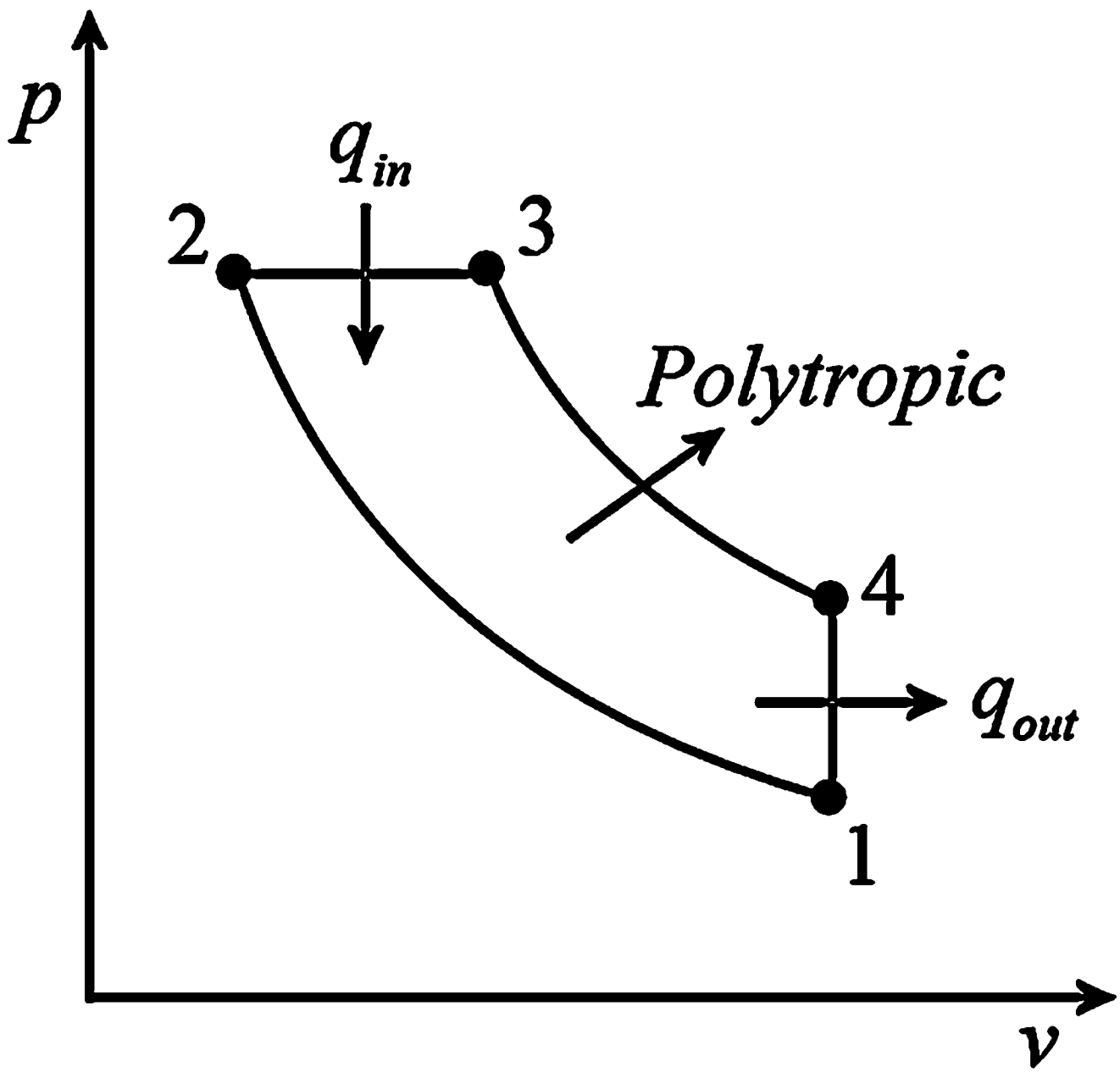
Specific heat of air at constant pressure, $c_p = 1.005\ \text{kJ/kg}\cdot\text{K}$

Specific heat of air at constant volume, $c_v = 0.718\ \text{kJ/kg}\cdot\text{K}$

Specific heats ratio, $k = 1.4$

Step 2 of 11

Draw the $p-v$ diagram for the diesel cycle.



Step 3 of 11

The Diesel cycle consists of four internally reversible processes:

1-2 isentropic compression

2-3 Constant-pressure heat addition

3-4 Polytropic expansion

4-1 Constant-volume heat rejection

Consider the process 1-2 is isentropic compression:

Using the following equation to determine the temperature at state 2.

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

Here, r is the compression ratio, T_1 is the temperature at state 1.

Substitute 20 for r and $293\ \text{K}$ for T_1.

$$T_2 = 293(20)^{1.4-1}$$
$$= 971.1\ \text{K}$$

Step 4 of 11

Process 2-3 is constant pressure heat addition:

Find the cut-off ratio as follows:

$$\frac{v_3}{v_2} = \frac{T_3}{T_2}$$

Here, v_3 is the volume after the heat addition at constant pressure, v_2 is the volume before the heat addition at constant pressure and T_3 is the temperature after the heat addition.

Substitute $2200\ \text{K}$ for T_3 and $971.1\ \text{K}$ for T_2.

$$\frac{v_3}{v_2} = \frac{2200}{971.1}$$
$$= 2.265$$
$$v_3 = 2.265 v_2$$

Step 5 of 11

Consider the process 3-4 is Polytropic expansion:

Calculate the temperature at state 4 using the following equation:

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{n-1}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1}$$

Since the process 4-1 is constant volume heat rejection, $v_4 = v_1$

$$\frac{T_3}{T_4} = \left(\frac{v_3}{v_4}\right)^{n-1}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1}$$

Since the process 4-1 is constant volume heat rejection, $v_4 = v_1$

Substitute $2.265v_2$ for v_3 and v_1 for v_4 .

$$T_4 = T_3\left(\frac{2.265v_2}{v_1}\right)^{n-1}$$

Substitute $\frac{1}{r}$ for $\frac{v_2}{v_1}$.

$$T_4 = T_3\left(\frac{2.265}{r}\right)^{n-1}$$

Substitute $2200\,\text{K}$ for T_3 , 20 for r and 1.35 for n .

$$T_4 = (2200)\left(\frac{2.265}{20}\right)^{1.35-1}$$
$$= 1026\,\text{K}$$

Step 6 of 11

Calculate the heat supplied per unit mass using the following equation.

$$q_{in} = h_3 - h_2$$
$$= c_p(T_3 - T_2)$$

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p , $2200\,\text{K}$ for T_3 and $971.1\,\text{K}$ for T_2 .

$$q_{in} = (1.005)(2200\text{-}971.1)$$
$$= 1235\,\text{kJ/kg}$$

Calculate the heat rejected per unit mass using the following equation.

$$q_{out,41} = u_4 - u_1$$
$$= c_v(T_4 - T_1)$$

Substitute $0.718\,\text{kJ/kg}\cdot\text{K}$ for c_v , $1026\,\text{K}$ for T_4 and $293\,\text{K}$ for T_1 .

$$q_{out,41} = 0.718(1026 - 293)$$
$$= 526.3\,\text{kJ/kg}$$

Note that q_{out} in this case does not represent the entire heat rejected, since some heat is also rejected during the polytropic process.

Step 7 of 11

Calculate the work produced in process 3-4, using the following equation:

$$w_{34} = \frac{R(T_4 - T_3)}{1-n}$$

Substitute $0.287\,\text{kJ/kg}\cdot\text{K}$ for R , $1026\,\text{K}$ for T_4 , $2200\,\text{K}$ for T_3 and 1.35 for n .

$$w_{34} = \frac{(0.287)(1026 - 2200)}{1-1.35}$$
$$= 963\,\text{kJ/kg}$$

Apply the energy balance in process 3-4:

$$E_{in} - E_{out} = \Delta E_{system}$$
$$q_{in,34} - w_{34} = u_4 - u_3$$
$$q_{in,34} = (u_4 - u_3) + w_{34}$$
$$q_{in,34} = c_v(T_4 - T_3) + w_{34}$$

Substitute $963\,\text{kJ/kg}$ for w_{34} , $1026\,\text{K}$ for T_4 , $2200\,\text{K}$ for T_3 and $0.718\,\text{kJ/kg}\cdot\text{K}$ for c_v .

$$q_{in,34} = 0.718(1026 - 2200) + 963$$
$$= 120.1\,\text{kJ/kg}$$

It indicates that $120.1\,\text{kJ/kg}$ of heat is transferred to the combustion gases during the polytropic expansion process. This is unrealistic since the gas at higher temperatures than the surroundings, and a hot gas loses heat during polytropic expansion.

Step 8 of 11

Now calculate the heat rejected during the process 3-4 using the properties of air tables:

Obtain the following values from the table "ideal gas properties of air".

At $2200\,\text{K}$: $u = 1872.4\,\text{kJ/kg}$

At $1026\,\text{K}$: $u = 781.3\,\text{kJ/kg}$

$$q_{out,34} = w_{34} + (u_4 - u_3)$$

Substitute $1872.4\,\text{kJ/kg}$ for u_3 , $781.3\,\text{kJ/kg}$ for u_4 and $963\,\text{kJ/kg}$ for w_{34} .

$$q_{out,34} = 963 + (781.3 - 1872.4)$$
$$= -128.1\,\text{kJ/kg}$$

Which is a heat loss expected.

Calculate the total heat rejected from the cycle as follows:

$$q_{out} = q_{out,41} + q_{out,34}$$

Substitute $128.1\,\text{kJ/kg}$ for $q_{out,34}$ and $526.3\,\text{kJ/kg}$ for $q_{out,41}$.

$$q_{out} = 128.1 + 526.3$$
$$= 654.4\,\text{kJ/kg}$$

Step 9 of 11

$$q_{out} = 128.1 + 526.3$$
$$= 654.4\ \text{kJ/kg}$$

Step 9 of 11

(a)

Calculate the thermal efficiency.

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Substitute $654.4\ \text{kJ/kg}$ for q_{out} and $1235\ \text{kJ/kg}$ for q_{in}.

$$\eta_{th} = 1 - \frac{654.4}{1235}$$
$$= 0.470$$
$$= 47.0\%$$

Therefore, the thermal efficiency is $\boxed{47.0\%}$.

(b)

Calculate the Mean effective pressure as follows:

$$MEP = \frac{w_{net}}{v_1 - v_2}$$
$$= \frac{w_{net}}{v_1\left(1 - \frac{1}{r}\right)} \quad \text{..... (1)}$$

Step 10 of 11

Find the net work done as follows:

$$w_{net} = q_{in} - q_{out}$$

Substitute $654.4\ \text{kJ/kg}$ for q_{out} and $1235\ \text{kJ/kg}$ for q_{in}.

$$w_{net} = 1235 - 654.4$$
$$= 580.6\ \text{kJ/kg}$$

Calculate the specific volume at state 1 using the following equation:

$$v_1 = \frac{RT_1}{P_1}$$

Substitute $0.287\ \text{kJ/kg}\cdot\text{K}$ for R, $293\ \text{K}$ for T_1 and $95\ \text{kPa}$ for P_1.

$$v_1 = \frac{0.287 \times 293}{95}$$
$$= 0.885\ \text{m}^3/\text{kg}$$

Step 11 of 11

Substitute $0.885\ \text{m}^3/\text{kg}$ for v_1, $580.6\ \text{kJ/kg}$ for w_{net} and 20 for r in equation (1).

$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{1}{r}\right)}$$
$$= \frac{580.6}{(0.885)\left(1 - \frac{1}{20}\right)}$$
$$= 690.57\ \text{kPa}$$

Therefore, the mean effective pressure is $\boxed{690.57\ \text{kPa}}$.

Question: 9.057P



Reconsider Prob. 9–56. Using EES (or other) software, study the effect of varying the compression ratio from 14 to 24. Plot the network output, mean effective pressure, and thermal efficiency as a function of the compression ratio. Plot the *T-s* and *P-V* diagrams for the cycle when the compression ratio is 20.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

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Question: 9.058P

A four-cylinder two-stroke 2,0-L diesel engine that operates on an ideal Diesel cycle has a compression ratio of 22 and a cutoff ratio of 1.8. Air is at 70°C and 97 kPa at the beginning of the compression process. Using the cold-air-standard assumptions, determine how much power the engine will deliver at 2300 rpm.

Step 1 of 10

Given data:

Compression ratio of the given cycle, $r = 22$

Cutoff ratio of the given cycle, $r_c = 1.8$

Initial pressure of the given cycle, $P_1 = 97\,\text{kPa}$

Initial temperature of the given cycle, $T_1 = 70°\text{C} = 343\text{K}$

Volume of the cylinder, $V = 2\text{L} = 2\times10^{-3}\,\text{m}^3$

Speed of the engine, $N = 2300\,\text{rpm}$

Step 2 of 10

Considering constant specific heats at room temperature

Gas constant $R = 0.287\,\text{kJ/kg.K}$

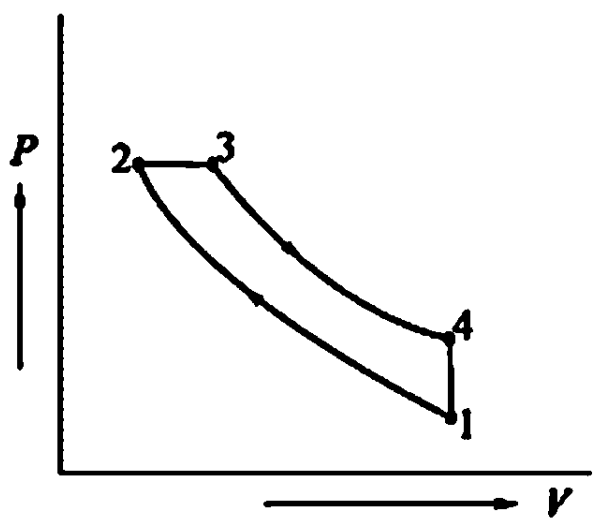
$c_p = 1.005\,\text{kJ/kg.K}$

$c_v = 0.718\,\text{kJ/kg.K}$

$k = 1.4$

Step 3 of 10

Representation of the ideal diesel cycle:



Step 4 of 10

An ideal Diesel cycle consists of four internally reversible processes:

1-2 isentropic compression

2-3 Constant-pressure heat addition

3-4 isentropic expansion

4-1 Constant-volume heat rejection

Step 5 of 10

From the isentropic compression process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 343\left(22^{1.4-1}\right)$$

$$= 1181\text{K}$$

Step 6 of 10

For the Process 2-3:

Constant pressure heat addition, we have

$$\frac{T_3}{T_2} = \frac{v_3}{v_2}$$

$$\frac{T_3}{T_2} = r_c$$

$$T_3 = 1181(1.8)$$

$$= 2125.8\text{K}$$

Step 7 of 10

For the Process 3-4:

From isentropic expansion, we have

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_2}\times\frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(r_c\times\frac{1}{r}\right)^{k-1}$$

$$T_4 = (2125.8)\left(\frac{1.8}{22}\right)^{1.4-1}$$

$$= 781\text{K}$$

Step 8 of 10

Heat supplied for the given diesel cycle,

$$q_{in} = h_3 - h_2$$

$$= c_p(T_3 - T_2)$$

$$q_{in} = (1.005)(2125.8 - 1181)$$

$$= 949.524\,\text{kJ/kg}$$

Step 9 of 10

The heat rejected per unit mass for the given cycle,

$$q_{out} = u_4 - u_1$$

$$= c_v(T_4 - T_1)$$

$$= 0.718(781 - 343)$$

$$= 314.484\text{K}$$

Net work done of the given cycle,

$$w_{net} = q_{in} - q_{out}$$

$$= 949.524 - 314.484$$

$$= 635.04\,\text{kJ/kg}$$

Step 10 of 10

Mass flow rate of air for the given cycle,

$$m = \frac{P_1V}{RT_1}$$

$$m = \frac{97\times2\times10^{-3}}{0.287\times343}$$

$$= 1.97\times10^{-3}\,\text{kg}$$

The power delivered by the cycle,

$$P = m\times N\times w_{net}$$

$$P = 1.97\times10^{-3}\times\frac{2300}{60}\times635.04$$

$$= 47.956\text{kW}$$

Therefore the power developed by the complete cycle, $\boxed{P = 47.956\text{kW}}$

Question: 9.059P

Repeat Prob. 9–58 using nitrogen as the working fluid.

Step 1 of 10

Given data:

Compression ratio of the given cycle, $r = 22$

Cutoff ratio of the given cycle, $r_c = 1.8$

Initial pressure of the given cycle, $P_1 = 97\ \text{kPa}$

Initial temperature of the given cycle, $T_1 = 70^\circ\text{C} = 343\text{K}$

Volume of the cylinder, $V = 2\text{L} = 2\times10^{-3}\text{m}^3$

Speed of the engine, $N = 2300\text{rpm}$

Step 2 of 10

Considering constant specific heats at room temperature for Nitrogen

Gas constant $R = 0.2968\ \text{kJ/kg.K}$

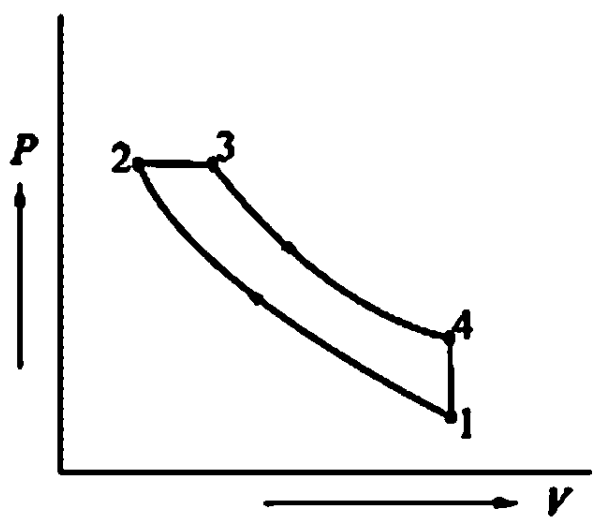
$c_p = 1.039\ \text{kJ/kg.K}$

$c_v = 0.743\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 10

Representation of the ideal diesel cycle:



Step 4 of 10

An ideal Diesel cycle consists of four internally reversible processes:

1-2 isentropic compression

2-3 Constant-pressure heat addition

3-4 isentropic expansion

4-1 Constant-volume heat rejection

Step 5 of 10

From the isentropic compression process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 343\left(22^{1.4-1}\right)$$

$$= 1181\text{K}$$

Step 6 of 10

For the Process 2-3:

Constant pressure heat addition, we have

$$\frac{T_3}{T_2} = \frac{v_3}{v_2}$$

$$\frac{T_3}{T_2} = r_c$$

$$T_3 = 1181(1.8)$$

$$= 2125.8\text{K}$$

Step 7 of 10

For the Process 3-4:

From isentropic expansion, we have

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_2}\times\frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_4}{T_3} = \left(r_c\times\frac{1}{r}\right)^{k-1}$$

$$T_4 = (2125.8)\left(\frac{1.8}{22}\right)^{1.4-1}$$

$$= 781\text{K}$$

Step 8 of 10

Heat supplied for the given diesel cycle,

$$q_{in} = h_3 - h_2$$

$$= c_p(T_3 - T_2)$$

$$q_{in} = (1.039)(2125.8 - 1181)$$

$$= 981.6472\text{kJ/kg}$$

Step 9 of 10

The heat rejected per unit mass for the given cycle,

$$q_{out} = u_4 - u_1$$

$$= c_v(T_4 - T_1)$$

$$= 0.743(781 - 343)$$

$$= 325.434\text{K}$$

Net work done of the given cycle,

$$w_{net} = q_{in} - q_{out}$$

$$= 981.6472 - 325.434$$

$$= 656.2132\text{kJ/kg}$$

Step 10 of 10

Mass flow rate of air for the given cycle,

$$m = \frac{P_1V}{RT_1}$$

$$= \frac{97\times2\times10^{-3}}{0.2968\times343}$$

$$= 1.9\times10^{-3}\text{kg}$$

The power delivered by the cycle,

$$P = m\times N\times w_{net}$$

$$= 1.9\times10^{-3}\times\frac{2300}{60}\times656.2134$$

$$= 47.92\text{kW}$$

Therefore the power developed by the complete cycle, $\boxed{P = 47.92\text{kW}}$

Question: 9.060P pg 1

An air-standard dual cycle has a compression ratio of 18 and a cutoff ratio of 1.1. The pressure ratio during constant-volume heat addition process is 1.1. At the beginning of the compression, $P1$= 90 kPa, $T1$= 18°C, and $V1$= 0.003 m3. How much power will this cycle produce when it is executed 4000 times per minute? Use constant specific heats at room temperature.

Step 1 of 14

Given

$$\text{Compression ratio } r = \frac{v_1}{v_2} = 18$$

$$\text{Cutoff ratios } r_c = \frac{v_4}{v_3} = 1.1$$

$$\text{Pressure ratio } r_p = \frac{p_3}{p_2} = 1.1$$

$$\text{Initial pressure } P_1 = 90\ \text{kPa}$$

$$\text{Initial temperature } T_1 = 18°\text{C}$$

$$= 18 + 273$$

$$T_1 = 291\text{K}$$

$$\text{Initial volume } V_1 = 0.003\ \text{m}^3$$

$$\text{Revolutions per minute } N = 4000\text{RPM}$$

Step 2 of 14

Considering constant specific heats at room temperature for air

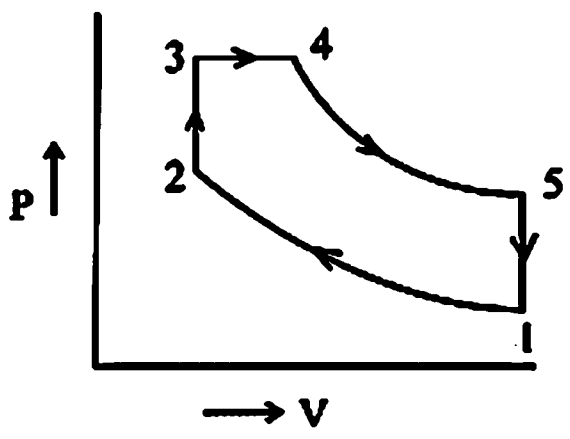
Gas constant $R = 0.287$ kJ/kg.K

$c_p = 1.005$ kJ/kg.K

$c_v = 0.718$ kJ/kg.K

$k = 1.4$

Step 3 of 14



Step 4 of 14

An ideal dual cycle consists of five internally reversible processes:

1-2 Isentropic compression

2-3 Constant-volume heat addition

3-4 Constant-pressure heat addition

4-5 Isentropic expansion

5-1 Constant-volume heat rejection

Step 5 of 14

1-2 process is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 291\left(18^{1.4-1}\right)$$

$$T_2 = 924.7\text{K}$$

Step 6 of 14

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^k$$

$$\frac{P_2}{P_1} = r^k$$

$$P_2 = 90\left(18^{1.4}\right)$$

$$P_2 = 5147.8\ \text{kPa}$$

Step 7 of 14

Given

$$\frac{P_3}{P_2} = r_p$$

$$P_3 = 1.1(5147.8)$$

$$P_4 = 5662.58\ \text{kPa}$$

Step 8 of 14

Process 2-3 is constant volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$\frac{T_3}{T_2} = r_p$$

$$T_3 = 924.7(1.1)$$

$$T_3 = 1017.17\text{K}$$

Step 9 of 14

Process 3-4 is constant pressure heat addition

$$\frac{T_4}{T_3} = \frac{v_4}{v_3}$$

Step 9 of 14

Process 3-4 is constant pressure heat addition

$$\frac{T_4}{T_3}=\frac{v_4}{v_3}$$

$$\frac{T_4}{T_3}=r_c$$

$$T_4=1017.17(1.1)$$

$$T_4=1118.88\text{K}$$

Step 10 of 14

Process 4-5 is isentropic expansion

$$\frac{T_5}{T_4}=\left(\frac{v_4}{v_5}\right)^{k-1}$$

$$\frac{T_5}{T_4}=\left(\frac{v_4}{v_1}\right)^{k-1}$$

$$\frac{T_5}{T_4}=\left(\frac{v_4}{v_3}\times\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_5}{T_4}=\left(\frac{v_4}{v_3}\times\frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_5}{T_4}=\left(\frac{r_c}{r}\right)^{k-1}$$

$$T_5=1118.88\left(\frac{1.1}{18}\right)^{1.4-1}$$

$$T_5=365.78\text{K}$$

Step 11 of 14

Heat addition is done partly in constant volume and partly in constant pressure process

$$q_{in}=c_v(T_3-T_2)+c_p(T_4-T_3)$$

$$=(0.718)(1017.17-924.7)+(1.005)(1118.88-1017.17)$$

$$=66.39+102.218$$

$$q_{in}=168.608\text{ kJ/kg}$$

Step 12 of 14

Heat rejection in the process 5-1 is

$$q_{out}=c_v(T_5-T_1)\text{ kJ/kg}$$

$$=(0.718)(365.78-291)$$

$$q_{out}=53.69\text{ kJ/kg}$$

Step 13 of 14

Net work is expressed as

$$w_{net}=q_{in}-q_{out}$$

$$=168.608-53.69$$

$$w_{net}=114.918\text{ kJ/kg}$$

Mass of air

$$m=\frac{P_1V}{RT_1}$$

$$=\frac{90\times0.003}{0.287\times291}$$

$$m=3.232\times10^{-3}\text{kg}$$

Step 14 of 14

The power delivered by the cycle

$$P=m\times N\times w_{net}\text{ kW}$$

$$P=3.232\times10^{-3}\times\frac{4000}{60}\times114.918$$

$$\boxed{P=24.76\text{ kW}}$$

Question:

Repeat Prob. 9–60 if the isentropic compression efficiency is 85 percent and the isentropic expansion efficiency is 90 percent.

Step 1 of 15

Given

Compression ratio $r = \frac{v_1}{v_2} = 18$

Cutoff ratios $r_c = \frac{v_4}{v_3} = 1.1$

Pressure ratio $r_p = \frac{p_3}{p_2} = 1.1$

Initial pressure $P_1 = 90\ \text{kPa}$

Initial temperature $T_1 = 18°\text{C}$

$= 18 + 273$

$T_1 = 291\text{K}$

Initial volume $V_1 = 0.003\ \text{m}^3$

Revolutions per minute $N = 4000\text{RPM}$

Isentropic compressor efficiency $\eta_c = 85\%$

Isentropic expansion efficiency $\eta_e = 90\%$

Step 2 of 15

Considering constant specific heats at room temperature for air

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 15

Dual cycle consists of five internally reversible processes:

1-2 Compression process

2-3 Constant-volume heat addition

3-4 Constant-pressure heat addition

4-5 Expansion process

5-1 Constant-volume heat rejection

Step 4 of 15

1-2' process is isentropic compression

$$\frac{T_2'}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2'}{T_1} = r^{k-1}$$

$$T_2' = 291\left(18^{1.4-1}\right)$$

$$T_2' = 924.7\text{K}$$

Step 5 of 15

Since the isentropic efficiency of the compression is given by

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$0.85 = \frac{924.7 - 291}{T_2 - 291}$$

$$T_2 = 1036.5\text{K}$$

Step 6 of 15

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^k$$

$$\frac{P_2}{P_1} = r^k$$

$$P_2 = 90\left(18^{1.4}\right)$$

$$P_2 = 5147.8\ \text{kPa}$$

Step 7 of 15

Given

$$\frac{P_3}{P_2} = r_p$$

$$P_3 = 1.1(5147.8)$$

$$P_4 = 5662.58\ \text{kPa}$$

Step 8 of 15

Process 2-3 is constant volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$\frac{T_3}{T_2} = r_p$$

$$T_3 = 1036.5(1.1)$$

$$T_3 = 1140.15\text{K}$$

Step 9 of 15

Process 3-4 is constant pressure heat addition

$$\frac{T_4}{T_3} = \frac{v_4}{v_3}$$

Step 9 of 15

Process 3-4 is constant pressure heat addition

$$\frac{T_4}{T_3}=\frac{v_4}{v_3}$$

$$\frac{T_4}{T_3}=r_c$$

$$T_4=1140.15(1.1)$$

$$T_4=1254.165\text{K}$$

Step 10 of 15

Process 4-5' is isentropic expansion

$$\frac{T_5'}{T_4}=\left(\frac{v_4}{v_5}\right)^{k-1}$$

$$\frac{T_5'}{T_4}=\left(\frac{v_4}{v_1}\right)^{k-1}$$

$$\frac{T_5'}{T_4}=\left(\frac{v_4}{v_3}\times\frac{v_3}{v_1}\right)^{k-1}$$

$$\frac{T_5'}{T_4}=\left(\frac{v_4}{v_3}\times\frac{v_2}{v_1}\right)^{k-1}$$

$$\frac{T_5'}{T_4}=\left(\frac{r_c}{r}\right)^{k-1}$$

$$T_5'=1254.165\left(\frac{1.1}{18}\right)^{1.4-1}$$

$$T_5'=410.0178\text{K}$$

Step 11 of 15

Since the isentropic efficiency of the expansion is given

$$\eta_{ex}=\frac{T_4-T_5}{T_4-T_5'}$$

$$0.90=\frac{1254.165-T_5}{1254.165-410.0178}$$

$$T_5=494.43\text{K}$$

Step 12 of 15

Heat addition is done partly by constant volume and partly by constant pressure process

$$q_{in}=c_v(T_3-T_2)+c_p(T_4-T_3)$$

$$=(0.718)(1140.15-1036.5)+(1.005)(1254.165-1140.15)$$

$$=74.4207+114.58$$

$$q_{in}=189\text{ kJ/kg}$$

Step 13 of 15

Heat rejection takes place in the process 5-1 at constant volume

$$q_{out}=c_v(T_5-T_1)\text{ kJ/kg}$$

$$=(0.718)(494.43-291)$$

$$q_{out}=146.062\text{ kJ/kg}$$

Step 14 of 15

Net work is expressed as

$$w_{net}=q_{in}-q_{out}$$

$$=188.9-146.062$$

$$w_{net}=42.838\text{ kJ/kg}$$

Mass of air

$$m=\frac{P_1V}{RT_1}$$

$$=\frac{90\times0.003}{0.287\times291}$$

$$m=3.232\times10^{-3}\text{kg}$$

Step 15 of 15

The power delivered by the cycle

$$P=m\times N\times w_{net}\text{ kW}$$

$$P=3.232\times10^{-3}\times\frac{4000}{60}\times42.838$$

$$\boxed{P=9.23016\text{ kW}}$$

Question: 9.062P

An ideal dual cycle has a compression ratio of 15 and a cutoff ratio of 1.4. The pressure ratio during constant-volume heat addition process is 1.1. The state of the air at the beginning of the compression is $P1$, = 14.2 psia and $T1$, = 75°F. Calculate the cycle's net specific work, specific heat addition, and thermal efficiency. Use constant specific heats at room temperature.

Step 1 of 11

Given

Compression ratio $r = \frac{V_1}{V_2} = 15$

Cutoff ratio $r_c = \frac{V_4}{V_3} = 1.4$

Pressure ratio $r_p = \frac{p_3}{p_2} = 1.1$

Initial pressure $P_1 = 14.2$ psia

Initial temperature $T_1 = 75°F$

$= 75+460$

$T_1 = 535R$

Step 2 of 11

Considering constant specific heats at room temperature

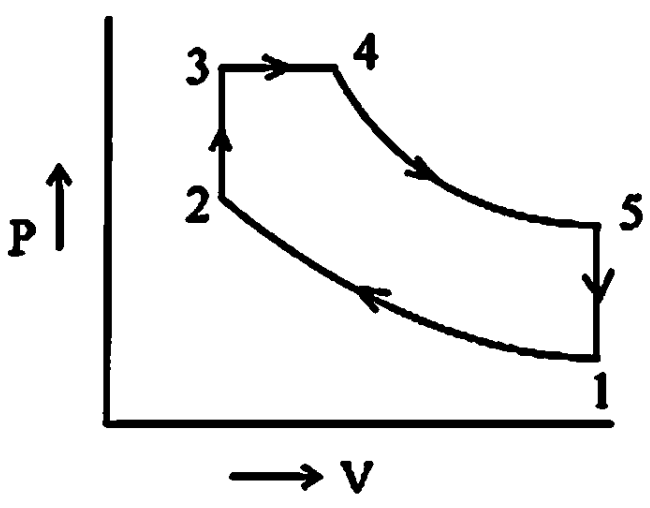
Gas constant $R = 0.3704$ psia.ft^3/lbm.R

Specific heat at constant pressure $c_p = 0.24$ Btu/lbm.R

Specific heat at constant volume $c_v = 0.171$ Btu/lbm.R

Ratio of specific heats $k = 1.4$

Step 3 of 11



Step 4 of 11

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

$$\frac{T_2}{535} = 15^{1.4-1}$$

$$T_2 = 1580.48^0 R$$

Step 5 of 11

Process 2-3 is constant-volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$\frac{T_3}{1580.48} = 1.1$$

$$T_3 = 1738.528^0 R$$

Step 6 of 11

Process 3-4 is constant pressure heat addition

$$\frac{T_4}{T_3} = \frac{V_4}{V_3}$$

$$\frac{T_4}{1738.528} = 1.4$$

$$T_4 = 2433.93^0 R$$

Step 7 of 11

Process 4-5 is isentropic expansion

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{k-1}$$

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_1}\right)^{k-1}$$

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_2} \times \frac{V_2}{V_1}\right)^{k-1}$$

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_3} \times \frac{V_2}{V_1}\right)^{k-1}$$

$$\frac{T_5}{T_4} = \left(r_c \times \frac{1}{r}\right)^{k-1}$$

$$\frac{T_5}{2433.93} = \left(1.4 \times \frac{1}{15}\right)^{1.4-1}$$

$$T_5 = 942.589^0 R$$

Step 8 of 11

Heat addition is done partly by constant volume and constant pressure process

$$q_{in} = c_v(T_3 - T_2) + c_p(T_4 - T_3)$$

$$= (0.171)(1738.528 - 1580.48) + (0.24)(2433.93 - 1738.528)$$

$$= 27.0262 + 166.896$$

$$\boxed{q_{in} = 193.922 \text{ Btu/lbm}}$$

Step 9 of 11

Heat rejection in the process 5-1 takes place at constant volume

$$q_{out} = c_v(T_5 - T_1)$$

$$= (0.171)(942.589 - 535)$$

$$q_{out} = 69.697 \text{ Btu/lbm}$$

Step 10 of 11

Net specific work

$$w_{net} = q_{in} - q_{out}$$

$$= 193.922 - 69.697$$

$$\boxed{w_{net} = 124.225 \text{ Btu/lbm}}$$

Step 11 of 11

The thermal efficiency is defined as

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{124.225}{193.922}$$

$$= 0.6405$$

$$\boxed{\eta_{th} = 64.05\%}$$

Question:

A six-cylinder, four-stroke, 3.2-L compression-ignition engine operates on the ideal diesel cycle with a compression ratio of 19. The air is at 95 kPa and 67°C at the beginning of the compression process and the engine speed is 1750 rpm. The engine uses light diesel fuel with a heating value of 42,500 kJ/kg, an air-fuel ratio of 28, and a combustion efficiency of 98 percent. Using constant specific heats at 850 K, determine (*a*) the maximum temperature in the cycle and the cutoff ratio (*b*) the net work output per cycle and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the net power output, and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the network produced.

Step 1 of 14

Given data:

Compression ratio, $r = \frac{V_1}{V_2} = 19$

Initial pressure, $P_1 = 95\ \text{kPa}$

Initial temperature, $T_1 = 67°\text{C} = 340\ \text{K}$

Swept volume, $V_s = 3.2\ \text{L} = 0.0032\ \text{m}^3$

Speed of the engine, $N = 1750\ \text{rpm}$

Heating value of diesel fuel, $HV = 42{,}500\ \text{kJ/kg}$

Air fuel ratio, $\frac{m_a}{m_f} = 28$

Combustion efficiency, $\eta_{comb} = 98\%$

Number of cylinders, $n = 6$

Step 2 of 14

Taking constant specific heats at temperature 850K from the table of ideal gas specific heats of various common gases, we get

Gas constant, $R = 0.287\ \text{kJ/kg}\cdot\text{K}$

Specific heat at constant pressure, $c_p = 1.11\ \text{kJ/kg}\cdot\text{K}$

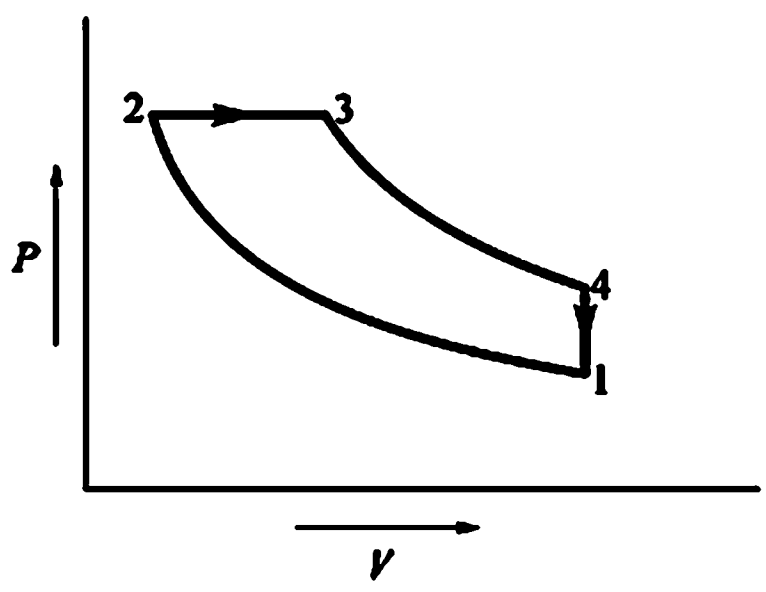
Specific heat at constant volume, $c_v = 0.823\ \text{kJ/kg}\cdot\text{K}$

The ratio of specific heats,

$$k = \frac{c_p}{c_v} = \frac{1.11}{0.823} = 1.349$$

Step 3 of 14

The *P-V* diagram of the process:



An ideal Diesel cycle consists of four internally reversible processes:

1-2 – An Isentropic compression

2-3 – A Constant-pressure heat addition

3-4 – An Isentropic expansion

4-1 – A Constant-volume heat rejection

Step 4 of 14

As the process 1-2 is isentropic compression, we can write

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = T_1 r^{k-1} = (340\ \text{K})(19)^{1.349-1} = 950\ \text{K}$$

Step 5 of 14

Let $V_2 = V_c$ be the volume at the end of the compression.

Thus, total volume of the cylinder at the beginning of the compression is

$V_1 = V_c + V_s$

We have,

$$r = \frac{V_1}{V_2} = \frac{V_c + V_s}{V_c}$$

$$19 = \frac{V_c + 0.0032}{V_c}$$

$$19V_c = V_c + 0.0032$$

$$V_c = 0.00018\ \text{m}^3$$

Thus,

$$V_1 = V_c + V_s = (0.00018\ \text{m}^3) + (0.0032\ \text{m}^3) = 0.00338\ \text{m}^3$$

Step 6 of 14

The total mass in the cylinder is

$$\frac{P_1 V_1}{}$$

Step 6 of 14

The total mass in the cylinder is

$$m = \frac{P_1 V_1}{R T_1}$$

$$= \frac{(95\ \text{kPa})(0.00338\ \text{m}^3)}{(0.287)(340\ \text{K})}$$

$$= 0.00329\ \text{kg}$$

The total mass is the sum of the air mass and fuel mass in the cylinder.

$$m = m_a + m_f$$

Air-fuel ratio is given by

$$\frac{m_a}{m_f} = 28$$

$$\frac{m - m_f}{m_f} = 28$$

$$0.00329 - m_f = 28 m_f$$

$$m_f = \frac{0.00329}{29}$$

$$= 0.00011\ \text{kg}$$

Step 7 of 14

(a)

The heat supplied is expressed in terms of combustion of fuel and heat addition takes place at constant pressure process

$$Q_{in} = mc_p(T_3 - T_2)$$

$$m_f \times HV \times \eta_{cc} = mc_p(T_3 - T_2)$$

$$(0.00011\ \text{kg})(42{,}500\ \text{kJ/kg})(0.98) = (0.00329\ \text{kg})(1.11\ \text{kJ/kg·K})(T_3 - 949.3\ \text{K})$$

$$T_3 - 949.3 = 1294.7$$

$$T_3 = 2244\ \text{K}$$

Hence, the maximum temperature is $\boxed{T_3 = 2244\ \text{K}}$

Step 8 of 14

The cutoff ratio is expressed as

From process 2-3

$$r_c = \frac{V_3}{V_2} = \frac{T_3}{T_2}$$

$$= \frac{2244}{950}$$

$$= 2.36$$

Hence, the cutoff ratio is $\boxed{r_c = 2.36}$

Step 9 of 14

(b)

Process 4-1 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

Since $\frac{v_3}{v_4} = \frac{r_c}{r}$,

$$\frac{T_4}{T_3} = \left(\frac{r_c}{r}\right)^{k-1}$$

$$\frac{T_4}{2244} = \left(\frac{2.36}{19}\right)^{1.349-1}$$

$$T_4 = 1084\ \text{K}$$

Step 10 of 14

The heat rejection during the constant volume process is

$$Q_{out} = mc_v(T_4 - T_1)$$

$$= (0.00329\ \text{kg})(0.823)(1220.5 - 340)$$

$$= 2.01\ \text{kJ}$$

Heat input during the isobaric process,

$$Q_{in} = mc_p(T_3 - T_2)$$

$$= (0.00329\ \text{kg})(1.11)(2244 - 950)$$

$$= 4.73\ \text{kJ}$$

Step 11 of 14

The net work is expressed as

$$W_{net} = Q_{in} - Q_{out}$$

$$= 4.73 - 2.01$$

$$= 2.72\ \text{kJ}$$

Hence, the net work output is $\boxed{W_{net} = 2.72\ \text{kJ}}$

The thermal efficiency is expressed as

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

$$= \frac{2.72}{4.73}$$

$$= 0.5755 \text{ or } 57.55\%$$

$\boxed{\eta_{th} = 57.55\%}$

Step 12 of 14

(c)

Step 12 of 14

(c)

Mean effective pressure

$$MEP = \frac{W_{net}}{V_1 - V_2}$$

$$= \frac{W_{net}}{V_1\left(1 - \frac{1}{r}\right)}$$

$$= \frac{2.72}{(0.00338)\left(1 - \frac{1}{19}\right)}$$

$$= 849 \text{ kPa}$$

Hence, the mean effective pressure is $\boxed{MEP = 849 \text{ kPa}}$

Step 13 of 14

(d)

The net power output

$$P = \frac{N}{2} \times W$$

$$= \frac{1750}{2 \times 60}(2.72)$$

$$= 39.7 \text{ kW}$$

Hence, the net power output is $\boxed{P = 45.2 \text{ kW}}$

Step 14 of 14

(e) The specific fuel consumption is given by

$$SFC = \frac{m_f}{W_{net}}$$

$$= \frac{0.00011}{2.72} \times 10^3 \times 3600$$

$$= 150 \text{ g/kWh}$$

Hence, the specific fuel consumption is $\boxed{SFC = 150 \text{ g/kWh}}$

Question: 9.064P

What four processes make up the simple ideal Brayton cycle?

Step 1 of 1

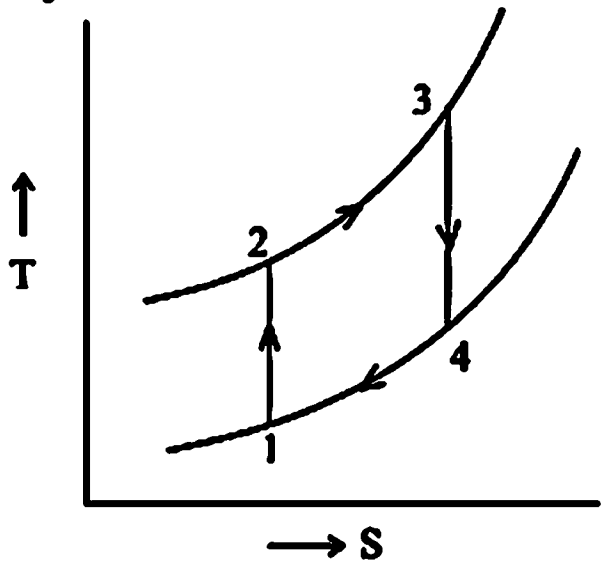
An ideal Brayton cycle consists of four processes:

1-2 isentropic compression

2-3 Constant pressure heat addition

3-4 isentropic expansion

4-1 Constant pressure heat rejection



Question: 9.065P

For fixed maximum and minimum temperatures, what is the effect of the pressure ratio on (*a*) the thermal efficiency and (*b*) the network output of a simple ideal Brayton cycle?

Step 1 of 1

The efficiency of a Brayton cycle is expressed as

$$\eta = 1 - \frac{1}{r_p^{\frac{k-1}{k}}}$$

(a) As the pressure ratio increases the thermal efficiency also increases.

(b) The work output first increases then decreases because at very high pressure ratios the increased turbine inlet temperature will destroy the blades for turbine will leads to loss in work output.

Question: 9.066P

What is the back work ratio? What arc typical back work ratio values for gas-turbine engines?

Step 1 of 1

The fraction of the turbine work used to drive the compressor is called the back work ratio.

$$r_{bw} = \frac{\dot{W}_{comp,in}}{\dot{W}_{turb,out}}$$

In gas turbine power plants the back work ratio is high compared to the steam power plants. In gas turbine engines the back work ratio lies between 0.4 to 0.6

Question: 9.067P

Why are the back work ratios relatively high in gas-turbine engines?

Step 1 of 1

The fraction of the turbine work used to drive the compressor is called the back work ratio. The back work ratio is usually very high in gas turbine power plants; usually more than one half of the turbine work output is used to drive the compressor.

In steam power plants the back work ratio is only a few percent. Liquid is compressed in steam power plants where as in gas turbine, gas is compressed and the steady-flow work is proportional to the specific volume of the working fluid.

Question: 9.068P

How do the inefficiencies of the turbine and the compressor affect (*a*) the back work ratio and (*b*) the thermal efficiency of a gas-turbine engine?

Step 1 of 1

Usually more than one half of the turbine work output is used to drive the compressor. The back work of the gas turbine still increases when the isentropic efficiencies of the compressor and the turbine are low and the efficiency of the gas turbine decreases. So the inefficiencies of the turbine and the compressor

(a) The back work ratio of the gas turbine engine increases

(b) The thermal efficiency decreases.

Question: 9.069P

A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 10. The air enters the compressor at 520 R and the turbine at 2000 R. Accounting for the variation of specific heats with temperature, determine (*a*) the air temperature at the compressor exit, (*b*) the back work ratio, and (*c*) the thermal efficiency.

Step 1 of 12

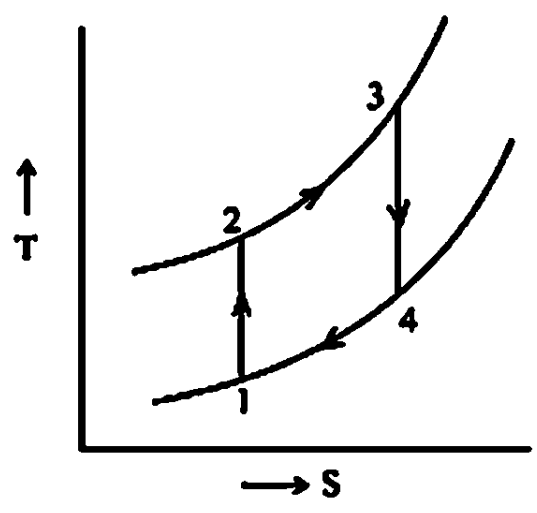
Given

Pressure ratio $r_p = 10$

Compressor inlet temperature $T_1 = 520\text{R}$

Turbine inlet temperature to $T_3 = 2000\text{R}$

Step 2 of 12



Step 3 of 12

Since we are considering variable specific heats the specific heat at constant pressure and specific heat at constant volume will be different at different temperatures.

At $T_1 = 520\text{R}$ from ideal gas properties of air tables

$h_1 = 124.27\ \text{Btu/lbm}$

$P_{r1} = 1.2147$

At $T_3 = 2000\text{R}$ from ideal gas properties of air tables

$h_3 = 504.71\ \text{Btu/lbm}$

$P_{r3} = 174.0$

Step 4 of 12

(a)

Process 1-2 is isentropic compression

$$\frac{P_2}{P_1} = \frac{P_{r2}}{P_{r1}}$$

$$P_{r2} = P_{r1}\frac{P_2}{P_1}$$

$$P_{r2} = 1.2147 \times 10$$

$$P_{r2} = 12.147$$

At $P_{r2} = 12.147$ from ideal gas properties of air

$$T_2 = 980 + (1000 - 980)\left(\frac{12.147 - 11.43}{12.30 - 11.43}\right)$$

$$\boxed{T_2 = 996.48\text{R}}$$

$$h_2 = 236.02 + (240.98 - 236.02)\left(\frac{12.147 - 11.43}{12.30 - 11.43}\right)$$

$$h_2 = 240.107\ \text{Btu/lbm}$$

Step 5 of 12

(b)

Process 3-4 is isentropic expansion

We know that

$$\frac{P_4}{P_3} = \frac{P_{r4}}{P_{r3}}$$

$$\frac{P_1}{P_2} = \frac{P_{r4}}{P_{r3}}$$

$$P_{r4} = P_{r3} \times \frac{P_1}{P_2}$$

$$P_{r4} = 174 \times \frac{1}{10}$$

$$P_{r4} = 17.4$$

Step 6 of 12

At $P_{r4} = 17.4$ from ideal gas properties of air

$$T_4 = 1080 + (1120 - 1080)\left(\frac{17.4 - 16.28}{18.60 - 16.28}\right)$$

$$T_4 = 1099.31\text{R}$$

$$h_4 = 260.97 + (271.03 - 260.97)\left(\frac{17.4 - 16.28}{18.60 - 16.28}\right)$$

$$h_4 = 265.826\ \text{Btu/lbm}$$

Step 7 of 12

The work required by compressor is expressed as

$$w_c = h_2 - h_1$$

$$w_c = 240.107 - 124.27$$

$$w_c = 115.837\ \text{Btu/lbm}$$

Step 8 of 12

The work done by turbine is expressed as

$$w_t = h_3 - h_4$$

$$w_t = 504.71 - 265.826$$

$$w_t = 238.884\ \text{Btu/lbm}$$

Step 9 of 12

The back work ratio

$$r_{bw} = \frac{w_c}{w_t}$$

$$= \frac{115.837}{238.884}$$

$$\boxed{r_{bw} = 0.484}$$

Step 10 of 12

(c) The thermal efficiency

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

Heat supplied

$$q_{in} = h_3 - h_2$$

$$= 504.71 - 240.107$$

$$q_{in} = 264.603\ \text{Btu/lbm}$$

Step 11 of 12

Net work is expressed as

$$w_{net} = w_t - w_c$$

$$= 238.884 - 115.837$$

$$w_{net} = 123.047\ \text{Btu/lbm}$$

Step 12 of 12

Therefore the thermal efficiency

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{123.047}{264.603}$$

$$= 0.465$$

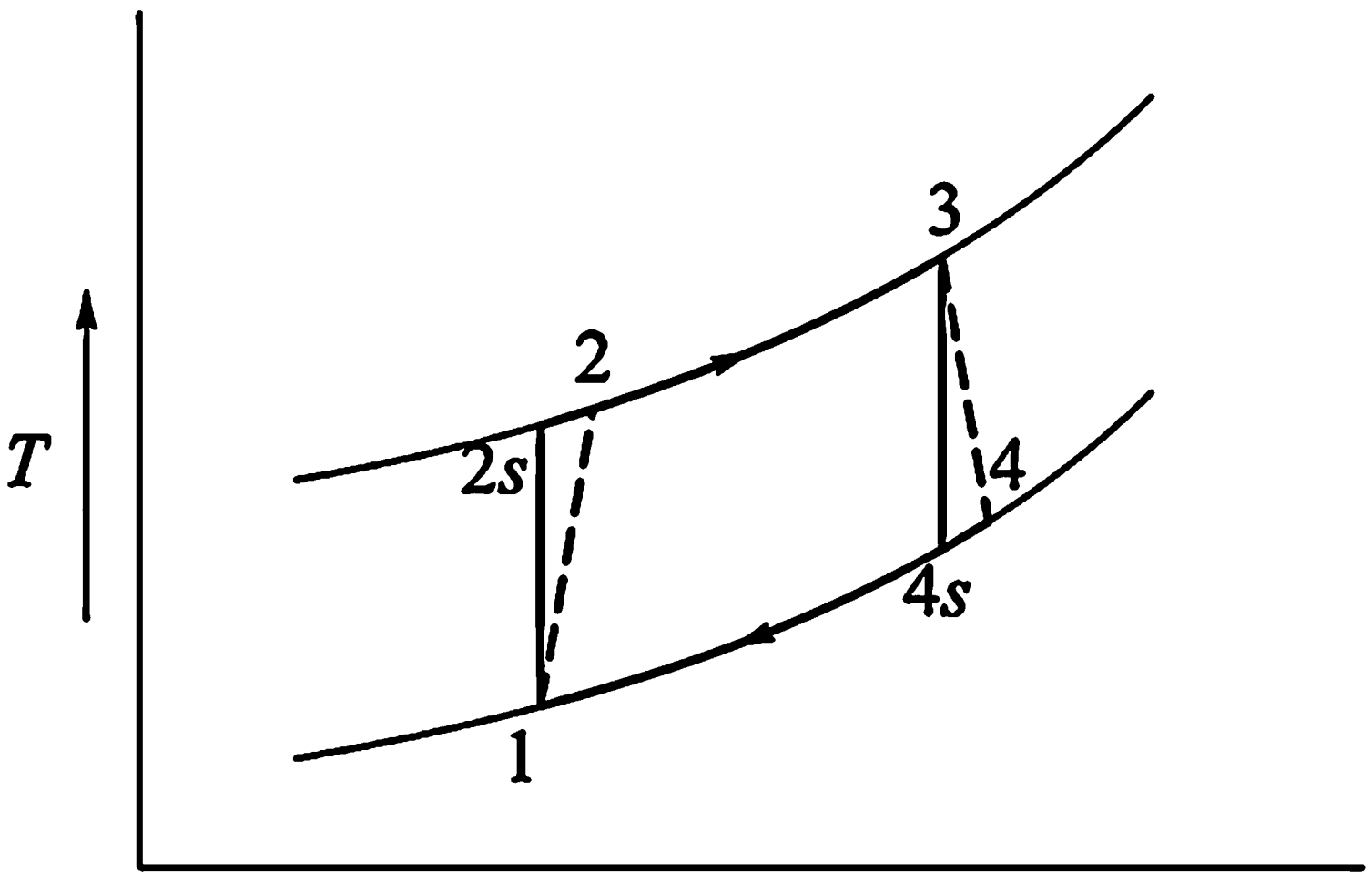
$$\boxed{\eta_{th} = 46.5\%}$$

Question:

A simple Brayton cycle using air as the working fluid has a pressure ratio of 10. The minimum and maximum temperatures in the cycle are 295 and 1240 K. Assuming an isentropic efficiency of 83 percent for the compressor and 87 percent for the turbine, determine (*a*) the air temperature at the turbine exit, (*b*) the net work output, and (*c*) the thermal efficiency.

Step 1 of 8

The *T-s* diagram of the process:



Step 2 of 8

Find the pressure using isentropic compression from process 1-2*s*.

$$\frac{P_2}{P_1} = \frac{P_{r2s}}{P_{r1}}$$

$$P_{r2s} = \left(\frac{P_2}{P_1}\right) P_{r1}$$

$$= r_p P_{r1}$$

Here, pressure ration is r_p and pressure at 295 K is P_{r1}

Form ideal gas properties of air table at $295\ \text{K}$, saturation pressure is $1.3068\ \text{kPa}$.

Substitute, 10 for r_p and $1.3068\ \text{kPa}$ for P_{r1} .

$$P_{r2s} = r_p P_{r1}$$

$$= 10 \times (1.3068)$$

$$= 13.068\ \text{kPa}$$

Step 3 of 8

Form ideal gas properties of air table at pressure is $13.068\ \text{kPa}$ interpolate for enthalpy.

$$h_{2s} = 565.17 + (575.59 - 565.17)\left(\frac{13.068 - 12.66}{13.5 - 12.66}\right)$$

$$= 570.23\ \text{kJ/kg}$$

Find the enthalpy at stage 2 using isentropic efficiency of the compressor is given by

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.83 = \frac{570.23 - 295.17}{h_2 - 295.17}$$

$$h_2 = 626.57\ \text{kJ/kg}$$

Step 4 of 8

Form ideal gas properties of air table at $1240\ \text{K}$. pressure is $272.3\ \text{kPa}$.

Process 3-4*s* is an isentropic expansion

$$\frac{P_4}{P_3} = \frac{P_{r4s}}{P_{r3}}$$

$$\frac{P_1}{P_2} = \frac{P_{r4s}}{P_{r3}}$$

$$P_{r4s} = \left(\frac{P_1}{P_2}\right) P_{r3}$$

$$P_{r4s} = \frac{1}{r_p} P_{r3}$$

Substitute, 10 for r_p and $272.3\ \text{kPa}$ for P_{r2} .

$$P_{r4s} = \frac{1}{r_p} P_{r3}$$

$$= \left(\frac{1}{10}\right)(272.3)$$

$$= 27.23$$

Interpolate for enthalpy at $27.23\ \text{kPa}$ from ideal gas properties of air.

$$h_{4s} = 691.82 + (702.52 - 691.82)\left(\frac{27.23 - 25.85}{27.29 - 25.85}\right)$$

$$= 702.07\ \text{kJ/kg}$$

Find the actual enthalpy at stage 4 isentropic efficiency of the turbine is given by

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$= 702.07 \text{ kJ/kg}$

Find the actual enthalpy at stage 4 isentropic efficiency of the turbine is given by

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$0.87 = \frac{1324.93 - h_4}{1324.93 - 702.07}$$

$$h_4 = 783.05 \text{ kJ/kg}$$

Step 5 of 8

(a)

Find the temperature at turbine exit by interpolating for enthalpy at stage 4.

$$T_4 = 760 + (780 - 760)\left(\frac{783.05 - 778.18}{800.03 - 778.18}\right)$$

$$= 764.45 \text{ K}$$

Therefore, the temperature at the exit of the turbine is $\boxed{T_4 = 764.45 \text{ K}}$

Step 6 of 8

(b)

Find work input to compressor is

$$w_c = h_2 - h_1$$

Form ideal gas properties of air table at 295 K, enthalpy is 295.17 kJ/kg.

Substitute, 626.57 kJ/kg for h_2 and 295.17 kJ/kg for h_1.

$$w_c = h_2 - h_1$$
$$= 626.57 - 295.17$$
$$= 331.4 \text{ kJ/kg}$$

Step 7 of 8

Find work input to compressor is

$$w_t = h_3 - h_4$$

Form ideal gas properties of air table at 1240 K, enthalpy is 1324.93 kJ/kg.

Substitute, 1324.93 kJ/kg for h_3 and 783.05 kJ/kg for h_1.

$$w_t = h_3 - h_4$$
$$= 1324.93 - 783.05$$
$$= 541.88 \text{ kJ/kg}$$

Find Net-work can be expressed as the difference of the turbine output and compressor input.

$$w_{net} = w_t - w_c$$
$$= 541.88 - 331.4$$
$$= 210.48$$

Hence, the net-work output of the cycle is $\boxed{w_{net} = 210.48 \text{ kJ/kg}}$

Step 8 of 8

(c)

The thermal efficiency is given by

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$
$$= \frac{w_{net}}{h_3 - h_2}$$

Substitute, 210.48 kJ/kg for w_{net}, 1324.93 kJ/kg for h_3 and 626.57 kJ/kg for h_2

$$\eta_{th} = \frac{w_{net}}{h_3 - h_2}$$
$$= \frac{210.48}{1324.93 - 626.57}$$
$$= 0.3014$$
$$= 30.14\%$$

Therefore, the thermal efficiency is $\boxed{\eta_{th} = 30.14\%}$

Question: 9.071P

Reconsider Prob. 9–70. Using EES (or other) software, allow the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor to vary. Assume the compressor inlet pressure is kPa. Develop a general solution for the problem by taking advantage of the diagram window method for supplying data to EES software.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 9.072P

Repeat Prob. 9–70 using constant specific heats at room temperature.

Step 1 of 8

Pressure ratio, $r = \frac{P_2}{P_1} = 10$

Minimum temperature of the cycle, $T_1 = 295\text{K}$

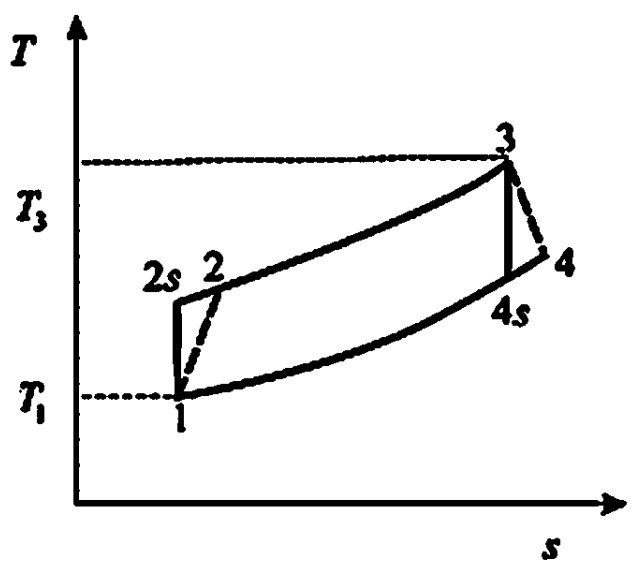
Maximum temperature of the cycle, $T_3 = 1240\text{K}$

Isentropic efficiency of the compressor, $\eta_c = 0.83$

Isentropic turbine efficiency of the compressor, $\eta_T = 0.87$

Step 2 of 8

T-s diagram of cycle:



Step 3 of 8

a)

From the isentropic relation of the compressor is

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{2s}}{295} = (10)^{\frac{1.4-1}{1.4}}$$

$$T_{2s} = 569.55\text{K}$$

Similarly, for turbine

$$\frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{4s}}{1240} = \left(\frac{1}{10}\right)^{\frac{1.4-1}{1.4}}$$

$$T_{4s} = 642.254\text{K}$$

Step 4 of 8

From the relation of isentropic efficiency of the compressor is

$$\eta_c = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)}$$

$$0.83 = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c}$$

$$= 295 + \frac{569.55 - 295}{0.83}$$

$$= 625.783\text{K}$$

Step 5 of 8

Similarly for turbine, isentropic efficiency of the turbine is given as

$$\eta_T = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})}$$

$$0.87 = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$T_4 = T_3 - \eta(T_3 - T_{4s})$$

$$= 1240 - 0.87(1240 - 642.254)$$

$$= 719.96\text{K}$$

Therefore, turbine exit temperature is $\boxed{T_4 = 719.96\text{K}}$

Step 6 of 8

b)

Heat input to the compressor is

$$q_{23} = c_p(T_3 - T_2)$$

$$= 1.005(1240 - 625.783)$$

$$= 617.288\ \text{kJ/kg}$$

Power output from the turbine is

$$q_{41} = c_p(T_4 - T_1)$$

$$= 1.005(719.96 - 295)$$

$$= 427.08\text{kJ/kg}$$

Step 7 of 8

Net work done of the cycle is

$$W_{net} = q_{23} - q_{41}$$

$$= 617.288 - 427.08$$

$$= 190.208\text{kJ/kg}$$

Therefore, net work done of the turbine is $\boxed{W_{net} = 190.208\text{kJ/kg}}$

Step 8 of 8

c)

Thermal efficiency of the cycle is

$$\eta = \frac{W_{net}}{q_{23}}$$

$$= \frac{190.208}{617.288}$$

$$= 0.308$$

$$= 30.81\%$$

Therefore, turbine efficiency is $\boxed{\eta = 30.81\%}$

Question: 9.073P

A simple ideal Brayton cycle uses helium as the working fluid; operates with 12 psia and 60°F at the compressor inlet; has a pressure ratio of 14; and a maximum cycle temperature of 1300°F. How much power will this cycle produce when the rate at which the helium is circulated about the cycle is 100 lbm/min? Use constant specific heats at room temperature.

Step 1 of 7

Given,

Temperature of gas at compressor inlet, $T_1 = 60^0\text{F}$

Pressure of gas at compressor inlet, $P_1 = 12$ psia

Pressure ratio, $r_p = 14$

Maximum temperature, $T_3 = 1300^0\text{F}$

$T_3 = 1760$ R

Mass flow rate of gas, $\dot{m} = 100$ lbm/min

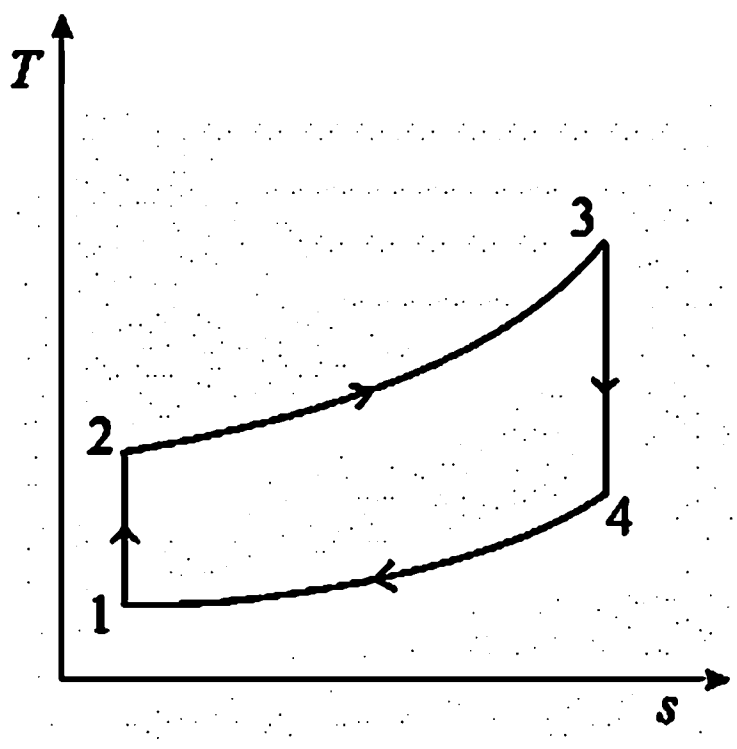
Properties of helium, $R = 0.5$ Btu/lbmF

$c_p = 1.25$ Btu/lbmF

$c_v = 0.75$ Btu/lbmF

$k = 1.667$

Step 2 of 7



Step 3 of 7

Process 1-2 is of Isentropic Compression and hence it follows $\frac{P_1^{(k-1)/k}}{T_1} = \frac{P_2^{(k-1)/k}}{T_2}$

$$T_2 = T_1 \frac{P_2^{(k-1)/k}}{P_1^{(k-1)/k}}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$T_2 = T_1 \times (r)^{(k-1)/k}$$

$$T_2 = (60+460) \times (14)^{(1.667-1)/1.667}$$

$$T_2 = 1494.8 \text{ R}$$

Step 4 of 7

Process 3-4 is of Isentropic expansion and hence it follows $\frac{P_3^{(k-1)/k}}{T_3} = \frac{P_4^{(k-1)/k}}{T_4}$

$$T_4 = T_3 \frac{P_4^{(k-1)/k}}{P_3^{(k-1)/k}}$$

$$T_4 = T_3 \times \left(\frac{P_1}{P_2}\right)^{(k-1)/k}$$

$$T_4 = T_3 \times \left(\frac{1}{r}\right)^{(k-1)/k}$$

$$T_4 = (1760) \times \left(\frac{1}{14}\right)^{(1.667-1)/1.667}$$

$$T_4 = 612.24 \text{ R}$$

Step 5 of 7

Process 2-3 is Constant pressure heat addition hence heat supplied is

$$Q_{in} = mc_p (T_3 - T_2)$$

$$Q_{in} = \frac{100}{60} \times 1.25 \times (1760 - 1494.8)$$

$$Q_{in} = 552.5 \text{ Btu/s}$$

Hence heat input is $\boxed{Q_{in} = 552.5 \text{ Btu/s}}$

Step 6 of 7

Process 4-1 is a Constant pressure heat rejection hence heat rejected is,

$$Q_{out} = mc_p (T_4 - T_1)$$

$$Q_{out} = \frac{100}{60} \times 1.25 \times (612.24 - 520)$$

$$Q_{out} = 192.17 \text{ Btu/s}$$

Hence heat rejected is $\boxed{Q_{out} = 192.17 \text{ Btu/s}}$

Step 7 of 7

Net work output obtained is given by,

$$W_{net} = Q_{in} - Q_{out}$$

$$W_{net} = 552.5 - 192.17$$

$$W_{net} = 360.33 \text{ Btu/s}$$

Net work output is $\boxed{W_{net} = 360.33 \text{ Btu/s}}$

Question: 9.074P

Repeat Prob. 9-73E when the isentropic efficiency of the compressor is 95 percent.

Step 1 of 8

Given,

Temperature of gas at compressor inlet, $T_1 = 60^0\text{F}$

Pressure of gas at compressor inlet, $P_1 = 12$ psia

Pressure ratio, $r_p = 14$

Maximum temperature, $T_3 = 1300^0\text{F}$

$T_3 = 1760$ R

Mass flow rate of gas, $m = 100$ lbm/min

Isentropic efficiency of $\eta_{comp} = 0.95$

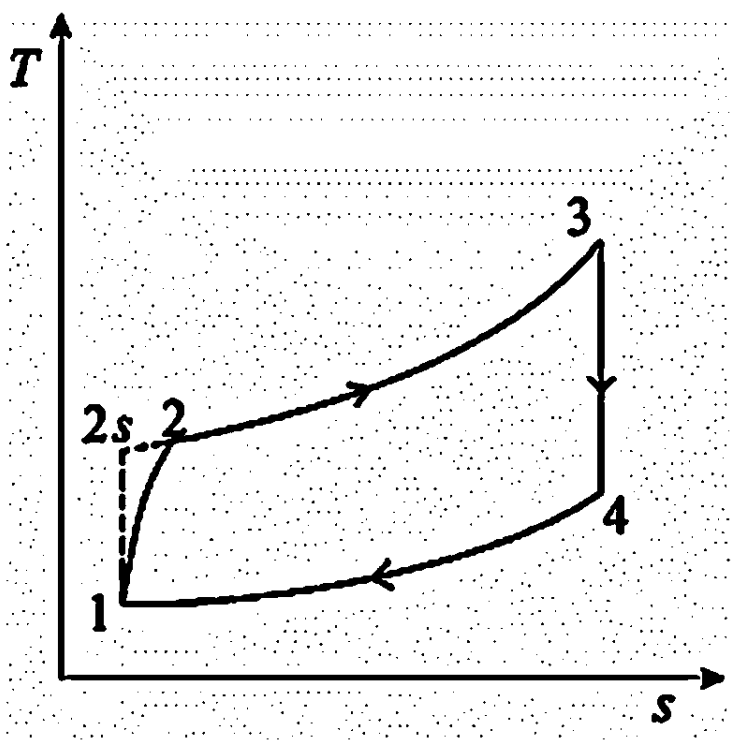
Properties of helium, $R = 0.5$ Btu/lbmF

$c_p = 1.25$ Btu/lbmF

$c_v = 0.75$ Btu/lbmF

$k = 1.667$

Step 2 of 8



Step 3 of 8

Process 1-2s is of Isentropic Compression and hence it follows $\frac{P_1^{(k-1)/k}}{T_1} = \frac{P_2^{(k-1)/k}}{T_{2s}}$

$$T_{2s} = T_1 \frac{P_2^{(k-1)/k}}{P_1^{(k-1)/k}}$$

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$T_{2s} = T_1 \times (r)^{(k-1)/k}$$

$$T_{2s} = (60+460) \times (14)^{(1.667-1)/1.667}$$

$$T_{2s} = 1494.8 \text{ R}$$

Step 4 of 8

Compressor isentropic efficiency is

$$\eta_{comp} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$0.95 = \frac{1494.8 - 520}{T_2 - 520}$$

$$T_2 - 520 = \frac{1494.8 - 520}{0.95}$$

$$T_2 = 520 + \frac{1494.8 - 520}{0.95}$$

$$T_2 = 1546.1 \text{ R}$$

Step 5 of 8

Process 3-4 is of isentropic expansion and hence it follows

$$\frac{P_3^{(k-1)/k}}{T_3} = \frac{P_4^{(k-1)/k}}{T_4}$$

$$T_4 = T_3 \frac{P_4^{(k-1)/k}}{P_3^{(k-1)/k}}$$

$$T_4 = T_3 \times \left(\frac{P_1}{P_2}\right)^{(k-1)/k}$$

$$T_4 = T_3 \times \left(\frac{1}{r}\right)^{(k-1)/k}$$

$$T_4 = (1760) \times \left(\frac{1}{14}\right)^{(1.667-1)/1.667}$$

$$T_4 = 612.24 \text{ R}$$

Step 6 of 8

Process 2-3 is Constant pressure heat addition hence heat supplied is

$$Q_{in} = mc_p (T_3 - T_2)$$

$$Q_{in} = \frac{100}{60} \times 1.25 \times (1760 - 1546.1)$$

$$Q_{in} = 445.625 \text{ Btu/s}$$

Hence heat input is $\boxed{Q_{in} = 445.625 \text{ Btu/s}}$

Step 7 of 8

Process 4-1 is a Constant pressure heat rejection hence heat rejected is,

$$Q_{out} = mc_p (T_4 - T_1)$$

$$Q_{out} = \frac{100}{60} \times 1.25 \times (612.24 - 520)$$

$$Q_{out} = 192.17 \text{ Btu/s}$$

Hence heat rejected is $\boxed{Q_{out} = 192.17 \text{ Btu/s}}$

Step 8 of 8

Net work output obtained is given by,

$$W_{net} = Q_{in} - Q_{out}$$

$$W_{net} = 445.625 - 192.17$$

$$W_{net} = 253.455 \text{ Btu/s}$$

Net work output is $\boxed{W_{net} = 253.455 \text{ Btu/s}}$

Question: 9.075P pg 1

Consider a simple Brayton cycle using air as the working fluid; has a pressure ratio of 12; has a maximum cycle temperature of 600°C; and operates the compressor inlet at 90 kPa and 15°C. Which will have the greatest impact on the back-work ratio: a compressor isentropic efficiency of 90 percent or a turbine isentropic efficiency of 90 percent? Use constant specific heats at room temperature.

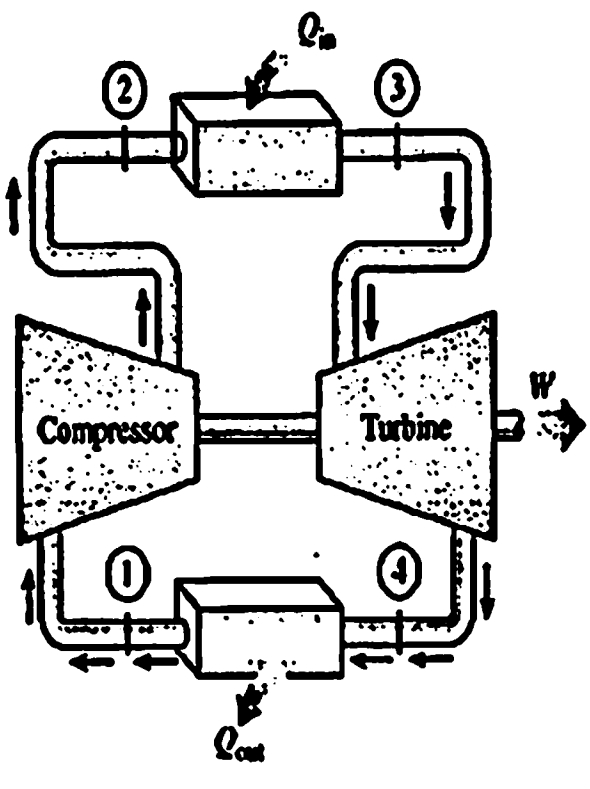


FIGURE P9–75

Step 1 of 11

Given data:

Pressure ratio, $r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4} = 12$

The maximum temperature, $T_3 = 600°C = 873\ K$

Inlet temperature to the compressor, $T_1 = 15°C = 288\ K$

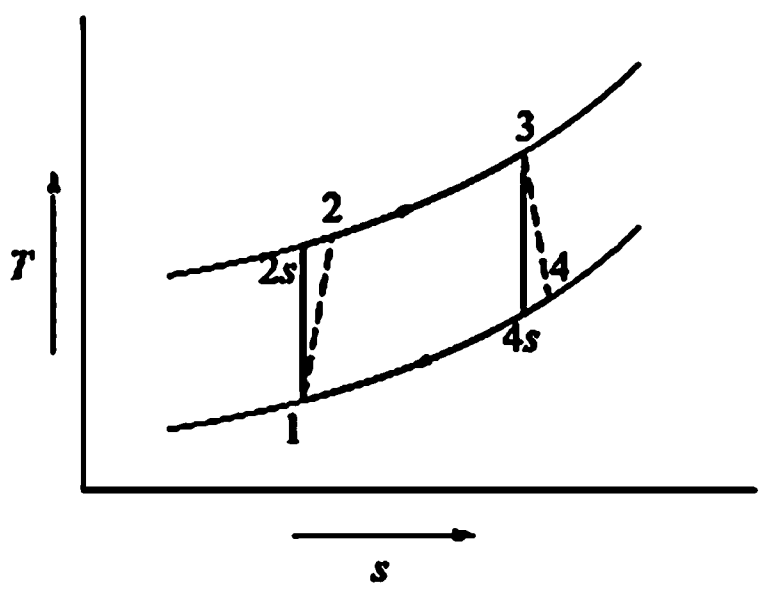
Inlet pressure to the compressor, $P_1 = 90\ kPa$

Isentropic efficiency of the compressor, $\eta_c = 90\%$

Isentropic efficiency of the turbine, $\eta_t = 90\%$

Step 2 of 11

The *T*-*s* diagram of the process:



Step 3 of 11

Since we are considering variable specific heats the specific heats at constant pressure and constant volume will be different at different temperatures

At $T_1 = 288\ K$, from ideal gas properties of air table and interpolation, we get

$$h_1 = (285.14\ kJ/kg) + (290.16\ kJ/kg - 285.14\ kJ/kg)\left(\frac{288\ K - 285\ K}{290\ K - 285\ K}\right)$$

$$= 288.15\ kJ/kg$$

$$P_{r1} = 1.1584 + (1.2311 - 1.1584)\left(\frac{288\ K - 285\ K}{290\ K - 285\ K}\right)$$

$$= 1.2020$$

At $T_3 = 873\ K$, from ideal gas properties of air table and interpolation, we get

$$h_3 = (888.27\ kJ/kg) + (910.56\ kJ/kg - 888.27\ kJ/kg)\left(\frac{873\ K - 860\ K}{880\ K - 860\ K}\right)$$

$$= 902.76\ kJ/kg$$

$$P_{r3} = 63.09 + (68.98 - 63.09)\left(\frac{873\ K - 860\ K}{880\ K - 860\ K}\right)$$

$$= 66.92$$

Step 4 of 11

Process 1-2*s* is isentropic compression

$$\frac{P_2}{P_1} = \frac{P_{r2s}}{P_{r1}}$$

$$P_{r2s} = \left(\frac{P_2}{P_1}\right) P_{r1}$$

$$= 12 \times (1.2020)$$

$$= 14.424$$

At $P_{r2s} = 14.424$ from ideal gas properties of air and interpolation, we get

$$h_{2s} = (586.04\ kJ/kg) + (596.52\ kJ/kg - 586.04\ kJ/kg)\left(\frac{14.424 - 14.38}{15.31 - 14.38}\right)$$

$$= 586.54\ kJ/kg$$

Step 5 of 11

Process 3-4*s* is an isentropic expansion

$$\frac{P_4}{P_3} = \frac{P_{r4s}}{P_{r3}}$$

$$P_{r4s} = \left(\frac{P_4}{P_3}\right) P_{r3}$$

$$= \left(\frac{1}{12}\right)(66.92)$$

$$= 5.577$$

$$= \left(\frac{1}{12}\right)(66.92)$$
$$= 5.577$$

At $P_{r4,s} = 5.577$ from ideal gas properties of air and interpolation, we get

$$h_{4,s} = (441.61\ \text{kJ/kg}) + (451.80\ \text{kJ/kg} - 441.61\ \text{kJ/kg})\left(\frac{5.577 - 5.332}{5.775 - 5.332}\right)$$
$$= 447.24\ \text{kJ/kg}$$

Step 6 of 11

To figure out the impact of the isentropic efficiencies on the back work ratios, we shall consider three cases to evaluate back work ratio.

Case 1:
When both the turbine and compressor are ideal having an isentropic efficiency of 100%
Then, $h_2 = h_{2,s} = 586.54\ \text{kJ/kg}$ and $h_4 = h_{4,s} = 447.24\ \text{kJ/kg}$

Input to the compressor,
$$w_{comp,in} = h_2 - h_1$$
$$= 586.54\ \text{kJ/kg} - 288.15\ \text{kJ/kg}$$
$$= 298.39\ \text{kJ/kg}$$

Output from the turbine,
$$w_{turbine,out} = h_3 - h_4$$
$$= 902.76\ \text{kJ/kg} - 447.24\ \text{kJ/kg}$$
$$= 455.52\ \text{kJ/kg}$$

Step 7 of 11

The back work ratio can be calculated as

$$r_{bw} = \frac{w_{comp,in}}{w_{turbine,out}}$$
$$= \frac{298.39\ \text{kJ/kg}}{455.52\ \text{kJ/kg}}$$
$$= 0.655$$

Hence, the back work ratio in this case is $r_{bw} = 0.655$

Step 8 of 11

Case 2:
When the compressor's isentropic efficiency is 90% and turbine is ideal
Then, $h_4 = h_{4,s} = 447.24\ \text{kJ/kg}$

Isentropic efficiency of the compressor is given by

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$h_2 - h_1 = \frac{h_{2s} - h_1}{\eta_c}$$
$$= \frac{586.54\ \text{kJ/kg} - 288.15\ \text{kJ/kg}}{0.90}$$
$$= 331.54\ \text{kJ/kg}$$

The input to the compressor,
$$w_{comp,in} = h_2 - h_1$$
$$= 331.54\ \text{kJ/kg}$$

Step 9 of 11

Output from the turbine,
$$w_{turbine,out} = h_3 - h_4$$
$$= 902.76\ \text{kJ/kg} - 447.24\ \text{kJ/kg}$$
$$= 455.52\ \text{kJ/kg}$$

The back work ratio can be calculated as

$$r_{bw} = \frac{w_{comp,in}}{w_{turbine,out}}$$
$$= \frac{331.54\ \text{kJ/kg}}{455.52\ \text{kJ/kg}}$$
$$= 0.728$$

Hence, the back work ratio in this case is $r_{bw} = 0.728$

Therefore, if the isentropic compressor efficiency is 90%, the back work ratio increased by 0.073.

Step 10 of 11

Case 2:
When the turbine's isentropic efficiency is 90% and compressor is ideal
Then, $h_2 = h_{2,s} = 586.54\ \text{kJ/kg}$

Isentropic efficiency of the turbine is given by

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4,s}}$$

$$h_3 - h_4 = \eta_t (h_3 - h_{4,s})$$
$$= 0.90(902.76\ \text{kJ/kg} - 447.24\ \text{kJ/kg})$$
$$= 408.97\ \text{kJ/kg}$$

Work output from the turbine,
$$w_{turbine,out} = h_3 - h_4$$
$$= 408.97\ \text{kJ/kg}$$

Work output from the turbine, 9.075P pg 3

$$w_{turbine,out} = h_3 - h_4$$
$$= 408.97 \text{ kJ/kg}$$

Step 11 of 11

Work input to the compressor,

$$w_{comp,in} = h_2 - h_1$$
$$= 586.54 \text{ kJ/kg} - 288.15 \text{ kJ/kg}$$
$$= 298.39 \text{ kJ/kg}$$

The back work ratio can be calculated as

$$r_{bw} = \frac{w_{comp,in}}{w_{turbine,out}}$$
$$= \frac{298.39 \text{ kJ/kg}}{409.97 \text{ kJ/kg}}$$
$$= 0.728$$

Hence, the back work ratio in this case is $r_{bw} = 0.728$

Therefore, if the isentropic turbine efficiency is 90%, the back work ratio increased by 0.073.

Thus, the impact is same for both the cases of turbine having an isentropic efficiency of 90% and compressor having an isentropic efficiency of 90%.

Question:

Air is used as the working fluid in a simple ideal Brayton cycle that has a pressure ratio of 12, a compressor inlet temperature of 300 K, and a turbine inlet temperature of 1000 K. Determine the required mass flow rate of air for a net power output of 70 MW, assuming both the compressor and the turbine have an isentropic efficiency of (*a*) 100 percent and (*b*) 85 percent. Assume constant specific heats at room temperature.

Step 1 of 14

Given

Pressure ratio $r_p = \frac{P_2}{P_1} = 12$

Maximum temperature $T_3 = 1000\text{K}$

Minimum temperature $T_1 = 300\text{K}$

Power output $P = 70\,\text{MW}$

Step 2 of 14

Considering constant specific heats at room temperature

Gas constant $R = 0.287\,\text{kJ/kg.K}$

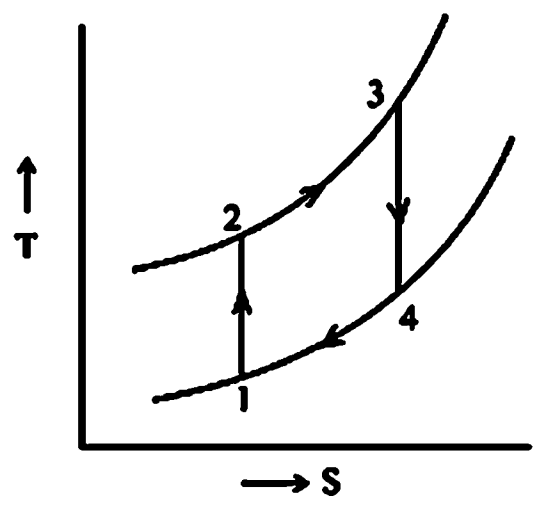
$c_p = 1.005\,\text{kJ/kg.K}$

$c_v = 0.718\,\text{kJ/kg.K}$

$k = 1.4$

Step 3 of 14

(a) When isentropic efficiency of compressor and turbine each are having 100%



Step 4 of 14

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_2}{300} = (12)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 610.18\text{K}$$

Step 5 of 14

Similarly process 3-4 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{1000} = \left(\frac{1}{12}\right)^{\frac{1.4-1}{1.4}}$$

$$T_4 = 491.657\text{K}$$

Step 6 of 14

The net work produced per unit mass is given by

$$w_{net} = w_t - w_c$$

$$w_{net} = c_p(T_3 - T_4) - c_p(T_2 - T_1)$$

$$= (1.005)\{(1000 - 491.657) - (610.18 - 300)\}$$

$$w_{net} = 199.153\,\text{kJ/kg}$$

Step 7 of 14

Power output is expressed as

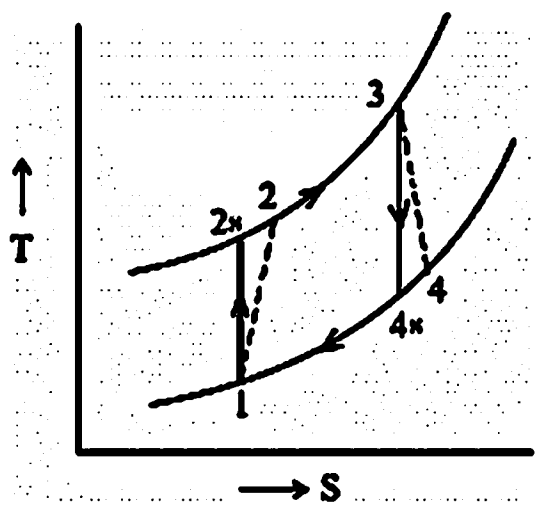
$$P = \dot{m} w_{net}$$

$$70\times10^3 = \dot{m}(199.153)$$

$$\boxed{\dot{m} = 351.48\,\text{kg/s}}$$

Step 8 of 14

(b) When isentropic efficiencies of turbine and compressor each are having $\eta_c = \eta_t = 85\%$



Step 9 of 14

Process 1-2s is isentropic compression

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Step 9 of 14

Process 1-2s is isentropic compression

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{2s}}{300} = (12)^{\frac{1.4-1}{1.4}}$$

$$T_{2s} = 610.18\text{K}$$

Step 10 of 14

Since the isentropic efficiency of the compressor is given by

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$0.85 = \frac{610.18 - 300}{T_2 - 300}$$

$$T_2 = 664.9176\text{K}$$

Step 11 of 14

Similarly process 3-4s is isentropic expansion

$$\frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{4s}}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{4s}}{1000} = \left(\frac{1}{12}\right)^{\frac{1.4-1}{1.4}}$$

$$T_{4s} = 491.657\text{K}$$

Step 12 of 14

Since the isentropic efficiency of the turbine is given by

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$0.85 = \frac{1000 - T_4}{1000 - 491.657}$$

$$T_4 = 567.9\text{K}$$

Step 13 of 14

The net work produced per unit mass is given by

$$w_{net} = w_t - w_c$$

$$w_{net} = c_p(T_3 - T_4) - c_p(T_2 - T_1)$$

$$= (1.005)\{(1000 - 567.9) - (664.9176 - 300)\}$$

$$= 1.005\{432.1 - 364.9176\}$$

$$w_{net} = 67.518\ \text{kJ/kg}$$

Step 14 of 14

Power output is expressed as

$$P = \dot{m} w_{net}$$

$$70 \times 10^3 = \dot{m}(67.518)$$

$$\boxed{\dot{m} = 1036.76\ \text{kg/s}}$$

Question: 9.077P

Air enters the compressor of a gas-turbine engine at 300 K and 100 kPa, where it is compressed to 700 kPa and 580 K. Heat is transferred to air in the amount of 950 kJ/kg before it enters the turbine. For a turbine efficiency of 86 percent. determine (*a*) the fraction of the turbine work output used t I drive the compressor and (*b*) the thermal efficiency. Assume variable specific heats for air.

Step 1 of 11

Given

Initial pressure $P_1 = 100\ \text{kPa}$

Initial temperature $T_1 = 300\text{K}$

Compressor exit pressure $P_2 = 700\ \text{kPa}$

Compressor exit temperature $T_2 = 580\text{K}$

Heat input $q_{in} = 950\ \text{kJ/kg}$

Turbine efficiency $\eta_t = 86\%$

Step 2 of 11

Since we are considering variable specific heats the specific heat at constant pressure and at constant volume will be different at different temperatures.

At $T_1 = 300\text{K}$ from ideal gas properties of air

$h_1 = 300.19\ \text{kJ/kg}$

$P_{r1} = 1.386$

At $T_2 = 580\text{K}$ from ideal gas properties of air

$h_2 = 586.04\ \text{kJ/kg}$

$P_{r3} = 14.38$

Step 3 of 11

Pressure ratio

$$r_p = \frac{P_2}{P_1}$$

$$r_p = \frac{700}{100}$$

$$r_p = 7$$

Step 4 of 11

The heat input is expressed as

$$q_{in} = h_3 - h_2$$

$$950 = h_3 - 586.04$$

$$h_3 = 1536.04\ \text{kJ/kg}$$

Step 5 of 11

At $h_3 = 1536.04\ \text{kJ/kg}$ from ideal gas properties of air

$$P_{r3} = 450.5 + (478.0 - 450.5)\left(\frac{1536.04 - 1515.42}{1539.44 - 1515.42}\right)$$

$$P_{r3} = 474.107$$

Step 6 of 11

Process 3-4s is isentropic expansion

$$\frac{P_4}{P_3} = \frac{P_{r4s}}{P_{r3}}$$

$$P_{r4s} = P_{r3} \times \frac{P_4}{P_3}$$

$$P_{r4} = 474.107 \times \frac{1}{7}$$

$$P_{r4s} = 67.729$$

Step 7 of 11

At $P_{r4s} = 67.729$ from ideal gas properties of air

$$h_{4s} = 888.27 + (910.56 - 888.27)\left(\frac{67.729 - 63.09}{68.98 - 63.09}\right)$$

$$h_{4s} = 905.825\ \text{kJ/kg}$$

Step 8 of 11

Since the isentropic efficiency of the turbine is given by

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$0.86 = \frac{1536.04 - h_4}{1536.04 - 905.825}$$

$$h_4 = 994.055\text{kJ/kg}$$

Step 9 of 11

The work required to compressor is expressed as

$$w_c = h_2 - h_1$$

$$w_c = 586.04 - 300.19$$

$$w_c = 285.85\ \text{kJ/kg}$$

The work done by turbine is expressed as

$$w_t = h_3 - h_4$$

$$w_t = 1536.04 - 994.055$$

$$w_t = 541.985\text{kJ/kg}$$

Step 10 of 11

(a) The fraction of turbine work output used to drive the compressor

$$r_{bw} = \frac{w_c}{w_t}$$

$$= \frac{285.85}{541.985}$$

$$r_{bw} = 0.527$$

Fraction of 0.527 of work is delivered by turbine to run the compressor

Step 11 of 11

(b) The thermal efficiency

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{w_t - w_c}{q_{in}}$$

$$\eta_{th} = \frac{541.985 - 285.85}{950}$$

$$= 0.2696$$

$$\boxed{\eta_{th} = 26.96\%}$$

Question: 9.078P pg 1

Repeat Prob. 9–77 using constant specific heats at room temperature.

Step 1 of 7

Solving the Prob. 9-78 using constant specific heats at room temperature yields the same result obtained for the Prob. 9-78.

Instead of that, solve the Prob. 9-100 using constant specific heats at room temperature as follows:

Obtain the properties of air from the **Table A-2a**.

At room temperature,

Gas constant, $R = 0.287\,\text{kJ/kg}\cdot\text{K}$

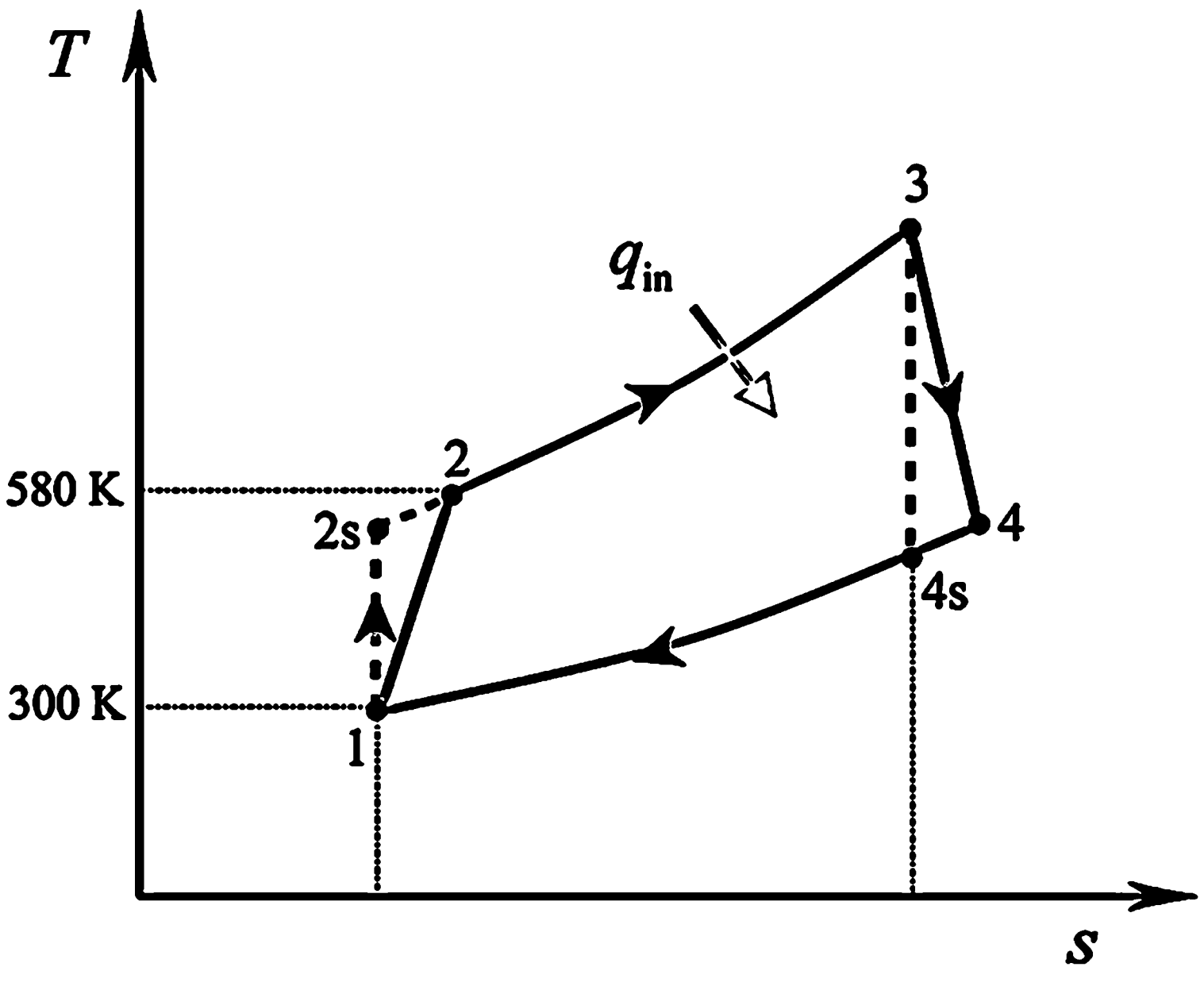
Specific heats at constant pressure, $c_p = 1.005\,\text{kJ/kg}\cdot\text{K}$

Specific heats at constant volume, $c_v = 0.718\,\text{kJ/kg}\cdot\text{K}$

Ratio of specific heats (or) Specific heat ratio, $k = 1.4$

Step 2 of 7

Draw the cyclical process on $T-s$ coordinates as follow:



Step 3 of 7

Write an expression for the heat input to the system during the constant pressure heat addition process $2-3$.

$$q_{in,2\text{-}3} = c_p(T_3 - T_2)$$

Here, the heat input to the system during the process $2-3$ is $q_{in,2\text{-}3}$, the temperature of air at the beginning of heat addition process is T_2, the temperature of air at the end of heat addition process is T_3 and the specific heat at constant pressure is c_p.

Substitute $950\,\text{kJ/kg}$ for $q_{in,2\text{-}3}$, $580\,\text{K}$ for T_2 and $1.005\ \text{kJ/kg}\cdot\text{K}$ for c_p.

$$950 = 1.005(T_3 - 580)$$
$$T_3 = 1525.27\,\text{K}$$

Step 4 of 7

Consider the process $3-4s$ (Isentropic Expansion process):

Calculate the temperature at the end of expansion in an isentropic expansion process (T_{4s}) by using the following equation,

$$\frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\left(\frac{k-1}{k}\right)}$$

$$\frac{T_{4s}}{T_3} = \left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)} \qquad \left(\because P_1 = P_{4s} = P_4 \text{ and } P_{2s} = P_2 = P_3\right)$$

$$T_{4s} = T_3\left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)}$$

Here, the temperature of air before expansion is T_3, the pressure of air before expansion is P_3, the temperature at the end of expansion in an isentropic expansion process is T_{4s} and the pressure of air at the end of expansion is P_4.

Substitute $1525.27\,\text{K}$ for T_3, 1.4 for k, $100\,\text{kPa}$ for P_1, and $700\,\text{kPa}$ for P_2.

$$T_{4s} = 1525.27 \times \left(\frac{100}{700}\right)^{\left(\frac{1.4-1}{1.4}\right)}$$
$$= 874.76\,\text{K}$$

Step 5 of 7

Calculate the work done in compressing the air by using the following equation,

$$w_{C,in} = h_2 - h_1$$
$$= c_p(T_2 - T_1)$$

Here, the temperature of air before compression is T_1, and the temperature of air after compression is T_2.

Substitute $300\,\text{K}$ for T_1, $580\,\text{K}$ for T_2 and $1.005\ \text{kJ/kg}\cdot\text{K}$ for c_p.

$$= c_p(T_2 - T_1)$$

Here, the temperature of air before compression is T_1, and the temperature of air after compression is T_2.

Substitute $300\ \text{K}$ for T_1, $580\ \text{K}$ for T_2 and $1.005\ \text{kJ/kg·K}$ for c_p.

$$\begin{aligned} w_{C,in} &= c_p(T_2 - T_1) \\ &= 1.005(580 - 300) \\ &= 281.4\ \text{kJ/kg} \end{aligned}$$

Calculate the actual work developed by the turbine during the expansion process.

$$\begin{aligned} (w_{T,out})_{Actual} &= \eta_T (w_{T,out})_{Ideal} \\ &= \eta_T (h_3 - h_{4s}) \\ &= \eta_T c_p (T_3 - T_{4s}) \end{aligned}$$

Here, the efficiency of the turbine is η_T.

Substitute $1525.27\ \text{K}$ for T_3, $874.76\ \text{K}$ for T_{4s}, $1.005\ \text{kJ/kg·K}$ for c_p and 0.86 for η_T.

$$\begin{aligned} (w_{T,out})_{Actual} &= \eta_T c_p (T_3 - T_{4s}) \\ &= 0.86 \times 1.005(1525.27 - 874.76) \\ &= 562.23\ \text{kJ/kg} \end{aligned}$$

Step 6 of 7

(a)

Calculate the fraction of turbine work, used to drive the compressor by using the following equation,

$$r_{bw} = \frac{w_{C,in}}{(w_{T,out})_{Actual}}$$

Substitute $281.4\ \text{kJ/kg}$ for $w_{C,in}$ and $562.23\ \text{kJ/kg}$ for $(w_{T,out})_{Actual}$.

$$\begin{aligned} r_{bw} &= \frac{281.4\ \text{kJ/kg}}{562.23\ \text{kJ/kg}} \\ &= 0.5005 \\ &= 50.05\% \end{aligned}$$

Therefore, a fraction of $\boxed{50.05\%}$ of the turbine work is used to drive the compressor.

Step 7 of 7

(b)

Calculate the net work done by the system.

$$w_{act,out} = (w_{T,out})_{Actual} - w_{C,in}$$

Substitute $281.4\ \text{kJ/kg}$ for $w_{C,in}$ and $562.23\ \text{kJ/kg}$ for $(w_{T,out})_{Actual}$.

$$\begin{aligned} w_{act,out} &= 562.23\ \text{kJ/kg} - 281.4\ \text{kJ/kg} \\ &= 280.83\ \text{kJ/kg} \end{aligned}$$

Calculate the thermal efficiency of the cycle by using the following equation,

$$\eta_{th} = \frac{w_{act,out}}{q_{in,2\text{-}3}}$$

Substitute 950kJ/kg for $q_{in,2\text{-}3}$, and $280.83\ \text{kJ/kg}$ for $w_{act,out}$.

$$\begin{aligned} \eta_{th} &= \frac{280.83\ \text{kJ/kg}}{950\ \text{kJ/kg}} \\ &= 0.2956 \\ &= 29.56\% \end{aligned}$$

Therefore, the thermal efficiency of the cycle is $\boxed{29.56\%}$.

Question: 9.079P

An aircraft engine operates on a simple ideal Brayton cycle with a pressure ratio of 10. Heat is added to the eyclj at a rate of 500 kW; air passes through the engine at a rate of 1 kg/s; and the air at the beginning of the compression is ati 70 kPa and 0°C Determine the power produced by this engine and its thermal efficiency. Use constant specific heats at room temperature.

Step 1 of 8

Given

Pressure ratio $r_p = 10$

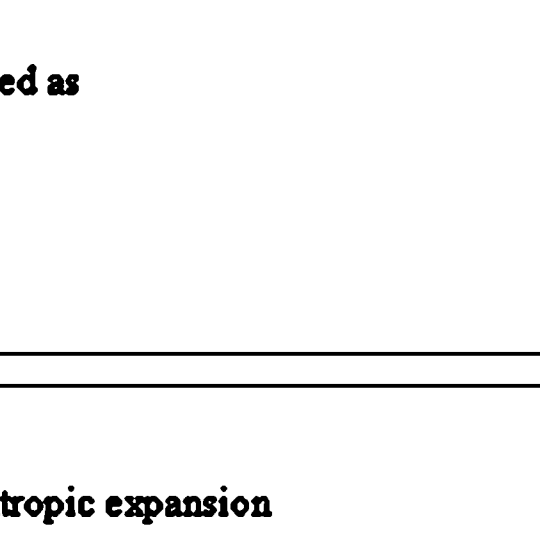
Initial pressure $P_1 = 70\ \text{kPa}$

Inlet temperature $T_1 = 0°\text{C} + 273 = 273\text{K}$

Heat supplied $\dot{Q}_{in} = 500\ \text{kW}$

Mass flow rate $\dot{m} = 1\ \text{kg/s}$

Step 2 of 8



Step 3 of 8

Considering constant specific heats at room temperature the properties of air will be

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 4 of 8

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_2}{273} = (10)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 527.08\text{K}$$

Step 5 of 8

The heat supplied is expressed as

$$\dot{Q}_{in} = \dot{m}c_p(T_3 - T_2)$$

$$500 = 1 \times 1.005(T_3 - 527)$$

$$T_3 = 1024.5\text{K}$$

Step 6 of 8

Similarly process 3-4 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{1024.5} = \left(\frac{1}{10}\right)^{\frac{1.4-1}{1.4}}$$

$$T_4 = 530.637\text{K}$$

Step 7 of 8

The Power produced by this engine

$$\dot{W}_{net} = \dot{m}(w_T - w_c)$$

$$\dot{W}_{net} = \dot{m}\{c_p(T_3 - T_4) - c_p(T_2 - T_1)\}$$

$$= (1)(1.005)\{(1024.5 - 530.637) - (527.08 - 273)\}$$

$$= 1.005\{493.863 - 254.08\}$$

$$\boxed{\dot{W}_{net} = 240.98\ \text{kW}}$$

Step 8 of 8

The thermal efficiency is defined as

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$= \frac{240.98}{500}$$

$$= 0.48196$$

$$\boxed{\eta_{th} = 48.19\%}$$

Question: 9.080P

Repeat Prob. 9–79 for a pressure ratio of 15.

Step 1 of 8

Given

Pressure ratio $r_p = 15$

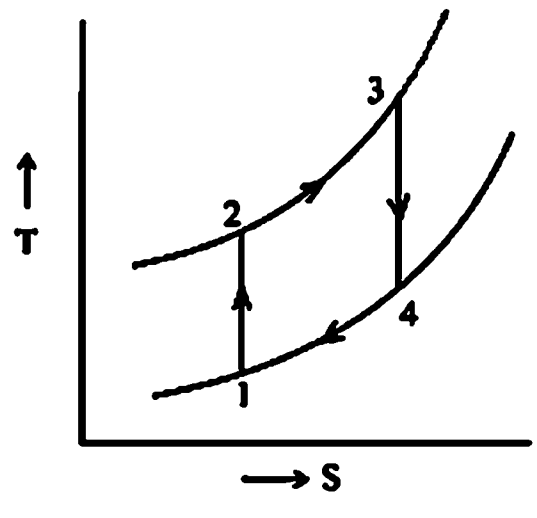
Initial pressure $P_1 = 70\ \text{kPa}$

Inlet temperature $T_1 = 0°\text{C} + 273 = 273\text{K}$

Heat supplied $\dot{Q}_{in} = 500\ \text{kW}$

Mass flow rate $\dot{m} = 1\ \text{kg/s}$

Step 2 of 8



Step 3 of 8

Considering constant specific heats at room temperature the properties of air will be

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 4 of 8

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_2}{273} = (15)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 591.818\text{K}$$

Step 5 of 8

The heat supplied is expressed as

$$\dot{Q}_{in} = \dot{m}c_p(T_3 - T_2)$$

$$500 = 1 \times 1.005(T_3 - 591.818)$$

$$T_3 = 1089.33\text{K}$$

Step 6 of 8

Similarly process 3-4 is isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{1089.33} = \left(\frac{1}{15}\right)^{\frac{1.4-1}{1.4}}$$

$$T_4 = 502.496\text{K}$$

Step 7 of 8

The Power produced by this engine

$$\dot{W}_{net} = \dot{m}(w_T - w_c)$$

$$\dot{W}_{net} = \dot{m}\{c_p(T_3 - T_4) - c_p(T_2 - T_1)\}$$

$$= (1)(1.005)\{(1089.33 - 502.496) - (591.818 - 273)\}$$

$$= 1.005(586.834 - 318.818)$$

$$\boxed{\dot{W}_{net} = 269.356\ \text{kW}}$$

Step 8 of 8

The thermal efficiency is defined as

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$= \frac{269.356}{500}$$

$$= 0.5387$$

$$\boxed{\eta_{th} = 53.87\%}$$

Question: 9.081P pg 1

A gas-turbine power plant operates on the simple Brayton cycle between the pressure limits of 100 and 2000 kPa. The working fluid is air, which enters the compressor at 40°C at a] rate of 700 m3/Vmin and leaves the turbine at 650°C. Using variable specific heats for air and assuming a compressor isentropic efficiency of 85 percent and a turbine isentropic efficiency of 88 percent, determine *(a)* the net power /output *(b)* the back work ratio, and (c) the thermal efficiency.

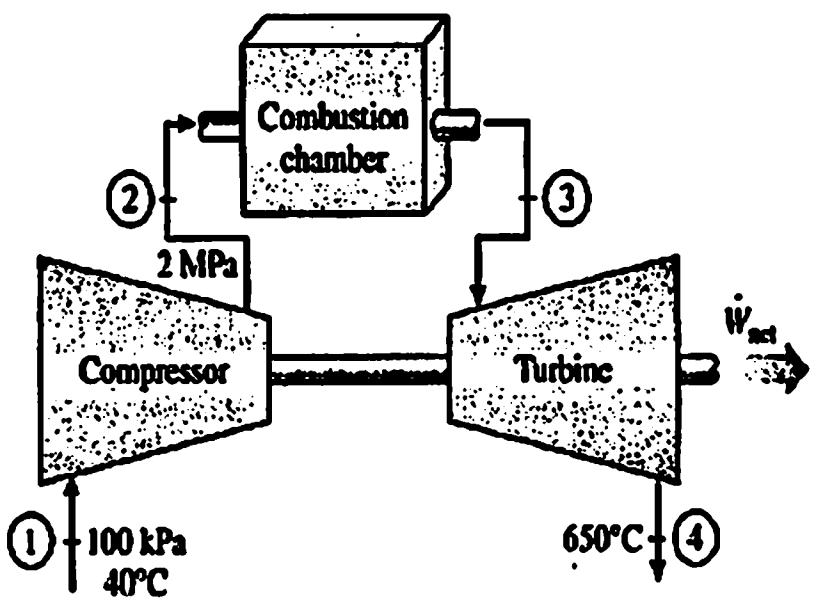


FIGURE P9–81

Step 1 of 11

Pressure inlet to the compressor, $P_1 = 100\,\text{kPa}$

Pressure outlet to the compressor, $P_2 = 2000\,\text{kPa}$

Inlet temperature of working fluid enters into compressor, $T_1 = 40°C = 313\,\text{K}$

Mass flow rate of the working fluid, $V_1 = 700\,\text{m}^3/\text{min}$

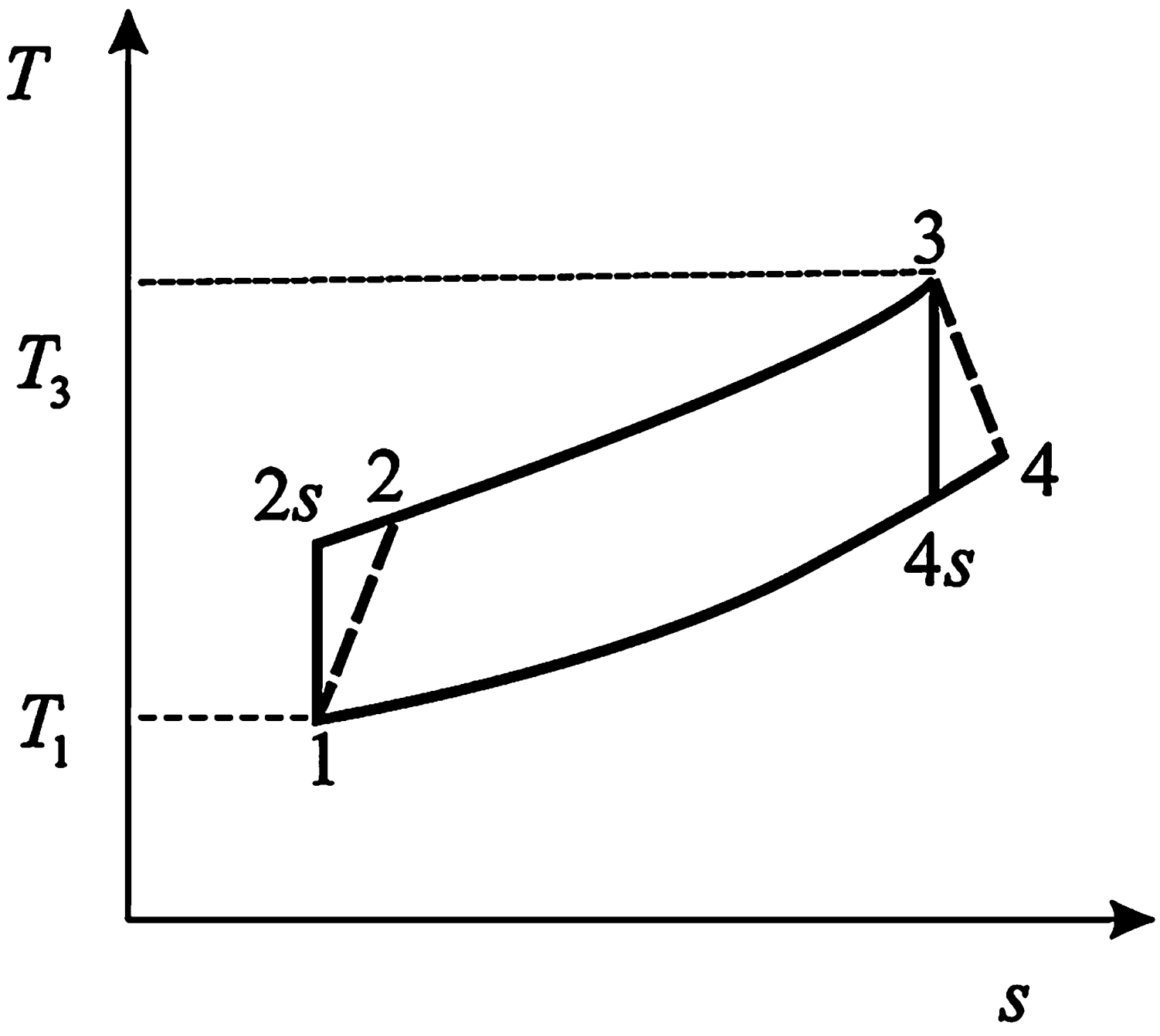
Exit temperature of the working fluid from the turbine, $T_4 = 650°C$

Isentropic efficiency of the compressor, $\eta_c = 0.85$

Isentropic efficiency of the turbine, $\eta_T = 0.88$

Step 2 of 11

Ts diagram:



Step 3 of 11

Properties of the gas at various states can be finding by using EES software only. Because in this problems we have more number of unknowns,

From the properties of the ideal gas air

At $T_1 = 313\,\text{K}$

By using the method of interpolation

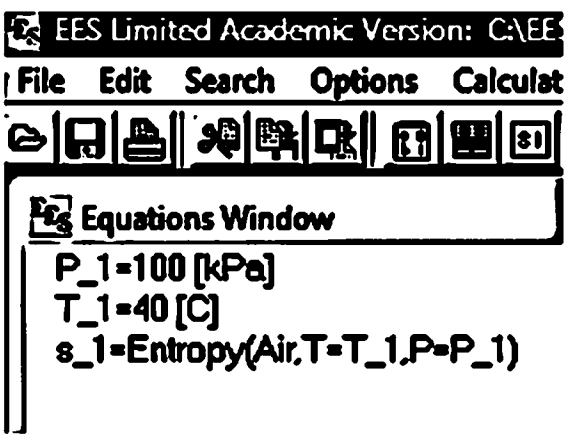
$$h_1 = 310.24 + \frac{(313-310)(315.27-310.24)}{(315-310)}$$

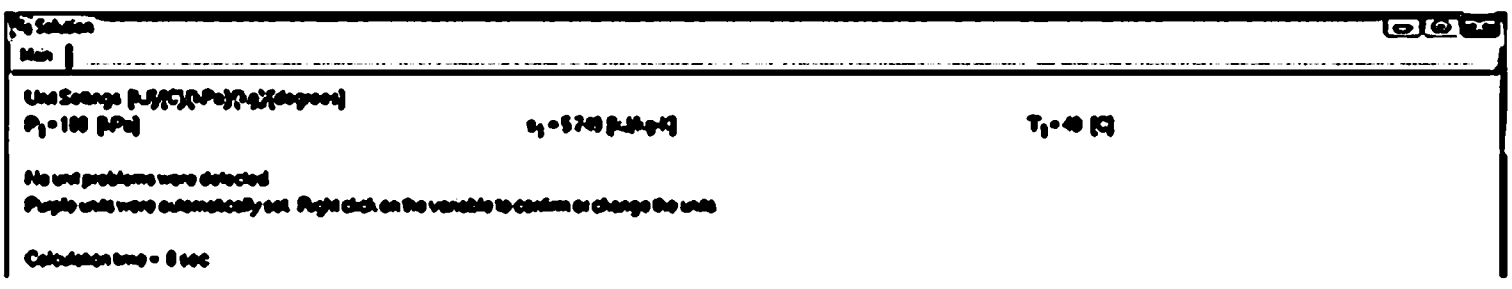
$$= 313.258\,\text{kJ/kg}$$

By using EES software

At pressure $P_1 = 100\,\text{kPa}$ and $T_1 = 313\,\text{K}$

Step 4 of 11





Unit Settings: [kJ/(C)/(kPa)/[kg]/[degree]
$P_1 = 100$ [kPa] $s_1 = 5.749$ [kJ/kg-K] $T_1 = 40$ [C]

No unit problems were detected.
Duplicate units were automatically set. Right click on the variable to confirm or change the units.

Calculation time = 0 sec

Entropy $s_1 = 5.749\,\text{kJ/kg}\cdot\text{K}$

Step 5 of 11

From the table of "Ideal gas properties of the air"

At $T_4 = 650°\text{C}$

$$h_4 = 955.38 + \frac{(923-920)(977.92-955.38)}{(940-920)}$$
$$= 958.761\,\text{kJ/kg}$$

Step 6 of 11

Enthalpy at state 2s is

Equations Window

```
P_2=2000 [kPa]
s_2=5.749 [kJ/kgK]
h_2s=Enthalpy(Air,P=P_2,s=s_2)
```

Solution
Main

Unit Settings: [kJ/(C)/(kPa)/[kg]/[degree]
$h_{2s} = 737$ $P_2 = 2000$ [kPa] $s_2 = 5.749$ [kJ/kgK]

2 potential unit problems were detected. Check Units

Calculation time = 0 sec

Then, from the relation of isentropic efficiency is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$
$$0.85 = \frac{737 - 313.258}{h_2 - 313.258}$$
$$h_2 = 811.778\,\text{kJ/kg}$$

Similarly, for turbine is given as

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$
$$0.88 = \frac{h_3 - 958.761}{h_3 - 313.258}$$

Step 7 of 11

The enthalpy at state 3 can be finding by using the EES software.

Equations Window

```
h_3=Enthalpy(Air,T=T_3)
s_3=Entropy(Air,T=T_3,P=P_2)
h_4s=Enthalpy(Air,P=P_1,s=s_3)
0.88=h_3-959.2/h_3-h_4s
P_1=100 [kPa]
P_2=2000 [kPa]
```

Solution
Main

Unit Settings: [kJ/(C)/(kPa)/[kg]/[degree]
$h_3 = 1873$ $T_3 = 1421$ [C] $s_3 = 6.736$ [kJ/kgK]

2 potential unit problems were detected. Check Units

Calculation time = 0 sec

From the above solution window, we get

Enthalpy at state 3 is $h_3 = 1873\,\text{kJ/kg}$

Temperature at state 3 is $T_3 = 1421°\text{C}$

Entropy at state 3 is $s_3 = 6.736\,\text{kJ/kg}\cdot\text{K}$

Step 8 of 11

Mass flow rate of air in compressor is given as

$$P_1V_1 = mRT_1$$
$$m = \frac{\left(100 \times \frac{700}{60}\,\text{m}^3/\text{s}\right)}{0.287 \times (40+273)}$$
$$= 12.99\,\text{kg/s}$$

Work input to the compressor is

$$W_{in,c} = m(h_2 - h_1)$$
$$= 12.99(811.778 - 313.258)$$
$$= 6475.774\,\text{kW}$$

Step 9 of 11

Power produced by turbine is

$$W_T = m(h_3 - h_4)$$
$$= 12.99(1873 - 958.761)$$
$$= 11875.96\,\text{kW}$$

Net power produced by cycle is

$$W_{net} = W_T - W_{in,c}$$
$$= 11875.96 - 6475.774$$
$$= 5400.1906\,\text{kW}$$

Net power produced by cycle is

$$\begin{aligned} W_{net} &= W_T - W_{in,c} \\ &= 11875.96 - 6475.774 \\ &= 5400.1906\,\text{kW} \end{aligned}$$

Therefore, net power produced by the cycle is $\boxed{W_{net} = 5400.1906\,\text{kW}}$

Step 10 of 11

b)

The back work ratio is

$$\begin{aligned} W_{back} &= \frac{W_{in,c}}{W_T} \\ &= \frac{6475.774}{11875.96} \\ &= 0.5452 \end{aligned}$$

Therefore, back work ratio is $\boxed{W_{back} = 0.5452}$

Step 11 of 11

c)

Heat input to the combustion chamber is

$$\begin{aligned} q_{in} &= m(h_3 - h_2) \\ &= 12.99(1873 - 811.778) \\ &= 13785.273\,\text{kW} \end{aligned}$$

Thermal efficiency is

$$\begin{aligned} \eta_{Th} &= \frac{W_{net}}{q_{in}} \\ &= \frac{5400.1906}{13785.273} \\ &= 0.3917 \\ &= 39.17\% \end{aligned}$$

Therefore, thermal efficiency of the cycle is $\boxed{\eta_{Th} = 39.17\%}$

Question: 9.082P

A gas-turbine power plant operates on the simple Brayton cycle between the pressure limits of 100 and 800 kPa. Air enters the compressor at 30°C and leaves at 3300C at a mass flow rate of 200 kg/s. The maximum cycle temperature is 1400 K. During operation of the cycle, the net power output is measured experimentally to be 60 MW. Assume constant properties for air at 300 K with *cv* -0.718 kJ/kg-K*k cp* =1.005 kJ/kg-K. *R*= 0.287 kJ/kg.K *k* = 1.4.

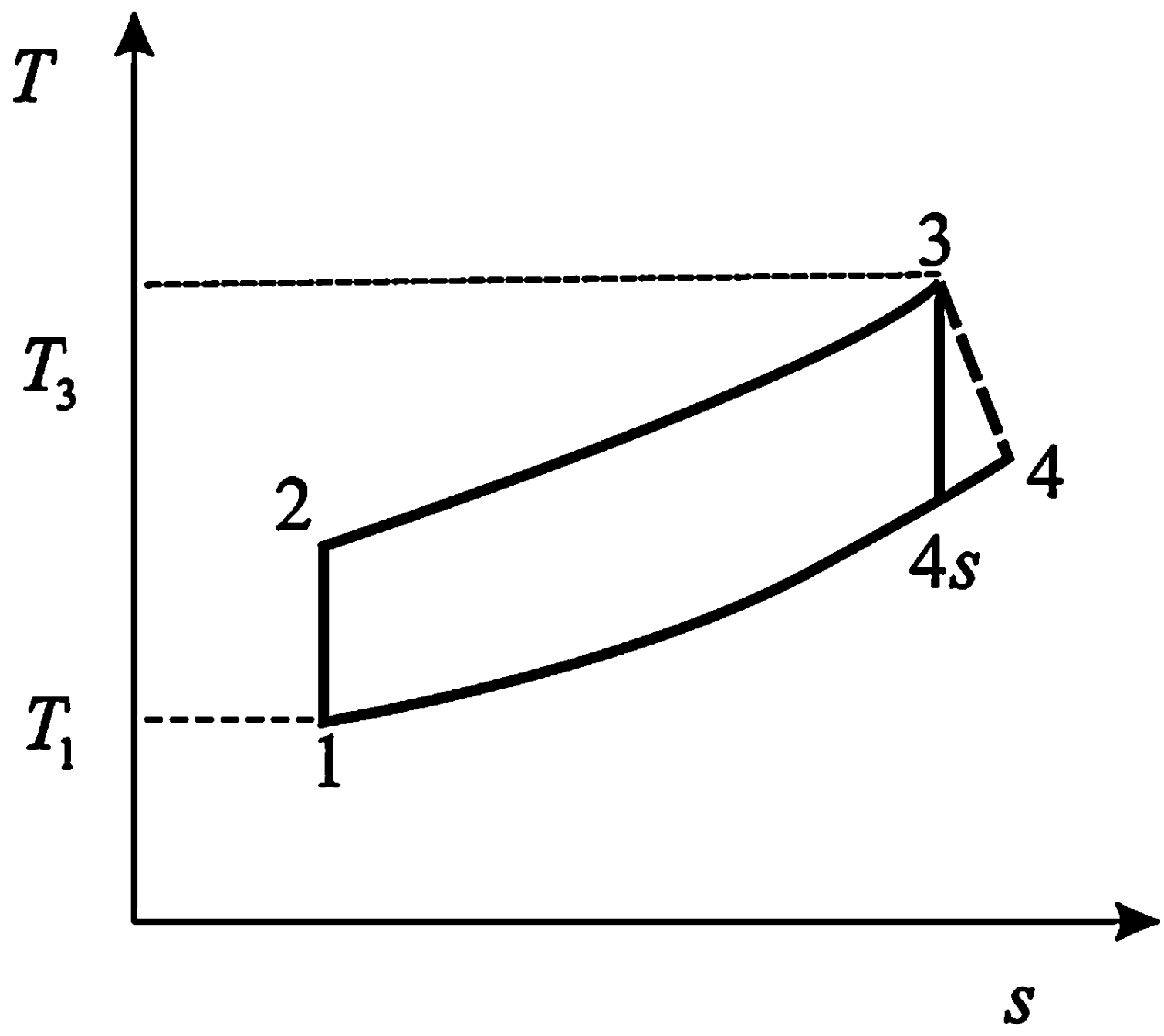
(a) Sketch the *T-s* diagram for the cycle.

(b) Determine the isentropic efficiency of the turbine for these operating conditions.

(c) Determine the cycle thermal efficiency.

Step 1 of 6

T-s diagram for gas turbine cycle shown in below:



Process 1-2 is isentropic compression in the compressor

Process 2-3 is constant pressure heat addition,

Process 3-4s is isentropic expansion

Process 3-4 is actual expansion in the turbine

Process 4s-1 is constant pressure heat rejection.

Step 2 of 6

From the relation of isentropic process,

$$\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}}=\frac{T_3}{T_4}$$

Here, P_3 and P_4 are the pressure at state point 3 and 4 respectively and T_3 and T_4 are the temperature at state points 3 and 4 respectively.

Substitute $800\,\text{kPa}$ for P_3, $100\,\text{kPa}$ for P_4 and $1400\,\text{K}$ for T_3

$$\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}}=\frac{T_3}{T_4}$$

$$\left(\frac{800}{100}\right)^{\frac{1.4-1}{1.4}}=\frac{1400}{T_4}$$

$$T_4=772.862\,\text{K}$$

Step 3 of 6

Calculate the actual work done turbine

$$W_{turb}=mc_p(T_3-T_4)$$
$$=(200)(1.005)(1400-772.862)$$
$$=126054.738\,\text{kW}$$

Step 4 of 6

Calculate the compressor work

$$W_{comp}=mc_p(T_2-T_1)$$

Here, c_p is the specific heat and m is the mass flow rate

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p and $200\,\text{kg/s}$ for m $330°\text{C}$ for T_2 and $30°\text{C}$ for T_1

$$W_{comp}=mc_p(T_2-T_1)$$
$$=200(1.005)(330-30)$$
$$=60300\,\text{kW}$$

Step 5 of 6

Calculate the turbine work done by using the relation of net work done by cycle

$$W_{net}=-W_{comp}+W_{turb}$$
$$W_{turb}=W_{comp}+W_{net}$$

Substitute $60\times10^3\,\text{kW}$ for W_{net}, $60300\,\text{kW}$ for W_{comp}

$$W_{turb}=W_{comp}+W_{net}$$
$$=60300+\left(60\times10^3\right)$$
$$=120300\,\text{kW}$$

Step 6 of 6

b)

Calculate the isotropic efficiency of the turbine is

$$\eta_{turb}=\frac{W_{turb}}{W_{turb,s}}$$
$$=\frac{120300}{126054.738}$$
$$=0.9543$$
$$=95.43\%$$

Hence, isentropic efficiency of the turbine is $\boxed{95.43\%}$

c)

Calculate the Thermal efficiency of the cycle

$$\eta_{Thur}=\frac{P}{q_{in}}$$
$$=\frac{P}{c_p(T_3-T_2)}$$
$$=\frac{60\times10^3}{200\times1.005(1400-603)}$$
$$=0.374$$

$$=37.4\%$$

Therefore, thermal efficiency of the cycle is $\boxed{37.4\%}$

Question: 9.083P pg 1

A gas-turbine power plant operates on a modified Brayton cycle shown in the figure with an overall pressure ratio of 8- Air enters the compressor at 0°C and 100 kPa. The maximum cycle temperature is 1500 K. The compressor and the turbines are isentropic. The high pressure turbine develops just enough power to run the compressor. Assume constant properties for air at 300 K with cv = 0.718 kJ/kg-K, Cp = 1.005 kJ/kg.K, *R* =0.287 kJ/kg.K, *k* =1.4.

(a)Sketch the *T-s* diagram for the cycle. Label the data states.

(b) Determine the temperature and pressure at state 4, the exit of the high pressure turbine.

(c) If the net power output is 200 MW, determine mass flow rate of the air into the compressor, in kg/s.

Answers: *(b)*1279 K, 457 kPa, (c) 442 kg/s

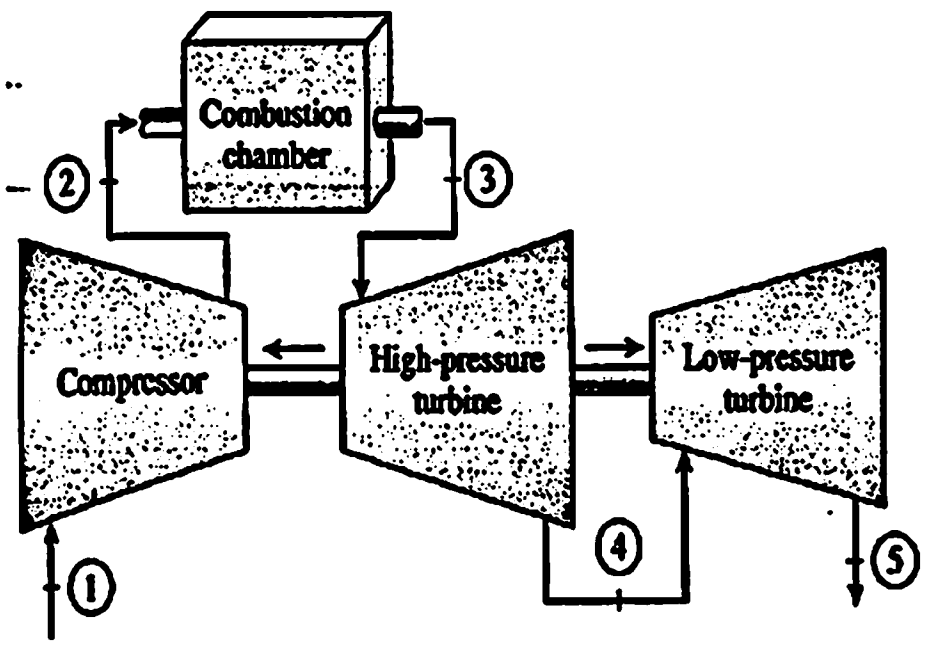


FIGURE P9–83

Step 1 of 7

Overall pressure ratio $r = 8$

Inlet temperature of the air enters into the compressor, $T_1 = 0°C$

Inlet Pressure of the air enters into the compressor, $P_1 = 100\,kPa$

Maximum cycle temperature, $T_3 = 1500\,K$

Properties of the air at 300 K temperature are:

Specific heat of air at constant volume, $c_v = 0.718\,kJ/kg \cdot K$

Specific heat of air at constant pressure, $c_p = 1.005\,kJ/kg \cdot K$

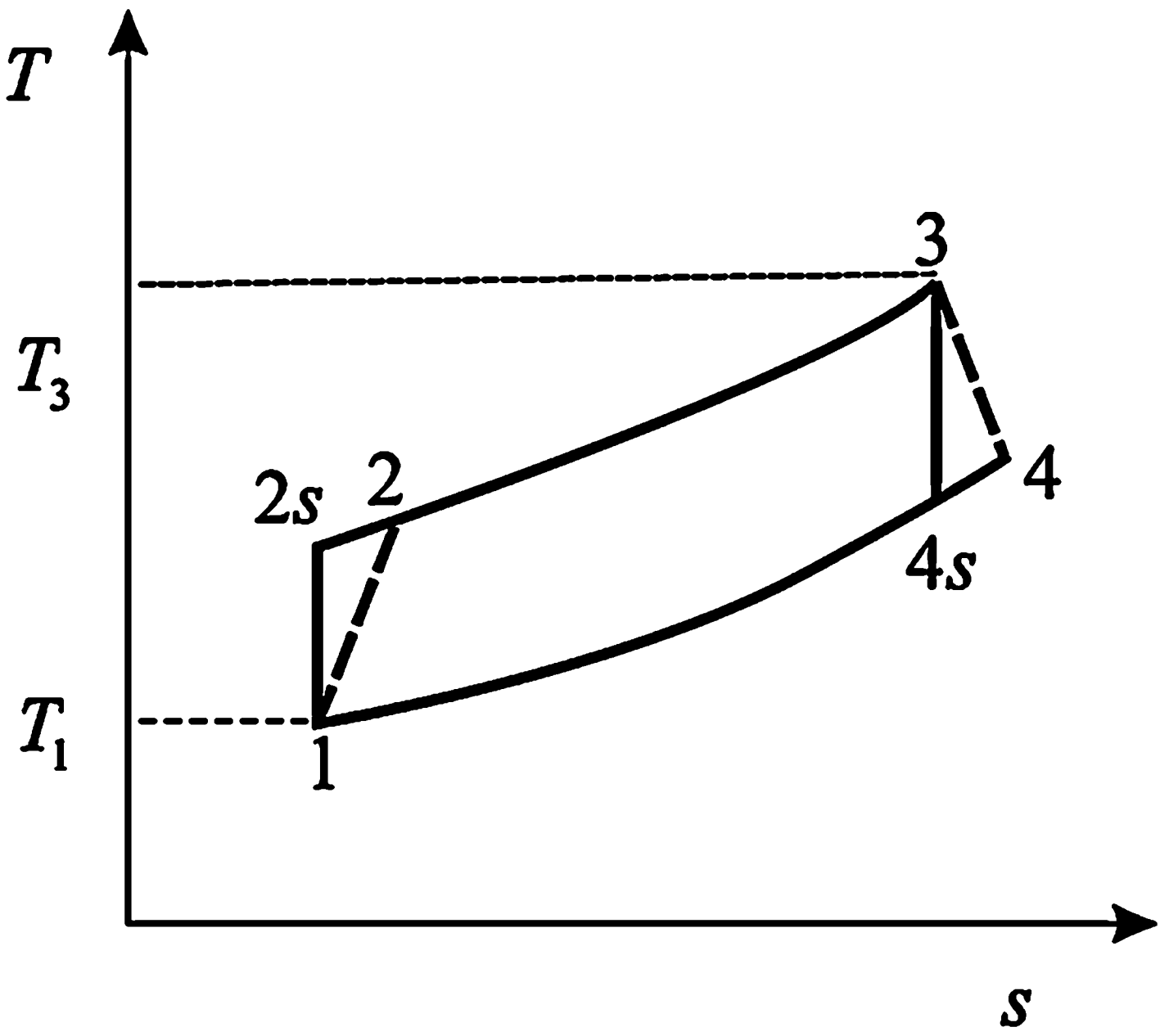
Gas constant, $R = 0.287\,kJ/kg \cdot K$

Specific heat ratio, $k = 1.4$

Step 2 of 7

(a)

The *T-s* diagram is shown below:



Step 3 of 7

(b)

For the process 1-2: isentropic compression

From the relation of isentropic process, we have

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{1}}$$

$$\frac{T_2}{273} = (8)^{\frac{1.4-1}{1}}$$

$$T_2 = 494.819\,K$$

Step 4 of 7

Since turbine and compressor both are mounted on the same shaft, therefore compressor power input to the compressor is equal to turbine output power.

By applying the energy balance equation, we have

$$mc_p(T_2 - T_1) = mc_p(T_3 - T_4)$$

$$T_2 - T_1 = T_3 - T_4$$

$$494.819 - 273 = 1500 - T_4$$

$$T_4 = 1278.181\,K$$

$$mc_p(T_2-T_1)=mc_p(T_3-T_4)$$
$$T_2-T_1=T_3-T_4$$
$$494.819-273=1500-T_4$$
$$T_4=1278.181\,\text{K}$$

Therefore, exit temperature of the high pressure turbine is $\boxed{T_4=1278.181\,\text{K}}$

Step 5 of 7

For the process 3-4: isentropic expansion

From the relation of isentropic process, we have

$$\frac{T_4}{T_3}=\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{1}}$$
$$\frac{1278.181}{1500}=\left(\frac{P_4}{r\times P_1}\right)^{\frac{1.4-1}{1}} \qquad \left(\because r=\frac{P_2}{P_1}=\frac{P_3}{P_4}\right)$$
$$\frac{1278.181}{1500}=\left(\frac{P_4}{8\times 100}\right)^{\frac{1.4-1}{1}}$$
$$P_4=457.179\,\text{kPa}$$

Therefore, exit pressure of the high pressure turbine is $\boxed{P_4=457.179\,\text{kPa}}$.

Step 6 of 7

(c)

For the process 4-5: isentropic expansion

From the relation of isentropic process, we have

$$\frac{T_5}{T_4}=\left(\frac{P_5}{P_4}\right)^{\frac{k-1}{1}}$$
$$\frac{T_5}{1278.181}=\left(\frac{100}{457.179}\right)^{\frac{1.4-1}{1.4}}$$
$$T_5=827.575\,\text{K}$$

Step 7 of 7

Mass flow rate of the air into the compressor is

$$P=mc_p(T_4-T_5)$$
$$200\times 10^3=m\times 1.005(1278.181-827.575)$$
$$m=441.638\,\text{kg/s}$$

Therefore, mass flow rate of the air into the compressor is $\boxed{m=441.638\,\text{kg/s}}$.

Question: 9.084P pg 1

A gas-turbine power plant operating on the simple Brayton has a pressure ratio of 7. Air enters the compressor at 00C and 100 kPa. The maximum cycle temperature is 1500 K. The compressor has an isentropic efficiency of 80 percent, and the turbine has an isentropic efficiency of 90 percent. Assume constant properties for air at 300 K with *cv*= 0.718 kJ/kg.K, *cp* =1.005 kJ/kg.K, *R*= 0.287 kJ/kg-K, *k*=1.4.

(a) Sketch the *T-s* diagram for the cycle.

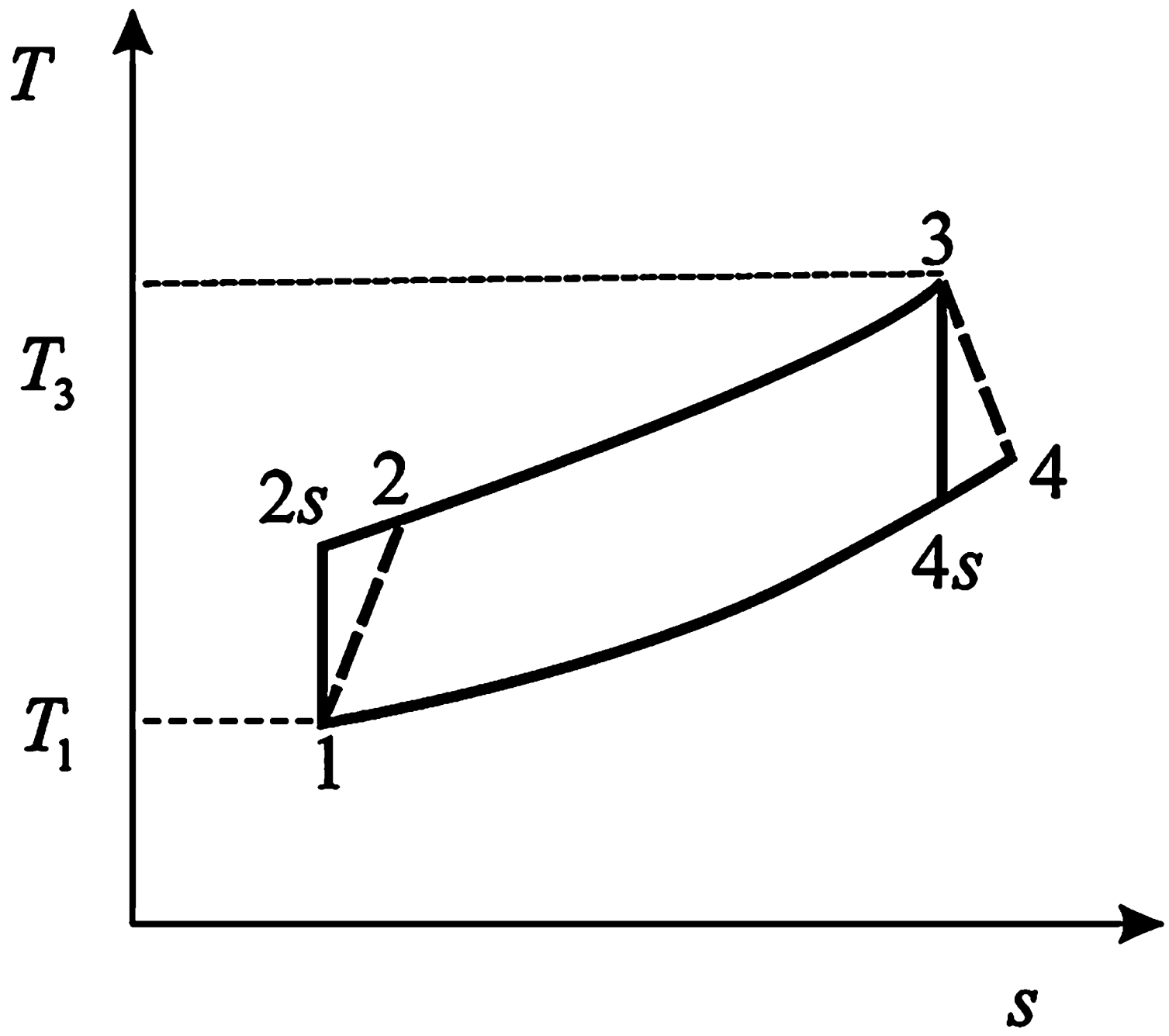
(b) If the net power output is 150 MW, determine the volume flow rate of the air into the compressor, in m3/Vs.

(c) For a fixed compressor inlet velocity and flow area, explain the effect of increasing compressor inlet temperature (i.e., summertime operation versus wintertime operation) on the inlet mass flow rate and the net power output with all other parameters of the problem being the same.

Step 1 of 7

a)

Draw the *T-s* diagram of the simple Brayton cycle:



Process 1-2s is isentropic compression

Process 1-2 is actual compression in the compressor

Process 2-3 is constant pressure heat addition,

Process 3-4s is isentropic expansion

Process 3-4 is actual expansion in the turbine

Process 4s-1 is constant pressure heat rejection.

Step 2 of 7

b)

Use the relation for process 1-2

$$\frac{T_{2s}}{T_1}=\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{1}}$$

Here, T_{2s} is the temperature at state point 2s, T_1 is the temperature at state point 1 and P_2 and P_1 are the pressure at state point 2 and 1 respectively.

Substitute 7 for $\frac{P_2}{P_1}$, 1.4 for k, $273\,\text{K}$ for T_1

$$\frac{T_{2s}}{T_1}=\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{1}}$$
$$\frac{T_{2s}}{273}=(7)^{\frac{1.4-1}{1.4}}$$
$$T_{2s}=476.278\,\text{K}$$

Step 3 of 7

Calculate the temperature at state point 2 by using the relation of isentropic efficiency of the compressor

$$\eta_C=\frac{T_{2s}-T_1}{T_2-T_1}$$

Here, η_C is the isentropic efficiency of the compressor,

Substitute $476.278\,\text{K}$ for T_{2s} and 0.80 for η_c

$$\eta_C=\frac{T_{2s}-T_1}{T_2-T_1}$$
$$0.80=\frac{476.278-273}{T_2-273}$$
$$T_{2s}=527.097\,\text{K}$$

Step 4 of 7

Calculate the temperature at state point 4s by using the relation of isentropic process

$$\frac{T_{4s}}{T_3}=\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

Substitute $1500\,\text{K}$ for T_3, $\frac{1}{7}$ for $\frac{P_4}{P_3}$

$$\frac{T_{4s}}{T_3}=\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$\frac{T_4}{T_3}\left(\frac{P_4}{P_3}\right)$

Substitute $1500\,\text{K}$ for T_3, $\frac{1}{7}$ for $\frac{P_4}{P_3}$

$$\frac{T_{4s}}{T_3}=\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{4s}}{1500}=\left(\frac{1}{7}\right)^{\left(\frac{1.4-1}{1.4}\right)}$$

$$T_{4s}=859.791\,\text{K}$$

Step 5 of 7

Calculate the temperature at state point 2 by using the relation of isentropic efficiency of the turbine

$$\eta_T=\frac{T_3-T_4}{T_3-T_{4s}}$$

Here, η_T is the isentropic efficiency of the turbine,

Substitute $859.791\,\text{K}$ for T_{4s} and 0.90 for η_T

$$\eta_T=\frac{T_3-T_4}{T_3-T_{4s}}$$

$$0.90=\frac{1500-T_4}{1500-859.791}$$

$$T_4=923.811\,\text{K}$$

Calculate the mass flow rate through a turbine by using the net work done in cycle

$$W_{net}=mc_p\left((T_3-T_4)-(T_2-T_1)\right)$$

Here, c_p is the specific heat of the air and W_{net} is the net work done

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p, $923.811\,\text{K}$ for T_3

$$W_{net}=mc_p\left((T_3-T_4)-(T_2-T_1)\right)$$

$$150000=m\times1.005\left((1500-923.811)-(527.097-273)\right)$$

$$m=463.388\,\text{kg/s}$$

Step 6 of 7

Calculate the specific volume by using the ideal gas relation

$$P_1v_1=RT_1$$

$$v_1=\frac{RT_1}{P_1}$$

Here, R is the gas constant

Substitute $0.287\,\text{kJ/kg}\cdot\text{K}$ for R

$$v_1=\frac{RT_1}{P_1}$$

$$=\frac{0.287\times273}{100}$$

$$=0.7835\,\text{m}^3/\text{kg}$$

Calculate the volume flow rate

$$V=mv_1$$

$$=0.78363.35\times463.388$$

$$=363.0644\,\text{m}^3/\text{s}$$

Hence, Volume flow rate is $\boxed{363.0644\,\text{m}^3/\text{s}}$

Step 7 of 7

c)

Effect of increasing compressor inlet temperature

From the ideal gas equation

$$PV=mRT$$

$$P=\rho RT$$

$$T\propto\frac{1}{\rho}$$

From the above we say that temperature is inversely proportional to the density of the gas.

Mass flow rate through a compressor is given as

$$m=\rho AV$$

Given inlet velocity and flow area remains constant.

Hence,

$$m\propto\rho$$

$$m\propto\rho\propto\frac{1}{T}$$

And power input to the compressor is

$$\text{Power}(\text{P})=mc_p\Delta T$$

$$P\propto m\propto\rho\propto\frac{1}{T}$$

From the above two relation we concluded that the effect of increasing the temperature mass flow rate of the air decreases, thereby decreases the power input to the compressor.

Question: 9.085P

How does regeneration affect the efficiency of a Brayton cycle, and how does it accomplish it?

Step 1 of 1

The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. This, in turn, decreases the heat input (thus fuel) requirements for the same net work output.

Question: 9.086P

Somebody claims that at very high pressure ratios, the use of regeneration actually decreases the thermal efficiency of a gas-turbine engine. Is there any truth in this claim? Explain.

Step 1 of 1

The Brayton cycle with regeneration increases the thermal efficiency of the cycle. However the use of regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature. Otherwise heat will flow in the reverse direction (heat addition to the exhaust gases) decreasing the efficiency.
At very high pressure ratios the gas temperature at the turbine exit may be lower than the temperature at the compressor exit which allows heat to flow in the reverse direction (heat addition to the exhaust gases), decreasing the efficiency.

Question: 9.087P

Define the effectiveness of a regenerator used in gas-turbine cycles.

Step 1 of 1

The extent to which a regenerator approaches an ideal regenerator is called the effectiveness and is defined as the ratio of the actual enthalpy increase of the air flowing through the compressor side of the regenerator to the maximum theoretical enthalpy increase. That is

$$\varepsilon = \frac{q_{regen,act}}{q_{regen,max}}$$

Question: 9.088P

In an ideal regenerator, is the air leaving the compressor heated to the temperature at *(a)* turbine inlet, *(b)* turbine exit, (c) slightly above turbine exit?

Step 1 of 1

Regeneration means the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. In an ideal regenerator the air leaving the compressor heated to the temperature at turbine exit temperature.

So the correct option is (b) Turbine exit

Question: 9.089P

In 1903, Aegidius Elling of Norway designed and built an 11-hp gas turbine that used steam injection between the combustion chamber and the turbine to cool the combustion gases to a safe temperature for the materials available at the time. Currently there are several gas-turbine power plants that use steam injection to augment power and improve thermal efficiency. For example, the thermal efficiency of the General Electric LM5000 gas turbine is reported to increase from 35.8 percent in simple-cycle operation to 43 percent when steam injection is used. Explain why steam injection increases the power output and the efficiency of gas turbines. Also, explain how you would obtain the steam.

Step 1 of 1

Steam injection increases the mass flow rate of steam in the turbine and this leads to increase the power output from the turbine.

We know thermal efficiency

$$\eta = \frac{W}{Q_{in}}$$

Since work output increases and heat addition remains constant so thermal efficiency increases. The process of steam injection is like regeneration process. The steam can be obtained from the exhaust gases which are coming from exit of turbine.

Question: 9.090P

A gas turbine for an automobile is designed with a regenerator. Air enters the compressor of this engine at 100 kPa and 20°C. The compressor pressure ratio is 8; the maximum cycle temperature is 800°C. and the cold air stream leaves the regenerator 10°C cooler than the hot air stream at the inlet of the regenerator. Assuming both the compressor and the turbine to be isentropic, determine the rates of heat addition and rejection for this cycle when it produces 150 kW. Use constant specific heats at room temperature.

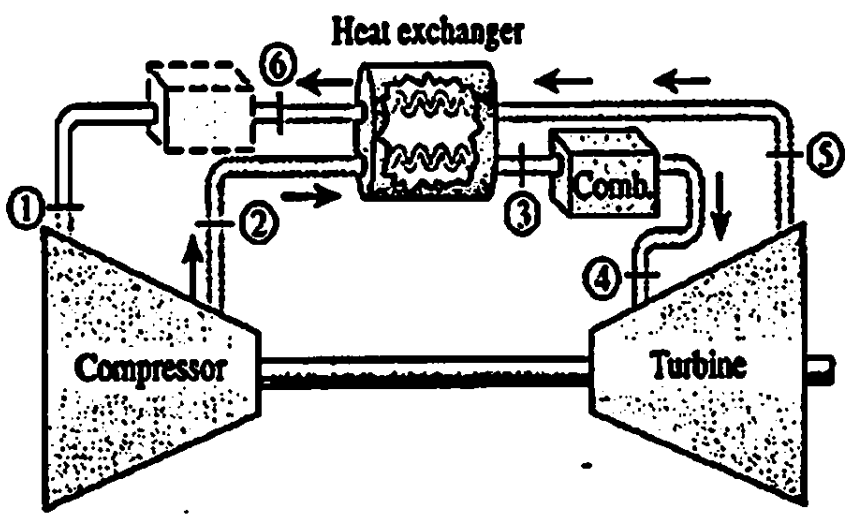


FIGURE P9-90

Step 1 of 10

Given

Compressor pressure ratio $r_p = 8$

Compressor inlet pressure $P_1 = 100\ \text{kPa}$

Compressor inlet temperature $T_1 = 20°C + 273 = 293K$

Maximum cycle temperature $T_4 = 800°C + 273 = 1073K$

Power produced $\dot{W}_{net} = 150\ \text{kW}$

The regeneration temperature $T_3 = T_5 - 10$

Step 2 of 10

Considering constant specific heats at room temperature

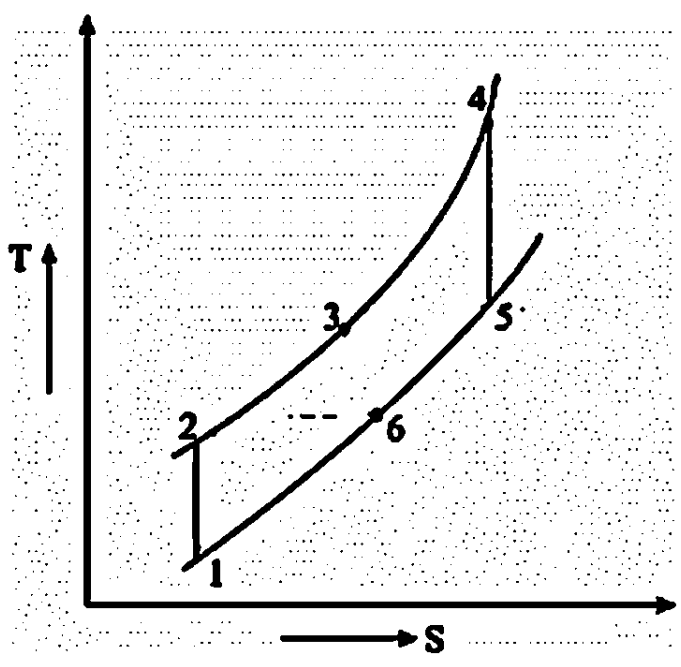
Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 10



Step 4 of 10

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_2}{293} = (8)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 530.75K$$

Step 5 of 10

Similarly process 4-5 is isentropic expansion

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{k-1}{k}}$$

$$\frac{T_5}{T_4} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

$$\frac{T_5}{1073} = \left(\frac{1}{8}\right)^{\frac{1.4-1}{1.4}}$$

$$T_5 = 592.344K$$

Step 6 of 10

Temperature at state 3

$$T_3 = T_5 - 10$$

$$T_3 = 592.344 - 10$$

$$T_3 = 582.344K$$

The temperature of the gas after regeneration is determined by using energy balance to the regenerator

$$T_5 - T_6 = T_3 - T_2$$

$$T_6 = T_5 - (T_3 - T_2)$$

$$T_6 = 592.344 - (582.344 - 530.75)$$

$$T_6 = 540.75K$$

Step 7 of 10

Net work output of the cycle is expressed as

$$w_{net} = w_t - w_c$$

$$w_{net} = c_p(T_4 - T_5) - c_p(T_2 - T_1)$$

$$= (1.005)\{(1073 - 592.344) - (530.75 - 293)\}$$

$$w_{net} = 244.12\ \text{kJ/kg}$$

Step 8 of 10

Power produced

$$\dot{W}_{net} = \dot{m} w_{net}$$

$$150 = \dot{m} \times 244.12$$

$$\dot{m} = 0.6145\text{kg/sec}$$

Step 9 of 10

The rate of heat addition

$$\dot{Q}_{in} = \dot{m} c_p (T_4 - T_3)$$

$$= (0.6145)(1.005)(1073 - 582.344)$$

$$\boxed{\dot{Q}_{in} = 303.015\ \text{kW}}$$

Step 10 of 10

The rate of heat rejection

$$\dot{Q}_{out} = \dot{m} c_p (T_6 - T_1)$$

$$= (0.6145)(1.005)(540.75 - 293)$$

$$\boxed{\dot{Q}_{out} = 153\ \text{kW}}$$

Question: 9.091P

Rework Prob. 9–90 when the compressor isentropic efficiency is 87 percent and the turbine isentropic efficiency is 93 percent.

Step 1 of 13

Given

Compressor pressure ratio $r_p = 8$

Compressor inlet pressure $P_1 = 100\ \text{kPa}$

Compressor inlet temperature $T_1 = 20°\text{C} + 273 = 293\text{K}$

Maximum cycle temperature $T_4 = 800°\text{C} + 273 = 1073\text{K}$

The isentropic efficiency of the turbine $\eta_t = 93\%$

The isentropic efficiency of the compressor $\eta_c = 87\%$

Power produced $\dot{W}_{net} = 150\ \text{kW}$

The regeneration temperature $T_3 = T_5 - 10$

Step 2 of 13

Considering constant specific heats at room temperature

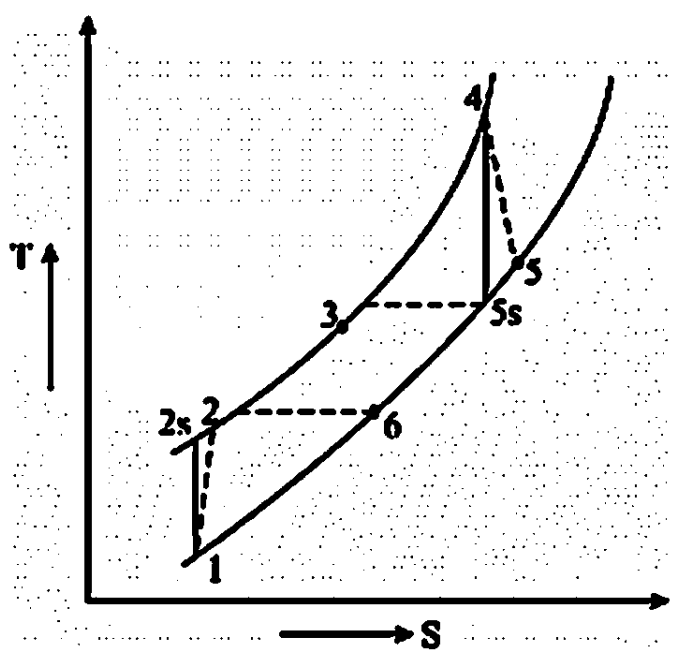
Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4$

Step 3 of 13



Step 4 of 13

Process 1-2s is isentropic compression

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{2s}}{293} = (8)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 530.75\text{K}$$

Step 5 of 13

Since the isentropic efficiency of the compression is given

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$0.87 = \frac{530.75 - 293}{T_2 - 293}$$

$$T_2 = 566.275\text{K}$$

Step 6 of 13

Similarly process 4-5s is isentropic expansion

$$\frac{T_{5s}}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{5s}}{T_4} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

$$\frac{T_{5s}}{1073} = \left(\frac{1}{8}\right)^{\frac{1.4-1}{1.4}}$$

$$T_{5s} = 592.344\text{K}$$

Step 7 of 13

Turbine isentropic efficiency is given by

$$\eta_T = \frac{T_4 - T_5}{T_4 - T_{5s}}$$

$$0.93 = \frac{1073 - T_5}{1073 - 592.344}$$

$$T_5 = 625.98\text{K}$$

Step 8 of 13

Temperature at state 3

$$T_3 = T_5 - 10$$

$$T_3 = 625.98 - 10$$

$$T_3 = 615.98\text{K}$$

Step 9 of 13

The temperature of the gas after regeneration is determined by using energy balance to the regenerator

$$T_5 - T_6 = T_3 - T_2$$

$$T_6 = T_5 - (T_3 - T_2)$$

$$T_6 = 625.98 - (615.98 - 566.275)$$

$$T_6 = 576.275\text{K}$$

Step 10 of 13

Net work output of the cycle is expressed as

$$w_{net} = w_t - w_c$$

$$w_{net} = c_p(T_4 - T_5) - c_p(T_2 - T_1)$$

$$= (1.005)\{(1073 - 625.98) - (566.275 - 293)\}$$

$$w_{net} = 174.613\ \text{kJ/kg}$$

Step 11 of 13

Power produced

$$\dot{W}_{net} = \dot{m} \times w_{net}$$

$$150 = \dot{m} \times 174.613$$

$$\dot{m} = 0.859\ \text{kg/s}$$

Step 12 of 13

The rate of heat addition

$$\dot{Q}_{in} = \dot{m} c_p (T_4 - T_3)$$

$$= (0.859)(1.005)(1073 - 615.98)$$

$$\boxed{\dot{Q}_{in} = 394.543\ \text{kW}}$$

Step 13 of 13

The rate of heat rejection

$$\dot{Q}_{out} = \dot{m} c_p (T_6 - T_1)$$

$$= (0.859)(1.005)(576.275 - 293)$$

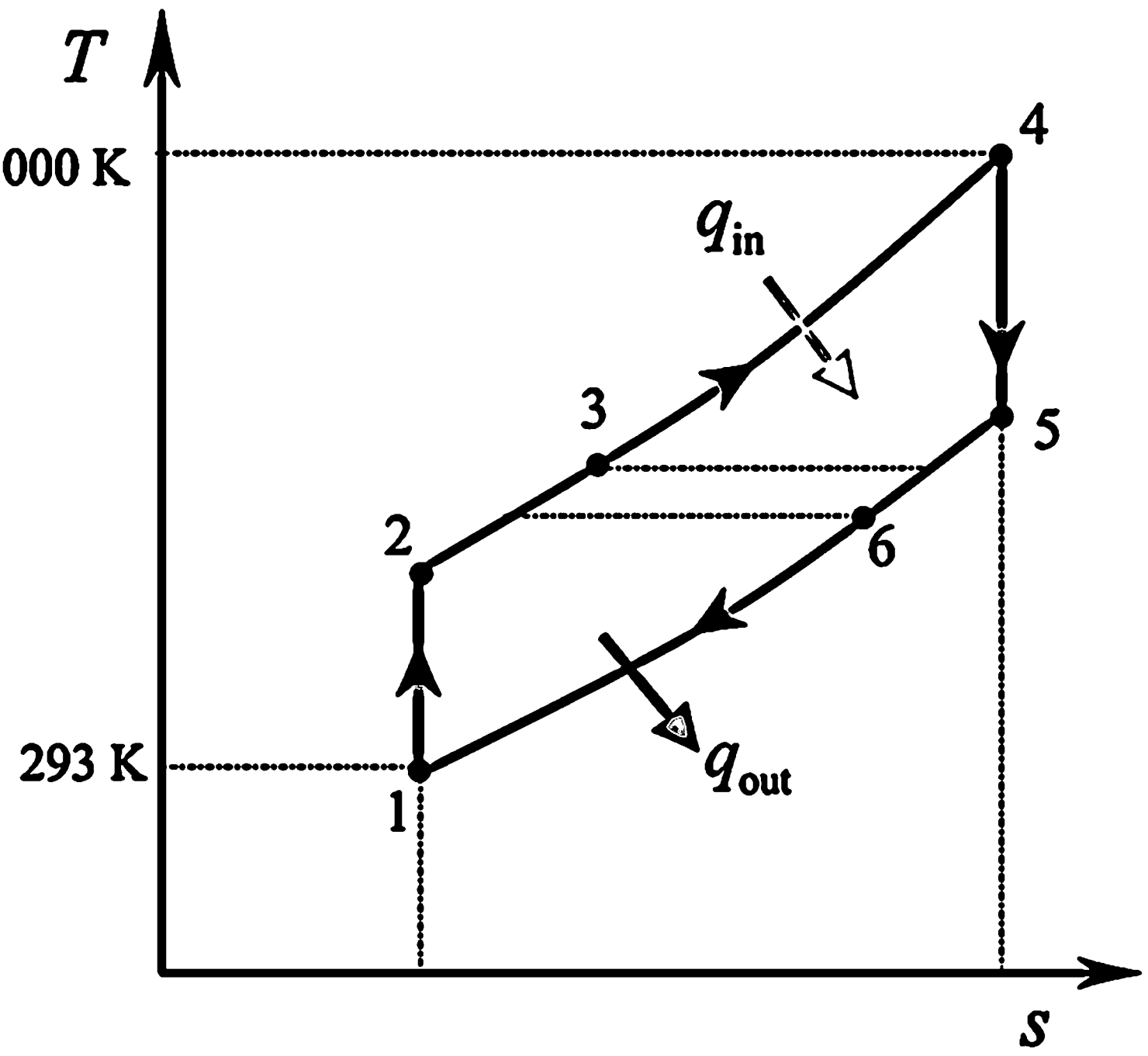
$$\boxed{\dot{Q}_{out} = 244.549\ \text{kW}}$$

Question: 9.092P pg 1

A gas turbine engine operates on the ideal Brayton cycle with regeneration, as shown in Fig. P9-90. Now the regenerator is rearranged so that the air streams of states 2 and 5 enter at one end of the regenerator and streams 3 and 6 exit at the other end (i.e., parallel flow arrangement of a heat exchanger). Consider such a system when air enters the compressor at 100 kPa and 20°C; the compressor pressure ratio is 7; the maximum cycle temperature is 727°C; and the difference between the hot and cold air stream temperatures is 6°C at the end of the regenerator where the cold stream leaves the regenerator. Is the cycle arrangement shown in the figure more or less efficient than this arrangement? Assume both the compressor and the turbine are isentropic, and use constant specific heats at room temperature.

Step 1 of 11

Draw the property curve between T and s for an Ideal Brayton cycle with regeneration as follows:



Step 2 of 11

Obtain the properties of air from the **Table A-2a**.

At room temperature,

Gas constant, $R = 0.287\,\text{kJ/kg}\cdot\text{K}$

Specific heats at constant pressure, $c_p = 1.005\,\text{kJ/kg}\cdot\text{K}$

Specific heats at constant volume, $c_v = 0.718\,\text{kJ/kg}\cdot\text{K}$

Ratio of specific heats (or) Specific heat ratio, $k = 1.4$

Step 3 of 11

Consider the process $1-2$ **(Isentropic Compression process)**:

Calculate the temperature at the end of compression in an isentropic compression process by using the following equation,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}$$

$$T_2 = T_1\left(r_p\right)^{\left(\frac{k-1}{k}\right)}$$

Here, the temperature of air before compression is T_1, the pressure of air before compression is P_1, the temperature of air at the end of compression is T_2, the pressure of air after compression is P_2, the compressor pressure ratio is r_p and the specific heat ratio is k.

Substitute $293\,\text{K}$ for T_1, 7 for r_p, and 1.4 for k.

$$\begin{aligned} T_2 &= T_1\left(r_p\right)^{\left(\frac{k-1}{k}\right)} \\ &= 293(7)^{\left(\frac{1.4-1}{1.4}\right)} \\ &= 510.88\,\text{K} \end{aligned}$$

Step 4 of 11

Consider the process $4-5$ **(Isentropic Expansion process)**:

Calculate the temperature at the end of expansion in an isentropic expansion process by using the following equation,

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\left(\frac{k-1}{k}\right)}$$

Substitute P_1 for P_5 and P_2 for P_4.

$$\frac{T_5}{T_4} = \left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{\left(\frac{k-1}{k}\right)}$$

Here, the temperature of air at the end of expansion is T_5, the pressure of air at the end of expansion is P_5, the temperature of air before expansion is T_4, and the pressure of air before expansion is P_4.

Substitute $1000\,\text{K}$ for T_4, 7 for r_p, and 1.4 for k.

$$T_5 = 1000\left(\frac{1}{\quad}\right)^{\left(\frac{1.4-1}{1.4}\right)}$$

the temperature of air before expansion is T_4, and the pressure of air before expansion is P_4.

Substitute $1000\ \text{K}$ for T_4, 7 for r_p, and 1.4 for k.

$$T_5 = 1000\left(\frac{1}{7}\right)^{\left(\frac{1.4-1}{1.4}\right)}$$
$$= 573.51\ \text{K}$$

Step 5 of 11

Write the relation between the hot and the cold air stream temperatures at the outlet of the regenerator as follows:

$$T_5 - T_3 = 6$$

Here, the temperature of hot air entering the heat addition process is T_3.

Substitute $573.51\ \text{K}$ for T_5.

$$573.51 - T_3 = 6$$
$$T_3 = 567.51\ \text{K}$$

Step 6 of 11

Apply the energy balance equation for the regenerator.

$$\begin{pmatrix}\text{Heat gained by the cold air}\\ \text{during the process 2-3}\end{pmatrix} = \begin{pmatrix}\text{Heat lost by the hot air}\\ \text{during the process 5-6}\end{pmatrix}$$
$$mc_p(T_3 - T_2) = mc_p(T_5 - T_6)$$
$$(T_3 - T_2) = (T_5 - T_6)$$

Here, the temperature of cold air at the outlet of the regenerator is T_6.

Substitute $510.88\ \text{K}$ for T_2, $567.51\ \text{K}$ for T_3 and $573.51\ \text{K}$ for T_5.

$$(567.51 - 510.88) = (573.51 - T_6)$$
$$T_6 = 516.88\ \text{K}$$

Step 7 of 11

Calculate the thermal efficiency of the cycle by using the following equation,

$$\eta_{th} = \frac{q_{in} - q_{out}}{q_{in}}$$
$$= 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{c_p(T_6 - T_1)}{c_p(T_4 - T_3)}$$
$$= 1 - \frac{(T_6 - T_1)}{(T_4 - T_3)}$$

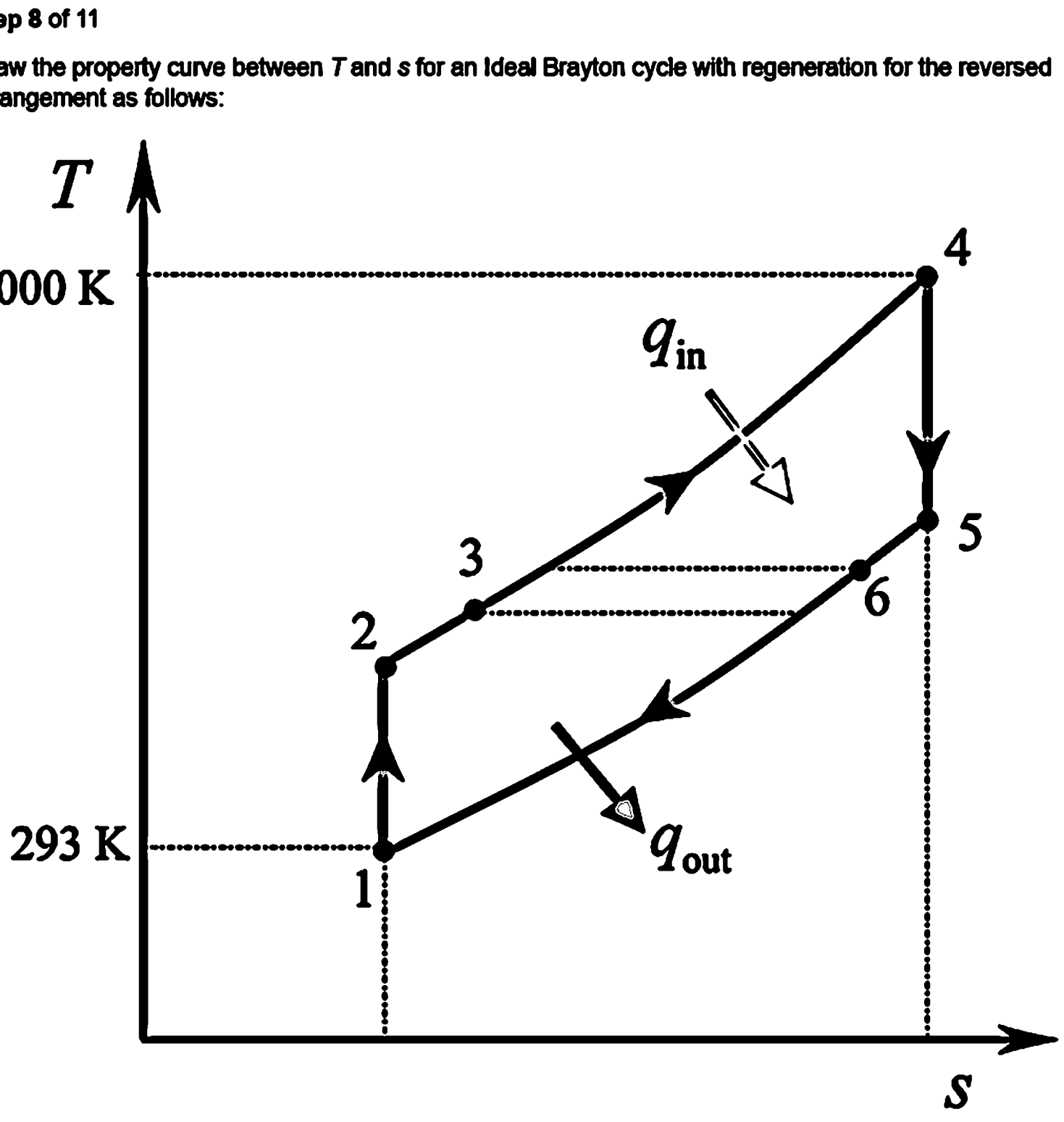
Substitute $516.88\ \text{K}$ for T_6, $567.51\ \text{K}$ for T_3, $293\ \text{K}$ for T_1, and $1000\ \text{K}$ for T_4.

$$\eta_{th} = 1 - \frac{(516.88 - 293)}{(1000 - 567.51)}$$
$$= 1 - \frac{223.88}{432.49}$$
$$= 0.4823$$
$$= 48.23\%$$

Therefore, the thermal efficiency of the cycle is 48.23%.

Step 8 of 11

Draw the property curve between *T* and *s* for an Ideal Brayton cycle with regeneration for the reversed arrangement as follows:



Step 9 of 11

Write the relation between the hot and the cold air stream temperatures at the outlet of the regenerator as follows:

$$T_6 - T_3 = 6$$
$$T_6 = T_3 + 6 \quad \text{..... (1)}$$

Here, the temperature of hot air entering the heat addition process is T_3 and the temperature of cold air at

follows:

$$T_6 - T_3 = 6$$
$$T_6 = T_3 + 6 \quad \text{..... (1)}$$

Here, the temperature of hot air entering the heat addition process is T_3 and the temperature of cold air at the outlet of the regenerator is T_6.

Apply the energy balance equation for the regenerator.

$$\begin{pmatrix}\text{Heat gained by the cold air}\\ \text{during the process 2-3}\end{pmatrix} = \begin{pmatrix}\text{Heat lost by the hot air}\\ \text{during the process 5-6}\end{pmatrix}$$
$$mc_p(T_3 - T_2) = mc_p(T_5 - T_6)$$
$$(T_3 - T_2) = (T_5 - T_6)$$

Here, the temperature of cold air at the outlet of the regenerator is T_6.

Substitute $510.88\ \text{K}$ for T_2, and $573.51\ \text{K}$ for T_5.

$$(T_3 - 510.88) = (573.51 - T_6)$$
$$T_3 + T_6 = 1084.39\ \text{K} \quad \text{..... (2)}$$

Step 10 of 11

Substitute equation (1) in (2).

$$T_3 + (T_3 + 6) = 1084.39\ \text{K}$$
$$2T_3 = 1078.39$$
$$T_3 = 539.195\ \text{K}$$

Substitute $539.195\ \text{K}$ for T_3 in equation (1).

$$T_6 = T_3 + 6$$
$$= 539.195 + 6$$
$$= 545.195\ \text{K}$$

Step 11 of 11

Calculate the thermal efficiency of the cycle for the reversed arrangement by using the following equation,

$$\eta_{th} = \frac{q_{in} - q_{out}}{q_{in}}$$
$$= 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{c_p(T_6 - T_1)}{c_p(T_4 - T_3)}$$
$$= 1 - \frac{(T_6 - T_1)}{(T_4 - T_3)}$$

Substitute $545.195\ \text{K}$ for T_6, $539.195\ \text{K}$ for T_3, $293\ \text{K}$ for T_1, and $1000\ \text{K}$ for T_4.

$$\eta_{th} = 1 - \frac{(545.195 - 293)}{(1000 - 539.195)}$$
$$= 1 - \frac{252.195}{460.805}$$
$$= 0.4527$$
$$= 45.27\%$$

Therefore, the thermal efficiency of the cycle for the reversed arrangement is 45.27%.

Therefore, from the obtained values of the thermal efficiencies for both the arrangements, it is clear that, the cycle arrangement shown in the figure P9-111 is more efficient than the reversed arrangement.

Question: 9.093P pg 1

The 7FA gas turbine manufactured by General W Electric is reported to have an efficiency of 35.9 percent in the simple-cycle mode and to produce 159 MW of net power. The pressure ratio is 14.7 and the turbine inlet temperature is 1288°C The mass flow rate through the turbine is 1,536,000 kg/h. Taking the ambient conditions to be 30°C and 100 kPa, determine the isentropic efficiency of the turbine and the compressor. Also, determine the thermal efficiency of this gas turbine if a regenerator with an effectiveness of 65 percent is added.

Step 1 of 12

Efficiency of the gas turbine, $\eta_{th} = 35.9\% = 0.359$

Net power $P_{net} = 159\,\text{MW}$

Pressure ratio, $r = 14.7$

Turbine inlet temperature, $T_3 = 1288\,^{\circ}\text{C}$

Mass flow rate through the turbine, $m = 1536000\,\text{kg/h}$

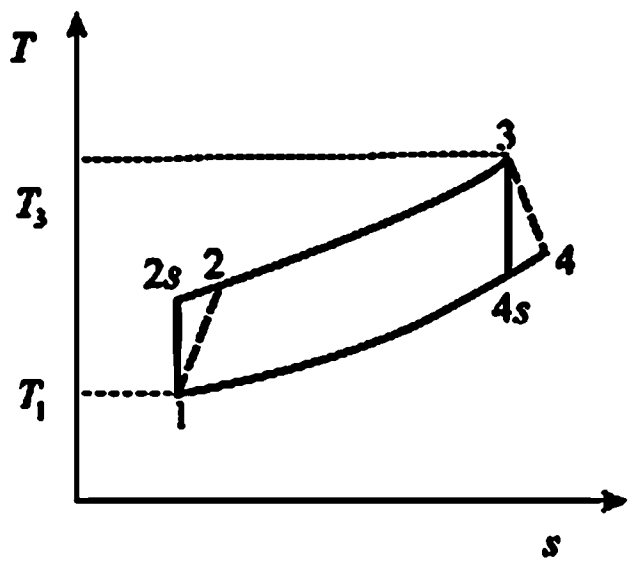
Ambient temperature $T_1 = 30^{\circ}\text{C}=303\text{K}$

Inlet pressure $P_1 = 100\,\text{kPa}$

Effectiveness of the regenerator, $\varepsilon = 0.65$

Step 2 of 12

Ts diagram:



Step 3 of 12

From the properties of the ideal gas air,

At $T_1 = 303\text{K}$

By using the method of interpolation

$$h_1 = 300.19 + \frac{(303-300)(305.22-300.19)}{(305-300)}$$

$$= 303.21\,\text{kJ/kg}$$

Pressure ratio is

$$\text{Pr}_1 = 1.3860 + \frac{(303-300)(1.4686-1.3860)}{(305-300)}$$

$$= 1.4356$$

Step 4 of 12

From the relation of isentropic process, we have

$$\frac{P_2}{P_1} = \left(\frac{\text{Pr}_2}{\text{P}_{r1}}\right)$$

$$14.7 \times 1.4356 = \text{P}_{r2}$$

$$\text{P}_{r2} = 21.10$$

From the properties of the ideal gas air,

At $\text{P}_{r2} = 21.10$

By using the method of interpolation

$$h_{2s} = 649.22 + \frac{(21.10-20.64)(659.84-649.22)}{(21.86-20.64)}$$

$$= 653.25\,\text{kJ/kg}$$

Step 5 of 12

From the properties of the ideal gas air,

At $T_3 = 1561\text{K}$

By using the method of interpolation

$$h_3 = 1708.82 + \frac{(1561-1560)(1733.17-1708.82)}{(1580-1560)}$$

$$= 1710\,\text{kJ/kg}$$

Pressure ratio is

$$\text{P}_{r3} = 710.5 + \frac{(1561-1560)(750-710.5)}{(1580-1560)}$$

$$= 712.5$$

Step 6 of 12

From the relation of isentropic process, we have

$$\frac{P_4}{P_3} = \left(\frac{\text{Pr}_4}{\text{P}_{r3}}\right)$$

$$\frac{1}{14.7} \times 712.5 = \text{P}_{r4}$$

$$\text{P}_{r2} = 48.47$$

From the properties of the ideal gas air,

At $\text{P}_{r4} = 48.47$

By using the method of interpolation

$$h_{4s} = 821.95 + \frac{(48.47-47.75)(843.98-821.95)}{(52.59-47.75)}$$

$$= 825.23\,\text{kJ/kg}$$

Step 7 of 12

Net work done per unit mass is

$$W_{net} = \frac{P_{net}}{m}$$

$$= \frac{159000\,\text{kW}}{} \left(\frac{3600\,\text{sec}}{}\right)$$

Net work done per unit mass is

$$W_{net} = \frac{P_{net}}{m}$$
$$= \frac{159000\,\text{kW}}{1536000\,\text{kg/h}}\left(\frac{3600\,\text{sec}}{1\text{h}}\right)$$
$$= 372.66\,\text{kJ/kg}$$

Heat input is

$$q_{in} = \frac{w_{net}}{\eta_{th}}$$
$$= \frac{372.66}{0.359}$$
$$= 1038\,\text{kJ/kg}$$

Step 8 of 12

Enthalpy at state 2 is

$$q_{in} = h_3 - h_2$$
$$h_2 = 1710 - 1038$$
$$= 672\,\text{kJ/kg}$$

Heat output of the turbine is

$$W_{net} = q_{in} - q_{out}$$
$$q_{out} = q_{in} - W_{net}$$
$$= 1038 - 372.66$$
$$= 665.34\,\text{kJ/kg}$$

And

$$q_{out} = h_4 - h_1$$
$$665.34 = h_4 - 303.21$$
$$h_4 = 968.55\,\text{kJ/kg}$$

Step 9 of 12

Then, isentropic efficiency of the turbine is

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$
$$= \frac{1710 - 968.55}{1710 - 825.33}$$
$$= 0.838$$
$$= 83.8\%$$

Therefore, isentropic turbine efficiency is $\boxed{\eta_T = 83.8\%}$

Step 10 of 12

Then, isentropic efficiency of the compressor is

$$\eta_c = \frac{h_1 - h_{2s}}{h_1 - h_2}$$
$$= \frac{303.21 - 653.25}{303.21 - 672}$$
$$= 0.9491$$
$$= 94.91\%$$

Therefore, isentropic compressor efficiency is $\boxed{\eta_C = 94.91\%}$

Step 11 of 12

By using the regenerator, the new heat input becomes

$$q_{gen} = \varepsilon(h_4 - h_2)$$
$$= 0.65(968.55 - 672)$$
$$= 192.80\,\text{kJ/kg}$$

The new heat input becomes

$$q_{in,reg} = q_{in} - q_{gen}$$
$$= 1038 - 192.8$$
$$= 845.20\,\text{kJ/kg}$$

Step 12 of 12

Then, thermal efficiency of the cycle is

$$\eta_{T,gen} = \frac{W_{net}}{q_{in}}$$
$$= \frac{372.66}{845.20}$$
$$= 0.4409$$
$$= 44.09\%$$

Therefore, new thermal efficiency due to regenerator is $\boxed{\eta_{T,gen} = 44.09\%}$

Question: 9.094P



Reconsider Prob. 9–93. Using EES (or other) software, develop a solution that allows different isentropic efficiencies for the compressor and turbine and study the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle. Plot the *T*-*s* diagram for the cycle.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question:

A Brayton cycle with regeneration using air as the working fluid has a pressure ratio of 7. The minimum and maximum temperatures in the cycle are 310 and 1150 K. Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine and an effectiveness of 65 percent for the regenerator, determine *(a)* the air temperature at the turbine exit, *(b)* the net work output, and (c) the thermal efficiency.
Answers: *(a)*783 K, *(b)*108.1 kJ/kg, (c) 22.5 Dercent

Step 1 of 13

Given

Pressure ratio $r_p = 7$

Minimum temperature $T_1 = 310\text{K}$

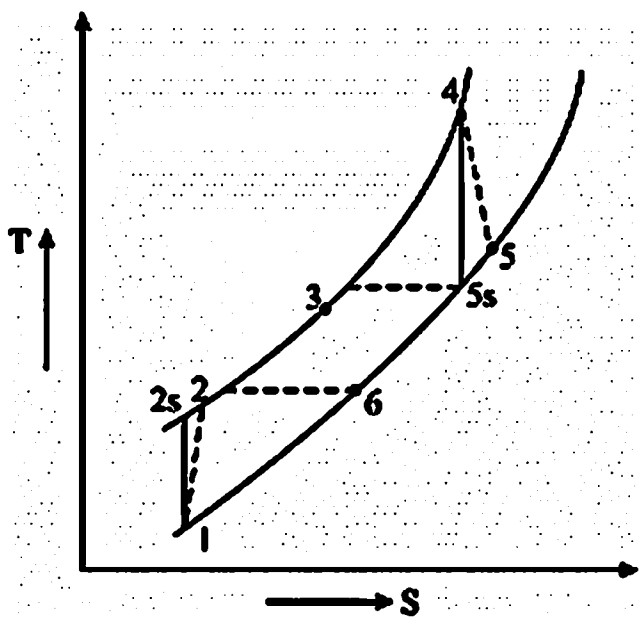
Maximum temperature $T_4 = 1150\text{K}$

Isentropic efficiency of the turbine $\eta_t = 82\%$

Isentropic efficiency of the compressor $\eta_c = 75\%$

The regenerator effectiveness $\varepsilon = 65\%$

Step 2 of 13



Step 3 of 13

At $T_1 = 310\text{K}$ from ideal gas properties of air

$h_1 = 310.24\text{kJ/kg}$

$P_{r_1} = 1.5546$

Process 1-2s is isentropic compression

$$\frac{P_{r_{2s}}}{P_{r_1}} = \left(\frac{P_2}{P_1}\right)$$

$$P_{r_{2s}} = P_{r_1}\left(\frac{P_2}{P_1}\right)$$

$$P_{r_{2s}} = 1.5546 \times 7$$

$$P_{r_{2s}} = 10.8822$$

At $P_{r_{2s}} = 10.8822$ from ideal gas properties of air

$$h_{2s} = 533.98 + (544.35 - 533.98)\left(\frac{10.8822 - 10.37}{11.10 - 10.37}\right)$$

$$h_{2s} = 541.256\text{kJ/kg}$$

Step 4 of 13

Isentropic efficiency of the compression is given by

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.75 = \frac{541.256 - 310.24}{h_2 - 310.24}$$

$$h_2 = 618.261\text{kJ/kg}$$

Step 5 of 13

At $T_4 = 1150\text{K}$ from ideal gas properties of air

$$h_4 = 1207.57 + (1230.92 - 1207.57)\left(\frac{1150 - 1140}{1160 - 1140}\right)$$

$$h_4 = 1219.245\text{kJ/kg}$$

$$P_{r_4} = 193.1 + (207.2 - 193.1)\left(\frac{1150 - 1140}{1160 - 1140}\right)$$

$$P_{r_4} = 200.15$$

Step 6 of 13

Process 4-5s is isentropic expansion

$$\frac{P_{r_4}}{P_{r_{5s}}} = \frac{P_4}{P_5}$$

$$\frac{P_{r_4}}{P_{r_{5s}}} = \frac{P_2}{P_1}$$

$$\frac{200.15}{P_{r_{5s}}} = 7$$

$$P_{r_{5s}} = 28.592$$

Step 7 of 13

At $P_{r_{5s}} = 28.592$ from ideal gas properties of air

$$h_{5s} = 702.52 + (713.27 - 702.52)\left(\frac{28.592 - 27.29}{28.80 - 27.29}\right)$$

$$h_{5s} = 711.789\text{kJ/kg}$$

Step 8 of 13

Since the isentropic efficiency of the turbine is given by

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}}$$

Step 8 of 13

Since the isentropic efficiency of the turbine is given by

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}}$$

$$0.82 = \frac{1219.245 - h_5}{1219.245 - 711.789}$$

$$h_5 = 803.131 \text{kJ/kg}$$

Step 9 of 13

(a)The exit temperature of air from turbine

At $h_5 = 803.131$kJ/kg from ideal gas properties of air

$$T_5 = 780 + (800 - 780)\left(\frac{803.13 - 800.03}{821.95 - 800.03}\right)$$

$$\boxed{T_5 = 782.82\text{K}}$$

Step 10 of 13

(b) The net work output

We know the regenerator effectiveness

$$\varepsilon = \frac{q_{reg,act}}{q_{reg,max}}$$

$$\varepsilon = \frac{h_3 - h_2}{h_5 - h_2}$$

$$0.65 = \frac{h_3 - 618.26}{803.13 - 618.26}$$

$$h_3 = 738.425 \text{kJ/kg}$$

Step 11 of 13

The net work out of the cycle is expressed as

$$w_{net} = w_t - w_c$$

$$w_{net} = (h_4 - h_5) - (h_2 - h_1)$$

$$= \{(1219.245 - 803.131) - (618.26 - 310.24)\}$$

$$\boxed{w_{net} = 108.094 \text{ kJ/kg}}$$

Step 12 of 13

(c) The heat supplied is expressed as

$$q_{in} = (h_4 - h_3)$$

$$= (1219.245 - 738.425)$$

$$q_{in} = 480.82 \text{ kJ/kg}$$

Step 13 of 13

The thermal efficiency is defined as

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{108.094}{480.82}$$

$$= 0.2248$$

$$\boxed{\eta_{th} = 22.48\%}$$

Question: 9.096P pg 1

A stationary gas-turbine power plant operates on an ideal regenerative Brayton cycle (ϵ = 100 percent) with air as the working fluid. Air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air from an external source at a rate of 75,000 kJ/s. Determine the power delivered by this plant (a) assuming constant specific heats for air at room temperature and *(b)* accounting for the variation of specific heats with temperature.

Step 1 of 9

(a)

Obtain the values of constant specific heats at room temperature.

Gas constant $R = 0.287\ \text{kJ/kg.K}$

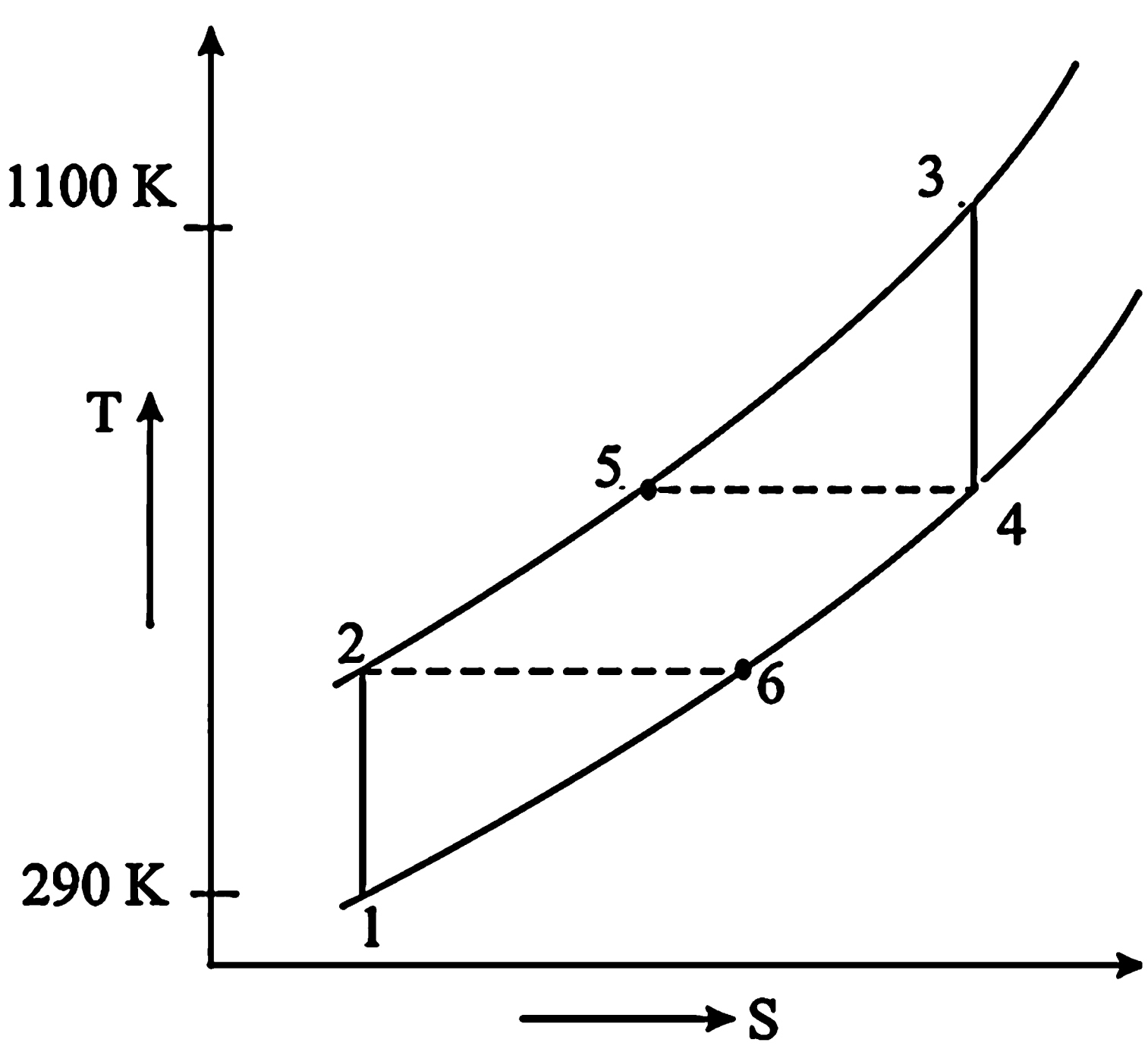
Specific heat at constant pressure, $c_p = 1.005\ \text{kJ/kg.K}$

Specific heat at constant volume, $c_v = 0.718\ \text{kJ/kg.K}$

Ratio of specific heats, $k = 1.4$

Step 2 of 9

Draw the T-s diagram for the Brayton cycle.



Step 3 of 9

For the Process 1-2(isentropic compression.):

Calculate the temperature at state 2 using the relation:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Here, compressor inlet pressure is P_1, temperature is T_1, and exit pressure is P_2.

Substitute $760\,\text{kPa}$ for P_2, $95\,\text{kPa}$ for P_1, 290 K for T_1.

$$\frac{T_2}{290} = \left(\frac{760}{95}\right)^{\frac{1.4-1}{1.4}}$$

$$T_2 = 525.319\text{K}$$

Step 4 of 9

Similarly for process 3-4 (isentropic expansion):

Calculate the temperature at state 4, using the relation:

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

Here, turbine exit pressure is P_4, temperature is T_4, turbine inlet temperature is T_3 and pressure P_3.

Substitute $760\,\text{kPa}$ for P_3, $95\,\text{kPa}$ for P_4, and 1100 K for T_3.

$$\frac{T_4}{1100} = \left(\frac{95}{760}\right)^{\frac{1.4-1}{1.4}}$$

$$T_4 = 607.249\text{K}$$

Step 5 of 9

For 100% regenerator effectiveness,

Take $T_5 = T_4$ also $h_5 = h_4$.

$T_6 = T_2$ also $h_6 = h_2$.

Calculate the power delivered by the gas turbine.

$$\eta = \frac{\dot{W}}{\dot{Q}_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$\frac{\dot{W}}{\dot{Q}} = 1 - \frac{c_p(T_2 - T_1)}{c_p(T_3 - T_4)}$$

$$\frac{\dot{W}}{\dot{Q}} = 1 - \frac{c_p(T_2 - T_1)}{c_p(T_3 - T_4)}$$

Substitute $1.005\ \text{kJ/kg.K}$ for c_p, 525.319 K for T_2, 290K for T_1, 1100K for T_3, and 607.249K for T_4.

$$\frac{\dot{W}}{75,000} = 1 - \frac{(525.319 - 290)}{(1100 - 607.249)}$$

$$\frac{\dot{W}}{75,000} = 0.52243$$

$$\dot{W} = 39182.25\text{kW}$$

Therefore the power delivered by the plant is $\boxed{39.18\ \text{MW}}$.

(b)

Obtain the properties of air from, "TABLE-17 Ideal-gas properties of air", in the text book

by using variable specific heats

At temperature, $T_1 = 290\text{K}$.

Specific enthalpy, $h_1 = 290.16\text{kJ/kg}$

Relative pressure, $P_{r1} = 1.2311$

For Process 1-2 (isentropic compression):

Calculate the relative pressure at state 2 using the relation:

$$\frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1}$$

$$\frac{P_{r2}}{1.2311} = \frac{760}{95}$$

$$P_{r2} = 9.8488$$

Step 6 of 9

Obtain the value of specific enthalpy from "TABLE-17 Ideal-gas properties of air", in the text book.

At $P_{r2} = 9.8488$

By the method of interpolation, calculate the specific enthalpy as shown:

$$h_2 = 523.63 + (533.98 - 523.63)\left(\frac{9.8488 - 9.684}{10.37 - 9.684}\right)$$

$$h_2 = 526.1164\,\text{kJ/kg}$$

Step 7 of 9

Obtain the value of specific enthalpy from "TABLE-17 Ideal-gas properties of air", in the text book.

At $T_3 = 1100\text{K}$

The specific enthalpy, $h_3 = 1161.07\text{kJ/kg}$

Relative pressure, $P_{r3} = 167.1$

For Process 4-5 (isentropic expansion process):

Calculate the relative pressure at state 5.

$$\frac{P_{r3}}{P_{r4}} = \frac{P_3}{P_4}$$

$$\frac{167.1}{P_{r4}} = \frac{760}{95}$$

$$P_{r4} = 20.89$$

Step 8 of 9

Obtain the value of specific enthalpy from "TABLE-17 Ideal-gas properties of air", in the text book.

At $P_{r4} = 20.89$ from ideal gas properties of air

$$h_4 = 649.22 + (659.84 - 649.22)\left(\frac{20.89 - 20.64}{21.86 - 20.64}\right)$$

$$h_4 = 651.374\,\text{kJ/kg}$$

Step 9 of 9

Calculate the power delivered by the gas turbine.

$$\eta = \frac{\dot{W}}{\dot{Q}_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$\frac{\dot{W}}{\dot{Q}} = 1 - \frac{(h_6 - h_1)}{(h_3 - h_5)}$$

Substitute $75000\,\text{kW}$ for $\dot{Q}$, $651.374\,\text{kJ/kg}$ for h_5, $526.1164\,\text{kJ/kg}$ for h_6, 1161.07kJ/kg for h_3, $290.16\,\text{kJ/kg}$ for h_1.

$$\frac{\dot{W}}{75,000} = 1 - \frac{(526.1164 - 290.16)}{(1161.07 - 651.374)}$$

$$\frac{\dot{W}}{75,000} = 0.53706$$

$$\dot{W} = 40283\,\text{kW}$$

Therefore, the power delivered by the plant is $\boxed{\dot{W} = 40.283\,\text{MW}}$

$$\frac{\dot{W}}{75{,}000} = 0.53706$$

9.096P pg 3

$$\dot{W} = 40283\,\text{kW}$$

Therefore, the power delivered by the plant is $\boxed{\dot{W} = 40.283\,\text{MW}}$

Question: 9.097P

Air enters the compressor of a regenerative gas-turbine engine at 310 K and 100 kPa, where it is compressed to 900 kPa and 650 K. The regenerator has an effectiveness of 80 percent, and the air enters the turbine at 1400 K. For a turbine efficiency of 90 percent, determine *(a)* the amount of heat transfer in the regenerator and *(b)* the thermal efficiency. Assume variable specific heats for air. Answers: *(a)*193 kJ/kg, *(b)*40.0 percent

Step 1 of 6

(a)

Consider constant specific heats at room temperature (From table A-2)

Gas constant

$$R = 0.287\ \text{kJ/kg}\cdot\text{K}$$

Specific heats,

$$c_v = 0.718\ \text{kJ/kg}\cdot\text{K}$$

$$c_p = 1.005\ \text{kJ/kg}\cdot\text{K}$$

Specific heat ratio

$$k = 1.4$$

Calculate the compression ratio

$$r_p = \frac{P_2}{P_1}$$

Here, Pressures at state 1 and 2 are P_1, P_2 respectively

Substitute 900 kPa for P_2, 100 kPa for P_1

$$\begin{aligned} r_p &= \frac{P_2}{P_1} \\ &= \frac{900}{100} \\ &= 9 \end{aligned}$$

Step 2 of 6

Obtain the enthalpies from properties of air table (TABLE A17) at each stage as below:

For temperature $T_1 = 310\ \text{K}$, enthalpy $h_1 = 310.24\ \text{kJ/kg}$

For temperature $T_2 = 650\ \text{K}$, enthalpy $h_2 = 659.84\ \text{kJ/kg}$

For temperature $T_3 = 1400\ \text{K}$, enthalpy $h_3 = 1515.42\ \text{kJ/kg}$ and corresponding pressure ratio is $P_{r3} = 450.5$

Find the pressure ratio between point 4 and point 3

$$P_{r4} = \frac{P_4}{P_3} P_{r3}$$

Here, Pressure at state 3 and 4 is P_3, P_4 respectively.

Substitute 100 kPa for P_4, 900 kPa for P_3, 238 for P_{r3} in the above equation

$$\begin{aligned} P_{r4} &= \frac{100}{900} \times 450.5 \\ &= 50.05 \end{aligned}$$

Obtain isentropic enthalpy at state 4s from the table at $P_{r4} = 50.05$

$$h_{4s} = 835.12\ \text{kJ/kg}$$

Step 3 of 6

Find enthalpy at state 4 using below relation

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

Here, η_T is efficiency of turbine.

Substitute $1515.42\ \text{kJ/kg}$ for h_3, 835.12 for h_{4s}, 0.90 for η_T.

$$\begin{aligned} h_4 &= 1515.42 - (0.90)(1515.42 - 835.12) \\ &= 903.15\ \text{kJ/kg} \end{aligned}$$

Step 4 of 6

Calculate the amount of heat transfer in the regenerator having effectiveness $\varepsilon = 0.8$ as follows,

$$q_R = \varepsilon (h_4 - h_2)$$

Substitute $903.15\ \text{kJ/kg}$ for h_4, $659.84\ \text{kJ/kg}$ for h_2, and 0.80 for ε

$$\begin{aligned} q_R &= 0.80(903.15 - 659.84) \\ &= 194.68\ \text{kJ/kg} \end{aligned}$$

Therefore, the amount of heat transfer in the regenerator is $\boxed{194.648\ \text{kJ/kg}}$

Step 5 of 6

(b)

Find the net work developed

$$\begin{aligned} w_{net} &= w_{T,out} - w_{C,in} \\ &= (h_3 - h_4) - (h_2 - h_1) \end{aligned}$$

Substitute $903.15\ \text{kJ/kg}$ for h_4, $659.84\ \text{kJ/kg}$ for h_2, $1515.42\ \text{kJ/kg}$ for h_3, and $310.24\ \text{kJ/kg}$ for h_1

$$\begin{aligned} w_{net} &= (1515.42 - 903.15) - (659.84 - 310.24) \\ &= 262.67\ \text{kJ/kg} \end{aligned}$$

Step 6 of 6

Find the heat input to the system

$$q_{in} = (h_3 - h_2) - q_R$$

Substitute $194.68\ \text{kJ/kg}$ for q_R, $659.84\ \text{kJ/kg}$ for h_2, and $1515.42\ \text{kJ/kg}$ for h_3.

$$\begin{aligned} q_{in} &= (1515.42 - 659.84) - 194.68 \\ &= 660.9\ \text{kJ/kg} \end{aligned}$$

Find the thermal efficiency of the system

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

Substitute $262.67\ \text{kJ/kg}$ for w_{net}, and $660.9\ \text{kJ/kg}$ for q_{in}.

$$\begin{aligned} \eta_{th} &= \frac{267.62}{660.9} \\ &= 40\ \% \end{aligned}$$

Therefore, the thermal efficiency is $\boxed{40\%}$.

Question: 9.098P

Repeat Prob. 9–97 using constant specific heats at room temperature.

Step 1 of 6

a)

Step 2 of 6

Calculate the actual temperature at stage 4 (T_{4s}) from isentropic process 3-4 as follows.

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

Step 3 of 6

Substitute 100 kPa for P_4, 900 kPa for P_3, and 1.4 for k.

$$T_{4s} = 1400\left(\frac{100}{900}\right)^{\frac{0.4}{1.4}}$$
$$= 747.28\ \text{K}$$

Step 4 of 6

Find enthalpy at state 4 as follows,

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$
$$\eta_T = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})}$$
$$T_4 = T_3 - \eta_T(T_3 - T_{4s})$$

Here, η_T is turbine efficiency.

Substitute 1400 K for T_3, 0.90 for η_T, and 747.28 K for T_{4s}.

$$T_4 = 1400 - 0.90(1400 - 747.28)$$
$$= 812.552\ \text{K}$$

Find the amount of heat transfer in the regenerator having effectiveness $\varepsilon = 0.80$ as follows,

$$q_R = \varepsilon c_p(T_4 - T_2)$$

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p, 650 K for T_2, 0.90 for η_T, and 812.552 K for T_4.

$$q_R = 0.90(1.005)(812.552 - 650)$$
$$= 147.02\ \text{kJ/kg}$$

Therefore, the amount of heat transfer in the regenerator is $\boxed{147.02\ \text{kJ/kg}}$.

Step 5 of 6

(b)

Calculate the net-work developed as follows,

$$w_{net} = w_{T,out} - w_{C,in}$$
$$= c_p(T_3 - T_4) - c_p(T_2 - T_1)$$
$$w_{net} = c_p\left[(T_3 - T_4) - (T_2 - T_1)\right]$$

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p, 310 K for T_1, 1400 K for T_3, 650 K for T_2, and 812.552 K for T_4.

$$w_{net} = 1.005\left[(1400 - 812.552) - (650 - 310)\right]$$
$$= 248.68\,\text{kJ/kg}$$

Step 6 of 6

Find the heat input to the system

$$q_{in} = (h_3 - h_2) - q_R$$
$$= c_p(T_3 - T_2) - q_R$$

Substitute $1.005\,\text{kJ/kg}\cdot\text{K}$ for c_p, 1400 K for T_3, 650 K for T_2, and $147.02\ \text{kJ/kg}$ for q_R.

$$q_{in} = (1.005\ \text{kJ/kg K})(1400 - 650)\,\text{K} - 147.02\,\text{kJ/kg}$$
$$= 606.73\,\text{kJ/kg}$$

Find the thermal efficiency as follows,

$$\eta_T = \frac{w_{net}}{q_{in}}$$

Substitute $248.68\,\text{kJ/kg}$ for w_{net}, and $606.73\,\text{kJ/kg}$ for q_{in}

$$\eta_T = \frac{248.68\ \text{kJ/kg}}{606.73\ \text{kJ/kg}}$$
$$\approx 41\,\%$$

Therefore, the thermal efficiency is $\boxed{41\,\%}$.

Question: 9.099P

Repeat Prob. 9–97 for a regenerator effectiveness of 70 percent.

Step 1 of 5

a)

Consider constant specific heats at room temperature (From table A-2)

Gas constant

$$R = 0.287\ \text{kJ/kg}\cdot\text{K}$$

Specific heats,

$$c_v = 0.718\ \text{kJ/kg}\cdot\text{K}$$

$$c_p = 1.005\ \text{kJ/kg}\cdot\text{K}$$

Specific heat ratio

$$k = 1.4$$

Calculate the compression ratio

$$r_p = \frac{P_2}{P_1}$$

Here, Pressures at state 1 and 2 are P_1, P_2 respectively

Substitute 900 kPa for P_2, and 100 kPa for P_1

$$\begin{aligned} r_p &= \frac{P_2}{P_1} \\ &= \frac{900}{100} \\ &= 9 \end{aligned}$$

Step 2 of 5

Obtain the enthalpies from properties of air table (TABLE A17) at each stage as below:

For temperature $T_1 = 310\ \text{K}$, enthalpy $h_1 = 310.24\ \text{kJ/kg}$

For temperature $T_2 = 650\ \text{K}$, enthalpy $h_2 = 659.84\ \text{kJ/kg}$

For temperature $T_3 = 1400\ \text{K}$, enthalpy $h_3 = 1515.42\ \text{kJ/kg}$ and corresponding pressure ratio is $P_{r3} = 450.5$

Find the pressure ratio between point 4 and point 3

$$P_{r4} = \frac{P_4}{P_3} P_{r3}$$

Here, Pressure at state 3 and 4 is P_3, P_4 respectively.

Substitute 100 kPa for P_4, 900 kPa for P_3 , 238 for P_{r3} in the above equation

$$\begin{aligned} P_{r4} &= \frac{100}{900} \times 450.5 \\ &= 50.05 \end{aligned}$$

Obtain isentropic enthalpy at state 4s from the table at $P_{r4} = 50.05$

$$h_{4s} = 835.12\ \text{kJ/kg}$$

Step 3 of 5

Find enthalpy at state 4 using below relation

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

Here, η_T is efficiency of turbine.

Substitute $1515.42\ \text{kJ/kg}$ for h_3 , 835.12 for h_{4s} , 0.90 for η_T .

$$\begin{aligned} h_4 &= 1515.42 - (0.90)(1515.42 - 835.12) \\ &= 903.15\ \text{kJ/kg} \end{aligned}$$

Step 4 of 5

Calculate the amount of heat transfer in the regenerator having effectiveness $\varepsilon = 0.70$ as follows

$$q_R = \varepsilon (h_4 - h_2)$$

Substitute $903.15\ \text{kJ/kg}$ for h_4 , $659.84\ \text{kJ/kg}$ for h_2 , and 0.70 for ε in equation

$$\begin{aligned} q_R &= 0.70(903.15 - 659.84) \\ &= 170.317\ \text{kJ/kg} \end{aligned}$$

Therefore, the amount of heat transfer in the regenerator is $\boxed{170.317\ \text{kJ/kg}}$.

Step 5 of 5

(b)

Find the net work developed

$$\begin{aligned} w_{net} &= w_{T,out} - w_{C,in} \\ &= (h_3 - h_4) - (h_2 - h_1) \end{aligned}$$

Substitute $903.15\ \text{kJ/kg}$ for h_4 , $659.84\ \text{kJ/kg}$ for h_2 , $1515.42\ \text{kJ/kg}$ for h_3 , and $310.24\ \text{kJ/kg}$ for h_1

$$\begin{aligned} w_{net} &= (1515.42 - 903.15) - (659.84 - 310.24) \\ &= 262.67\ \text{kJ/kg} \end{aligned}$$

Find the heat input to the system

$$q_{in} = (h_3 - h_2) - q_R$$

Substitute $170.317\ \text{kJ/kg}$ for q_R , $659.84\ \text{kJ/kg}$ for h_2 , and $1515.42\ \text{kJ/kg}$ for h_3 .

$$\begin{aligned} q_{in} &= (1515.42 - 659.84) - 170.317 \\ &= 685.263\ \text{kJ/kg} \end{aligned}$$

Find the thermal efficiency

$$\eta_T = \frac{w_{net}}{q_{in}}$$

Substitute $262.67\ \text{kJ/kg}$ for w_{net} , and $685.263\ \text{kJ/kg}$ for q_{in} .

$$\begin{aligned} \eta_T &= \frac{262.67}{685.263} \\ &= 38\ \% \end{aligned}$$

Therefore, the thermal efficiency is $\boxed{38\%}$.

Question: 9.100P

Develop an expression for the thermal efficiency 0 f an ideal Brayton cycle with an ideal regenerator of effectiveness 100 percent. Use constant specific heats at room temperature

Step 1 of 4

The efficiency of the cycle is defined as

$$\eta_{bry} = 1 - \frac{q_{out}}{q_{in}}$$

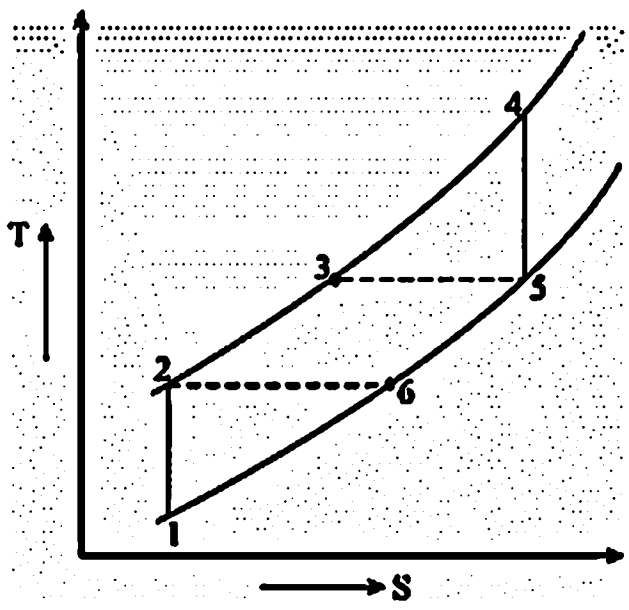
The rate of heat addition

$$q_{in} = c_p (T_4 - T_3)$$

The rate of heat rejection

$$q_{out} = c_p (T_6 - T_1)$$

Step 2 of 4



Step 3 of 4

For an ideal regenerator of 100% effectiveness

$T_5 = T_3$ And $T_2 = T_6$

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Assuming pressure ratio $r_p = \frac{P_2}{P_1}$ then

$$\frac{T_2}{T_1} = (r_p)^{\frac{k-1}{k}}$$

Similarly process 4-5 is isentropic expansion

$$\frac{T_4}{T_5} = \left(\frac{P_4}{P_5}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{T_5} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{T_5} = (r_p)^{\frac{k-1}{k}}$$

Step 4 of 4

We know thermal efficiency of the cycle is

$$\eta = 1 - \frac{q_{out}}{q_{in}}$$

$$\eta = 1 - \frac{T_6 - T_1}{T_4 - T_3}$$

$$= 1 - \frac{T_2 - T_1}{T_4 - T_5}$$

$$= 1 - \frac{T_1}{T_4}\left(\frac{\frac{T_2}{T_1} - 1}{1 - \frac{T_5}{T_4}}\right)$$

$$= 1 - \frac{T_1}{T_4}\left[\frac{(r_p)^{\frac{k-1}{k}} - 1}{1 - \frac{1}{(r_p)^{\frac{k-1}{k}}}}\right]$$

$$\eta = 1 - \frac{T_1}{T_4}(r_p)^{\frac{k-1}{k}}$$

Therefore efficiency of an ideal Brayton cycle with an ideal regenerator of effectiveness 100% is

$$\boxed{\eta = 1 - \frac{T_1}{T_4}(r_p)^{\frac{k-1}{k}}}$$

Question: 9.101P

Why is the Carnot cycle not a realistic model for steam power plants?

Step 1 of 1

Carnot cycle is not a realistic model for steam power plant due to the following reasons.

(1) Limiting the heat transfer processes to two-phase systems for maintaining isothermal conditions severely limits the maximum temperature that can be used in the cycle.

(2) Turbine handles steam with high moisture content that causes erosion.

(3) It is not easy to control the condensation process so precisely as to end up with the desired quality at state.

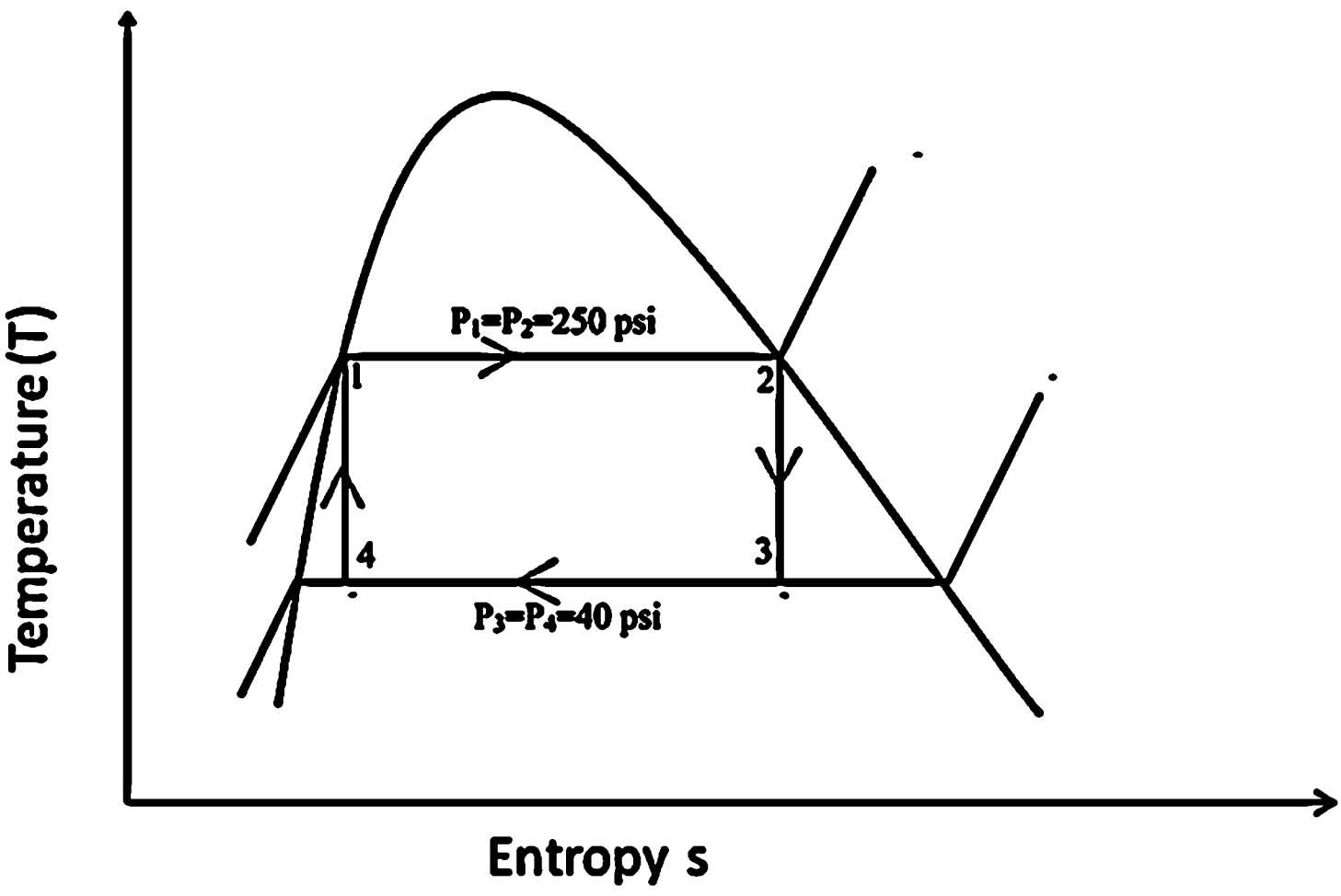
(4) Designing a compressor that can handle two phases is not practiacal.

Question: 9.102P pg 1

Water enters the boiler of a steady-flow Carnot engine as a saturated liquid at 250 psia and leaves with a quality of 0.95. Steam leaves the turbine at a pressure of 40 psia. Show the cycle on a *T-s* diagram relative to the saturation lines, and determine (a) the thermal efficiency *(b)* the quality at the end of the isothermal heat-rejection process, and *(c)* the net work output.

Step 1 of 9

Draw the T-s diagram of the Carnot cycle.



Step 2 of 9

(a)

Obtain the properties at state1 from the saturated water pressure table,

At $P_1 = 250$ psia Saturated liquid, at this pressure,

$$T_1 = T_{sat} = 400.97°F$$
$$h_1 = (h_f) = 376.09 \text{ Btu/lbm}$$
$$s_1 = (s_f) = 0.56784 \text{ Btu/lbm°F}$$

Here, T_1 is the saturation temperature at state 1 h_1 is the enthalpy at state 1 and s_1 is the entropy at state 1.

Step 3 of 9

At state 2:

Obtain the properties at state 2 from the saturated water pressure table,

$P_1 = 250$ psi and is 95% saturated vapor

$$T_2 = T_1 = 400.97°F$$

Calculate the enthalpy at state 2 by using the equation,

$$h_2 = (h_f) + x(h_{fg})$$

Here, h_2 is the enthalpy at state 2, h_f is the enthalpy of saturated liquid at state 2 and h_{fg} is the enthalpy of wet steam at state 2.

Substitute 376.09 Btu/lbm for h_f . 0.95 for x and 825.47 Btu/lbm for h_{fg} .

$$h_2 = 376.09 + (0.95 \times 825.47)$$
$$= 1160.286 \text{ Btu/lbm}$$

Calculate the entropy at state 2 by using equation,

$$s_2 = (s_f) + x(s_{fg})$$

Here, s_2 is the entropy at state 2 , s_f is the entropy of saturated liquid at state 2 and s_{fg} is the entropy of wet steam at state 2.

Substitute $0.56784 \text{ Btu/lbm°F}$ for s_f . 0.95 for x and $0.95912 \text{ Btu/lbm°F}$ for s_{fg} .

$$s_2 = 0.56784 + (0.95 \times 0.95912)$$
$$= 1.479 \text{ Btu/lbm°F}$$

Step 4 of 9

State 3:

Obtain the properties of steam at state 3 from the saturated water table.

Since process 2-3 isentropic process.

So $s_3 = s_2 = 1.479 \text{ Btu/lbm°F}$

From the saturated water pressure table,

At $P_3 = 40$ psi

At this pressure and entropy,

$$T_3 = (T_{sat}) = 267.22°F$$
$$(s_g) = 1.6766 \text{ Btu/lbm°F}$$
$$(s_f) = 0.39213 \text{ Btu/lbm°F}$$

$$h_f = 236.14 \text{ Btu/lbm}$$
$$h_g = 1169.8 \text{ Btu/lbm}$$
$$h_{fg} = 933.66 \text{ Btu/lbm}$$

Step 5 of 9

Calculate the entropy at state 3 by using the equation,

Step 5 of 9

Calculate the entropy at state 3 by using the equation,

$$s_3 = s_2 = (s_f) + x_3(s_g - s_f)$$

Here, s_2 is the entropy at state 2 , s_f is the entropy of saturated liquid a state 3, s_g is the entropy of saturated vapour state 3 and x_3 is the quality of steam at state 3.

Substitute $1.479\,\text{Btu/lbm°F}$ for s_2 , $0.39213\,\text{Btu/lbm°F}$ for s_f and $1.6766\ \text{Btu/lbm°F}$ for s_g .

$$1.479 = 0.39213 + x_3(1.6766 - 0.39213)$$
$$x_3 = 0.846$$

Calculate the enthalpy at state 3 by using the equation,

$$h_3 = (h_f) + x_3(h_g - h_f)$$

Here, h_3 is the enthalpy at state 3 , h_f is the enthalpy of saturated liquid at state 3 and h_g is the enthalpy of saturated vapour at state 3.

Substitute $236.14\,\text{Btu/lbm}$ for h_f , 0.846 for x_3 and $1169.8\,\text{Btu/lbm}$ for h_g .

$$h_3 = 236.14 + 0.846 \times (1169.8 - 236.14)$$
$$h_3 = 1026.01\ \text{Btu/lbm}$$

Step 6 of 9

State 4:

Obtain the properties of steam at state 4 from the saturated water table.

Since Process 4-1 isentropic process

$$s_4 = s_1 = 0.56784\,\text{Btu/lbm°F}$$

$$P_3 = 40\ \text{psi}$$

Calculate the entropy at state 4 by using the equation,

$$s_4 = s_1 = (s_f) + x_4(s_g - s_f)$$

Here, s_1 is the entropy at state 1, s_f is the entropy of saturated liquid a state 4 and s_{fg} is the entropy of wet steam state 4 and x_4 is the quality of steam at the end of the heat rejection.

Substitute $0.56784\,\text{Btu/lbm°F}$ for s_4 , $0.39213\,\text{Btu/lbm°F}$ for s_f and $1.6766\ \text{Btu/lbm°F}$ for s_g .

$$0.56784 = 0.39213 + x_4(1.6766 - 0.39213)$$
$$x_4 = 0.137$$

Therefore, the quality of steam at the end of the heat rejection is $\boxed{x_4 = 0.137}$.

Step 7 of 9

Calculate the entropy at state 4 by using the equation,

$$h_4 = (h_f) + x_4(h_g - h_f)$$

Here, h_4 is the enthalpy at state 4 , h_f is the enthalpy of saturated liquid at state 4 and h_g is the enthalpy of saturated vapour state 4.

Substitute $236.14\,\text{Btu/lbm}$ for h_f , 0.137 for x_4 and $1169.8\,\text{Btu/lbm}$ for h_g .

$$h_4 = 236.14 + [0.137 \times (1169.8 - 236.14)]$$
$$h_4 = 364.051\ \text{Btu/lbm}$$

Step 8 of 9

(b)

Calculate the thermal efficiency of the Carnot cycle by using the equation,

$$\eta_{Carnot} = 1 - \frac{q_{rej}}{q_{in}}$$
$$= 1 - \frac{h_3 - h_4}{h_2 - h_1}$$

Here, η_{carnot} is the thermal efficiency of Carnot cycle.

Substitute $376.09\,\text{Btu/lbm}$ for h_1 , $1160.286\,\text{Btu/lbm}$ for h_2 , $1026.01\,\text{Btu/lbm}$ for h_3 and $364.0514\,\text{Btu/lbm}$ for h_4 .

$$\eta_{Carnot} = 1 - \left(\frac{1026.01 - 364.0514}{1160.286 - 376.09}\right)$$
$$= 0.155$$

Therefore, the efficiency of the Carnot cycle is $\boxed{\eta_C = 15.5\%}$.

Step 9 of 9

(c)

Calculate the net work output of the cycle by using the equation,

$$W_{net} = (h_2 - h_3) - (h_1 - h_4)$$

Here, W_{net} is the net work output of the cycle.

Substitute $376.09\,\text{Btu/lbm}$ for h_1 , $1160.286\,\text{Btu/lbm}$ for h_2 , $1026.01\,\text{Btu/lbm}$ for h_3 and $364.0514\,\text{Btu/lbm}$ for h_4 .

$$W_{net} = ((1160.286 - 1026.57) - (376.09 - 364.0514))$$
$$= 133.716 - 12.0386$$
$$= 121.6774$$

364.0514 Btu/lbm for h_4.

9.102P pg 3

$$W_{net} = ((1160.286 - 1026.57) - (376.09 - 364.0514))$$

$$= 133.716 - 12.0386$$

$$= 121.6774$$

$$\simeq 122 \text{ Btu/lbm}$$

Therefore, the net work output of the cycle is $W_{net} = 122$ Btu/lbm .

Question: 9.103P

A steady-flow Carnot cycle uses water as the working fluid. Water changes from saturated liquid to saturated vapor as heat is transferred to it from a source at 250°G Heat rejection takes place at a pressure of 20 kPa. Show the cycle on a *Ts* diagram relative to the saturation lines, and determine *(a)* the thermal efficiency, *(b)* the amount of heat rejected, and

(c) the net work output.

Step 1 of 5

Given

Temperature of the source $T_H = 250°C$

$= 250 + 273K$

$= 523K$

Temperature at which heat is rejected $= T_L$

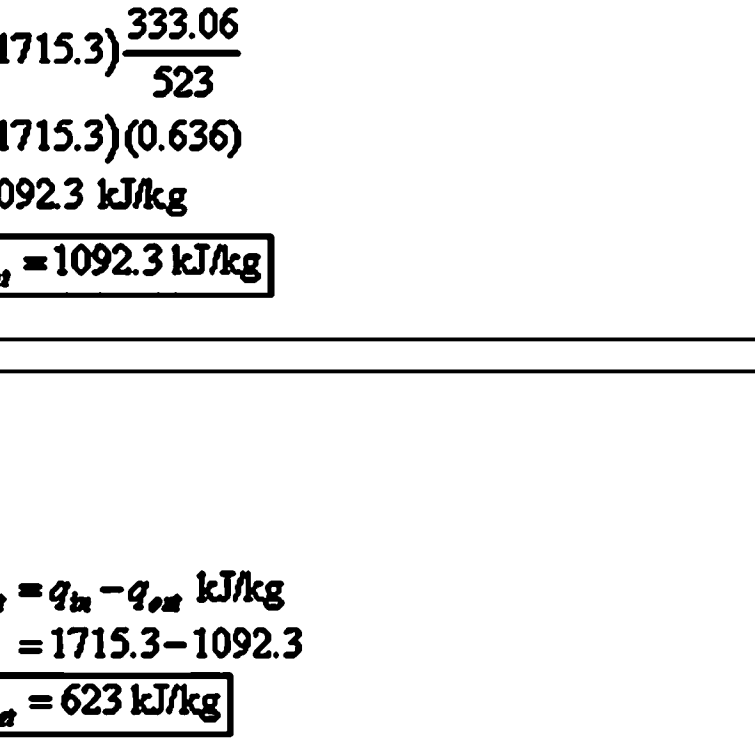
$T_L = T_{sat20kpa}$

$= 60.06°C$ (From Saturated water – Pressure table)

$= 273 + 60.06K$

$T_L = 333.06K$

Step 2 of 5



Step 3 of 5

(a)The Carnot cycle efficiency is defined as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$= 1 - \frac{333.06}{523}$$

$$= 1 - 0.636$$

$$= 0.363$$

$$\boxed{\eta_{th} = 36.3\%}$$

Step 4 of 5

b)

The heat supplied during this cycle is simply the enthalpy of vaporization

$q_{in} = h_{fg} = 1715.3$ kJ/kg (From Saturated water – Pressure table)

Heat rejected $q_{out} = q_{in}\frac{T_L}{T_H}$

$$= (1715.3)\frac{333.06}{523}$$

$$= (1715.3)(0.636)$$

$$= 1092.3 \text{ kJ/kg}$$

$$\boxed{q_{out} = 1092.3 \text{ kJ/kg}}$$

Step 5 of 5

(c)

The work is expressed as

$$w_{net} = q_{in} - q_{out} \text{ kJ/kg}$$

$$= 1715.3 - 1092.3$$

$$\boxed{w_{net} = 623 \text{ kJ/kg}}$$

Question: 9.104P

Repeat Prob. 9–103 for a heal rejection pressure of 10 kPa.

Step 1 of 5

Given

Temperature of the source $T_H = 250°C$

$= 250 + 273K$

$= 523K$

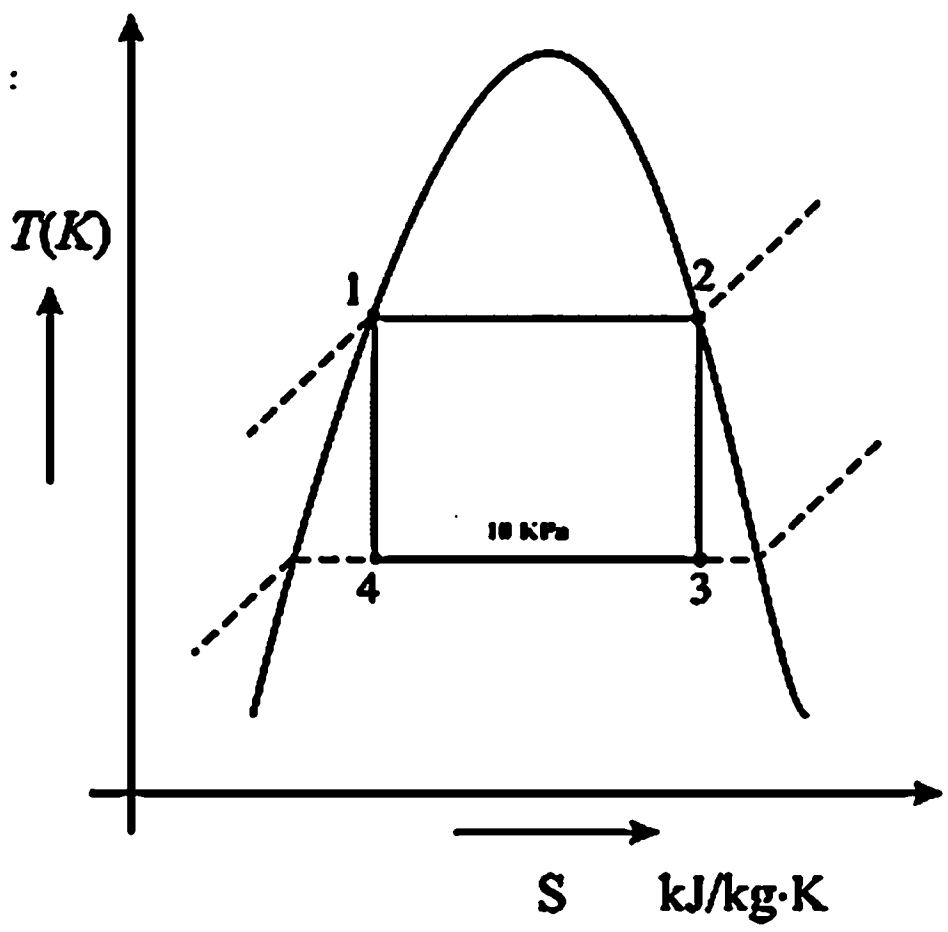
Temperature at which heat is rejected $= T_L$

$T_L = T_{sat@10kpa} = 45.81°C$ (From Saturated water – Pressure table)

$= 45.81 + 273K$

$= 318.81K$

Step 2 of 5



Step 3 of 5

(a)

The Carnot cycle efficiency is defined as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$= 1 - \frac{318.81}{523}$$

$$= 1 - 0.609$$

$$\boxed{\eta_{th} = 39\%}$$

Step 4 of 5

(b)

The heat supplied during this cycle is simply the enthalpy of vaporization

$q_{in} = h_{fg} = 1715.3$ kJ/kg (From Saturated water – Pressure table)

Heat rejected $q_{out} = q_{in} \frac{T_L}{T_H}$

$$= (1715.3)\frac{318.81}{523}$$

$$= (1715.3)(0.609)$$

$$\boxed{q_{out} = 1045.6 \text{ kJ/kg}}$$

Step 5 of 5

(c)

The work is expressed as

$$w_{net} = q_{in} - q_{out} \text{ kJ/kg}$$

$$= 1715.3 - 1045.6$$

$$\boxed{w_{net} = 670 \text{ kJ/kg}}$$

Question: 9.105P

Consider a steady-flow Carnot cycle with water as the working fluid. The maximum and minimum temperatures in the cycle are 350 and 60°C. The quality of water is 0.891 at the beginning of the heat-rejection process and 0.1 at the end. Show the cycle on a *T-s* diagram relative to the saturation lines, and determine *(a)* the thermal efficiency, *(b)* the pressure at the turbine inlet, and (c) the net work output. Answers:(a) 0.465, *[b)*1.40 MPa, (c) 1623 kJ/kg

Step 1 of 6

Given

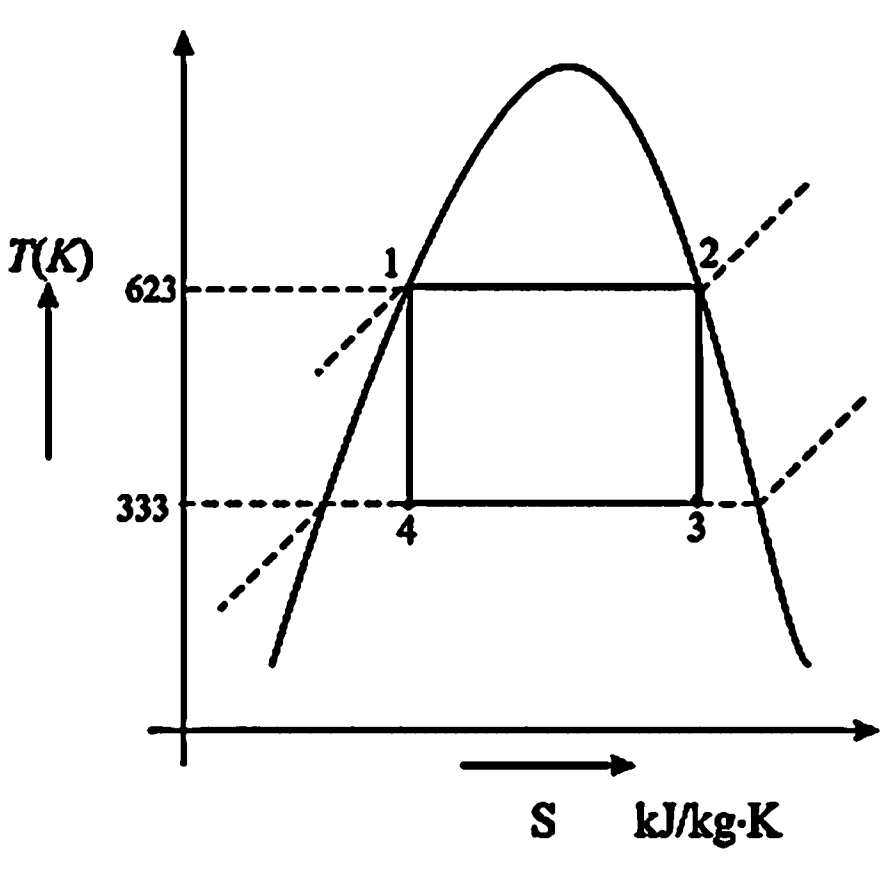
Maximum temperature of Carnot cycle $T_H = 350°C = 350+273 = 623K$

Minimum temperature of Carnot cycle $T_L = 60°C = 333K$

Dryness fraction at point turbine outlet (At point 3) = $x_3 = 0.891$

Dryness fraction at condenser outlet (At point 4) = $x_4 = 0.1$

Step 2 of 6



Step 3 of 6

(a)

The Carnot cycle efficiency is defined as

$$\eta_{th} = 1 - \frac{T_L}{T_H}$$

$$= 1 - \frac{333}{623} = 0.465$$

$$\boxed{\eta_{th} = 46.5\%}$$

Step 4 of 6

(b)

At $T_3 = 60°C \rightarrow s_f = 0.8313$ kJ/kg.K , $s_{fg} = 7.0769$ kJ/kg.K (From Saturated water – Temperature table)

We know that $s_2 = s_3 = s_f + x_3 s_{fg}$

$$s_3 = 0.8313 + (0.891)7.0769$$

$$s_2 = s_3 = 7.1368 \text{ kJ/kg.K}$$

Step 5 of 6

At $T_2 = 350°C$, $s_2 = 7.1368$ kJ/kg.K $\rightarrow P_2 = 1.40$ MPa (From Superheated water table)

$$\boxed{P_2 = 1.40 \text{ MPa}}$$

Step 6 of 6

(c) The net work output

At $T_3 = 60°C \rightarrow s_f = 0.8313$ kJ/kg.K , $s_{fg} = 7.0769$ kJ/kg.K (From Saturated water – Temperature table)

We know that $s_4 = s_f + x_4 s_{fg}$

$$s_4 = 0.8313 + (0.1)7.0769$$

$$s_4 = 1.538 \text{ kJ/kg.K}$$

The work is expressed as

$$w_{net} = (T_H - T_L)(s_3 - s_4) \text{ kJ/kg}$$

$$= (623-333)(7.14-1.538)$$

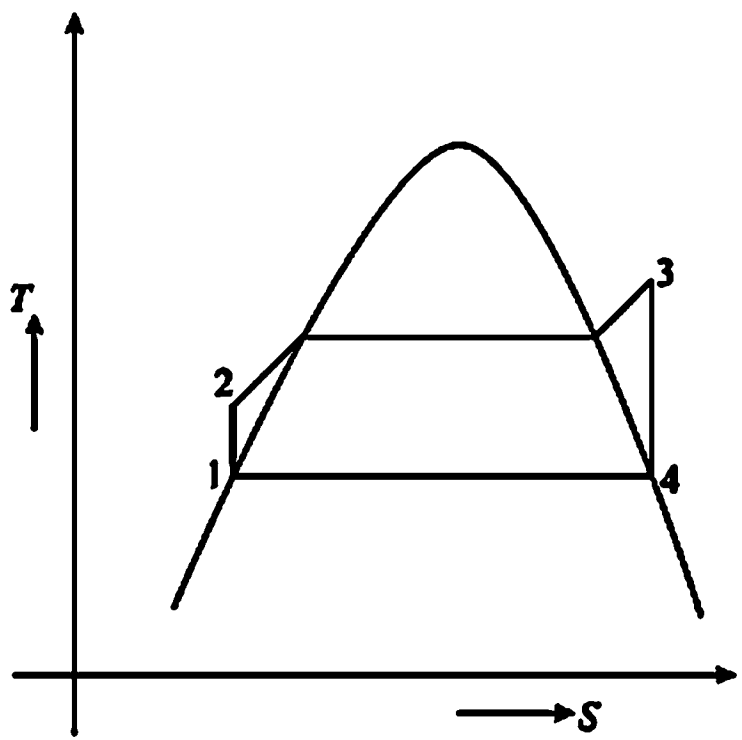
$$\boxed{w_{net} = 1624 \text{ kJ/kg}}$$

Question: 9.106P

What four processes make up the simple ideal Rankine cycle?

Step 1 of 2

T-S Diagram of Rankine cycle



Step 2 of 2

The ideal Rankine cycle does not involve any internal irreversibility's and consists of the following four processes:

1-2 Isentropic compression in a pump
2-3 Constant pressure heat addition in a boiler
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection in a condenser

Question: 9.107P

Consider a simple ideal Rankine cycle with fixed turbine inlet conditions. What is the effect of lowering the ! condenser pressure on

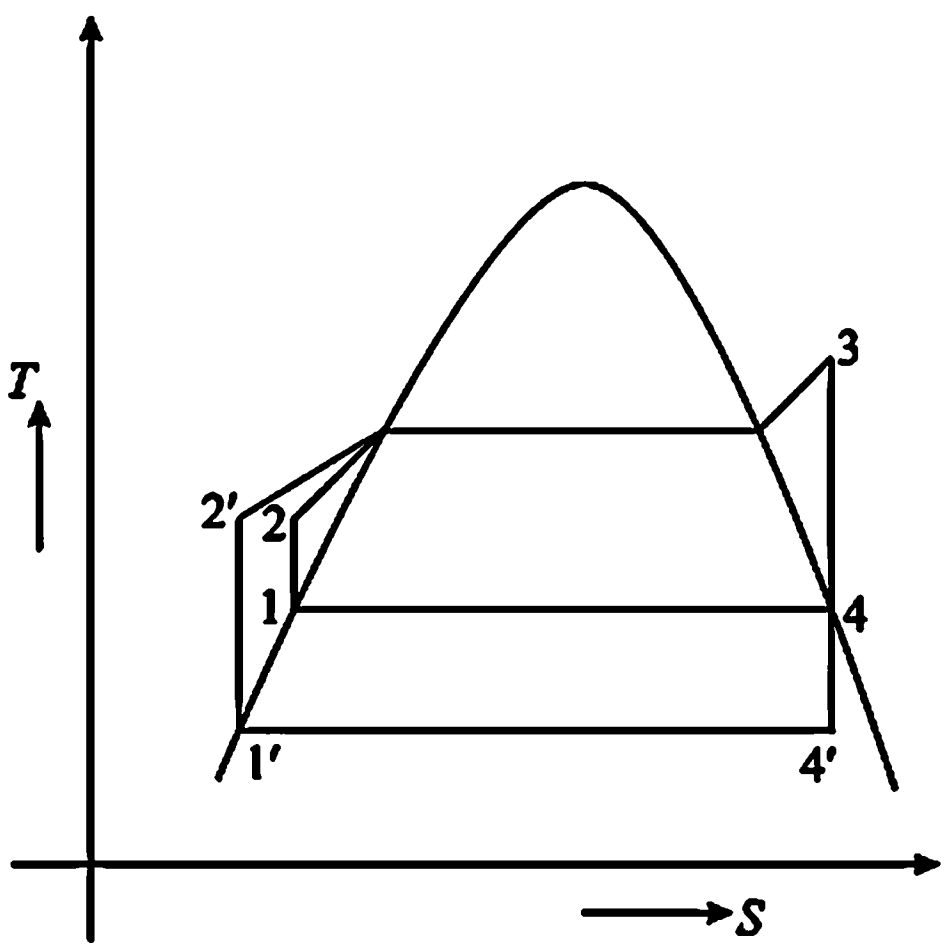
Pump work input:	(a) increases, (b) decreases, (c)remains the same
Turbine work ouput:	(a) increases (b) decreases, (c) remains the same
Heat supplied:	(a) increases (b) decreases, (c) remains the same
Heat rejected:	(a) increases, (b) decreases, (c) remains the same
Cycle efficiency:	(a) increases, (b) decreases, (c) remains the same
Moisture content at turbine exit:	(a) increases, (b) decreases, (c) remains the same

Step 1 of 8

Given: Turbine inlet conditions are fixed

Step 2 of 8

T-S Diagram of Rankine Cycle



Step 3 of 8

Pump work input: (a) increases

As the pressure difference has increased in process $1'-2'$ compared to process $1-2$ hence pump work increases

Step 4 of 8

Turbine work output: (a) increases

As the enthalpy difference has increased in process $3-4'$ compared to process $3-4$ hence turbine output work increases

Step 5 of 8

Heat supplied: (a) increases

As the area under $2'-3$ has increased compared to process $2-3$ hence heat supplied increases

Step 6 of 8

Heat rejected: (b) decreases

As the area under $4'-1'$ has increased compared to process $4-1$ hence heat rejected decreases

Step 7 of 8

Cycle efficiency: (a) increases

As the compressor pressure is reduced net work increases hence efficiency increases

Step 8 of 8

Moisture content at turbine exit: (a) increases

As point 4 changes its position into $4'$ hence moisture content will increases

Question: 9.108P

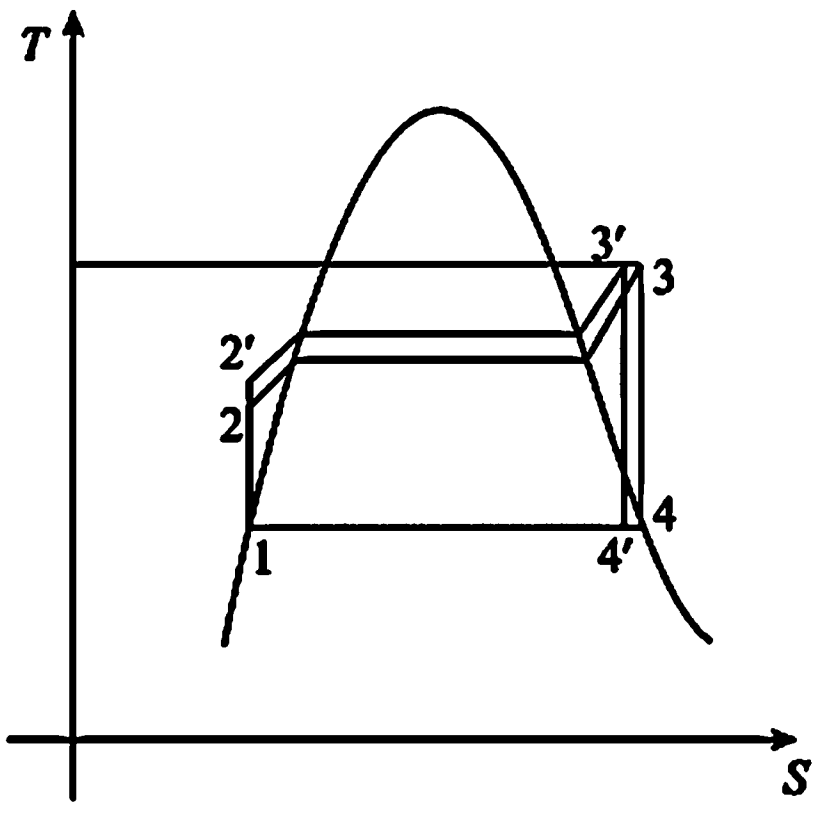
Consider a simple ideal Rankine cycle with fixed turbine intel temperature and condenser pressure. What is the effect of increasing the boiler pressure on

Pump work input:	(*a*) increases, (*b*) decreases, (c)remains the same
Turbine work ouput:	(*a*) increases (b) decreases, (c) remains the same
Heat supplied:	(*a*) increases (b) decreases, (c) remains the same
Heat rejected:	(*a*) increases, (b) decreases, (c) remains the same
Cycle efficiency:	(*a*) increases, (b) decreases, (c) remains the same
Moisture content at turbine exit:	(*a*) increases, (b) decreases, (c) remains the same

Step 1 of 8

Given: Turbine inlet temperature is constant
Condenser pressure is constant

Step 2 of 8



Step 3 of 8

Pump work input: (a) increases
If boiler pressure increases pump work increases as $1-2$ has increased to $1-2'$

Step 4 of 8

Turbine work output: (a) increases
If the boiler pressure increases enthalpy difference increases from $3-4$ to $3'-4$ and hence turbine work output increases

Step 5 of 8

Heat supplied: (a) increases
If the boiler pressure increases area under $2-3$ has increased to area under $2'-3'$ and hence heat supplied increases

Step 6 of 8

Heat rejected: (b) decreases
If the boiler pressure increases the turbine outlet point goes less than saturated vapor region for given conditions and hence heat rejected decreases

Step 7 of 8

Cycle efficiency: (a) increases
If the boiler pressure increases heat supplied increases and hence cycle efficiency increases

Step 8 of 8

Moisture content at turbine exit: (a) increases
If the boiler pressure increases the turbine outlet point goes less than saturated vapor region for given conditions and hence moisture content decreases

Question: 9.109P

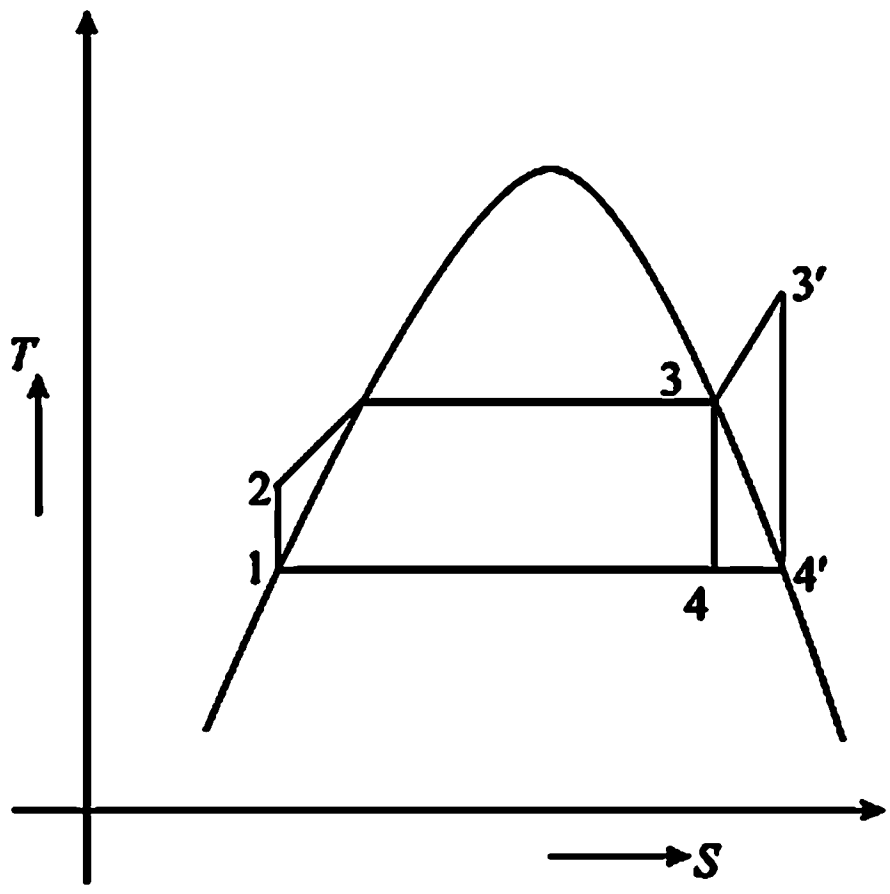
Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. What is the effect of super- heating the steam to a higher temperature on

Pump work input:	(a) increases, (b) decreases, (c)remains the same
Turbine work ouput:	(a) increases (b) decreases, (c) remains the same
Heat supplied:	(a) increases (b) decreases, (c) remains the same
Heat rejected:	(a) increases, (b) decreases, (c) remains the same
Cycle efficiency:	(a) increases, (b) decreases, (c) remains the same
Moisture content at turbine exit:	(a) increases, (b) decreases, (c) remains the same

Step 1 of 8

Given: Boiler pressure is constant
Condenser pressure is constant

Step 2 of 8



Step 3 of 8

Pump work input: (c) remains the same
As there is no change in the curve $1-2$ pump work remains same

Step 4 of 8

Turbine work output: (a) increases
As the enthalpy has been from $3-4$ to $3'-4'$ turbine work output increases

Step 5 of 8

Heat supplied: (a) increases
As the area under curve $2-3'$ has increased in boiler compared to $2-3$ hence heat supplied increases

Step 6 of 8

Heat rejected: (a) increases
As the area under curve $1-4'$ has increased in boiler compared to $1-4$ hence heat supplied increases

Step 7 of 8

Cycle efficiency: (a) increases
As the heat supplied is increased cycle efficiency increases

Step 8 of 8

Moisture content at turbine exit: (b) decreases
As the point 4 has tend towards saturated and supersaturated region (point $4'$) moisture content decreases

Question: 9.110P

How do actual vapor power cycles differ from idealized ones?

Step 1 of 4

The actual vapor power cycles differ from the idealized ones as a result of irreversibilities in various components . Fluid friction and heat loss are the common sources of irreversibilities.

Step 2 of 4

Fluid friction:

It causes pressure drops in the boiler, the condenser, and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure. Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to pressure drop in the connecting pipes.

Step 3 of 4

Heat loss:

There will be heat loss from the steam to the surroundings as the steam flows through various components. To maintain the same level of net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesired heat losses. As a result cycle efficiency decreases.

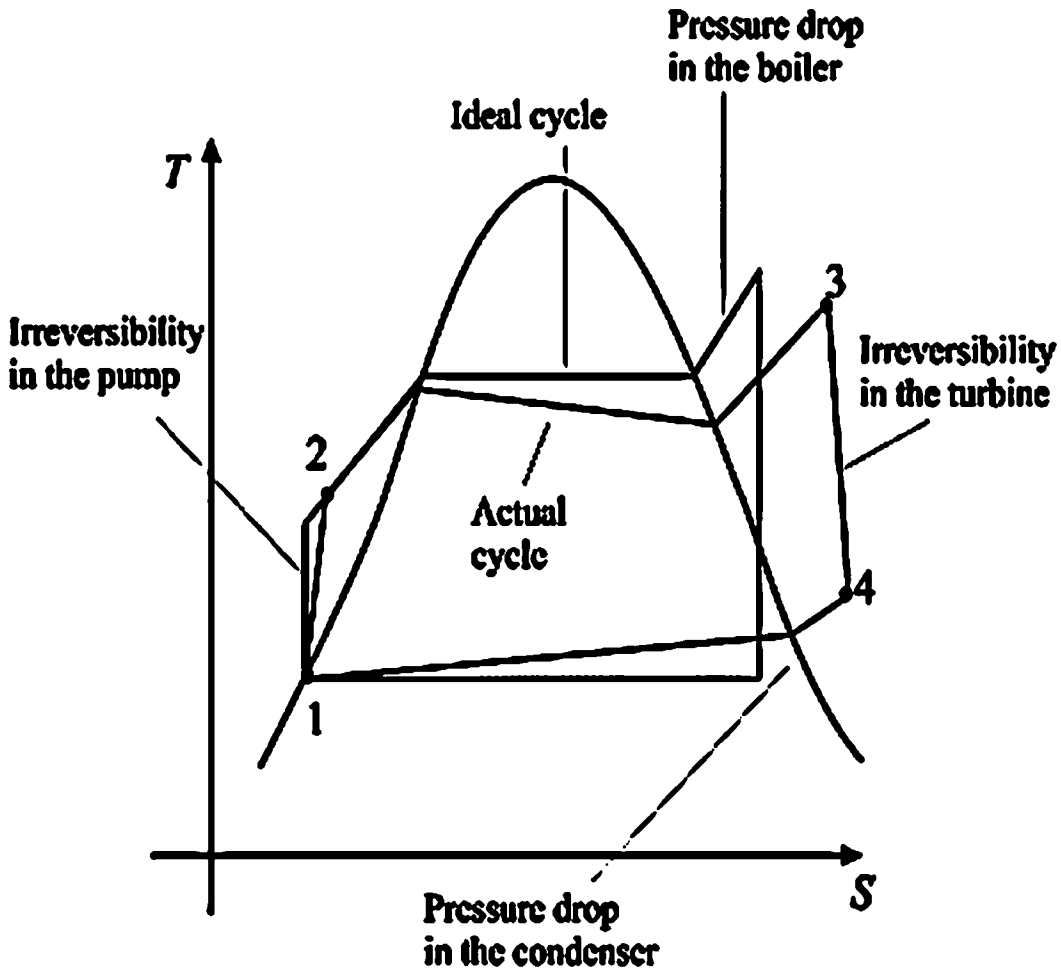
Step 4 of 4

Other irreversibilities are bearings at moving parts, steam leakage and air entering into the condenser.

Question: 9.111P

Compare the pressures at the inlet and the exit of the boiler for *(a)* actual and *ib)* ideal cycles,

Step 1 of 3



Step 2 of 3

(a) In actual cases the boiler exit pressure will be lower than the inlet pressure. Due to fluid friction there will be pressure drop in the boiler. As a result steam leaves the boiler at a somewhat lower pressure.

Step 3 of 3

(b) In ideal case inlet and outlet pressure will be same.

Question:

9.112P

The entropy of steam increases in actual steam turbines as a result of irreversibilities. In an effort to control entropy increase, it is proposed to cool the steam in the turbine by running cooling water around the turbine casing. It is argued that this will reduce the entropy and the enthalpy of the steam at the turbine exit and thus increase the work output. How would you evaluate this proposal?

Step 1 of 1

I will not agree the proposal, as turbine work output depends on enthalpy at boiler exit point. If the boiler is cooled by cool water there will be heat lost from boiler and thus enthalpy decreases. As a result turbine work output decreases.

Question: 9.113P

Is it possible to maintain a pressure of 10 kPa in a condenser that is being cooled by river water entering at 20°C?

Step 1 of 1

Yes, because the saturation temperature of steam at 10 KPa is 45.81°C, Cooling water temperature is 20°c which is less than that of saturation temperature of steam .And hence cooling water with given temperature can be used.

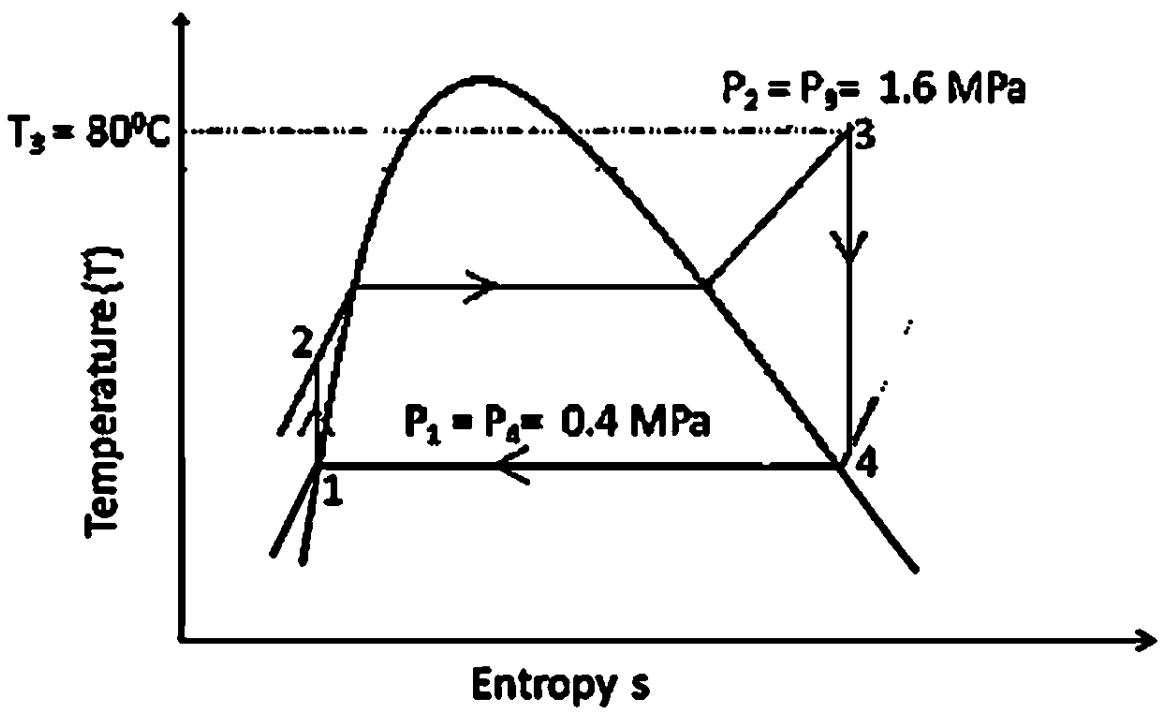
Question:

9.114P

An engineer has proposed that a simple ideal Rankine cycle that uses refrigerant-134a be used to produce work with heat from a low-temperature thermal energy reservoir. The boiler operates at 1.6 MPa. the condenser at 0.4 MPa, and the turbine inlet at 80°C. Determine the R-134a mass flow rate needed for this cycle to produce 750 kW of power and the thermal efficiency of the cycle.

Step 1 of 7

The rankine cycle representing the operating conditions is,



Step 2 of 7

At state1 from the diagram:

From the Saturated R-134a Pressure table
At a pressure of 0.4 MPa ($P_1 = 0.4$ MPa) the properties of enthalpy and specific volume are,

$$h_1 = (h_f) = 63.94 \text{ kJ/kg}$$

$$v_1 = (v_f) = 0.0007907 \text{ m}^3/\text{kg}$$

Step 3 of 7

At state2 from the diagram:

From the Saturated R-134a Pressure table
At a pressure of 1.6 MPa ($P_2 = 1.6$ MPa),

$$P_2 = 1.6 \text{ MPa} = 1600 \text{ kPa}$$

From process 1-2,
The work done in the compression process is,

$$w_{pump} = v_1 \times (P_2 - P_1)$$

$$w_{pump} = 0.0007907 \left(\frac{\text{m}^3}{\text{kg}}\right) \times (1600 \text{ kPa} - 400 \text{ kPa}) \left(\frac{1 \text{ kJ}}{1 \left(\text{kPa m}^3\right)}\right)$$

$$w_{pump} = 0.949 \text{ kJ/kg}$$

Enthalpy at state 2 is,

$$h_2 = h_1 + w_{pump} = 63.94 + 0.949 = 64.889 \text{ kJ/kg}$$

Step 4 of 7

At state3 from the diagram:

From the Superheated refrigerant-134a tables,
At pressure of 1.6 MPa ($P_3 = P_2 = 1.6$ MPa) and temperature of 80°C ($T_3 = 80^0$C) the properties of enthalpy and specific volume are,

$$h_3 = 305.07 \text{ kJ/kg}$$

$$s_3 = 0.9875 \text{ kJ/kg} \cdot \text{K}$$

Step 5 of 7

At state4 from the diagram:

Consider isentropic expansion of turbine at pressure ,

$$P_4 = 0.4 \text{ MPa}$$

From the Superheated refrigerant -134a,
The enthalpies along the vertical line are equal so,

$$s_4 = s_3 = 0.9875 \text{ kJ/kg} \cdot \text{K}$$

By using the interpolation method in between 20°c and 30°c at 0.4 MPa the enthalpy at state 4 is,

$$h_4 = 265.86 + \frac{(275.07 - 265.86)(0.9875 - 0.9628)}{(0.9937 - 0.9628)}$$

$$h_4 = 273.22 \text{ kJ/kg}$$

Step 6 of 7

(a)
Consider the net power output as 750 kJ/s.
The work obtained in the expansion process is,

$$W_{net} = \dot{m} (h_3 - h_4)$$

$$750 = \dot{m} (305.07 - 273.22)$$

$$\dot{m} = 23.54 \text{ kg/s}$$

mass flow rate of R-134a is $\boxed{23.54 \frac{\text{kg}}{\text{s}}}$.

Step 7 of 7

(b)
Thermal efficiency of the Rankin cycle is the net power produced per unit input of heat supplied. So,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)} = \frac{(305.07 - 273.22) - (64.889 - 63.94)}{(305.07 - 64.889)} = 0.128$$

Therefore the efficiency of the cycle is, $\boxed{12.8\ \%}$.

Question: 9.115P

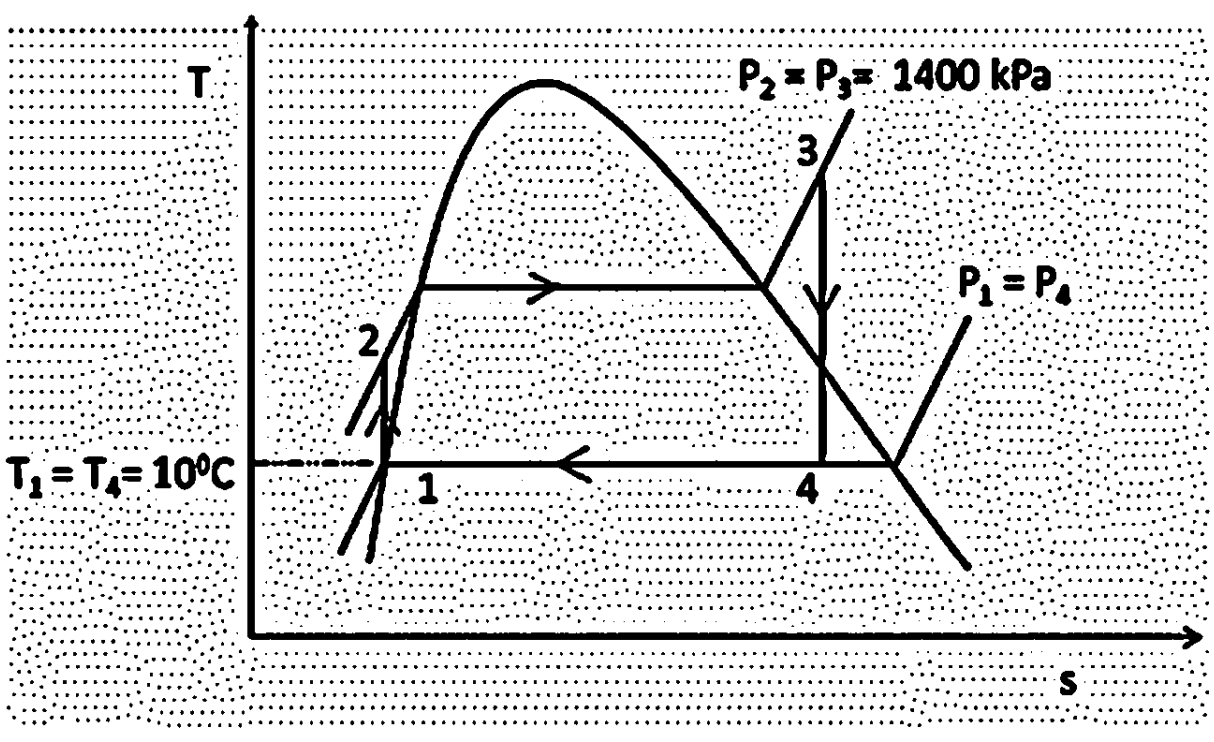
Refrigerant-134a is used as the working fluid in a simple ideal Rankine cycle which operates the boiler at 1400 kPa and the condenser at 10°C. The mixture at the exit of the turbine has a quality of 98 percent. Determine the turbine inlet temperature, the cycle thermal efficiency, and the back-work ratio of this cycle.

Step 1 of 11

Given:
Process: Rankine cycle.
Working fluid: 134a.
The boiler operates pressure $P_2=P_3=1400$ kPa.
The condenser operating Temperature $T_1 = T_4 = 10^\circ C$.
The mixture at the exit of the turbine has a quality of 0.98.

Step 2 of 11

The Ideal Rankin cycle operating between the given conditions is shown below.



Step 3 of 11

State 1:
From the saturated refrigerant-134a
At $T_1 = 10^\circ C$
$P_1 = P_{sat} = 414.89$ kPa
$h_1 = h_f = 65.43$ kJ/kg
$h_g = 256.16$ kJ/kg
$s_1 = s_f = 0.25286$ kJ/kg K
$s_g = 0.92641$ kJ/kg K
$v_1 = v_f = 0.0007930\ m^3/kg$

Step 4 of 11

State 2:
$P_2 = 1.4$ MPa
$= 1400$ kPa
The work done in the compression process is given by,

$$w_{pump} = v_1 \times (P_2 - P_1)$$

$$w_{pump} = 0.0007930\left(\frac{m^3}{kg}\right) \times (1400\ kPa - 414.89\ kPa)\left(\frac{1\ kJ}{1\ (kPa\ m^3)}\right)$$

$$w_{pump} = 0.781\ kJ/kg$$

Step 5 of 11

Enthalpy at state 2 is

$$h_2 = h_1 + w_{pump}$$
$$= 65.43 + 0.781$$
$$= 66.211\ kJ/kg$$

Step 6 of 11

State 4:
From the saturated refrigerant-134a Temperature table
At $T_4 = 10^\circ C$ And the quality is given as $x_4 = 0.98$

$$h_4 = (h_f) + x_4(h_{fg})$$
$$h_4 = 65.43 + 0.98 \times (190.73)$$
$$= 252.35 kJ/kg$$
$$s_4 = (s_f) + x_4(s_{fg})$$
$$s_4 = 0.25286 + 0.98 \times (0.67356)$$
$$= 0.91295\ kJ/kg^\circ K$$

Step 7 of 11

State 3:
Assume the expansion in turbine is isentropic in nature,
Then $s_3 = s_4 = 0.91295\ kJ/kg^\circ K$
From the saturated refrigerant-134a Temperature table
At $P_4 = P_1 = 1400$ kPa

$$\boxed{T_4 = 52.40^0 C}$$

And $h_3 = h_g = 276.12 kJ/kg$

Step 8 of 11

Heat input is

$$q_{in} = h_3 - h_2$$
$$= 276.12 - 66.21$$
$$= 209.91 kJ/kg$$

Step 9 of 11

Heat rejected

$$q_{out} = h_4 - h_1$$
$$= 252.35 - 65.43$$
$$= 186.92 kJ/kg$$

Step 10 of 11

Thermal efficiency of the Rankin cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}}$$
$$= 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{186.92}{209.91}$$
$$= 0.11$$

$$\boxed{\eta_{Rankin} = 11\ \%}$$

Step 11 of 11

Back work ratio

$$r_{bw} = \frac{W_{in}}{W_{Tout}}$$
$$= \frac{W_{Pin}}{h_3 - h_4}$$
$$= \frac{0.781}{(276.12 - 252.35)}$$
$$= 0.03285$$

$$\boxed{r_{bw} = 0.03285}$$

Question: 9.116P pg 1

A simple ideal Rankine cycle which uses water as the working fluid operates its condenser at 40°C and its boiler at 300°C. Calculate the work produced by the turbine, the heat supplied in the boiler, and the thermal efficiency of this cycle when the steam enters the turbine without any superheating,

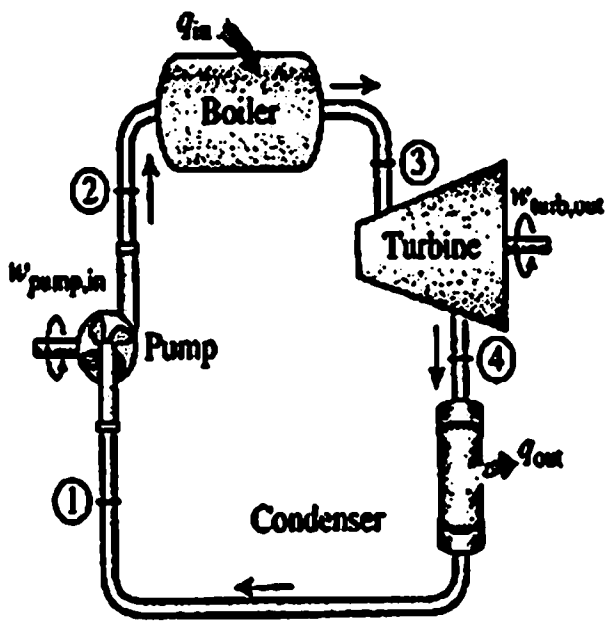
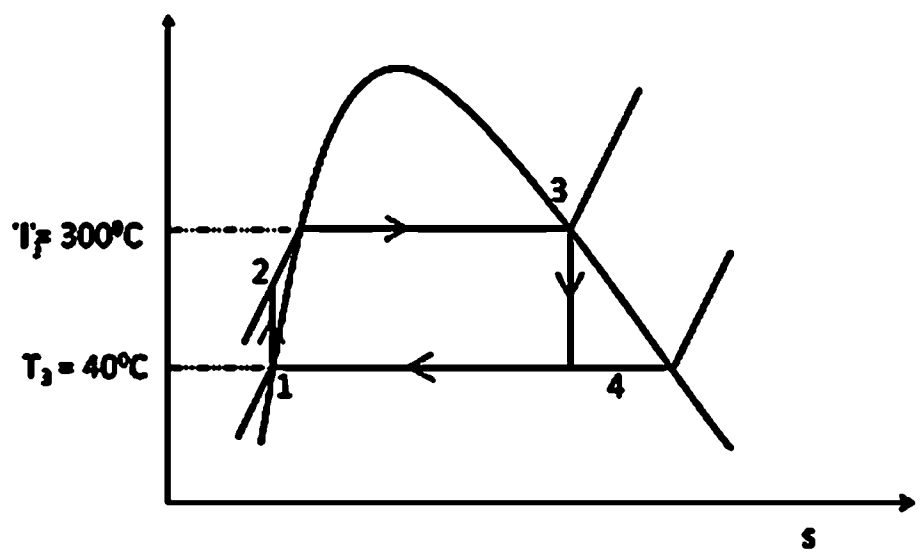


FIGURE P9–116

Step 1 of 10

Draw the different processes of Rankine cycle on T-s diagram as shown below:



Step 2 of 10

At state1 of the cycle:

From the saturated water Temperature table at temperature of 40°C(T_1 = 40°C) the properties of pressure, enthalpy, entropy, and specific volume are,

$$P_1 = P_{sat} = 7.3851 \text{ kPa}$$

$$h_1 = h_f = 167.53 \text{ kJ/kg}$$

$$s_1 = s_f = 0.5724 \text{ kJ/kg} \cdot \text{K}$$

$$v_1 = v_f = 0.001008 \text{ m}^3/\text{kg}$$

Step 3 of 10

At state2 of the cycle:

The pump work done in the compression process is,

$$w_{pump} = v_1 \times (P_2 - P_1)$$

Here, P_2 is the pressure at state 2.

At temperature of $T_3 = 300°\text{C}$ from saturated water table temperature table the pressure P_2 is 8587.9 kPa.

Substitute the corresponding values in the pump work equation.

$$w_{pump} = v_1 \times (P_2 - P_1)$$

$$= 0.001008 \text{ m}^3/\text{kg} \times (8587.9 \text{ kPa} - 7.3851 \text{ kPa})(\text{kJ/ kPa m}^3)$$

$$= 8.649 \text{ kJ/kg}$$

Enthalpy corresponding to state 2 is,

$$h_2 = h_1 + w_{pump}$$

$$= 167.53 + 8.649$$

$$= 176.189 \text{ kJ/kg}$$

Step 4 of 10

At state3 of the cycle:

At temperature of $T_3 = 300°\text{C}$, from saturated water table temperature table the enthalpy and entropy are,

$$h_3 = h_g = 2749.6 \text{ kJ/kg}$$

$$s_3 = s_g = 5.7059 \text{ kJ/kg} \cdot \text{K}$$

Step 5 of 10

At state 4 of the cycle:

Process 3-4 is isentropic expansion of the turbine.

The entropy remains constant during the process 3-4

$$s_4 = s_3 = s_g = 5.7059 \text{ kJ/kg} \cdot \text{K}$$

From the saturated water Temperature table

Find the dryness fraction of the of the liquid-vapor mixture at state 4 by using the equation

$$s_4 = s_f + x_4 s_{fg}$$

Here s_f is the entropy of the saturated liquid, x_4 is the dryness fraction corresponding to point 4 and s_{fg} is the entropy of evaporation.

Substitute corresponding values.

$$s_4 = s_f + x_4 s_{fg}$$

$$5.7059 = 0.5724 + x_4 (7.6832)$$

$$x_4 = 0.6681$$

So, the dryness fraction of liquid-vapor at state 4 is 0.6681.

Step 6 of 10

Calculate the enthalpy at state 4 by using the equation,

$$h_4 = h_f + x_4 h_{fg}$$

Here h_f is the enthalpy of the saturated liquid, x_4 is the dryness fraction corresponding to

Step 6 of 10

Calculate the enthalpy at state 4 by using the equation,

$$h_4 = h_f + x_4 h_{fg}$$

Here h_f is the enthalpy of the saturated liquid, x_4 is the dryness fraction corresponding to point 4 and h_{fg} is the enthalpy of evaporation.

Substitute the corresponding values in the above equation.

$$\begin{aligned} h_4 &= h_f + x_4 h_{fg} \\ &= 167.53 + 0.6681(2406.0) \\ &= 1774.978 \text{kJ/kg} \end{aligned}$$

Step 7 of 10

The work obtained in the turbine is,

$$W_T = (h_3 - h_4)$$

Substitute the corresponding values in the equation above.

$$\begin{aligned} W_T &= (h_3 - h_4) \\ &= 2749.6 - 1774.978 \\ &= 974.622 \text{kJ/kg} \end{aligned}$$

Therefore the work obtained in the turbine is

$\boxed{974.622 \text{ kJ/kg}}$

Step 8 of 10

The amount of the heat supplied to the boiler is,

$$Q_{in} = (h_3 - h_2)$$

Substitute the corresponding values.

$$\begin{aligned} Q_{in} &= (h_3 - h_2) \\ &= (2749.6 - 176.189) \\ &= 2573.411 \text{kJ/kg} \end{aligned}$$

Therefore the heat supplied to the boiler is $\boxed{2573.411 \text{kJ/kg}}$

Step 9 of 10

The amount of Heat rejected in condenser is,

$$Q_{out} = h_4 - h_1$$

Substitute the corresponding values

$$\begin{aligned} Q_{out} &= h_4 - h_1 \\ &= 1774.978 - 167.53 \\ &= 1607.448 \text{kJ/kg} \end{aligned}$$

Step 10 of 10

Find the thermal Efficiency of the Rankine cycle by using the equation,

$$\begin{aligned} \eta_{Rankine} &= 1 - \frac{Q_{out}}{Q_{in}} \\ &= 1 - \frac{1607.448}{2573.411} \\ &= 0.375 \\ &= 0.375 \times 100 \\ &= 37.5\% \end{aligned}$$

Therefore the thermal efficiency of the Rankine cycle is $\boxed{37.5\ \%}$

Question:

9.117P

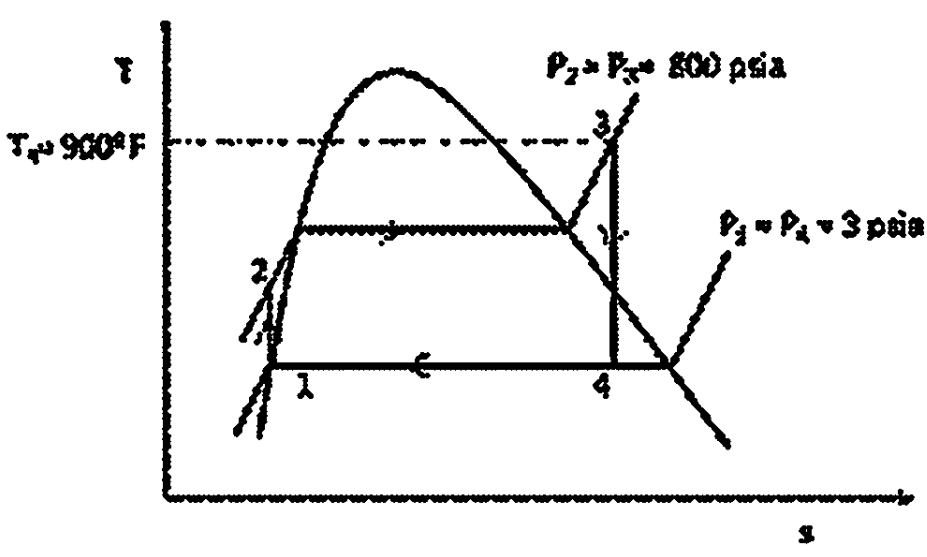
The turbine of a steam power plant operating on a simple ideal Rankine cycle produces 1750 kW of power when the boiler is operated at 800 psia, the condenser at 3 psia, and the temperature at the turbine entrance is 900°F. Determine the rate of heat supply in the boiler, the rate of heat rejection in the condenser, and the thermal efficiency of the cycle.

Step 1 of 11

Given,
Process: Rankine cycle.
The boiler operates Pressure $P_2 = P_3 = 800\text{psia}$.
The condenser Pressure $P_1 = P_4 = 3\text{psia}$.
Turbine inlet temperature $T_3 = 900^0\text{F}$.
Power produced by the cycle = 1750 kW.

Step 2 of 11

The Ideal Rankin cycle operating between the given conditions is shown below.



Step 3 of 11

State1:
From the saturated water pressure table
$P_1 = 3$ psia At this pressure,
$h_1 = h_f = 109.40$ Btu/lb
$v_1 = v_f = 0.01630\ \text{ft}^3/\text{lb}$

Step 4 of 11

State2:
Pressure at state 2 is $P_2 = 800$ psia
The work done in the compression process is given by,

$$w_{pump} = v_1 \times (P_2 - P_1)$$
$$w_{pump} = 0.01630\ (\text{ft}^3/\text{lb}) \times (800 - 3)\text{psia}$$
$$w_{pump} = 13\ (\text{psia ft}^3/\text{lb})(1\text{Btu}/5.40395\ \text{psia}\cdot\text{ft}^3)$$
$$= 2.4\ \text{Btu/lbm}$$

Step 5 of 11

Enthalpy at state 2 is

$$h_2 = h_1 + w_{pump}$$
$$= (109.40 + 2.4)$$
$$= 111.8\text{Btu/lbm}$$

Step 6 of 11

State3:
From the superheated water
$P_3 = P_2 = 800$ psia And $T_3 = 900^0$F at this pressure and temperature,
$h_3 = 1456$ Btu/lbm
$s_3 = 1.6413\text{Btu/lbm}\cdot\text{R}$

Step 7 of 11

State4:
Assume the expansion in turbine is isentropic in nature,
then $s_3 = s_4$
$s_4 = s_3 = 1.6413\text{Btu/lbm}\cdot\text{R}$
From the saturated water pressure table
At $P_4 = P_1 = 3$ psia and $s_4 = s_3 = 1.6413\text{Btu/lbm}\cdot\text{R}$

$$s_4 = s_f + x_4 s_{fg}$$
$$1.6413 = 0.20090 + x_4 (1.6849)$$
$$x_4 = 0.8549$$

Step 8 of 11

Enthalpy at state 4 is

$$h_4 = h_f + x_4 h_{fg}$$
$$= 109.40 + 0.8549(1012.8)$$
$$= 975.24\text{Btu/lbm}$$

Step 9 of 11

power output from the turbine = 1750 kW

$$W_{net} = \dot{m}\ (h_3 - h_4)$$
$$(1750\ \text{kW})\left(\frac{3412.14(\text{Btu/h})}{1\ \text{kW}}\right) = \dot{m}\ (1456.0 - 975.24)\ \text{Btu/lbm}$$
$$\dot{m} = 12418.1\ (\text{lbm/h})(1\text{h}/3600\ \text{s})$$
$$= 3.45\ \text{lbm/s}$$
$$\boxed{\dot{m} = 3.45\ \text{lbm/s}}$$

Step 10 of 11

Rate flow of heat supply is given by,

$$Q_{in} = \dot{m}\ (h_3 - h_2)$$
$$Q_{in} = 3.45 \times (1456.0 - 111.81)$$
$$= 4637.45\ (\text{Btu/s})(3600/1\text{h})(1\text{kW}/3412.14\text{Btu/h})$$
$$= 4892.776\text{kW}$$
$$\boxed{Q_{in} = 4892.766\ \text{kW}}$$

Step 11 of 11

Thermal efficiency of the Rankin cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}}$$
$$= \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}$$
$$= \frac{(1456.0 - 975.24) - (111.81 - 109.40)}{(1456.0 - 111.81)}$$
$$= 0.355$$
$$= 35.5\%$$
$$\boxed{\eta_{Rankin} = 35.5\%}$$

Question: 9.118P

A simple ideal Rankine cycle with water as the working fluid operates between the pressure limits of 2500 psia in the boiler and 5 psia in the condenser. What is the minimum temperature required at the turbine inlet such that the quality of the steam leaving the turbine is not below 80 percent. When operated at this temperature, what is the thermal efficiency of this cycle?

Step 1 of 11

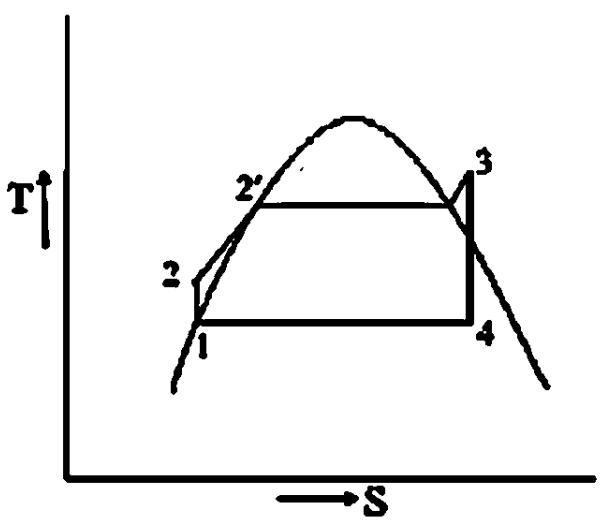
Given

High pressure in the boiler $P_3 = P_2 = 2500\,\text{psia}$

Low pressure in the condenser $P_4 = P_1 = 5\,\text{psia}$

Quality at the exit of the turbine $x_4 = 80\%$

Step 2 of 11



Step 3 of 11

At $P_4 = 5\,\text{psia}$ from saturated water tables

$h_f = 130.18\,\text{Btu/lbm}$

$h_{fg} = 1000.5\,\text{Btu/lbm}$

$s_f = 0.23488\,\text{Btu/lbm.R}$

$s_{fg} = 1.60894\,\text{Btu/lbm.R}$

Step 4 of 11

Enthalpy at state-4

$h_4 = h_f + x_4 h_{fg}$

$h_4 = 130.18 + 0.80 \times 1000.5$

$h_4 = 930.58\,\text{Btu/lbm}$

Step 5 of 11

Similarly entropy at state-4

$s_4 = s_f + x_4 s_{fg}$

$s_4 = 0.23488 + 0.80 \times 1.60894$

$s_4 = 1.522\,\text{Btu/lbm.R}$

Step 6 of 11

The expansion process through the turbine is isentropic

At $P_3 = 2500\,\text{psia}$ and $s_3 = s_4 = 1.522\,\text{Btu/lbm.R}$ from superheated water tables

$$h_3 = 1423.3 + (1458.2 - 1423.3)\left(\frac{1.522 - 1.5028}{1.5271 - 1.5028}\right)$$

$h_3 = 1450.875\,\text{Btu/lbm}$

$$T_3 = 950 + (1000 - 950)\left(\frac{1.522 - 1.5028}{1.5271 - 1.5028}\right)$$

$T_3 = 989.506\,^0\text{R}$

Therefore minimum temperature required at turbine inlet is

$\boxed{T_3 = 989.506\,^0\text{R}}$

Step 7 of 11

Enthalpy at state-1

$h_1 = h_{f@5\text{Psia}} = 130.18\,\text{Btu/lbm}$

Specific volume at state-1

$v_1 = v_{f@5\text{Psia}} = 0.01641\,\text{m}^3/\text{kg}$

Pump work

$w_{pump} = -v_{f@5\text{psia}}(P_2 - P_1)$

$$w_{pump} = -0.010641\frac{\text{ft}^3}{\text{lbm}} \times (2500 - 5)\frac{\text{lbf}}{\text{in}^2} \times 144\frac{\text{in}^2}{\text{ft}^2} \times \frac{1}{778}\frac{\text{Btu}}{\text{lbf-ft}}$$

$w_{pump} = -7.578\,\text{Btu/lbm}$

Step 8 of 11

By applying first law to the process 1-2

$h_1 = h_2 + w_{pump}$

$h_2 = h_1 - w_{pump}$

$h_2 = 130.18 + 7.578$

$h_2 = 137.758\,\text{Btu/lbm}$

Step 9 of 11

Heat addition in the boiler

$q_{in} = (h_3 - h_2)$

$q_{in} = 1450.875 - 137.758$

$q_{in} = 1313.117\,\text{Btu/lbm}$

Step 10 of 11

Heat rejection in the condenser

$q_{out} = (h_4 - h_1)$

$q_{out} = 930.58 - 130.18$

$q_{out} = 800.4\,\text{Btu/lbm}$

Step 11 of 11

Thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - \frac{800.4}{1313.117}$$

$\boxed{\eta_{th} = 39.045\%}$

Question: 9.119P pg 1

A steam Rankine cycle operates between the pressure limits of 2500 psia in the boiler and 1 psia in the condenser. The turbine inlet temperature is 800°F. The turbine isentropic efficiency is 90 percent, the pump losses are negligible, and the cycle is sized to produce 1000 kW of power. Calculate the mass flow rate through the boiler, the power produced by the turbine, the rate of heat supply in the boiler, and the thermal efficiency.

Step 1 of 16

Given

Boiler pressure $P_3 = P_2 = 2500\,\text{psia}$

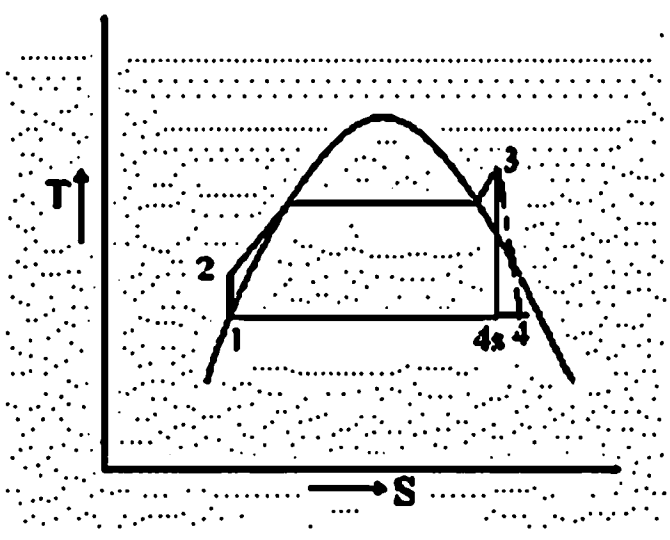
Condenser pressure $P_4 = P_1 = 1\,\text{psia}$

Turbine inlet temperature $T_3 = 800^0\text{F}$

Turbine isentropic efficiency $\eta_T = 90\%$

Power developed $\dot{W}_{net} = 1000\,\text{kW}$

Step 2 of 16



Step 3 of 16

At $P_1 = 1\,\text{psia}$ from saturated water tables

$h_1 = h_f = 69.72\ \text{Btu/lbm}$

$h_{fg} = 1035.7\,\text{Btu/lbm}$

$s_f = 0.13262\,\text{Btu/lbm.R}$

$s_{fg} = 1.84495\,\text{Btu/lbm.R}$

$v_1 = v_{f\,@\,1\,psia} = 0.01614\ \text{ft}^3\text{/lbm}$

Step 4 of 16

Pump work

$$w_{pump} = -v_1(P_2 - P_1)$$

$$w_{pump} = -0.01614\ \text{ft}^3\text{/lbm} \times (2500-1)\frac{\text{lbf}}{\text{in}^2} \times 144 \times \frac{\text{in}^2}{\text{ft}^2} \times \frac{1}{778}\frac{\text{Btu}}{\text{lbf-ft}}$$

$$w_{pump} = -7.46\,\text{Btu/lbm}$$

Step 5 of 16

Applying energy equation to the process 1-2

$$h_1 = h_2 + w_{pump}$$

$$h_2 = h_1 - w_{pump}$$

$$h_2 = 69.72 + 7.46$$

$$h_2 = 77.18\ \text{Btu/lbm}$$

Step 6 of 16

At $P_3 = 2500$ psia and $T_3 = 800^0$ F from superheated water tables

$h_3 = 1302\ \text{Btu/lbm}$

$s_3 = 1.4116\ \text{Btu/lbm.R}$

Step 7 of 16

Process 3-4s is isentropic process entropy remains constant

$$s_4 = s_3$$

$$s_3 = s_{f@1psia} + x_4 s_{fg@psia}$$

$$x_{4s} = \frac{(s_4 - s_f)}{s_{fg}}$$

$$x_{4s} = \frac{(1.4116 - 0.13262)}{1.84495}$$

$$x_{4s} = 0.6932$$

Step 8 of 16

Enthalpy at state-4s,

$$h_{4s} = h_f + x_{4s} h_{fg}$$

$$h_{4s} = 69.72 + (0.6932 \times 1035.7)$$

$$h_{4s} = 787.7\ \text{Btu/lbm}$$

Step 9 of 16

Isentropic efficiency of the turbine is defined as,

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

$$h_4 = 1302 - (0.9 \times (1302 - 787.7))$$

$$h_4 = 839.13\ \text{Btu/lbm}$$

Step 10 of 16

Heat addition in the boiler is given by

$$q_{in} = h_3 - h_2$$

$$q_{in} = 1302 - 77.18$$

$$q_{in} = 1224.82\ \text{Btu/lbm}$$

$q_{in} = h_3 - h_2$

$q_{in} = 1302 - 77.18$

$q_{in} = 1224.82 \text{ Btu/lbm}$

Step 11 of 16

Heat rejected is given by

$q_{out} = h_4 - h_1$

$q_{out} = 839.13 - 69.72$

$q_{out} = 769.41 \text{ Btu/lbm}$

Step 12 of 16

Network output is defined as

$w_{net} = q_{in} - q_{out}$

$w_{net} = 1224.82 - 769.41$

$w_{net} = 455.41 \text{ Btu/lbm}$

Step 13 of 16

The mass flow rate of steam can be obtained from the relation,

$W_{net} = \dot{m} w_{net}$

$$\dot{m} = \frac{W_{net}}{w_{net}}$$

$$\dot{m} = \frac{1000 \text{ kJ/s}}{455.41 \text{ Btu/lbm}} \left(\frac{0.9478 \text{Btu}}{1 \text{kJ}} \right)$$

$\boxed{\dot{m} = 2.081 \text{ lbm/s}}$

Step 14 of 16

The power output from the turbine

$W_T = \dot{m}(h_3 - h_4)$

$$W_T = 2.081 \frac{\text{lbm}}{\text{s}} \times (1302 - 839.13) \frac{\text{Btu}}{\text{lbm}} \times \frac{1 \text{kJ}}{0.9478 \text{Btu}}$$

$\boxed{W_T = 1016.28 \text{ kW}}$

Step 15 of 16

The rate of heat addition in the boiler is given by

$Q_{in} = \dot{m} q_{in}$

$Q_{in} = 2.081 \times (1224.82)$

$\boxed{Q_{in} = 2548.85 \text{Btu/s}}$

$$Q_{in} = 2548.85 \frac{\text{Btu}}{\text{s}} \times \frac{1}{0.9478} \times \frac{\text{kJ}}{\text{Btu}}$$

$\boxed{Q_{in} = 2689.227 \text{kW}}$

Step 16 of 16

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

$$\eta_{th} = \frac{1000}{2689.227}$$

$$= 0.3718$$

$\boxed{\eta_{th} = 37.18\%}$

Question:

Reconsider Prob. 10—119E. How much error is caused in the thermal efficiency if the power required by the pump were completely neglected?

Step 1 of 21

Given

Boiler pressure $P_3 = P_2 = 2500\,\text{psia}$

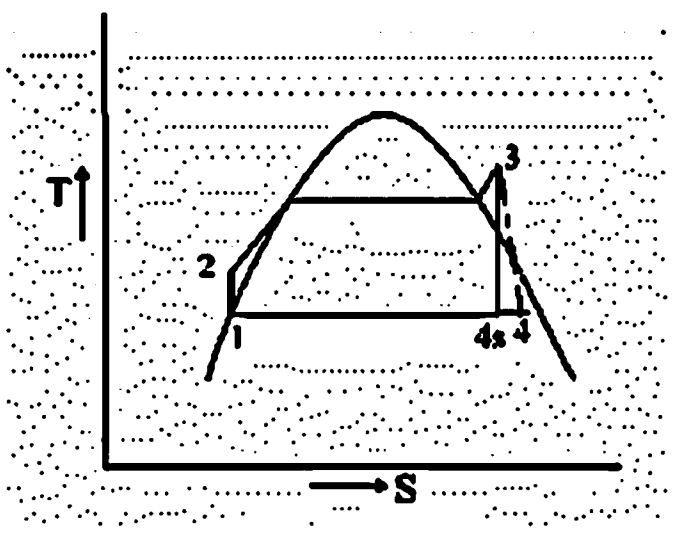
Condenser pressure $P_4 = P_1 = 1\,\text{psia}$

Turbine inlet temperature $T_3 = 800^0\text{F}$

Turbine isentropic efficiency $\eta_T = 90\%$

Power developed $\dot{W}_{net} = 1000\,\text{kW}$

Step 2 of 21



Step 3 of 21

At $P_1 = 1\,\text{psia}$ from saturated water tables

$h_1 = h_f = 69.72\ \text{Btu/lbm}$

$h_{fg} = 1035.7\,\text{Btu/lbm}$

$s_f = 0.13262\,\text{Btu/lbm.R}$

$s_{fg} = 1.84495\,\text{Btu/lbm.R}$

$v_1 = v_{f\,@\,1\text{psia}} = 0.01614\ \text{ft}^3\text{/lbm}$

Step 4 of 21

Pump work

$$w_{pump} = -v_1(P_2 - P_1)$$

$$w_{pump} = -0.01614\ \text{ft}^3\text{/lbm} \times (2500-1)\frac{\text{lbf}}{\text{in}^2} \times 144 \times \frac{\text{in}^2}{\text{ft}^2} \times \frac{1}{778}\frac{\text{Btu}}{\text{lbf-ft}}$$

$$w_{pump} = -7.46\,\text{Btu/lbm}$$

Step 5 of 21

Applying energy equation to the process 1-2

$$h_1 = h_2 + w_{pump}$$

$$h_2 = h_1 - w_{pump}$$

$$h_2 = 69.72 + 7.46$$

$$h_2 = 77.18\ \text{Btu/lbm}$$

Step 6 of 21

At $P_3 = 2500$ psia and $T_3 = 800^0$F from superheated water tables

$h_3 = 1302\ \text{Btu/lbm}$

$s_3 = 1.4116\ \text{Btu/lbm.R}$

Step 7 of 21

Process 3-4s is isentropic process entropy remains constant

$$s_4 = s_3$$

$$s_3 = s_{f@1\text{psia}} + x_4 s_{fg@\text{psia}}$$

$$x_{4s} = \frac{(s_4 - s_f)}{s_{fg}}$$

$$x_{4s} = \frac{(1.4116 - 0.13262)}{1.84495}$$

$$x_{4s} = 0.6932$$

Step 8 of 21

Enthalpy at state-4s,

$$h_{4s} = h_f + x_{4s} h_{fg}$$

$$h_{4s} = 69.72 + (0.6932 \times 1035.7)$$

$$h_{4s} = 787.7\ \text{Btu/lbm}$$

Step 9 of 21

Isentropic efficiency of the turbine is defined as,

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

$$h_4 = 1302 - (0.9 \times (1302 - 787.7))$$

$$h_4 = 839.13\ \text{Btu/lbm}$$

Step 10 of 21

Heat addition in the boiler is given by

$$q_{in} = h_3 - h_2$$

$$q_{in} = 1302 - 77.18$$

$$q_{in} = 1224.82\ \text{Btu/lbm}$$

Step 11 of 21

Heat rejected is given by

$$q_{out} = h_4 - h_1$$

Step 11 of 21

Heat rejected is given by

$$q_{out} = h_4 - h_1$$
$$q_{out} = 839.13 - 69.72$$
$$q_{out} = 769.41 \text{ Btu/lbm}$$

Step 12 of 21

Work output from the turbine

$$w_T = (h_3 - h_4)$$
$$= 1302 - 839.13$$
$$w_T = 462.87 \text{ Btu/lbm}$$

Step 13 of 21

The mass flow rate of steam can be obtained from the relation,

$$W_{net} = \dot{m} w_{net}$$
$$\dot{m} = \frac{W_{net}}{w_T}$$
$$\dot{m} = \frac{1000 \text{ kJ/s}}{462.87 \text{ Btu/lbm}} \left(\frac{0.9478 \text{Btu}}{1 \text{kJ}} \right)$$
$$\dot{m} = 2.047 \text{ lbm/s}$$

Step 14 of 21

The power output from the turbine

$$W_T = \dot{m}(h_3 - h_4)$$
$$W_T = 2.047 \frac{\text{lbm}}{s} \times (1302 - 839.13) \frac{\text{Btu}}{\text{lbm}} \times \frac{1 \text{kJ}}{0.9478 \text{Btu}}$$
$$W_T = 999.678 \text{ kW}$$

Step 15 of 21

The rate of heat addition in the boiler is given by

$$Q_{in} = \dot{m} q_{in}$$
$$Q_{in} = 2.047 \times (1224.82)$$
$$Q_{in} = 2507.206 \text{Btu/s}$$
$$Q_{in} = 2507.206 \frac{\text{Btu}}{s} \times \frac{1}{0.9478} \times \frac{\text{kJ}}{\text{Btu}}$$
$$Q_{in} = 2645.29 \text{kW}$$

Step 16 of 21

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$
$$\eta_{th} = \frac{1000}{2645.29}$$
$$= 0.378$$
$$\eta_{th} = 37.8\%$$

Step 17 of 21

When pump work is considered

Network output is defined as

$$w_{net} = q_{in} - q_{out}$$
$$w_{net} = 1224.82 - 769.41$$
$$w_{net} = 455.41 \text{ Btu/lbm}$$

Step 18 of 21

The mass flow rate of steam can be obtained from the relation,

$$W_{net} = \dot{m} w_{net}$$
$$\dot{m} = \frac{W_{net}}{w_{net}}$$
$$\dot{m} = \frac{1000 \text{ kJ/s}}{455.41 \text{ Btu/lbm}} \left(\frac{0.9478 \text{Btu}}{1 \text{kJ}} \right)$$
$$\dot{m} = 2.081 \text{ lbm/s}$$

Step 19 of 21

The rate of heat addition in the boiler is given by

$$Q_{in} = \dot{m} q_{in}$$
$$Q_{in} = 2.081 \times (1224.82)$$
$$Q_{in} = 2548.85 \text{Btu/s}$$
$$Q_{in} = 2548.85 \frac{\text{Btu}}{s} \times \frac{1}{0.9478} \times \frac{\text{kJ}}{\text{Btu}}$$
$$Q_{in} = 2689.227 \text{kW}$$

Step 20 of 21

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$
$$\eta_{th} = \frac{1000}{2689.227}$$
$$= 0.3718$$
$$\eta_{th} = 37.18\%$$

Step 21 of 21

Thermal efficiency obtained when considered the pump work is

$$\eta_{th1} = 0.3718$$

Thermal efficiency obtained when neglecting the pump work is

$$\eta_{th2} = 0.378$$

Error caused in the thermal efficiency is

Thermal efficiency obtained when considered the pump work is

$\eta_{th1} = 0.3718$

Thermal efficiency obtained when neglecting the pump work is

$\eta_{th2} = 0.378$

Error caused in the thermal efficiency is

$$error = \frac{\eta_{th2} - \eta_{th1}}{\eta_{th1}}$$

$$error = \frac{0.378\text{-}0.3718}{0.3718} \times 100$$

$$\boxed{error = 1.66\%}$$

Question: 9.121P

Consider a steam power plant that operates on a simple ideal Rankine cycle and has a net power output of 45 MW. Steam enters the turbine at 7 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa by running cooling water from a lake through the tubes of the condenser at a rate of 2000 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine *(a)* the thermal efficiency of the cycle, (b) the mass flow rate of the steam, and (c) the temperature rise of the cooling water.

Step 1 of 13

Given

Turbine inlet pressure $P_3 = 7\text{MPa}$

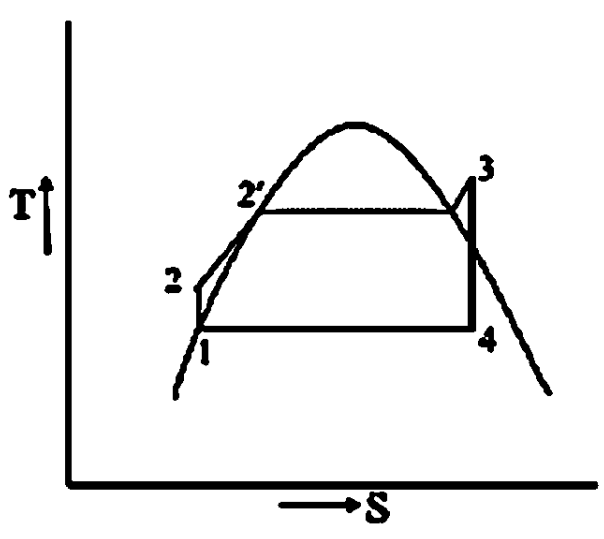
Turbine inlet temperature $T_3 = 500^0\text{C}$

Condenser pressure $P_4 = P_1 = 10\text{kPa}$

Net power output $W_{net} = 45\text{MW}$

Mass flow rate of cooling water $\dot{m}_w = 2000\text{kg/s}$

Step 2 of 13



Step 3 of 13

At $P_1 = 10\text{kPa}$ from saturated water tables

$h_1 = h_{f,10kPa} = 191.81\text{ kJ/kg}$

$v_1 = v_{f,10kPa} = 0.00101\text{ m}^3\text{/kg}$

$h_{fg} = 2392.1\text{kJ/kg}$

$s_f = 0.6492\text{kJ/kg.K}$

$s_{fg} = 7.4996\text{kJ/kg.K}$

Step 4 of 13

Pump work during the process 1-2 is

$w_{pump} = -v_1(P_2 - P_1)$

$w_{pump} = -0.00101 \times (7000 - 10)$

$w_{pump} = -7.06\text{kJ/kg}$

Step 5 of 13

Applying first law to the process 1-2

$h_1 = h_2 + w_{pump}$

$h_2 = h_1 - w_{pump}$

$h_2 = 191.81 + 7.06$

$h_2 = 198.87\text{ kJ/kg}$

Step 6 of 13

At $P_3 = 7\text{MPa}$ and $T_3 = 500^0\text{C}$ from superheated water tables

$h_3 = 3411.4\text{ kJ/kg}$

$s_3 = 6.8\text{ kJ/kg.K}$

Step 7 of 13

Process 3-4 is isentropic expansion

$s_3 = s_4$

$s_3 = s_f + x_4 s_{fg}$

$x_4 = \dfrac{(s_3 - s_f)}{s_{fg}}$

$x_4 = \dfrac{(6.8 - 0.649)}{7.4996}$

$x_4 = 0.82$

Step 8 of 13

Enthalpy at state-4

$h_4 = h_f + x_4 h_{fg}$

$h_4 = 191.81 + (0.82 \times 2392.1)$

$h_4 = 2153.332\text{ kJ/kg}$

Step 9 of 13

Heat addition is given by

$q_{in} = h_3 - h_2$

$q_{in} = 3411.4 - 198.87$

$q_{in} = 3212.53\text{ kJ/kg}$

Step 10 of 13

Heat rejected in the condenser is

$q_{out} = h_4 - h_1$

$q_{out} = 2153.332 - 191.81$

$q_{out} = 1961.522\text{ kJ/kg}$

Step 11 of 13

(a)

Thermal efficiency of the cycle,

$\eta_{th} = \dfrac{w_{net}}{q_{in}}$

$\eta_{th} = \dfrac{q_{in} - q_{out}}{q_{in}}$

$\eta_{th} = \dfrac{3212.53 - 1961.522}{3212.53}$

$= 0.3894$

$\boxed{\eta_{th} = 38.94\%}$

Step 12 of 13

(b)

Steam mass flow rate is given by

$\dot{m} = \dfrac{W_{net}}{w_{net}}$

$\dot{m} = \dfrac{W_{net}}{q_{in} - q_{out}}$

$\dot{m} = \dfrac{45000}{3212.53 - 1961.522}$

$\boxed{\dot{m} = 35.97\text{ kg/s}}$

Step 13 of 13

(c)

Mass flow rate of the water

By applying energy balance to the condenser

$Q_{out} = m_w \times C_y \Delta T_{coolingwater}$

Where specific heat of water is

$C_y = 4.18\text{kJ/kg.K}$

The temperature rise of water is

$\Delta T_{coolingwater} = \dfrac{Q_{out}}{m_w \times C_y}$

$\Delta T_{coolingwater} = \dfrac{\dot{m} q_{out}}{m_w \times C_y}$

$\Delta T_{coolingwater} = \dfrac{35.97 \times 1961.522}{2000 \times 4.18}$

$\boxed{\Delta T_{coolingwater} = 8.439^0\text{C}}$

Question: 9.122P pg 1

Repeat Prob. 9–121 assuming an isentropic efficiency of 87 percent for both the turbine and the pump

Step 1 of 15

Given

Turbine inlet pressure $P_3 = 7\text{MPa}$

Turbine inlet temperature $T_3 = 500^0\text{C}$

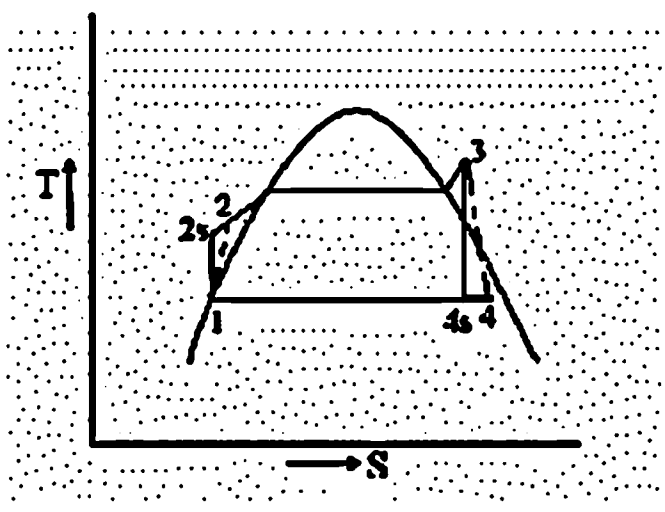
Condenser pressure $P_4 = P_1 = 10\text{kPa}$

Net power output $W_{net} = 45\text{MW}$

Mass flow rate of cooling water $\dot{m}_w = 2000\text{kg/s}$

Isentropic efficiency of turbine and pump is $\eta_T = \eta_p = 87\%$

Step 2 of 15



Step 3 of 15

At $P_1 = 10\text{kPa}$ from saturated water tables

$h_1 = h_{f,10kPa} = 191.81\text{ kJ/kg}$

$v_1 = v_{f,10kPa} = 0.00101\text{ m}^3\text{/kg}$

$h_{fg} = 2392.1\text{kJ/kg}$

$s_f = 0.6492\text{kJ/kg.K}$

$s_{fg} = 7.4996\text{kJ/kg.K}$

Step 4 of 15

Isentropic pump work during the process 1-2s is

$w_{isenp} = -v_1(P_{2s} - P_1)$

$w_{isenp} = -0.00101 \times (7000 - 10)$

$w_{isenp} = -7.06\text{kJ/kg}$

Step 5 of 15

Isentropic efficiency of the pump is

$$\eta_p = \frac{w_{isenp}}{w_{actp}}$$

$$0.87 = \frac{-7.06}{w_{actp}}$$

$$w_{actp} = -8.114\text{kJ/kg}$$

Step 6 of 15

Applying first law to the process 1-2

$h_1 = h_2 + w_{actp}$

$h_2 = h_1 - w_{actp}$

$h_2 = 191.81 + 8.114$

$h_2 = 199.924\text{ kJ/kg}$

Step 7 of 15

At $P_3 = 7\text{MPa}$ and $T_3 = 500^0\text{C}$ from superheated water tables

$h_3 = 3411.4\text{ kJ/kg}$

$s_3 = 6.8\text{ kJ/kg.K}$

Step 8 of 15

Process 3-4s is isentropic expansion

$s_3 = s_{4s}$

$s_3 = s_f + x_{4s}s_{fg}$

$$x_{4s} = \frac{(s_3 - s_f)}{s_{fg}}$$

$$x_{4s} = \frac{(6.8 - 0.649)}{7.4996}$$

$x_{4s} = 0.82$

Step 9 of 15

Enthalpy at state-4s

$h_{4s} = h_f + x_{4s}h_{fg}$

$h_{4s} = 191.81 + (0.82 \times 2392.1)$

$h_{4s} = 2153.332\text{ kJ/kg}$

Step 10 of 15

Isentropic efficiency of the turbine is

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$0.87 = \frac{3411.4 - h_4}{3411.4 - 2153.332}$$

$h_4 = 2316.88\text{kJ/kg}$

Step 11 of 15

$$h_4 = \frac{3411.4 - 2153.332}{}$$
$h_4 = 2316.88\,\text{kJ/kg}$

9.122P pg 2

Step 11 of 15

Heat addition is given by

$$q_{in} = h_3 - h_2$$
$$q_{in} = 3411.4 - 199.924$$
$$q_{in} = 3211.476\,\text{kJ/kg}$$

Step 12 of 15

Heat rejected in the condenser is

$$q_{out} = h_4 - h_1$$
$$q_{out} = 2316.88 - 191.81$$
$$q_{out} = 2125.07\,\text{kJ/kg}$$

Step 13 of 15

(a)
Thermal efficiency of the cycle,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$
$$\eta_{th} = \frac{q_{in} - q_{out}}{q_{in}}$$
$$\eta_{th} = \frac{3211.476 - 2125.07}{3211.476}$$
$$= 0.3382$$
$$\boxed{\eta_{th} = 33.82\%}$$

Step 14 of 15

(b)
Steam mass flow rate is given by

$$\dot{m} = \frac{W_{net}}{w_{net}}$$
$$\dot{m} = \frac{W_{net}}{q_{in} - q_{out}}$$
$$\dot{m} = \frac{45000}{3211.476 - 2125.07}$$
$$\boxed{\dot{m} = 41.42\,\text{kg/s}}$$

Step 15 of 15

(c)
Mass flow rate of the water
By applying energy balance to the condenser

$$Q_{out} = m_w \times C_p \Delta T_{coolingwater}$$

Where specific heat of water is

$$C_p = 4.18\text{kJ/kg.K}$$

The temperature rise of water is

$$\Delta T_{coolingwater} = \frac{Q_{out}}{m_w \times C_p}$$
$$\Delta T_{coolingwater} = \frac{\dot{m} q_{out}}{m_w \times C_p}$$
$$\Delta T_{coolingwater} = \frac{41.42 \times 2125.07}{2000 \times 4.18}$$
$$\boxed{\Delta T_{coolingwater} = 10.528^0\text{C}}$$

Question: 9.123P pg 1

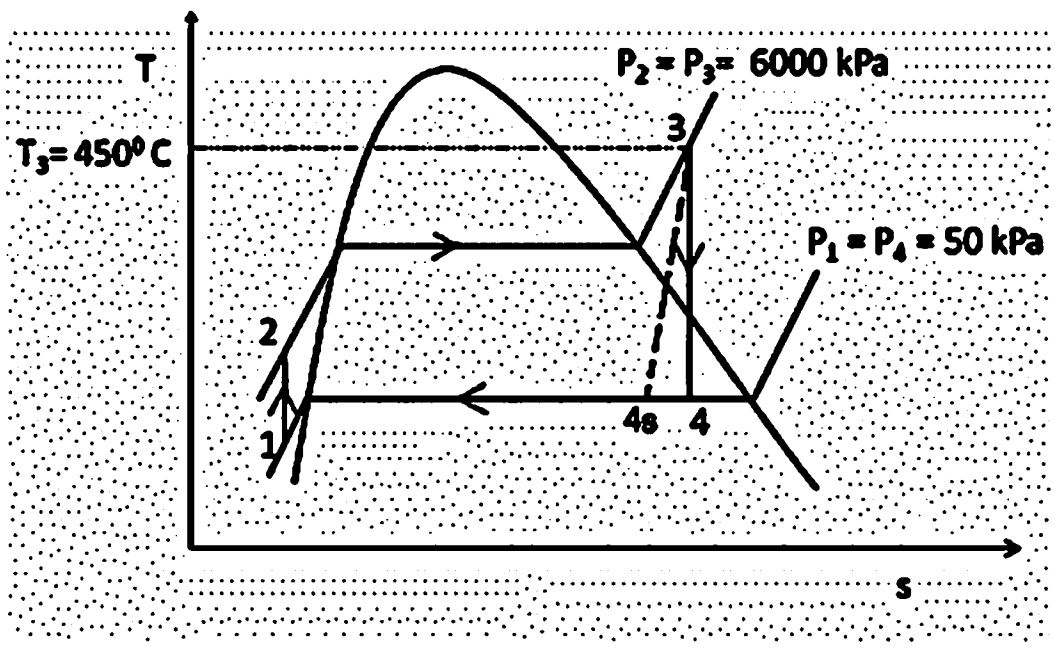
A simple Rankine cycle uses water as the working fluid. The boiler operates at 6000 kPa and the condenser at 50 kPa. At the entrance to the turbine, the temperature is 450°C. The isentropic efficiency of the turbine is 94 percent, pressure and pump losses are negligible, and the water leaving the condenser is subcooled by 6.3°C. The boiler is sized for a mass flow rate of 20 kg/s. Determine the rate at which heat is added in the boiler, the power required to operate the pumps, the net power produced by the cycle, and the thermal efficiency. Answers:59,660 kW, 122 kW, 18,050 kW, 30.3 percent.

Step 1 of 12

Given:
Process: Rankine cycle.
Working fluid: water.
The boiler operates Pressure $P_2 = P_3 = 6000\,\text{kPa}$.
The condenser operating pressure $P_1 = P_4 = 50\text{kPa}$.
Turbine inlet temperature $T_3 = 450\,^\circ\text{C}$.
Isentropic efficiency of the turbine $\eta_T = 0.94$.
Water leaving the condenser is sub cooled by 6.3^0C.
Mass flow rate $m = 20\ \text{kg}/s$.
Pressure drop across the boiler $P_{drop} = 50\text{kPa}$.

Step 2 of 12

The Ideal Rankin cycle



Step 3 of 12

State1:
From the Saturated water pressure table
$P_1 = 50\ \text{kPa}$
$(T_{sat}) = 81.32^\circ\text{C}$
$(h_f) = 340.54\ \text{kJ/kg}$
$(v_f) = 0.001030\ \text{m}^3/\text{kg}$
From the above quantities,
At this pressure and temperature is sub cooled by 6.3^0C,
Then,
$T_1 = T_{sat} - 6.3^\circ\text{C}$
$= 81.3 - 6.3$
$= 75^\circ\text{C}$
Now from the Saturated water temperature table
At $T_1 = 75^\circ\text{C}$
$h_1 = h_f = 314.03\ \text{kJ/kg}$

Step 4 of 12

State 2:
Exit pressure of the pump $P_2 = 6000\ \text{kPa}$
The work done in the compression process is given by,
$w_{pump} = v_1 \times (P_2 - P_1)$
$w_{pump} = 0.001030 \times (6000\ \text{kPa} - 50\ \text{kPa})$
$w_{pump} = 6.12\text{kJ/kg}$

Step 5 of 12

Enthalpy at state 2 is
$h_2 = h_1 + w_{pump}$
$= 314.03 + 6.1$
$= 320.15\text{kJ/kg}$

Step 6 of 12

State 3:
From the superheated water table
At $P_3 = P_2 = 6000\ \text{kPa}$ And $T_3 = 450^\circ\text{C}$
$h_3 = 3302.9\ \text{kJ/kg}$
$s_3 = 6.7219\ \text{kJ/kg} \cdot \text{K}$

Step 7 of 12

State4:
Ideally the expansion in turbine is isentropic in nature, then $s_4 = s_3 = 6.7219\ \text{kJ/kg} \cdot \text{K}$
From the saturated water pressure table
At $P_4 = P_1 = 50\ \text{kPa}$ and $s_4 = s_3 = 6.7219\ \text{kJ/kg} \cdot \text{K}$
$s_4 = s_f + x_4 s_{fg}$
$6.7219 = 1.0912 + x_4 (6.5019)$
$x_4 = 0.866$
$h_4 = h_f + x_4 h_{fg}$
$= 340.54 + 0.866(2304.7)$
$= 2336.410\text{kJ/kg}$

Step 8 of 12

Work obtained from the turbine (ideally)

$= 2336.410 \text{ kJ/kg}$

Step 8 of 12

Work obtained from the turbine (ideally)

$W_{Ts} = 20 \times (3302.76 - 2336.41)$

$= 19327 \text{ kW}$

$= 19.32 \text{ MW}$

But it is given that the turbine is only 94% efficient, hence the actual work obtained from the turbine is,

$W_{Ta} = \eta_T \times W_{Ts}$

$W_{Ta} = 0.94 \times 19.32 \text{ MW}$

$= 18.17 \text{ MW}$

Step 9 of 12

Work done in pump is

$\dot{W}_{pump} = \dot{m} W_{pump}$

$= 20 \times 6.1$

$= 122 \text{ kJ/sec}$

$= 122 \text{ kW}$

Step 10 of 12

The net output of the Rankin cycle is given by,

$W_{net} = W_{Ta} - W_{pump}$

$= 18.17 - 0.122 \text{ MW}$

$= 18.05 \text{ M}$

$\boxed{W_{net} = 18.05 \text{ MW}}$

Step 11 of 12

The Rate at which heat added is,

$Q_{in} = \dot{m} (h_3 - h_2)$

$Q_{in} = 20 \times (3302.76 - 320.15)$

$= 59652.2 \text{ kJ/s}$

$= 59.65 \text{ MW}$

$\boxed{Q_{in} = 59.65 \text{ MW}}$

Step 12 of 12

Thermal efficiency of the Rankin cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}}$$

$$= \frac{18.05 \text{ MW}}{59.65 \text{ MW}}$$

$= 0.303$

$= 30.3 \%$

$\boxed{\eta_{Rankin} = 30.3\%}$

Question: 9.124P

 Using EES (or other) software, determine how much the thermal efficiency of the cycle in Prob. 9–123 would change if there were a 50 kPa pressure drop across the boiler.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question:

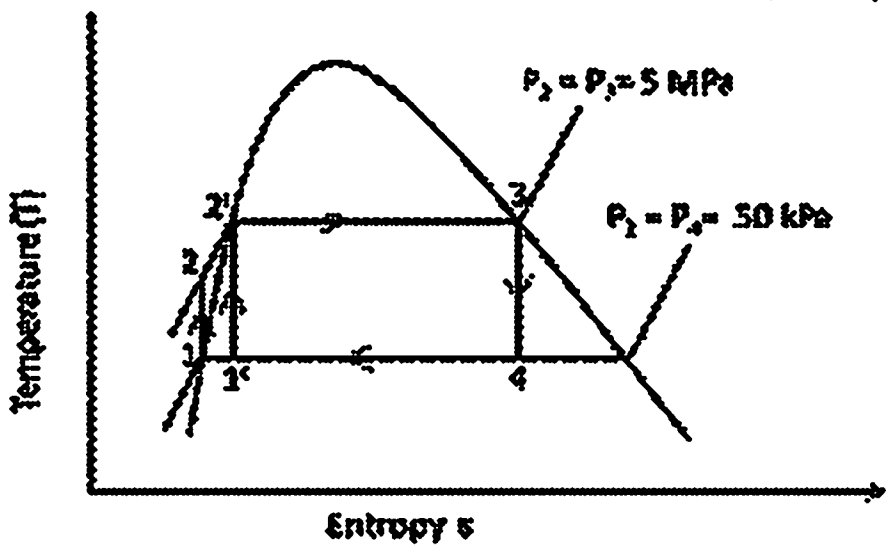
The net work output and the thermal efficiency for the Carnot and the simple ideal Rankine cycles with steam as the working fluid are to be calculated and compared. Steam enters the turbine in both cases at 5 MPa as a saturated vapor, and the condenser pressure is 50 kPa. In the Rankine cycle, the condenser exit state is saturated liquid and in the Carnot cycle, the boiler inlet state is saturated liquid. Draw the *T-s* diagrams for both cycles.

Step 1 of 15

Given,
Both the Rankin and Carnot cycles are operated between same pressure limits.
Boiler pressure = 5 MPa,
Condenser pressure = 50 kPa.
In both the cases stem is entering the turbine as saturated vapor.
In Rankin cycle the condenser exit is saturated liquid, where as in Carnot cycle boiler nlet is saturated liquid.

Step 2 of 15

The Ideal Rankin cycle and Carnot cycle operating between the given conditions is shown below.



Step 3 of 15

a) **Carnot Cycle:**

State 2^1:

$P_{2'} = 5\ \text{MPa} = 5000\ \text{kPa}$ and the liquid entering the boiler is saturated liquid, hence

$h_{2'} = 1154.5\ \text{kJ/kg}$

$s_2 = 2.9207\ \text{kJ/kg}\cdot\text{K}$

Step 4 of 15

State 3:

$P_3 = P_{2'} = 5\ \text{MPa} = 5000\ \text{kPa}$

the liquid entering the turbine is saturated vapor.

$h_3 = h_g = 2794.3\ \text{kJ/kg}$

$s_3 = s_g = 5.973\ \text{kJ/kg}\cdot\text{K}$

Step 5 of 15

State 4:

Since the expansion is isentropic $s_4 = s_3$ and $P_4 = 50\ \text{kPa}$

At these conditions the enthalpy is calculated using the steam tables and is given by,

From the saturated water pressure table

At $P_4 = P_1 = 50\ \text{kPa}$ and $s_4 = s_3 = 5.973\ \text{kJ/kg}\cdot\text{K}$

$$s_4 = s_f + x_4 s_{fg}$$

$$5.973 = 1.0912 + x_4(6.5019)$$

$$x_4 = 0.75$$

$$h_4 = h_f + x_4 h_{fg}$$

$$= 340.54 + 0.75(2304.7)$$

$$= 2070.9\ \text{kJ/kg}$$

Step 6 of 15

State 1^1:

Since the expansion is isentropic $s_{1'} = s_2 = 2.9207\ \text{kJ/kg}\cdot\text{K}$

From the saturated water pressure table

$P_{1'} = 50\ \text{kPa}$

$$s_1 = s_f + x_4 s_{fg}$$

$$2.9207 = 1.0912 + x_4(6.5019)$$

$$x_1 = 0.28$$

$$h_1 = h_f + x_4 h_{fg}$$

$$= 340.54 + 0.2814(2304.7)$$

$$= 989.082\ \text{kJ/kg}$$

Step 7 of 15

The net work output from the Carnot cycle is given by,

$$W_{net} = (h_3 - h_4) - (h_{2'} - h_1)$$

$$W_{net} = (2794.3 - 2070.9) - (1154.5 - 989.082)$$

$$W_{net} = 557.982\ \text{kJ/kg}$$

Step 8 of 15

The efficiency of the Carnot cycle is given by,

$$\eta_{Carnot} = \frac{W_{net}}{Q_{sup}}$$

$$= \frac{(h_3 - h_4) - (h_{2'} - h_1)}{(h_3 - h_{2'})}$$

$$= \frac{557.982}{}$$

$$\eta_{Carnot} = \frac{Q_{sup}}{}$$

$$= \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}$$

$$= \frac{557.982}{(2794.3 - 1154.5)}$$

$$= 0.340$$

$$\boxed{\eta_{Carnot} = 0.340}$$

Step 9 of 15

b) Rankin cycle:

State 1:

$P_1 =$ 50 kPa And the water is at saturated liquid condition. At these conditions the enthalpy is calculated using the steam tables and is given by,

$h_1 =$ 340.54 kJ/kg

$v_1 =$ 0.001030 m^3/kg

Step 10 of 15

State 2:

$P_2 =$ 5 MPa = 5000 kPa

The work done in the compression process is given by,

$w_{pump} = v_1 \times (P_2 - P_1)$

$w_{pump} = 0.001030 \times (5000\ \text{kPa} - 50\ \text{kPa})$

$w_{pump} = 5.0985$kJ/kg

Step 11 of 15

Enthalpy at state 2 is

$h_2 = h_1 + w_{pump}$

$= 340.54 + 5.0985$

$=345.638$kJ/kg

Step 12 of 15

State 3:

$P_3 = 5$ MPa = 5000 kPa And the liquid entering the turbine is saturated vapor.

$h_3 =$ 2794.2 kJ/kg

$s_3 =$ 5.9737 kJ/kg K

Step 13 of 15

State 4:

Since the expansion is isentropic $s_4 = s_3$ and $P_4 = 50$ kPa

At these conditions the enthalpy is calculated using the steam tables and is given by,

From the saturated water pressure table

At $P_4 = P_1 = 50$ kPa and $s_4 = s_3 =$ 5.973 kJ/kg·K

$s_4 = s_f + x_4 s_{fg}$

$5.973 = 1.0912 + x_4 (6.5019)$

$x_4 = 0.75$

$h_4 = h_f + x_4 h_{fg}$

$= 340.54 + 0.75(2304.7)$

$= 2070.9$kJ/kg

Step 14 of 15

The net work output from the Rankin cycle is given by,

$W_{net} = (h_3 - h_4) - (h_2 - h_1)$

$W_{net} = (2794.3 - 2070.9) - (345.638 - 340.54)$

$= 718.302$kJ/kg

$\boxed{W_{net} = 718.302\text{kJ/kg}}$

Step 15 of 15

The efficiency of the Rankin cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{sup}}$$

$$= \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)}$$

$$= \frac{718.302}{(2794.3 - 345.638)}$$

$$= 0.293$$

$$\boxed{\eta_{Rankin} = 0.293}$$

Question: 9.126P

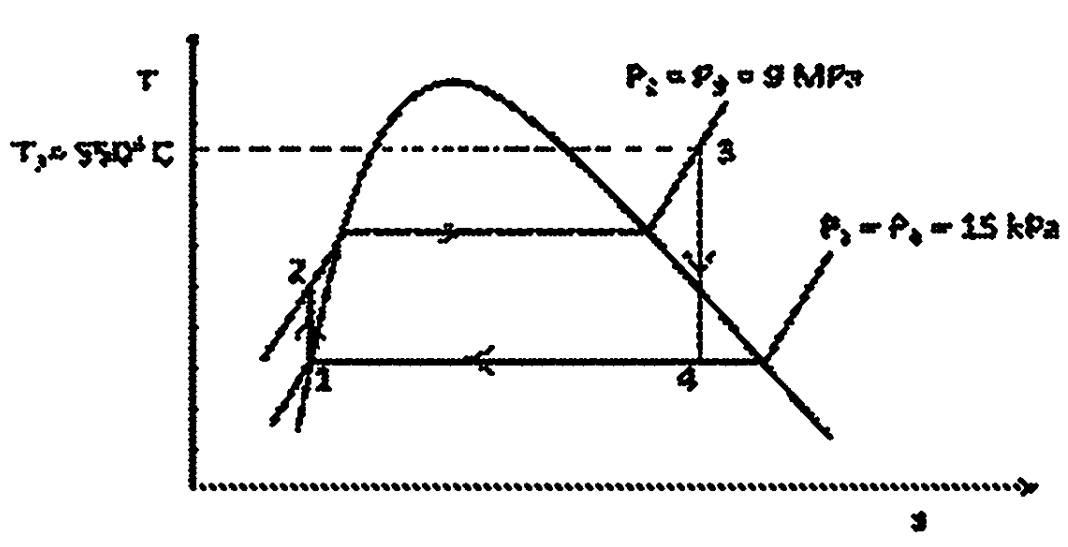
Consider a coal-fired steam power plant that produces 120 MW of electric power. The power plant operates on a simple ideal Rankine cycle with turbine inlet conditions of 9 MPa and 550°C and a condenser pressure of 15 kPa. The coal has a heating value (energy released when the fuel is burned) of 29,300 kJ/kg. Assuming that 75 percent of this energy is transferred to the steam in the boiler and that the electric generator has an efficiency of 96 percent, determine (a) the overall plant efficiency (the ratio of net electric power output to the energy input as fuel) and *(b)* the required rate of coal supply.

Step 1 of 11

Given,
Working fluid: water.
The boiler operating pressure $P_2=P_3=9\,\text{MPa}$.
the condenser operating pressure $P_1=P_4=15\,\text{kPa}$.
Turbine inlet temperature $T_3=550^\circ\text{C}$
The heating value of coal $C_v=29300\,\text{kJ/kg}$
It is mentioned that only 75 % of the energy is transferred to boiler.
Electric generator efficiency $\eta_g=0.96$.

Step 2 of 11

The Ideal Rankin cycle operating between the given conditions is shown below.



Step 3 of 11

State1:
From the Saturated water pressure table
$P_1 = 15\,\text{kPa}$
$h_1 = 225.94\,\text{kJ/kg}$
$v_1 = 0.001014\,\text{m}^3/\text{kg}$

Step 4 of 11

State 2:
The boiler operating pressure $P_2 = 9000\,\text{kPa}$
The work done in the compression process is given by,

$$w_{pump} = v_1 \times (P_2 - P_1)$$
$$w_{pump} = 0.001014 \times (9000 - 15)$$
$$= 9.11\,\text{kJ/kg}$$

And we know that

$$h_2 = h_1 + w_{pump}$$
$$= 225.93 + 9.11$$
$$= 235.04\,\text{kJ/kg}$$

Step 5 of 11

State 3:
From the superheated water
$P_3 = P_2 = 9000\,\text{kPa}$ And $T_3 = 550^0\,\text{C}$
$h_3 = 3512.0\,\text{kJ/kg}$
$s_3 = 6.81164\,\text{kJ/kg}\cdot\text{K}$

Step 6 of 11

State4:
Ideally the expansion in turbine is isentropic in nature,
Then $s_4 = s_3 = 6.81164\,\text{kJ/kg}\cdot\text{K}$
From the Saturated water pressure table
At $P_4 = P_1 = 15\,\text{kPa}$ and $s_4 = s_3 = 6.81164\,\text{kJ/kg}\cdot\text{K}$

$$h_4 = h_f + x_4 h_{fg}$$
$$= 225.94 + 0.8358(2372.3)$$
$$= 2208.708\,\text{kJ/kg}$$

Step 7 of 11

Work obtained from the turbine

$$W_T = (3512.0 - 2208.708)$$
$$= 1303.29\,\text{kJ/kg}$$

But it is given that the electric generator is only 96% efficient, The power obtained is 120 MW, hence,

$$120\text{MW} = m_{steam} \times W_T \times \eta_g$$
$$m_{steam} = \frac{120 \times 10^6}{0.96 \times 1303.2 \times 10^3}$$
$$= 95.917\,\text{kg/s}$$

Step 8 of 11

Heat supplied in the boiler is given by,

$$Q_{in} = \dot{m}\,(h_3 - h_2)$$
$$Q_{in} = 95.917 \times (3512.0 - 235.04)$$
$$= 314.316\,\text{MW}$$

Step 9 of 11

Since, only 75% of the energy is transferred to the boiler, total heat supplied by the steam is,

$$Q_{sup} = \frac{314.316}{0.75}$$
$$= 419.088\,\text{MW}$$

Step 10 of 11

The overall plant efficiency is given by,

$$\eta_{Overall} = \frac{W_T}{Q_{sup}}$$
$$= \frac{120}{419.088}$$
$$= 0.286$$

$$\boxed{\eta_{Overall} = 0.286}$$

Step 11 of 11

(b)

The required rate of coal is calculated by using following relation.

$$m_{coal} \times CV_{coal} = Q_{sup}$$
$$m_{coal} = \frac{419.088\text{MW}}{29300\,\text{kJ/kg}}\left(\frac{10^3\,\text{kJ/s}}{1\,\text{MW}}\right)$$
$$= 14.303\,\text{kg/s} \times \left(\frac{1\,\text{t}}{1000\,\text{kg}}\right) \times \left(\frac{3600\,\text{s}}{1\,\text{h}}\right)$$
$$= 51.49\,\text{t/h}$$

$$\boxed{m_{coal} = 51.49\,\text{t/h}}$$

Question: 9.127P pg 1

The schematic of a single-flash geothermal power plant with state numbers is given in Fig. P9-127. Geothermal resource exists as saturated liquid at 230°C. The geothermal liquid is withdrawn from the production well at a rate of 230 k.g/ s and is flashed to a pressure of 500 kPa by an essentially lisenthalpic flashing process where the resulting vapor is seprated from the liquid in a separator and directed to the turbine. The steam leaves the turbine at 10 kPa with a moisture content of 10 percent and enters the condenser where it is condensed and routed to a reinjection well along with the liquid coming off the separator. Determine (a) the mass flow rate of steam through the turbine, *(b)* the isentropic efficienc of the turbine, (c) the power output of the turbine, and *(d)* thermal efficiency of the plant (the ratio of the turbine work output to the energy of the geothermal fluid relative to standard ambient conditions).

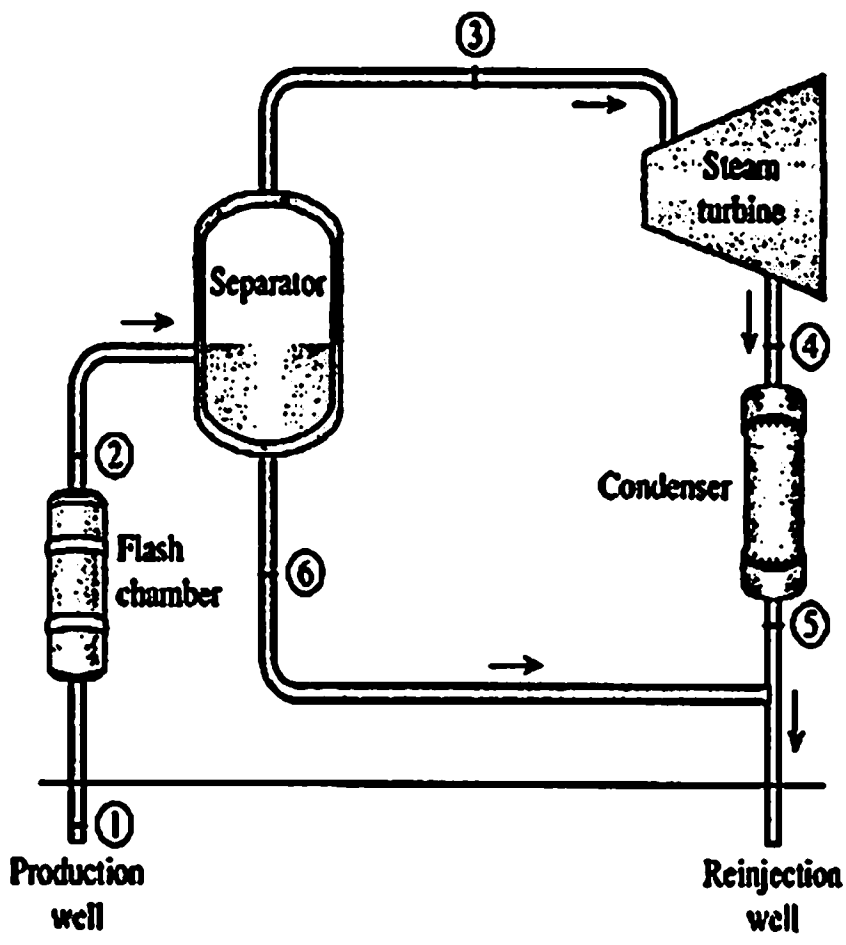
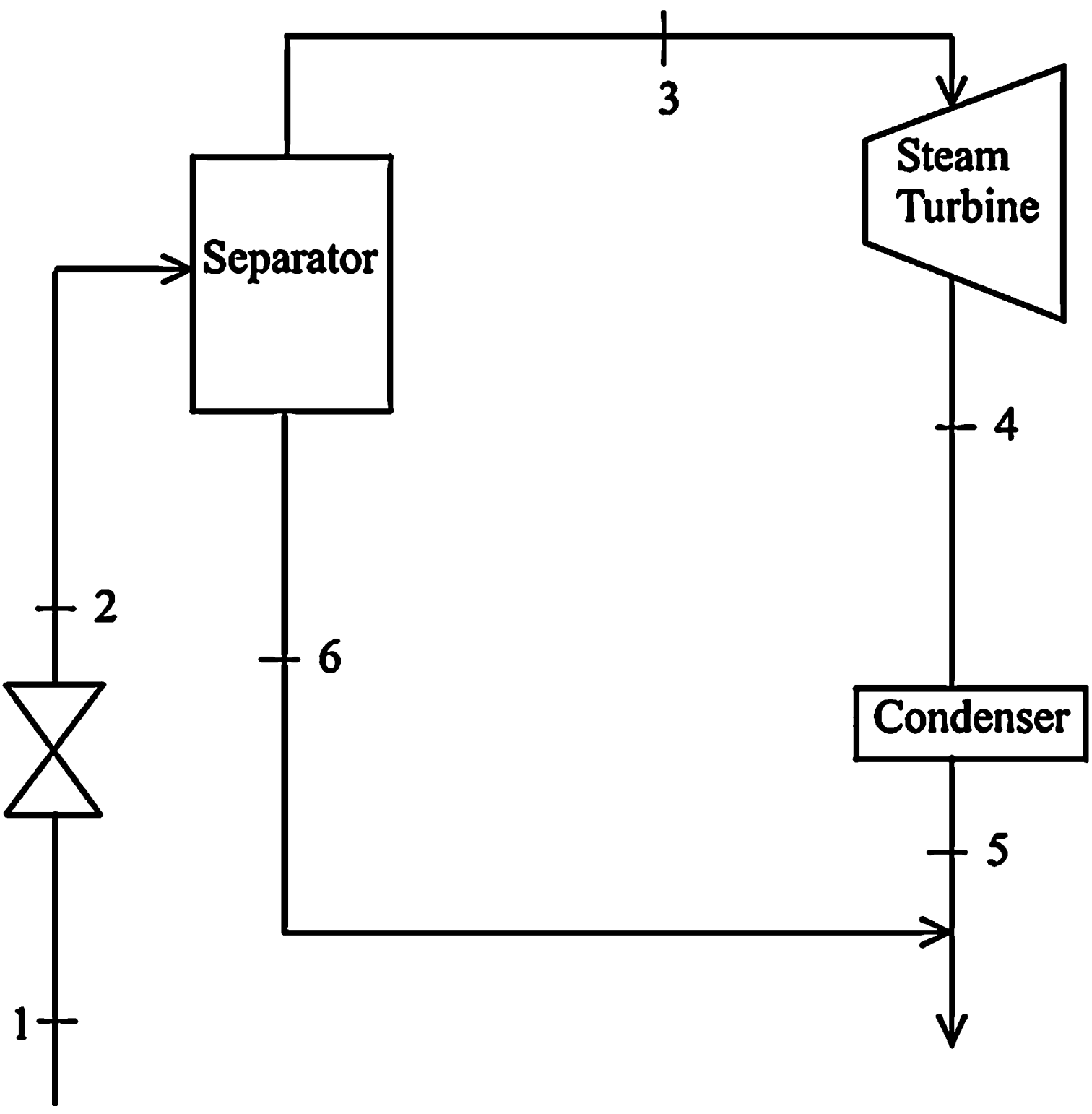


FIGURE P9–127

Step 1 of 6

Consider the schematic diagram of a single-flash geothermal power plant.



Step 2 of 6

Obtain the properties of steam from the table A.4, "Properties of saturated water-temperature table" at temperature, 230^0C .

Specific enthalpy, $h_1 = h_f = 990.14\ \text{kJ/kg}$

In the flash chamber, the enthalpy remains constant.

$$h_1 = h_2$$

Obtain the properties of steam from the table A.4, "Properties of saturated water-pressure table" at pressure, 500 kPa.

$$h_{f@500\text{kPa}} = 640.09\,\text{kJ/kg}$$

$$h_{fg@500\text{kPa}} = 2108\,\text{kJ/kg}$$

Determine the dryness fraction at state 2.

$$h_2 = h_{f@500\text{kPa}} + x_2 h_{fg@500\text{kPa}}$$

$$990.14 = 640.09 + x_2 \times 2108.0$$

$$x_2 = 0.166$$

Step 3 of 6

Determine the mass flow rate of steam through the turbine.

$$x_2 = \frac{\dot{m}_3}{\dot{m}_2}$$

$$x_2 = \frac{\dot{m}_3}{\dot{m}_1}$$

$$\dot{m}_3 = x_2 \times \dot{m}_1$$

Substitute 0.166 for x_2 , and $230\,\text{kg/s}$ for $\dot{m}_1$.

$\dot{m}_1$

$\dot{m}_3 = x_2 \times \dot{m}_1$

Substitute 0.166 for x_2 , and 230kg/s for $\dot{m}_1$.

$$\dot{m}_3 = 0.166 \times 230$$
$$= 38.18\text{kg/s}$$

Hence, the mass flow rate of steam through the turbine is, $\boxed{\dot{m}_3 = 38.18\text{kg/s}}$.

Step 4 of 6

(b)

Obtain the properties of saturated vapor from the table A.4, "Properties of saturated water-pressure table" at pressure, 500 kPa.

$h_3 = 2748.1\text{ kJ/kg}$

$s_3 = 6.8207\text{ kJ/kg.K}$

Obtain the properties of saturated vapor from the table A.4, "Properties of saturated water-pressure table" at pressure, 10 kPa.

$h_{f@10\text{kPa}} = 191.81\text{kJ/kg}$

$h_{fg@10\text{kPa}} = 2392.1\text{kJ/kg}$

Calculate the enthalpy at pressure , 10 kPa and dryness fraction, 0.9.

$h_4 = h_{f@10\text{kPa}} + x_4 h_{fg@10\text{kPa}}$

$h_4 = 191.81 + (0.9 \times 2392.1)$

$h_4 = 2344.7\text{ kJ/kg}$

Step 5 of 6

Obtain the properties of saturated vapor from the table A.4, "Properties of saturated water-pressure table" at pressure, 10 kPa.

$s_{f@10\text{kPa}} = 0.6492\text{kJ/kgK}$

$s_{fg@10\text{kPa}} = 7.4996\text{kJ/kgK}$

As the process 3-4s is reversible isentropic process,

$s_3 = s_{4s}$

$s_{4s} = s_{f@10\text{kPa}} + x_{4s} s_{fg@10\text{kPa}}$

$6.8207 = 0.6492 + x_{4s} \times 7.4996$

$x_{4s} = 0.8229$

Determine the enthalpy at state 4s.

$h_{4s} = h_{f@10\text{kPa}} + x_{4s} h_{fg@10\text{kPa}}$

$h_{4s} = 191.81 + 0.8229 \times 2392.1$

$h_{4s} = 2160.269\text{kJ/kg}$

Determine the isentropic efficiency of the turbine.

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\eta_T = \frac{2748.1 - 2344.7}{2748.1 - 2160.269}$$

$\eta_T = 0.686$

$\boxed{\eta_T = 0.686}$

(c)

Determine the power output from the turbine.

$W_T = \dot{m}_3 (h_3 - h_4)$

$W_T = 38.18 \times (2748.1 - 2344.7)$

$W_T = 15401.812\text{kW}$

Hence, the power output of the turbine is, $\boxed{W_T = 15.401\text{ MW}}$.

Step 6 of 6

(d)

Obtain the properties of saturated water from the table A.4, "Properties of saturated water-temperature table" at temperature, 25°C .

$h_0 = h_f = 104.83\text{ kJ/kg}$

Determine the total energy of the geothermal fluid.

$E_{in} = m_1 (h_1 - h_0)$

$E_{in} = 230 \times (990.14 - 104.83)$

$E_{in} = 203621.3\text{kW}$

Determine the thermal efficiency of the plant.

$$\eta_{th} = \frac{W_T}{E_{in}}$$

$$\eta_{th} = \frac{15401.812}{203621.3}$$

$= 0.07563$

Hence, the thermal efficiency of the plant is

$$\eta_{th} = \frac{15401.812}{203621.3}$$

$$= 0.07563$$

Hence, the thermal efficiency of the plant is, $\boxed{\eta_{th} = 7.563\%}$.

Question: 9.128P

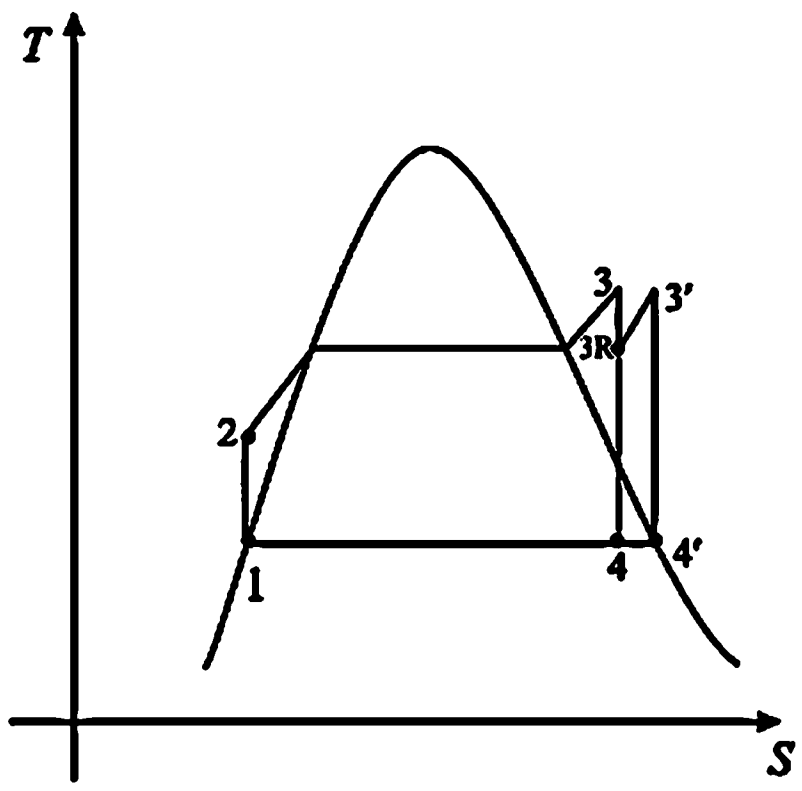
How do the following quantities change when a simple ideal Rankine cycle is modified with reheating? Assume the mass flow rate is maintained the same.

Pump work input:	(*a*) increases, (*b*) decreases, (c)remains the same
Turbine work ouput:	(*a*) increases (b) decreases, (c) remains the same
Heat supplied:	(*a*) increases (b) decreases, (c) remains the same
Heat rejected:	(*a*) increases, (b) decreases, (c) remains the same
Moisture content at turbine exit:	(*a*) increases, (b) decreases, (c) remains the same

Step 1 of 7

Given: Mass flow rate is constant

Step 2 of 7



Step 3 of 7

Pump work input: (c) remains the same

In reheating process $1-2$ will not change and hence pump work remains same

Step 4 of 7

Turbine work output: (a) increases

In reheating work output in turbine will take place in two stages first stage from $3-3R$ and second stage from $3'-4'$ and hence turbine work output increases

Step 5 of 7

Heat supplied: (a) increases

As it is reheat cycle additional heat is supplied and hence heat supplied increases

Step 6 of 7

Heat rejected: (a) increases

In reheat cycle turbine outlet point moves towards superheated region and hence heat rejected increases

Step 7 of 7

Moisture content at turbine exit: (b) decreases

In reheat cycle turbine outlet point moves towards superheated region and hence moisture content at turbine decreases

Question: 9.129P

Is there an optimal pressure for reheating the steam of a Rankine cycle? Explain.

Step 1 of 2

An optimal pressure for reheating generally allows for decrease of the moisture content of the steam at the low-pressure turbine exit. In general optimum reheat pressure is about the one-fourth of the maximum cycle pressure.

Step 2 of 2

The average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach an isothermal process at the maximum temperature. The use of more than two reheat stages, however, is not practical. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat. If the turbine inlet pressure is not high enough, double reheat would result in superheated exhaust. This is undesirable as it would cause the average temperature for heat rejection to increase and thus the cycle efficiency to decrease. Therefore, double reheat is used only on supercritical-pressure ($P > 22.06$ MPa) power plants. A third reheat stage would increase the cycle efficiency by about half of the improvement attained by the second reheat. This gain is too small to justify the added cost and complexity.

Question: 9.130P

Consider a simple ideal Rankine cycle and an ideal Rankine cycle with three reheat stages. Both cycles operate between the same pressure limits. The maximum temperature is 700°C in the simple cycle and 450°C in the reheat cycle. Which cycle do you think will have a higher thermal efficiency?

Step 1 of 2

The thermal efficiency of the simple ideal Rankine cycle will probably be higher when compared to ideal Rankine cycle in three stages since the average temperature at which heat (700°C) is added will be higher in simple cycle.

Step 2 of 2

Reason:
In ideal rankine cycle on addition of third reheat stage it would increase the cycle efficiency by about half of the improvement attained by the second reheat but gain is too small to justify the added cost and complexity. And also the average temperature at which heat added is low compared to simple ideal Rankine cycle.

Question: 9.131P

An ideal reheat Rankine cycle with water as the working fluid operates the inlet of the high-pressure turbine at 8000 kPa and 450T; the inlet of the low-pressure turbine at 500 kPa and 500°C; and the condenser at 10 kPa. Determine the mass flow rate through the boiler needed for this system to produce a net 5000 kW of power and the thermal efficiency of the cycle.

We don't have the solution to this problem yet.

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Question:

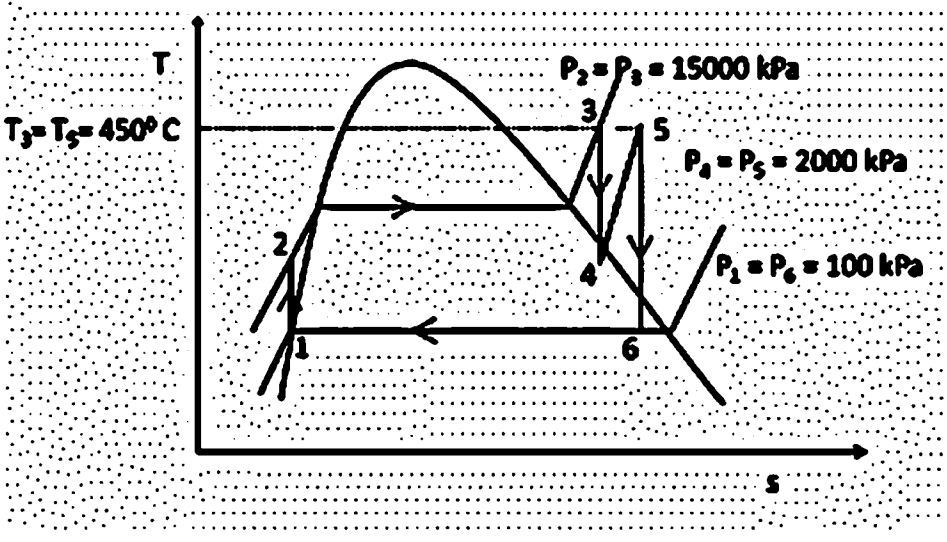
An ideal reheat Rankine cycle with water as the working fluid operates the boiler at 15,000 kPa, the reheater at 2000 kPa, and the condenser at 100 kPa. The temperature is 450CC at the entrance of the high-pressure and low-pressure turbines. The mass flow rate through the cycle is 1.74 kg/s. Determine the power used by pumps, the power produced by the cycle, the rate of heat transfer in the reheater, and the thermal efficiency of this system.

Step 1 of 13

Given,
Working fluid: water.
Pressure at the inlet of the high-pressure turbine $P_2 = P_3 = 15000$ kPa.
Temperature at the inlet of the high-pressure turbine $T = 450^0$C.
Pressure at the inlet of the low-pressure turbine $P_4 = P_5 = 2000$ kPa.
Temperature at the inlet of the low-pressure turbine $T = 450^0$C.
Condenser Pressure $P_1 = P_6 = 100$ kPa.
Mass flow rate $m = 1.74$ kg / s.

Step 2 of 13

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



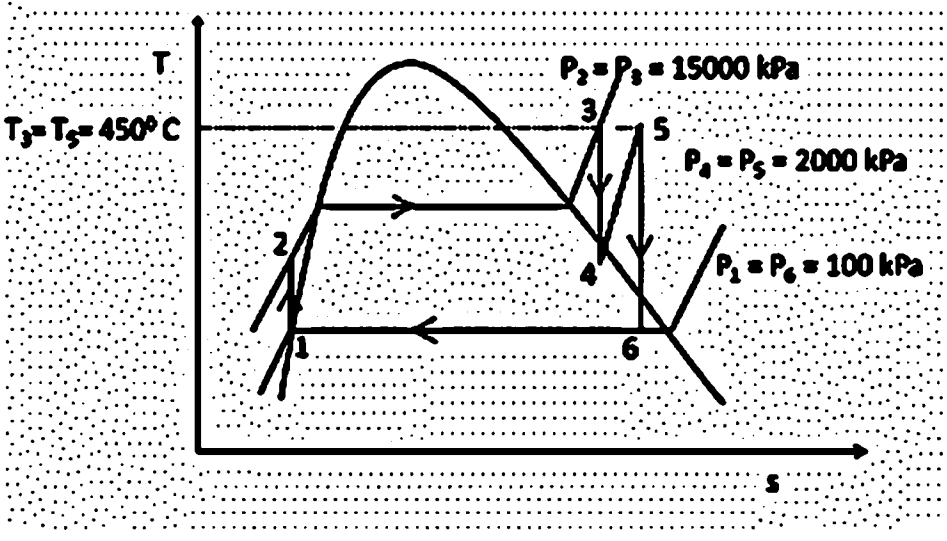
Step 3 of 13

State 1:
From the saturated water pressure table
$P_1 = 100$ kPa And saturated liquid, at these conditions
$h_1 = 417.51$ kJ/kg
$v_1 = 0.001043$ m^3/kg

Step 4 of 13

Step 2 of 13

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



Step 3 of 13

State 1:
From the saturated water pressure table
$P_1 = 100$ kPa And saturated liquid, at these conditions
$h_1 = 417.51$ kJ/kg
$v_1 = 0.001043$ m^3/kg

Step 7 of 13

State 3:
From the superheated water
$P_3 = P_2 = 15000$ kPa And $T_3 = 450^0$C
$h_3 = 3157.9$ kJ/kg
$s_3 = 6.1434$kJ/kg · K

Step 8 of 13

State 4:
From the saturated water pressure table
$P_4 = 2000$ kPa and $s_4 = s_3 = 6.143$ kJ/kg · K

$$s_4 = s_f + x_4 s_{fg}$$
$$6.143 = 2.4467 + x_4 (3.8923)$$
$$x_4 = 0.9497$$
$$h_4 = h_f + x_4 h_{fg}$$
$$= 908.47 + 0.9497(1889.8)$$
$$= 2703.321\text{kJ/kg}$$

Step 9 of 13

State 5:
From the superheated water
$P_5 = 2000$ kPa and $T_5 = 450^0$C
By using interpolation,

$$h_5 = 3248.4 + \frac{(450-400)(3468.3-3248.3)}{(500-400)}$$
$$= 3358.4\text{kJ/kg}$$

By using interpolation,

$$h_5 = 3248.4 + \frac{(450-400)(3468.3-3248.3)}{(500-400)}$$

$$= 3358.4\,\text{kJ/kg}$$

$$s_5 = 7.1292 + \frac{(450-400)(7.4337-7.1292)}{(500-400)}$$

$$= 7.281\,\text{kJ/kg}\cdot\text{K}$$

Step 10 of 13

State 6:

From the saturated water pressure table

$P_6 = 100\,\text{kPa}$ and $s_6 = s_5 = 7.281\,\text{kJ/kg}\cdot\text{K}$

$$s_6 = s_f + x_6 s_{fg}$$

$$7.281 = 1.3028 + x_6(6.0562)$$

$$x_6 = 0.9871$$

$$h_6 = h_f + x_6(h_{fg})$$

$$= 417.51 + 0.9871(2257.5)$$

$$= 2645.88\,\text{kJ/kg}$$

Step 11 of 13

Net power output is given by,

$$W_{net} = \dot{m} \times \left((h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)\right)$$

$$W_{net} = 1.74\left((3157.9 - 2703.32) + (3358.4 - 2645.88) - (432.97 - 417.51)\right)$$

$$= 2003.85\,\text{kW}$$

Step 12 of 13

Rate of heat transfer is given by,

$$Q_{in} = \dot{m} \times \left((h_3 - h_2) + (h_5 - h_4)\right)$$

$$Q_{in} = 1.74 \times \left((3157.9 - 432.97) + (3358.4 - 2703.321)\right)$$

$$= 5881.215\,\text{kJ/s}$$

$$= 5881.215\,\text{kW}$$

Step 13 of 13

Efficiency of the cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}}$$

$$= \frac{2000\,\text{kW}}{5881.215\,\text{kW}}$$

Step 11 of 13

Net power output is given by,

$$W_{net} = \dot{m} \times \left((h_3 - h_4) + (h_5 - h_6) - (h_2 - h_1)\right)$$

$$W_{net} = 1.74\left((3157.9 - 2703.32) + (3358.4 - 2645.88) - (432.97 - 417.51)\right)$$

$$= 2003.85\,\text{kW}$$

Question: 9.133P

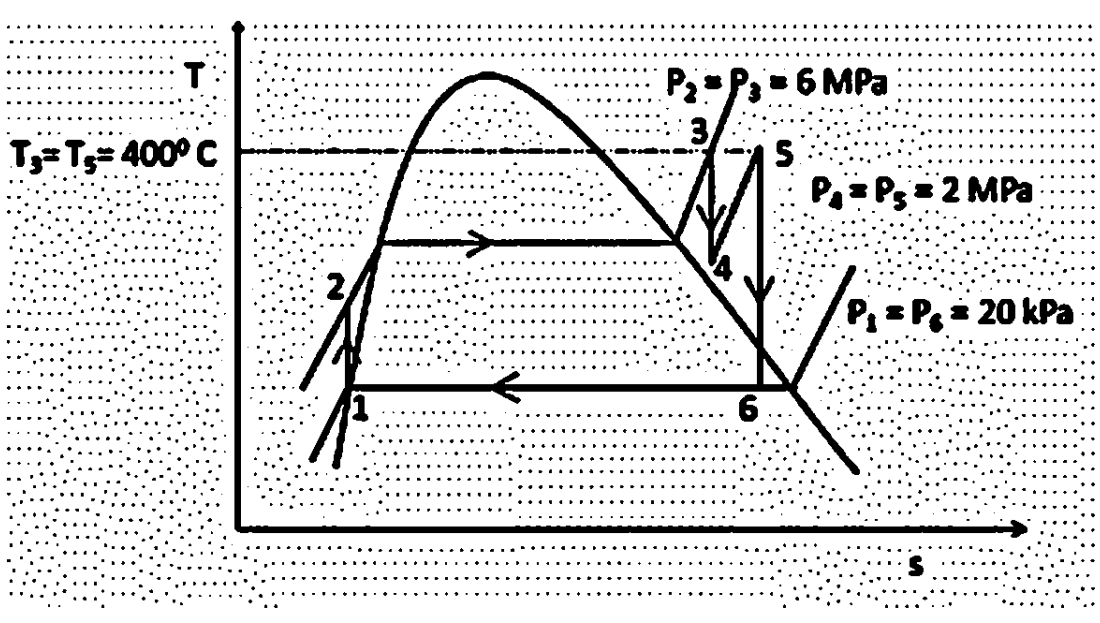
A steam power plant operates on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 6 MPa and 400°C and leaves at 2 MPa. Steam is then reheated at constant pressure to 400°C before it expands to 20 kPa in the low-pressure turbine. Determine the turbine work output, in kJ/kg. and the thermal efficiency of the cycle. Also, show the cycle on a *T-s* diagram with respect to saturation lines.

Step 1 of 13

Given,
Working fluid: water.
Pressure at the inlet of the high-pressure turbine $P_2 = P_3 = 6$ MPa.
Temperature at the inlet of the high-pressure turbine $T_3 = 400^0$C
Pressure at the inlet of the low-pressure turbine $P_4 = P_5 = 2$ MPa.
Temperature at the inlet of the low-pressure turbine $T_5 = 400^0$C.
Condenser Pressure $P_1 = P_6 = 20$ kPa.

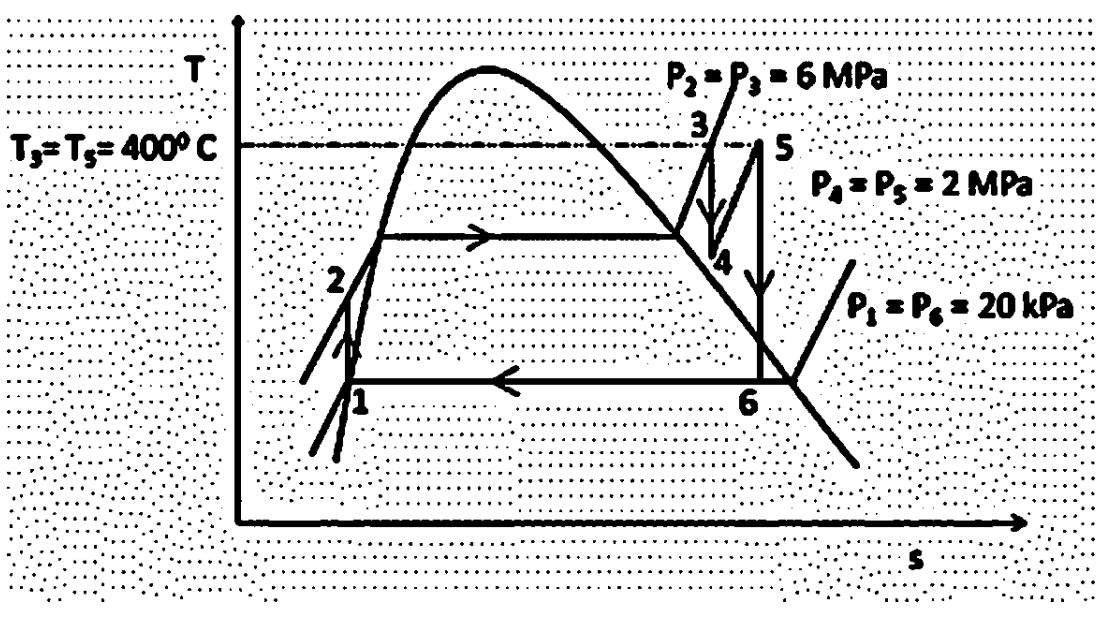
Step 2 of 13

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



Step 2 of 13

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



Step 3 of 13

State 1:
From the saturated water pressure table
$P_1 = 20$ kPa And saturated liquid,
$h_1 = 251.42$ kJ/kg
$v_1 = 0.001017$ m^3/kg

Step 3 of 13

State 1:
From the saturated water pressure table
$P_1 = 20$ kPa And saturated liquid,
$h_1 = 251.42$ kJ/kg
$v_1 = 0.001017$ m^3/kg

Step 7 of 13

State 4:
From the superheated water pressure table
$P_4 = 2$ MPa and $s_4 = s_3 = 6.5432$ kJ/kg·K

$$h_4 = 2836.1 + \frac{(6.5432 - 6.4160)(2903.3 - 2836.1)}{(6.5475 - 6.4160)}$$

$h_4 = 2901.10$ kJ/kg

Step 8 of 13

State 5:
From the superheated water pressure table
$P_5 = 2$ MPa and $T_5 = 400^0$C
$h_5 = 3248.4$ kJ/kg
$s_5 = 7.1292$ kJ/kg·K

Step 9 of 13

State 6:
From the saturated water pressure table
$P_6 = 20$ kPa and $s_6 = s_5 = 7.1292$ kJ/kg·K

$$s_6 = s_f + x_6(s_{fg})$$

$$7.1292 = 0.8320 + x_6(7.0752)$$

$$x_6 = 0.8900$$

$$h_6 = h_f + x_6(h_{fg})$$

$$= 251.42 + 0.8900(2357.5)$$

$$= 2349.685 \text{ kJ/kg}$$

Step 7 of 13

State 4:
From the superheated water pressure table
$P_4 = 2$ MPa and $s_4 = s_3 = 6.5432$ kJ/kg·K

$$h_4 = 2836.1 + \frac{(6.5432 - 6.4160)(2903.3 - 2836.1)}{(6.5475 - 6.4160)}$$

$h_4 = 2901.10$ kJ/kg

Step 8 of 13

State 5:
From the superheated water pressure table
$P_5 = 2$ MPa and $T_5 = 400^0$C
$h_5 = 3248.4$ kJ/kg
$s_5 = 7.1292$ kJ/kg·K

Step 10 of 13

Turbine work output is given by,

$$W_T = (h_3 - h_4) + (h_5 - h_6)$$

$$W_T = (3178.3 - 2901.10) + (3248.7 - 2349.685)$$

$$W_T = 1176.215 \text{ kJ/kg}$$

Step 11 of 13

Net work output is given by,

$$W_{net} = W_T - W_{pump}$$

$$W_{net} = 1176.215 - 6.08$$

$$= 1170.135 \text{ kJ/kg}$$

Question: 9.134DIAGR-CH



Reconsider Prob. 9–133. Using EES (or other) software, solve this problem by the diagram window data entry feature of EES. Include the effects of the turbine and pump efficiencies and also show the effects of reheat on the steam quality at the low-pressure turbine exit. Plot the cycle on a *T-s* diagram with respect to the saturation lines. Discuss the results of your parametric studies.

We don't have the solution to this problem yet.

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Question:

Consider a steam power plant that operates on the ideal reheat Rankine cycle. The plant maintains the inlet of the high-pressure turbine at 600 psia and 600°F, the inlet of the low-pressure turbine at 200 psia and 600°F, and the condenser at 10 psia. The net power produced by this plant is 5000 kW. Determine the rate of heat addition and rejection and the thermal efficiency of the cycle.

Step 1 of 18

Given:

Inlet pressure of high pressure turbine $P_3 = 600\,\text{psia}$

Inlet temperature of high pressure turbine $T_1 = 600^0\,\text{F}$

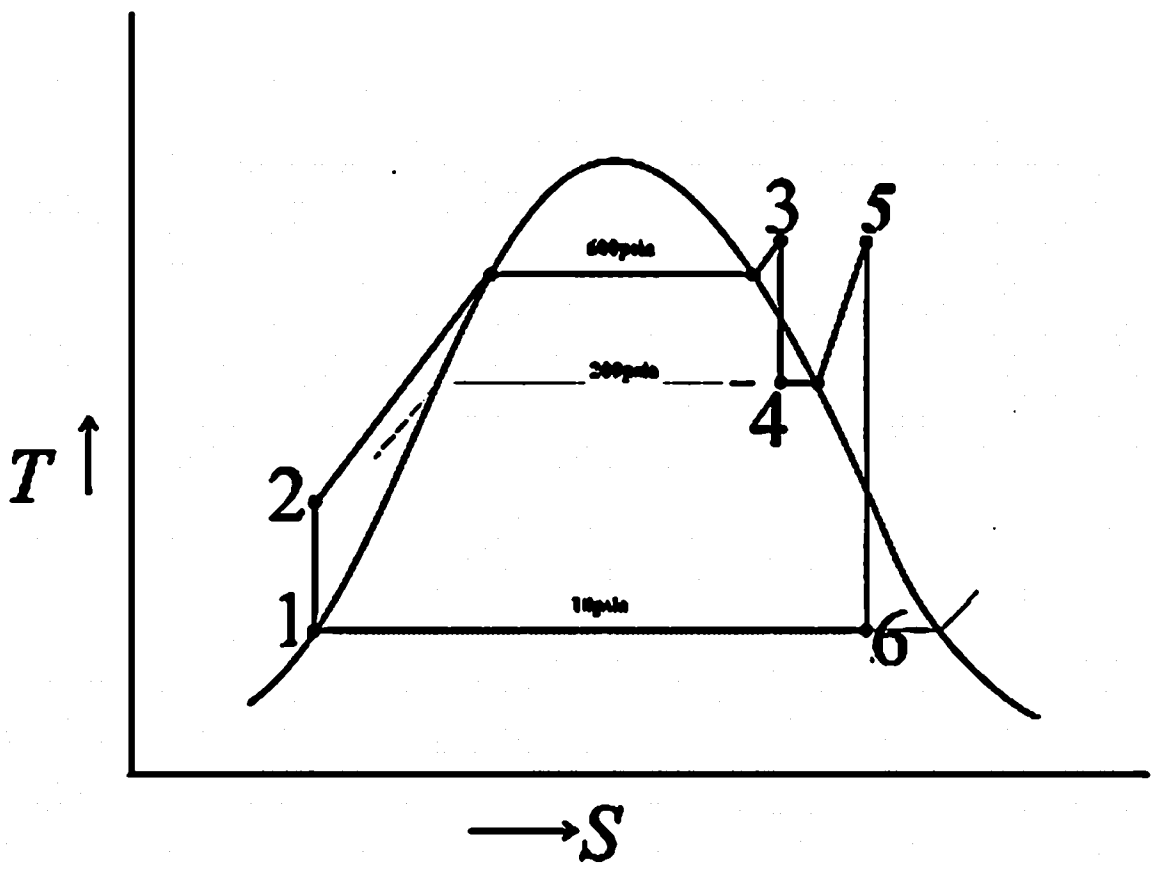
Inlet pressure of low pressure turbine $P_5 = 200\,\text{psia}$

Inlet temperature of low pressure turbine $T_5 = 600^0\,\text{F}$

Condenser pressure $P_6 = P_1 = 10\,\text{psia}$

Net power produced $W_{net} = 5000\,\text{kW}$

Step 2 of 18



Step 3 of 18

At $P_1 = 10\,\text{psia}$ from saturated water tables

$h_1 = h_f = 161.25\,\text{Btu/lbm}$

$v_1 = v_{f,10psia} = 0.01659\,\text{ft}^3/\text{lbm}$

Step 4 of 18

Pump work

$$w_{pump} = -v_1(P_2 - P_1) \qquad (\because P_2 = P_3)$$

$$w_{pump} = -0.01659\,\frac{\text{ft}^3}{\text{lbm}} \times (600-10)\frac{\text{lbf}}{\text{in}^2} \times 144\frac{\text{in}^2}{\text{ft}^2} \times \frac{1}{778}\frac{\text{Btu}}{\text{ft-lbf}}$$

$$w_{pump} = -1.81\,\text{Btu/lbm}$$

Step 5 of 18

Applying first law to the process 1-2

$h_1 = h_2 + w_{pump}$

$h_2 = h_1 - w_{pump}$

$h_2 = 161.25 + 1.81$

$h_2 = 163.1\,\text{Btu/lbm}$

Step 6 of 18

At $P_3 = 600\,\text{psia}$ and $T_3 = 600^0\,\text{F}$ from superheated water tables

$h_3 = 1289.9\,\text{Btu/lbm}$

$s_3 = 1.5325\,\text{Btu/lbm.R}$

Step 7 of 18

Process 3-4 is isentropic process

$s_4 = s_3$

At $P_4 = 200\,\text{psia}$ from saturated water tables

$s_4 = s_{f@200psia} + x_4 s_{fg@200psia}$

$1.5325 = 0.54379 + x_4 \times 1.00219$

$x_4 = 0.986$

Step 8 of 18

Enthalpy at state-4,

$h_4 = h_{f@200psia} + x_4 h_{fg@200psia}$

$h_4 = 355.46 + 0.986 \times 843.33$

$h_4 = 1186.98\,\text{Btu/lbm}$

Step 9 of 18

At state-5:

From superheated water tables at $P_5 = 200\,\text{psia}$ and $T_5 = 600^0\,\text{F}$

$h_5 = 1322.3\,\text{Btu/lbm}$

$s_5 = 1.6771\,\text{Btu/lbm.R}$

Step 10 of 18

Process 5-6 is isentropic expansion

$s_5 = s_6$

$s_5 = s_{f@10psia} + x_6 s_{fg@10psia}$

$1.6771 = 0.28362 = x_6 \times 1.50391$

$x_6 = 0.927$

Step 11 of 18

$1.6771 = 0.28362 = x_6 \times 1.50391$

$x_6 = 0.927$

Step 11 of 18

Enthalpy at state-6

$h_6 = h_{f@10\,psia} + x_6 h_{fg@psia}$

$h_6 = 161.25 + 0.927 \times 981.82$

$h_6 = 1071.39$ kJ/kg

Step 12 of 18

The heat input is

$q_{in} = (h_3 - h_2) + (h_5 - h_4)$

$q_{in} = (1289.9 - 163.1) + (1322.3 - 1186.98)$

$q_{in} = 1262.12$ Btu/lbm

Step 13 of 18

The heat rejected in the condenser is

$q_{out} = (h_6 - h_1)$

$q_{out} = (1071.39 - 161.25)$

$q_{out} = 910.14$ Btu/lbm

Step 14 of 18

Net power output

$w_{net} = q_{in} - q_{out}$

$w_{net} = 1262.12 - 910.14$

$w_{net} = 351.98$ Btu/lbm

Step 15 of 18

Mass flow rate

$$m_{steam} = \frac{W_{net}}{w_{net}}$$

$$m_{steam} = \frac{5000 \text{ kJ/s}}{351.98 \text{ Btu/lbm}} \times \frac{0.9478 \text{ Btu}}{1 \text{ kJ}}$$

$m_{steam} = 13.46$ lbm/s

Step 16 of 18

Rate of heat addition is

$Q_{in} = m_{steam} \times q_{in}$

$Q_{in} = 13.46 \times 1262.12$

$\boxed{Q_{in} = 16988.135 \text{Btu/s}}$

Step 17 of 18

Rate of heat rejection is

$Q_{out} = m_{steam} \times q_{out}$

$Q_{out} = 13.46 \times 910.14$

$\boxed{Q_{out} = 12250.48 \text{ Btu/s}}$

Step 18 of 18

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{351.98}{1262.12}$$

$\boxed{\eta_{th} = 27.88\%}$

Question: 9.136P pg 1

In Prob, 9-135E, is there any advantage to operating the reheat section of the boiler at 100 psia rather than 200 psia while maintaining the same low-pressure turbine inlet temperature?

Step 1 of 26

Given:

Inlet pressure of high pressure turbine $P_3 = 600\,\text{psia}$

Inlet temperature of high pressure turbine $T_1 = 600^0\,\text{F}$

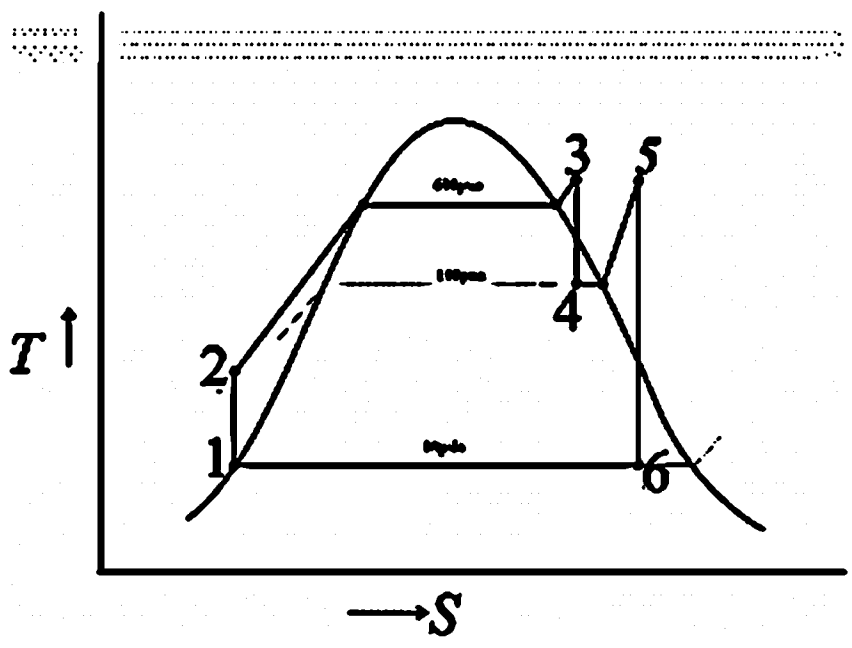
Inlet pressure of low pressure turbine $P_5 = 100\,\text{psia}$

Inlet temperature of low pressure turbine $T_5 = 600^0\,\text{F}$

Condenser pressure $P_6 = P_1 = 10\,\text{psia}$

Net power produced $W_{net} = 5000\,\text{kW}$

Step 2 of 26



Step 3 of 26

At $P_1 = 10\,\text{psia}$ from saturated water tables

$h_1 = h_f = 161.25\,\text{Btu/lbm}$

$v_1 = v_{f,10psia} = 0.01659\,\text{ft}^3/\text{lbm}$

Step 4 of 26

Pump work

$$w_{pump} = -v_1(P_2 - P_1) \qquad (\because P_2 = P_3)$$

$$w_{pump} = -0.01659\ \frac{\text{ft}^3}{\text{lbm}} \times (600-10)\frac{\text{lbf}}{\text{in}^2} \times 144\frac{\text{in}^2}{\text{ft}^2} \times \frac{1}{778}\frac{\text{Btu}}{\text{ft-lbf}}$$

$$w_{pump} = -1.81\,\text{Btu/lbm}$$

Step 5 of 26

Applying first law to the process 1-2

$h_1 = h_2 + w_{pump}$

$h_2 = h_1 - w_{pump}$

$h_2 = 161.25 + 1.81$

$h_2 = 163.1\,\text{Btu/lbm}$

Step 6 of 26

At $P_3 = 600\,\text{psia}$ and $T_3 = 600^0\,\text{F}$ from superheated water tables

$h_3 = 1289.9\,\text{Btu/lbm}$

$s_3 = 1.5325\,\text{Btu/lbm.R}$

Step 7 of 26

Process 3-4 is isentropic process

$s_4 = s_3$

From steam tables at $P_4 = 100\,\text{psia}$ and $s_4 = s_3 = 1.5325\,\text{Btu/lbm.R}$

$s_3 = s_{f@100psia} + x_4 s_{fg@100psia}$

$1.5325 = 0.47427 + x_4 \times 1.12888$

$x_4 = 0.937$

Step 8 of 26

Enthalpy at state-4

$h_4 = h_{f@100psia} + x_4 h_{fg@100psia}$

$h_4 = 298.51 + 0.937 \times 888.9$

$h_4 = 1131.409\,\text{Btu/lbm}$

Step 9 of 26

At state-5:

From steam tables at $P_5 = 100\,\text{psia}$ and $T_5 = 600^0\,\text{F}$

Enthalpy at state-5, $h_5 = 1329.4\,\text{Btu/lbm}$

Entropy at state-5, $s_5 = 1.7586\,\text{Btu/lbm.R}$

Step 10 of 26

Process 5-6 is isentropic expansion

$s_5 = s_6$

$s_5 = s_{f@10psia} + x_6 s_{fg@10psia}$

$1.7586 = 0.28362 + x_6 \times 1.50391$

$x_6 = 0.9807$

Step 11 of 26

Enthalpy at state-6,

$h_6 = h_{f@10psia} + x_6 h_{fg@10psia}$

$h_6 = 161.25 + 0.9807 \times 981.82$

$h_6 = 1124.12\,\text{kJ/kg}$

Enthalpy at state-6,

$h_6 = h_{f@10psia} + x_6 h_{fg@10psia}$

$h_6 = 161.25 + 0.9807 \times 981.82$

$h_6 = 1124.12$ kJ/kg

Step 12 of 26

The heat input is

$q_{in} = (h_3 - h_2) + (h_5 - h_4)$

$q_{in} = (1289.9 - 163.1) + (1329.4 - 1131.409)$

$q_{in} = 1324.791$ Btu/lbm

Step 13 of 26

The heat rejected in the condenser is

$q_{out} = (h_6 - h_1)$

$q_{out} = (1124.12 - 161.25)$

$q_{out} = 962.87$ Btu/lbm

Step 14 of 26

Net power output

$w_{net} = q_{in} - q_{out}$

$w_{net} = 1324.791 - 962.87$

$w_{net} = 361.921$ Btu/lbm

Step 15 of 26

Mass flow rate

$$m_{steam} = \frac{W_{net}}{w_{net}}$$

$$m_{steam} = \frac{5000 \text{ kJ/s}}{361.921 \text{ Btu/lbm}} \times \frac{0.9478 \text{ Btu}}{1 \text{ kJ}}$$

$m_{steam} = 13.094$ lbm/s

Step 16 of 26

Rate of heat addition is

$Q_{in} = m_{steam} \times q_{in}$

$Q_{in} = 13.094 \times 1324.791$

$\boxed{Q_{in} = 17346.813 \text{ Btu/s}}$

Step 17 of 26

Rate of heat rejection is

$Q_{out} = m_{steam} \times q_{out}$

$Q_{out} = 13.094 \times 962.87$

$\boxed{Q_{out} = 12607.819 \text{ Btu/s}}$

Step 18 of 26

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{361.921}{1324.791}$$

$= 0.2731$

$\boxed{\eta_{th} = 27.31\%}$

Step 19 of 26

If the reheat pressure is 200psia

Process 3-4 is isentropic process

$s_4 = s_3$

At $P_4 = 200$ psia from saturated water tables

$s_4 = s_{f@200psia} + x_4 s_{fg@200psia}$

$1.5325 = 0.54379 + x_4 \times 1.00219$

$x_4 = 0.986$

Step 20 of 26

Enthalpy at state-4,

$h_4 = h_{f@200psia} + x_4 h_{fg@200psia}$

$h_4 = 355.46 + 0.986 \times 843.33$

$h_4 = 1186.98$ Btu/lbm

Step 21 of 26

At state-5:

From superheated water tables at $P_5 = 200$ psia and $T_5 = 600^0$F

$h_5 = 1322.3$ Btu/lbm

$s_5 = 1.6771$ Btu/lbm.R

Step 22 of 26

Process 5-6 is isentropic expansion

$s_5 = s_6$

$s_5 = s_{f@10psia} + x_6 s_{fg@10psia}$

$1.6771 = 0.28362 = x_6 \times 1.50391$

$x_6 = 0.927$

Enthalpy at state-6

$h_6 = h_{f@10psia} + x_6 h_{fg@psia}$

$h_6 = 161.25 + 0.927 \times 981.82$

$h_6 = 1071.39$ kJ/kg

$h_6 = h_{f@10\,psia} + x_6 h_{fg@psia}$

$h_6 = 161.25 + 0.927 \times 981.82$

$h_6 = 1071.39$ kJ/kg

Step 23 of 26

The heat input is

$q_{in} = (h_3 - h_2) + (h_5 - h_4)$

$q_{in} = (1289.9 - 163.1) + (1322.3 - 1186.98)$

$q_{in} = 1262.12$ Btu/lbm

Step 24 of 26

The heat rejected in the condenser is

$q_{out} = (h_6 - h_1)$

$q_{out} = (1071.39 - 161.25)$

$q_{out} = 910.14$ Btu/lbm

Step 25 of 26

Net power output

$w_{net} = q_{in} - q_{out}$

$w_{net} = 1262.12 - 910.14$

$w_{net} = 351.98$ Btu/lbm

Step 26 of 26

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{351.98}{1262.12}$$

$$\boxed{\eta_{th} = 27.88\%}$$

Therefore there is a slight decrease in thermal efficiency if the pressure is decreased from 200psia to 100psia

Question: 9.137P pg 1

Consider a steam power plant that operates on the ideal reheat Rankine cycle. The plant maintains the boiler at 7000 kPa, the reheat section at 800 kPa, and the condenser at 10 kPa. The mixture quality at the exit of both turbines is 93 percent. Determine the temperature at the inlet of each turbine and the cycle's thermal efficiency. Answers:373°C, 416°C, 37.6 percent

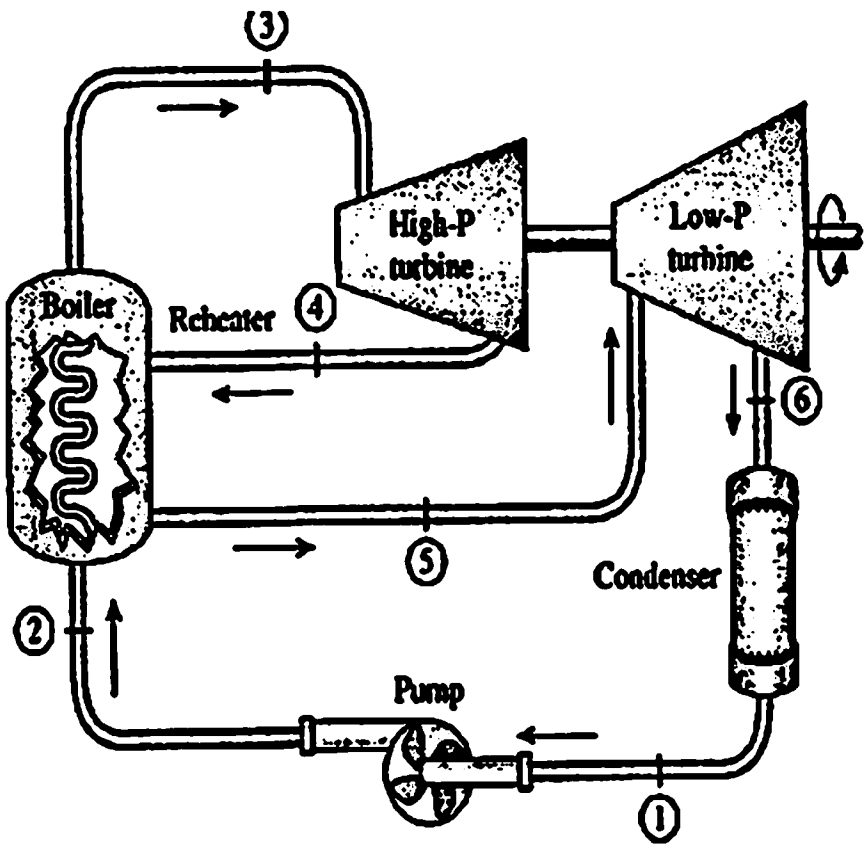


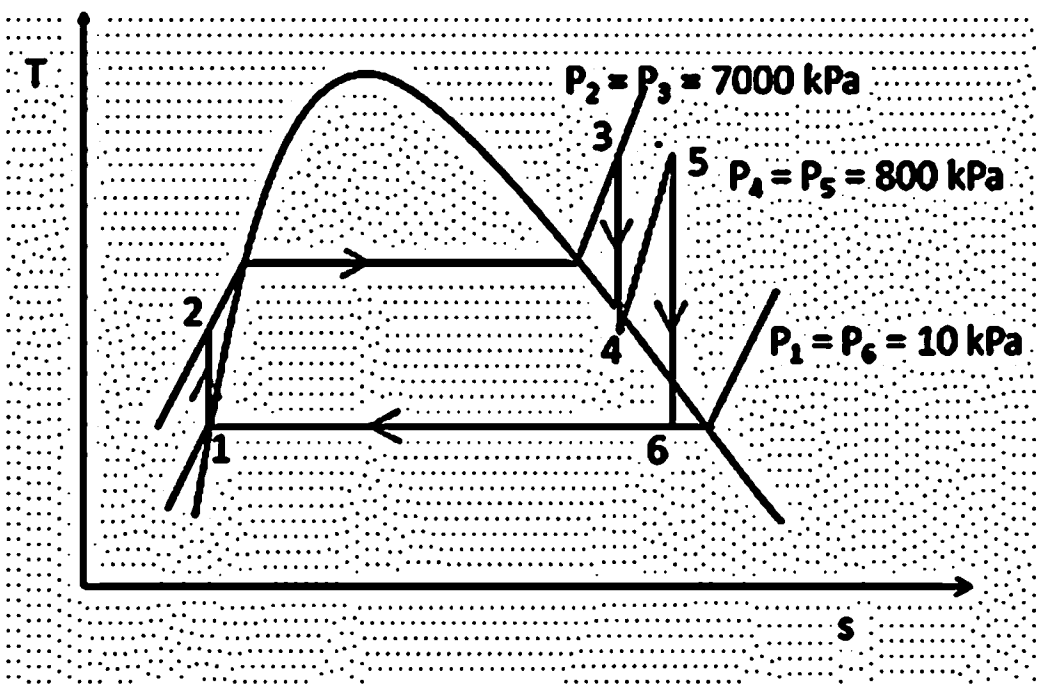
FIGURE P9–137

Step 1 of 13

Given,
Working fluid: water.
Pressure at the inlet of the high-pressure turbine $P_2 = P_3 = 7000$ kPa.
Pressure at the inlet of the low-pressure turbine $P_4 = P_5 = 800$ kPa.
Condenser Pressure $P_1 = P_6 = 10$ kPa.
The mixture quality at the exit of both the turbines is $x_r = 93\%$.

Step 2 of 13

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



Step 3 of 13

State1:
From the saturated water
$P_1 = 10$ kPa And saturated liquid,
$h_1 = h_f = 191.81$ kJ/kg
$v_1 = v_f = 0.001010\ m^3/kg$

Step 4 of 13

State 2:
$P_2 = 7000$ kPa
The work done in the compression process is given by,
$w_{pump} = v_1 \times (P_2 - P_1)$
$w_{pump} = 0.001010 m^3/kg \times (7000\ kPa - 10\ kPa) \times (1kJ/1\ kPa\ m^3)$
$w_{pump} = 7.06$ kJ/kg

Step 5 of 13

Enthalpy at state 2 is
$h_2 = h_1 + w_{pump}$
$= 191.81 + 7.06$
$= 198.87$ kJ/kg

Step 6 of 13

State 6:
$P_6 = P_1 = 10$ kPa and quality of steam after expension through turbine is 0.93
At these conditions,
$h_6 = h_f + x_6 h_{fg}$
$= 191.81 + 0.93(2392.1)$
$= 2416.463 kJ/kg$
And entropy is
$s_6 = s_f + x_6(s_{fg})$
$= 0.6492 + 0.93(7.4996)$
$= 7.623 kJ/kg \cdot K$

Step 7 of 13

State 5:
Since process 5-6 isentropic process.
From the superheated water
$P_5 = 800$ kPa and $s_5 = s_6 = = 7.623 kJ/kg \cdot K$
$$h_5 = 3267.7 + \frac{(7.623 - 7.5735)(3267.7 - 3162.2)}{(7.8692 - 7.5735)}$$
$= 3285.360 kJ/kg$

$P_5 = 800\,\text{kPa}$ and $s_5 = s_6 = = 7.629\,\text{kJ/kg}\,\text{K}$ P pg 2

$$h_5 = 3267.7 + \frac{(7.623 - 7.5735)(3267.7 - 3162.2)}{(7.8692 - 7.5735)}$$

$$= 3285.360\,\text{kJ/kg}$$

$$T_5 = 400 + \frac{(7.623 - 7.5735)(3267.7 - 3162.2)}{(7.8692 - 7.5735)}$$

$$= 417.66^\circ\text{C}$$

Temperature at the inlet of low pressure turbine inlet $\boxed{T_5 = 417.66^\circ\text{C}}$

Step 8 of 13

State 4:

$P_4 = 800\,\text{kPa}$ and and quality of steam after expension through turbine is 0.93 at these conditions,

From the saturated water pressure table

At $P_4 = 800\,\text{kPa}$

$$h_4 = h_f + x_4(h_{fg})$$

$$= 720.87 + 0.93(2047.5)$$

$$= 2625.0\,\text{kJ/kg}$$

And entropy is

$$s_4 = s_f + x_4(s_{fg})$$

$$= 2.0457 + 0.93(4.6160)$$

$$= 6.3385\,\text{kJ/kg}\cdot\text{K}$$

Step 9 of 13

State 3:

From the superheated water

$P_3 = 7000\,\text{kPa}$ and $s_3 = s_4 = 6.338\,\text{kJ/kg}\cdot\text{K}$

$$h_5 = 3016.9 + \frac{(6.3385 - 6.2305)(3159.2 - 3016.9)}{(6.4502 - 6.2305)}$$

$$= 3086.851\,\text{kJ/kg}$$

$$T_3 = 350 + \frac{(6.3385 - 6.2305)(400 - 350)}{(6.4502 - 6.2305)}$$

$$= 374.578^\circ\text{C}$$

Temperature at the inlet of high pressure turbine inlet $\boxed{T_5 = 374.578^0\text{C}}$

Step 10 of 13

Turbine work output is given by,

$$W_T = (h_3 - h_4) + (h_5 - h_6)$$

$$W_T = (3086.851 - 2625) + (3285.360 - 2416.463\ )$$

$$= 1330.748\,\text{kJ/kg}$$

Step 11 of 13

Net work output is given by,

$$W_{net} = W_T - W_{pump}$$

$$W_{net} = 1330.748\ - 7.06$$

$$= 1323.288\,\text{kJ/kg}$$

Step 12 of 13

Heat supplied is given by,

$$Q_{in} = (h_3 - h_2) + (h_5 - h_4)$$

$$= (3086.851 - 198.87) + (3285.360 - 2625)$$

$$= 3548.341\,\text{kJ/kg}$$

Step 13 of 13

Efficiency of the cycle is given by,

$$\eta_{Rankin} = \frac{W_{net}}{Q_{in}} = \frac{1323.288}{3548.341}$$

$$= 0.377$$

$\boxed{\eta_{Rankin} = 0.377}$

Question: 9.138P

A steam power plant operates on an ideal reheat Rankine cycle between the pressure limits of 15 MPa and 10 kPa. The mass flow rate of steam through the cycle is 12 kg/s. Steam enters both stages of the turbine at 500°C. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10 percent, determine *(a)* the pressure at which reheating takes place, *(b)* the total rate of heat input in the boiler, and (c) the thermal efficiency of the cycle. Also, show the cycle on a *T-s* diagram with respect to saturation lines.

Step 1 of 13

Given:

Pressure in the boiler $P_3 = P_2 = 15\text{MPa}$

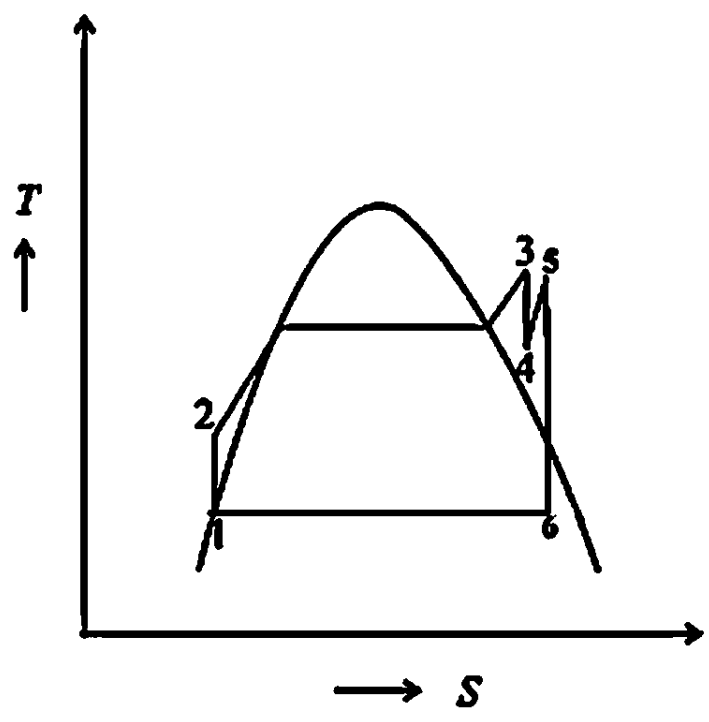
Condenser pressure $P_6 = P_1 = 10\text{kPa}$

Mass flow rate of steam $\dot{m} = 12\text{kg/s}$

Steam enters both stages of the turbine at temperature $T_3 = T_5 = 500^0\text{C}$

Dryness fraction at the exit of the low pressure turbine is $x_6 = 90\%$

Step 2 of 13



Step 3 of 13

At $P_6 = P_1 = 10\text{kPa}$ from saturated water tables

$h_1 = h_f = 191.81\text{ kJ/kg}$

$v_1 = v_f = 0.00101\text{ m}^3\text{/kg}$

Pump work

$w_{pump} = -v_1(P_2 - P_1)$

$w_{pump} = -0.00101 \times (15000 - 10)$

$w_{pump} = -15.14\text{ kJ/kg}$

Step 4 of 13

By applying first law to the process 1-2

$h_1 = h_2 + w_{pump}$

$h_2 = h_1 - w_{pump}$

$h_2 = 191.81 + 15.14$

$h_2 = 206.95\text{ kJ/kg}$

Step 5 of 13

At $P_3 = 15000\text{ kPa}$ and $T_3 = 500^0\text{C}$ from superheated water tables

$h_3 = 3310.8\text{ kJ/kg}$

$s_3 = 6.348\text{ kJ/kg.K}$

Step 6 of 13

Process 5-6 is isentropic expansion

$s_6 = s_5$

At $P_6 = 10\text{ kPa}$ from saturated water tables

$s_6 = s_{f@10kPa} + x_6 s_{fg@10kPa}$

$s_6 = 0.6492 + 0.9 \times 7.4996$

$s_6 = 7.3988\text{ kJ/kg.K}$

Enthalpy at state-6

$h_6 = h_f + x_6 h_{fg}$

$h_6 = 191.81 + 0.9 \times 2392.1$

$h_6 = 2344.7\text{ kJ/kg}$

Step 7 of 13

(a)

Process 5-6 is isentropic expansion

$s_5 = s_6$

At $T_5 = 500^0\text{C}$ and $s_5 = s_6 = 7.3988\text{kJ/kg.K}$ from superheated tables

$$P_5 = 2000 + (2500 - 2000)\left(\frac{7.3988 - 7.4337}{7.3254 - 7.4337}\right)$$

Pressure at state-5

$\boxed{P_5 = 2161.126\text{ kPa}}$

Step 8 of 13

Enthalpy at state-5

$$h_5 = 3468.3 + (3462.8 - 3468.3)\left(\frac{7.3988 - 7.4337}{7.3254 - 7.4337}\right)$$

$h_5 = 3466.527\text{ kJ/kg}$

Step 9 of 13

At $P_4 = P_5 = 2161.126\text{ kPa}$ and $s_4 = s_3 = 6.348\text{kJ/kg.K}$ by using EES software

Enthalpy at state-4

$h_4 = 2818.5\text{ kJ/kg}$

Step 10 of 13

Net work output from the turbine is

$w_{net} = (h_3 - h_4) + (h_5 - h_6)$

$w_{net} = (3310.8 - 2818.5) + (3466.527 - 2344.7)$

$w_{net} = 1614.127\text{ kJ/kg}$

Step 11 of 13

The heat input is

$q_{in} = (h_3 - h_2) + (h_5 - h_4)$

$q_{in} = (3310.8 - 206.95) + (3466.527 - 2818.5)$

$q_{in} = 3751.87\text{ kJ/kg}$

Step 12 of 13

The total rate of heat input in the boiler is

$Q_{in} = m_{steam} \times q_{in}$

$Q_{in} = 12 \times 3751.87$

$\boxed{Q_{in} = 45022.44\text{kW}}$

Step 13 of 13

(c)

Thermal efficiency of the cycle,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$\eta_{th} = \frac{1614.127}{3751.87}$$

$$= 0.4302$$

$\boxed{\eta_{th} = 43.02\%}$

Question: 9.139P pg 1

A steam power plant operates on the reheat Rankine cycle. Steam enters the high-pressure turbine at 12.5 MPa and 550°C at a rate of 7.7 kg/s and leaves at 2 MPa. Steam is then reheated at constant pressure to 450°C before it expands in the low-pressure turbine. The isentropic efficiencies of the turbine and the pump are 85 percent and 90 percent, respectively. Steam leaves the condenser as a saturated liquid. If the moisture content of the steam at the exit of the turbine is not to exceed 5 percent, determine (a) the condenser pressure, *(b)* the net power output, and *(c)* the thennal efficiency

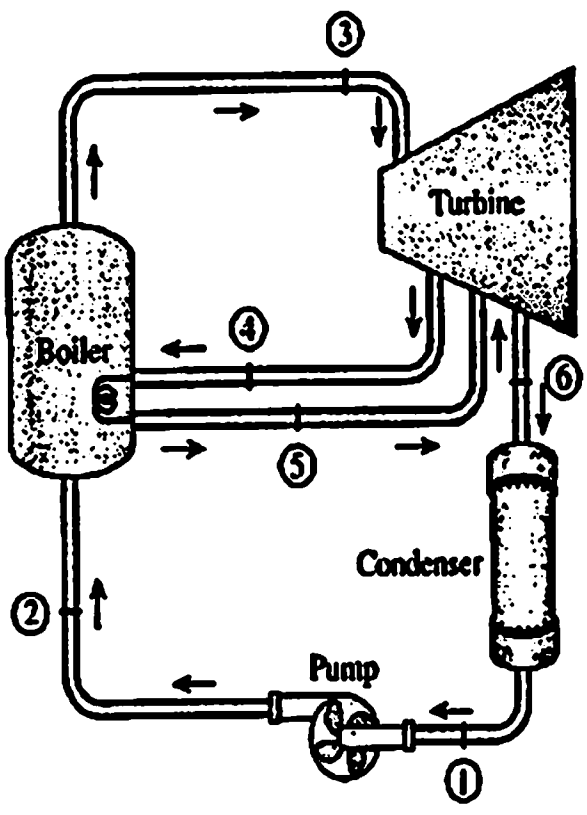
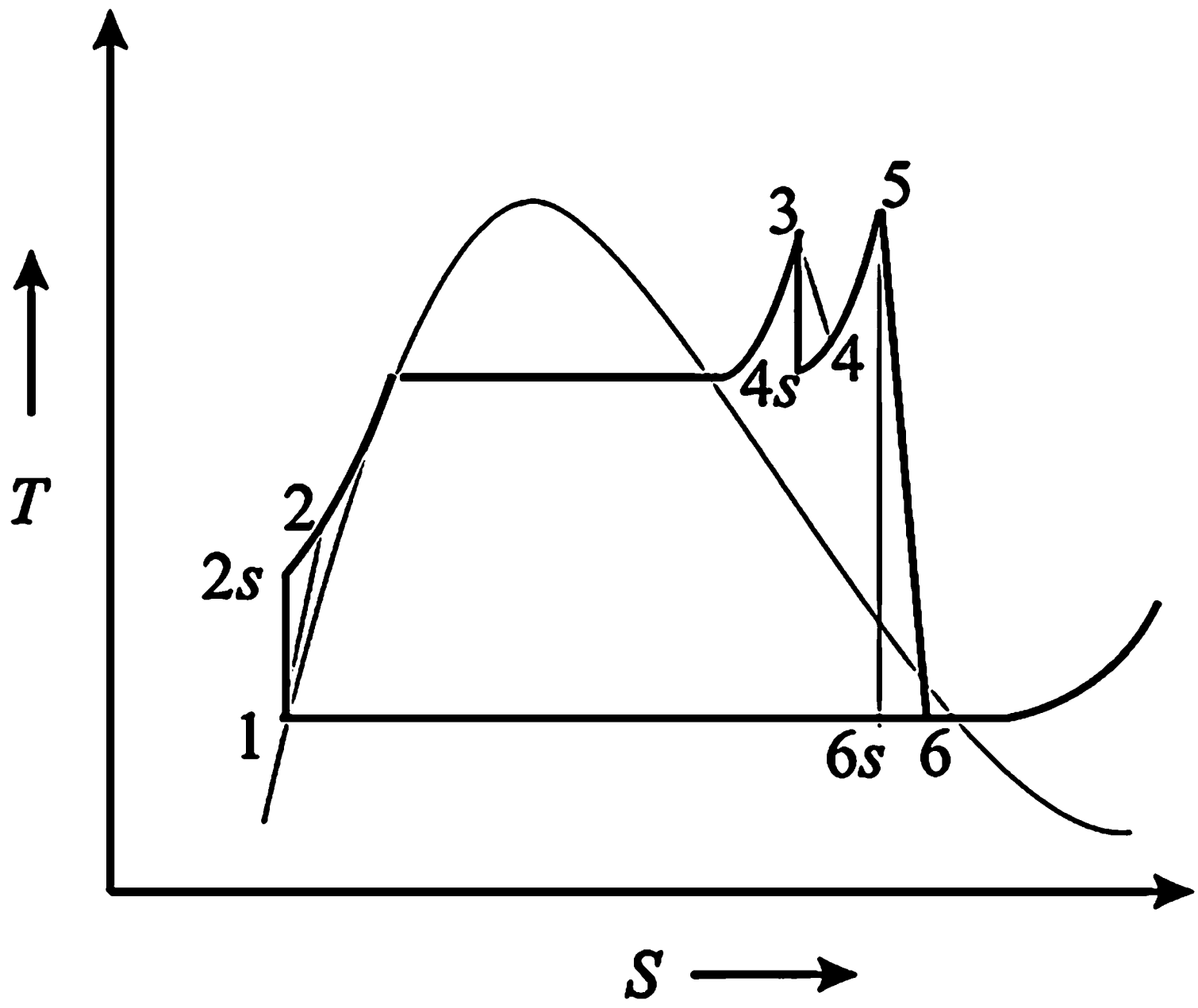


FIGURE P9–139

Step 1 of 12

Draw the *T-s* Diagram of the thermodynamic cycle is as follows:



Step 2 of 12

Obtain the properties of super-heated water vapor at $P_3 = 12.5\,\text{MPa}$ and $T_3 = 550\,^\circ\text{C}$ from Table A-6, "Properties of superheated vapor", from the text book.

Specific enthalpy, $h_3 = 3476.5\ \text{kJ/kg}$

Specific entropy, $s_3 = 6.6317\ \text{kJ/kg.K}$

Since the process 3-4*s* is isentropic process, entropies remains constant.

$$s_{4s} = s_3$$
$$= 6.6317\ \text{kJ/kg.K}$$

Obtain the properties of super-heated water vapor at $P_4 = 2\,\text{MPa}$ and $s_{4s} = 6.6317\ \text{kJ/kg.K}$ from Table A-6, "Properties of superheated vapor", from the text book by using interpolation between 250°C and 300°C.

$$\frac{h_{4s} - 2903.3}{3024.2 - 2903.3} = \frac{6.6317 - 6.5475}{6.7684 - 6.5475}$$
$$h_{4s} = 2949.38\ \text{kJ/kg}$$

Step 3 of 12

Calculate the actual enthalpy at state 4 by using following relation:

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

Substitute $3476.5\ \text{kJ/kg}$ for h_3, $2949.38\ \text{kJ/kg}$ for h_{4s}, and 0.85 for η_t.

$$0.85 = \frac{3476.5 - h_4}{3476.5 - 2949.38}$$
$$h_4 = 3028.44\ \text{kJ/kg}$$

Obtain the properties of super-heated water vapor at $P_5 = 2\,\text{MPa}$ and $T_5 = 450\,^\circ\text{C}$ from Table A-6, "Properties of superheated vapor", from the text book by using interpolation between 400°C and 500°C.

Specific enthalpy:

$$\frac{h_5 - 3248.4}{3468.3 - 3248.4} = \frac{450 - 400}{500 - 400}$$
$$h_5 = 3358.35\ \text{kJ/kg}$$

Specific entropy:

$$\frac{s_5 - 7.1292}{7.4337 - 7.1292} = \frac{450 - 400}{500 - 400}$$
$$s_5 = 7.28145\ \text{kJ/kg}$$

$$\frac{s_5 - 7.1292}{7.4337 - 7.1292} = \frac{450 - 400}{500 - 400}$$
$$s_5 = 7.28145 \text{ kJ/kg}$$

Step 4 of 12

(a)

Assume the pressure at condenser is 9 kPa and condition of the steam is 0.95.

Since the process 3-4*s* is isentropic process, entropies remains constant.

$$s_{6s} = s_5$$
$$= 7.2814 \text{ kJ/kg.K}$$

Obtain the isentropic enthalpy at exit of the second turbine at 9 kPa and $s_{6s} = 7.2814 \text{ kJ/kg.K}$ from the EES.

Specific enthalpy, $h_{6s} = 2293.4 \text{ kJ/kg}$

Obtain the properties of saturated water at 9 kPa from Table A-5, "Properties of saturated water", from the text book.

Specific enthalpy of saturated liquid, $h_f = 183.2 \text{ kJ/kg}$

Specific enthalpy of vaporization, $h_{fg} = 2396.9 \text{ kJ/kg}$

Step 5 of 12

Calculate the value of actual enthalpy at exit of the second turbine.

$$h_6 = h_{f6} + x_6 h_{fg6}$$

Substitute 183.2 kJ/kg for h_{f6}, 0.95 for x_6 and 2396.9 kJ/kg for h_{fg6}

$$h_6 = 183.2 + (0.95 \times 2396.9)$$
$$= 2460.255 \text{ kJ/kg}$$

Write the relation for isentropic efficiency of the turbine.

$$\eta_t = \frac{h_5 - h_6}{h_5 - h_{6s}}$$

Substitute 3358.35 kJ/kg for h_5, 2293.4 kJ/kg for h_{6s}, and 2460.255 kJ/kg for h_6.

$$\eta_t = \frac{3358.35 - 2460.255}{3358.35 - 2293.4}$$
$$= 0.843$$
$$\neq 0.85$$

Step 6 of 12

Assume the pressure at condenser is 10 kPa and condition of the steam is 0.95.

Since the process 3-4*s* is isentropic process, entropies remains constant.

$$s_{6s} = s_5$$
$$= 7.2814 \text{ kJ/kg.K}$$

Obtain the isentropic enthalpy at exit of the second turbine at 10 kPa and $s_{6s} = 7.2814 \text{ kJ/kg.K}$ from the EES.

Specific enthalpy, $h_{6s} = 2307 \text{ kJ/kg}$

Obtain the properties of saturated water at 10 kPa from Table A-5, "Properties of saturated water", from the text book.

Specific enthalpy of saturated liquid, $h_f = 191.81 \text{ kJ/kg}$

Specific enthalpy of vaporization, $h_{fg} = 2392.1 \text{ kJ/kg}$

Step 7 of 12

Calculate the value of actual enthalpy at exit of the second turbine.

$$h_6 = h_{f6} + x_6 h_{fg6}$$

Substitute 183.2 kJ/kg for h_{f6}, 0.95 for x_6 and 2396.9 kJ/kg for h_{fg6}

$$h_6 = 191.8 + (0.95 \times 2392.1)$$
$$= 2464.295 \text{ kJ/kg}$$

Write the relation for isentropic efficiency of the turbine.

$$\eta_t = \frac{h_5 - h_6}{h_5 - h_{6s}}$$

Substitute 3358.35 kJ/kg for h_5, 2307 kJ/kg for h_{6s}, and 2464.295 kJ/kg for h_6.

$$\eta_t = \frac{3358.35 - 2464.295}{3358.35 - 2307}$$
$$= 0.8503$$
$$\neq 0.85$$

Step 8 of 12

Therefore condenser pressure will be between 9 kPa and 10 kPa.

Calculate the condenser pressure by using interpolation between 9 kPa and 10 kPa:

$$\frac{0.85 - 0.843}{0.8503 - 0.843} = \frac{P_6 - 9}{10 - 9}$$
$$P_6 = 9.9 \text{ kPa}$$

Therefore the condenser pressure is, $\boxed{9.9 \text{ kPa}}$.

Step 9 of 12

Therefore the condenser pressure is, $\boxed{9.9\text{ kPa}}$ 9P pg 3

Step 9 of 12

Obtain the properties of the water at exit of the condenser at $P_6 = 9.9\text{ kPa}$ from Table A-5, "Properties of saturated water", from the text book.

Specific enthalpy, $h_1 = 190.9\text{ kJ/kg}$

Specific entropy, $s_1 = 0.646\text{ kJ/kg.K}$

Specific volume, $v_1 = 0.001\text{ m}^3\text{/kg}$

Calculate the work done by the pump by using the following relation:

$$w_p = v_1(P_2 - P_6)$$

Substitute $0.001\text{ m}^3\text{/kg}$ for v_1, 9.9 kPa for P_6, and 12.5 MPa for P_2.

$$\begin{aligned} w_p &= 0.001\left[\left(12.5\times10^3\right)-9.9\right] \\ &= 12.49\text{ kJ/kg} \end{aligned}$$

Step 10 of 12

Calculate the isentropic enthalpy at exit of the pump by using the following relation:

$$w_p = h_{2s} - h_1$$

Substitute 190.9 kJ/kg for h_1, and 12.49 kJ/kg for w_p.

$$\begin{aligned} 12.49 &= h_{2s} - 190.9 \\ h_{2s} &= 12.49 + 190.9 \\ h_{2s} &= 203.39\text{ kJ/kg} \end{aligned}$$

Calculate the actual enthalpy at exit of the pump by using the following relation:

$$\eta_p = \frac{h_{2s} - h_1}{h_2 - h_1}$$

Substitute 203.39 kJ/kg for h_{2s}, 190.9 kJ/kg for h_1, and 0.9 for η_p.

$$\begin{aligned} 0.9 &= \frac{203.39-190.9}{h_2-190.9} \\ h_2 &= 204.78\text{ kJ/kg} \end{aligned}$$

Step 11 of 12

(b)

Calculate the net power output.

$$P_{net} = \dot{m}\left[\left(h_3 - h_4\right)+\left(h_5 - h_6\right)\right]$$

Here, $\dot{m}$ is the mass flow rate of the steam.

Substitute 7.7 kg/s for $\dot{m}$, 3358.35 kJ/kg for h_5, 2464.295 kJ/kg for h_6, 3476.5 kJ/kg for h_3, and 3028.44 kJ/kg for h_4.

$$\begin{aligned} P_{net} &= 7.7\left[\left(3476.5-3028.44\right)+\left(3358.35-2464.295\right)\right] \\ &= 10334.2\text{ kW} \\ &= 10.3\text{ MW} \end{aligned}$$

Therefore, net power output is, $\boxed{10.4\text{ MW}}$.

Step 12 of 12

(c)

Calculate the thermal efficiency by using the following relation:

$$\begin{aligned} \eta_{th} &= \frac{P_{net}}{Q_{in}} \\ &= \frac{P_{net}}{\dot{m}\left[\left(h_3 - h_2\right)+\left(h_5 - h_4\right)\right]} \end{aligned}$$

Here, Q_{in} is the amount of heat supplied.

Substitute 7.7 kg/s for $\dot{m}$, 10468 kW for P_{net}, 3476.5 kJ/kg for h_3, 3358.35 kJ/kg for h_5, 204.78 kJ/kg for h_2, and 3028.44 kJ/kg for h_4.

$$\begin{aligned} \eta_{th} &= \frac{10334.2}{7.7\left[\left(3476.5-204.78\right)+\left(3358.35-3028.44\right)\right]} \\ &= \frac{10334.2}{27732.55} \\ &= 37\% \end{aligned}$$

Therefore, thermal efficiency of the cycle is, $\boxed{37\%}$.

Question: 9.140P

Why do we study the reversed Carnot cycle even though it is not a realistic model for refrigeration cycles?

Step 1 of 2

Carnot cycle is totally reversible cycle that consists of two reversible isothermal and two reversible isentropic processes. It has a maximum efficiency for given temperature limits. The two isothermal heat transfer processes are not difficult to achieve in practice since maintaining the constant pressure automatically fixes the temperature of a two phase mixture at the saturation value. So, it is difficult to maintain isothermal conditions during the heat absorption and heat rejection processes. Therefore we conclude that reverse Carnot cycle cannot be approximated in actual device and it is not realistic model for refrigeration cycle.

Step 2 of 2

And reverse Carnot cycle is most effective refrigeration cycle operating between two specified temperature levels. So it can serve as a standard against which actual refrigeration cycles are compared.

Question:

9.141P

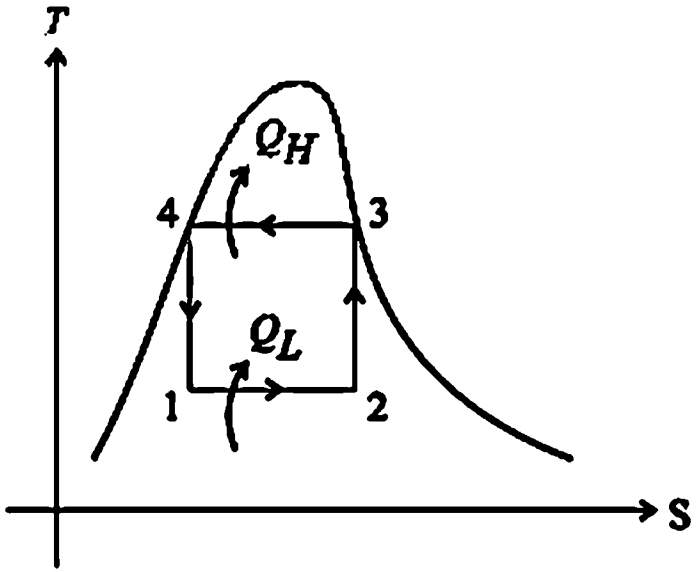
Why is the reversed Carnot cycle executed within the saturation dome not a realistic model for refrigeration cycles?

Step 1 of 1

The reversed Carnot cycle executed within the saturation dome is not a realistic model for refrigeration cycles.

Reason:

The two isothermal heat transfer processes (Processes 1-2 and 3-4) are not difficult to achieve in practice. The processes 2-3 and 4-1 cannot be approximated closely in practice. This is because process 2-3 involves the compression of a liquid-vapor mixture, which requires a compressor that will handle two phases, and process 4-1 involves the expansion of high-moisture content refrigerant in a turbine.



Question: 9.142P

A steady-flow Carnot refrigeration cycle uses refrigerant- 134a as the working fluid. The refrigerant changes from saturated vapor to saturated liquid at 40°C in the condenser as it rejects heat. The evaporator pressure is 100 kPa. Show the cycle on a *T-s* diagram relative to saturation lines, and determine *(a)* the coefficient of performance, *(b)* the amount of heat absorbed from the refrigerated space, and *(c)* the net work input. Answer: (a) 3.72, *(b)*128 kJ/kg, (c) 34.6 kJ/kg

Step 1 of 7

Given:

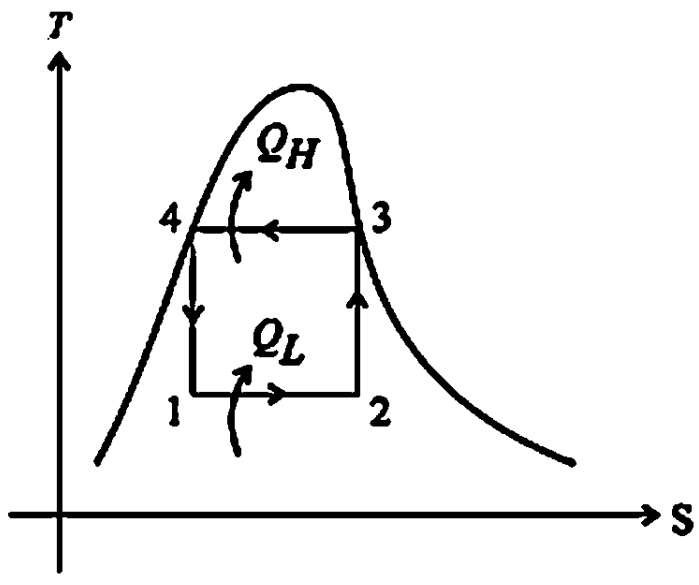
Refrigeration fluid: R-134a

Temperature of the condenser $T_H = 40^\circ C$

Pressure of evaporator $P_L = 100 kPa$

Step 2 of 7

T-S diagram for reversed Carnot refrigeration cycle :



Step 3 of 7

Condenser temperature

$T_H = 40^\circ C$

$= 273 + 40$

$= 313K$

Evaporator temperature

From the saturated refrigerant R-134a pressure table

At $P_L = 100 kPa$

$T_L = -26.37^\circ C$

$= 273 - 26.37^\circ C$

$= 246.63K$

Step 4 of 7

(a) Coefficient of performance of reversed Carnot refrigeration cycle is

$$(COP)_{R.C} = \frac{T_L}{T_H - T_L}$$

$$= \frac{246.63}{313 - 246.63}$$

$$= 3.715$$

$$\boxed{(COP)_{RC} = 3.715}$$

Step 5 of 7

b) Amount of heat absorbed from the refrigerant space, from the saturated refrigerant R-134a Temperature table

At $40^\circ C$

$$h_3 = h_g = 271.27 \frac{kJ}{kg}$$

$$h_4 = h_f = 108.26 \frac{kJ}{kg}$$

Thus,

Heat transferred is

$$q_H = h_3 - h_4$$

$$= \left(271.27 \frac{kJ}{kg}\right) - \left(108.26 \frac{kJ}{kg}\right)$$

$$= 163.01 \frac{kJ}{kg}$$

Step 6 of 7

Since reversed Carnot refrigeration cycle operates between the two temperature limits, so amount of heat supplied and rejection occurs at respective the temperature limits.

It gives

$$\frac{q_H}{q_L} = \frac{T_H}{T_L}$$

$$q_L = \frac{T_L}{T_H}(q_H)$$

$$= \frac{(246.63K)}{(313K)}\left(163.01 \frac{kJ}{kg}\right)$$

$$= 128.44 \frac{kJ}{kg}$$

$$\boxed{q_L = 128.44 \frac{kJ}{kg}}$$

Step 7 of 7

(c) The net work input

$$W_{net} = q_H - q_L$$

$$= \left(163.01 \frac{kJ}{kg}\right) - \left(128.44 \frac{kJ}{kg}\right)$$

$$= 34.57 \frac{kJ}{kg}$$

$$= 34.57 \frac{kJ}{kg}$$

$$\boxed{W_{net} = 34.57 \frac{kJ}{kg}}$$

Question: 9.143P

Refrigerant-134a enters the condenser of a steady-flow Carnot refrigerator as a saturated vapor at 90 psia, and it leaves with a quality of 0.05. The heat absorption from the refrigerated space takes place at a pressure of 30 psia. Show the cycle on a *T-s* diagram relative to saturation lines, and determine *(a)* the coefficient of performance, *(b)* the quality at the beginning of the heat-absorption process, and (c) the net work input.

Step 1 of 7

Given:
Working fluid: Refrigerant-134a
Pressure at warm space = 90 psia
Temperature of warm space,

$T_H = T_{sat@90psia} = 72.78^\circ F$

$T_H = 72.78+460.02\ R$

$T_H = 532.8\ R$

Pressure at refrigerated space = 30 psia
Temperature of refrigerated space,

$T_L = T_{sat\ @\ 30psia} = 15.37^\circ F$

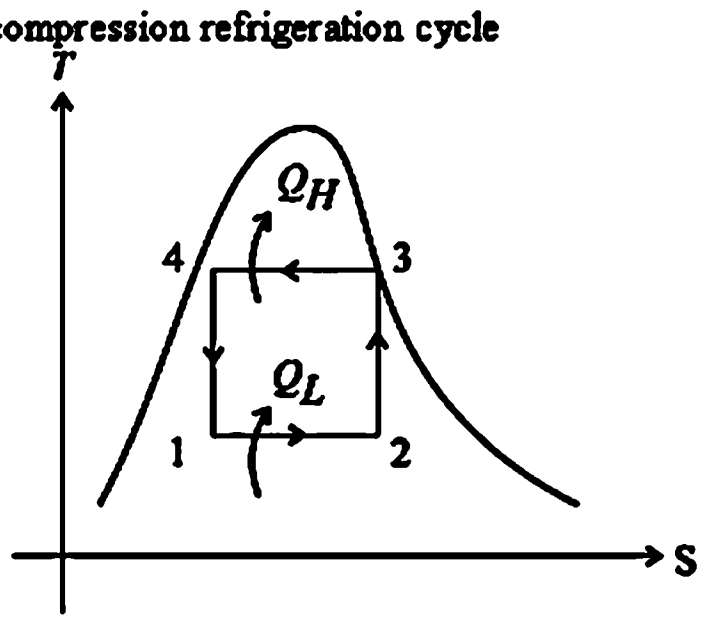
$T_L = 15.37+460.02R$

$T_L = 475.4\ R$

Given, quality $x_4 = 0.05$

Step 2 of 7

T-S diagram of vapor-compression refrigeration cycle



Step 3 of 7

From Saturated refrigerant-134a – Pressure table,

$s_{f@90psia} = 0.07481$ Btu/lbm.R

$s_{fg@90psia} = 0.14525$ Btu/lbm.R

$s_{g@90psia} = 0.22006$ Btu/lbm·R

$s_{f@30psia} = 0.03793$ Btu/lbm.R

$s_{fg@30psia} = 0.18589$ Btu/lbm.R

Step 4 of 7

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 5 of 7

(a) Coefficient of performance:
Coefficient of performance

$$COP = \frac{1}{\frac{T_H}{T_L}-1}$$

$$COP = \frac{1}{\frac{532.8}{475.4}-1}$$

$$COP = \frac{1}{0.12074}$$

$$\boxed{COP = 8.282}$$

Step 6 of 7

(b) The quality at the beginning of the heat-absorption process, x_1:
For the process 4-1,

$s_1 = s_4 = \left(s_f + x_4 \times s_{fg}\right)_{@90psia}$

$s_1 = 0.07481+0.05\times0.14525$

$s_1 = 0.0820725$ Btu/lbm.R

We know that

$s_1 = \left(s_f + x_1 \times s_{fg}\right)_{@30psia}$

$0.0820725 = 0.03793 + x_1 \times 0.18589$

$$x_1 = \frac{(0.0820725-0.03793)}{0.18589}$$

$$\boxed{x_1 = 0.2374}$$

Step 7 of 7

(c) Net work input:
Net work input = Area enclosed in the T-s diagram

$w_{net} = (T_H - T_L)\times(s_3 - s_4)$

Where $s_3 = s_{g\ @\ 90psia} = 0.22006$ Btu/lbm·R

$w_{net} = (532.8-475.4)\times(0.22006-0.0820725)$

$w_{net} = (57.4)\times(0.1379875)$

$$\boxed{w_{net} = 7.9205\ \text{Btu/lbm}}$$

Question: 9.144P

Does the ideal vapor-compression refrigeration cycle involve any internal irreversibilities?

Step 1 of 1

Yes. The ideal vapor-compression refrigeration cycle involves one irreversible process (throttling).

Explanation:
Unlike other ideal cycles (Carnot cycle), the ideal vapor compression refrigeration cycle is not an internally reversible cycle since it involves an irreversible (throttling) process. This process is maintained in the cycle to make it a more realistic model for the actual vapor-compression refrigeration cycle.

Question: 9.145P

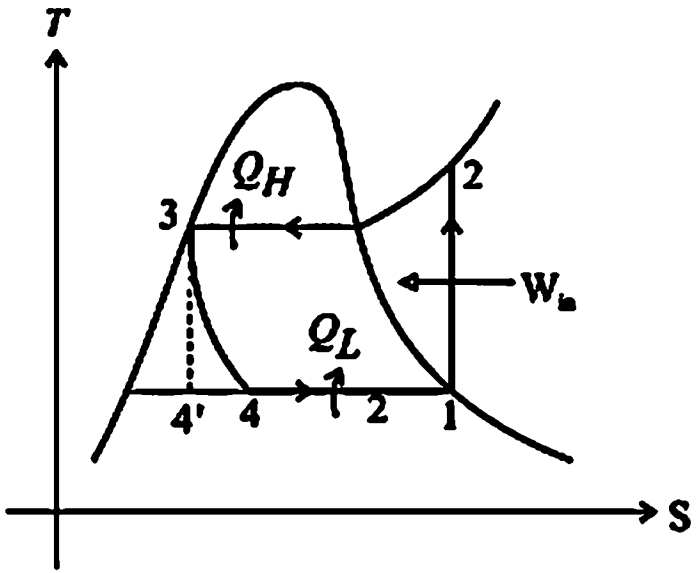
Why is the throttling valve not replaced by an isentropic turbine in the ideal vapor-compression refrigeration cycle?

Step 1 of 2

Replacing the expansion valve by a turbine is not practical.

Explanation:
If the throttling device were replaced by an isentropic turbine, the refrigerant would enter the evaporator at state $4'$ instead of state 4. As a result, the refrigeration capacity would increase and the net work input would decrease by the amount of work output of turbine. Even though there are benefits on adding the turbine but the added cost and complexity make it unpractical

Step 2 of 2



Question: 9.146P

It is proposed to use water instead of refrigerant-134a as the working fluid in air-conditioning applications where the minimum temperature never falls below the freezing point. Would you support this proposal? Explain.

Step 1 of 1

We can't use water instead of refrigerant R-134a.

Explanation:
For the refrigerants in evaporator saturated pressure for given temperature is very low which is not practical to maintain with water.

Question: 9.147P

In a refrigeration system, would you recommend condensing the refrigerant-134a at a pressure of 0.7 or 1.0 MPa if heat is to be rejected to a cooling medium at 15°C?Why?

Step 1 of 1

We must recommend the pressure of 0.7 MPa.

Explanation:
We must maintain minimum temperature difference of 10°C for effective heat transfer. Given that cooling medium is at 15°C. So, we must maintain condensation temperature of the refrigerant as 25°C. The saturation pressure corresponding to 25°C is very much near to 0.7 MPa (From Saturated refrigerant R-134a temperature table)

Question: 9.148P

Docs the area enclosed by the cycle on a *T-s* diagram represent the net work input for the reversed Carnot cycle? How about for the ideal vapor-compression refrigeration cycle?

Step 1 of 1

Yes, the area enclosed by the cyclic curve on a *T-s* diagram represents the net work input for the reversed Carnot cycle but not for the ideal vapor-compression refrigeration cycle.

Explanation:

The area enclosed by the cyclic curve on a *T-s* diagram represents the net work input for the reversed Carnot cycle as it contains all internally reversible processes where the ideal vapor-compression refrigeration cycle involves one irreversible process (throttling).

Question:

9.149P

Consider two vapor-compression refrigeration cycles. The refrigerant enters the throttling valve as a saturated liquid at 30°C in one cycle and as subcooled liquid at 30°C in the other one. The evaporator pressure for both cycles is the same. Which cycle do you think will have a higher COP?

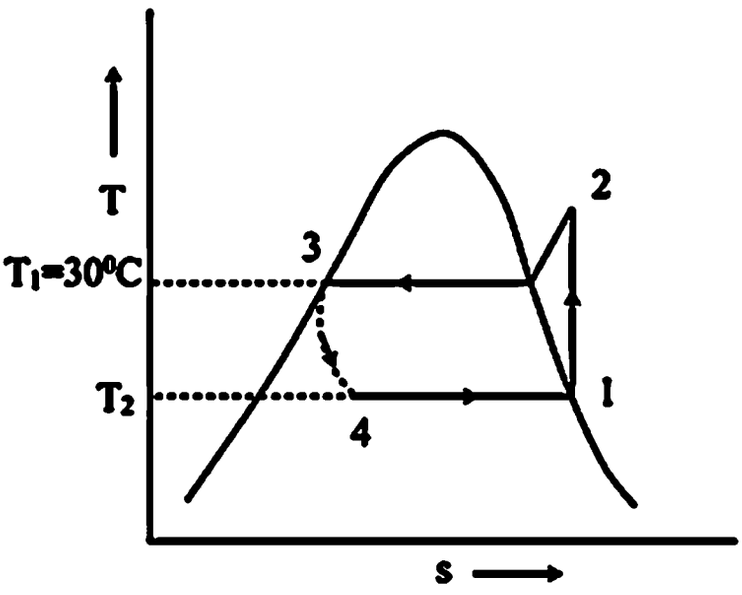
Step 1 of 3

Consider two vapor-compression refrigeration cycles. The refrigerant enters the throttling valve as a saturated liquid at 30°C in one cycle and as sub cooled liquid at 30°C in the other one. The evaporator pressure for both cycles is the same.

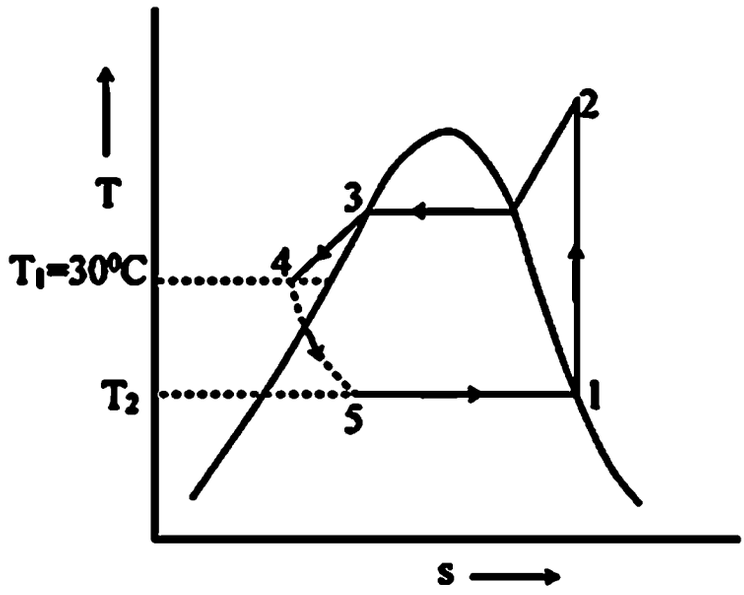
The cycle that involves saturated liquid at 30°C will have a higher COP than the cycle that involves sub cooled liquid at 30°C.

Step 2 of 3

Draw the T-s diagrams for the two cycles.



(A) Saturated vapour



(B) sub cooled vapour

Step 3 of 3

Explanation:

It require a smaller work input for the same refrigeration capacity if we use the cycle that involves saturated liquid at 30°C than the cycle that involves sub cooled liquid at 30°C.

COP can be defined as

$$COP_R = \frac{\text{Cooling effect}}{\text{Work input}}$$

So, The cycle that involves saturated liquid at 30°C will have a higher COP.

Question: 9.150P

The COP of vapor-compression refrigeration cycles improves when the refrigerant is subcooled before it enters the throttling valve. Can the refrigerant be subcooled indefinitely to maximize this effect, or is there a lower limit? Explain.

Step 1 of 1

It is undesirable to route the refrigerant to the throttling valve before the refrigerant is completely condensed. Therefore, the refrigerant is sub cooled somewhat before it enters the throttling valve. The COP of vapor-compression refrigeration cycles improves when the refrigerant is sub cooled before it enters the throttling valve.

But we can't sub cool the refrigerant indefinitely to maximize this effect. The reason is that heat has to be transferred from the refrigerant to cooling medium, so the minimum temperature to which the refrigerant can be sub cooled is the temperature of the cooling medium. As the temperature of the coolant cannot be at indefinite low temperature, sub cooling the refrigerant indefinitely is not possible. Hence the minimum temperature of the coolant is the lower limit.

Question:

9.151P

A refrigerator operates on the ideal vapor-comprcssion refrigeration cycle and uses refrigerant -134a as the working fluid. The condenser operates at 300 psia and the evaporator 20°F. If an adiabatic, reversible expansion device were avail able and used to expand the liquid leaving the condenser. how much would the COP improve by using this device instead of the throttle device?

Step 1 of 8

Given:
Working fluid: Refrigerant-134a
Pressure at condenser, P_2 =300 psia
Temperature at evaporator, T_1 =20°F

Step 2 of 8

T-S diagram of vapor-compression refrigeration cycle



Question:

A refrigerator operates on the ideal vapor-comprcssion refrigeration cycle and uses refrigerant -134a as the working fluid. The condenser operates at 300 psia and the evaporator 20°F. If an adiabatic, reversible expansion device were avail able and used to expand the liquid leaving the condenser. how much would the COP improve by using this device instead of the throttle device?

Step 1 of 8

Given:
Working fluid: Refrigerant-134a
Pressure at condenser, P_2 =300 psia
Temperature at evaporator, T_1 =20°F

Step 2 of 8

T-S diagram of vapor-compression refrigeration cycle



Step 5 of 8

From saturated refrigerant-134a pressure table
P_2 =300 psia, Saturated liquid,
$h_3 = h_{f@300\,psia} = 66.339$ Btu/lbm
$s_3 = s_{f@300\,psia} = 0.12715$ Btu/lbm.R
For throttling, $h_4 = h_3 = 66.339$ Btu/lbm
If we use adiabatic, reversible expansion device, $s_{4s} = s_3$
$h_{4s} = 59.8$ Btu/lbm

Step 6 of 8

Assumptions:
1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 7 of 8

If we use throttling device, $COP_1 = \dfrac{h_1 - h_4}{h_2 - h_1}$

$COP_1 = \dfrac{105.98 - 66.339}{125.68 - 105.98}$

$COP_1 = 2.01223$

Step 5 of 8

From saturated refrigerant-134a pressure table
P_2 =300 psia, Saturated liquid,
$h_3 = h_{f@300\,psia} = 66.339$ Btu/lbm
$s_3 = s_{f@300\,psia} = 0.12715$ Btu/lbm.R
For throttling, $h_4 = h_3 = 66.339$ Btu/lbm
If we use adiabatic, reversible expansion device, $s_{4s} = s_3$
$h_{4s} = 59.8$ Btu/lbm

Step 6 of 8

Assumptions:
1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Question: 9.152P

An ideal vapor-compression refrigeration cycle that uses refrigerant-134a as its working fluid maintains a condenser at 1000 kPa and the evaporator at 4°C. Determine this! system's COP and the amount of power required to service a400 kW cooling load.

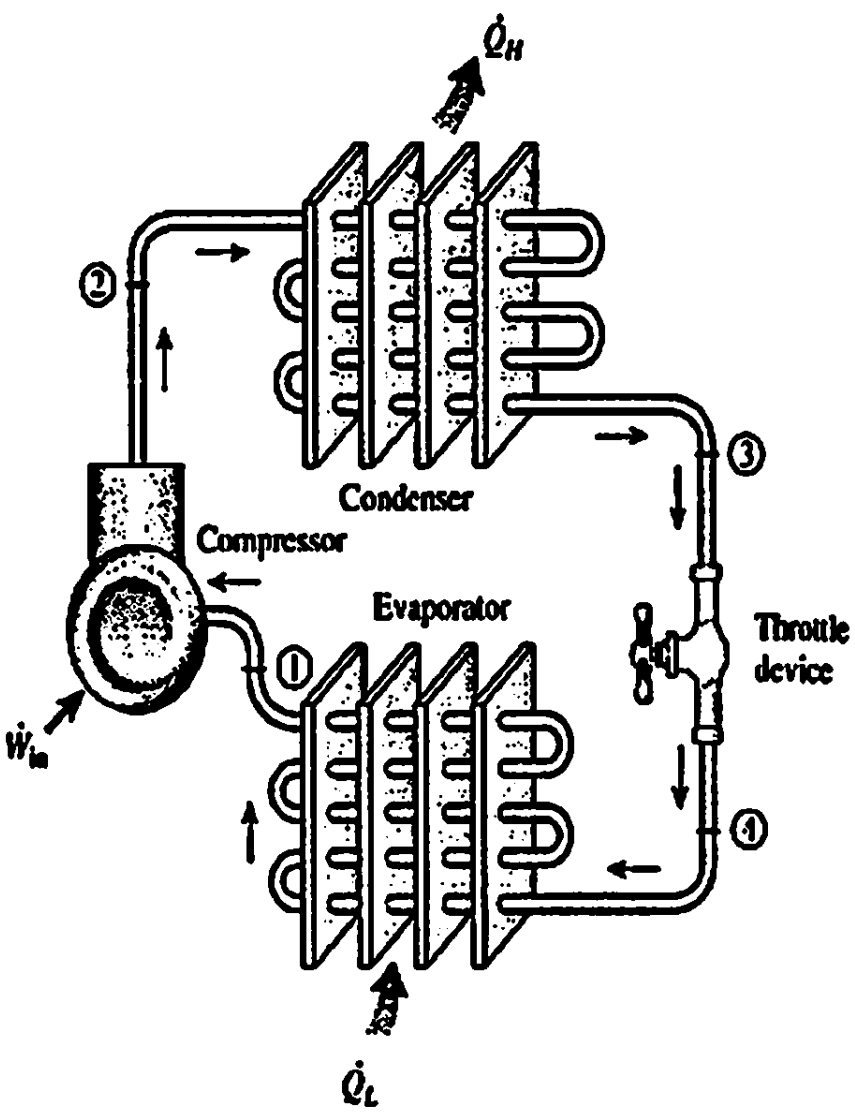


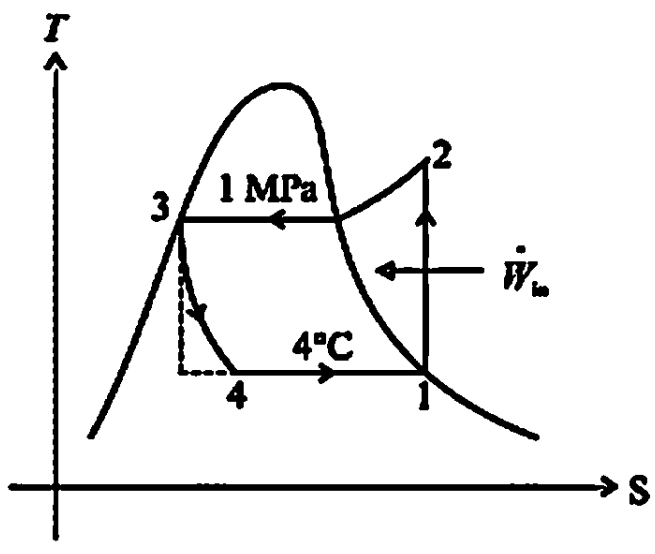
FIGURE P9–152

Step 1 of 9

Given:
Working fluid: Refrigerant-134a
Cooling load, $\dot{Q}_L = 400\ \text{kW}$
Pressure at condenser, $P_2 = 1\text{MPa}$
Temperature at evaporator, $T_1 = 4°\text{C}$

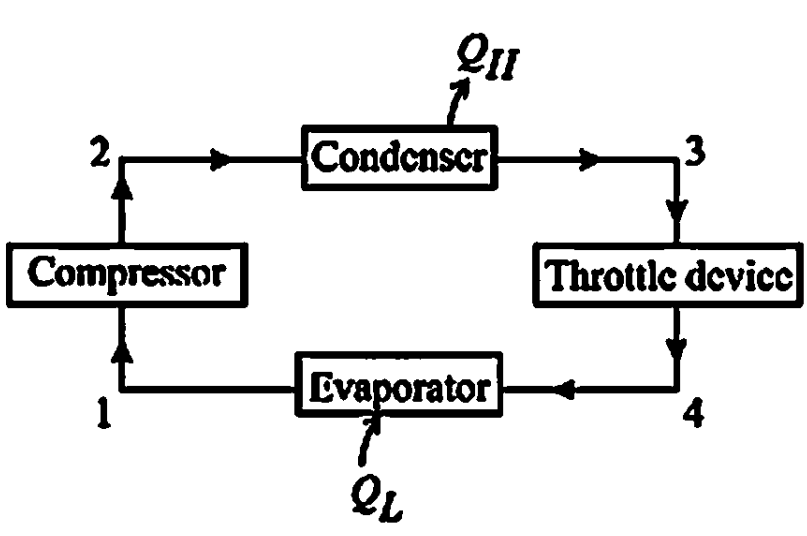
Step 2 of 9

T-S diagram of vapor-compression refrigeration cycle



Step 3 of 9

Block diagram of vapor-compression refrigeration cycle



Step 4 of 9

From Saturated refrigerant −134a temperature table
At $T_1 = 4°C$ saturated vapor,
$h_1 = h_{g@4°C} = 252.77\ \text{kJ/kg}$
$s_1 = s_{g@4°C} = 0.92927\ \text{kJ/kg.K}$

Step 5 of 9

From superheated refrigerant -134a table
$P_2 = 1\text{MPa}, s_2 = s_1 = 0.92927\ \text{kJ/kg.K}$

$$h_2 = 271.71 + \frac{0.92927 - 0.9179}{0.9525 - 0.9179} \times (282.74 - 271.71)$$

$h_2 = 275.3\ \text{kJ/kg}$

Step 6 of 9

From saturated refrigerant -134a pressure table
$P_2 = 1\text{MPa}$, Saturated liquid,
$h_3 = h_{f@1MPa} = 107.32\ \text{kJ/kg}$
For throttling, $h_4 = h_3 = 107.32\ \text{kJ/kg}$

Step 7 of 9

Assumptions:
1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 8 of 9

We know that cooling load, $\dot{Q}_L = \dot{m} \times (h_1 - h_4)$

So, Mass flow rate, $\dot{m} = \dfrac{\dot{Q}_L}{(h_1 - h_4)}$

$$\dot{m} = \frac{400}{(252.77 - 107.32)}$$

$\dot{m} = 2.75\ \text{kg/s}$

Power required, $\dot{W}_{in} = \dot{m} \times (h_2 - h_1)$

$\dot{W}_{in} = 2.75 \times (275.3 - 252.77)$

Power required, $\boxed{\dot{W}_{in} = 61.95\ \text{kW}}$

Step 9 of 9

COP of the refrigerator:

Coefficient of performance, $\text{COP} = \dfrac{\dot{Q}_L}{\dot{W}_{in}}$

$$\text{COP} = \frac{400}{61.95}$$

$\text{COP} = 6.4568$

$\boxed{\text{COP} = 6.46}$

Question: 9.153P

A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.12 and 0.7 MPa. The mass flow rate of the refrigerant is 0.05 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines. Determine *(a)* the rate of heat removal from the refrigerated space and the power input to the compressor, *(b)* the rate of heat rejection to the environment, and *(c)* the coefficient of performance. Answers:(a) 7.41 kW, 1.83 kW, *(b)*9.23 kW, (c) 4.06

Step 1 of 9

Given:

Working fluid: Refrigerant-134a

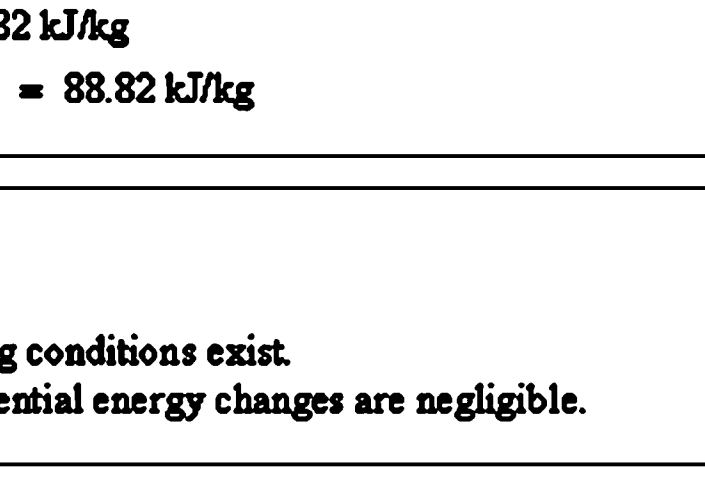
Mass flow rate of refrigerant, $\dot{m}_r = 0.05\ kg/s$

Pressure at condenser, $P_2 = 0.7\ MPa$

Pressure at evaporator, $P_1 = 0.12\ MPa$

Step 2 of 9

T-S diagram of vapor-compression refrigeration cycle



Step 3 of 9

From saturated refrigerant – 134a table

At $P_1 = 0.12\ MPa$ and saturated vapor,

$h_1 = h_{g@120\ kPa} = 236.97\ kJ/kg$

$s_1 = s_{g@120\ kPa} = 0.94779\ kJ/kg.K$

Step 4 of 9

From superheated refrigerant – 134a table

At $P_2 = 0.7\ MPa$, $s_2 = s_1 = 0.94779\ kJ/kg.K$

$$h_2 = 268.45 + \frac{0.94779 - 0.9313}{0.9641 - 0.9313} \times (278.57 - 168.45)$$

$h_2 = 273.5\ kJ/kg$

Step 5 of 9

At $P_2 = 0.7\ MPa$, Saturated liquid,

$h_3 = h_{f@0.7\ MPa} = 88.82\ kJ/kg$

For throttling, $h_4 = h_3 = 88.82\ kJ/kg$

Step 6 of 9

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 7 of 9

(a) Rate of heat removal from the refrigerated space, $\dot{Q}_L = \dot{m} \times (h_1 - h_4)$

$\dot{Q}_L = 0.05 \times (236.97 - 88.82)$

$\dot{Q}_L = 7.4075\ kW$

Rate of heat removal from the refrigerated space, $\boxed{\dot{Q}_L = 7.41\ kW}$

Power required, $\dot{W}_{in} = \dot{m} \times (h_2 - h_1)$

$\dot{W}_{in} = 0.05 \times (273.5 - 236.97)$

$\dot{W}_{in} = 1.8265$

Power required, $\boxed{\dot{W}_{in} = 1.83\ kW}$

Step 8 of 9

(b) The rate of heat rejection to the environment, $\dot{Q}_H = \dot{W}_{in} + \dot{Q}_L$

$\dot{Q}_H = 1.83 + 7.41$

$\boxed{\dot{Q}_H = 9.23\ kW}$

Step 9 of 9

(c) COP of the refrigerator:

Coefficient of performance, $COP = \dfrac{\dot{Q}_L}{\dot{W}_{in}}$

$$COP = \frac{7.41}{1.83}$$

$\boxed{COP = 4.05}$

Question: 9.154P

Repeat Prob. 9–153 for a condenser pressure of 0.9 MPa.

Step 1 of 9

Given:

Working fluid: Refrigerant-134a

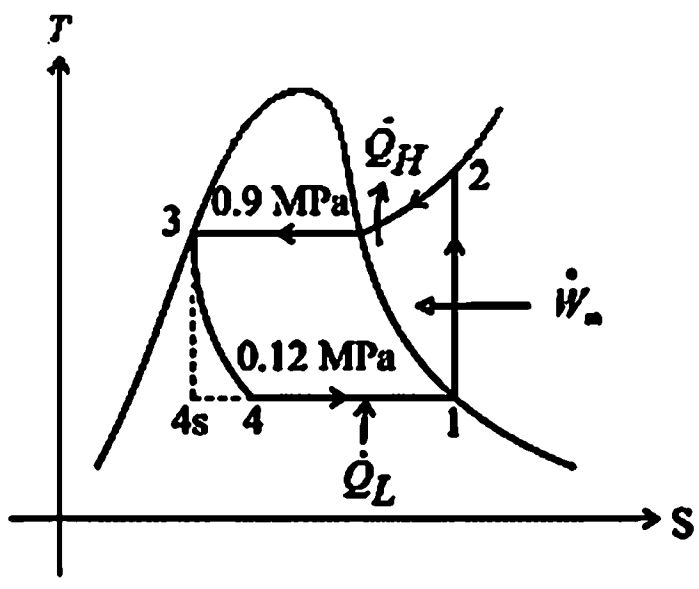
Mass flow rate of refrigerant, $\dot{m}_r = 0.05\ kg/s$

Pressure at condenser, $P_2 = 0.9\ MPa$

Pressure at evaporator, $P_1 = 0.12\ MPa$

Step 2 of 9

T-S diagram of vapor-compression cycle



Step 3 of 9

From saturated refrigerant- 134a pressure table

At $P_1 = 0.12\ MPa$ and saturated vapor,

$h_1 = h_{g@120\ kPa} = 236.97\ kJ/kg$

$s_1 = s_{g@120\ kPa} = 0.94779\ kJ/kg.K$

Step 4 of 9

At $P_2 = 0.9\ MPa$, $s_2 = s_1 = 0.94779\ kJ/kg.K$

$$h_2 = 274.17 + \frac{0.94779 - 0.9327}{0.966 - 0.9327} \times (284.77 - 274.17)$$

$h_2 = 278.97\ kJ/kg$

Step 5 of 9

At $P_2 = 0.9\ MPa$, Saturated liquid,

$h_3 = h_{f@0.9\ MPa} = 101.61\ kJ/kg$

For throttling, $h_4 = h_3 = 101.61\ kJ/kg$

Step 6 of 9

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 7 of 9

(a) Rate of heat removal from the refrigerated space, $\dot{Q}_L = \dot{m} \times (h_1 - h_4)$

$\dot{Q}_L = 0.05 \times (236.97 - 101.61)$

$\dot{Q}_L = 6.768\ kW$

Rate of heat removal from the refrigerated space, $\boxed{\dot{Q}_L = 6.768\ kW}$

Power required, $\dot{W}_{in} = \dot{m} \times (h_2 - h_1)$

$\dot{W}_{in} = 0.05 \times (278.97 - 236.97)$

$\dot{W}_{in} = 2.1 kW$

Power required, $\boxed{\dot{W}_{in} = 2.1\ kW}$

Step 8 of 9

(b) The rate of heat rejection to the environment, $\dot{Q}_H = \dot{W}_{in} + \dot{Q}_L$

$\dot{Q}_H = 2.1 + 6.768$

$\boxed{\dot{Q}_H = 8.868\ kW}$

Step 9 of 9

(c) COP of the refrigerator:

Coefficient of performance, $COP = \dfrac{\dot{Q}_L}{\dot{W}_{in}}$

$$COP = \frac{6.768}{2.1}$$

$\boxed{COP = 3.223}$

Question:

9.155P

If the throttling valve in Prob. 9–153 is replaced by an isentropic turbine, determine the percentage increase in the COP and in the rate of heat removal from the refrigerated space. Answers: 4.2 percent, 4.2 percent

Step 1 of 13

Given:

Working fluid: Refrigerant-134a

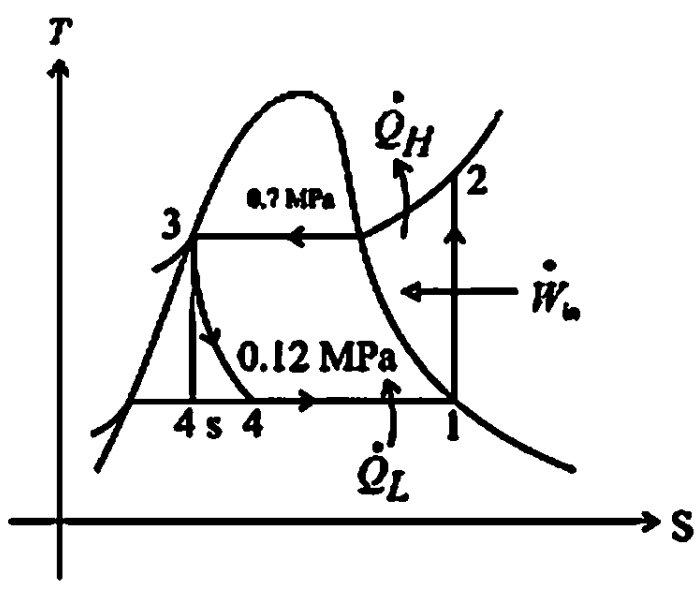
Mass flow rate of refrigerant, $\dot{m}_r = 0.05\ kg/s$

Pressure at condenser, $P_2 = 0.7\ MPa$

Pressure at evaporator, $P_1 = 0.12\ MPa$

Step 2 of 13

T-S diagram of vapor-compression refrigeration cycle



Step 3 of 13

From saturated refrigerant- 134a pressure table

At $P_1 = 0.12\ MPa$ and saturated vapor,

$h_1 = h_{g@120\ kPa} = 236.97\ kJ/kg$

$s_1 = s_{g@120\ kPa} = 0.94779\ kJ/kg.K$

Step 4 of 13

From superheated refrigerant- 134a table

$P_2 = 0.7\ MPa, s_2 = s_1 = 0.94779\ kJ/kg.K$

$$h_2 = 268.45 + \frac{0.94779 - 0.9313}{0.9641 - 0.9313} \times (278.57 - 168.45)$$

$h_2 = 273.5\ kJ/kg$

Step 5 of 13

From saturated refrigerant- 134a pressure table

$P_2 = 0.7\ MPa$, Saturated liquid,

$h_3 = h_{f@0.7\ MPa} = 88.82\ kJ/kg$

For throttling, $h_4 = h_3 = 88.82\ kJ/kg$

Step 6 of 13

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 7 of 13

The throttling valve is replaced by an isentropic turbine.

$(s_4)_{turbine} = s_3 = s_{f@0.7\ MPa} = 0.3323\ kJ/kg.K$

$s_{f@120kPa} = 0.09275\ kJ/kg.K$

$s_{fg@120kPa} = 0.85503\ kJ/kg.K$

$h_{f@120kPa} = 22.49\ kJ/kg$

$h_{fg@120kPa} = 214.48\ kJ/kg$

But $s_3 = s_{f@120kPa} + (x_4)_{turbine} \times s_{fg@120kPa}$

$$(x_4)_{turbine} = \frac{0.3323 - 0.09275}{0.85503}$$

$(x_4)_{turbine} = 0.280166$

Step 8 of 13

Enthalpy at turbine exit, $(h_4)_{turbine} = h_{f@120\ kPa} + (x_4)_{turbine} \times h_{fg@120kPa}$

$(h_4)_{turbine} = 22.49 + 0.280166 \times 214.48$

$(h_4)_{turbine} = 82.58\ kJ/kg$

Step 9 of 13

For throttling valve: Rate of heat removal from the refrigerated space,

$\dot{Q}_L = \dot{m} \times (h_1 - h_4)$

$\dot{Q}_L = 0.05 \times (236.97 - 88.82)$

$\dot{Q}_L = 7.4075\ kW$

For turbine: Rate of heat removal from the refrigerated space,

$(\dot{Q}_L)_{turbine} = \dot{m} \times (h_1 - (h_4)_{turbine})$

$(\dot{Q}_L)_{turbine} = 0.05 \times (236.97 - 82.58)$

$(\dot{Q}_L)_{turbine} = 7.7195\ kW$

Step 10 of 13

$$\text{Increase in } \dot{Q}_L = \frac{(\dot{Q}_L)_{turbine} - \dot{Q}_L}{\dot{Q}_L}$$

$$\%\dot{Q}_L = \frac{7.7195 - 7.41}{7.41}$$

$\%\dot{Q}_L = 4.18\%$

$\boxed{\%\dot{Q}_L = 4.2\%}$

Step 11 of 13

Power required, $\dot{W}_{in} = \dot{m} \times (h_2 - h_1)$

$\dot{W}_{in} = 0.05 \times (273.5 - 236.97)$

$\dot{W}_{in} = 1.8265$

Step 12 of 13

COP of the refrigerator if it is throttling valve:

Coefficient of performance, $COP = \frac{\dot{Q}_L}{\dot{W}_{in}}$

$$COP = \frac{7.41}{1.83}$$

$COP = 4.05$

COP of the refrigerator if it is turbine:

$$(COP)_{turbine} = \frac{(\dot{Q}_L)_{turbine}}{\dot{W}_{in}}$$

$$(COP)_{turbine} = \frac{7.7195}{1.83}$$

$(COP)_{turbine} = 4.2183$

Step 13 of 13

$$\text{Increase in COP, } = \frac{(COP)_{turbine} - COP}{COP}$$

$$\%COP = \frac{4.2183 - 4.05}{4.05}$$

$\boxed{\%COP = 4.2\%}$

Question: 9.156P

Refrigerant-134a enters the compressor of a refrigerator as superheated vapor at 0.20 MPa and −5°C at a rate of 007 kg/s, and it leaves at 1.2 MPa and 70°C. The refrigerant is cooled in the condenser to 44°C and 1.15 MPa, and it is throttled to 0.21 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components, show the cycle on a *T-s* diagram with respect to saturation lines, and determine *(a)* the rate of heat removal from the refrigerated space and the power input to the compressor, *(b)* the isentropic efficiency of the compressor, and *(c)* the COP of the refrigerator.

Step 1 of 10

Given:

Working fluid: R-134a

Inlet pressure of the compressor $P_1 = 0.20\,\text{MPa}$

Inlet temperature of the compressor $T_1 = -5^\circ\text{C}$

Mass flow rate of the refrigerant $m = 0.07\,\frac{\text{kg}}{s}$

Exit pressure of the compressor $P_2 = 1.2\,\text{MPa}$

Exit temperature of the compressor $T_2 = 70^\circ\text{C}$

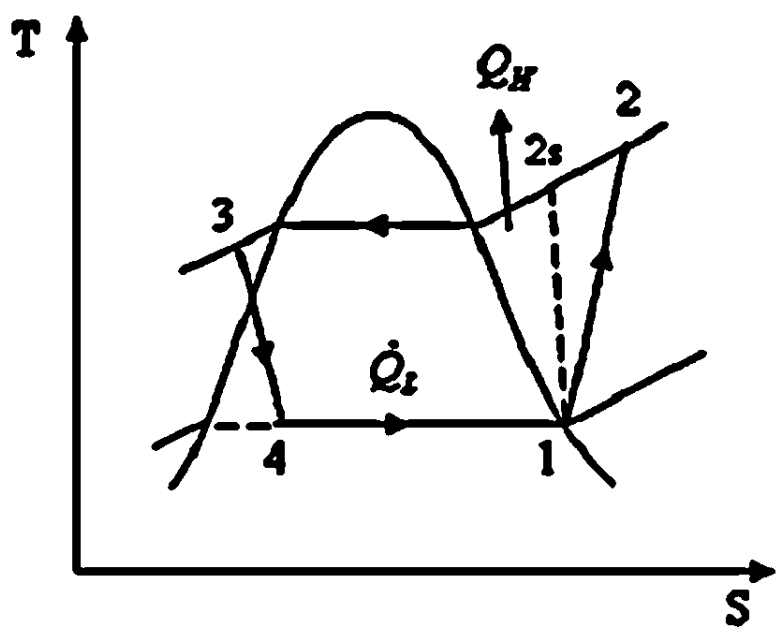
Exit pressure of the condenser $P_3 = 1.15\,\text{MPa}$

Exit temperature of the condenser $T_3 = 44^\circ\text{C}$

Throttled pressure $P_4 = 0.21\,\text{MPa}$

Step 2 of 10

T-S Diagram:



Step 3 of 10

State1:

Since refrigerant leaves the compressor is saturated liquid

From the Superheated refrigerant R-134a

At P_1=0.2MPa , $T_1 = -5^\circ\text{C}$

Using the interpolation

$$h_1 = 244.54 + \frac{(-10-(-5))(253.05-244.54)}{(-10-0)}$$

$$h_1 = 248.795\,\frac{\text{kJ}}{\text{kg}}$$

Entropy

At $T_1 = -5^\circ\text{C}$ and P_1=0.2MPa

Using the method of interpolation

$$s_1 = 0.9380 + \frac{(-10-(-5))(0.9698-0.9380)}{(-10-0)}$$

$$s_1 = 0.9539\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 4 of 10

State2:

Since refrigerant leaves the compressor in the form of saturated liquid

From the Superheated refrigerant R-134a

At $P_2 = 1.2\,\text{MPa}$ and $T_2 = 70^\circ\text{C}$

$$h_2 = 300.61\,\frac{\text{kJ}}{\text{kg}}$$

Step 5 of 10

State3:

Since refrigerant leaves the condenser in form of saturated liquid.

So, from the saturated refrigerant R-134a temperature table

$T_3 = 44^\circ\text{C}$ And P_3=1.15MPa

$$h_3 = h_f = 114.28\,\frac{\text{kJ}}{\text{kg}}$$

State4:

In throttling process, enthalpy remains constant.

$h_4 \cong h_3$ (Throttling)

$$h_3 = 114.28\,\frac{\text{kJ}}{\text{kg}}$$

Step 6 of 10

(a) The rate of heat removal from the refrigerated space and the power input to the compressor are determined from their definitions.

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

$$= \left(0.07\,\frac{\text{kg}}{s}\right)\left(\left(248.795\,\frac{\text{kJ}}{\text{kg}}\right)-\left(114.28\,\frac{\text{kJ}}{\text{kg}}\right)\right)$$

$$= 9.41\,\frac{\text{kJ}}{s}$$

$$\boxed{Q_L = 9.41\,\text{kW}}$$

Step 7 of 10

Work input is

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

$$= 0.07\,\frac{\text{kg}}{s}\left(\left(300.61\,\frac{\text{kJ}}{\text{kg}}\right)-\left(248.795\,\frac{\text{kJ}}{\text{kg}}\right)\right)$$

$$= 3.627\,\frac{\text{kJ}}{s}$$

$$\boxed{W_{in} = 3.627\,\text{kW}}$$

Step 8 of 10

(b) The isentropic efficiency of the compressor is determined from

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

Where the enthalpy at state $2s$ is

From the superheated refrigerant R-134a

$$P_{2s} = 1.2\,\text{MPa and } s_{2s} = s_2 = 0.9539\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$h_{2s} = = 278.27 + \frac{(0.9539-0.9297)(289.64-278.27)}{(0.9614-0.9267)}$$

$$h_{2s} = 286.199\,\frac{\text{kJ}}{\text{kg}}$$

Step 9 of 10

Isentropic efficiency of the compressor is

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$= \frac{(286.199-248.795)}{(300.61-248.795)}$$

$$= 0.721$$

$$\boxed{\eta_C = 72.1\%}$$

Step 10 of 10

(c) The COP of the refrigerator is determined from

$$(COP)_R = \frac{Q_L}{\dot{W}_{in}}$$

$$= \frac{9.41}{3.627}$$

$$= 2.59$$

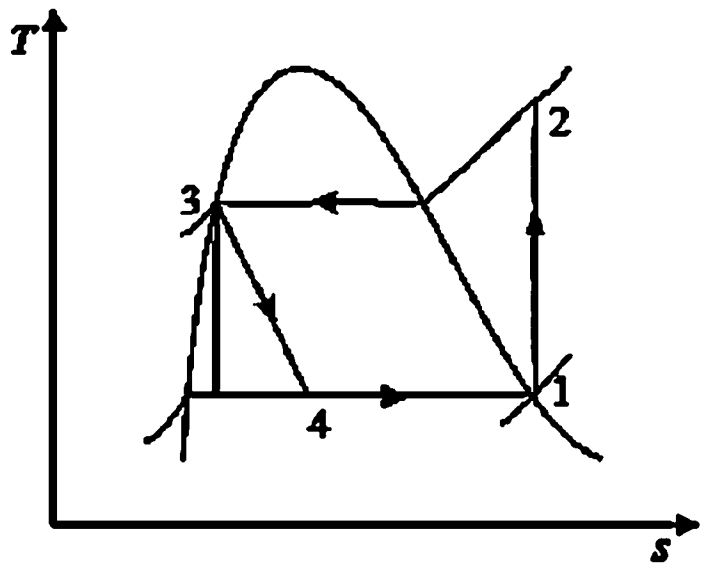
$$\boxed{\text{COP} = 2.59}$$

Question: 9.157P

A refrigerator uses refrigerant-134a as its working fluid and operates on the ideal vapor-compression refrigeration cycle. The refrigerant evaporates at 5°F and condenses at 180 psia. This unit serves a 45,000 Btu/h cooling load. Determine the mass flow rate of the refrigerant and the power that this unit will require.

Step 1 of 6

Draw the different processes on T-s diagram as shown below:



Process 1-2 is isentropic compression
Process 2-3 is condensation process
Process 3-4 is throttling expansion
Process 4-1 is evaporation

Step 2 of 6

The refrigerant enters into the compressor is saturated Vapor, and refrigerant leaves the condenser is saturated liquid.

Obtain the properties of refrigerant from "saturated refrigerant R-134a temperature tables" at $T_1 = 5°\text{F}$

Specific enthalpy of saturated vapor $h_1 = h_g = 103.82 \frac{\text{Btu}}{\text{lbm}}$

Specific entropy of saturated vapor $s_1 = s_g = 0.22485 \frac{\text{Btu}}{\text{lbm} \cdot \text{R}}$

Step 3 of 6

Process 1-2 is isentropic compression process, entropy remains constant.

$$s_1 = s_2$$

Obtain the properties at state 2 from "superheated refrigerant R-134a pressure table" at $P_2 = 180\text{psia}$ and $s_2 = 0.22485 \frac{\text{Btu}}{\text{lbm} \cdot \text{R}}$

$$h_2 = 118.63 + (124.17 - 118.63)\left(\frac{0.22485 - 0.21910}{0.22850 - 0.21910}\right)$$

$$= 122.01 \frac{\text{Btu}}{\text{lbm}}$$

Step 4 of 6

Obtain the properties at state 3 "from saturated refrigerant R-134a pressure table" at condenser pressure $P_3 = 180\text{psia}$

Specific enthalpy of saturated liquid $h_3 = h_f = 51.497 \frac{\text{Btu}}{\text{lbm}}$

In process 3-4 enthalpy remains constant. Since it is a throttling process

$$h_3 = h_4$$

$$h_4 = 51.497 \frac{\text{Btu}}{\text{lbm}}$$

Step 5 of 6

Find the mass flow rate of refrigerant by using the equation,

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

Here, $\dot{Q}_L$ is cooling load capacity

$$\dot{m} = \frac{\dot{Q}_L}{(h_1 - h_4)}$$

Substitute $45{,}000\,\text{Btu/h}$ for $\dot{Q}_L$

$$\dot{m} = \frac{45000}{(103.82 - 51.497)}$$

$$= 860.042\,\text{lbm/h}$$

Therefore, mass flow rate of the refrigerant is $\boxed{860.042 \frac{\text{lbm}}{\text{h}}}$

Step 6 of 6

Calculate the power input to the compressor by using the equation,

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

$$= \left(860.0424 \frac{\text{lbm}}{\text{h}}\right)\left(\left(122.01 \frac{\text{Btu}}{\text{lbm}}\right) - \left(103.82 \frac{\text{Btu}}{\text{lbm}}\right)\right)\left(\frac{1\text{kW}}{3412.14 \frac{\text{Btu}}{\text{h}}}\right)$$

$$= 4.58\text{kW}$$

Therefore, power required to the compressor is $\boxed{4.58\text{kW}}$

Question: 9.158P


Using EES for other) software, repeat *mM* Prob. 9-157E if ammonia is used in place of refrigerant-134a.

Step 1 of 10

Given:
Working fluid: ammonia
Operating temperature of the evaporator $T_1=5°F$
Operating pressure of the condenser $P_2=180\,\text{psia}$
Cooling load $Q_L=45000\frac{\text{Btu}}{\text{h}}$

Step 2 of 10

The problem is solved using EES, and the solution is given below.

```
"Given"
x[1]=1
T[1]=5 [F]
x[3]=0
P[3]=180 [psia]
Q_dot_L=45000 [Btu/h]
```

Step 3 of 10

```
"Analysis"
Fluid$='ammonia'
"compressor"
h[1]=enthalpy(Fluid$, T=T[1], x=x[1])
s[1]=entropy(Fluid$, T=T[1], x=x[1])
s[2]=s[1]
P[2]=P[3]
h[2]=enthalpy(Fluid$, P=P[2], s=s[2])
```

Step 4 of 10

```
"expansion valve"
h[3]=enthalpy(Fluid$, P=P[3], x=x[3])
h[4]=h[3]
m_dot_R=Q_dot_L/(h[1]-h[4])
W_dot_in=m_dot_R×(h[2]-h[1])×Convert(Btu/h, kW)
```

Step 5 of 10

Solution for ammonia

```
Fluid$='ammonia'
m_dot_R=33 [Btu/h]
Q_dot_L=45000 [Btu/h]
W_dot_in=-1.196 [kW]
```

Step 6 of 10

Given:
Working fluid: R-134a
Operating temperature of the evaporator $T_1=5°F$
Operating pressure of the condenser $P_2=180\,\text{psia}$
Cooling load $Q_L=45000\frac{\text{Btu}}{\text{h}}$

Step 7 of 10

The problem is solved using EES, and the solution is given below.

```
"Given"
x[1]=1
T[1]=5 [F]
x[3]=0
P[3]=180 [psia]
Q_dot_L=45000 [Btu/h]
```

Step 8 of 10

```
"Analysis"
Fluid$='ammonia'
"compressor"
h[1]=enthalpy(Fluid$, T=T[1], x=x[1])
s[1]=entropy(Fluid$, T=T[1], x=x[1])
s[2]=s[1]
P[2]=P[3]
h[2]=enthalpy(Fluid$, P=P[2], s=s[2])
```

Step 9 of 10

```
"expansion valve"
h[3]=enthalpy(Fluid$, P=P[3], x=x[3])
h[4]=h[3]
m_dot_R=Q_dot_L/(h[1]-h[4])
W_dot_in=m_dot_R×(h[2]-h[1])×Convert(Btu/h, kW)
```

Step 10 of 10

Solution for R-134a

```
Fluid$='R134a'
m_dot_R=206 [Btu/h]
Q_dot_L=45000 [Btu/h]
W_dot_in=-0.8038 [kW]
```

Question: 9.159P pg 1

A commercial refrigerator with refrigerant-134a as the working fluid is used to keep the refrigerated space at −30°C by rejecting its waste heat to cooling water that enters the condenser at 18°C at a rate of 0.25 kg/s and leaves at 26°C. The refrigerant enters the condenser at 1.2 MPa and 65°C and leaves at 42°C. The inlet state of the compressor is 60 kPa and −34°C and the compressor is estimated to gain a net heat of 450 W from the surroundings. Determine *(a)* the quality of the refrigerant at the evaporator inlet, *(b)* the refrigeration load, (c) the COP of the refrigerator, and *(d)* the theoretical maximum refrigeration load for the same power input to the compressor.

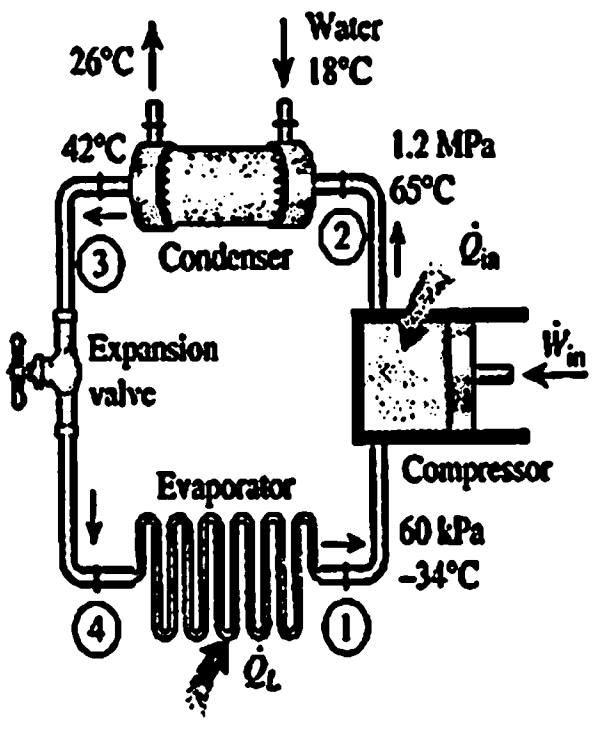


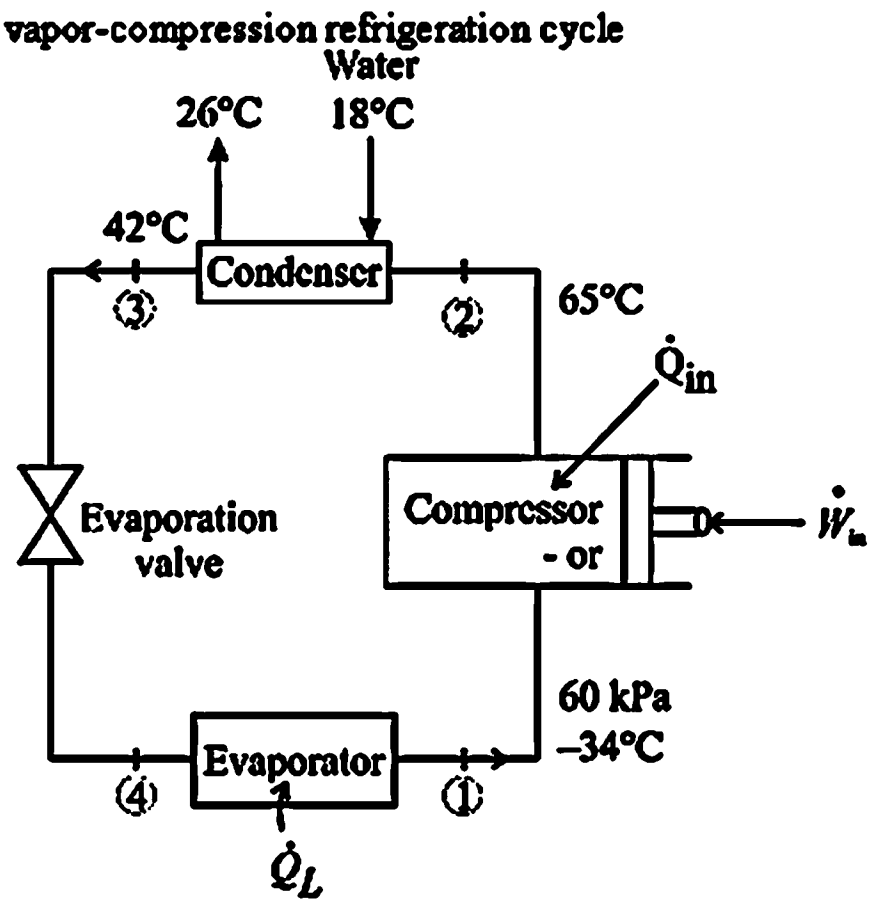
FIGURE P9–159

Step 1 of 12

Given:
Working fluid: Refrigerant-134a
Mass flow rate of water, $\dot{m}_w = 0.25\ kg/s$
The compressor is estimated to gain a net heat of 450 W from the surroundings, $\dot{Q}_{in} = 450\ W$

Step 2 of 12

Block diagram of vapor-compression refrigeration cycle



Step 3 of 12

From Superheated refrigerant- 134a table
At $P_1 = 60kPa$, $T_1 = -34°C$

$$h_1 = 227.79 + \frac{(-34-(-36.95))}{(-20-(-34))}(240.76-227.79)$$

$$h_1 = 230.5 kJ/kg$$

Step 4 of 12

From Superheated refrigerant- 134a table
At $P_2 = 1200kPa$, $T_2 = 65°C$

$$h_2 = 289.64 + \frac{(65-60)}{(70-60)}(300.61-289.64)$$

$$h_2 = 295.12\ kJ/kg$$

Step 5 of 12

At $P_3 = 1200kPa$, $T_3 = 42°C$

$h_3 \cong h_{f@42°C} = 111.26$ kJ/kg (From saturated refrigerant – 134a temperature table)

Step 6 of 12

Enthalpy of water before entering the condenser,
$h_{w\,before} = h_{f@18°C} = 75.47$ kJ/kg (From saturated water-temperature table)
Enthalpy of water after leaving the condenser,
$h_{w\,after} = h_{f@26°C} = 108.94$ kJ/kg (From saturated water-temperature table)

Step 7 of 12

Assumptions:
1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 8 of 12

(a) The quality of the refrigerant at the evaporator inlet:
From the diagram,
$h_4 = h_3 = 111.26$ kJ/kg
From saturated refrigerant – 134a pressure table
$P_4 = 60kPa$, $h_4 = 111.26$ kJ/kg

$$h_4 = h_{f4} + x_4 h_{fg4}$$

$$111.26 = 3.841 + x_4(223.95)$$

$h_4 = h_3 = 111.26$ kJ/kg

9.159P pg 2

From saturated refrigerant – 134a pressure table

$P_4 = 60$kPa, $h_4 = 111.26$ kJ/kg

$h_4 = h_{f4} + x_4 h_{fg4}$

$111.26 = 3.841 + x_4 (223.95)$

$$x_4 = \frac{111.26 - 3.841}{223.95}$$

$\boxed{x_4 = 0.4796}$

Step 9 of 12

(b) The refrigeration load:

Mass flow rate of the refrigerant, $\dot{m}_r = \dfrac{h_{w\text{-}after} - h_{w\text{-}before}}{h_2 - h_3} \times \dot{m}_w$

$$\dot{m}_r = \frac{108.94 - 75.47}{295.12 - 111.26} \times 0.25$$

$\dot{m}_r = 0.04551$ kg/s

Heat rejected to atmosphere, $\dot{Q}_H = \dot{m}_r \times (h_2 - h_3)$

$\dot{Q}_H = 0.04551 \times (295.12 - 111.26)$

$\dot{Q}_H = 8.3674$ kW

Step 10 of 12

Compressor power input, $\dot{W}_{in} = \dot{m}_r \times (h_2 - h_1) - \dot{Q}_{in}$

$\dot{W}_{in} = 0.04551 \times (295.12 - 230.5) - 0.45$

$\dot{W}_{in} = 2.9408 - 0.45$

$\dot{W}_{in} = 2.4908$ kW

Refrigeration load, $\dot{Q}_L = \dot{Q}_H - \dot{W}_{in}$

$\dot{Q}_L = 8.3674 - 2.4908$

$\boxed{\dot{Q}_L = 5.8766 \text{ kW}}$

Step 11 of 12

(c) COP of the refrigerator:

Coefficient of performance, $COP = \dfrac{\dot{Q}_L}{\dot{W}_{in}}$

$$COP = \frac{5.8766}{2.4908}$$

$\boxed{COP = 2.359}$

Step 12 of 12

(d) Theoretical maximum refrigeration load:

Maximum COP, $COP_{max} = \dfrac{1}{\dfrac{T_H}{T_L} - 1}$

$$COP_{max} = \frac{1}{\frac{291}{243} - 1}$$

$COP_{max} = 5.0625$

Theoretical maximum refrigeration load, $(\dot{Q}_L)_{max} = COP_{max} \times \dot{W}_{in}$

$(\dot{Q}_L)_{max} = 5.0625 \times 2.4908$

$\boxed{(\dot{Q}_L)_{max} = 12.609 \text{ kW}}$

Question: 9.160P pg 1

Refrigerant-134a enters the compressor of a refrigerator at 100 kPa and −20°C at a rate of 0.5 m3/min and leaves at 0.8 MPa. The isentropic efficiency of the compressor is 78 percent. The refrigerant enters the throttling valve at 0.75 MPa and 26°C and leaves the evaporator as saturated vapor at −26°C. Show the cycle on a T-s diagram with respect to saturation lines, and determine *(a)* the power input to the compressor, *(b)* the rate of heat removal from the refrigerated space, and (c) the pressure drop and rate of heat gain in the line between the evaporator and the compressor.

Step 1 of 11

Given:

Working fluid: R-134a

Inlet pressure of the compressor $P_1 = 100\text{kPa}$

Inlet Temperature of the compressor $T_1 = -20^\circ\text{C}$

Volume flow rate in compressor $V_1 = 0.5\dfrac{\text{m}^3}{\text{min}}$

Exit pressure of the compressor $P_2 = 0.8\text{MPa}$

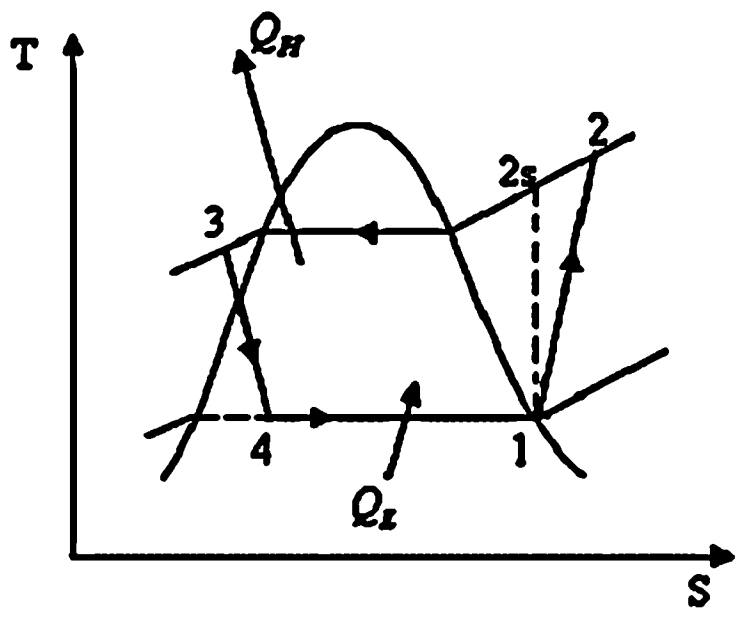
Inlet pressure of the throttle $P_3 = 0.75\text{MPa}$

Inlet temperature of the throttle $T_3 = 26^\circ\text{C}$

Inlet temperature of the evaporator $T_5 = -26^\circ\text{C}$

Step 2 of 11

T-S Diagram:



Step 3 of 11

(a)

State1:

Refrigerant enters into the compressor is superheated state. So

From the superheated refrigerant R-134a

At $P_1 = 100\text{kPa}$ and $T_1 = -20^\circ\text{C}$

$$v_1 = 0.19841\frac{\text{m}^3}{\text{kg}}$$

$$h_1 = 239.50\frac{\text{kJ}}{\text{kg}}$$

$$s_1 = 0.9721\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 4 of 11

State2:

Refrigerant leaves the compressor is superheated sate.

From the superheated refrigerant R-134a

At $P_2 = 0.8MPa$

And $s_{2s} = s_1 = 0.9721\dfrac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Using method of interpolation

$$h_{2s} = 276.45 + \frac{(0.9721-0.9480)(286.69-276.45)}{(0.9802-0.9480)}$$

$$h_{2s} = 284.11\frac{\text{kJ}}{\text{kg}}$$

Step 5 of 11

State3:

Refrigerant leaves the condenser is saturated liquid

From the saturated refrigerant R-134a Temperature table

At $P_3 = 0.75\text{MPa}, T_3 = 26^\circ\text{C}$

$$h_3 = h_f = 87.83\frac{\text{kJ}}{\text{kg}}$$

Throttling:

Since inn throttling enthalpy remains constant.

$$h_3 = h_4 = 87.83\frac{\text{kJ}}{\text{kg}}$$

Step 6 of 11

State5:

Leaving temperature of the evaporator is saturated vapor.

From the saturated refrigerant R-134a Temperature table

$T_5 = -26^\circ\text{C}$

$P_5 = P_{sat} = 101.73\text{kPa}$

$$h_5 = h_g = 234.68\frac{\text{kJ}}{\text{kg}}$$

Step 7 of 11

Then the mass flow rate of the refrigerant is

$$\dot{m} = \frac{V_1}{v_1}$$

$$= \frac{\left(\dfrac{0.5\,\text{m}^3}{60\,\text{sec}}\right)}{\left(0.19841\dfrac{\text{m}^3}{\text{kg}}\right)}$$

$$= 0.042\frac{\text{kg}}{s}$$

$$= \frac{\left(0.042\frac{\text{kg}}{\text{s}}\right)}{\left(0.19841\frac{\text{m}^3}{\text{kg}}\right)}$$

$$= 0.042\frac{\text{kg}}{\text{s}}$$

Step 8 of 11

The power input to the compressor is

$$\dot{W}_{in} = \frac{\dot{m}(h_{2s} - h_1)}{\eta_c}$$

$$= \frac{\left(0.042\frac{\text{kg}}{\text{s}}\right)\left(\left(284.11\frac{\text{kJ}}{\text{kg}}\right) - \left(239.50\frac{\text{kJ}}{\text{kg}}\right)\right)}{0.78}$$

$$= 2.40\frac{\text{kJ}}{\text{s}}$$

$$\boxed{\dot{W}_{in} = 2.40\text{kW}}$$

Step 9 of 11

(b) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4)$$

$$= \left(0.042\frac{\text{kg}}{\text{s}}\right)\left(\left(234.68\frac{\text{kJ}}{\text{kg}}\right) - \left(87.83\frac{\text{kJ}}{\text{kg}}\right)\right)$$

$$= 6.167\frac{\text{kJ}}{\text{s}}$$

$$\boxed{\dot{Q}_L = 6.167\text{kW}}$$

Step 10 of 11

(c) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1$$

$$= 101.73\text{kPa} - 100\text{kPa}$$

$$= 1.73\text{kPa}$$

$$\boxed{\Delta P = 1.73\text{kPa}}$$

Step 11 of 11

Rate of heat in between evaporator and compressor is

$$\dot{Q}_{gain} = \dot{m}(h_1 - h_5)$$

$$= \left(0.042\frac{\text{kg}}{\text{s}}\right)\left(\left(239.50\frac{\text{kJ}}{\text{kg}}\right) - \left(234.68\frac{\text{kJ}}{\text{kg}}\right)\right)$$

$$= 0.20244\frac{\text{kJ}}{\text{s}}$$

$$\boxed{\dot{Q}_{gain} = 0.20244\text{kW}}$$

Question:

Reconsider Prob. 9–160. Using EES (or other) software, investigate the effects of varying the compressor isentropic efficiency over the range 60 to 100 percent and the compressor inlet volume flow rate from 0.1 to 1.0 m3/min on the power input and the rate of refrigeration. Plot the rate of refrigeration and the power input to the compressor as functions of compressor efficiency for compressor inlet volume flow rates of 0.1, 0.5, and 1.0 mVmin, and discuss the results.

Step 1 of 22

Given:

Working fluid: R-134a

Inlet pressure of the compressor $P_1 = 100\text{kPa}$

Inlet Temperature of the compressor $T_1 = -20^\circ\text{C}$

Volume flow rate in compressor varies from $V_1 = 0.1\frac{\text{m}^3}{\text{min}}$ to $1\frac{\text{m}^3}{\text{min}}$

Exit pressure of the compressor $P_2 = 0.8\text{MPa}$

Inlet pressure of the throttle $P_3 = 0.75\text{MPa}$

Inlet temperature of the throttle $T_3 = 26^\circ\text{C}$

Inlet temperature of the evaporator $T_5 = -26^\circ\text{C}$

Step 2 of 22

```
P[1]=100 [kPa]
T[1]=-20 [C]
V_dot=0.5 [m^3/min]
P[2]=800 [kPa]
"Eta_C=0.78"
P[3]=750 [kPa]
T[3]=26  [C]
T[5]=-26 [C]
x[5]=1
```

Step 3 of 22

```
Analysis
Fluid$='R134a'
"Compressor"
h[1]=enthalpy(Fluid$, P=P[1], T=T[1])
s[1]=entropy(Fluid$, P=P[1], T=T[1])
v[1]=volume(Fluid$, P=P[1], T=T[1])
s_s[2]=s[1]
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])
```

Step 4 of 22

```
"expansion valve"
x[3]=0  "assumed saturated liquid"
h[3]=enthalpy(Fluid$, T=T[3], x=x[3])
h[4]=h[3]
```

Step 5 of 22

```
"evaporator exit"
h[5]=enthalpy(Fluid$, T=T[5], x=x[5])
P[5]=pressure(Fluid$, T=T[5], x=x[5])
```

Step 6 of 22

```
"cycle"
m_dot=V_dot/v[1]×Convert(kg/min, kg/s)
W_dot_in=m_dot×(h_s[2]-h[1])/η_C
Q_dot_L=m_dot×(h[5]-h[4])
DELTAP=P[5]-P[1]
Q_dot_gain=m_dot×(h[1]-h[5])
```

Step 7 of 22

Run the parameter table from the efficiency ranges from the 0.6 to 1%

Then select the η_c, $\dot{W}_{in}$ and $\dot{Q}_L$ then click ok

Step 8 of 22

Solution table:

η_{1_10}	η_c	$\dot{Q}_L$ [kW]	$\dot{W}_{in}$ [kW]
Run 1	0.6	6.169	3.12
Run 2	0.6444	6.169	2.905
Run 3	0.6889	6.169	2.718
Run 4	0.7333	6.169	2.553
Run 5	0.7778	6.169	2.407
Run 6	0.8222	6.169	2.277
Run 7	0.8667	6.169	2.16
Run 8	0.9111	6.169	2.055
Run 9	0.9556	6.169	1.959
Run 10	1	6.169	1.872

Step 9 of 22

Given:

$V_1 = 0.1\frac{\text{m}^3}{\text{min}}$

```
"Given"
P[1]=100 [kPa]
T[1]=-20 [C]
V_dot=0.1 [m^3/min]
P[2]=800 [kPa]
"Eta_C=0.78"
P[3]=750 [kPa]
T[3]=26  [C]
T[5]=-26 [C]
x[5]=1
```

Step 10 of 22

T[5]=-26 [C]
x[5]=1

Step 10 of 22

"Analysis"
Fluid$='R134a'
"Compressor"
h[1]=enthalpy(Fluid$, P=P[1], T=T[1])
s[1]=entropy(Fluid$, P=P[1], T=T[1])
v[1]=volume(Fluid$, P=P[1], T=T[1])
s_s[2]=s[1]
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])

Step 11 of 22

"expansion valve"
x[3]=0 "assumed saturated liquid"
h[3]=enthalpy(Fluid$, T=T[3], x=x[3])
h[4]=h[3]

Step 12 of 22

"evaporator exit"
h[5]=enthalpy(Fluid$, T=T[5], x=x[5])
P[5]=pressure(Fluid$, T=T[5], x=x[5])

Step 13 of 22

"cycle"
m_dot=V_dot/v[1]×Convert(kg/min, kg/s)
W_dot_in=m_dot×(h_s[2]-h[1])/η_C
Q_dot_L=m_dot×(h[5]-h[4])
DELTAP=P[5]-P[1]
Q_dot_gain=m_dot×(h[1]-h[5])

Step 14 of 22

Solution table:

▷ 1..10	1 η_C	2 $\dot{Q}_L$ [kW]	3 $\dot{W}_{in}$ [kW]
Run 1	0.6	1.234	0.624
Run 2	0.6444	1.234	0.581
Run 3	0.6889	1.234	0.5435
Run 4	0.7333	1.234	0.5106
Run 5	0.7778	1.234	0.4814
Run 6	0.8222	1.234	0.4554
Run 7	0.8667	1.234	0.432
Run 8	0.9111	1.234	0.4109
Run 9	0.9556	1.234	0.3918
Run 10	1	1.234	0.3744

Step 15 of 22

When the volume equal to $V_1 = 1\frac{m^3}{min}$

"Given"
P[1]=100 [kPa]
T[1]=-20 [C]
V_dot=01 [m^3/min]
P[2]=800 [kPa]
"Eta_C=0.78"
P[3]=750 [kPa]
T[3]=26 [C]
T[5]=-26 [C]
x[5]=1

Step 16 of 22

"Analysis"
Fluid$='R134a'
"Compressor"
h[1]=enthalpy(Fluid$, P=P[1], T=T[1])
s[1]=entropy(Fluid$, P=P[1], T=T[1])
v[1]=volume(Fluid$, P=P[1], T=T[1])
s_s[2]=s[1]
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])

Step 17 of 22

"expansion valve"
x[3]=0 "assumed saturated liquid"
h[3]=enthalpy(Fluid$, T=T[3], x=x[3])
h[4]=h[3]

Step 18 of 22

"evaporator exit"
h[5]=enthalpy(Fluid$, T=T[5], x=x[5])
P[5]=pressure(Fluid$, T=T[5], x=x[5])

Step 19 of 22

"cycle"
m_dot=V_dot/v[1]×Convert(kg/min, kg/s)
W_dot_in=m_dot×(h_s[2]-h[1])/η_C
Q_dot_L=m_dot×(h[5]-h[4])
DELTAP=P[5]-P[1]
Q_dot_gain=m_dot×(h[1]-h[5])

Step 20 of 22

Solution table :

▷ 1..10	1 η_C	2 $\dot{Q}_L$ [kW]	3 $\dot{W}_{in}$ [kW]

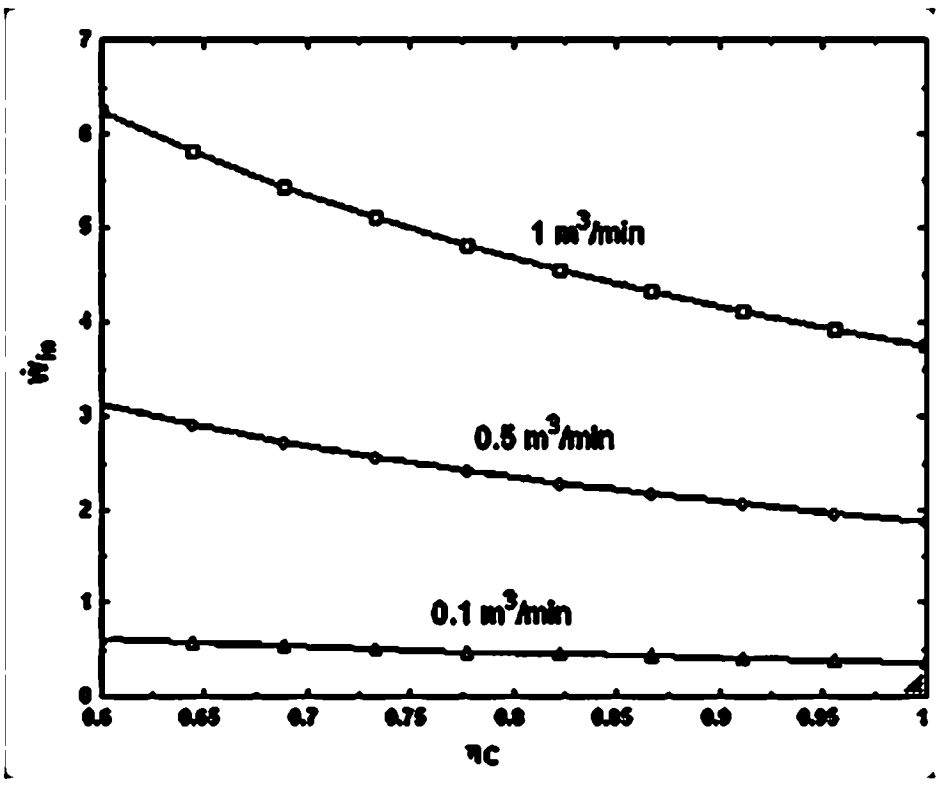
Step 20 of 22

Solution table :

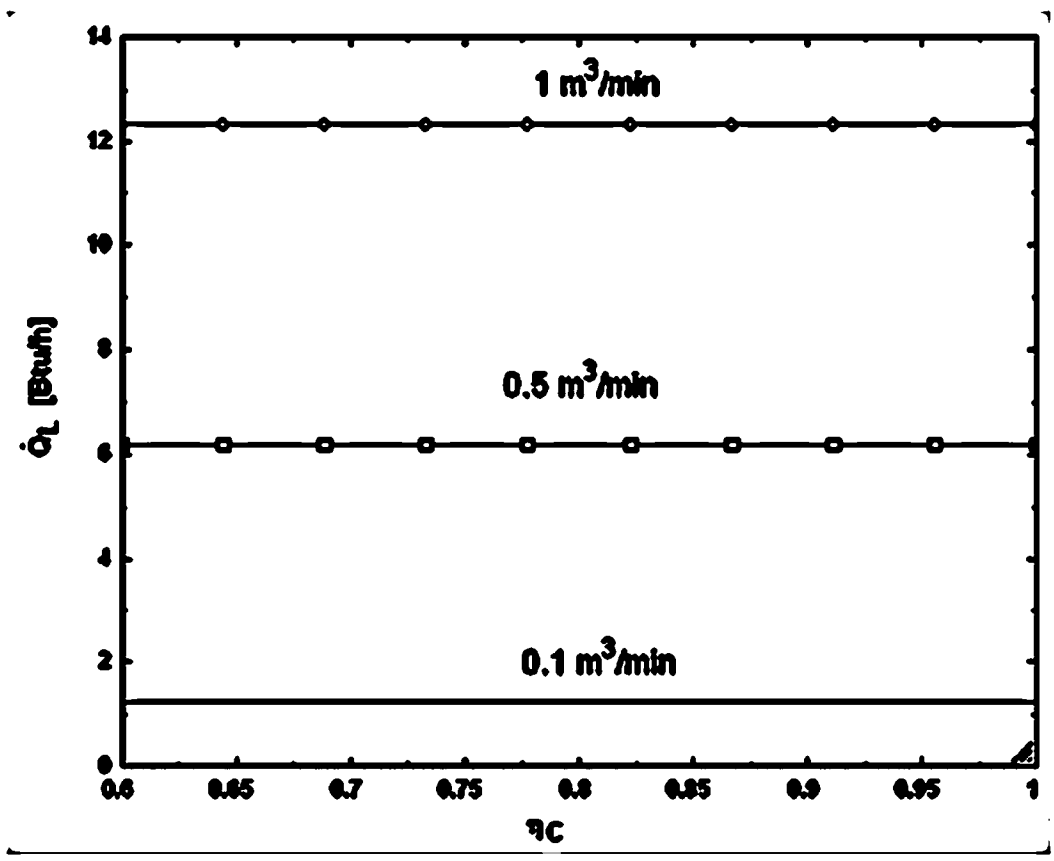
$\triangleright$ 1..10	1 η_C	2 $\dot{Q}_L$ [kW]	3 $\dot{W}_{in}$ [kW]
Run 1	0.6	12.34	6.24
Run 2	0.6444	12.34	5.81
Run 3	0.6889	12.34	5.435
Run 4	0.7333	12.34	5.106
Run 5	0.7778	12.34	4.814
Run 6	0.8222	12.34	4.554
Run 7	0.8667	12.34	4.32
Run 8	0.9111	12.34	4.109
Run 9	0.9556	12.34	3.918
Run 10	1	12.34	3.744

Step 21 of 22

Combining of three plots is



Step 22 of 22



Question: 9.162P

A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle except for the compression process. The refrigerant enters the evaporator at 120 kPa with a quality of 30 percent and leaves the compressor at 60°C. If the compressor consumes 450 W of power, determine (a) the mass flow rate of the refrigerant, *(b)* the condenser pressure, and (c) the COP of the refrigerator.

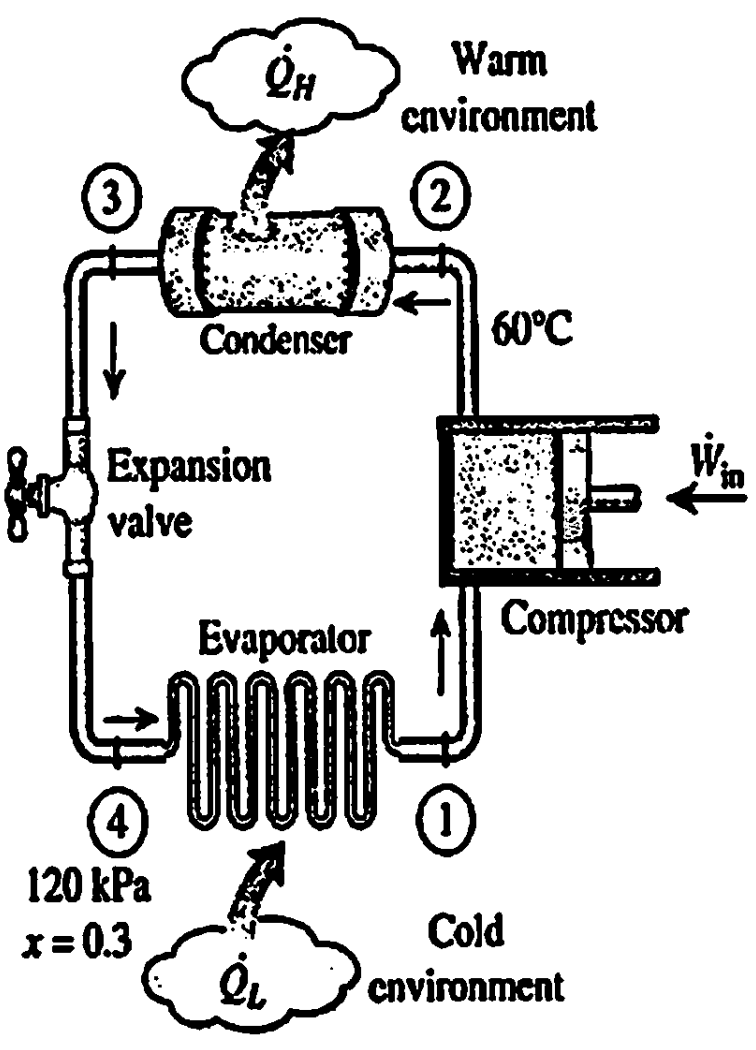


FIGURE P9–162

Step 1 of 11

Given:

Working fluid: Refrigerant-134a

Pressure of refrigerant at entrance of evaporator, $P_4 = 120$ kPa

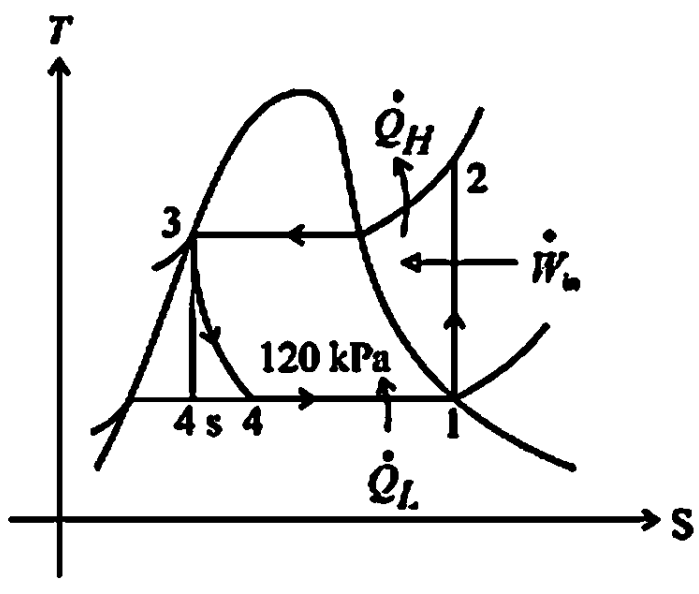
Quality $x_4 = 0.3$

It leaves the compressor at $T_2 = 60°C$

Power of compressor, $\dot{W}_{in} = 450$ W

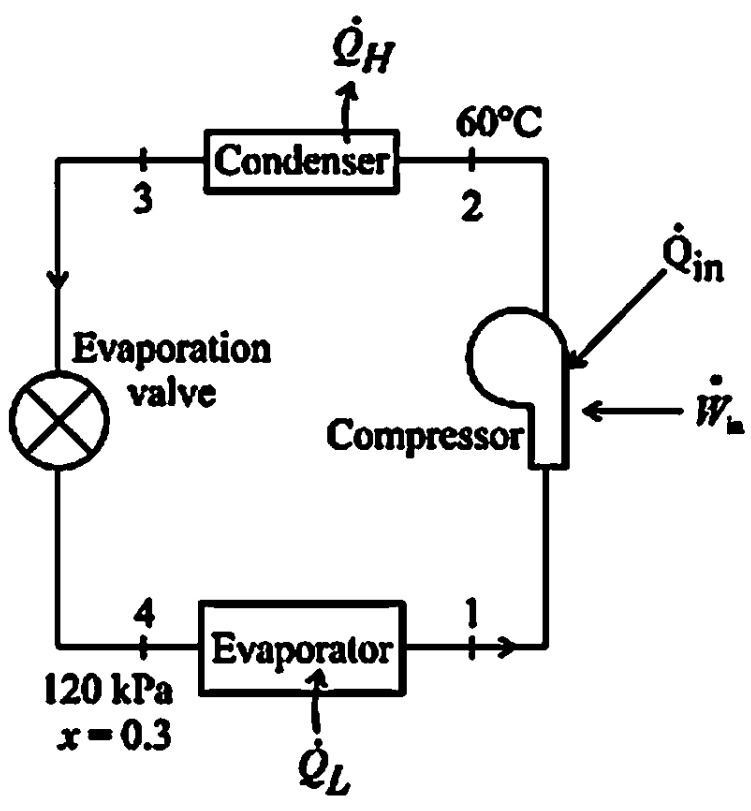
Step 2 of 11

T-S diagram of vapor-compression refrigeration cycle



Step 3 of 11

Block diagram of vapor-compression refrigeration cycle



Step 4 of 11

From Saturated refrigerant- 134a pressure table

At $P_4 = 120$ kPa and $x_4 = 0.3$

$h_4 = 22.49 + (0.3)(214.48)$

$h_3 = h_4 = 86.83$ kJ/kg

Step 5 of 11

From saturated refrigerant-134a pressure table

At $h_3 = 86.83$ kJ/kg, saturated liquid

$$P_3 = 650 + \frac{(86.83 - 85.26)}{(88.82 - 85.26)}(700 - 650)$$

$P_2 = P_3 = 672$ kPa

Step 6 of 11

From superheated refrigerant-134a table

At $P_2 = 672$ kPa, $T_2 = 60°C$

$$h_2 = 299.98 - \frac{(672 - 600)}{(700 - 600)}(299.98 - 298.42)$$

$h_2 = 298.85$ kJ/kg

Step 7 of 11

From saturated refrigerant-134a pressure table

At $P_1 = 120$ kPa, saturated vapor

$h_1 = 236.97$ kJ/kg

Step 8 of 11

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 9 of 11

(a) Mass flow rate of the refrigerant:

Mass flow rate of refrigerant, $\dot{m}_r = \dfrac{\dot{W}_{in}}{h_2 - h_1}$

$$\dot{m}_r = \frac{0.45}{298.85 - 236.97}$$

$\boxed{\dot{m}_r = 7.27 \times 10^{-3} \text{ kg/s}}$

Step 10 of 11

(b) The condenser pressure

Condenser pressure $\boxed{P_3 = 672 \text{ kPa}}$

Step 11 of 11

(c) COP of the refrigerator

Cooling load, $\dot{Q}_L = \dot{m}_r \times (h_1 - h_4)$

$\dot{Q}_L = 0.00727 \times (236.97 - 86.83)$

$\dot{Q}_L = 1.0915$ kW

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$\text{COP} = \frac{1.0915}{0.45}$$

$\boxed{\text{COP} = 2.425}$

Question: 9.163P

The manufacturer of an air conditioner claims a seasonal energy efficiency ratio (SEER) of 16 (Btu/h)/W for one of its units. This unit operates on the normal vapor compression refrigeration cycle and uses refrigerant-22 as the working fluid. This SEER is for the operating conditions when the evaporator saturation temperature is -5°C and the condenser saturation temperature is 45°C. Selected data for refrigerant-22 are provided in the table below.

T, °C	P_{sat}, kPa	h_f, kJ/kg	h_g, kJ/kg	s_g, kJ/kg·K
−5	421.2	38.76	248.1	0.9344
45	1728	101	261.9	0.8682

(*a*) Sketch the hardware and the *T*-*s* diagram for this air conditioner.

(*b*) Determine the heat absorbed by the refrigerant in the evaporator per unit mass of refrigeranl-22, in kJ/kg.

(*c*) Determine the work input to the compressor and the heat rejected in the condenser per unit mass of refrigerant-22, in kJ/kg.

Step 1 of 7

Given:

Working fluid: $R-22$

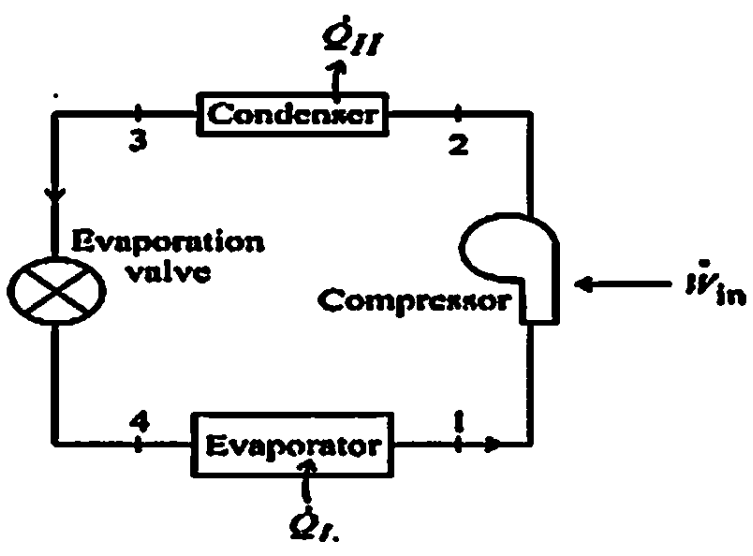
Seasonal energy efficiency $SEER = 16\dfrac{\text{Btu}}{\text{h}}$

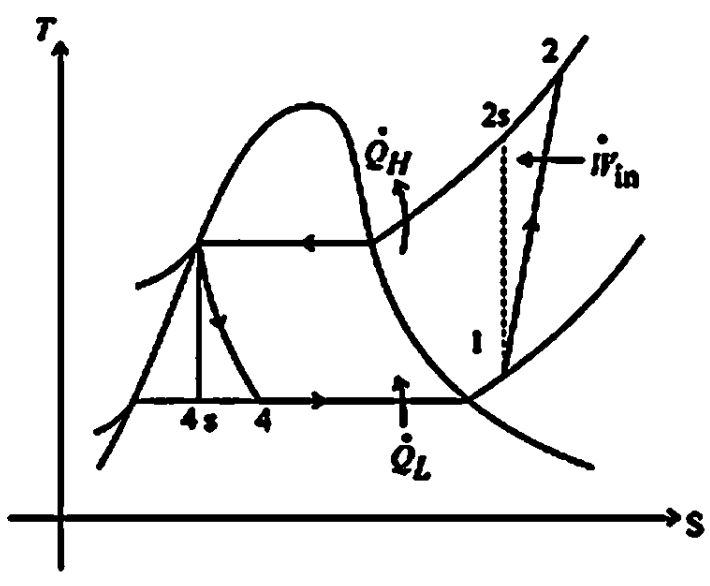
Saturation temperature of evaporator $T_1 = T_4 = -5^\circ C$

Saturation temperature of the condenser $T_3 = 45^\circ C$

Step 2 of 7

a)





Step 3 of 7

(b) State 3:

From the saturation temperature table

$$h_3 = h_f = 101\frac{\text{kJ}}{\text{kg}}$$

State4:

Since Throttling process enthalpy remains constant

$$h_4 = h_3 = 101\frac{\text{kJ}}{\text{kg}}$$

Step 4 of 7

State 1:

From the saturated refrigerant-22

At

$$h_1 = h_g = 248.1\frac{\text{kJ}}{\text{kg}}$$

The heat absorbed by the refrigerant in the evaporator is

$$q_L = h_1 - h_4$$
$$= 248.1 - 101$$
$$= 147.1\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{q_L = 147.1\frac{\text{kJ}}{\text{kg}}}$$

Step 5 of 7

(c) The COP of the air conditioner is

$$\text{COP}_R = SEER\left(\frac{1W}{3.412Btu/h}\right)$$
$$= \left(16\frac{Btu/h}{W}\right)\left(\frac{1W}{3.412Btu/h}\right)$$
$$= 4.689$$

Step 6 of 7

The work input to the compressor is

$$\text{COP}_R = \frac{q_L}{w_{in}}$$

$$\Rightarrow w_{in} = \frac{q_L}{COP_R}$$
$$= \frac{\left(147.1\frac{\text{kJ}}{\text{kg}}\right)}{4.689}$$
$$= 31.36\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{W_{in} = 31.36\frac{\text{kJ}}{\text{kg}}}$$

Step 7 of 7

The enthalpy at the compressor exit is

$$w_{in} = h_2 - h_1$$
$$\Rightarrow h_2 = h_1 + w_{in}$$
$$= \left(248.1\frac{\text{kJ}}{\text{kg}}\right) + \left(31.36\frac{\text{kJ}}{\text{kg}}\right)$$
$$= 279.5\frac{\text{kJ}}{\text{kg}}$$

The heat rejected from the refrigerant in the condenser is then

$$q_H = h_2 - h_3$$
$$= \left(279.5\frac{\text{kJ}}{\text{kg}}\right) - \left(101\frac{\text{kJ}}{\text{kg}}\right)$$
$$= 178.5\frac{\text{kJ}}{\text{kg}}$$

$$\boxed{q_H = 178.5\frac{\text{kJ}}{\text{kg}}}$$

Question: 9.164P

An actual refrigerator operates on the vapor-compression refrigeration cycle with refrigerant-22 as the working fluid. The refrigerant evaporates at -15°C and condenses at 40°C. The isentropic efficiency of the compressor is 83 percent. The refrigerant is superheated by 5°C at the compressor inlet and subcooled by 5°C at the exit of the condenser. Determine (a) the heat removed from the cooled space and the work input, in kJ/kg and the COP of the cycle. Determine (*b*) the same parameters if the cycle operated on the ideal vapor-compression refrigeration cycle between the same evaporating and condensing temperatures.

The properties of R-22 in the case of actual operation are:*h*1 =402.49 kJ/kg. *h*2=454.00 U/kg. *h*3 = 243.19 kJ/kg

The properties of R-22 in the case of ideal operation are: *h*1 =399.04 kJ/kg, *h*2 = 440.71 Id/kg, *h*3, =249.80 kJ/kg Note: state 1: compressor inlet, state 2: compressor exit, state 3: condenser exit, state 4: evaporator inlet.

Step 1 of 9

Given:

Working fluid: $R-22$

Temperature of the evaporator $T_1 = T_4 = -15^\circ C$

Temperature of the condenser $T_2 = 40^\circ C$

Exit temperature of the condenser $T_3 = 5^\circ C$

Properties of $R-22$:

Enthalpy	Actual condition	Ideal condition
h_1	$402.49\frac{kJ}{kg}$	$399.04\frac{kJ}{kg}$
h_2	$454.00\frac{kJ}{kg}$	$440.71\frac{kJ}{kg}$
h_3	$243.19\frac{kJ}{kg}$	$249.80\frac{kJ}{kg}$

Step 2 of 9

Actual refrigerator:

(a) Heat removed from the cold space is

From the properties of $R-22$

$$h_1 = 402.49\frac{kJ}{kg}$$

$$h_4 = 243.19\frac{kJ}{kg}$$

$$\dot{Q}_L = h_1 - h_4$$

$$= \left(402.49\frac{kJ}{kg}\right) - \left(243.19\frac{kJ}{kg}\right)$$

$$= 159.3\frac{kJ}{kg}$$

$$\boxed{\dot{Q}_L = 159.3\frac{kJ}{kg}}$$

Step 3 of 9

Work input to the compressor

From the properties of $R-22$

$$h_2 = 454\frac{kJ}{kg}$$

$$h_1 = 402.49\frac{kJ}{kg}$$

$$W_{in} = h_2 - h_1$$

$$= \left(454\frac{kJ}{kg}\right) - \left(402.49\frac{kJ}{kg}\right)$$

$$= 51.51\frac{kJ}{kg}$$

$$\boxed{W_{in} = 51.51\frac{kJ}{kg}}$$

Step 4 of 9

COP of the cycle

$$(COP)_{actual} = \frac{Q_L}{W_{in}}$$

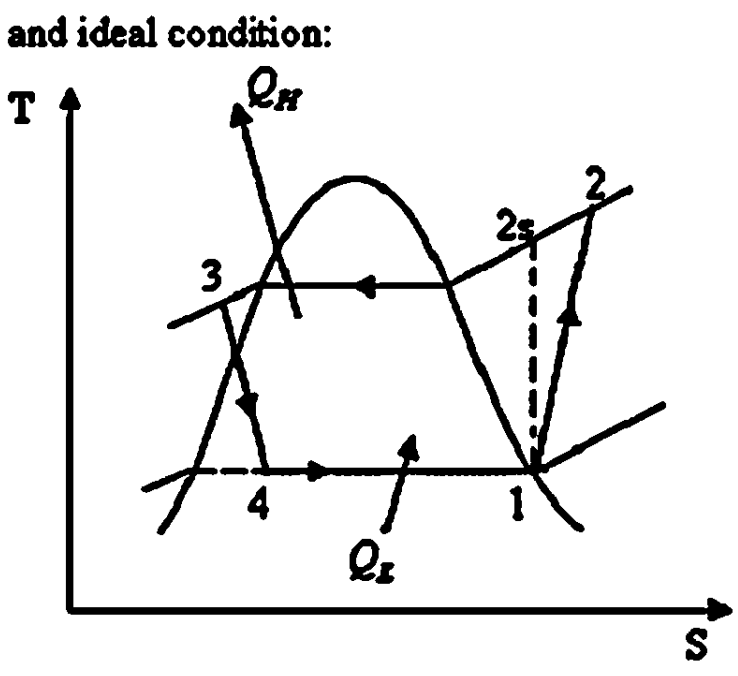
$$= \frac{159.3}{51.51}$$

$$= 3.092$$

$$\boxed{(COP)_{actual} = 3.092}$$

Step 5 of 9

T-*s* Diagram for actual and ideal condition:



Step 6 of 9

Ideal Cycle:

Heat removed from the refrigerated space

From the properties of $R-22$

$$h_1 = 399.04\frac{kJ}{kg}$$

$$h_4 = 249.8\frac{kJ}{kg}$$

Step 7 of 9

$$\dot{Q}_L = h_1 - h_4$$

$$= \left(399.04\frac{kJ}{kg}\right) - \left(249.8\frac{kJ}{kg}\right)$$

$$= 149.24\frac{kJ}{kg}$$

$$\boxed{\dot{Q}_L = 149.24\frac{kJ}{kg}}$$

Step 8 of 9

Work input to compressor

From the properties of $R-22$

$$h_1 = 399.04\frac{kJ}{kg}$$

$$h_2 = 440.71\frac{kJ}{kg}$$

$$W_{in} = (h_2 - h_1)$$

$$= \left(440.71\frac{kJ}{kg}\right) - \left(399.04\frac{kJ}{kg}\right)$$

$$= 41.67\frac{kJ}{kg}$$

$$\boxed{W_{in} = 41.67\frac{kJ}{kg}}$$

Step 9 of 9

COP of the ideal cycle

$$(COP)_{ideal} = \frac{Q_L}{W_{in}}$$

$$= \frac{149.24}{41.67}$$

$$= 3.58147$$

$$\boxed{(COP)_{ideal} = 3.58147}$$

Question:

A refrigerator uses refrigerant-134a as the working fluid and operates on the vapor-compression refrigeration cycle. The evaporator and condenser pressures are 200 kPa and 1400 kPa, respectively. The isentropic efficiency of the compressor is 88 percent. The refrigerant enters the compressor at a rate of 0.025 kg/s superheated by 10.1°C and leaves the condenser sub cooled by 4.4°C. Determine (*a*) the rate of cooling provided by the evaporator, the power input, and the COP. Determine (*b*) the same parameters if the cycle operated on the ideal vapor-compression refrigeration cycle between the same pressure limits.

Step 1 of 17

Given:

Working fluid: $R-134a$

Evaporator pressure $P_1 = P_4 = 200\text{kPa}$

Condenser pressure $P_2 = P_3 = 1400\text{kPa}$

Isentropic efficiency of the compressor $\eta_C = 0.88$

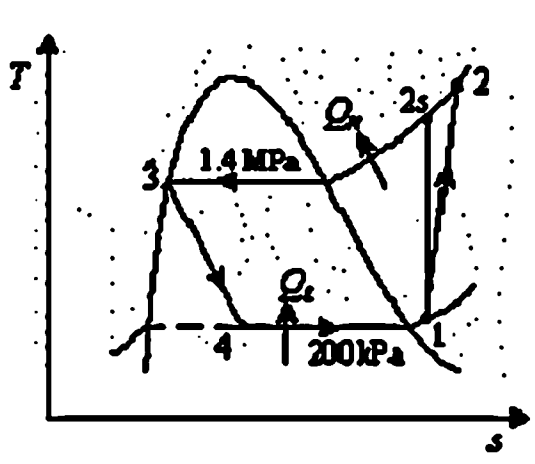
Mass flow rate to compressor $m = 0.025\dfrac{\text{kg}}{s}$

Superheat temperature $T_2 = 10.1^\circ\text{C}$

Sub cooled temperature $T_{3'} = 4.4^\circ\text{C}$

Step 2 of 17

T-s Diagram:



Step 3 of 17

(a)

State 1:

Since refrigerant enters into the compressor is superheated state. So

From the superheated refrigerant $R-134a$

At $P_1 = 200\text{kPa}$

$T_{sat} = -10.09^\circ\text{C}$

$\cong -10.1^\circ\text{C}$

$T_1 = (-10.1^\circ\text{C}) + (10.1^\circ\text{C})$

$= 0^\circ\text{C}$

Now take the properties at $T_1 = 0^\circ\text{C}$ and $P_1 = 200\text{kPa}$

$h_1 = 253.05\dfrac{\text{kJ}}{\text{kg}}$

$s_1 = 0.9698\dfrac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Step 4 of 17

State 2:

From the superheated refrigerant $R-134a$

At $P_2 = 1400\text{kPa}$ And $s_1 = s_2 = 0.9698\dfrac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Using the interpolation

$$h_{2s} = 285.47 + \frac{(0.9698-0.9389)(297.10-285.47)}{(0.9733-0.9389)}$$

$h_{2s} = 295.91\dfrac{\text{kJ}}{\text{kg}}$

$T_{sat} = 52.40^\circ\text{C}$

Step 5 of 17

State 3 and State 4:

$P_3 = 1400kPa$

$T_3 = 52.40^\circ\text{C} - T_{3'}$

$= 52.40^\circ\text{C} - 4.4^\circ\text{C}$

$= 48^\circ\text{C}$

From the Saturated refrigerant $R-134a$ Temperature table

At $T_3 = 48^\circ\text{C}$

$h_3 \cong h_f = 120.39\dfrac{\text{kJ}}{\text{kg}}$

Since state 3 to state 4 is the throttling process so enthalpy remains constant.

$h_4 = h_3 = 120.39\dfrac{\text{kJ}}{\text{kg}}$

Step 6 of 17

Isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.88 = \frac{\left(295.91\dfrac{\text{kJ}}{\text{kg}}\right) - \left(253.05\dfrac{\text{kJ}}{\text{kg}}\right)}{h_2 - \left(253.05\dfrac{\text{kJ}}{\text{kg}}\right)}$$

$$\Rightarrow h_2 = 301.75\frac{\text{kJ}}{\text{kg}}$$

Step 7 of 17

Amount of heat removed in evaporator

$\dot{Q}_L = \dot{m}(h_1 - h_4)$

$= \left(0.025\dfrac{\text{kg}}{s}\right)\left(253.05\dfrac{\text{kJ}}{\text{kg}} - 120.39\dfrac{\text{kJ}}{\text{kg}}\right)$

$= 3.317\dfrac{\text{kJ}}{s}$

$$= \left(0.025\frac{\text{kg}}{s}\right)\left(253.05\frac{\text{kJ}}{\text{kg}} - 120.39\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 3.317\frac{\text{kJ}}{s}$$

$$= 3.317\text{kW}$$

$$\boxed{\dot{Q}_L = 3.317\text{kW}}$$

Step 8 of 17

Amount of heat removed in condenser

$$\dot{Q}_H = \dot{m}(h_2 - h_3)$$

$$= \left(0.025\frac{\text{kg}}{s}\right)\left(301.75\frac{\text{kJ}}{\text{kg}} - 120.39\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 4.534\frac{\text{kJ}}{s}$$

$$= 4.534\text{kW}$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

$$= \left(0.025\frac{\text{kg}}{s}\right)\left(301.75\frac{\text{kJ}}{\text{kg}} - 253.05\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 1.217\frac{\text{kJ}}{s}$$

$$= 1.217\text{kW}$$

$$\boxed{\dot{W}_{in} = 1.217\text{kW}}$$

Step 9 of 17

$$COP = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$= \frac{(3.317\text{kW})}{(1.217\text{kW})}$$

$$= 2.725$$

$$\boxed{(COP)_{actual} = 2.725}$$

Step 10 of 17

(b)

Ideal vapor-compression refrigeration cycle

State 1:

Since refrigerant enters into the compressor is superheated state. So

From the superheated refrigerant $R-134a$

At $P_1 = 200\text{kPa}$

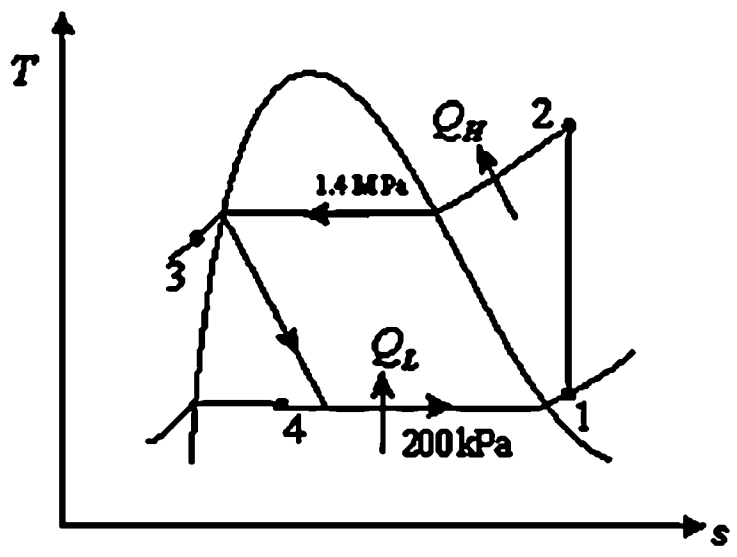
$$x_1 = 1$$

$$h_1 = 244.46\frac{\text{kJ}}{\text{kg}}$$

$$s_1 = 0.9377\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 11 of 17

T-s Diagram:



Step 12 of 17

State 2:

From the superheated $R-134a$ table

$$P_2 = 1400\text{kPa}$$

$$s_1 = s_2 = 0.9377\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Using the interpolation

$$h_2 = 276.12 + \frac{(0.9377 - 0.9105)(285.47 - 276.12)}{(0.9389 - 0.9105)}$$

$$h_2 = 285.08\frac{\text{kJ}}{\text{kg}}$$

Step 13 of 17

State 3:

From the Saturated refrigerant $R-134a$ Pressure table

$$P_3 = 1400kPa$$

$$x_1 = 0$$

$$h_3 = h_g = 127.22\frac{\text{kJ}}{\text{kg}}$$

Since state 3 to state 4 is the throttling process so enthalpy remains constant

$$h_4 = h_3 = 127.22\frac{\text{kJ}}{\text{kg}}$$

Step 14 of 17

Amount of heat removed in evaporator

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

$$= \left(0.025\frac{\text{kg}}{s}\right)\left(244.46\frac{\text{kJ}}{\text{kg}} - 127.22\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 2.931\frac{\text{kg}}{s}$$

$$= 2.931\text{kW}$$

$$= \left(0.025\frac{kg}{s}\right)\left(244.46\frac{kJ}{kg} - 127.22\frac{kJ}{kg}\right)$$

$$= 2.931\frac{kg}{s}$$

$$= 2.931 kW$$

$$\boxed{\dot{Q}_L = 2.931 kW}$$

9.165P pg 3

Step 15 of 17

Amount of heat rejection in condenser

$$\dot{Q}_H = \dot{m}(h_2 - h_3)$$

$$= \left(0.025\frac{kg}{s}\right)\left(285.08\frac{kJ}{kg} - 127.22\frac{kJ}{kg}\right)$$

$$= 3.947\frac{kg}{s}$$

$$= 3.947 kW$$

Step 16 of 17

Work input to the compressor

$$\dot{W}_H = \dot{m}(h_2 - h_1)$$

$$= \left(0.025\frac{kg}{s}\right)\left(285.08\frac{kJ}{kg} - 244.46\frac{kJ}{kg}\right)$$

$$= 1.016\frac{kJ}{s}$$

$$= 1.016 kW$$

$$\boxed{\dot{W}_H = 1.016 kW}$$

Step 17 of 17

COP of the ideal refrigeration cycle

$$COP = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$= \frac{(2.931 kW)}{(1.016 kW)}$$

$$= 2.884$$

$$\boxed{COP = 2.884}$$

Question: 9.166P

Do you think a heat pump system will be more cost-effective in New York or in Miami? Why?

Step 1 of 1

Heat pumps are most competitive in areas that have a large cooling load during the cooling season and a relatively small heating load during the heating season, such as in Miami. In these areas, the heat pump can meet the entire cooling and heating needs of residential or commercial buildings. The heat pump is least competitive in areas where the heating load is very large and the cooling load is small, such as in New York.

Question: 9.167P

What is a water-source heat pump? How does the COP of a water-source heat pump system compare to that of an air-source system?

Step 1 of 1

A water-source heat pump is a heat pump in which water is used as energy source for the heat pump instead of air. Water-source systems usually use well water from depths of up to 80 m in the temperature range of 5 to 18°C, and they do not have a frosting problem. They typically have higher COPs but are more complex and require easy access to a large body of water such as underground water.

Question:

9.168P

A heat pump uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The condenser pressure is 100 psia while the evaporator temperature is 40°F. What is the COP of this heat pump?

Step 1 of 7

Given:

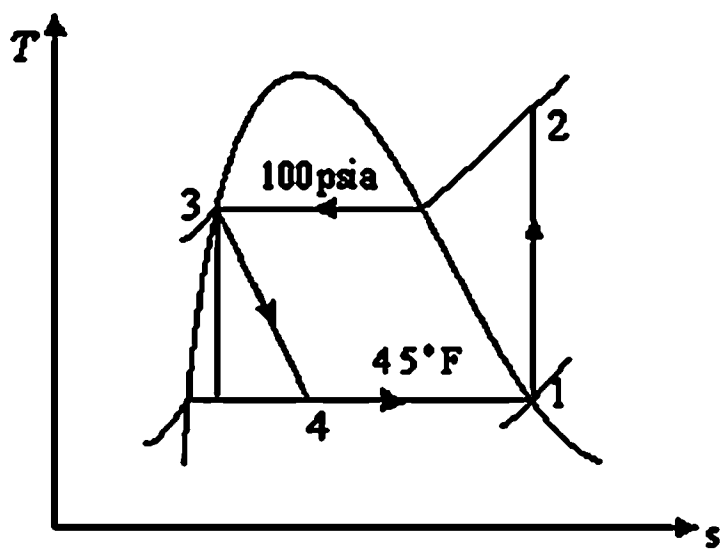
Working fluid: $R-134a$

Pressure of the condenser $P_3 = P_2 = 100\,\text{psia}$

Temperature of evaporator $T_1 = 40^\circ\text{F}$

Step 2 of 7

T-s Diagram:



Step 3 of 7

State 1:

Since refrigerant enters into the compressor is saturated vapor, So

From the Saturated refrigerant -134a Temperature table

At $T_1 = 40^\circ\text{F}$

$$h_1 = h_g = 108.78\frac{\text{Btu}}{\text{lbm}}$$

$$s_1 = 0.22189\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$$

Step 4 of 7

Sate 2:

Since ideal refrigeration cycle so entropy remains constant. And also refrigerant leaving to the compressor superheated condition.

$$s_2 = s_1 = 0.22189\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$$

From the Superheated refrigerant-134a

At $P_2 = 100\text{psia}$ and $s_2 = 0.22189\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$

By using the interpolation

$$h_2 = 114.05 + \frac{(0.22189-0.22016)(118.90-114.05)}{(0.22900-0.22016)}$$

$$h_2 = 114.99\frac{\text{Btu}}{\text{lbm}}$$

Step 5 of 7

State 3:

Since refrigerant leaves the condenser is saturated liquid.

From the saturated refrigerant -134a pressure table

$P_3 = 100\text{psia}$

$$h_3 = h_f = 37.869\frac{\text{Btu}}{\text{lbm}}$$

Step 6 of 7

State 4:

Since process 3 to 4 throttling so enthalpy remains constant.

$$h_4 = h_3 = 37.869\frac{\text{Btu}}{\text{lbm}}$$

$$\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$$

From the Superheated refrigerant-134a

At $P_2 = 100\text{psia}$ and $s_2 = 0.22189\frac{\text{Btu}}{\text{lbm}\cdot\text{R}}$

By using the interpolation

$$h_2 = 114.05 + \frac{(0.22189-0.22016)(118.90-114.05)}{(0.22900-0.22016)}$$

$$h_2 = 114.99\frac{\text{Btu}}{\text{lbm}}$$

Step 5 of 7

Question: 9.169P

A heat pump operates on the ideal vapor-compression refrigeration cycle and uses refrigerant-134a as the working fluid. The condenser operates at1000 kPa and the evaporator at 200 kPa. Determine this system's COP and the rate of heat supplied to the evaporator when the compressor consumes 6 kw.

Step 1 of 9

Given:

Working fluid: $R-134a$

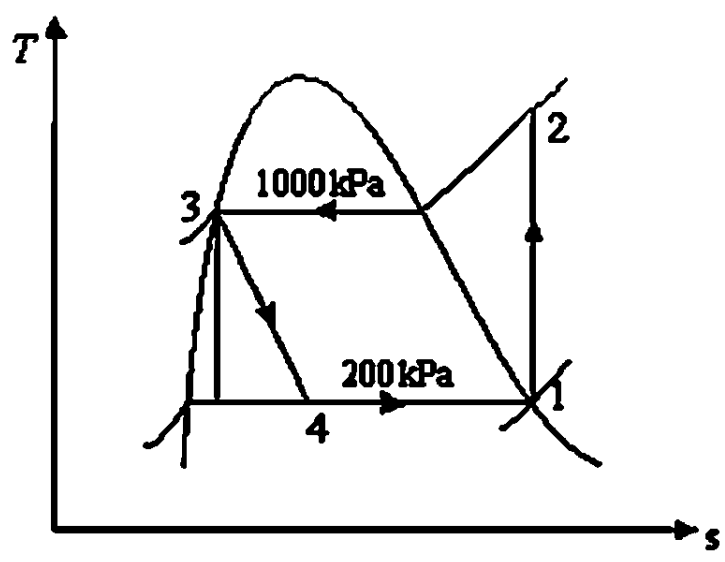
Operating pressure of the condenser $P_2 = P_3 = 1000\text{kPa}$

Operating pressure of the evaporator $P_1 = P_4 = 200\text{kPa}$

Power input to the compressor $W_{in} = 6\text{kW}$

Step 2 of 9

T-s diagram:



Step 3 of 9

State 1:

Since refrigerant enters into the compressor is saturated vapor, so

From the Saturated refrigerant -134a Pressure table

At $P_1 = 200\text{kPa}$

$$h_1 = h_g = 244.46\frac{\text{kJ}}{\text{kg}}$$

$$s_1 = s_g = 0.93773\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 4 of 9

Sate 2:

Since ideal refrigeration cycle so entropy remains constant. And also refrigerant leaving to the compressor superheated condition.

$$s_1 = s_2 = 0.93773\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

From the superheated refrigerant-134a

At $P_2 = P_3 = 1000\text{kPa}$ and $s_1 = s_2 = 0.93773\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Using method of interpolation

$$h_2 = 271.71 + \frac{(0.93773 - 0.9179)(282.74 - 271.71)}{(0.9525 - 0.9179)}$$

$$h_2 = 278.031\frac{\text{kJ}}{\text{kg}}$$

Step 5 of 9

Mass flow rate in to the compressor is

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

$$\Rightarrow \dot{m} = \frac{\dot{W}_{in}}{(h_2 - h_1)}$$

$$= \frac{6\text{kW}}{\left(277.031\frac{\text{kJ}}{\text{kg}} - 244.46\frac{\text{kJ}}{\text{kg}}\right)}$$

$$= 0.1842\frac{\text{kg}}{s}$$

Step 6 of 9

State 3:

Since refrigerant leaves the condenser is saturated liquid.

From the saturated refrigerant -134a pressure table

$P_3 = 1000\text{kPa}$

$$h_3 = h_f = 107.32\frac{\text{kJ}}{\text{kg}}$$

Step 7 of 9

State 4:

Since process 3 to 4 throttling so enthalpy remains constant.

$$h_4 = h_3 = 107.32\frac{\text{kJ}}{\text{kg}}.$$

Step 8 of 9

Rate of heat supplied to the evaporator

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

$$= \left(0.1842\frac{\text{kg}}{s}\right)\left(244.46\frac{\text{kJ}}{\text{kg}} - 107.32\frac{\text{kJ}}{\text{kg}}\right)$$

$$= 25.26\frac{\text{kJ}}{s}$$

$$\boxed{\dot{Q}_L = 25.26\text{kW}}$$

Step 9 of 9

COP of the heat pump is

$$(COP)_{HP} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$

$$COP = \frac{h_2 - h_3}{h_2 - h_1}$$

$$COP = \frac{278.031 - 107.32}{278.031 - 244.46}$$

$$= 5.085$$

$$\boxed{COP = 5.085}$$

Question: 9.170P

A heat pump operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. This heat pump is used to keep a space at 25°C by absorbing heat at the rate of 2.7 kW from geothermal water flowing through the evaporator. The evaporator operates at 20°C. and the condenser operates at 1400 kPa. The compressor receives work equal to 20 kJ for each kilogram of refrigerant flowing into it

(*a*) Sketch the hardware and the *T-s* diagram for this heat pump

(*b*) Determine the rate of heat transfer to the heated space at 25°C.

(*c*) Determine the COP of the heat pump.

Refrigerant-134a data: *T*= 20°C: *hf*= 79.3 kJ/kg

hg= 261.6 kJ/kg: *P*= 1400 kPa: *hf*= 127.2 kJ/kg

hg =276.2 kJ/kg

Step 1 of 9

Given:

Working fluid: $R-134a$

Temperature of heated space $T_R = 25^\circ C$

Rate of heat gain $\dot{Q}_L = 2.7 kW$

Operating temperature of the evaporator $T_{evap} = 20^\circ C$

Operating Pressure of the condenser $P_2 = P_3 = 1400 kPa$

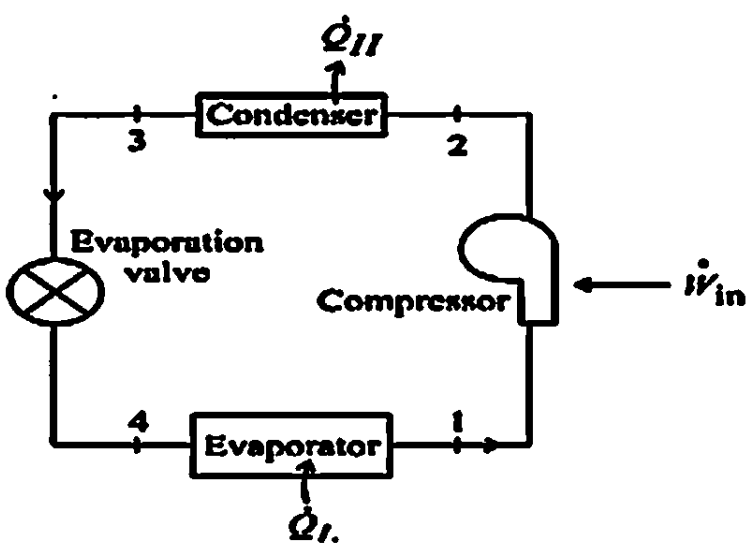
Compressor work $W_{in} = 20 kJ$

Properties R-134a:

At T=20°C	At P=1400kPa
$h_f = 79.3 \frac{kJ}{kg}$	$h_f = 127.2 \frac{kJ}{kg}$
$h_g = 261.6 \frac{kJ}{kg}$	$h_g = 276.2 \frac{kJ}{kg}$

Step 2 of 9

A) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic; the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.



Step 3 of 9

(b) State 1:

Since refrigerant enters in to the compressor is saturated vapor, so

From the above table

At $T_{evap} = 20^\circ C$

$$h_1 = h_g = 261.6 \frac{kJ}{kg}$$

Step 4 of 9

State 3:

From the above table

At $P_2 = P_3 = 1400 kPa$

$$h_3 = h_f = 127.2 \frac{kJ}{kg}$$

Step 5 of 9

State 4:

Process 3 to 4 throttling Process so enthalpy remains constant.

$$h_4 = h_3 = 127.2 \frac{kJ}{kg}$$

Step 6 of 9

The enthalpy at the compressor exit is

$$w_{in} = h_2 - h_1$$

$$\Rightarrow h_2 = h_1 + w_{in}$$

$$= 261.6 \frac{kJ}{kg} + 20 \frac{kJ}{kg}$$

$$= 281.6 \frac{kJ}{kg}$$

Step 7 of 9

The mass flow rate through the cycle is

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

$$\Rightarrow \dot{m} = \frac{\dot{Q}_L}{h_1 - h_4}$$

$$= \frac{2.7 \frac{kJ}{s}}{\left(261.6 \frac{kJ}{kg} - 127.2 \frac{kJ}{kg}\right)}$$

$$= 0.02009 \frac{kg}{s}$$

Step 8 of 9

The rate of heat transfer to the heated space is

$$\dot{Q}_H = \dot{m}(h_2 - h_3)$$

$$= \left(0.02009 \frac{kg}{s}\right)\left(281.6 \frac{kJ}{kg} - 127.2 \frac{kJ}{kg}\right)$$

$$= 3.10 \frac{kJ}{s}$$

$$= 3.10 kW$$

$$\boxed{\dot{Q}_H = 3.10 kW}$$

Step 9 of 9

(c) The COP of the heat pump is

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$

$$= \frac{\dot{Q}_H}{\dot{m} w_{in}}$$

$$= \frac{3.10 kW}{\left(0.02009 \frac{kg}{s}\right)\left(20 \frac{kJ}{kg}\right)}$$

$$= 7.72$$

$$\boxed{(COP)_{HP} = 7.72}$$

Question: 9.171P pg 1

A building requires a 2-ton heat pump for maintaining the interior space at 27°C when the outside temperature is 5°C. The heat pump operates on the normal vapor compression refrigeration cycle and uses refrigerant-134a as the working fluid. The heat pump operating conditions require an evaporator pressure of 240 kPa and a condenser pressure of 1600 kPa while the compressor has an isentropic efficiency] of 85 percent. Selected data for refrigerant-134a are provided in the table below.

P, kPa	T_{sat}, °C	h_f, kJ/kg	h_g, kJ/kg	s_g, kJ/kg·K
240	−5.37	43	244	0.9222
1600	57.92	134	275	0.8982

For R-134a at P = 1600 kPa and s = 0.9222 kJ/kg·K, h = 285 kJ/kg. Also, 1 ton = 211 kJ/min.

(*a*) Sketch the hardware and the *T*-*s* diagram for this heat pump system.

(*b*) Determine the power required to drive the heat pump, in kW and the COP.

Step 1 of 9

Given:
Working fluid: $R-134a$
Capacity of the heat pump $Q_H = 2\,\text{ton}$
Highest temperature $T_H = 27^\circ\text{C}$
Lowest temperature $T_L = 5^\circ\text{C}$
Operating pressure of evaporator $P_1 = P_4 = 240\,\text{kPa}$
Operating pressure of Condenser $P_2 = P_3 = 1600\,\text{kPa}$
Isentropic efficiency of the compressor $\eta_C = 0.85$
Properties of R-134a:

P,kPa	T_{sat},°C	h_f, kJ/kg	h_g, kJ/kg	s_g, kJ/kg·K
240	-5.37	43	244	0.92222
1600	57.92	134	275	0.8982

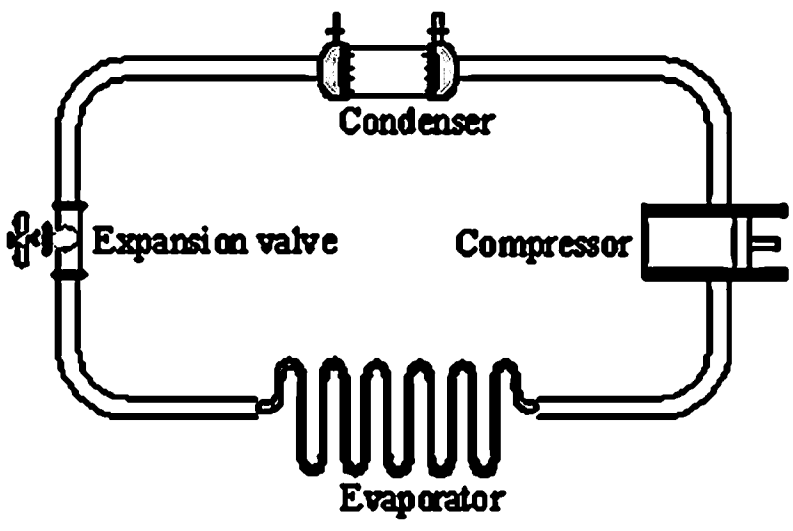
Superheated conditions:

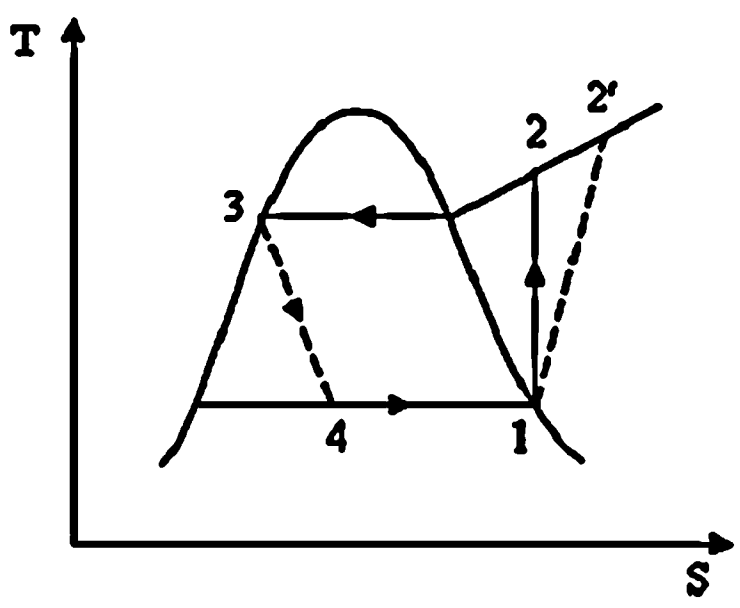
For R-134a at $P = 1600\,\text{kPa}$ and $s = 0.9222\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$

$h_{sup} = 285\,\frac{\text{kJ}}{\text{kg}}$ And $1\text{Ton} = 210\,\frac{\text{kJ}}{\text{min}} = 3.5\text{kW}$

Step 2 of 9

A)





Step 3 of 9

b) State1:
Since refrigerant enters into the compressor Saturated vapor ,so
From the above table
At $P_1 = P_4 = 240\,\text{kPa}$

$h_1 = h_g = 244\,\frac{\text{kJ}}{\text{kg}}$

$s_1 = s_g = 0.9222\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Step 4 of 9

State 2:
Since refrigerant leaving to the compressor is superheated conditions.So
From the above table
At $P = 1600\,\text{kPa}$

$h_2 = h_{sup} = 285\,\frac{\text{kJ}}{\text{kg}}$

Step 5 of 9

State 3:
Refrigerant leaving to the condenser is saturated liquid, So
From the above table
At $P = 1600\,\text{kPa}$

$h_3 = h_f = 134\,\frac{\text{kJ}}{\text{kg}}$

Step 6 of 9

We know that
Efficiency of the compressor

Step 6 of 9

We know that
Efficiency of the compressor

$$\eta_c = \frac{h_2 - h_1}{h_{2'} - h_1}$$

$$0.85 = \frac{\left(285\frac{kJ}{kg} - 244\frac{kJ}{kg}\right)}{\left(h_{2'} - 244\frac{kJ}{kg}\right)}$$

$$\Rightarrow h_{2'} = 292.24\frac{kJ}{kg}$$

Step 7 of 9

Capacity of the heat pump $Q_H = 2\text{ton}$
And we also know that

$$1\text{Ton} = 210\frac{kJ}{min}$$
$$= 3.5kW$$
$$\dot{Q}_H = 2\times 3.5kW$$
$$= 7\,kW$$

$$\dot{Q}_H = \dot{m}(h_{2'} - h_3)$$
$$\Rightarrow \dot{m} = \frac{7kW}{\left(292.24\frac{kJ}{kg} - 134\frac{kJ}{kg}\right)}$$
$$= 0.04424\frac{kg}{s}$$

Step 8 of 9

Power required for driving the pump

$$\dot{W}_{in} = \dot{m}(h_{2'} - h_1)$$
$$= 0.04424\frac{kg}{s}\left(292.24\frac{kJ}{kg} - 244\frac{kJ}{kg}\right)$$
$$= 2.134\frac{kJ}{s}$$

$$\boxed{\dot{W}_{in} = 2.134kW}$$

Step 9 of 9

COP of the heat pump is

$$(COP)_{HP} = \frac{\dot{Q}_H}{\dot{W}_{in}}$$
$$= \frac{7}{2.134}$$
$$= 3.28$$

$$\boxed{COP = 3.28}$$

Question: 9.172P pg 1

A heat pump with refrigerant-134a as the working ' fluid is used to keep a space at 25°C by absorbing heat from geothermal water that enters the evaporator at 50°C at a rate of 0.065 kg/s and leaves at 40°C. The refrigerant enters the evaporator at 20°C with a quality of 23 percent and leaves at the inlet pressure as saturated vapor. The refrigerant loses 300 W of heat to the surroundings as it flows through the compressor and the refrigerant leaves the compressor at 1.4 MPa at the same entropy as the inlet. Determine (*a*) the degrees of subcooling of the refrigerant in the condenser, (*b*) the mass flow rate of the refrigerant, (*c*) the heating load and the COP of the heat pump, and (*d*) the theoretical minimum power input to the compressor for the same heating load.

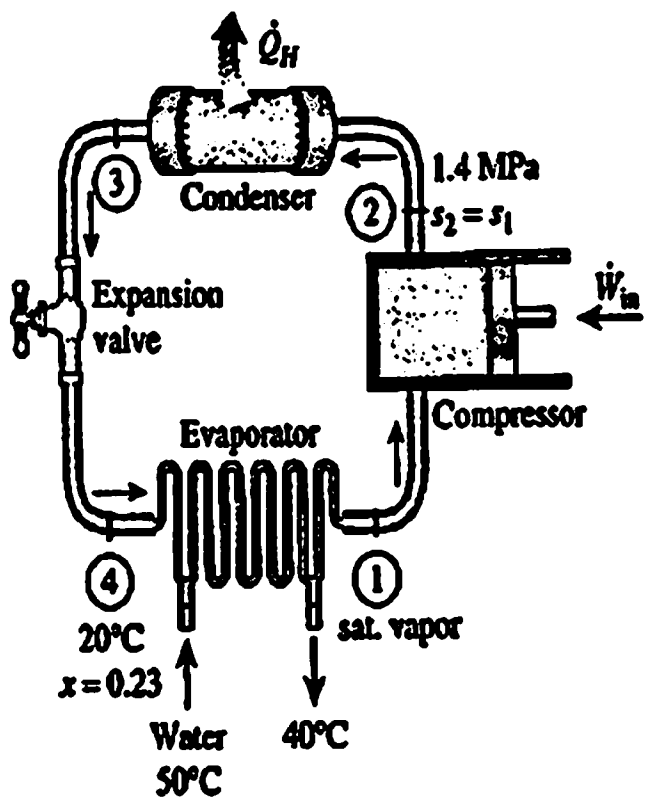
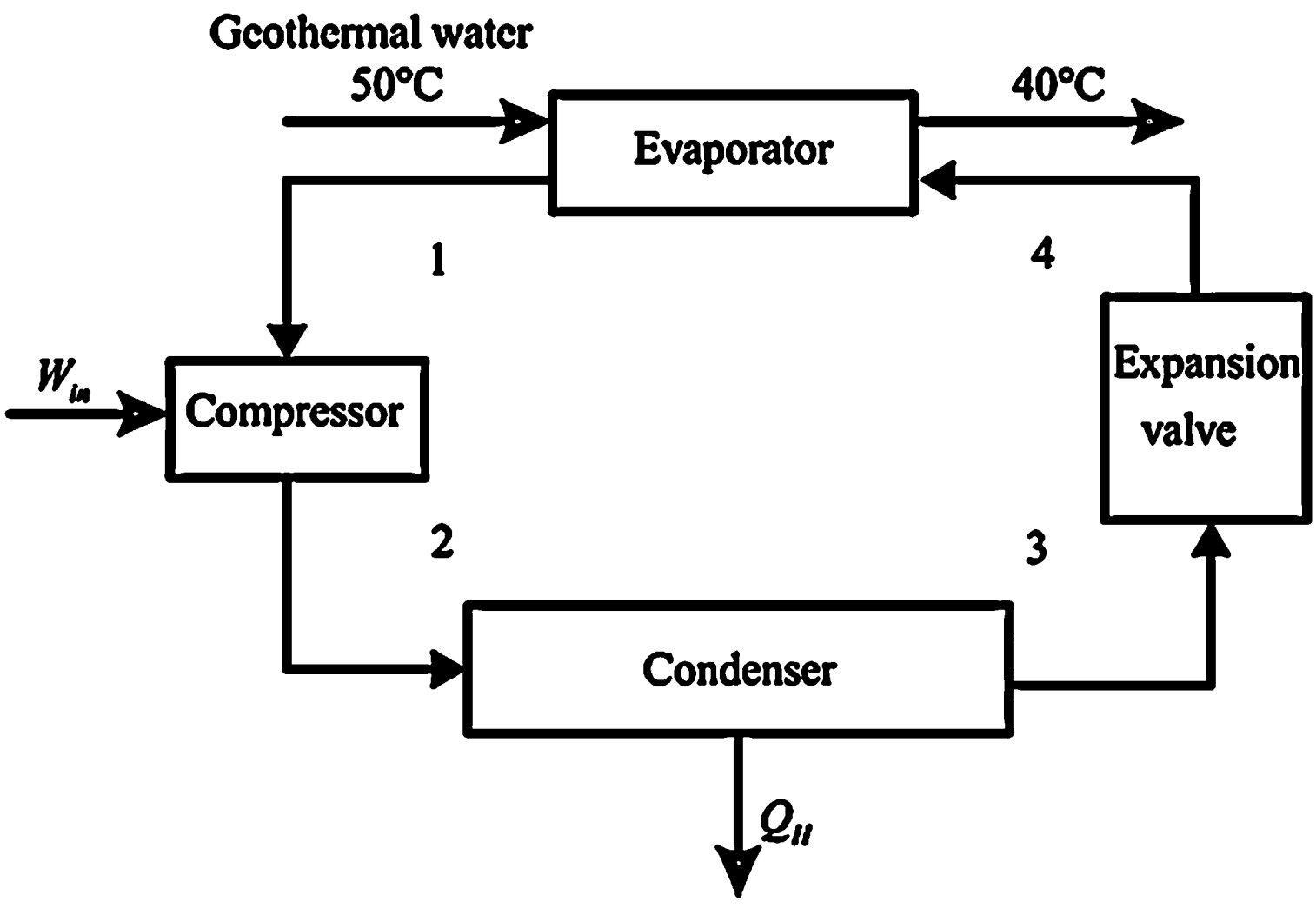


FIGURE P9–172

Step 1 of 12

Sketch the block diagram showing heat pump cycle:



Step 2 of 12

(a)

Calculate the enthalpy at the entry of the evaporator:

Obtain the enthalpies from the table A-11, "Saturated refrigerant-134a-Temperature Table" in the textbook.

The enthalpy of a saturated liquid h_{f4} is $79.32\left(\frac{\text{kJ}}{\text{kg}}\right)$.

The enthalpy of the evaporation h_{fg4} is $182.27\left(\frac{\text{kJ}}{\text{kg}}\right)$.

The saturated pressure P_4 is $572.07\ \text{kPa}$.

Hence substitute $79.32\left(\text{kJ}/\text{kg}\right)$ for h_{f4} and $182.27\left(\text{kJ}/\text{kg}\right)$ for h_{fg4}.

$$h_4 = h_{f4} + x_4 h_{fg4}$$

Here, h_{f4} is the enthalpy of a saturated liquid, h_{fg4} is enthalpy of the evaporation, and x_4 is the quality of the refrigerant at the entry of evaporator.

Substitute $79.32\,\text{kJ}/\text{kg}$ for h_{f4}, $182.27\left(\text{kJ}/\text{kg}\right)$ for h_{fg4} and 0.23 for x_4.

$$\begin{aligned} h_4 &= 79.32 + (0.23 \times 182.27) \\ &= 121.24\,\text{kJ/kg} \end{aligned}$$

Step 3 of 12

The pressure remains constant when refrigerant leaves the evaporator in a saturated stage. Hence the pressures at the inlet and the outlet are equal.

$$\begin{aligned} P_4 &= P_1 \\ &= 572.07\,\text{kPa} \end{aligned}$$

At saturated stage the enthalpy at the inlet of the compressor is equal to the enthalpy of the saturated vapor.

Hence $h_1 = h_{g1}$

$$= 261.59\,\text{kJ/kg}$$

And at the saturated stage the entropy at the inlet of the compressor is equal to the enthalpy of the saturated vapor.

Hence $s_1 = s_{g1}$

$$= 0.92234\,\text{kJ/kg}\cdot\text{K}$$

Step 4 of 12

Calculate the enthalpy at the exit of the compressor pressure by interpolation method:

Step 4 of 12

Calculate the enthalpy at the exit of the compressor pressure by interpolation method:

For h_{avg}, at $h_{max} = 285.47\ \text{kJ/kg}$ and $s_{max} = 0.9389\ \text{kJ/kg.k}$, $h_{min} = 276.12\ \text{kJ/kg}$ and $s_{min} = 0.9015\ \text{kJ/kg.k}$.

The formula for interpolation is follows:

$$h_{avg} = h_{min} + (h_{max} - h_{min})\left(\frac{s_{avg} - s_{min}}{s_{max} - s_{min}}\right)$$

$$h_2 = h_{0.9105} + (h_{0.9389} - h_{0.9105})\left(\frac{0.924 - 0.9105}{0.9389 - 0.9105}\right)$$

$$= 276.12 + (285.47 - 276.12)\left(\frac{0.924 - 0.9105}{0.9389 - 0.9105}\right)$$

$$= 280.56\ \text{kj/kg}$$

Therefore, the enthalpy at the exit of the compressor is $280.56\ \text{kJ/kg}$

Step 5 of 12

Obtain the enthalpy values for the geothermal water inlet and outlet respectively:
From the saturated water temperature table A-4 the values of enthalpy for the geothermal water inlet and outlet are follows;

$$\text{At } T_{GI} = 50°\text{C ; enthalpy } h_{GI} = 209.34\ \text{kJ/kg}$$
$$\text{At } T_{GO} = 40°\text{C ; enthalpy } h_{GO} = 167.53\ \text{kJ/kg}$$

From the A-12 saturated refrigerant of R134-a pressure table the saturated temperature at pressure of 1400 kPa the temperature is $T_{sat} = 52.4°\text{C}$.

We know that enthalpy is constant in expansion valve therefore, $h_4 = h_3$. Also inlet and outlet pressure are constant therefore, $P_2 = P_3 = 1.4\text{MPa}$.

Step 6 of 12

Calculate the temperature at state 3 by interpolation as follows:

Obtain the enthalpies at the following temperatures from the table A-11, "Saturated refrigerant -134a- Temperature table" in the textbook.

$$h_{@48°\text{C}} = 120.39\ \text{kJ/kg}$$
$$h_{@52°\text{C}} = 126.59\ \text{kJ/kg}$$

Use interpolation and find temperature at state 3 as follows:

$$T_3 = 48 + \left(\frac{52 - 48}{126.59 - 120.39}\right)(121.24 - 120.39)$$
$$= 48.6°\text{C}$$

Calculate the degree of sub cooling as follows:

$$T_{subcooling} = T_{sat} - T_3$$

Substitute $52.4°\text{C}$ for T_{sat} and $48.6°\text{C}$ for T_3.

$$T_{subcooling} = 52.4 - 48.6$$
$$= 3.8°\text{C}$$

Step 7 of 12

Therefore, the degree of sub cooling is $\boxed{3.8°\text{C}}$.

Step 8 of 12

(b)

Calculate the heat absorbed from geothermal water as follows.

$$\dot{Q} = \dot{m}(h_{GI} - h_{GO})$$

Here, $\dot{m}$ is the mass flow rate, h_{GI} is the enthalpy of the geo water inlet, and h_{GO} is the enthalpy of the geo water outlet.

Substitute 0.0065 for $\dot{m}$, $209.34\ \text{kJ/kg}$ for h_{GI} and $167.53\ \text{kJ/kg}$ for h_{GO}.

$$\dot{Q} = 0.065 \times (209.34 - 167.53)$$
$$= 2.71\ \text{kW}$$

Calculate the mass flow rate of the refrigerant R134-a as follows.

$$\dot{m}_R = \frac{\dot{Q}}{h_1 - h_4}$$

Here, $\dot{m}_R$ is the mass flow rate of the refrigerant, h_1 is the enthalpy at the entry of the compressor and h_4 is the enthalpy at the entry of the evaporator and is the heat absorbed by the geothermal water $\dot{Q}$.

Substitute, 2.71 kW for $\dot{Q}$, $261.59\ \text{kJ/kg}$ for h_1, and $121.74\ \text{kJ/kg}$ for h_4.

$$\dot{m}_R = \frac{2.71}{260.03 - 121.74}$$
$$= 0.01965\ \text{kg/s}$$

Therefore, mass flow rate of the refrigerant is $\boxed{0.01965\ \text{kg/s}}$.

Step 9 of 12

(c)

Calculate the Power input to the heat pump as follows.

$$\dot{W}_{in} = \dot{m}_R(h_2 - h_1) + \dot{Q}_{II}$$

Here, $\dot{m}_R$ mass flow rate of the refrigerant, h_2, h_1 is the enthalpy at the entry of the compressor and the exit of the compressor and $\dot{Q}_{II}$ is the heat loss through the compressor.

Calculate the Power input to the heat pump as follows:

$$w_{in} = \dot{m}_R (h_2 - h_1) + Q_H$$

Here, $\dot{m}_R$ mass flow rate of the refrigerant, h_2, h_1 is the enthalpy at the entry of the compressor and the exit of the compressor and Q_H is the heat loss through the compressor.

Substitute, $0.01965\,^{kg}/_{s}$ for $\dot{m}_R$, $261.59\,^{kJ}/_{kg}$ for h_1, $280.5\,^{kJ}/_{kg}$ for h_2 and $0.3\,kW$ for Q_H.

$$\begin{aligned} w_{in} &= 0.01965(280.5 - 261.59) + 0.3 \\ &= 0.671\,kW \end{aligned}$$

Step 10 of 12

Calculate the heating effect of the condenser:

$$\dot{Q}_H = \dot{m}_R (h_2 - h_3)$$

Here, $\dot{Q}_H$ is the heating effect of the condenser, h_2, h_3 is the enthalpy at the entry of the compressor and the exit of the condenser and $\dot{m}_R$ is the mass flow rate of the refrigerant.

Substitute $0.01965\,^{kg}/_{s}$ for $\dot{m}_R$, $280.5\,^{kJ}/_{kg}$ for h_2, and $121.74\,^{kJ}/_{kg}$ for h_4

$$\begin{aligned} \dot{Q}_H &= 0.01965 \times (280.5 - 121.74) \\ &= 3.119\,kW \end{aligned}$$

Therefore, the heating load is $\boxed{3.119\,kW}$.

Step 11 of 12

Calculate the coefficient of performance of the heat pump:

$$COP = \frac{\dot{Q}_H}{w_{in}}$$

Here, *COP* is the coefficient of performance, $\dot{Q}_H$ is the heating effect of the condenser and w_{in} is the work input.

Substitute 3.119 kW for $\dot{Q}_H$ and 0.671 kW for w_{in}.

$$\begin{aligned} COP &= \frac{3.119}{0.671} \\ &= 4.648 \end{aligned}$$

Therefore, COP of the heat pump is $\boxed{4.648}$.

Step 12 of 12

(d)

Calculate the reversible COP as follows:

$$COP_{rev} = \frac{1}{1 - \frac{T_L}{T_H}}$$

Substitute 298 K for T_L and 323 K for T_H.

$$\begin{aligned} COP_{rev} &= \frac{1}{1 - \frac{298}{323}} \\ &= 12.29 \end{aligned}$$

Calculate the minimum power input as follows:

$$w_{in-min} = \frac{\dot{Q}_H}{COP_{rev}}$$

Substitute 3.119 kW for $\dot{Q}_H$ and 12.29 for COP_{rev}.

$$\begin{aligned} w_{in-min} &= \frac{3.119}{12.92} \\ &= 0.241\,kW \end{aligned}$$

Therefore, the minimum power input to the compressor is $\boxed{0.241\,kW}$.

Question: 9.173P pg 1

Refrigerant-134a enters the condenser of a residential heat pump at 800 kPa and 55°C at a rate of 0.018 kg/s and leaves at 750 kPa subcooled by 3°C. The refrigerant enters the compressor at 200 kPa superheated by 4°C. Determine (*a*) the isentropic efficiency of the compressor, (*b*) the rate of heat supplied to the heated room, and (*c*) the COP of the heat pump. Also, determine (*d*) the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the pressure limits of 200 and 800 kPa.

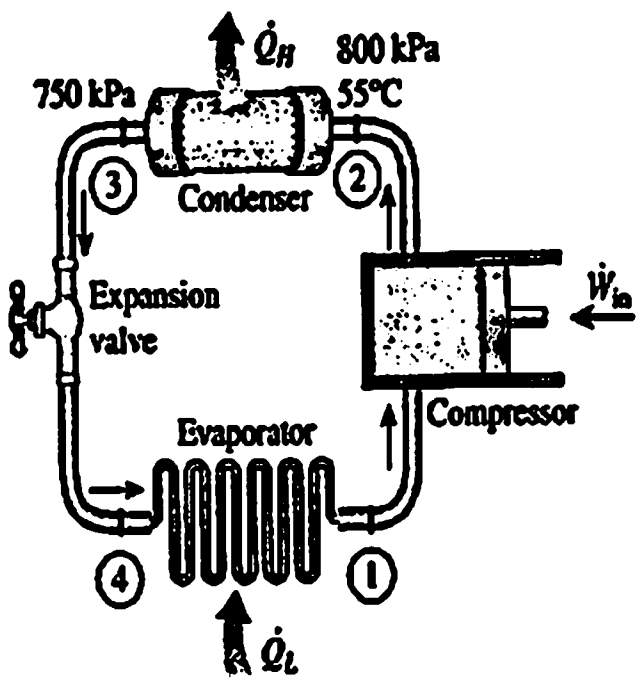


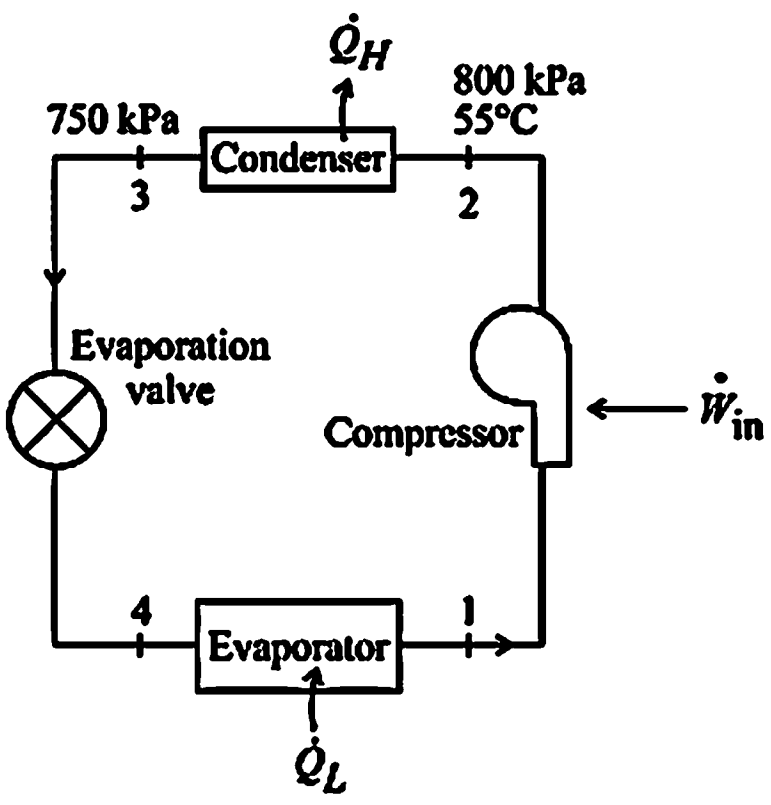
FIGURE P9–173

Step 1 of 16

Given:
Working fluid: R-134a
Pressure at the inlet of condenser $P_2 = 800\ \text{kPa}$
Temperature at the inlet of condenser $T_2 = 55°\text{C}$
Pressure at the exit of condenser $P_3 = 750\ \text{kPa}$
Pressure at the exit of evaporator $P_1 = 200\ \text{kPa}$
Mass flow rate $\dot{m} = 0.018\ \text{kg/s}$

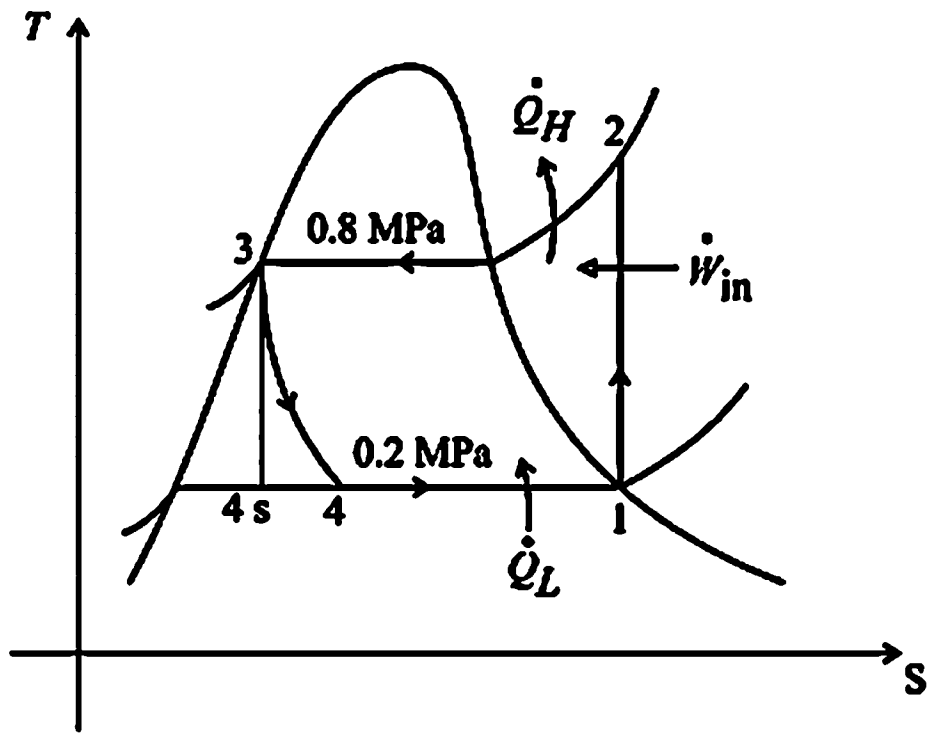
Step 2 of 16

Block diagram of vapor-compression refrigeration cycle



Step 3 of 16

T-S diagram of vapor-compression refrigeration cycle



Step 4 of 16

From superheated refrigerant-134a table

At $P_2 = 800\ \text{kPa}, T_2 = 55°\text{C}$

$$h_2 = 286.69 + \frac{(55-50)}{(60-50)}(296.81-286.69)$$

$$h_2 = 291.75\ \text{kJ/kg}$$

Step 5 of 16

From saturated refrigerant-134a pressure table

$T_{sat@750kPa} = 29.06°\text{C}$

At $P_3 = 750\ \text{kPa}$ and $T_3 = (29.06-3)°\text{C} = 26.06°\text{C}$

From saturated refrigerant-134a temperature table

$$h_{3@26.06°C} = 87.83 + \left(\frac{26.06-26}{28-26}\right)(90.69-87.83)$$

$$h_{3@26.06°C} = 87.91\ \text{kJ/kg}$$

Step 6 of 16

For throttling, $h_4 = h_3 = 87.91\ \text{kJ/kg}$

From saturated refrigerant-134a pressure table

$T_{sat@200kPa} = -10.09°\text{C}$

Step 7 of 16

From superheated refrigerant-134a table

$T_{sat@200kPa} = -10.09\,^{\circ}C$

Step 7 of 16

From superheated refrigerant-134a table

At $P_1 = 200\,kPa, T_1 = (-10.09+4)^{\circ}C = -6.09^{\circ}C$

$$h_1 = 244.54 + \frac{(-6.09-(-10))}{(0-(-10))}(253.05-244.54)$$

$$h_1 = 247.87\,kJ/kg$$

$$s_1 = 0.9380 + \frac{(-6.09-(-10))}{(0-(-10))}(0.9698-0.9380)$$

$$s_1 = 0.9504\,kJ/kg.K$$

Step 8 of 16

From superheated refrigerant-134a table

At $P_2 = 800\,kPa, s_2 = s_1 = 0.9504\,kJ/kg.K$

$$h_{2s} = 276.45 + \frac{(0.9504-0.9480)}{(0.9802-0.9480)}(286.69-276.45)$$

$$h_{2s} = 277.21\,kJ/kg$$

Step 9 of 16

Assumptions:

1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 10 of 16

(a) Isentropic efficiency of the compressor:

Isentropic efficiency of the compressor, $\eta_C = \dfrac{h_{2s}-h_1}{h_2-h_1}$

$$\eta_C = \frac{277.21-247.87}{291.75-247.87}$$

$$\boxed{\eta_C = 0.6686}$$

Step 11 of 16

(b) Rate of heat supplied to the room:

Rate of heat supplied to the room, $\dot{Q}_H = \dot{m}\times(h_2-h_3)$

$$\dot{Q}_H = 0.018\times(291.75-87.91)$$

$$\boxed{\dot{Q}_H = 3.67\,kW}$$

Step 12 of 16

(c) COP:

Power input, $\dot{W}_{in} = \dot{m}\times(h_2-h_1)$

$$\dot{W}_{in} = 0.018\times(291.75-247.87)$$

$$\dot{W}_{in} = 0.79\,kW$$

But, $COP = \dfrac{\dot{Q}_H}{\dot{W}_{in}}$

$$COP = \frac{3.67}{0.79}$$

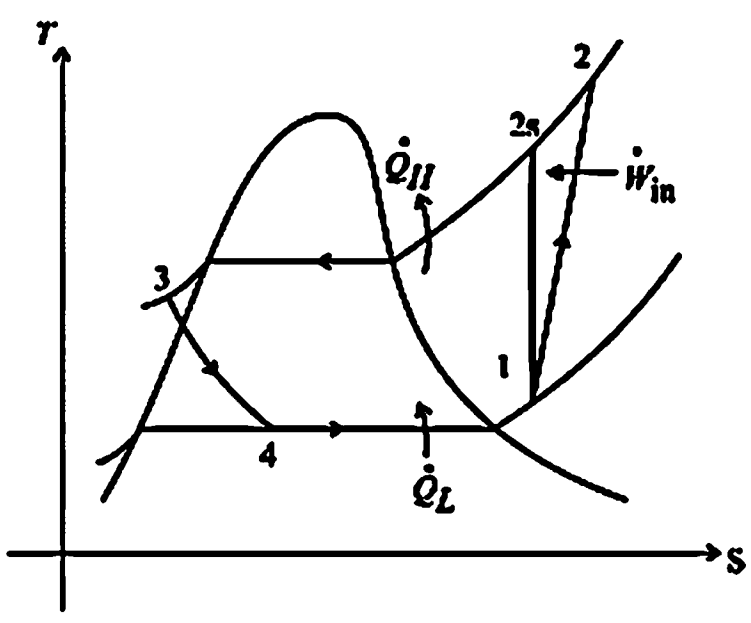
$$\boxed{COP = 4.646}$$

Step 13 of 16

(d) COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the pressure limits 200 kPa and 800 kPa:

Step 14 of 16

T-S diagram of vapor-compression refrigeration cycle



Step 15 of 16

From saturated refrigerant-134a pressure table

$h_1 = h_{g@200kPa} = 244.46\,kJ/kg$

$s_1 = s_{g@200kPa} = 0.9377\,kJ/kg.K$

From superheated refrigerant-134a table

At $P_2 = 800\,kPa, s_2 = s_1 = 0.9377\,kJ/kg.K$

$$h_2 = 267.29 + \frac{(0.9377-0.9183)}{(0.9480-0.9183)}(276.45-267.29)$$

$$h_2 = 273.27\,kJ/kg$$

$h_4 = h_3 = h_{f@800kPa} = 95.47\,kJ/kg$

But, $COP = \dfrac{\dot{Q}_H}{\dot{W}_{in}}$

$$COP = \frac{(h_2-h_3)}{(h_2-h_1)}$$

$$COP = \frac{273.27-95.47}{273.27-244.46}$$

$$\boxed{COP = 6.171}$$

$$COP = \frac{273.27-95.47}{273.27-244.46}$$

$$\boxed{COP = 6.171}$$

Rate of heat supplied to the room, $\dot{Q}_H = \dot{m} \times (h_2 - h_3)$

$$\dot{Q}_H = 0.018 \times (273.27-95.47)$$

$$\boxed{\dot{Q}_H = 3.2\ \text{kW}}$$

Step 16 of 16

Question:

9.174P

An Otto cycle with a compression ratio of 7 begins its compression at 90 kPa and 15°C. The maximum cycle temperature is 1000°C. Utilizing air-standard assumptions, determine the thermal efficiency of this cycle using (*a*) constant specific heats at room temperature and(*b*) variable specific heats

Step 1 of 6

Given:

Compression ratio of the Otto cycle is $r = 7$

Initial Temperature (before compression), $T_1 = 15^0\text{C}$

Initial Pressure (before compression), $P_1 = 90\ \text{kPa}$

Specific heat at constant pressure, $C_P = 1.005\ \text{kJ/KgK}$

Specific heat at constant temperature, $C_V = 0.7\ \text{kJ/KgK}$

Gas constant of air, $R = 0.3\ \text{kJ/KgK}$

Specific heat ratio, $k = 1.4$

Step 2 of 6

(a)

With Constant specific heats, the efficiency of the Otto cycle is given as

$$\eta_{otto} = 1 - \left(\frac{1}{r}\right)^{k-1}$$

$$= 1 - \left(\frac{1}{7}\right)^{k-1}$$

$$= 1 - 0.4591$$

$$= 0.5409$$

Hence the thermal efficiency of Otto cycle is $\boxed{\eta_{otto} = 54.09\%}$

Step 3 of 6

(b)

From ideal gas properties table of air at $T_1\,(288\ \text{K})$,

$V_{r1} = 688.1$ and

$u_1 = 205.48\ \text{kJ/kg}$

But compression ratio, $r = \frac{V_1}{V_2} = \frac{V_{r1}}{V_{r2}}$ Hence

$$V_{r2} = \frac{V_{r1}}{r}$$

$$= \frac{688.1}{7}$$

$$= 98.3$$

From ideal gas properties table of air, the internal energy corresponding to V_{r2} is

$$u_2 = 477.62\ \text{kJ/kg}$$

Step 4 of 6

Temperature after constant volume heat addition is $T_3 = 1273\ \text{K}$

From ideal gas properties table of air at T_3,

$$V_{r3} = 12.045 \text{ and}$$

$$u_3 = 998.51\ \text{kJ/kg}$$

Hence amount of heat added is

$$q_{in} = u_3 - u_2$$

$$= 998.51 - 447.62$$

$$= 550.89\ \text{kJ/kg}$$

Step 5 of 6

From compression ratio, $r = \frac{V_4}{V_3} = \frac{V_{r4}}{V_{r3}}$ Hence

$$V_{r4} = rV_{r3}$$

$$= 7 \times 12.045$$

$$= 84.32$$

From ideal gas properties table of air, the internal energy corresponding to V_{r4} is

$$u_4 = 475.54\ \text{kJ/kg}$$

Hence amount of heat rejected is

$$q_{out} = u_4 - u_1$$

$$= 475.54 - 205.48$$

$$= 270.06\ \text{kJ/kg}$$

Step 6 of 6

Efficiency of the cycle,

$$\eta_{Otto} = 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - \frac{270.06}{550.89}$$

$$= 0.5098$$

Hence efficiency of Otto cycle is $\boxed{\eta_{Otto} = 50.98\%}$

Question: 9.175P

A Diesel cycle has a compression ratio of 20 and begins its compression at 13 psia and 45°F. The maximum cycle temperature is 1800°F. Utilizing air-standard assumptions, determine the thermal efficiency of this cycle using (*a*) constant specific heats at room temperature and (*b*) variable specific heats.

Step 1 of 9

Given:
Compression ratio of the Diesel cycle, $r = 20$
Initial temperature (before compression), $T_1 = 45^0\text{F}$
Initial pressure (before compression), $P_1 = 13$ psia
Maximum cycle temperature, $T_{max} = 1800^0\text{F}$
Specific heat at constant pressure, $C_p = 0.24$ Btu/lbmR
Specific heat at constant temperature, $C_v = 0.171$ Btu/lbmR
Specific heat ratio, $k = 1.4$

Step 2 of 9

(a)
For Isentropic Compression,

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1}$$
$$= T_1(r)^{k-1}$$
$$= (45+460)(7)^{1.4-1}$$
$$= 1673.8\text{ R}$$

For constant pressure heat addition,

$$V_3 = \frac{T_3}{T_2}V_2$$
$$= \frac{2260}{1673.8}\times V_2$$
$$= 1.35\times V_2$$

Step 3 of 9

For isentropic expansion,

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1}$$
$$= T_3\left(\frac{1.35\times V_2}{V_1}\right)^{k-1}$$
$$= (2260)\left(\frac{1.35}{20}\right)^{1.4-1}$$
$$= 768.8\text{ R}$$

Heat added to the cycle is

$$q_{in} = C_p(T_3 - T_2)$$
$$= (0.24)(2260-1673.8)$$
$$= 140.7\text{ Btu/lbm}$$

Heat rejected from the cycle is

$$q_{out} = C_v(T_4 - T_1)$$
$$= (0.171)(768.8-505)$$
$$= 45.11\text{ Btu/lbm}$$

Step 4 of 9

Efficiency of the cycle is

$$\eta_{Diesel} = 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{45.11}{140.7}$$
$$= 0.6784$$

Hence the efficiency of the diesel cycle is $\boxed{\eta_{Diesel} = 67.84\%}$

Step 5 of 9

(b)
From Ideal gas properties of air, at $T_1\,(505\text{ R})$

$$V_{r1} = 170.82 \text{ and}$$
$$u_1 = 86.06\text{ Btu/lbm}$$

But compression ratio, $r = \frac{V_1}{V_2} = \frac{V_{r1}}{V_{r2}}$ Hence

$$V_{r2} = \frac{V_{r1}}{r}$$
$$= \frac{170.82}{20}$$
$$= 8.541$$

Step 6 of 9

From ideal gas properties table of air, the internal energy corresponding to V_{r2} is

$$T_2 = 1582.3\text{ R}$$
$$h_2 = 391.01\text{ Btu/lbm}$$

Temperature after constant pressure heat addition is

$$T_3 = 2260\text{ R}$$

Hence volume after constant pressure heat addition is

$$V_3 = \left(\frac{T_3}{T_2}\right)V_2$$
$$= \left(\frac{2260}{1582.3}\right)V_2$$
$$= 1.428\times V_2$$

Step 7 of 9

From ideal gas properties table of air at T_3,

$$V_{r3} = 2.922 \text{ and}$$
$$h_3 = 577.52\text{ Btu/lbm}$$

Hence amount of heat added is

$$q_{in} = h_3 - h_2$$
$$= 577.52 - 86.06$$
$$= 186.51\text{ Btu/lbm}$$

Step 8 of 9

For isentropic expansion process,

$$V_{r4} = \left(\frac{V_4}{V_3}\right)V_{r3}$$
$$= \left(\frac{V_1}{1.428\times V_2}\right)V_{r3}$$
$$= \left(\frac{20}{1.428}\right)2.922$$
$$= 40.92$$

From ideal gas properties table of air, the internal energy corresponding to V_{r4} is

$$u_4 = 152.65\text{ Btu/lbm}$$

Hence amount of heat rejected is

$$q_{out} = u_4 - u_1$$
$$= 152.65 - 86.06$$
$$= 66.59\text{ Btu/lbm}$$

Step 9 of 9

Efficiency of the cycle,

$$\eta_{Diesel} = 1 - \frac{q_{out}}{q_{in}}$$
$$= 1 - \frac{66.59}{186.51}$$
$$= 0.643$$

Hence efficiency of Diesel cycle is $\boxed{\eta_{Diesel} = 64.3\%}$

Question: 9.176P

A Brayton cycle with a pressure ratio of 12 operates with air entering the compressor at 13 psia and 20°F, and the turbine at 1000°F. Calculate the net specific work produced by this cycle treating the air as an ideal gas with (*a*) constant specific heats at room temperature and (*b*) variable specific heats.

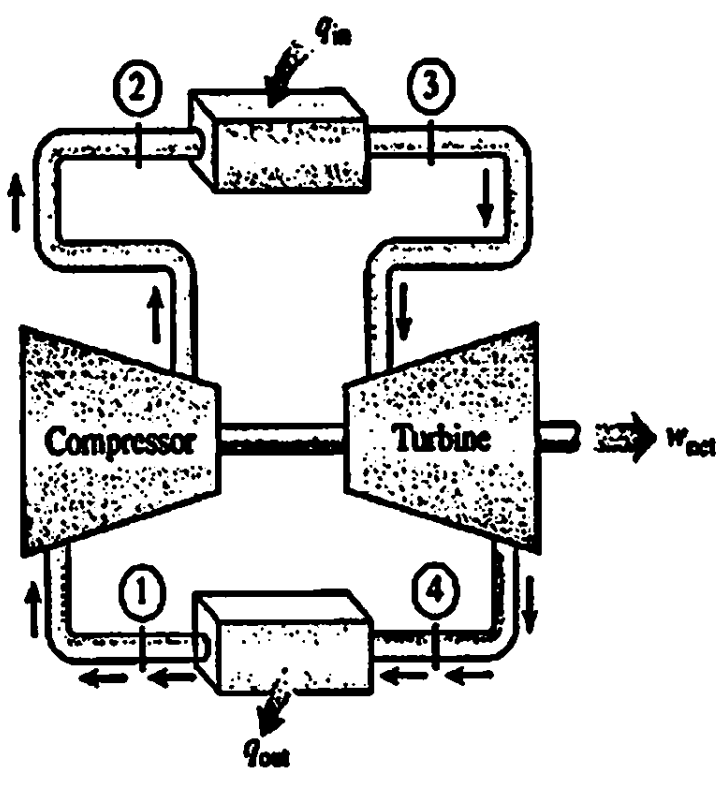


FIGURE P9–176E

Step 1 of 5

Obtain the properties of air at room temperature from properties of air table

Specific heat of air at constant pressure $c_p = 0.24\,\text{Btu/lbm}\cdot\text{R}$

Ratio of specific heats $k = 1.4$

(a)

Find the net specific work by using constant specific heats:

Find the exit temperature of the compressor by using isentropic relation of the process 1- 2

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Here, T_1, T_2 are the temperatures of compressor inlet, exit and P_1, P_2 are the pressures of compressor inlet, exit.

Substitute pressure ratio, $\frac{P_2}{P_1} = 12$ and $20°\text{F} = 480\,\text{R}$ for T_1

$$\frac{T_2}{480} = (12)^{\frac{1.4-1}{1.4}}$$
$$T_2 = 976.28\,\text{R}$$

Step 2 of 5

Find the exit temperature of turbine by using isentropic expansion of the turbine of the process 3-4

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}}$$
$$= \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \qquad (\text{since } P_3 = P_2, P_4 = P_1)$$

Here, a point 3 and 4 indicates turbine inlet and exit respectively.

Find T_4 by substituting $1000°\text{F} = 1460\,\text{R}$ for T_3

$$\frac{1460}{T_4} = 12^{\frac{1.4-1}{1.4}}$$
$$T_4 = 717.81\,\text{R}$$

Step 3 of 5

Find the specific work output by the Brayton cycle by using the expression,

$$w_{net} = w_{turbine} - w_{compressor}$$
$$= c_p(T_3 - T_4) - c_p(T_2 - T_1)$$
$$= 0.24(1460 - 717.81) - 0.24(976.28 - 480)$$
$$= 178.125 - 119.107$$
$$= 59.018\,\text{Btu/lbm}$$

Therefore, net specific work produced by the cycle by using constant specific heats is **59.018 Btu/lbm**

Step 4 of 5

(b)

Find the net specific work by using variable specific heats:

Obtain the properties of air at compressor inlet temperature at $T_1 = 480\,\text{R}$ from table "Ideal gas properties of air"

Enthalpy $h_1 = 114.69\,\text{Btu/lbm}$

Relative pressure $P_{r1} = 0.9182$

Similarly obtain the properties of air at turbine inlet temperature at $T_3 = 1460\,\text{R}$

Specific enthalpy

$$h_3 = 353.37 + (363.89 - 353.37)\left(\frac{1460 - 1440}{1480 - 1440}\right)$$
$$= 358.63\,\text{Btu/lbm}$$

Relative pressure

$$P_{r3} = 47.74 + (53.04 - 47.75)\left(\frac{1460 - 1440}{1480 - 1440}\right)$$
$$= 50.385$$

Step 5 of 5

Find the relative pressure at compressor exit by using the relation,

$$P_{r2} = \frac{P_2}{P_1} P_{r1}$$
$$= 12 \times 0.9182$$
$$= 11.0184$$

Find the enthalpy at $P_{r2} = 11.0184$ from ideal gas properties of air table,

$$h_2 = 231.06 + (236.02 - 231.06)\left(\frac{11.0184 - 10.61}{11.43 - 10.61}\right)$$
$$= 233.53\,\text{Btu/lbm}$$

Find the relative pressure at turbine exit by using the relation,

$$P_{r4} = \frac{P_4}{P_3} P_{r3}$$
$$= \frac{P_1}{P_2} P_{r3}$$
$$= \frac{1}{12} \times 50.385$$
$$= 4.1987$$

Find the enthalpy at $P_{r4} = 4.1987$ from ideal gas properties of air table,

$$h_4 = 177.23 + (182.08 - 177.23)\left(\frac{4.1987 - 4.193}{4.607 - 4.193}\right)$$
$$= 177.296\,\text{Btu/lbm}$$

Find the specific work output by the Brayton cycle by using the expression,

$$w_{net} = w_{turbine} - w_{compressor}$$
$$= (h_3 - h_4) - (h_2 - h_1)$$
$$= (358.63 - 177.296) - (233.53 - 114.69)$$
$$= 181.361 - 118.84$$
$$= 62.521\,\text{Btu/lbm}$$

Therefore, net specific work produced by the cycle by using variable specific heats is **62.521 Btu/lbm**

Question: 9.177P

A four-stroke turbocharged V-16 diesel engine built by GE Transportation Systems to power fast trains produces 3500 hp at 1200 rpm. Determine the amount of power produced per cylinder per (*a*) mechanical cycle and (*b*) thermodynamic cycle.

Step 1 of 3

Given

Power produced $P = 3500\,\text{hp}$

Speed of train $N = 1200\,\text{rpm}$

Number of cylinders $n = 16$

Step 2 of 3

(a) Mechanical cycle

$$P_{mech} = \frac{\text{Total power}}{(\text{Number of cylinders})(\text{Number of mechanical cycles})}$$

$$= \frac{3500 \times 746}{(16)\left(\frac{1200}{60}\right)} \qquad (\text{Since } 1\text{hp} = 746 \text{ watts})$$

$$= 8159.375\,\text{J/cycle}$$

$$\boxed{P_{mech} = 8.16\,\text{kJ/cycle}}$$

Step 3 of 3

(b) Thermodynamic cycle

$$P_{ther} = \frac{\text{Total power}}{(\text{Number of cylinders})(\text{Number of thermodynamic cycles})}$$

$$= \frac{3500 \times 746}{(16)\left(\frac{1200}{2 \times 60}\right)}$$

$$= 16318.75\,\text{J/cycle}$$

$$\boxed{P_{ther} = 16.318\,\text{kJ/cycle}}$$

Question: 9.178P

Consider a simple ideal Brayton cycle operating between the temperature limits of 300 and 1500 K. Using constant specific heats at room temperature, determine the pressure ratio for which the compressor and the turbine exit temperatures of air are equal.

Step 1 of 4

Given

Initial temperature $T_1 = 300\text{K}$

Maximum temperature $T_3 = 1500\text{K}$

Considering $k = 1.4$ (specific heat ratio)

Step 2 of 4

During the process isentropic compression (1-2)

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$\frac{T_2}{T_1} = \left(r_p\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1\left(r_p\right)^{\frac{k-1}{k}} \text{ ---------- (1)}$$

Step 3 of 4

Similarly during the process isentropic expansion (3-4)

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

Since $\frac{P_2}{P_1} = \frac{P_3}{P_4} = r_p$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$\frac{T_4}{T_3} = \left(\frac{1}{r_p}\right)^{\frac{k-1}{k}}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{\frac{k-1}{k}} \text{ ---------- (2)}$$

Step 4 of 4

If compressor and the turbine exit temperatures of air are equal then

$T_2 = T_4$

Therefore from expressions (1) and (2)

$$T_1\left(r_p\right)^{\frac{k-1}{k}} = T_3\left(\frac{1}{r_p}\right)^{\frac{k-1}{k}}$$

$$\frac{T_3}{T_1} = \left(r_p\right)^{\frac{2(k-1)}{k}}$$

$$r_p = \left(\frac{T_3}{T_1}\right)^{\frac{k}{2(k-1)}}$$

$$r_p = \left(\frac{1500}{300}\right)^{\frac{1.4}{2(1.4-1)}}$$

$$\boxed{r_p = 16.718}$$

Question: 9.179P

A four-cylinder spark-ignition engine has a compression ratio of 10.5, and each cylinder has a maximum volume of 0.4 L. At the beginning of the compression process, the air is at 98 kPa and 37°C, and the maximum temperature in the cycle is 2100 K. Assuming the engine to operate on the ideal Otto cycle, determine (*a*) the amount of heat supplied per cylinder, (*b*) the thermal efficiency, and (*c*) the number of revolutions per minute required for a net power output of 45 kW. Assume variable specific heats for air.

Step 1 of 4

Obtain the properties of air at compressor inlet temperature $T_1 = 37°C = 310\,K$ from ideal gas properties of air.

Specific internal energy $u_1 = 221.25\,kJ/kg$

Relative volume $v_{r1} = 572.3$

Find the relative volume at state 2 by applying the isentropic relation during the compression process 1-2

$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1}$$

Here, $\frac{v_1}{v_2}$ is compression ratio

Substitute 10.5 for $\frac{v_1}{v_2}$

$$\frac{v_{r2}}{v_{r1}} = \frac{1}{\frac{v_1}{v_2}}$$

$$\frac{v_{r2}}{572.3} = \frac{1}{10.5}$$

$$v_{r2} = 54.50$$

Step 2 of 4

Obtain the internal energy at state 2 at $v_{r2} = 54.50$ from ideal gas properties of air

$$u_2 = 560.01 + (576.12 - 560.01)\left(\frac{54.50 - 55.54}{51.64 - 55.54}\right)$$

$$= 564.306\,kJ/kg$$

Similarly obtain the properties of air at maximum temperature in the cycle $T_3 = 2100\,K$

Specific internal energy $u_3 = 1775.3\,kJ/kg$

Relative volume $v_{r3} = 2.356$

By using the isentropic relation during the expansion process 3-4, find the relative volume v_{r4}

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3}$$

$$\frac{v_{r4}}{v_{r3}} = \frac{v_1}{v_2} \qquad (\text{since } v_4 = v_1 \text{ and } v_3 = v_2)$$

$$\frac{v_{r4}}{2.356} = 10.5$$

$$v_{r4} = 24.738$$

Obtain the internal energy at state 4 at $v_{r4} = 24.738$ from ideal gas properties of air

$$u_4 = 758.94 + (776.10 - 758.94)\left(\frac{24.738 - 25.17}{23.72 - 25.17}\right)$$

$$= 764.052\,kJ/kg$$

Find mass of air by using ideal gas equation,

$m = \frac{P_1 V_1}{R T_1}$ Here, P_1 is initial pressure, V_1 is initial volume, R is gas constant and T_1 is initial temperature

Substitute 98 kPa for P_1, $0.4 \times 10^3\,m^3$ for V_1, $0.287\,kJ/kg \cdot K$ for R and $310\,K$ for T_1

$$m = \frac{98 \times 0.4 \times 10^{-3}}{0.287 \times 310}$$

$$= 0.00044\,kg$$

Step 3 of 4

(a) Find the amount of heat supplied per cylinder by using the equation,

$$Q_{2-3} = m(u_3 - u_2)$$

$$= 0.00044(1775.3 - 564.306)$$

$$= 0.5328\,kJ$$

Therefore, the amount of heat supplied per cylinder is $\boxed{0.5328\,kJ}$

Step 4 of 4

(b)

Find the amount of heat rejection during the cycle by using the equation,

$$Q_{4-1} = m(u_4 - u_1)$$

$$= 0.00044(764.052 - 221.25)$$

$$= 0.2388\,kJ$$

Find the net work done by using the equation,

$$W_{net} = Q_{2-3} - Q_{4-1}$$

$$= 0.5328 - 0.2388$$

$$= 0.294\,kJ$$

Find the thermal efficiency of the cycle by using the equation,

$$\eta = \frac{W_{net}}{Q_{2-3}}$$

$$= \frac{0.294}{0.5328}$$

$$= 0.5518$$

$$= 53.18\%$$

Therefore, thermal efficiency of the cycle is $\boxed{53.18\%}$

(c)

Find the number of revolutions per minute by using the equation,

$$\dot{n} = 2\frac{\dot{W}_{net}}{n_{cyl} W_{net}}$$

Here, $\dot{W}_{net}$ is net power output, n_{cyl} is number of cylinders

Substitute 45 kW for $\dot{W}_{net}$ and 4 for n_{cyl}

$$\dot{n} = 2 \times \frac{45}{4 \times 0.294}$$

$$= 76.53\,rps$$

$$= 76.53 \times 60$$

$$= 4591.836\,rpm$$

Therefore, the number of revolutions per minute is $\boxed{4591.836\,rpm}$

Question: 9.180P

Reconsider Prob. 9–179. Using EES (or other) software, study the effect of varying the compression ratio from 5 to 11 on the net work done and the efficiency of the cycle. Plot the *P-v* and *T-s* diagrams for the cycle, and discuss the results.

Step 1 of 3

Now to solve the equation for a range of compression ratio, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.

A dailog box will open. Now, select "No. of Runs" = 7

Select "variables in equations" as ratio , Eta_th and w_net

Press 'Add' and then press 'ok'

A table will appear and enter the values of compression ration 'ratio' as 5, 6, 7, 8, 9, 10, and 11. Then click on 'Solve table' of 'Calculate' menu.

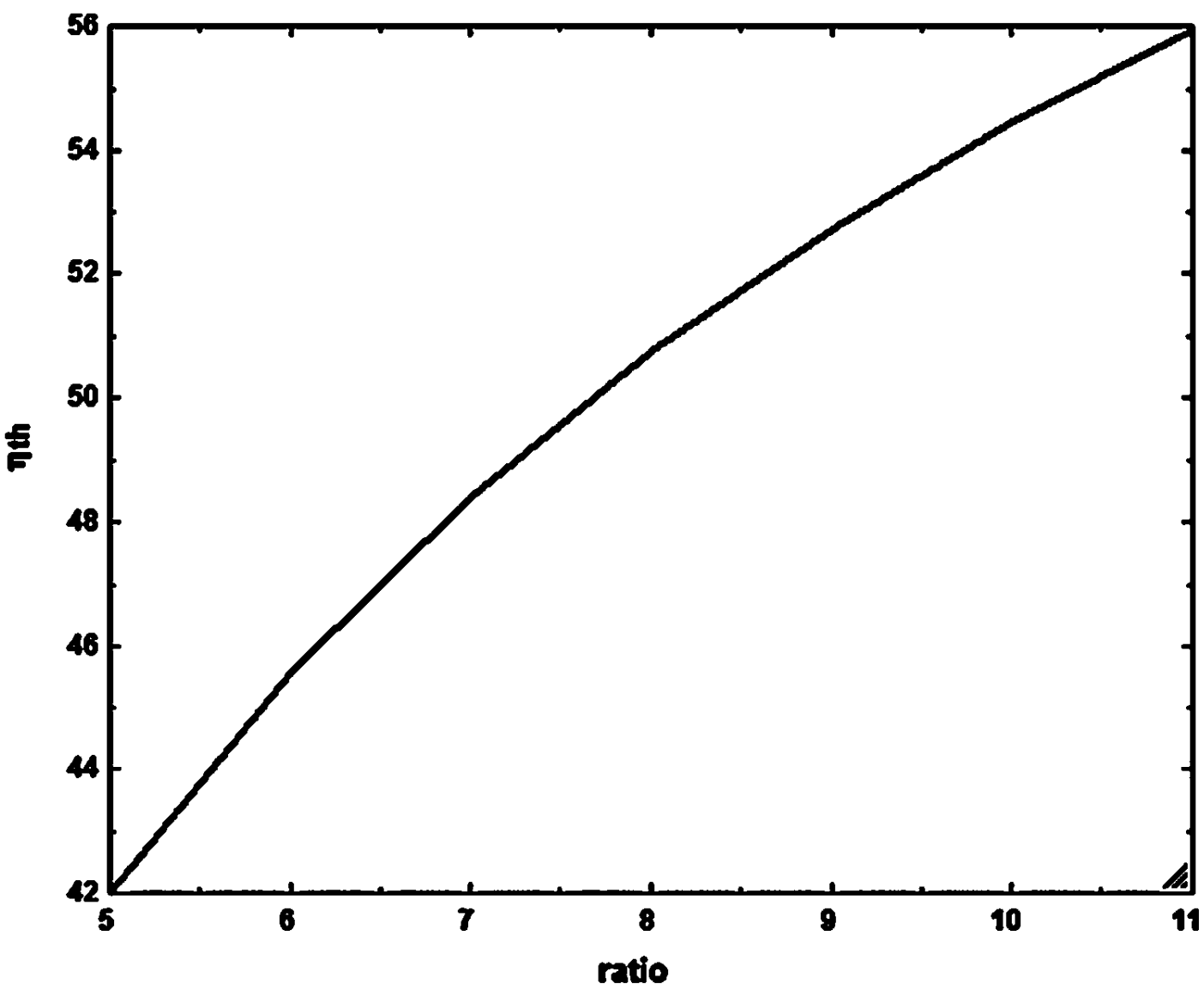
The Solution will appear as follows:

1..7	1 ratio	2 η_{th}	3 w_{net}
Run 1	5	42.01	568.3
Run 2	6	45.56	602
Run 3	7	48.4	625.8
Run 4	8	50.75	643
Run 5	9	52.73	655.5
Run 6	10	54.45	664.7
Run 7	11	55.95	671.3

In order to find the variation of thermal efficiency with compression ratio, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as ratio and Y-axis as Eta_th. Press OK

The Graph will be plotted as follows:



Step 2 of 3

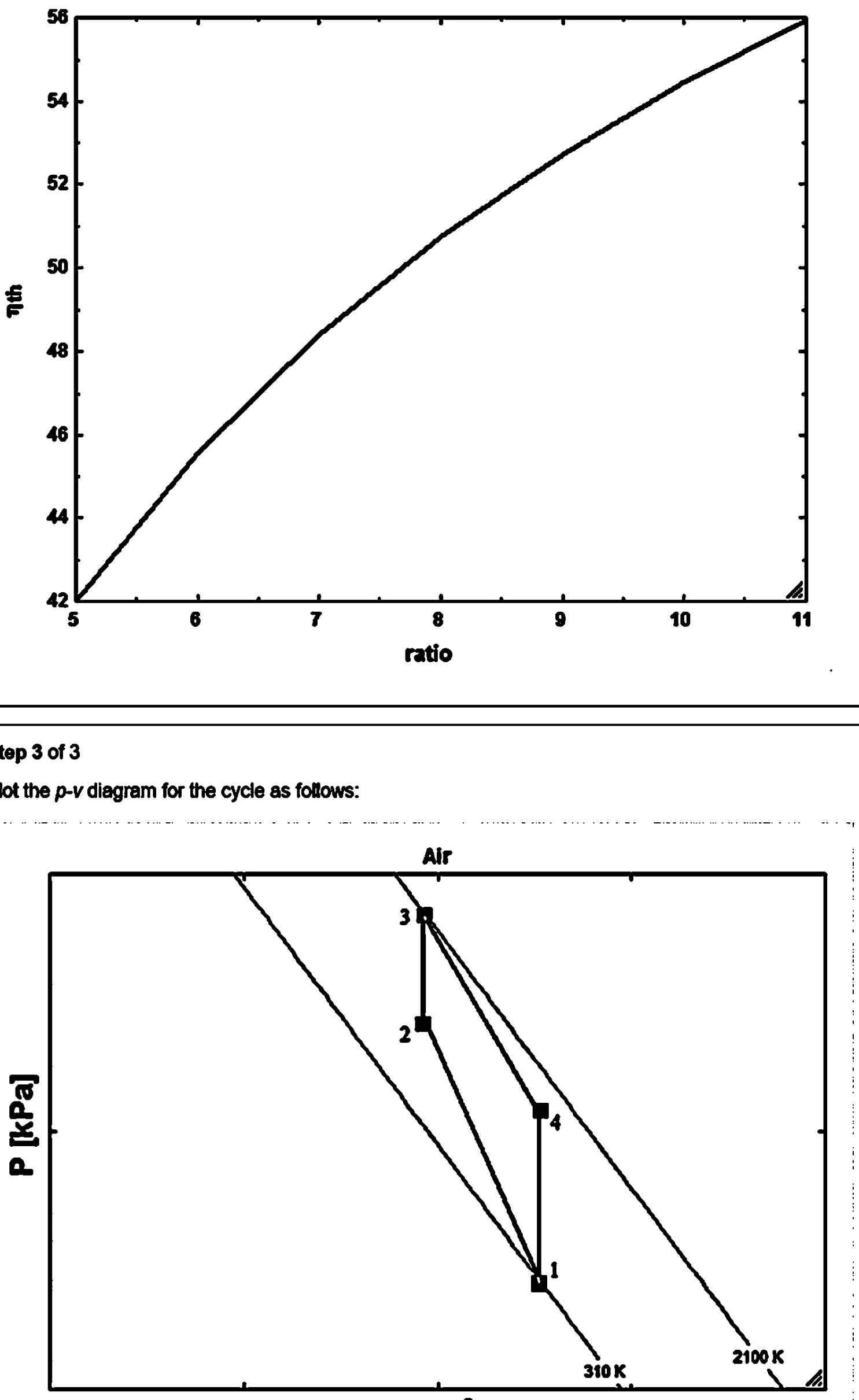
Inorder to find the variation of thermal efficiency with compression, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.

Select X-axis as ratio and Y-axis as Eta_th. Press OK

The Graph will be plotted as follows:

Step 3 of 3

Plot the *p-v* diagram for the cycle as follows:



Plot the *T-s* diagram for the cycle as follows:

Question: 9.181P

An ideal gas Carnot cycle uses helium as the working fluid and rejects heat to a lake at 15°C. Determine the pressure ratio, compression ratio, and minimum temperature of the heat source for this cycle to have a thermal efficiency of 50 percent.

Step 1 of 4

Given

Thermal efficiency $\eta_{th} = 0.5$

Low temperature reservoir $T_L = 15°C = 15+273 = 288K$

Step 2 of 4

The efficiency of a Carnot cycle is defined as

$$\eta = 1 - \frac{T_{min}}{T_{max}}$$

$$\eta = 1 - \frac{T_L}{T_H}$$

$$0.5 = 1 - \frac{288}{T_H}$$

$$T_H = 576K$$

Temperature of heat source

$$\boxed{T_H = 576K}$$

Step 3 of 4

The pressure ratio is defined as

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{T_H}{T_L}\right)^{\frac{k}{k-1}}$$

Specific heat ratio for helium is taking from ideal gas specific heats of various common gases table

$k = 1.667$

$$\frac{P_2}{P_1} = \left(\frac{576}{288}\right)^{\frac{1.667}{1.667-1}}$$

$$= 5.6539$$

Pressure ratio

$$\boxed{\frac{P_2}{P_1} = 5.6539}$$

Step 4 of 4

Compression ratio

$$\frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{1/k}$$

$$\frac{v_1}{v_2} = (5.6539)^{1/1.667}$$

$$= 2.8269$$

Compression ratio

$$\boxed{\frac{v_1}{v_2} = 2.83}$$

Question: 9.182P

Repeat Prob. 9–181 when the lake is at 60°F and the Carnot cycle's thermal efficiency is to be 60 percent

Step 1 of 4

Given

Thermal efficiency $\eta_{th} = 0.6$

Low temperature reservoir $T_L = 60°F = 60+460 = 520R$

Step 2 of 4

The efficiency of a Carnot cycle is defined as

$$\eta = 1 - \frac{T_{min}}{T_{max}}$$

$$\eta = 1 - \frac{T_L}{T_H}$$

$$0.6 = 1 - \frac{520}{T_H}$$

$$T_H = 1300R$$

Temperature of heat source

$$\boxed{T_H = 1300R}$$

Step 3 of 4

The pressure ratio is defined as

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}}$$

Specific heat ratio for helium is taking from ideal gas specific heats of various common gases table

$k = 1.667$

$$\frac{P_2}{P_1} = \left(\frac{1300}{520}\right)^{\frac{1.667}{1.667-1}}$$

$$\boxed{\frac{P_2}{P_1} = 9.875}$$

Step 4 of 4

Compression ratio

$$\frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{1/k}$$

$$\frac{v_1}{v_2} = (9.875)^{1/1.667}$$

$$\frac{v_1}{v_2} = 3.95$$

Compression ratio

$$\boxed{\frac{v_1}{v_2} = 3.95}$$

Question: 9.183P

A typical hydrocarbon fuel produces 43,000 kJ/kg of heat when used in a spark-ignition engine. Determine the compression ratio required for an ideal Otto cycle to use 0.035 grams of fuel to produce 1 kJ of work. Use constant specific heats at room temperature.

Step 1 of 2

Determine the heat energy of hydrocarbon fuel by using the equation,

$$Q = mq$$

Here, m is mass of fuel and q is specific energy of hydrocarbon fuel.

Substitute $0.035\,\text{grams} = 0.035\times10^{-3}\,\text{kg}$ for m and $43,000\,\text{kJ/kg}$ for q

$$\begin{aligned} Q &= 0.035\times10^{-3}\times43,000 \\ &= 1.505\,\text{kJ} \end{aligned}$$

Step 2 of 2

Write the thermal efficiency of a heat engine,

$$\eta = \frac{W}{Q_{in}} \quad \text{..... (1)}$$

Here, W is work output from the engine and Q_{in} is heat supplied to the engine

Also, write the thermal efficiency of an ideal Otto cycle in terms of compression ratio,

$$\eta = 1 - \frac{1}{r^{k-1}} \quad \text{..... (2)}$$

Here, r is compression ratio and k is specific heat ratio.

Equate equations (1) and (2)

$$\eta = \frac{W}{Q_{in}} = 1 - \frac{1}{r^{k-1}}$$

$$\frac{W}{Q_{in}} = 1 - \frac{1}{r^{k-1}}$$

Substitute 1 kJ for W and 1.4 for k

$$\frac{1}{1.505} = 1 - \frac{1}{r^{1.4-1}}$$

$$0.664 = 1 - \frac{1}{r^{0.4}}$$

$$0.336 = \frac{1}{r^{0.4}}$$

$$r^{0.4} = 2.976$$

$$r = 15.27$$

Therefore, compression ratio of Otto cycle is $\boxed{15.27}$

Question:

An ideal Otto cycle has a compression ratio of 9.2 and uses air as the working fluid. At the beginning of the compression process, air is at 98 kPa and 27°C. The pressure is doubled during the constant-volume heat-addition process. Accounting for the variation of specific heats with temperature, determine (*a*) the amount of heat transferred to the air,(*b*) the net work output, (*c*) the thermal efficiency, and.(*d*) the mean effective pressure for the cycle.

Step 1 of 16

Given

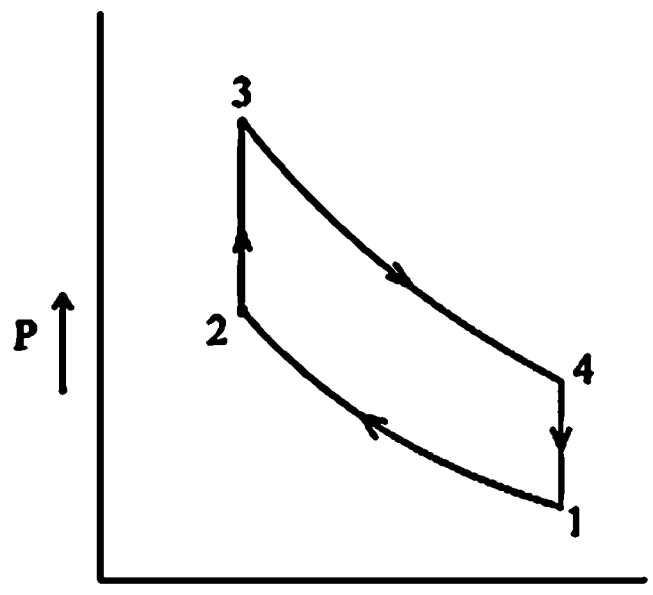
Compression ratio $r = 9.2$

Initial pressure $P_1 = 98\ \text{kPa}$

Initial temperature $T_1 = 27°\text{C} = 27 + 273 = 300\text{K}$

Pressure during the constant volume heat addition process is $P_3 = 2P_2$

Step 2 of 16



Step 3 of 16

a)

Since we are considering variable specific heats the specific heat at constant pressure and specific heat at constant volume will be different at different temperatures.

At $T_1 = 300\text{K}$ from ideal gas properties of air

$u_1 = 214.07\ \text{kJ/kg}$

$v_{r1} = 621.2$

Step 4 of 16

Process 1-2 is isentropic compression process

$$\frac{v_{r_1}}{v_{r_2}} = \frac{v_2}{v_1}$$

$$v_{r2} = v_{r1}\frac{v_2}{v_1}$$

$$= 621.2 \times \frac{1}{9.2}$$

$$v_{r_2} = 67.52$$

Step 5 of 16

At $v_{r_2} = 67.52$ from ideal gas properties of air

$$u_2 = 512.33 + (520.23 - 512.33)\left(\frac{67.52 - 69.76}{67.07 - 69.76}\right)$$

$$u_2 = 518.9\text{kJ/kg}$$

$$T_2 = 700 + (710 - 700)\left(\frac{67.52 - 69.76}{67.07 - 69.76}\right)$$

$$T_2 = 708.32\text{K}$$

Step 6 of 16

Applying ideal gas equation to process 1-2

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

$$P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1$$

$$= (9.2) \times \frac{708.32}{300} \times 98$$

$$P_2 = 2128.737\ \text{kPa}$$

Step 7 of 16

Process 2-3 is constant volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$T_3 = \frac{P_3}{P_2} \times T_2$$

$$T_3 = 2(708.32)$$

$$T_3 = 1416.64\text{K}$$

Step 8 of 16

At $T_3 = 1416.64\text{K}$ from ideal gas properties of air

$$u_3 = 1113.52 + (1131.77 - 1113.52)\left(\frac{1416.64 - 1400}{1420 - 1400}\right)$$

$$u_3 = 1128.704\ \text{kJ/kg}$$

$$v_{r_3} = 8.919 + (8.526 - 8.919)\left(\frac{1416.64 - 1400}{1420 - 1400}\right)$$

$$v_{r3} = 8.592$$

Step 9 of 16

(a) The amount of heat transferred to the air is

$$q_{in} = (u_3 - u_2)\ \text{kJ/kg}$$

$$= (1128.704 - 518.9)$$

(a) The amount of heat transferred to the air is

$$q_{in} = (u_3 - u_2) \text{ kJ/kg}$$
$$= (1128.704 - 518.9)$$
$$\boxed{q_{in} = 609.804 \text{ kJ/kg}}$$

Step 10 of 16

(b) The net work output

Process 3-4 is isentropic expansion

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3}$$
$$v_{r4} = \frac{v_4}{v_3} v_{r3}$$
$$v_{r4} = \frac{v_1}{v_2} v_{r3}$$
$$= 9.2(8.592)$$
$$v_{r_4} = 79.0464$$

Step 11 of 16

At $v_{r_4} = 79.0464$ from ideal gas properties of air

$$u_4 = 481.01 + (488.81 - 481.01)\left(\frac{79.0464 - 81.89}{78.61 - 81.89}\right)$$
$$u_4 = 487.77 \text{ kJ/kg}$$

Step 12 of 16

The amount of heat rejection is expressed as

$$q_{out} = (u_4 - u_1) \text{ kJ/kg}$$
$$= (487.77 - 214.07)$$
$$q_{out} = 273.7 \text{ kJ/kg}$$

Step 13 of 16

The net work output

$$w_{net} = q_{in} - q_{out}$$
$$= 609.804 - 273.7$$
$$\boxed{w_{net} = 336.104 \text{ kJ/kg}}$$

Step 14 of 16

(c) The thermal efficiency

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$
$$= \frac{336.104}{609.804}$$
$$= 0.551$$
$$\boxed{\eta_{th} = 55.1\%}$$

Step 15 of 16

(d) The mean effective pressure for the cycle is

$$MEP = \frac{w_{net}}{v_1 - v_2}$$
$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{v_2}{v_1}\right)}$$

Where v_1 can be calculated by using ideal gas equation

$$v_1 = \frac{RT_1}{P_1}$$
$$= \frac{(0.287)(300)}{98}$$
$$v_1 = 0.8785 \text{ m}^3/\text{kg}$$

Step 16 of 16

Mean effective pressure is

$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{v_2}{v_1}\right)}$$
$$MEP = \frac{336.104}{0.8785\left(1 - \frac{1}{9.2}\right)}$$
$$\boxed{MEP = 429.245 \text{ kPa}}$$

Question: 9.185P pg 1

Repeat Prob. 9–184 using constant specific heats at room temperature.

Step 1 of 13

Given

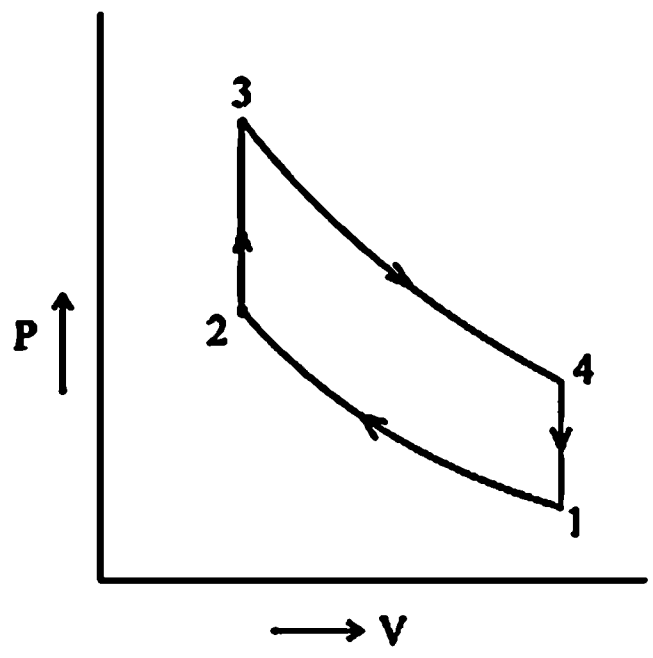
Compression ratio $r = 9.2$

Initial pressure $P_1 = 98\ \text{kPa}$

Initial temperature $T_1 = 27°\text{C} = 27 + 273 = 300\text{K}$

Pressure during the constant volume heat addition process is $P_3 = 2P_2$

Step 2 of 13



Step 3 of 13

a) The amount of heat supplied

Considering constant specific heats at room temperature

From table ideal gas specific heats of various common gases

Gas constant $R = 0.287\ \text{kJ/kg.K}$

$c_p = 1.005\ \text{kJ/kg.K}$

$c_v = 0.718\ \text{kJ/kg.K}$

$k = 1.4\,(\text{specific heat ratio})$

Step 4 of 13

Process 1-2 is isentropic compression

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 300(9.2)^{1.4-1}$$

$$T_2 = 728.847\text{K}$$

Step 5 of 13

Applying ideal gas equation to process 1-2

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

$$P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1$$

$$= (9.2)\frac{728.847}{300} \times 98$$

$$P_2 = 2190.428\ \text{kPa}$$

Step 6 of 13

Process 2-3 is constant volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$T_3 = \frac{P_3}{P_2} \times T_2$$

$$T_3 = 2(728.847)$$

$$\frac{T_2}{T_1} = r^{k-1}$$

$$T_2 = 300(9.2)^{1.4-1}$$

$$T_2 = 728.847\text{K}$$

Step 5 of 13

Applying ideal gas equation to process 1-2

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

$$P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1$$

$$= (9.2)\frac{728.847}{300} \times 98$$

$$P_2 = 2190.428\ \text{kPa}$$

Step 6 of 13

Process 2-3 is constant volume heat addition

$$\frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$T_3 = \frac{P_3}{P_2} \times T_2$$

$$T_3 = 2(728.847)$$

$$\boxed{q_{in} = 523.312\ \text{kJ/kg}}$$

Step 9 of 13

(b) The net work output

The amount of heat rejection during the process 4-1 is

$$q_{out} = (u_4 - u_1)\ \text{kJ/kg}$$

Step 9 of 13

(b) The net work output

The amount of heat rejection during the process 4-1 is

$$q_{out} = (u_4 - u_1)\ \text{kJ/kg}$$

$$= c_v (T_4 - T_1)$$

$$= 0.718(600 - 300)$$

$$q_{out} = 215.4\ \text{kJ/kg}$$

Step 10 of 13

The net work output

$$w_{net} = q_{in} - q_{out}$$

$$= 523.312 - 215.4$$

$$\boxed{w_{net} = 307.912\ \text{kJ/kg}}$$

Step 11 of 13

c)

The thermal efficiency is defined as

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{307.912}{523.312}$$

$$= 0.588$$

$$\boxed{\eta_{th} = 58.8\%}$$

Step 12 of 13

(d) Mean effective pressure

We know that

$$MEP = \frac{w_{net}}{v_1 - v_2}$$

$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{v_2}{v_1}\right)}$$

Where v_1 can be calculated by using ideal gas equation

$$v_1 = \frac{RT_1}{P_1}$$

$$= \frac{(0.287)(300)}{98}$$

$$v_1 = 0.8785\ \text{m}^3/\text{kg}$$

Step 13 of 13

Mean effective pressure is

$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{v_2}{v_1}\right)}$$

$$MEP = \frac{w_{net}}{v_1\left(1 - \frac{1}{r}\right)}$$

$$= \frac{307.912}{0.8785\left(1 - \frac{1}{9.2}\right)}$$

$$= 393.062\text{kPa}$$

$$\boxed{MEP = 393.062\ \text{kPa}}$$

Question: 9.186P

An ideal dual cycle has a compression ratio of 14 and uses air as the working fluid. At the beginning of the compression process, air is at 14.7 psia and 120°F, and occupies a volume of 98 in 3. During the heat-addition process, 0.6 Btu of heat is transferred to air at constant volume and 1.1 Btu at constant pressure. Using constant specific heats evaluated at room temperature, determine the thermal efficiency of the cycle.

Step 1 of 7

Obtain the properties of air at room temperature from table "Properties of ideal gases"

Gas constant of air $R = 0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$

Specific heat at constant pressure $c_p = 0.24\,\text{Btu/lbm}\cdot\text{R}$

Specific heat at constant volume $c_v = 0.171\,\text{Btu/lbm}\cdot\text{R}$

Specific heat ratio, $k = 1.4$

Draw the different processes of Dual cycle on *T-s* diagram as shown below:

Question:

An ideal dual cycle has a compression ratio of 14 and uses air as the working fluid. At the beginning of the compression process, air is at 14.7 psia and 120°F, and occupies a volume of 98 in 3. During the heat-addition process, 0.6 Btu of heat is transferred to air at constant volume and 1.1 Btu at constant pressure. Using constant specific heats evaluated at room temperature, determine the thermal efficiency of the cycle.

Step 1 of 7

Obtain the properties of air at room temperature from table "Properties of ideal gases"

Gas constant of air $R = 0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}$

Specific heat at constant pressure $c_p = 0.24\,\text{Btu/lbm}\cdot\text{R}$

Specific heat at constant volume $c_v = 0.171\,\text{Btu/lbm}\cdot\text{R}$

Specific heat ratio, $k = 1.4$

Draw the different processes of Dual cycle on *T-s* diagram as shown below:

Here, $\frac{V_1}{V_2}$ is compression ratio.

Substitute 14 for $\frac{V_1}{V_2}$

$$\frac{T_2}{580} = 14^{1.4-1}$$
$$T_2 = 1666.78\,\text{R}$$

Step 4 of 7

Find the temperature at state 3 after constant volume heat addition process,

$$Q_{2-3} = mc_v(T_3 - T_2)$$

Here, Q_{2-3} heat addition in constant volume process

Substitute 0.6 Btu for Q_{2-3}

$$0.6 = 0.00388 \times 0.171 \times (T_3 - 1666.78)$$
$$T_3 = 2571.10\,\text{R}$$

Step 5 of 7

Find the temperature at state 4 after constant pressure heat addition process,

Here, $\frac{V_1}{V_2}$ is compression ratio.

Substitute 14 for $\frac{V_1}{V_2}$

$$\frac{T_2}{580} = 14^{1.4-1}$$
$$T_2 = 1666.78\,\text{R}$$

Step 4 of 7

Find the temperature at state 3 after constant volume heat addition process,

$$Q_{2-3} = mc_v(T_3 - T_2)$$

Here, Q_{2-3} heat addition in constant volume process

Substitute 0.6 Btu for Q_{2-3}

$$0.6 = 0.00388 \times 0.171 \times (T_3 - 1666.78)$$
$$T_3 = 2571.10\,\text{R}$$

Step 5 of 7

Find the temperature at state 4 after constant pressure heat addition process,

$$Q_{3-4} = mc_p(T_4 - T_3)$$

Here, Q_{3-4} heat addition in constant volume process

Substitute 1.1 Btu for Q_{3-4}

$$1.1 = 0.00388 \times 0.24 \times (T_4 - 2571.10)$$
$$T_4 = 3752.37\,\text{R}$$

Find the temperature at state 5 by using the isentropic relation,

$$\frac{T_4}{T_5} = \left(\frac{V_5}{V_4}\right)^{k-1}$$
$$= \left(\frac{V_1}{V_4}\right)^{k-1} \qquad (\text{Since } V_5 = V_1)$$
$$= \left(\frac{V_1}{V_2} \times \frac{V_2}{V_4}\right)^{k-1}$$
$$= \left(\frac{V_1}{V_2} \times \frac{V_3}{V_4}\right)^{k-1} \qquad (\text{Since } V_2 = V_3)$$

Substitute the relation $\frac{V_3}{V_4} = \frac{T_3}{T_4}$, since the process 3-4 is constant pressure heat addition process.

$$\frac{T_4}{T_5} = \left(\frac{V_1}{V_4} \times \frac{T_3}{T_4}\right)^{k-1}$$
$$= \frac{Q_{2-3} + Q_{3-4} - Q_{5-1}}{Q_{2-3} + Q_{3,4}}$$
$$= \frac{0.6 + 1.1 - 0.6229}{0.6 + 1.1}$$
$$= 0.6335$$

Therefore, thermal efficiency of the ideal dual cycle is **63.35%**

$$= \frac{Q_{2-3} + Q_{3-4} - Q_{5-1}}{Q_{2-3} + Q_{3,4}}$$
$$= \frac{0.6 + 1.1 - 0.6229}{0.6 + 1.1}$$
$$= 0.6335$$

Therefore, thermal efficiency of the ideal dual cycle is **63.35%**

Question:

Consider a simple ideal Brayton cycle with air as the working fluid. The pressure ratio of the cycle is 6, and the minimum and maximum temperatures are 300 and 1300 K, respectively. Now the pressure ratio is doubled without changing the minimum and maximum temperatures in the cycle. Determine the change in (*a*) the net work output per unit mass and (*b*) the thermal efficiency of the cycle as a result of this modification. Assume variable specific heats for air.

Step 1 of 18

Given Data:

Pressure ratio of the cycle, $r_p = 6$

The minimum temperature of the cycle, $T_1 = 300\ \text{K}$

The maximum temperature of the cycle, $T_3 = 1300\ \text{K}$

Step 2 of 18

The *T*-*s* diagram for the given ideal Brayton cycle is shown:

Question:

Consider a simple ideal Brayton cycle with air as the working fluid. The pressure ratio of the cycle is 6, and the minimum and maximum temperatures are 300 and 1300 K, respectively. Now the pressure ratio is doubled without changing the minimum and maximum temperatures in the cycle. Determine the change in (*a*) the net work output per unit mass and (*b*) the thermal efficiency of the cycle as a result of this modification. Assume variable specific heats for air.

Step 1 of 18

Given Data:

Pressure ratio of the cycle, $r_p = 6$

The minimum temperature of the cycle, $T_1 = 300\ \text{K}$

The maximum temperature of the cycle, $T_3 = 1300\ \text{K}$

Step 2 of 18

The *T*-*s* diagram for the given ideal Brayton cycle is shown:

The pressure ratio is given by,

$$r_p = \frac{P_2}{P_1}$$

$$r_p = \frac{P_{r_2}}{P_{r_1}}$$

$$6 = \frac{P_{r_2}}{1.3860}$$

$$P_{r_2} = 8.316$$

Step 5 of 18

By interpolating the values, enthalpy h_2 is calculated as follows:

$$\frac{7.824 - P_{r_2}}{7.824 - 8.411} = \frac{h_{7824} - h_2}{h_{7824} - h_{8.411}}$$

$$\frac{7.824 - 8.316}{7.824 - 8.411} = \frac{492.74 - h_2}{492.74 - 503.02}$$

$$h_2 = 501.36\ \text{kJ/kg}$$

Step 6 of 18

The pressure ratio is given by,

$$\frac{1}{} = \frac{P_4}{}$$

The pressure ratio is given by,

$$r_p = \frac{P_2}{P_1}$$

$$r_p = \frac{P_{r_2}}{P_{r_1}}$$

$$6 = \frac{P_{r_2}}{1.3860}$$

$$P_{r_2} = 8.316$$

Step 5 of 18

By interpolating the values, enthalpy h_2 is calculated as follows:

$$\frac{7.824 - P_{r_2}}{7.824 - 8.411} = \frac{h_{7824} - h_2}{h_{7824} - h_{8.411}}$$

$$\frac{7.824 - 8.316}{7.824 - 8.411} = \frac{492.74 - h_2}{492.74 - 503.02}$$

$$h_2 = 501.36\ \text{kJ/kg}$$

Step 6 of 18

The pressure ratio is given by,

$$\frac{1}{} = \frac{P_4}{}$$

Heat energy out of the system per unit mass is given by,

$$q_{out} = h_4 - h_1$$

$$= 855.27 - 300.19$$

$$= 555.08\ \text{kJ/kg}$$

Step 9 of 18

Work done by the system per unit mass is given by,

$= 555.08 \text{ kJ/kg}$

Step 9 of 18

Work done by the system per unit mass is given by,

$$w_{net} = q_{in} - q_{out}$$
$$= 894.61 - 555.08$$
$$= 339.53 \text{ kJ/kg}$$

Thermal efficiency of the system is given by,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$
$$= \frac{339.53}{894.61}$$
$$= 37.95\%$$

Step 10 of 18

Since, the pressure ratio of the cycle is doubled. Therefore, the pressure ratio of the system, $r_p' = 12$

From ideal gas properties of air tables,

At temperature $T_1 = 300 \text{ K}$ the following values are obtained:

Enthalpy $h_1 = 300.19 \text{ kJ/kg}$, reduced pressure, $P_{r_1} = 1.3860$

At temperature $T_3' = 1300 \text{ K}$ the following values are obtained:

Enthalpy $h_3' = 1395.97 \text{ kJ/kg}$, reduced pressure, $P_{r_3}' = 330.90$

Step 11 of 18

The pressure ratio is given by,

$$r_p' = \frac{P_2'}{P_1}$$

$$r_p' = \frac{P_{r_2}'}{P_{r_1}}$$

$$12 = \frac{P_{r_2}'}{1.3860}$$

$$P_{r_2}' = 16.6320$$

Step 12 of 18

By interpolating the values, enthalpy h_2' is calculated as follows:

$$\frac{16.28 - P_{r_2}'}{16.28 - 17.30} = \frac{h_{16.28} - h_2'}{h_{16.28} - h_{17.30}}$$

$$\frac{16.28 - 16.6320}{16.28 - 17.30} = \frac{607.02 - h_2}{607.02 - 617.53}$$

$$h_2' = 610.65 \text{ kJ/kg}$$

Step 13 of 18

The pressure ratio is given by,

$$\frac{1}{r_p'} = \frac{P_4'}{P_3'}$$

$$\frac{1}{r_p} = \frac{P_{r_4}'}{P_{r_3}'}$$

$$\frac{1}{12} = \frac{P_{r_4}'}{330.90}$$

$$P_{r_4}' = 27.575$$

Step 14 of 18

By interpolating the values, enthalpy h_4' is calculated as follows:

$$\frac{27.29 - P_{r_2}'}{16.28 - 17.30} = \frac{h_{16.28} - h_2'}{h_{16.28} - h_{17.30}}$$

$$\frac{16.28 - 16.6320}{16.28 - 17.30} = \frac{607.02 - h_2}{607.02 - 617.53}$$

$$h_2' = 610.65 \text{ kJ/kg}$$

Step 13 of 18

The pressure ratio is given by,

$$\frac{1}{r_p'} = \frac{P_4'}{P_3'}$$

$$\frac{1}{r_p} = \frac{P_{r_4}'}{P_{r_3}'}$$

$$\frac{1}{12} = \frac{P_{r_4}'}{330.90}$$

$$P_{r_4}' = 27.575$$

Step 14 of 18

By interpolating the values, enthalpy h_4' is calculated as follows:

$27.29 - P_{r_2}'$ h $- h'$

Thermal efficiency of the system is given by,

$$\eta_{th}' = \frac{w_{net}'}{q_{in}'}$$

27.29 – P. 4 –4' 9.187P pg 3

Thermal efficiency of the system is given by,

$$\eta_{th}' = \frac{w_{net}'}{q_{in}'}$$
$$= \frac{380.96}{785.32}$$
$$= 48.51\%$$

Step 17 of 18

(a)

Change in the net work output per unit mass of the cycle due to this modification is given by,

$$\Delta w_{net} = w_{net}' - w_{net}$$
$$= 380.96 - 339.53$$
$$= 41.43 \text{ kJ/kg}$$

Therefore, the change in the net work output per unit mass of the cycle due to this modification, $\boxed{\Delta w_{net} = 41.43 \text{ kJ/kg}}$

Step 18 of 18

(b)

Change in thermal efficiency of the cycle due to this modification is given by,

Thermal efficiency of the system is given by,

$$\eta_{th}' = \frac{w_{net}'}{q_{in}'}$$
$$= \frac{380.96}{785.32}$$
$$= 48.51\%$$

$$\Delta \eta_{th} = \eta_{th}' - \eta_{th}$$
$$= 48.51 - 37.95$$
$$= 10.56\%$$

Therefore, the change in thermal efficiency of the cycle due to this modification, $\boxed{\Delta \eta_{th} = 10.56\%}$

Question: 9.188P pg 1

Repeat Prob. 9–187 using constant specific heats at room temperature.

Step 1 of 13

Given Data:

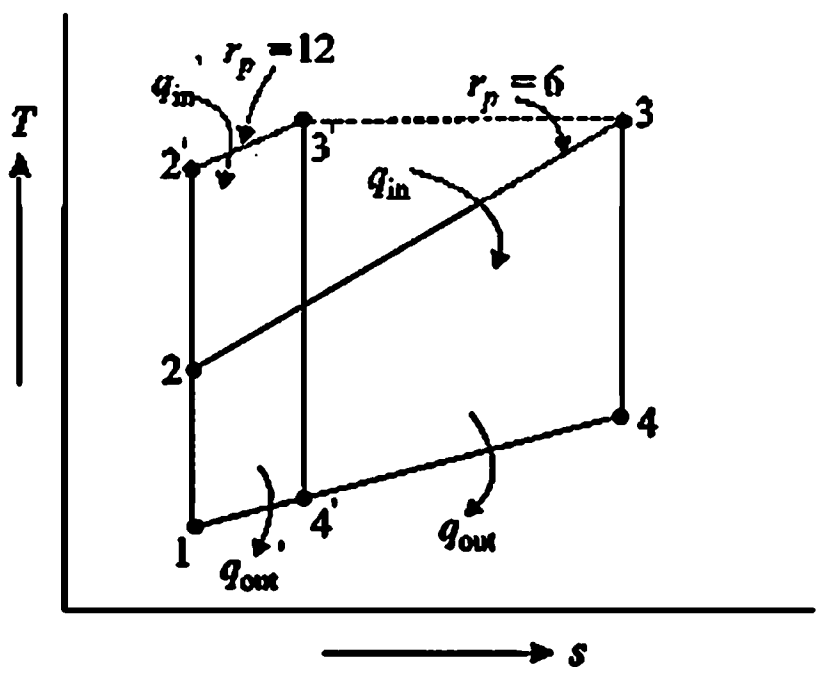
Pressure ratio of the cycle, $r_p = 6$

The minimum temperature of the cycle, $T_1 = 300\ \text{K}$

The maximum temperature of the cycle, $T_3 = 1300\ \text{K}$

Step 2 of 13

The T-s diagram for the given ideal Brayton cycle is shown:



Step 3 of 13

From ideal-gas specific heats of various common gases tables; the properties of air are,

Gas constant, $R = 0.2870\ \text{kJ/kg}\cdot\text{k}$

Specific heat at constant pressure, $c_p = 1.005\ \text{kJ/kg}\cdot\text{k}$

Specific heat at constant volume, $c_v = 0.718\ \text{kJ/kg}\cdot\text{k}$

Specific heat ratio, $k = 1.400$

The pressure ratio is given by,

$$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

Step 4 of 13

Process 1-2:

During the process 1-2, isentropic compression of the gas takes place. Therefore,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$\frac{T_2}{300} = \left(r_p\right)^{(1.4-1)/1.4}$$

$$T_2 = 300\times(6)^{0.4/1.4}$$

$$= 500.5531\ \text{K}$$

Step 5 of 13

Process 3-4:

During the process 3-4, isentropic expansion of the gas takes place. Therefore,

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k}$$

$$\frac{1300}{T_4} = \left(r_p\right)^{(1.4-1)/1.4}$$

$$\frac{1300}{T_4} = (6)^{0.4/1.4}$$

$$T_4 = 779.1381\ \text{K}$$

Step 6 of 13

Heat energy given to the system per unit mass is given by,

$$q_{in} = c_p\left(T_3 - T_2\right)$$

$$= 1.005(1300 - 500.5531)$$

$$= 803.4441\ \text{kJ/kg}$$

Heat energy given out of the system per unit mass is given by,

$$q_{out} = c_p\left(T_4 - T_1\right)$$

$$= 1.005(779.1381 - 300)$$

$$= 481.5338\ \text{kJ/kg}$$

Step 7 of 13

The net work done by the system per unit mass is given by,

$$w_{net} = q_{in} - q_{out}$$

$$= 803.4441 - 481.5338$$

$$= 321.9103\ \text{kJ/kg}$$

Thermal efficiency of the system is given by,

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{321.9103}{803.9103}$$

$$= 40.07\%$$

Step 8 of 13

Since, the pressure ratio of the cycle is doubled. Therefore, the pressure ratio of the system, $r_p' = 12$

The pressure ratio is given by,

$$r_p' = \frac{P_2'}{P_1} = \frac{P_3'}{P_4'}$$

system, $r_p' = 12$

The pressure ratio is given by,

$$r_p' = \frac{P_2'}{P_1} = \frac{P_3'}{P_4'}$$

During the process $1\text{-}2'$, isentropic compression of the gas takes place. Therefore,

$$\frac{T_2'}{T_1} = \left(\frac{P_2'}{P_1}\right)^{(k-1)/k}$$

$$\frac{T_2'}{300} = \left(r_p'\right)^{(1.4-1)/1.4}$$

$$T_2' = 300 \times (12)^{0.4/1.4}$$

$$= 610.1811 \text{ K}$$

Step 9 of 13

During the process $3'\text{-}4'$, isentropic expansion of the gas takes place. Therefore,

$$\frac{T_3'}{T_4'} = \left(\frac{P_3'}{P_4'}\right)^{(k-1)/k}$$

$$\frac{1300}{T_4'} = \left(r_p'\right)^{(1.4-1)/1.4}$$

$$\frac{1300}{T_4} = (12)^{0.4/1.4}$$

$$T_4 = 639.1545 \text{ K}$$

Step 10 of 13

Heat energy given to the system per unit mass is given by,

$$q_{in}' = c_p\left(T_3 - T_2'\right)$$

$$= 1.005(1300 - 610.1811)$$

$$= 693.2680 \text{ kJ/kg}$$

Heat energy out of the system per unit mass is given by,

$$q_{out}' = c_p\left(T_4' - T_1'\right)$$

$$= 1.005(639.1545 - 300)$$

$$= 340.8502 \text{ kJ/kg}$$

Step 11 of 13

Work done by the system per unit mass is given by,

$$w_{net}' = q_{in}' - q_{out}'$$

$$= 693.2680 - 340.8502$$

$$= 352.4178 \text{ kJ/kg}$$

Thermal efficiency of the system is given by,

$$\eta_{th}' = \frac{w_{net}'}{q_{in}'}$$

$$= \frac{352.4178}{693.2680}$$

$$= 50.83\%$$

Step 12 of 13

(a)

Change in the net work output per unit mass of the cycle due to this modification is given by,

$$\Delta w_{net} = w_{net}' - w_{net}$$

$$= 352.4178 - 321.9103$$

$$= 30.51 \text{ kJ/kg}$$

Therefore, the change in the net work output per unit mass of the cycle due to this modification, $\boxed{\Delta w_{net} = 30.51 \text{ kJ/kg}}$

Step 13 of 13

(b)

Change in thermal efficiency of the cycle due to this modification is given by,

$$\Delta \eta_{th} = \eta_{th}' - \eta_{th}$$

$$= 50.83 - 40.07$$

$$= 10.76\%$$

Therefore, the change in thermal efficiency of the cycle due to this modification, $\boxed{\Delta \eta_{th} = 10.76\%}$

Question:

A steam power plant operating on a simple ideal Rankine cycle maintains the boiler at 6000 kPa, the turbine inlet at 600°C, and the condenser at 50 kPa. Compare the thermal efficiency of this cycle when it is operated so that the liquid enters the pump as a saturated liquid against that when the liquid enters the pump 11.3°C cooler than a saturated liquid at the condenser pressure.

Step 1 of 17

Given:

Boiler pressure $P_3 = 6000\text{kPa}$

Inlet temperature of the turbine $T_3 = 600°\text{C}$

Condenser pressure $P_1 = 50\text{kPa}$

Step 2 of 17

T-s Diagram:

Step 3 of 17

At state-1:

At $P_1 = 50\text{ kPa}$ and $x_1 = 0$

Since saturated liquid

Enthalpy at state-1,

$$h_1 = h_f = 340.54\ \frac{\text{kJ}}{\text{kg}}$$

Specific volume at -1,

$$v_1 = 0.001030\ \frac{\text{m}^3}{\text{kg}}$$

Work of pump

$$w_{pump} = v_1(P_2 - P_1)$$

$$w_{pump} = 0.001030\ \frac{\text{m}^3}{\text{kg}}(6000-50)\text{kPa} \times \left(\frac{1\text{kJ}}{1\text{kPa.m}^3}\right)$$

$$w_{pump} = 6.12\ \frac{\text{kJ}}{\text{kg}}$$

Step 4 of 17

At state-2:

Enthalpy at state-2

$$h_2 = h_1 + w_{pump}$$

$$h_2 = (340.54 + 6.12)$$

$$h_2 = 346.66\ \frac{\text{kJ}}{\text{kg}}$$

Step 5 of 17

At state-3:

From the superheated water

At $P_3 = 6000\text{ kPa}$ and $T_3 = 600°\text{C}$

Enthalpy at state-3,

$$h_3 = 3658.8\ \frac{\text{kJ}}{\text{kg}}$$

Entropy at state-3,

$$s_3 = 7.1693\ \frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Step 6 of 17

At state-4:

At $P_4 = 50\text{ kPa}$ and $s_4 = s_3 = 7.1693\ \frac{\text{kJ}}{\text{kg}\cdot\text{K}}$

It's an isentropic process

From the saturated water

At $P_4 = 50\text{ kPa}$

$$s_f = 1.0912\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

$$s_{fg} = 6.5019\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$$

Entropy at state-4,

$$s_4 = 7.169\text{ kJ/kg.K}$$

Dryness fraction at state-4,

$$s_4 = s_f + x_4 s_{fg}$$

$$\Rightarrow x_4 = \frac{s_4 - s_f}{s_{fg}}$$

$$x_4 = \frac{7.169 - 1.0912}{6.5019}$$

$$x_4 = 0.934$$

Step 7 of 17

Enthalpy at state-4,

From the saturated water

At $P_4 = 50\text{ kPa}$

$$h_f = 340.54\frac{\text{kJ}}{\text{kg}}$$

$$h_{fg} = 2304.7\frac{\text{kJ}}{\text{kg}}$$

$$h_f = 340.54 \frac{\text{kJ}}{\text{kg}}$$

$$h_{fg} = 2304.7 \frac{\text{kJ}}{\text{kg}}$$

$$h_4 = h_f + x_4 h_{fg}$$

$$h_4 = 340.5 + 0.934 \times 2304.7$$

$$h_4 = 2494.87 \ \frac{\text{kJ}}{\text{kg}}$$

Step 8 of 17

Total heat addition in the cycle occurs at process 2-3, so
Heat input per unit mass,

$$q_{in} = (h_3 - h_2)$$

$$q_{in} = (3658.8 - 346.66)\text{kJ/kg}$$

$$q_{in} = 3312.14 \ \frac{\text{kJ}}{\text{kg}}$$

Step 9 of 17

Heat rejected during process 4-1, so
Heat rejected per unit mass,

$$q_{out} = h_4 - h_1$$

$$q_{out} = (2494.87 - 340.54)$$

$$q_{out} = 2154.33 \ \frac{\text{kJ}}{\text{kg}}$$

Step 10 of 17

Total work output from the three turbines,

$$w_{turbine} = q_{in} - q_{out}$$

$$w_{turbine} = (3312.14 - 2154.33)$$

$$w_{turbine} = 1157.81 \ \frac{\text{kJ}}{\text{kg}}$$

Step 11 of 17

Thermal efficiency of the plant is

$$\eta_{th} = \frac{w_{turbine}}{q_{in}}$$

$$\eta_{th} = \frac{1157.81}{3312.14}$$

$$= 0.349$$

$$= 34.9\%$$

$$\boxed{\eta_{th} = 34.9\%}$$

Step 12 of 17

Case 2:
When liquid enters the pump at a temperature 11.3°C cooler than $T_{1,50kPa} = 81.3^\circ\text{C}$
At state-1:
At $P_1 = 50\ \text{kPa}$
Saturation temperature at $P_1 = 50\ \text{kPa}$ is $T_{1,50kPa} = 81.3$
So temperature at -1,

$$T_1 = T_{1,50kPa} - 11.3^\circ\text{C}$$

$$T_1 = 81.3^\circ\text{C} - 11.3^\circ\text{C}$$

$$T_1 = 70^\circ\text{C}$$

From the saturated water
At $T_1 = 70^\circ\text{C}$
Enthalpy at state-1,

$$h_1 = h_f = 293.07 \ \frac{\text{kJ}}{\text{kg}}$$

Specific volume at -1,

$$v_1 = 0.001023 \ \frac{\text{m}^3}{\text{kg}}$$

Step 13 of 17

Work of pump,

$$w_{pump} = v_1 (P_2 - P_1)$$

$$w_{pump} = 0.001023 \ \frac{\text{m}^3}{\text{kg}} (6000 - 50)\text{kPa} \times \left(\frac{1\text{kJ}}{1\text{kPa.m}^3} \right)$$

$$w_{pump} = 6.09 \ \frac{\text{kJ}}{\text{kg}}$$

Step 14 of 17

At state-2:
Enthalpy at state-2, $h_2 = h_1 + w_{pump}$

$$h_2 = (293.1 + 6.09)$$

$$h_2 = 299.2 \ \frac{\text{kJ}}{\text{kg}}$$

Step 15 of 17

Total heat addition in the cycle occurs at process 2-3, So
Heat input per unit mass,

$$q_{in} = (h_3 - h_2)$$

$$q_{in} = (3658.8 - 299.2)$$

$$q_{in} = 3359.6 \ \frac{\text{kJ}}{\text{kg}}$$

Step 16 of 17

Heat rejected during process 4-1, so

Step 16 of 17

Heat rejected during process 4-1, so
Heat rejected per unit mass,

$$q_{out} = h_4 - h_1$$

$$q_{out} = (2494.87 - 293.07)$$

$$q_{out} = 2201.8 \ \frac{kJ}{kg}$$

Total work output from the three turbines,

$$w_{turbine} = q_{in} - q_{out}$$

$$w_{turbine} = (3359.6 - 2201.8)$$

$$w_{turbine} = 1157.8 \ \frac{kJ}{kg}$$

Step 17 of 17

Thermal efficiency of the plant is

$$\eta_{th} = \frac{w_{turbine}}{q_{in}}$$

$$\eta_{th} = \frac{1157.8}{3359.6}$$

$$= 0.345$$

$$= 34.5\%$$

$$\boxed{\eta_{th} = 34.5\%}$$

So η_{th} slightly decreases as a effect of sub-cooling liquid at pump.

Question: 9.190P

The Stillwater geothermal power plant in Nevada, which started full commercial operation in 1986, is designedto operate with seven identical units. Each of these seven units consists of a pair of power cycles, labeled Level I and Level II, operating on the simple Rankine cycle using anorganic fluid as the working fluid.

The heat source for the plant is geothermal water (brine) entering the vaporizer (boiler) of Level I of each unit at 325°F at a rate of 384,286 lbm/h and delivering 22.79 MBtu/h ("M" stands for "million"). The organic fluid that enters the vaporizer at 202.2°C at a rate of 157,895 lbm/h leaves it at 282.4°F and 225.8 psia as saturated vapor. This saturated vapor expands in the turbine to 95.8°F and 19.0 psia and produces 1271 kW of electric power. About 200 kW of this power is used by the pumps, the auxiliaries, and the six fans of the condenser- Subsequently, the organic working fluid is condensed ■ an air-cooled condenser by air that enters the condenser at 55°F at a rate of 4,195,100 lbm/h and leaves at 84.5°C. The working fluid is pumped and then preheated in a preheater to 202.2°F by absorbing 11.14 MBtu/h of heat from the geothermal water (coming from the vaporizer of Level II) entering the preheater at 211.8°F and leaving at 154.0°F.

Taking the average specific heat of the geothermal water to be 1.03 Btu/lbm °F, determine (*a*) the exit temperature of the geothermal water from the vaporizer, (*b*) the rate of heat rejection from the working fluid to the air in the condenser, (*c*) the mass flow rate of the geothermal water at the preheater, and (*d*) the thermal efficiency of the Level I cycle of this geothermal power plant.

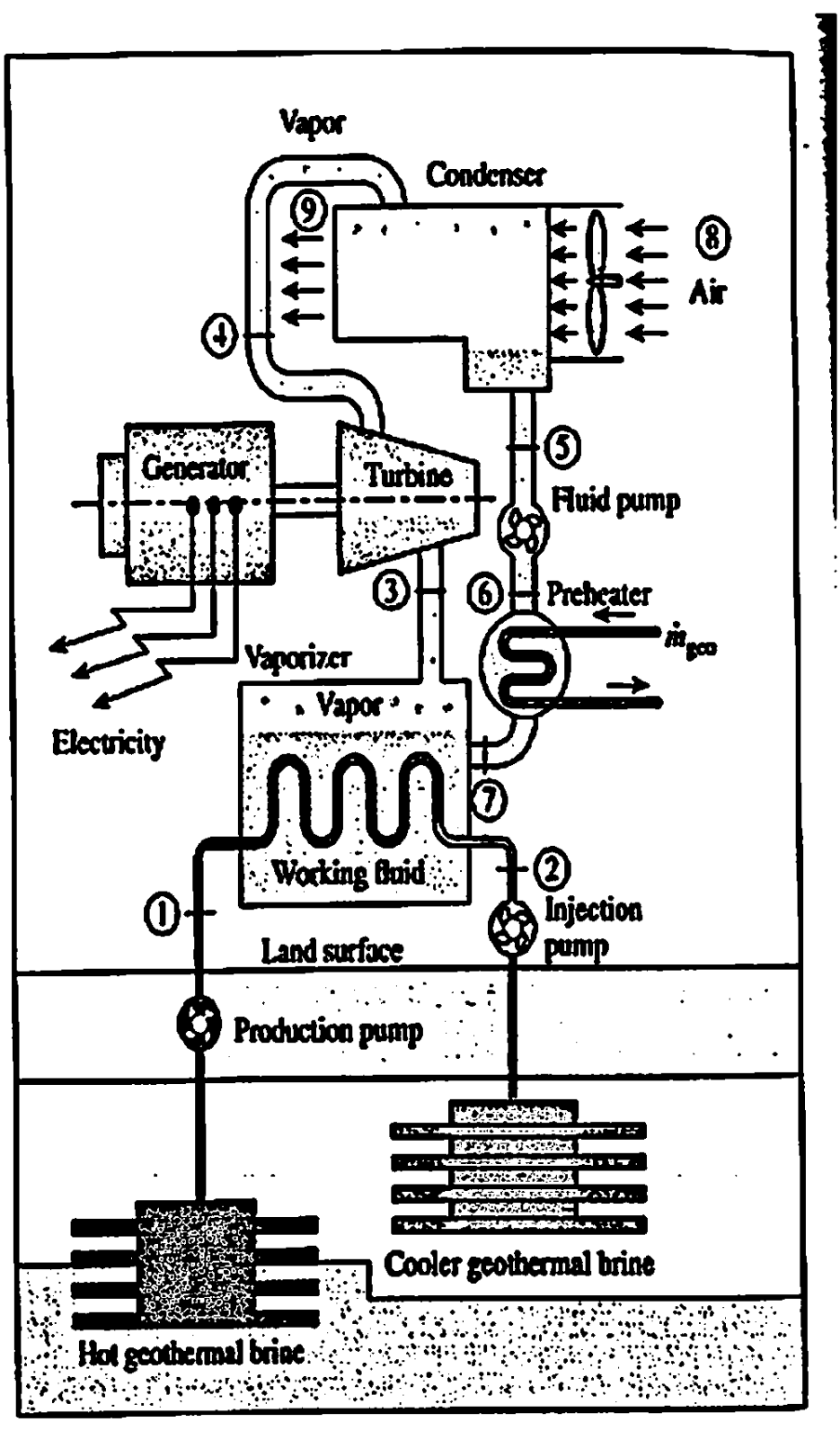


Figure P9-190

Schematic of a binary geothermal power plant.

Step 1 of 7

Given data:

Geothermal water entering the vaporizer at temperature $T_1 = 325^0\text{F}$

Mass flow rate of geothermal water $\dot{m}_{geowater} = 384286\text{lbm/h}$

Amount of heat delivered from geothermal water $Q_{geowater} = -22.79\text{MBtu/h}$

Specific heat of geothermal water $c_p = 1.03\text{Btu/lbm}$

Work output from the turbine $W_T = 1271\text{kW}$

Power used by the pumps, the auxiliaries and fans are $W_p = -200\text{kW}$

Air enters the condenser at temperature $T_8 = 55^0\text{F}$

Mass flow rate of air $\dot{m}_{air} = 4195100\text{lbm/h}$

Air leaves the condenser at temperature $T_9 = 84.5^0\text{C}$

The amount of heat absorbed to the working fluid $Q_{working\,fluid} = 11.14\text{MBtu/h}$

Geothermal water entering the preheater at temperature $T_6 = 211.8^0\text{F}$

Geothermal water leaving the preheater at temperature $T_7 = 154.0^0\text{F}$

Step 2 of 7

(a)

The exit temperature of the geothermal water from the vaporizer is determined using the steady flow energy balance on the geothermal water

$$Q_{geowater} = m_{geowater} \times c_p \times (T_2 - T_1)$$

$$-22790000 = 384286 \times 1.03 \times (T_2 - 325)$$

$$\boxed{T_2 = 267.42^0\text{F}}$$

Step 3 of 7

(b)

The rate of heat rejection from the working fluid to the air in the condenser

Applying steady flow energy balance on air

$$Q_{air} = m_{air} \times c_p \times (T_9 - T_8)$$

$$Q_{air} = 4195100 \times 0.24 \times (84.5 - 55)$$

$$Q_{air} = 29701308\text{Btu/hr}$$

$$\boxed{Q_{air} = 29.7\text{ MBtu/hr}}$$

Step 4 of 7

(c)

The mass flow rate of geothermal water at the preheater

Applying steady flow energy balance on the geothermal water in preheater

$$Q_{geo} = m_{geo} \times c_{p,geo} \times (T_{out} - T_{in})$$

We know the amount of heat absorbed by working fluid is equal to the amount of heat rejected by the geothermal water

$$-11140000 = m_{geo} \times 1.03 \times (154 - 211.8)$$

$$\boxed{m_{geo} = 187119.96\text{ lbm/hr}}$$

Step 5 of 7

(d)

The heat addition is given by,

$$Q_{in} = Q_{vaporizer} + Q_{preheater}$$

$$Q_{in} = 22790000 + 11140000$$

$$Q_{in} = 33930000\text{ Btu/hr.}$$

Step 6 of 7

The network output is

$$W_{net} = W_T + W_p$$

$$W_{net} = 1271 - 200$$

$$W_{net} = 1071\text{ kW}$$

Step 7 of 7

Thermal efficiency of the level I cycle of this geothermal power plant is

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

$$\eta_{th} = \frac{1071\text{ kW}}{33930000\text{ Btu/hr}}$$

Since we know that $1\text{kWhr} = 3412.14\text{Btu}$

$$\eta_{th} = \frac{1071\text{ kW}}{33930000\text{ Btu/hr}} \times \frac{3412.14\text{Btu}}{1\text{kWhr}}$$

$$\eta_{th} = 0.1077$$

$$\boxed{\eta_{th} = 10.77\%}$$

Question: 9.191P

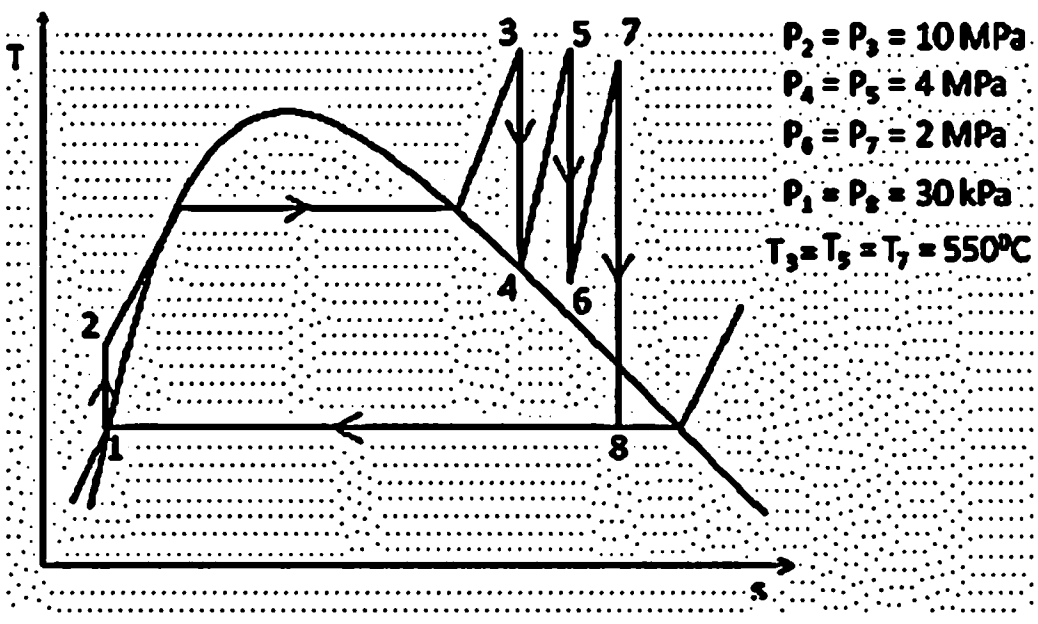
A steam power plant operates on an ideal Rankine cycle with two stages of reheat and has a net power output of 75 MW. Steam enters all three stages of the turbine at 550°C. The maximum pressure in the cycle is 10 MPa, and the minimum pressure is 30 kPa. Steam is reheated at 4 MPa the first time and at 2 MPa the second time. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the thermal efficiency of the cycle, and (*b*) the mass flow rate of the steam.

Step 1 of 7

Given,
The net output of the power plant, $W_{net} = 75\ \text{MW}$,
Steam enters all three stages of the turbine at 550^0C
The maximum pressure in the cycle = 10 MPa,
The minimum pressure in the cycle = 30 kPa,

Step 2 of 7

The Ideal Reheat Rankin cycle operating between the given conditions is shown below.



Step 3 of 7

From saturated steam tables, at State-1 (30 kPa)

$$h_1 = \left(h_f\right)_{\text{at } 30\text{ kPa}} = 289.22\ \text{kJ/kg}$$

$$v_1 = \left(v_f\right)_{\text{at } 30\text{ kPa}} = 0.001022\ \text{m}^3\text{/kg}$$

The work done in the compression process is given by,

$$w_{pump} = v_1 \times (P_2 - P_1)$$
$$= \left(0.001022\ \text{m}^3\text{/kg}\right) \times (10000\ \text{kPa} - 30\ \text{kPa})$$
$$= 10.2\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{pump}$$
$$= 289.22 + 10.2$$
$$= 299.42\ \text{kJ/kg}$$

Step 4 of 7

From superheated steam tables at State-3 (10 MPa and 550^0C)

$h_3 = 3501.9\ \text{kJ/kg}$

$s_3 = 6.7583\ \text{kJ/kgK}$

For isentropic expansion, $s_4 = s_3 = 6.7583\ \text{kJ/kgK}$

From superheated steam tables at State-4 (4 MPa and $s_4 = 6.7583\ \text{kJ/kgK}$)

$h_4 = 3205.7\ \text{kJ/kg}$

From superheated steam tables at State-5 (4 MPa and 550^0C)

$h_5 = 3560.2\ \text{kJ/kg}$

$s_5 = 7.2352\ \text{kJ/kgK}$

Step 5 of 7

For isentropic expansion, $s_6 = s_5 = 7.2352\ \text{kJ/kgK}$

From superheated steam tables at State-6 (2 MPa and $s_6 = 7.2352\ \text{kJ/kgK}$)

$h_6 = 3321.5\ \text{kJ/kg}$

From superheated steam tables at State-7 (2 MPa and 550^0C)

$h_7 = 3578.9\ \text{kJ/kg}$

$s_7 = 7.5723\ \text{kJ/kgK}$

For isentropic expansion, $s_8 = s_7 = 7.5723\ \text{kJ/kgK}$

From saturated steam tables at State-8 (30 kPa and $s_8 = 7.5723\ \text{kJ/kgK}$)

$h_8 = 2557.8\ \text{kJ/kg}$

Step 6 of 7

(b)

Now net work output is given as

$$W_{net} = \dot{m}\left[(h_3 - h_4) + (h_5 - h_6) + (h_7 - h_8) - (h_2 - h_1)\right]$$
$$75000 = \dot{m}[(3501.9 - 3205.7) + (3560.2 - 3321.5) + (3578.9 - 2557.8) - (299.42 - 289.22)]$$
$$75000 = \dot{m} \times 1545.8$$
$$\dot{m} = 48.51\ \text{kg/s}$$

Hence the mass flow rate is $\boxed{\dot{m} = 48.51\ \text{kg/s}}$

Step 7 of 7

(a)

Heat supplied to the cycle is

$$Q_{in} = \dot{m} \times \left[(h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6)\right]$$
$$= 48.51 \times \left[(3501.9 - 299.42) + (3560.2 - 3205.7) + (3578.9 - 3321.5)\right]$$
$$= 185\ \text{MW}$$

The thermal efficiency of the Rankin cycle is given by,

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$
$$= \frac{75}{185}$$
$$= 0.405$$

Hence thermal efficiency of the cycle is $\boxed{\eta_{th} = 40.5\%}$

Question: 9.192P

Consider a steam power plant operating on the ideal Rankine cycle with reheat between the pressure limits of 25 MPa and 10 kPa with a maximum cycle temperature of 600°C and a moisture content of 8 percent at the turbine exit. For a reheat temperature of 600°C, determine the reheat pressures of the cycle for the cases of (*a*) single and (*b*) double reheat.

Step 1 of 7

Given data:

Maximum pressure in the cycle $P_3 = 25\,\text{MPa}$

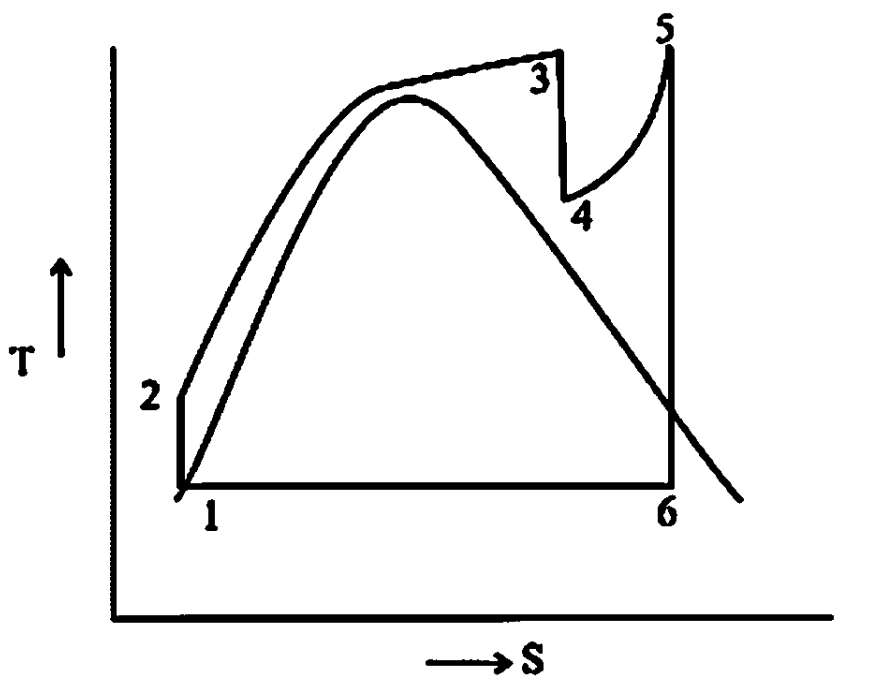
Minimum pressure in the cycle $P_1 = 10\,\text{kPa}$

Maximum cycle temperature $T_3 = 600^0\,\text{C}$

Reheat temperature $T_5 = 600^0\,\text{C}$

Dryness fraction at the turbine exit $x_6 = 0.92$

Step 2 of 7



Step 3 of 7

(a)

For a single reheat cycle

At $P_6 = 10\,\text{kPa}$ and $x_6 = 0.92$

Enthalpy at state -6 is given by

$h_6 = h_f + x_6 h_{fg}$

$h_6 = 191.81 + (0.92 \times 2392.1)$

$h_6 = 2392.54\ \text{kJ/kg}$

Entropy at state-6 is given by

$s_6 = s_f + x_6 s_{fg}$

$s_6 = 0.6492 + (0.92 \times 7.4996)$

$s_6 = 7.5488\ \text{kJ/kg.K}$

Step 4 of 7

Process 5-6 is isentropic expansion then entropy remains constant

$s_5 = s_6$

At $T_6 = 600^0\,\text{C}$ and $s_6 = s_5 = 7.5488\,\text{kJ/kg.K}$ from superheated water tables

$$P_6 = 2500 + (3000 - 2500)\left(\frac{7.5488 - 7.5979}{7.5103 - 7.5979}\right)$$

$P_6 = 2780.251\,\text{kPa}$

Therefore reheat pressure of the cycle for the case of single reheat is

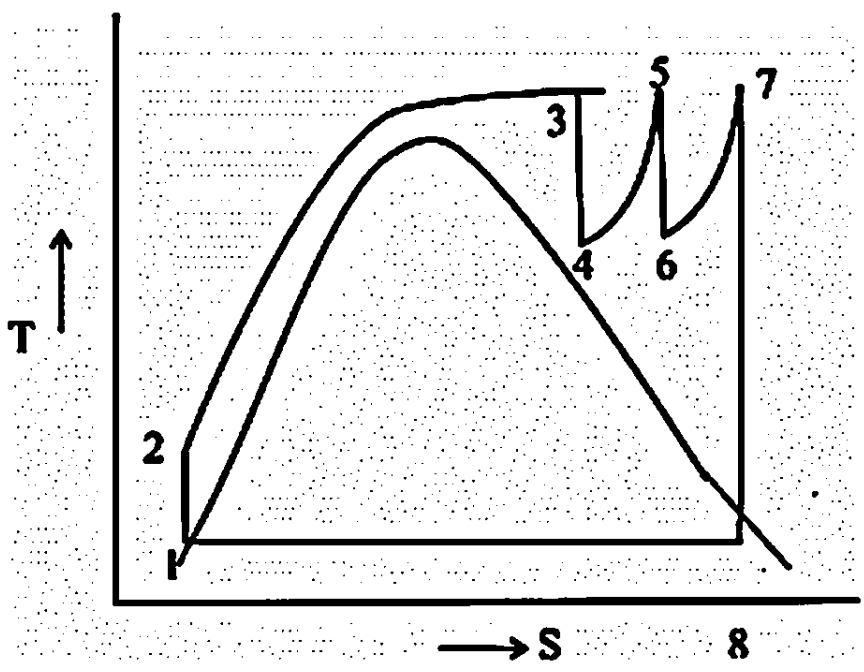
$\boxed{P_5 = 2780.251\ \text{kPa}}$ Or

$\boxed{P_5 = 2.780\ \text{MPa}}$

Step 5 of 7

(b)

Double reheat cycle:



Step 6 of 7

At $P_8 = 10\ \text{kPa}$ and $x_8 = 0.92$

Enthalpy at state -8 is given by

$h_8 = h_f + x_8 h_{fg}$

$h_8 = 191.81 + (0.92 \times 2392.1)$

$h_8 = 2392.54\ \text{kJ/kg}$

Entropy at state-8 is given by

$s_8 = s_f + x_8 s_{fg}$

$s_8 = 0.6492 + (0.92 \times 7.4996)$

$s_8 = 7.5488\ \text{kJ/kg.K}$

Step 7 of 7

Process 7-8 is isentropic expansion then entropy remains constant

$s_7 = s_8$

At $T_7 = 600^0\,\text{C}$ and $s_7 = s_8 = 7.5488\,\text{kJ/kg.K}$ from superheated water tables

$$P_7 = 2500 + (3000 - 2500)\left(\frac{7.5488 - 7.5979}{7.5103 - 7.5979}\right)$$

$P_7 = 2780.251\,\text{kPa}$

$P_7 = 2.78\,\text{MPa}$

Any pressure P_x selected between the limits of 25Mpa and 2.78Mpa will satisfy the requirements, and can be used for the double reheat pressure.

Question:

Using EES (or other) software, investigate the effect of the condenser pressure on the performance of a simple ideal Rankine cycle. Turbine inlet conditions of steam are maintained constant at 10 MPa and 550°C while the condenser pressure is varied from 5 to 100 kPa. Determine the thermal efficiency of the cycle and plot it against the condenser pressure, and discuss the results.

Step 1 of 6

The procedure for developing solution by using EES software: Open the new equations window EES software and type all the information which is used to solve the problem For developing table: 1) Click on table followed by new parametric table and add the variables, which are varying variable and required answer variable, then one table will form with that variables 2) Click on the variable which is varying and enter initial value and final value. 3) For getting the required answer variable values corresponding to the varying variable values click on the green colour button and then the table will form with required answers For developing graph: 1) Click on plot followed by new plot window followed by x-y plot. 2) Select the x-axis variable and y-axis variable and then press ok, then the graph will form with the given values.

Step 2 of 6

Write the program for the given system in EES software:

EES Limited Academic Version: C:\EES (LIMITED ACADEMIC)\HELLO.TXT - [Equations ...

File Edit Search Options Calculate Tables Plots Windows Help

Thermal-Fluid Sciences

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
x4$=''
if (x4>1) then x4$='(superheated)'
if (x4<0) then x4$='(compressed)'
end
P[3] = 10000 [kPa]
T[3] = 550 [C]
"P[4] = 5 [kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"
"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0 "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"

x4s$=x4$(x[4])
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"
"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

Step 3 of 6

Calculate the range of the thermal efficiencies and network done varying the power out flow:

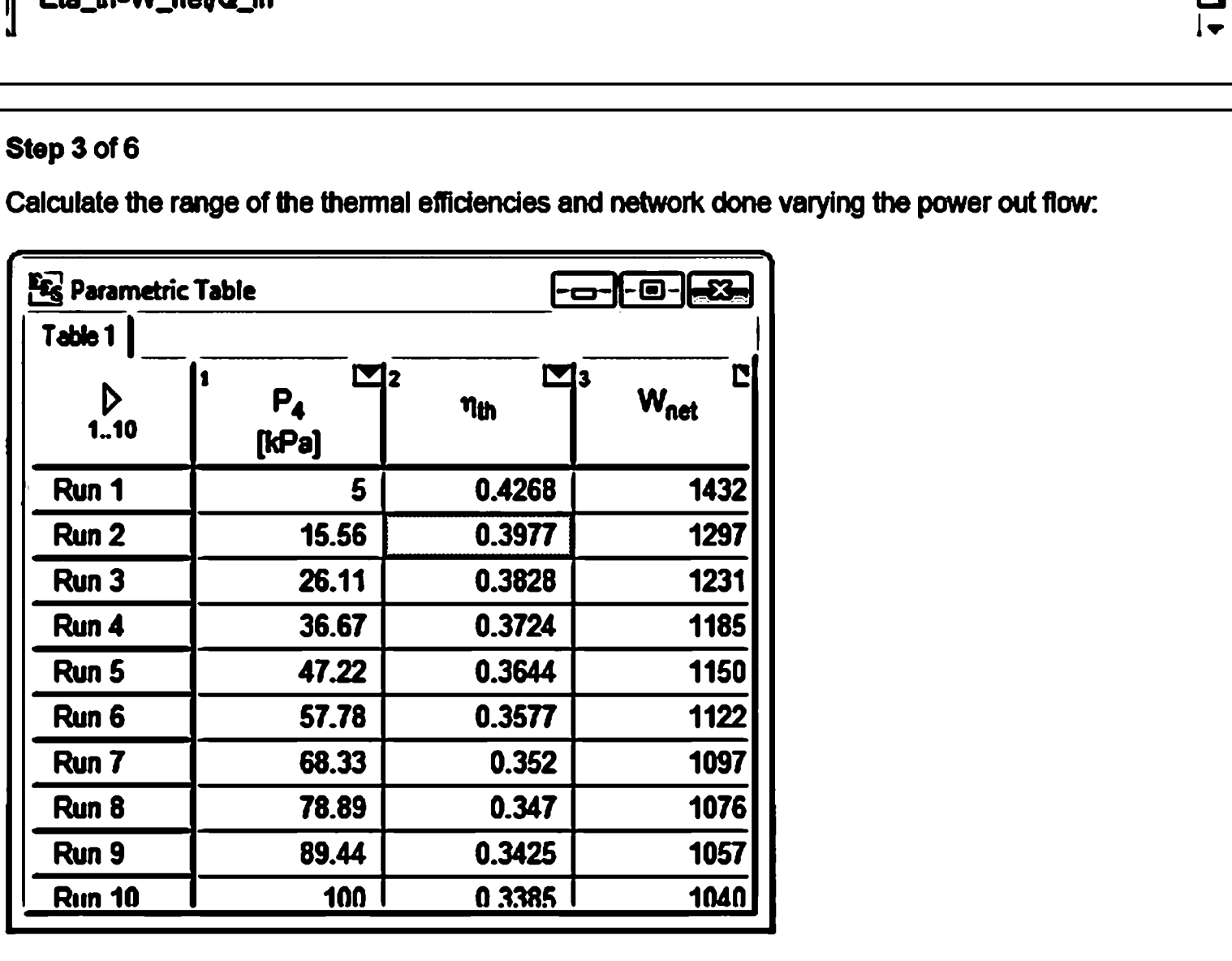
Parametric Table

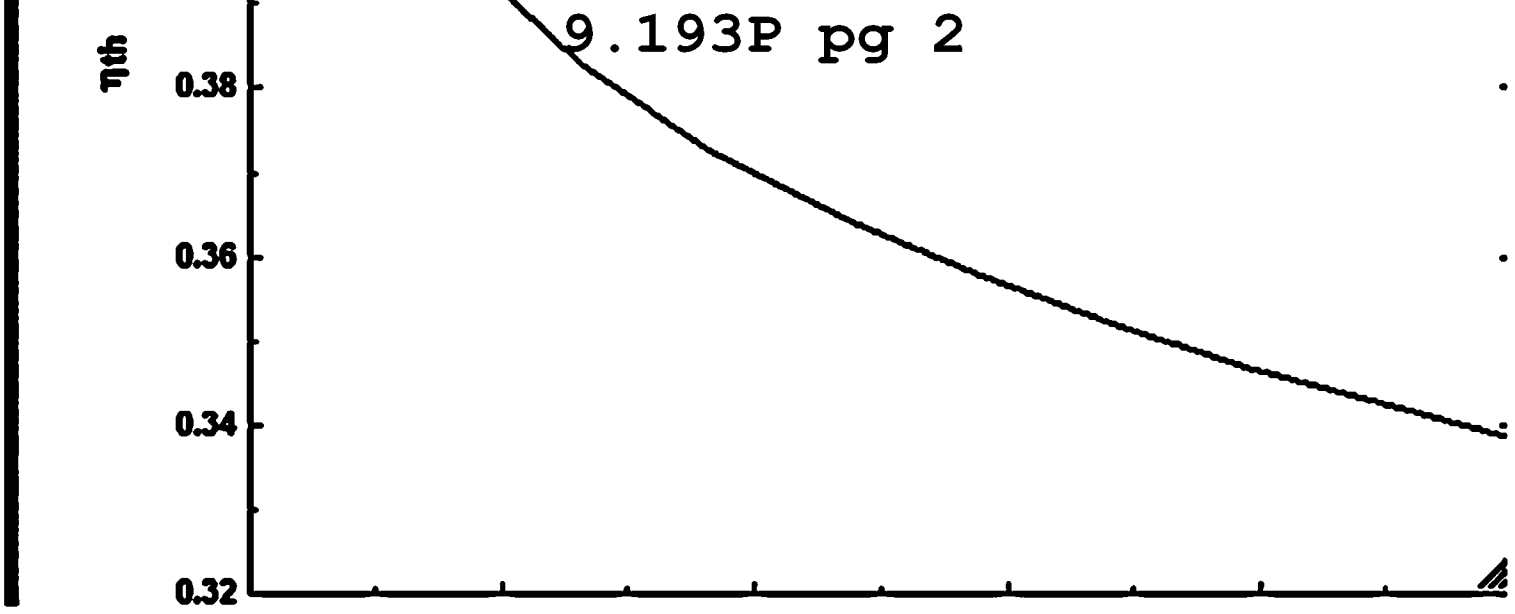
Table 1

1..10	1 P_4 [kPa]	2 η_{th}	3 W_{net}
Run 1	5	0.4268	1432
Run 2	15.56	0.3977	1297
Run 3	26.11	0.3828	1231
Run 4	36.67	0.3724	1185
Run 5	47.22	0.3644	1150
Run 6	57.78	0.3577	1122
Run 7	68.33	0.352	1097
Run 8	78.89	0.347	1076
Run 9	89.44	0.3425	1057
Run 10	100	0.3385	1040

Step 4 of 6

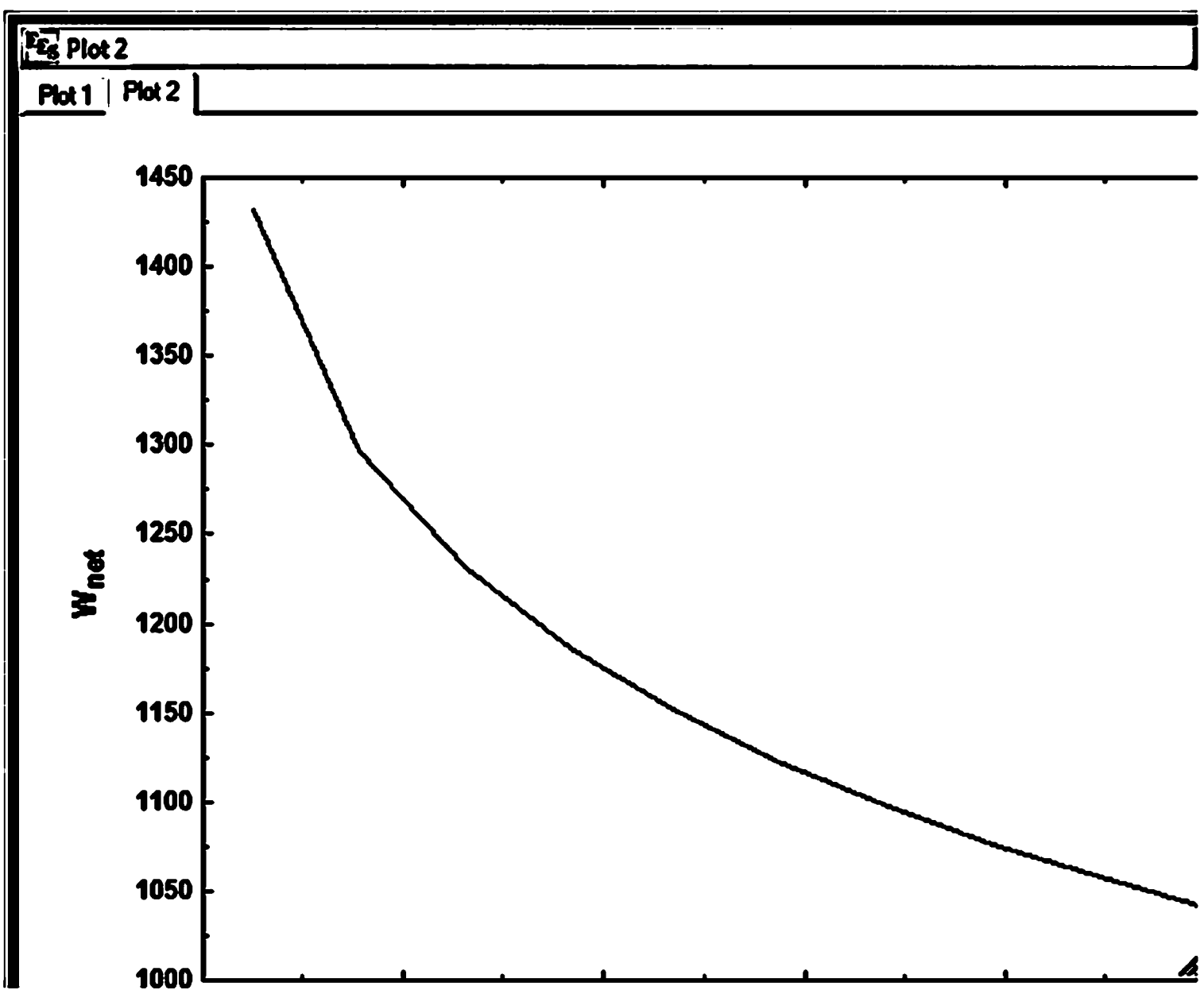
Plot the graph between the pressure at point at point 4 and the thermal efficiency:





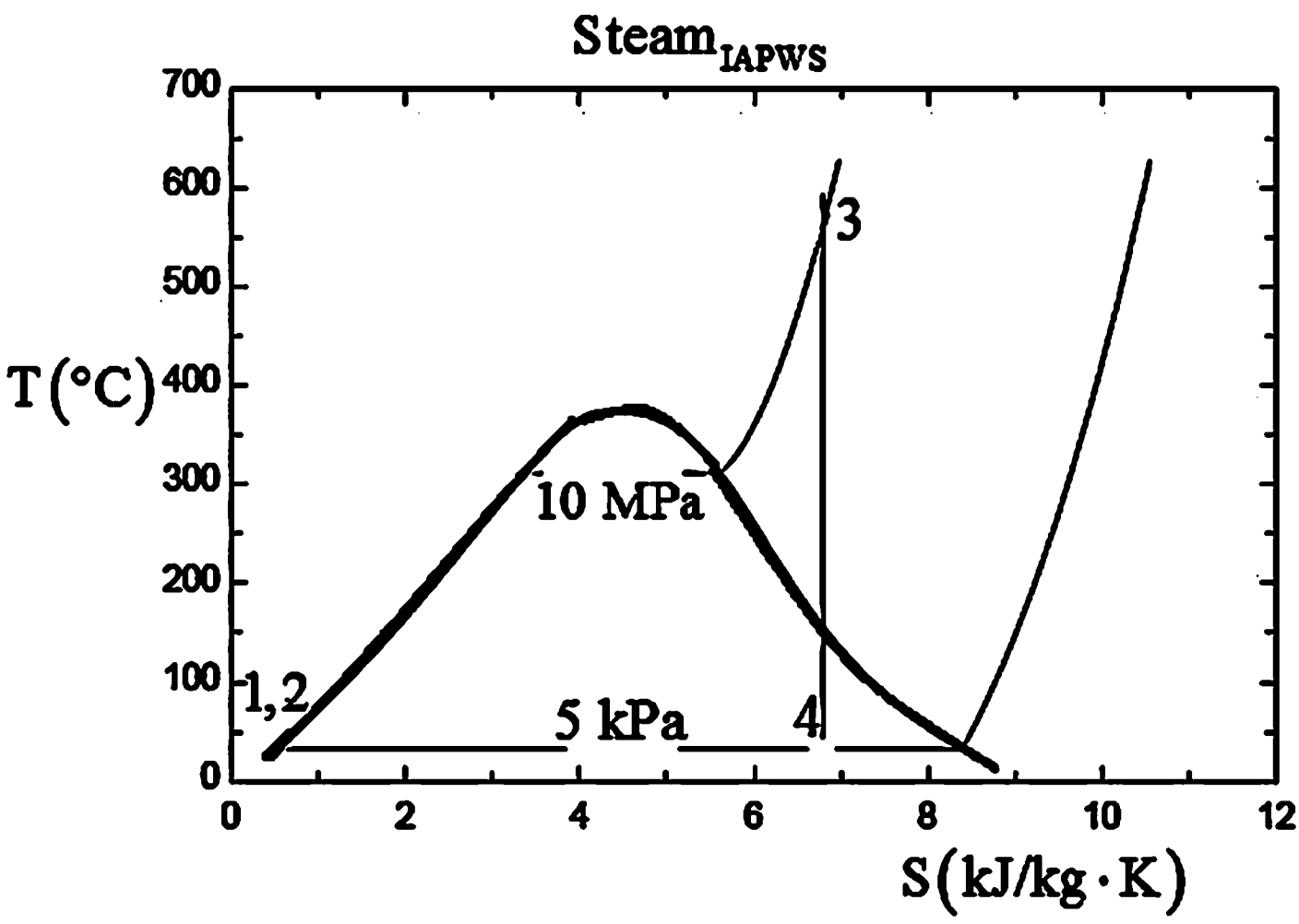
Step 5 of 6

Plot the graph between the pressure at point at point 4 and the network done:



Step 6 of 6

Draw the graph between the temperature and the entropy:



Question: 9.194P



Using EES (or other) software, investigate the effect of superheating the steam on the performance of a simple ideal Rankine cycle. Steam enters the turbine at 3 MPa and exits at 10 kPa. The turbine inlet temperature is varied from 250 to 1100°C. Determine the thermal efficiency of the cycle and plot it against the turbine inlet temperature, and discuss the results.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 9.195P



Using EES (or other) software, investigate the effect of number of reheat stages on the performance of an ideal Rankine cycle. The maximum and minimum pressures in the cycle are 15 MPa and 10 kPa, respectively, and steam enters all stages of the turbine at 500°C. For each case, maintain roughly the same pressure ratio across each turbine stage. Determine the thermal efficiency of the cycle and plot it against the number of reheat stages 1, 2, 4, and 8, and discuss the results.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 9.196P

Consider a steady-flow Carnot refrigeration cycle that uses refrigerant-134a as the working fluid. The maximum and minimum temperatures in the cycle are 30 and -20°C. respectively. The quality of the refrigerant is 0.15 at the beginning of the heat absorption process and 0.80 at the end. Show the cycle on a *T*-*s* diagram relative to saturation lines, and determine (*a*) the coefficient of performance, (*b*) the condenser and evaporator pressures, and (*c*) the net work input.

Step 1 of 8

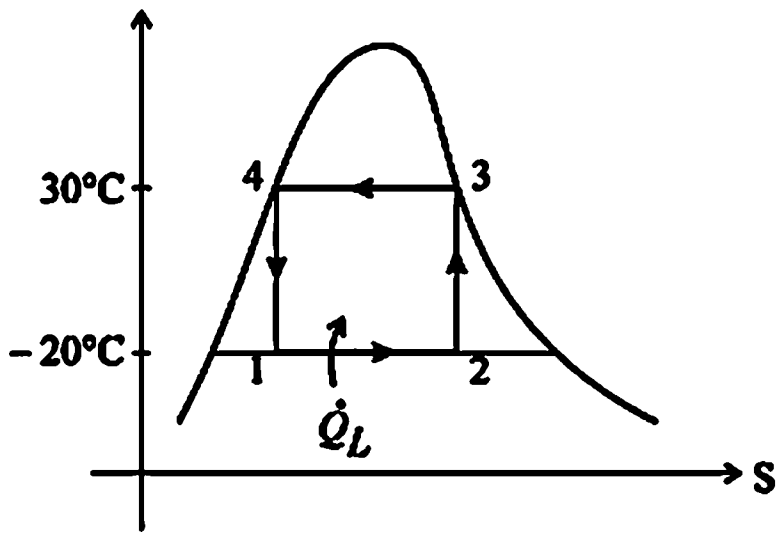
Given:
Working fluid: R-134a
Maximum temperature $T_H = 30°C = 30+273 = 303K$
Minimum temperature $T_L = -20°C = -20+273 = 253K$

Step 2 of 8

Assumptions:
1. Steady operating conditions exist.
2. Kinetic and potential energy changes are negligible.

Step 3 of 8

T-S diagram of carnot refrigeration cycle



Step 4 of 8

From Saturated water-temperature table
$P_{sat@-20°C} = 132.82\ kPa$
$P_{sat@30°C} = 770.64\ kPa$
$h_{f@-20°C} = 25.49\ kJ/kg$
$h_{fg@-20°C} = 212.91\ kJ/kg$

Step 5 of 8

(a) COP of refrigeration cycle:

$$COP = \frac{1}{\frac{T_H}{T_L} - 1}$$

$$COP = \frac{1}{\frac{303}{253} - 1}$$

$$\boxed{COP = 5.06}$$

Step 6 of 8

(b) Condenser and evaporator pressures:
From Saturated water-temperature table
Pressure of condenser, $\boxed{P_c = P_{sat@30°C} = 770.64\ kPa}$
Pressure of evaporator, $\boxed{P_e = P_{sat@-20°C} = 132.82\ kPa}$

Step 7 of 8

(c) Net work input:

$$h_1 = h_{f@-20°C} + x_1 \times h_{fg@-20°C}$$
$$h_1 = 25.49 + 0.15 \times 212.91$$
$$h_1 = 57.4265\ kJ/kg$$
$$h_2 = h_{f@-20°C} + x_2 \times h_{fg@-20°C}$$
$$h_2 = 25.49 + 0.8 \times 212.91$$
$$h_2 = 195.818\ kJ/kg$$

Step 8 of 8

$$\text{But, } COP = \frac{\dot{Q}_L}{\dot{W}_{in}}$$

$$COP = \frac{(h_2 - h_1)}{\dot{W}_{in}}$$

$$\dot{W}_{in} = \frac{195.818 - 57.4265}{5.06}$$

$$\boxed{\dot{W}_{in} = 27.35\ kJ/kg}$$

Question:

9.197P

Rooms with floor areas of up to 15-m2 are cooled adequately by window air conditioners whose cooling capacity is 5000 Btu/h. Assuming the COP of the air conditioner to be 3.5, determine the rate of heat gain of the room, in Btu/h, when the air conditioner is running continuously to maintain a constant room temperature.

Step 1 of 2

Assumptions:

1. The heat gain includes heat transfer through walls and the roof, infiltration heat gain, solar heat gain etc.
2. Steady Operation condition exist

Step 2 of 2

The rate of heat gain of the room in steady operation is equal to the cooling rate of the air conditioning system.

So, $\boxed{\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling rate}} = 5000\text{Btu/h}}$

Question:

9.198P

A heat pump water heater (HPWH) heats water by absorbing heat from the ambient air and transferring it to water. The heat pump has a COP of 3.4 and consumes 6 kW of electricity when running. Determine if this heat pump can be used to meet the cooling needs of a room most of the time for "free" by absorbing heat from the air in the room. The rate of heat gain of a room is usually less than 45,000 kJ/h.

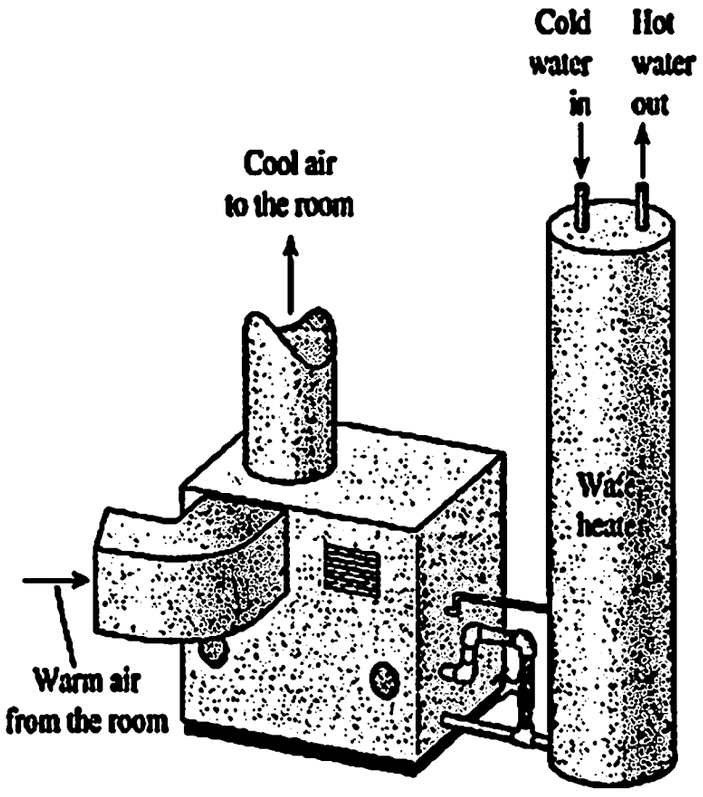


FIGURE P9–198

Step 1 of 3

Given:

COP of the heat pump $(COP)_{HP} = 3.4$

Work input to the compressor $W_{in} = 6\,\text{kW}$

Rate of heat gain $Q_H = 45000\,\frac{\text{kJ}}{\text{h}}$

Step 2 of 3

We know that

Coefficient of performance

$$COP = \frac{Q_H}{W_{in}}$$

$$\Rightarrow \dot{Q}_H = \dot{W}_{in} \times COP$$

$$= 6 \times 3.4$$

$$= 20.4\,\text{kW}$$

Step 3 of 3

But according to problem, we have

$$\dot{Q}_H = 45000\,\frac{\text{kJ}}{\text{h}}$$

$$= 12.5\,\text{kW}$$

$$Q_H = 20.4\,\text{kW} > Q_{\text{rabbum}}$$

So, heat pump can be used to meet the cooling need of the room most of the time.

Question: 9.199P

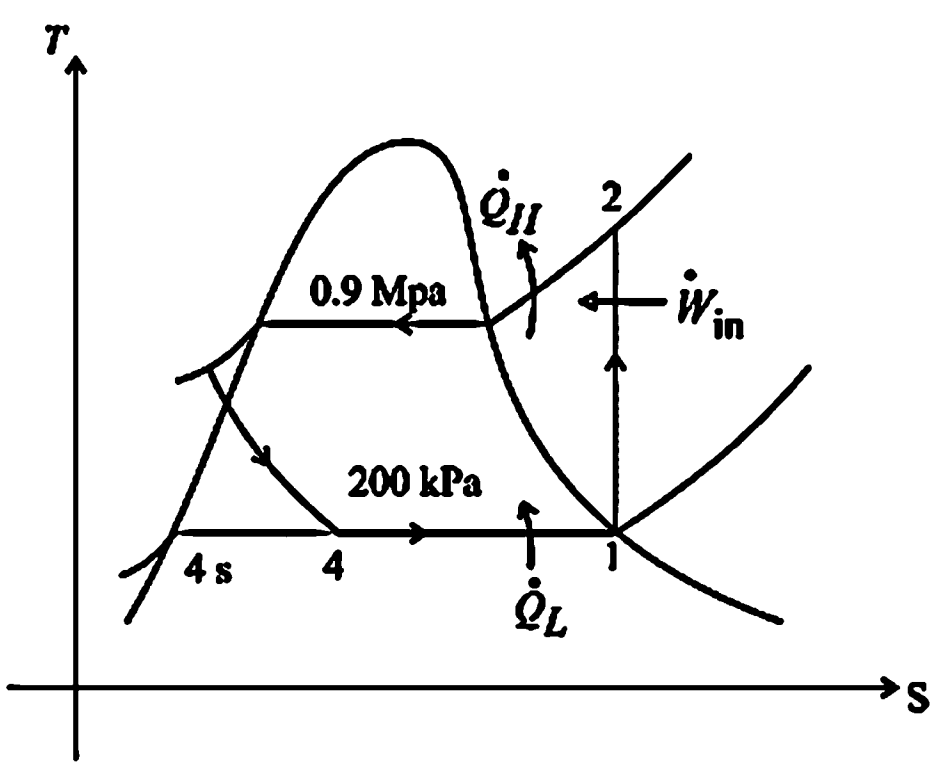
A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat a house. The mass flow rate of the refrigerant is 0.32 kg/s. The condenser and evaporator pressures are 900 and 200 kPa, respectively. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the rate of heat supply to the house, (*b*) the volume flow rate of the refrigerant at the compressor inlet, and (*c*) the COP of this heat pump.

Step 1 of 7

Given:
Working fluid: R-134a
Mass flow rate = 0.32 kg/s
Condenser pressure $P_2 = 900\,\text{kPa}$
Evaporator pressure $P_1 = 200\,\text{kPa}$

Step 2 of 7

T-S diagram of vapor compression cycle



Step 3 of 7

From saturated water-pressure table
At $P_1 = 200$ kPa and saturated vapor,
$h_1 = h_{g@200kPa} = 244.46$ kJ/kg,
$s_1 = s_{g@200kPa} = 0.93773$ kJ/kg.K,
$v_1 = v_{g@200kPa} = 0.099867\ \text{m}^3/\text{kg}$
At $P_2 = 900$ kPa and $s_2 = s_1 = 0.93773$ kJ/kg.K,

$$h_2 = 274.17 + \frac{0.93773 - 0.9327}{0.9660 - 0.9327} \times (284.77 - 274.17)$$

$h_2 = 275.77$ kJ/kg,
At $P_3 = P_2 = 900$ kPa and saturated liquid,
$h_3 = h_{f@0.9MPa} = 101.61$ kJ/kg,
For throttling, $h_4 = h_3 = 101.61$ kJ/kg,

Step 4 of 7

Assumptions:
1. Steady operating conditions exist.
2 Kinetic and potential energy changes are negligible.

Step 5 of 7

(a) Rate of heat supply to the house:
Rate of heat supply to the house, $\dot{Q}_H = \dot{m} \times (h_2 - h_3)$
$\dot{Q}_H = 0.32 \times (275.77 - 101.61)$
$\boxed{\dot{Q}_H = 55.7312\ \text{kW}}$

Step 6 of 7

(b) Volume flow rate of the refrigerant at the compressor inlet:
Volume flow rate of the refrigerant at the compressor inlet, $\dot{V}_1 = \dot{m} \times v_1$
$\dot{V}_1 = 0.32 \times 0.099867$
$\boxed{\dot{V}_1 = 0.03196\ \text{m}^3/\text{s}}$

Step 7 of 7

(c) COP of the heat pump:
COP of the heat pump, $\text{COP} = \dfrac{\dot{Q}_L}{\dot{W}_{in}}$

$$\text{COP} = \frac{h_2 - h_3}{h_2 - h_1}$$

$$\text{COP} = \frac{275.77 - 101.61}{275.77 - 244.46}$$

$\boxed{\text{COP} = 5.5624}$

Question: 9.200P

A ground-coupled heat pump operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. This heat pump has a COP equal to 6 in this cooling mode and maintains a space at 25°C when absorbing heat from the space at the rate of 18 kW in the evaporator. The evaporator operates at 20°C, and the condenser operates at 1400 kPa. Ground water taken from a well at the rate of 0.32 kg/s absorbs the heat transferred from the refrigerant in the condenser.

(a) Sketch the hardware and the *T-s* diagram for this heat pump,

(b) If the ground water enters the condenser at 10°C, what will be the temperature of the cooling water as it is returned to the ground? Take *cp*= 4.18 kJ/kg.K for cooling water.

(c) Determine the COP of the heat pump.

RefrigeranM34a data: *T*= 20°C: *hf* =79.3 kJ/kg, *hg* = 261.6 kJ/kg; *P* =1400 kPa: *hf* = *Mil* kJ/kg,

hg =276.2 kJ/kg

Step 1 of 7

Given:

Working fluid: R-134a

COP of the heat pump $(COP)_R = 6$

Refrigeration space temperature $T_1 = 25^\circ C$

Capacity of the evaporator $Q_L = 18\,kW$

Operating temperature of evaporator $T_4 = 20^\circ C$

Operating pressure of condenser $P_2 = P_3 = 1400\,kPa$

Mass flow rate of water $m = 0.32\frac{kg}{s}$

Inlet temperature of the ground water $T_{w1} = 10^\circ C$

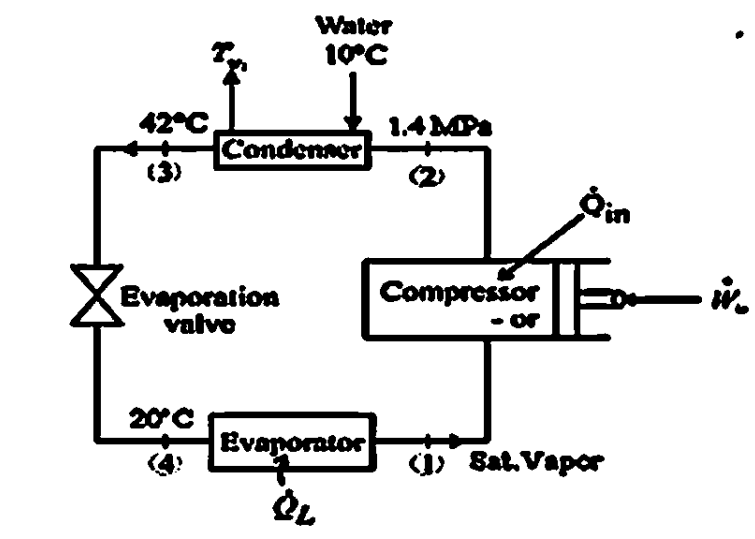
Specific heat of water $C_p = 4.18\frac{kJ}{kg \cdot K}$

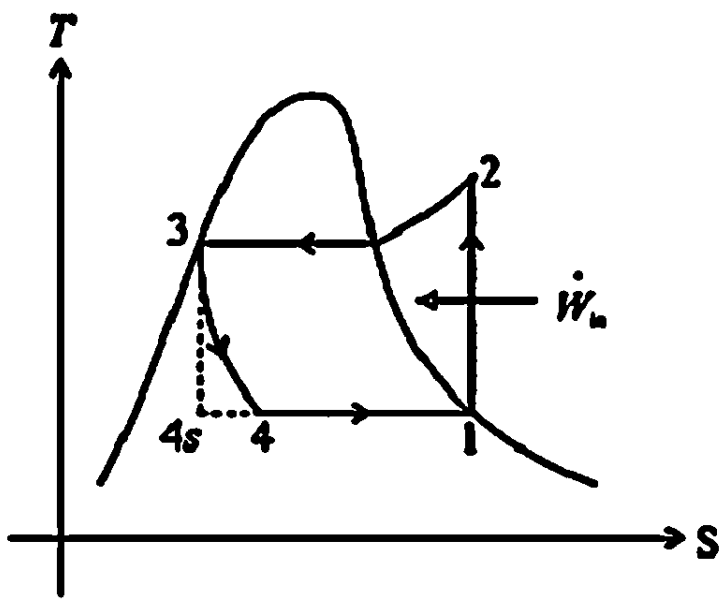
Properties of R-134a

$T = 20^\circ C$	$h_f = 79.3\frac{kJ}{kg}$	$h_g = 261.6\frac{kJ}{kg}$
$P_2 = P_3 = 1400\,kPa$	$h_f = 127.2\frac{kJ}{kg}$	$h_g = 276.2\frac{kJ}{kg}$

Step 2 of 7

(a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic; the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure.





Step 3 of 7

(b) State 3:

From the properties of R-134a

At $P_2 = P_3 = 1400\,kPa$

$$h_3 = h_f = 127.2\frac{kJ}{kg}$$

Step 4 of 7

State 4:

Since process 3 to 4 is throttling so enthalpy remains constant.So

$$h_4 = h_3 = 127.2\frac{kJ}{kg}$$

State1:

Since refrigerant enters into the compressor is saturated Vapor. So

$$h_1 = h_g = 261.6\frac{kJ}{kg}$$

Step 5 of 7

The rate of heat transfer in the condenser is determined from

$$COP = \frac{\dot{Q}_L}{\dot{Q}_H - \dot{Q}_L}$$

$$\Rightarrow \dot{W}_{in} = \frac{\dot{Q}_L}{COP_R}$$

$$\Rightarrow \dot{Q}_H - \dot{Q}_L = \frac{\dot{Q}_L}{COP_R}$$

$$\dot{Q}_H = \dot{Q}_L\left(1 + \frac{1}{COP_R}\right)$$

$$= (18)\left(1 + \frac{1}{6}\right)$$

$$= 21\,kW$$

Step 6 of 7

An energy balance on the condenser gives

Heat supplied is equal to the heat gain.

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = \dot{m}_w c_{pw}(T_{2w} - T_{1w})$$

$$T_{2w} = T_{1w} + \frac{\dot{Q}_H}{\dot{m}_w c_{pw}}$$

$$= 10^\circ C + \frac{21\,kW}{\left(0.32\frac{kg}{s}\right)\left(4.18\frac{kJ}{kg \cdot K}\right)}$$

$$= 25.69^\circ C$$

$$\boxed{T_{2w} = 25.69^\circ C}$$

Step 7 of 7

(c) The COP of the heat pump is

$$COP_{HP} = \frac{\dot{Q}_H}{\dot{Q}_H - \dot{Q}_L}$$

$$= \frac{21}{21 - 18}$$

$$= 7$$

$$\boxed{COP_{HP} = 7}$$

Question: 9.201P pg 1

A refrigeration unit operates on the ideal vapor compression refrigeration cycle and uses refrigerant-22 as the working fluid. The operating conditions for this unit are evaporator saturation temperature of -5°C and the condenser saturation temperature of 45°C. Selected data for refrigerant-22 are provided in the table below.

T, °C	P_{sat}, kPa	h_f, kJ/kg	h_g, kJ/kg	s_g, kJ/kg·K
−5	421.2	38.76	248.1	0.9344
45	1728	101	261.9	0.8682

For R-22 at P =1728 kPa and s =0.9344 kJ/kg-K,

T =68.15T and h =283.7 kJ/kg. Also, take

cp,air =1.005 kJ/kg-K.

(*a*) Sketch the hardware and the *T-s* diagram for this heat pump application.

(*b*) Determine the COP for this refrigeration unit.

(*c*)The evaporator of this unit is located inside the air handler of the building. The air flowing through the air handler enters the air handler at 27°C and is limited to a 20°C temperature drop. Determine the ratio of volume flow rate of air entering the air handler (*m*3air/min) to mass flow rate of R-22 (kgR-22/s) through the air handler, in $(m^3_{air}/\text{min})/(\text{kg}_{R\text{-}22}/\text{s})$. Assume the air pressure is 100 kPa.

Step 1 of 7

Given:

Evaporation saturation temperature $T_1 = -5°C$

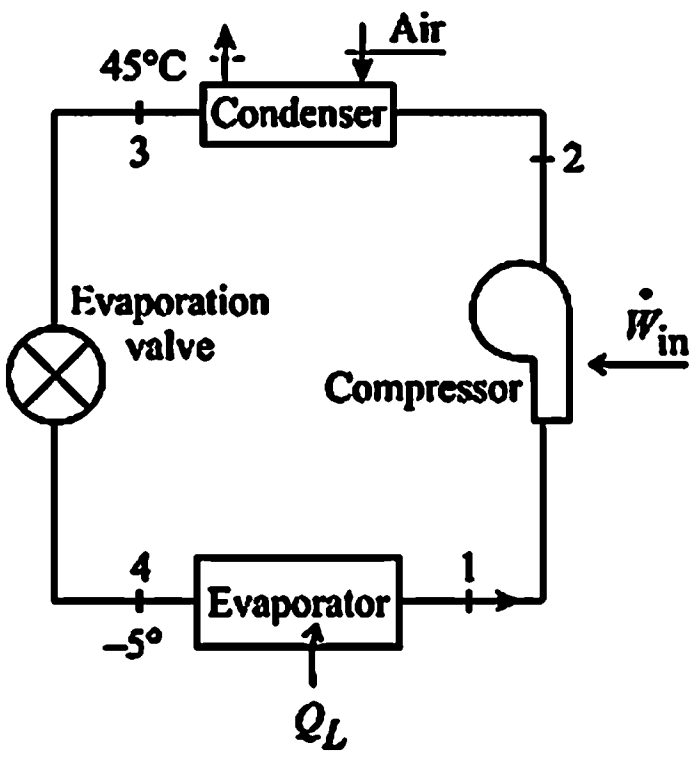
Condenser saturation temperature $T_2 = 45°C$

Air flowing through air handler is limited to $\Delta T = 20°C$

Step 2 of 7

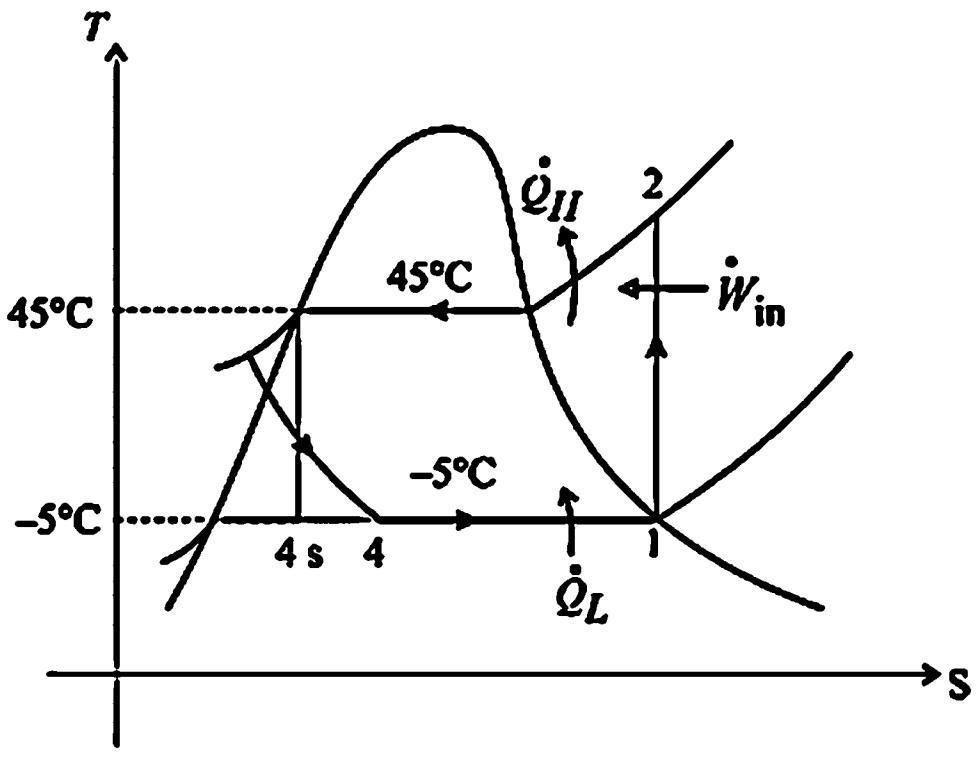
(a)

Hardware for heat pump application



Step 3 of 7

T-S diagram of heat pump application



Step 4 of 7

(b)

From the refrigerant-22 table

At $T_1 = -5°C$, saturated vapor

$h_1 = h_{g@-5°C} = 248.1 \text{kJ/kg}$, $s_1 = s_{g@-5°C} = 0.9344 \text{kJ/kg.K}$

At $P_2 = 1728 \text{kPa}$, $s_2 = s_1 = 0.9344 \text{kJ/kg.K}$

$h_2 = 283.7 \text{kJ/kg}$

At $P_3 = 1728 \text{kPa}$, saturated liquid

$h_3 = h_{f@1728\text{kPa}} = 101 \text{kJ/kg}$

For Throttling, $h_4 \cong h_3 = 101 \text{kJ/kg}$

Step 5 of 7

COP of the heat pump:

$$\text{COP}_{HP} = \frac{q_H}{w_{in}}$$

$$\text{COP}_{HP} = \frac{h_2 - h_3}{h_2 - h_1}$$

$$\text{COP}_{HP} = \frac{283.7 - 101}{283.7 - 248.1}$$

$$\boxed{\text{COP}_{HP} = 5.132}$$

Step 6 of 7

(c)

Ratio of volume flow rate of air entering the air handler to mass flow rate of R-22 through the air handler:

Energy balance on the condenser, $\dot{Q}_H = \dot{m}_R (h_2 - h_3)$ and $\dot{Q}_H = \dot{m}_a c_p \Delta T$

$$\dot{m}_R (h_2 - h_3) = \frac{\dot{V}_a}{v_a} c_p \Delta T$$

through the air handler: 9.201P pg 2

Energy balance on the condenser, $\dot{Q}_H = \dot{m}_R (h_2 - h_3)$ and $\dot{Q}_H = \dot{m}_a c_p \Delta T$

$$\dot{m}_R (h_2 - h_3) = \frac{\dot{V}_a}{v_a} c_p \Delta T$$

$$\frac{\dot{V}_a}{\dot{m}_R} = \frac{(283.7 - 101)}{(1/0.847) \times (1.005) \times (20)}$$

$$\frac{\dot{V}_a}{\dot{m}_R} = \frac{182.7}{23.7308}$$

$$\frac{\dot{V}_a}{\dot{m}_R} = 7.699 \left(m^3 air / s \right) / (kg R22 / s)$$

$$\boxed{\frac{\dot{V}_a}{\dot{m}_R} = 461.94 \left(m^3 air / \min \right) / (kg R22 / s)}$$

Step 7 of 7

Question: 9.202P

Consider an ice producing plant that operates on the ideal vapor-compression refrigeration cycle and uses refrigerant-134a as the working fluid. The refrigeration cycle operating conditions require an evaporator pressure of 140 kPa and the condenser pressure of 1200 kPa. Cooling water flows through the water jacket surrounding the condenser and is supplied at the rate of 200 kg/s. The cooling water has a 10°C temperature rise as it flows through the water jacket. To produce ice, potable water is supplied to the chiller section of the refrigeration cycle. For each kg of ice produced,333 kj of energy must be removed from the potable water supply |

(*a*)Sketch the hardware for all three working fluids of this refrigerant-ice making system and the *T-s* diagram for refrigeration cycle.

(*b*)Determine the mass flow rate of the refrigerant, in kg/s

(*c*)Determine the mass flow rate of the potable water supply, in kg/s.

Step 1 of 8

Given:

Working fluid: R-134a

Evaporator pressure $P_1 = P_4 = 140\text{kPa}$

Condenser pressure $P_2 = P_3 = 1200\text{kPa}$

Mass flow rate of water $\dot{m}_w = 200\text{kg/s}$

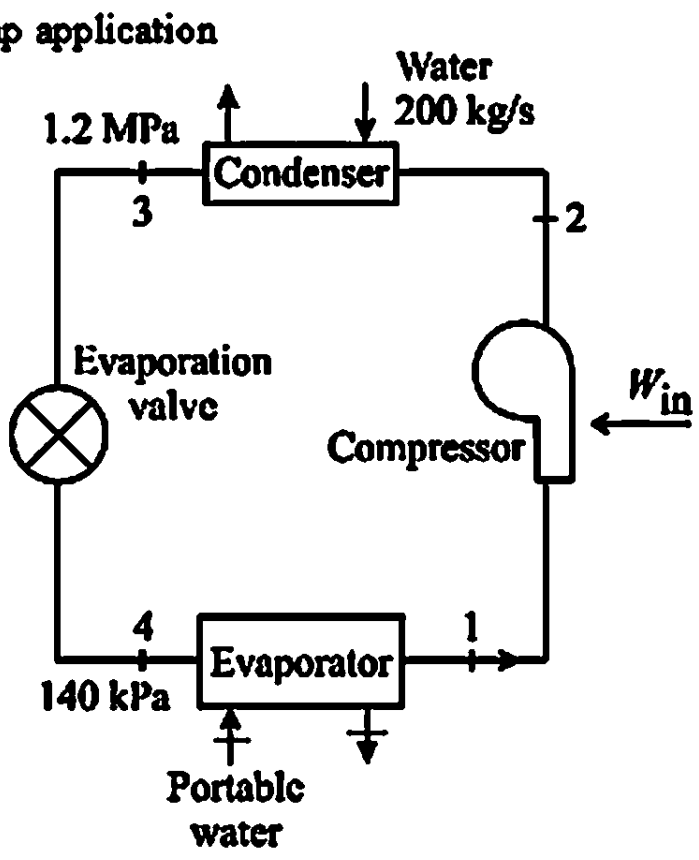
Rise in temperature of water $\Delta T = 10°\text{C}$

Amount of energy to be removed for each kg of ice $h_{if} = 333\text{kJ/kg}$

Step 2 of 8

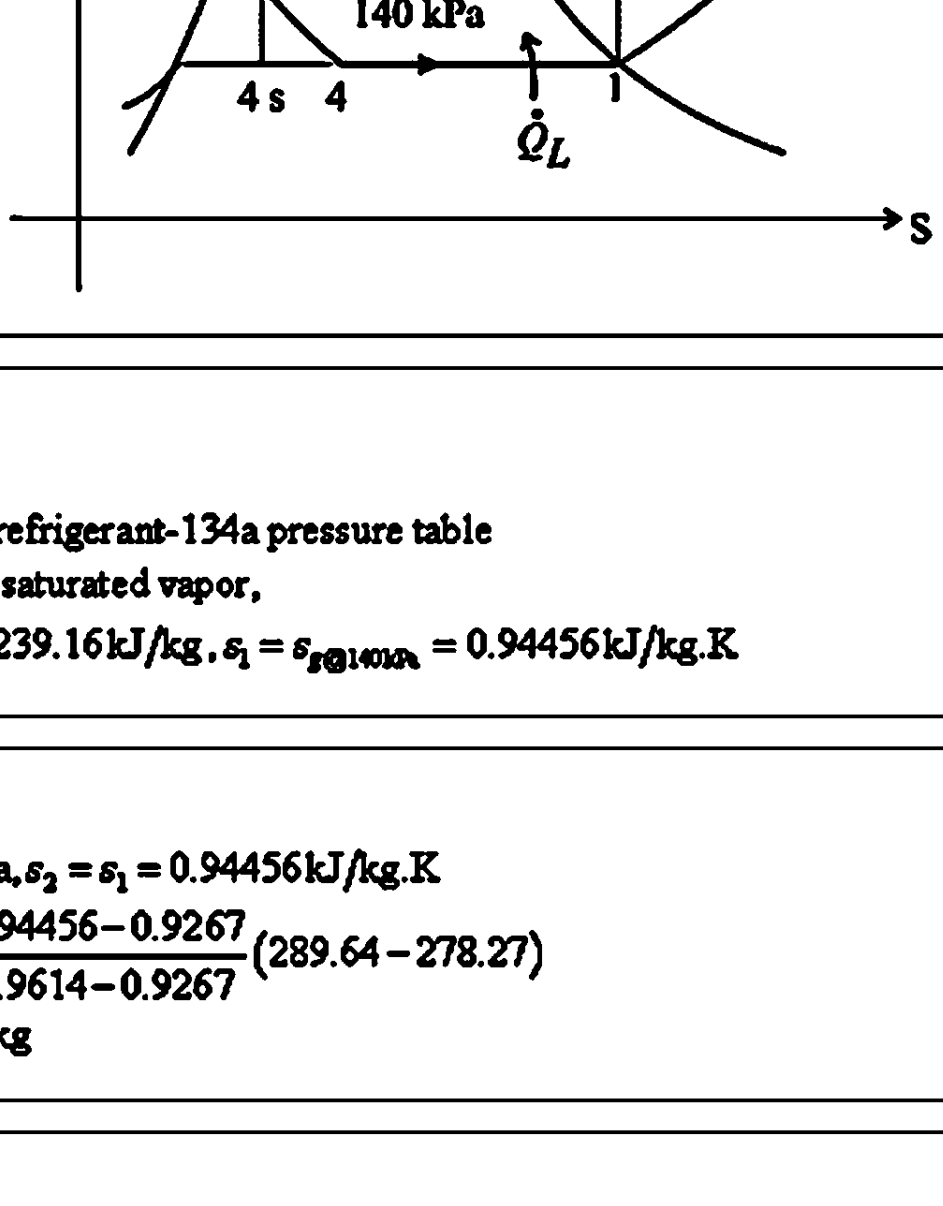
(a)

Hardware for Heat pump application



Step 3 of 8

T-S diagram of heat pump application



Step 4 of 8

(b)

From saturated refrigerant-134a pressure table

At $P_1 = 140\text{kPa}$, saturated vapor,

$h_1 = h_{g@140\text{kPa}} = 239.16\text{kJ/kg}$, $s_1 = s_{g@140\text{kPa}} = 0.94456\text{kJ/kg.K}$

Step 5 of 8

At $P_2 = 1200\text{kPa}$, $s_2 = s_1 = 0.94456\text{kJ/kg.K}$

$$h_2 = 278.27 + \frac{0.94456 - 0.9267}{0.9614 - 0.9267}(289.64 - 278.27)$$

$$h_2 = 284.12\text{kJ/kg}$$

Step 6 of 8

From saturated refrigerant-134a pressure table

At $P_3 = 1200\text{kPa}$, saturated liquid

$h_3 = h_{f@1200\text{kPa}} = 117.77\text{kJ/kg}$

For Throttling, $h_4 \cong h_3 = 117.77\text{kJ/kg}$

Step 7 of 8

Mass flow rates of the refrigerant:

Energy balance on the condenser, $\dot{Q}_H = \dot{m}_R(h_2 - h_3) = \dot{m}_w c_p \Delta T$

$$\dot{m}_R = \frac{\dot{m}_w c_p \Delta T}{h_2 - h_3}$$

$$\dot{m}_R = \frac{(200)\times(4.18)(10)}{(284.12 - 117.77)}$$

$$\dot{m}_R = \frac{8360}{166.35}$$

$$\boxed{\dot{m}_R = 50.255\text{ kg/s}}$$

Step 8 of 8

(c)

Mass flow rate of the potable water:

Energy balance on the evaporator, $\dot{Q}_L = \dot{m}_R(h_1 - h_4) = \dot{m}_w h_{if}$

$$\dot{m}_w = \frac{\dot{m}_R(h_1 - h_4)}{h_{if}}$$

$$\dot{m}_w = \frac{50.271\times(239.16 - 117.77)}{333}$$

$$\dot{m}_w = \frac{6102.39}{333}$$

$$\boxed{\dot{m}_w = 18.325\text{ kg/s}}$$

CHAPTER 10
SOLUTIONS

Question: 10.001P

Define internal, external, and open-channel flows.

Step 1 of 3

External flow: The flow of an unbounded fluid over a surface is known as external flow. The flow of water in river is example for external flow.

Step 2 of 3

Internal flow: If the fluid is completely bounded by a solid surface, then the flow is known as internal flow. Water flow in a pipe is example for internal flow.

Step 3 of 3

Open channel flow

The flow of liquids in a duct is called open channel flow, if the duct is partially filled liquid and there is a free surface. Internal flows are dominated by the influence of viscosity, where as in solid fluids viscous effects are limited to boundaries.

Discussion: Classification of flows is useful in flow analysis.

Question: 10.002P

Define incompressible flow and incompressible fluid. Must the flow of a compressible fluid necessarily be treated as compressible?

Step 1 of 2

Incompressible Flow:

Flow is incompressible means; the fluid density remains constant throughout. If the density changes in a flow, the density of such a fluid (like liquid) is practically independent.

Incompressible flow is satisfy the below equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho w}{\partial z}=0$$

If flow is satisfy the continuity equation then flow is called incompressible.

Step 2 of 2

Compressible Flow:

Flow is compressible means the density of the fluid changes in the fluid flow due to change in temperature and pressure.

Example: Gases are considered compressible fluid.

If flow is not satisfy the continuity equation then flow is called compressible.

Question: 10.003P

Consider the flow of air over the wings of an aircraft. Is this flow internal or external? How about the flow of gases through a ajet engine?

Step 1 of 2

External flow: The flow of an unbounded fluid over a surface is known as External flow. The flow of water in river is an example for external flow.

Therefore we can say that the air flow over the wings of aircraft is external flow. Here the flow is over unbounded surface.

Step 2 of 2

Internal flow: If the fluid is completely bounded by a solid surface, then the flow is known as internal flow. Water flow in a pipe is example for internal flow.

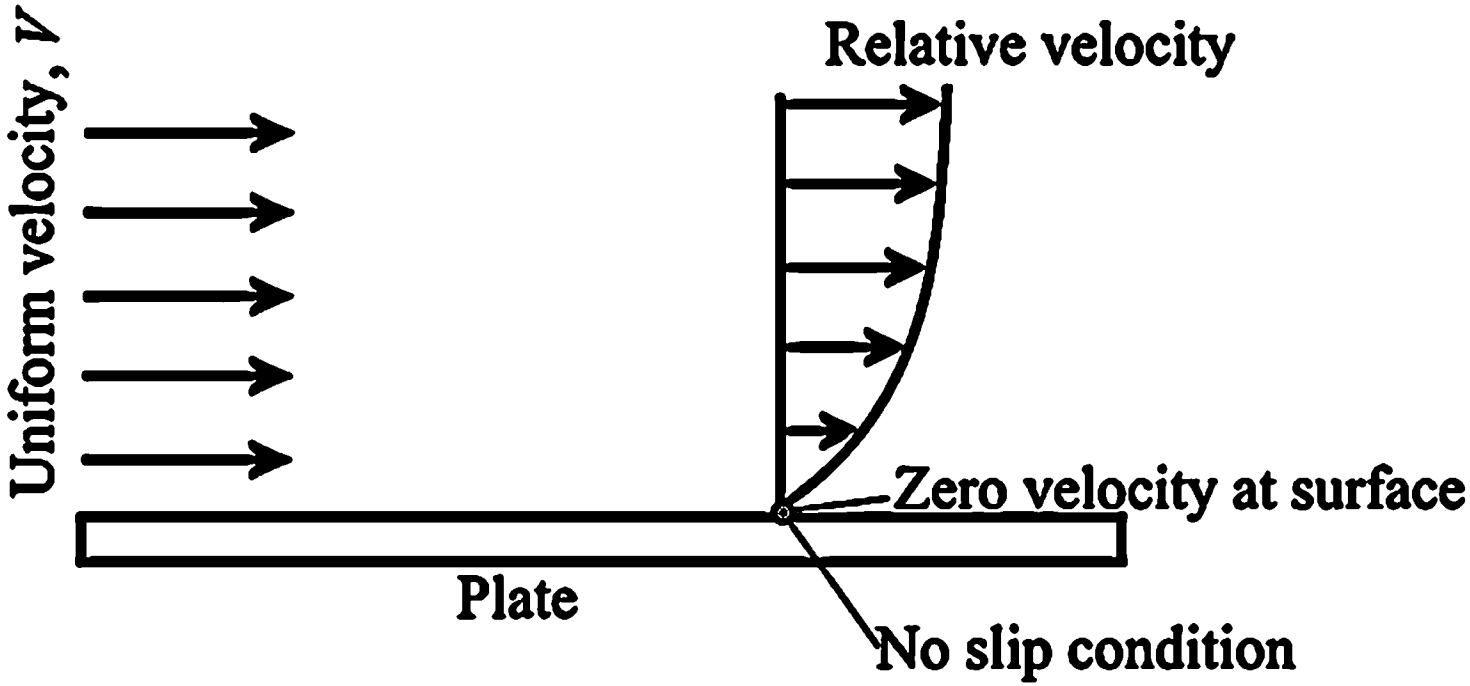
Flow of gases through jet engine is internal flow because in engines, fluids (gases) are bounded by solid surface.

Question: 10.004P

What is the no-slip condition? What causes it?

Step 1 of 2

Sketch the no slip condition exits, when a fluid flows over the plate.



Step 2 of 2

Consider the surface up on which the fluid flows as nonporous. Thus when the fluid in motion comes in contact with a nonporous surface then the fluid approaches to a complete stop at the surface and considers a zero velocity relative to the surface. That is a fluid in direct contact with solid, sticks to the surface, and there will be no slip occurs on the surface. This condition is called to be as no-slip condition.

Viscosity of the fluid is responsible for no-slip condition to occur.

Question: 10.005P

What is forced flow? How does it differ from natural flow? Is flow caused by winds forced or natural flow?

Step 1 of 1

Forced Flow:
If a fluid is forced to flow over a surface or in a pipe by means of external source such as pump or fan, then the flow is known as forced flow.

Natural Flow:
In natural flow, the fluid motion is due to natural means such as buoyancy effect. Flow caused by winds is natural flow. This flow is due to change in temperatures and pressures in the atmosphere, so the flow of wind is natural flow.

Question: 10.006P

What is a boundary layer? What causes a boundary layer to develop?

Step 1 of 2

Boundary layer:
At the surface, the fluid is at rest. The velocity of the fluid is zero at the surface and gradually increases. The flow region adjacent to the wall in which the viscous effects are significant is called the boundary layer.

Step 2 of 2

Boundary layer is developed due to the no-slip condition. No slip condition means, the fluid motion is zero at the contact surface of the solid. The property viscosity is responsible for no slip condition and development of boundary layer. Viscosity gives the relation between rate of deformation and shear stress. It is the main cause of boundary layer.

Question: 10.007P

What is vapor pressure? How is it related to saturation pressure?

Step 1 of 1

The vapor pressure of a pure substance is defined as the pressure exerted by its vapor in an equilibrium phase with its liquid at a given temperature. It is denoted as P_v. At a given temperature, the pressure at which the substance changes its phase is known as the saturation pressure, P_{sat}.

Relation between both is given as

$P_v = P_{sat}$ for pure vapor.

Question: 10.008P

Does water boil at higher temperatures at higher pressures? Explain.

Step 1 of 1

The saturation temperature of the pure substance depends upon pressure; when pressure increases saturation temperature also increases.

For example: Water boils at 134° C in a pressure cooker operating at 3 atm absolute pressure, but it boils at 93° C in an ordinary pan at 2000 m elevation where the atmospheric pressure is 0.8 atm. Therefore the vapour pressure increases with temperature.

Question: 10.009P

If the pressure of a substance is increased during a boiling process, will the temperature also increase or will it remain constant? Why?

Step 1 of 1

Analysis:

If the pressure of the substance increases during a boiling process, the temperature will also increase.

During boiling, the liquid is converted into vapor so we can apply the ideal gas equation. Here the pressure and temperature are linearly proportional, therefore as pressure increases, the temperature also increases.

Question: 10.010P

What is cavitation? What causes it?

Step 1 of 1

Cavitation is the phenomenon of formation of vapor bubbles of a flowing liquid in a region where the pressure of the liquid falls below its vapor pressure at that temperature and the sudden collapsing of these vapor bubbles in a region of higher pressure. When these vapor bubbles collapse, a very high pressure is created which causes pitting on the metallic surfaces termed as cavitation.

Question: 10.011P

In a piping system, the water temperature remains under 35°C. Determine the minimum pressure allowed in the system to avoid cavitation.

Step 1 of 1

Given data:

The temperature of water, $t = 35^\circ C$

Let the minimum pressure that can exist in the piping system to avoid cavitations be P_{min}.

The vapor pressure of water at, $t = 35^\circ C$ is

$P = 5.63 kPa$.

i.e. $P_{min} = 5.63\ kPa$

Therefore the pressure should not be maintained more than $P = 5.63 kPa$ in given water piping system to avoid cavitation.

Question: 10.012P

The analysis of a propeller that operates in water at 20°C shows that the pressure at the tips of the propeller drops to 2 kPa at high speeds. Determine if there is a dange of cavitation for this propeller.

Step 1 of 6

Given data :

The temperature of water, $t = 20°C$

The actual pressure at the tip of the propeller, $P_{act} = 2\,kPa$

Step 2 of 6

Obtain the saturation pressure of water $20°C$ from table A-4, "Saturated water-temperature table "from the textbook.

$P = 2.34\,kPa$

Step 3 of 6

At the given temperature, to avoid cavitation, the pressure in the flow must be greater than the vapor (or saturation) pressure.

Step 4 of 6

Since the actual pressure $P_{act} = 2\,kPa$ is less than the vapor pressure of the water at $20°C$, there is a chance of cavitation.

Step 5 of 6

Hence, there is a danger of cavitation for this propeller.

Step 6 of 6

Question: 10.013P

The analysis of a propeller that operates in water at 70°F shows that the pressure at the tips of the propeller drops to 0.1 psia at high speeds. Determine if there is a danger of cavitation for this propeller.

Step 1 of 2

Given data:

The temperature of water, $t = 70°F$

The actual pressure at the tip of the propeller, $P_{act} = 0.1 psia$

Step 2 of 2

From the saturated water temperature table at $t = 70°F$ the given values of saturated pressure is $P_{sat} = 0.3632$ psia

At the given temperature, to avoid cavitation, pressure in the flow must be greater than the vapour (or saturation) pressure.

There is danger of cavitation in the pump because the minimum pressure in the pump is $P_{act} = 0.1 psia$. This is less than the vapour pressure.

Question: 10.014P

What is viscosity? What is the cause of it in liquids and in gases? Do liquids or gases have higher dynamic viscosities?

Step 1 of 1

Viscosity is defined as the property of fluid which offers resistance to the movement of one layer of the fluid over another adjacent layer.

The viscous forces in a fluid are due to cohesive forces and momentum transfer. In liquids, the cohesive force predominates the molecular momentum transfer, due to closely packed molecules and with the increase in temperature, the cohesive force decreases with the result of decreasing viscosity.

In case of gases, the cohesive forces are small and molecular momentum transfer predominates. With the increase in temperature, molecular momentum transfer increases and hence viscosity increases.

Viscosity is the measure of resistance of fluid. Viscosity of liquids is more than viscosity of gases.

Question: 10.015P

What is a Newtonian fluid? Is water a Newtonian fluid?

Step 1 of 1

Fluid which satisfy the Newton's law of viscosity is known as Newtonian fluid. Newton's law of viscosity states that a fluid's shear stress is proportional to its velocity gradient.

$$\tau = \mu \frac{du}{dy}$$

Here

μ : Coefficient of viscosity or dynamic viscosity of the fluid.

Water is an example of Newtonian fluid.

Question: 10.016P

Consider two identical small glass balls dropped into two identical containers, one filled with water and the other with oil. Which ball will reach the bottom of the container first? Why?

Step 1 of 1

From the property of viscosity ball is dropped in the water container will reach faster to the bottom than the ball dropped in oil container.

The viscosity of water is less than oil. So the viscosity of oil restricts the motion of ball.

Question: 10.017P

How does the dynamic viscosity of (*a*) liquids and (*b*) gases vary with temperature?

Step 1 of 1

Dynamic viscosity is depends upon temperature.

Dynamic Viscosity of liquids decreases with increases in temperature, where as in gases dynamic viscosity increases with increase in temperature. In liquids cohesive force dominates the molecular momentum. In gases molecular momentum dominates the cohesive force.

When temperature increases cohesive force decreases in liquids so dynamic viscosity decreases.

In gases molecular momentum increases with increase in temperature so the dynamic viscosity increases.

Question: 10.018P

How does the kinematic viscosity of (*a*) liquids and (*b*) gases vary with temperature?

Step 1 of 1

Kinematics viscosity is depends on the temperature .Kinematic Viscosity of liquids reduces with raise in temperature, where as in gases kinematic viscosity raises with raise in temperature.

Question: 10.019P

For flow over a plate, the variation of velocity with vertical distance y from the plate is given as $u(y) = ay - by2$ where a and b are constants. Obtain a relation for the wall shear stress in terms of a, b, and μ.

Step 1 of 1

The velocity distribution is given as

$$u(y) = ay - by^2$$

Now, the velocity gradient is

$$\frac{du}{dy} = a - 2by$$

Shear stress would be:

$$\tau = \mu \frac{du}{dy}$$

The shear stress $\boxed{\tau = \mu(a - 2by)}$

Question: 10.020P

A 50-cm × 30-cm × 20-cm block weighing 150 N is to be moved at a constant velocity of 0.8 m/s on an inclined surface with a friction coefficient of 0.27. (*a*) Determine the force *F* that needs to be applied in the horizontal direction. (*b*) If a 0.4-mm-thick oil film with a dynamic viscosity of 0.012 Pa•s is applied between the block and inclined surface, determine the percent reduction in the required force.

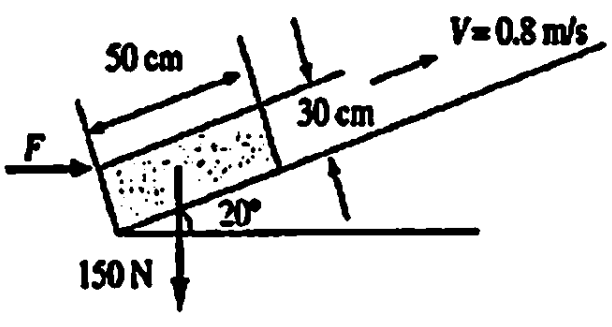


FIGURE P10–20

Step 1 of 9

Given data:

The weight of the block is $W = 150\ \text{N}$.

The velocity of the block is constant and is $v = 0.8\ \text{m/s}$.

The coefficient of friction of the surface is $\mu_s = 0.27$.

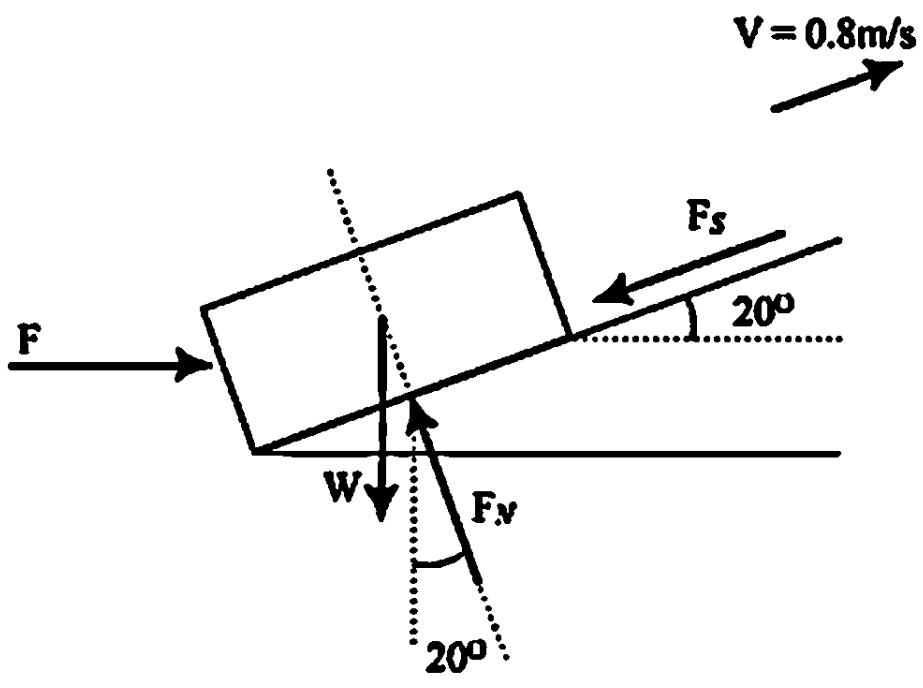
The thickness of the oil film is $\ell = 0.4\ \text{mm}$.

The dynamic viscosity between the block and inclined surface is $\mu = 0.012\ \text{Pa}\cdot\text{s}$.

The velocity $V = 0.8\ \text{m/s}$.

Step 2 of 9

Let us draw the free body diagram of the block.



Step 3 of 9

(a) Apply the condition of equilibrium.

Then

$$+\uparrow \sum F_y = 0$$

$$F_N \cos 20° - W - F_S \sin 20° = 0$$

Where

The frictional force applied by the surface

$$F_S = \mu_s F_N$$

Then

$$F_N \cos 20° - W - \mu_s F_N \sin 20° = 0$$

$$0.939F_N - (0.27)F_N(0.342) = 150$$

$$F_N = 177.16\ \text{N}$$

Step 4 of 9

And

$$+\rightarrow \sum F_x = 0$$

$$F - F_N \sin 20° - F_S \cos 20° = 0$$

Then

$$F = F_N \sin 20° + \mu_s F_N \cos 20°$$

$$= (0.342 + 0.27 \times 0.939)F_N$$

$$= 105.5\ \text{N}$$

Hence the required horizontal force to be applied is $\boxed{F = 105.5\ \text{N}}$.

Step 5 of 9

(b)

An oil film of thickness 4 mm and dynamic viscosity of $\mu = 0.012\ \text{Pa}\cdot\text{s}$ is introduced between the block and surface.

The shear force exerted

$$F_{sf} = \mu A \frac{V}{\ell}$$

Where the area of contact between the oil film and block

$$A = (50 \times 20)\ \text{cm}^2$$

$$= 1000\ \text{cm}^2$$

$$= 0.1\ \text{m}^2$$

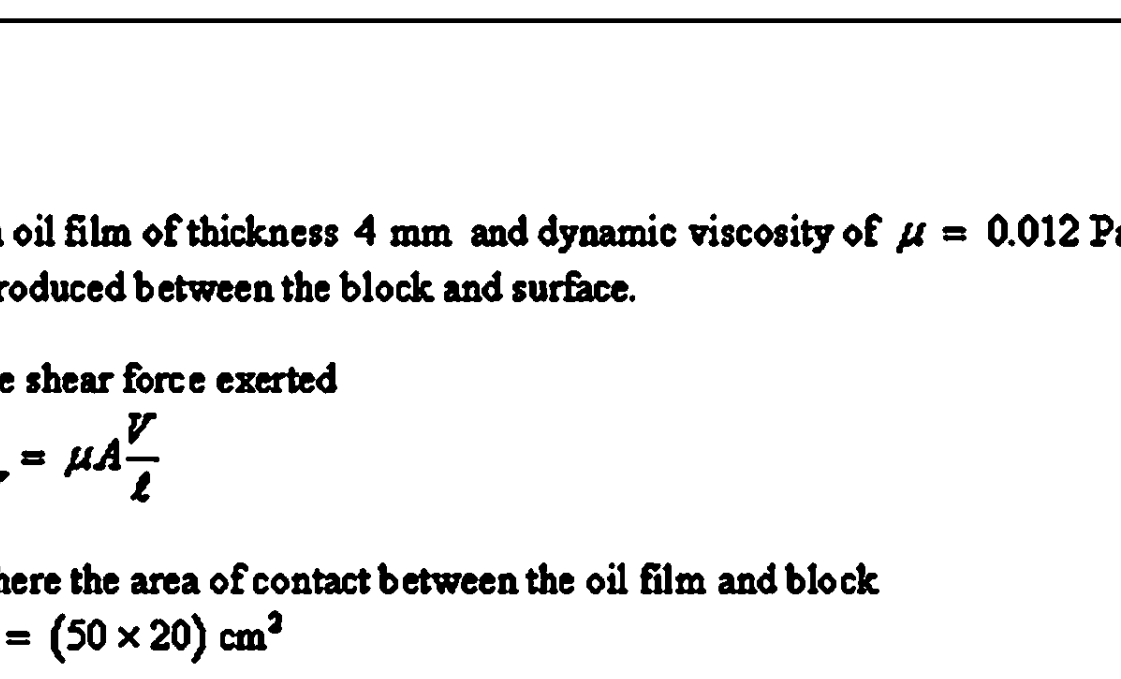
Then

$$F_{sf} = (0.012)(0.1)\frac{0.8}{(0.4 \times 10^{-3})}$$

$$= 2.4\ \text{N}$$

Step 6 of 9

Let us draw the free body diagram



Step 7 of 9

Apply the condition of equilibrium.

Then

$$+\uparrow \sum F_y = 0$$

$$F_N \cos 20° - W - F_{sf} \sin 20° = 0$$

$$F_N \cos 20° - 150 - 2.4 \sin 20° = 0$$

$$F_N = 160.5\ \text{N}$$

Step 8 of 9

And

$$+\rightarrow \sum F_x = 0$$

$$F - F_N \sin 20° - F_{sf} \cos 20° = 0$$

Then

$$F = F_N \sin 20° + F_{sf} \cos 20°$$

$$= 160.5 \sin 20° + 2.4 \cos 20°$$

$$= 57.14\ \text{N}$$

The required horizontal force to be applied when the oil film is used,

$F = 57.14\ \text{N}$.

Step 9 of 9

The percentage reduction in applied force is

$$\%\text{reduction} = \left(\frac{105.5\ \text{N} - 57.14\ \text{N}}{105.5\ \text{N}}\right) \times 10$$

$$= 45.83\%$$

Hence the percentage reduction in the applied force when the oil film is used is

$\boxed{\text{Percentage of reduction} = 45\%}$.

Question: 10.021P

Consider the flow of a fluid with viscosity μ through a circular pipe. The velocity profile in the pipe is given as $u(r) = u\max(1 - rn/Rn)$, where $u\max$ is the maximum flow velocity, which occurs at the centerline; r is the radial distance from the centerline; and $u(r)$ is the flow velocity at any position r. Develop a relation for the drag force exerted on the pipe wall by the fluid in the flow direction per unit length of the pipe.

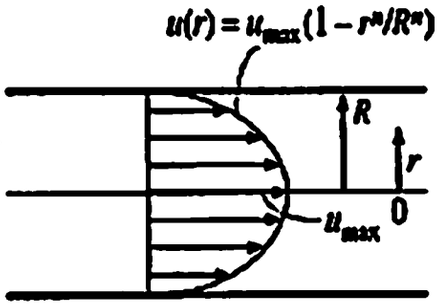


FIGURE P10–21

Step 1 of 2

Velocity profile in the pipe, $u(r) = u_{max}\left(1 - \frac{r^n}{R^n}\right)$

Drag force, $F = \mu A \frac{u}{y}$

μ is the coefficient of viscosity

Area of the profile $A = 2\pi Rl$

Velocity gradient

$$\frac{u}{y} = \frac{du}{dy}$$

$$= -\frac{du}{dr}$$

$$= -\frac{d}{dr}\left[u_{max}\left(1 - \frac{r^n}{R^n}\right)\right]$$

$$= -u_{max}\left(\frac{-nr^{n-1}}{R^n}\right)_{r=R}$$

$$= \frac{u_{max} n}{R}$$

Step 2 of 2

So drag force becomes;

$$F = \mu(2\pi RL)\left(\frac{u_{max} n}{R}\right)$$

$$= 2\pi n u_{max} \mu L$$

$$\frac{F}{L} = 2\pi n u_{max} \mu$$

Therefore drag force per unit length is $\boxed{2\pi n u_{max} \mu}$

Question: 10.022P

A thin 40-cm × 40-cm flat plate is pulled at 2 m/s horizontally through a 3.6-mm-thick oil layer sandwiched between two plates, one stationary and the other moving at a constant velocity of 0.3 m/s, as shown in Fig. P10–22. The dynamic viscosity of oil is 0.027 Pa·s. Assuming the velocity in each oil layer to vary linearly, (*a*) plot the velocity profile and find the location where the oil velocity is zero and (*b*) determine the force that needs to be applied on the plate to maintain this motion.

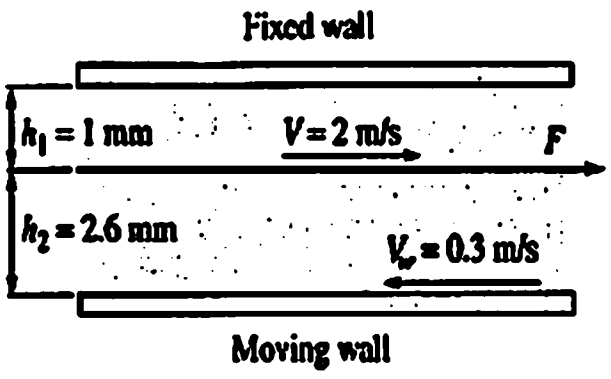


FIGURE P10–22

Step 1 of 5

Given data:

Dimensions of the thin flat plate, 40 cm ×40 cm

The velocity of the thin plate, $V = 2\,m/s$

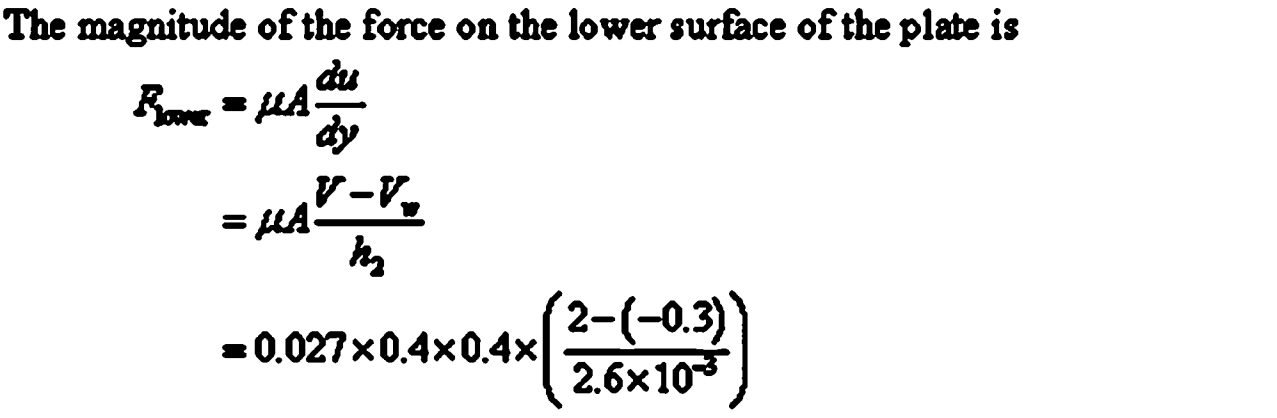
Thickness of the oil layer, 3.6 mm

Dynamic viscosity of oil, $\mu = 0.027\,N.s/m^2$

Velocity of the moving plate, $V_w = 0.3\,m/s$

Step 2 of 5

a) Consider a point O at a distance of x from the moving wall where the velocity is zero as shown in figure



Step 3 of 5

By symmetric triangles below the thin plate

$$\frac{2.6-x}{x} = \frac{2}{0.3}$$

$$(2.6-x)\times 0.3 = 2x$$

$$0.78 - 0.3x = 2x$$

$$0.78 = 2.3x$$

$$x = \frac{0.78}{2.3}$$

$$\boxed{x = 0.339 \text{ mm}}$$

Therefore the location where the oil velocity is zero, $x = 0.339$ mm .

Step 4 of 5

(b) The magnitude of the force on the upper surface of the plate

$$F_{upper} = \mu A \frac{du}{dy}$$

$$= \mu A \left(\frac{V-0}{h_1}\right)$$

$$= 0.027\times 0.4\times 0.4\times \left(\frac{2}{1\times 10^{-3}}\right)$$

$$F_{upper} = 8.64\text{ N}$$

Therefore force on the upper surface of the plate, $\boxed{F_{upper} = 8.64\text{ N}}$.

Step 5 of 5

The magnitude of the force on the lower surface of the plate is

$$F_{lower} = \mu A \frac{du}{dy}$$

$$= \mu A \frac{V - V_w}{h_2}$$

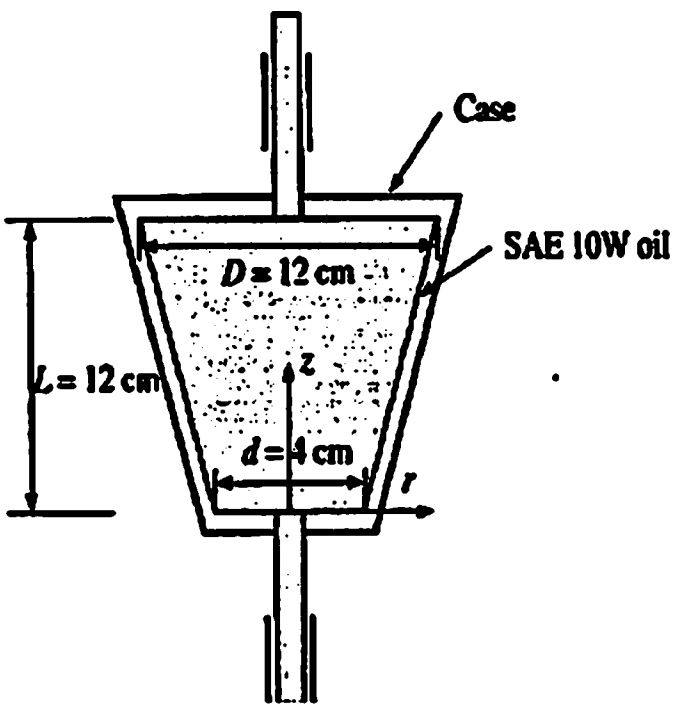
$$= 0.027\times 0.4\times 0.4\times \left(\frac{2-(-0.3)}{2.6\times 10^{-3}}\right)$$

$$F_{lower} = 3.82\text{ N}$$

Therefore force on the lower surface of the plate, $\boxed{F_{lower} = 3.82\text{ N}}$

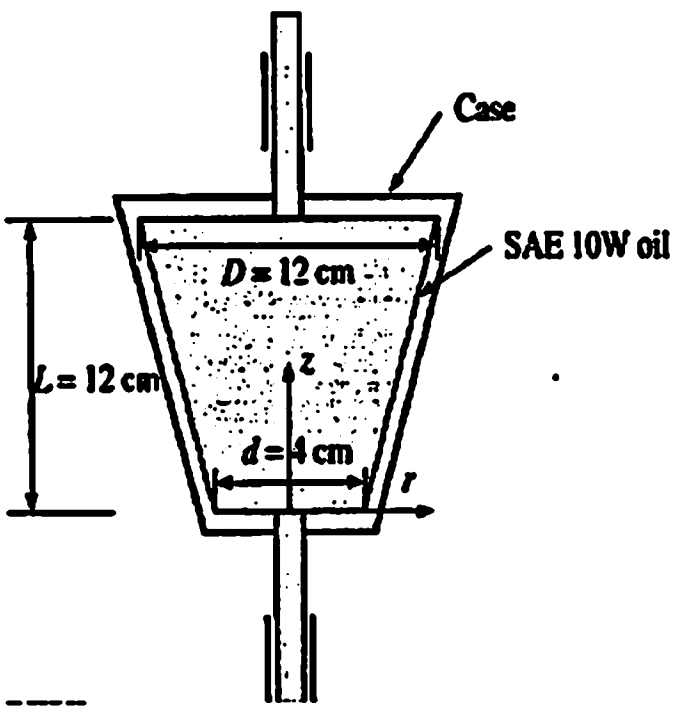
Question: 10.023P pg 1

A frustum-shaped body is rotating at a constant angular speed of 200 rad/s in a container filled with SAE 10W oil at 20°C ($\mu = 0.1$ Pa·s), as shown in Fig. P10–23. If the thickness of the oil film on all sides is 1.2 mm, determine the power required to maintain this motion. Also determine the reduction in the required power input when the oil temperature rises to 80°C ($\mu = 0.0078$ Pa·s).



Question:

A frustum-shaped body is rotating at a constant angular speed of 200 rad/s in a container filled with SAE 10W oil at 20°C ($\mu = 0.1$ Pa·s), as shown in Fig. P10–23. If the thickness of the oil film on all sides is 1.2 mm, determine the power required to maintain this motion. Also determine the reduction in the required power input when the oil temperature rises to 80°C ($\mu = 0.0078$ Pa·s).



$$\tau = \mu\frac{du}{dr}$$
$$= \mu\frac{V}{h}$$
$$= \mu\frac{\omega r}{h} \qquad \ldots\ldots (1)$$

Step 3 of 10

The shear force acting on the surface is
$F = \tau A$
Consider an area dA on which the shear force acts.
Then expression can be written as
$dF = \tau dA \qquad \ldots\ldots (2)$
Substituting equation (1) in equation (2), we get
$$dF = \mu\frac{\omega r}{h}dA$$
Using relation,
The torque is given by
$T = FR$
For a radius $R = r$ and force dF, the torque generated is
$dT = (dF)(r) \qquad \ldots\ldots (3)$
Substitute the dF in equation (3), we get
$$dT = \left(\mu\frac{\omega r}{h}dA\right)(r)$$
$$= \mu\frac{\omega r^2}{h}dA$$

Step 4 of 10

We have for power,
$P = T\omega \qquad \ldots\ldots (4)$
Where
Where
$$T = \int dT$$
$$= \int \mu\frac{\omega r^2}{h}dA$$
Substituting T in equation (4), we get
$$P = \left(\int \mu\frac{\omega r^2}{h}dA\right)\omega$$
$$= \int \mu\frac{\omega^2 r^2}{h}dA$$
$$= \mu\frac{(\omega r)^2}{h}\int dA$$

Step 5 of 10

Consider the side surface of the frustum shaped body.

We can express the power as
$$P_{side} = \mu\frac{(\omega r)^2}{h}\int dA \qquad \ldots\ldots (5)$$
Where
$dA = (2\pi r)dz$
The mean radius for the body can be expressed as
$$r = \frac{d}{2} + \frac{D-d}{2}$$

$$P_{side} = \mu\frac{(\omega r)^2}{h}\int dA \qquad \ldots\ldots (5)$$

Where

$$dA = (2\pi r)dz$$

The mean radius for the body can be expressed as

$$r = \frac{d}{2} + \frac{D-d}{2L}$$

Differentiating r with respect to z

$$dr = \frac{D-d}{2L}dz$$

$$dz = \frac{2L}{D-d}dr$$

Step 6 of 10

Substituting dz in dA, we get

$$dA = 2\pi r(dz)$$

$$= 2\pi r\left(\frac{2L}{D-d}dr\right)$$

$$= (rdr)\left(\frac{4L\pi}{D-d}\right)$$

Substituting r and dA in equation (5), we get

$$P_{side} = \mu\frac{(\omega r)^2}{h}\int dA$$

The top radius is $D/2$ and the bottom radius is $d/2$.

$$P_{side} = \mu\frac{(\omega r)^2}{h}\int_{d/2}^{D/2} dA$$

$$= \mu\frac{(\omega r)^2}{h}\int_{d/2}^{D/2}(rdr)\left(\frac{4L\pi}{D-d}\right)$$

$$= \frac{\mu\omega^2}{h}\left(\frac{4L\pi}{D-d}\right)\int_{d/2}^{D/2} r^3 dr$$

Step 7 of 10

Applying the limits to integration, we get

$$P_{side} = \frac{\mu\omega^2}{h}\left(\frac{4L\pi}{D-d}\right)\left(\frac{r^4}{4}\right)_{d/2}^{D/2}$$

$$= \frac{\mu\omega^2\pi L}{h(D-d)}\left(\frac{D^4-d^4}{16}\right)$$

Consider the top surface.
We can express the power as

$$P_{top} = \mu\frac{(\omega r)^2}{h}\int dA \qquad \ldots\ldots (6)$$

Where the area

$$dA = (2\pi r)dr$$

Substituting dA in equation (6), we get

$$P_{top} = \mu\frac{(\omega r)^2}{h}\int 2\pi r dr$$

The top radius is $D/2$ and the bottom radius is 0.

$$P_{top} = \mu\frac{(\omega r)^2}{h}\int_0^{D/2} 2\pi r dr$$

$$= \mu\frac{2\pi\omega^2}{h}\left(\frac{r^4}{4}\right)_0^{D/2}$$

$$= \mu\frac{\pi\omega^2 D^4}{32h}$$

Step 8 of 10

Consider the bottom surface.
We can express the power as

$$P_{bottom} = \mu\frac{(\omega r)^2}{h}\int dA \qquad \ldots\ldots (7)$$

Where the area

$$dA = (2\pi r)dr$$

Substituting dA in equation (7), we get

$$P_{top} = \mu\frac{(\omega r)^2}{h}\int 2\pi r dr$$

The top radius is $d/2$ and the bottom radius is 0.

$$P_{top} = \mu\frac{(\omega r)^2}{h}\int_0^{d/2} 2\pi r dr$$

$$= \mu\frac{2\pi\omega^2}{h}\left(\frac{r^4}{4}\right)_0^{d/2}$$

$$= \mu\frac{\pi\omega^2 d^4}{32h}$$

Step 9 of 10

The total power required to maintain the motion is

$$P_{total} = P_{side} + P_{top} + P_{bottom}$$

$$= \left[\frac{\mu\omega^2\pi L}{h(D-d)}\left(\frac{D^4-d^4}{16}\right)\right] + \left[\mu\frac{\pi\omega^2 D^4}{32h}\right] + \left[\mu\frac{\pi\omega^2 d^4}{32h}\right]$$

$$= \frac{\mu\omega^2\pi}{16h}\left[L\frac{(D^4-d^4)}{D-d} + \left(\frac{D^4+d^4}{2}\right)\right]$$

Substituting the given values in P_{total}, we get

$$P_{total} = \frac{(0.1)(200)^2\pi}{16(0.0012)}\left[(0.12)\frac{(0.12^4-0.04^4)}{0.12-0.04} + \left(\frac{0.12^4+0.04^4}{2}\right)\right]$$

$$= 269.75\ \text{W}$$

The power required to maintain the motion at 20°C is

$$\boxed{(P_{total})_{20} = 269.75\ \text{W}}$$

$= 269.75\ \text{W}$

The power required to maintain the motion at 20°C is

$$\boxed{(P_{\text{total}})_{20} = 269.75\ \text{W}}$$

Step 10 of 10

The power required to maintain the motion when $\mu = 0.0078\ \text{Pa}\cdot s$ and the temperature 80°C can be expressed as

The power associated is directly proportional to the viscosity of the fluid.

$$\frac{(P_{\text{total}})_{20}}{(P_{\text{total}})_{80}} = \frac{\mu_{20}}{\mu_{80}}$$

$$\frac{(P_{\text{total}})_{20}}{(P_{\text{total}})_{80}} = \frac{0.1}{0.0078}$$

$$\frac{(P_{\text{total}})_{80}}{269.75} = \frac{0.0078}{0.1}$$

$$(P_{\text{total}})_{80} = \frac{0.0078}{0.1}(269.75)$$

$$= 21.04\ \text{W}$$

The reduction in the required power input when the oil temperature is 80°C

$$\boxed{(P_{\text{total}})_{80} = 21.04\ \text{W}}.$$

Question: 10.024P

The clutch system shown in Fig. P10–24 is used to transmit torque through a 2-mm-thick oil film with μ = 0.38 N·s/m2 between two identical 30-cm-diameter disks. When the driving shaft rotates at a speed of 1450 rpm, the driven shaft is observed to rotate at 1398 rpm. Assuming a linear velocity profile for the oil film, determine the transmitted torque.

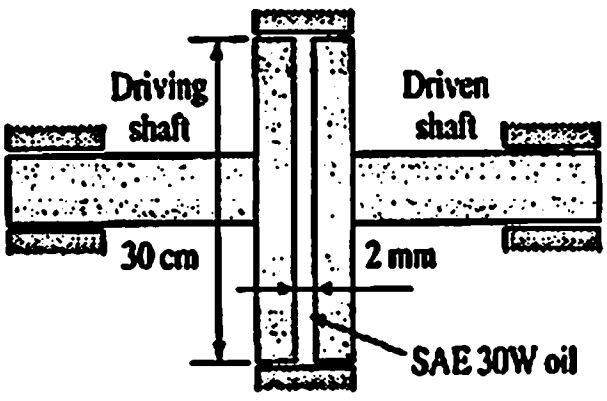


FIGURE P10–24

Step 1 of 4

Given data:

Oil film thickness, $h = 2\text{mm} = 0.002\text{m}$

The absolute viscosity of SAE30 oil is $\mu = 0.38\text{N.s/m}^2$

The Speed of the driving shaft, $n_1 = 1450\text{rpm}$

The Speed of the driven shaft, $n_2 = 1398\text{rpm}$

The diameter of a clutch $D = 30\text{cm} = 0.3\text{m}$

Step 2 of 4

Assume that one of the plate is stationary, linear velocity of other plate $V = (\omega_1 - \omega_2)r$

Consider a small area dA on the surface of the disc.

Then the shear force on the disc is on the area dA is $dF = \mu(dA)\dfrac{V}{h}$

Here $dA = (2\pi r)dr$

Now

$$dF = \mu(dA)\frac{V}{h}$$

$$= \mu(2\pi r)\frac{(\omega_1 - \omega_2)r}{h}dr$$

Now the torque on the small area dA is $dT = (dF)r$

Now $dT = \mu(2\pi)\dfrac{(\omega_1 - \omega_2)r^3}{h}dr$

Step 3 of 4

By integrating the dT we get torque on the disc

$$T = \int_{r=0}^{D/2} dT$$

$$= \int_{r=0}^{D/2} \mu 2\pi \frac{(\omega_1 - \omega_2)r^3}{h}dr$$

$$= \mu 2\pi \frac{(\omega_1 - \omega_2)}{h}\int_0^{D/2} r^3 dr$$

$$= \mu 2\pi \frac{(\omega_1 - \omega_2)}{h}\left(\frac{r^4}{4}\right)_0^{D/2}$$

$$T = \frac{\pi\mu(\omega_1 - \omega_2)D^4}{32h}$$

Step 4 of 4

Calculating:

$$\omega_1 - \omega_2 = 2\pi(n_1 - n_2)$$

$$= 2\pi(1450 - 1398)\frac{1}{60}$$

$$= 5.445\text{rad/s}$$

Also absolute viscosity of SAE30 oil is $0.38\text{N}\cdot\text{s/m}^2$

By substituting the values we get

$$T = \frac{\pi\mu(\omega_1 - \omega_2)D^4}{32h}$$

$$= \frac{0.38\pi \times 5.445 \times (0.3)^4}{32 \times 0.002}$$

$$= 0.823\text{N}\cdot\text{m}$$

Therefore transmitted torque is $\boxed{T = 0.823\text{N}\cdot\text{m}}$

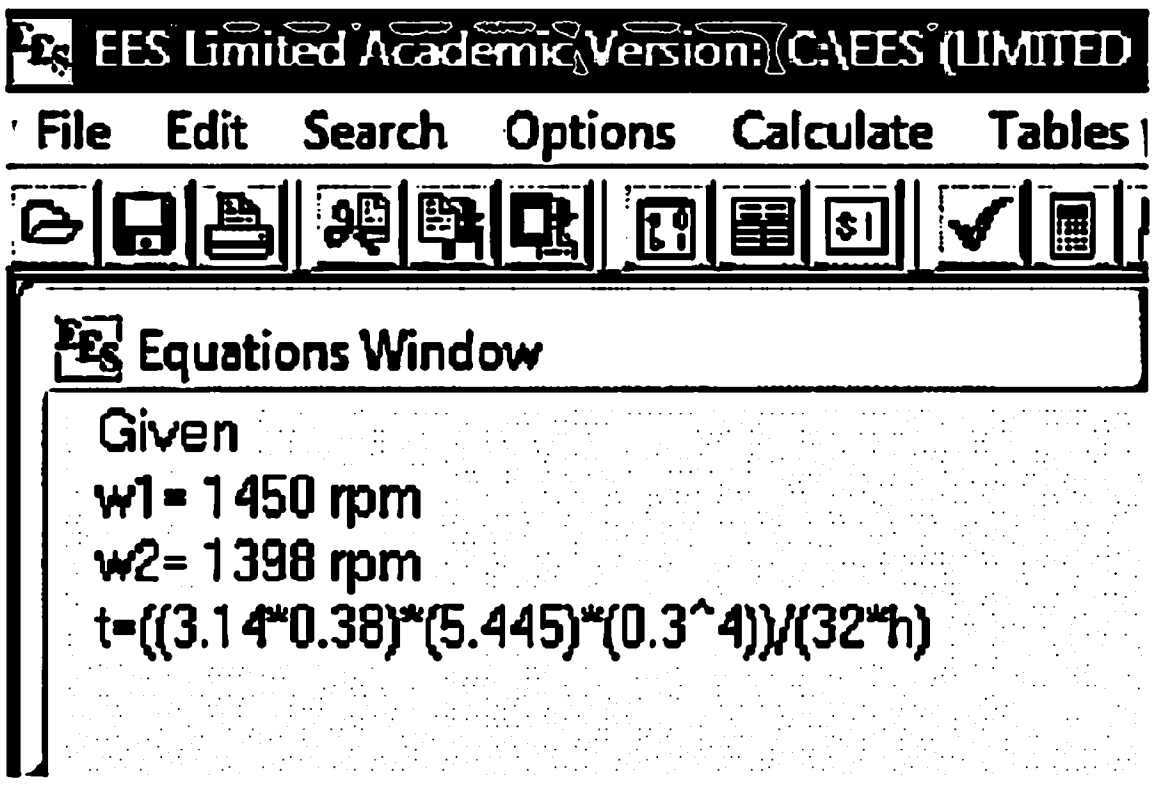
Question: 10.025P

Reconsider Prob. 10–24. Using EES (or other) software, investigate the effect of oil film thickness on the torque transmitted. Let the film thickness vary from 0.1 mm to 10 mm. Plot your results, and state your conclusions.

Step 1 of 3

The equation used is $T = \frac{\pi\mu(\omega_1 - \omega_2)D^4}{32h}$

By Using the EES software, we have to write the above equation in new window.



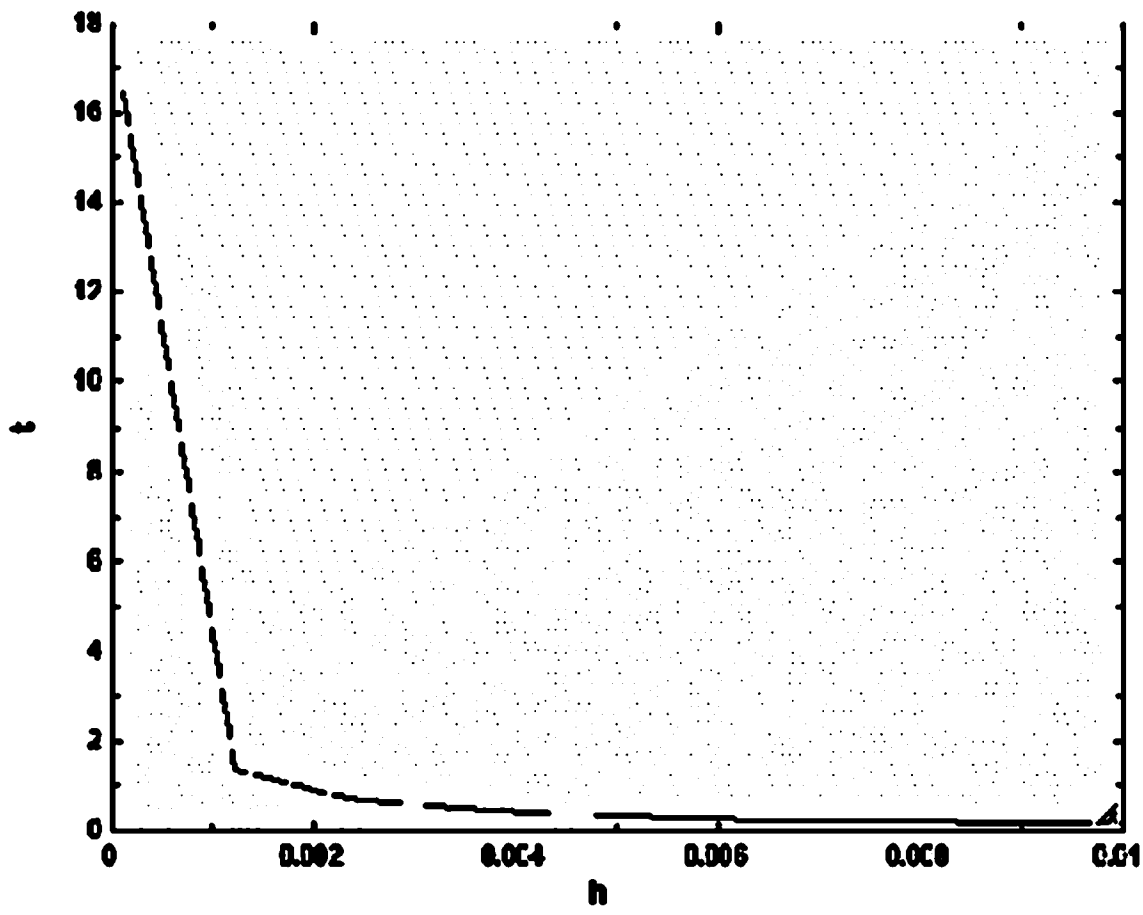
Step 2 of 3

After writing the equations in the Equation window then click on table in menu bar. Then select new parameter table. Then varying the h value from 0.0001 to 0.01m we get

1..10	h	t
Run 1	0.0001	16.45
Run 2	0.0012	1.37
Run 3	0.0023	0.715
Run 4	0.0034	0.4837
Run 5	0.0045	0.3655
Run 6	0.0056	0.2937
Run 7	0.0067	0.2455
Run 8	0.0078	0.2108
Run 9	0.0089	0.1848
Run 10	0.01	0.1645

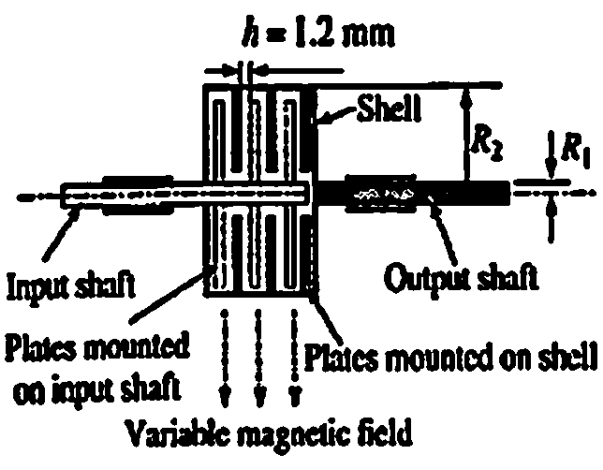
Step 3 of 3

By plotting the graph between torque Vs thickness of oil film:



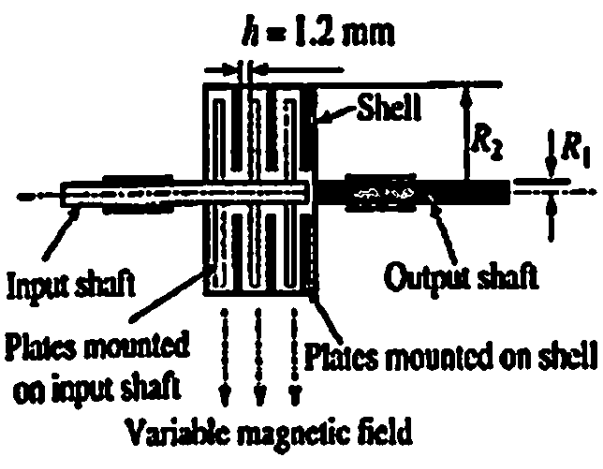
Question: 10.026P

The viscosity of some fluids changes when a strong electric field is applied on them. This phenomenon is known as the electrorheological (ER) effect, and fluids that exhibit such behavior are known as ER fluids. The Bingham plastic model for shear stress, which is expressed as $\tau = \tau_y + \mu(du/dy)$ is widely used to describe ER fluid behavior because of its simplicity. One of the most promising applications of ER fluids is the ER clutch. A typical multidisk ER clutch consists of several equally spaced steel disks of inner radius $R1$ and outer radius $R2$, N of them attached to the input shaft. The gap h between the parallel disks is filled with a viscous fluid. (*a*) Find a relationship for the torque generated by the clutch when the output shaft is stationary and (*b*) calculate the torque for an ER clutch with $N = 11$ for $R1 = 50$ mm, $R2 = 200$ mm, and $\dot{n} = 2400$ rpm if the fluid is SAE 10 with $\mu = 0.1$ Pa•s, $\tau_y = 2.5$ kPa, and $h = 1.2$ mm.



Question:

The viscosity of some fluids changes when a strong electric field is applied on them. This phenomenon is known as the electrorheological (ER) effect, and fluids that exhibit such behavior are known as ER fluids. The Bingham plastic model for shear stress, which is expressed as $\tau = \tau_y + \mu(du/dy)$ is widely used to describe ER fluid behavior because of its simplicity. One of the most promising applications of ER fluids is the ER clutch. A typical multidisk ER clutch consists of several equally spaced steel disks of inner radius $R1$ and outer radius $R2$, N of them attached to the input shaft. The gap h between the parallel disks is filled with a viscous fluid. (*a*) Find a relationship for the torque generated by the clutch when the output shaft is stationary and (*b*) calculate the torque for an ER clutch with $N = 11$ for $R1 = 50$ mm, $R2 = 200$ mm, and $\dot{n} = 2400$ rpm if the fluid is SAE 10 with $\mu = 0.1$ Pa•s, $\tau_y = 2.5$ kPa, and $h = 1.2$ mm.



We have,

$$\tau = \tau_y + \mu\left(\frac{du}{dy}\right)$$

$$= \tau_y + \mu\left(\frac{V}{h}\right) \qquad \ldots\ldots (1)$$

Where,
The velocity of the input shaft is
$V = r\omega$
Substituting V in equation (1), we get

$$\tau = \tau_y + \mu\left(\frac{r\omega}{h}\right)$$

Step 3 of 6

Consider an area dA on the plate surface.
Then the force can be written as
$dF = \tau dA$
Substituting τ in dF, we get

$$dF = \left[\tau_y + \mu\left(\frac{r\omega}{h}\right)\right](2\pi r\,dr)$$

The torque generated can write as
$dT = r\,dF$
Integrating on both sides
$\int dT = \int r\,dF$
Substituting dF in dT, we get
We have,

$$\tau = \tau_y + \mu\left(\frac{du}{dy}\right)$$

$$= \tau_y + \mu\left(\frac{V}{h}\right) \qquad \ldots\ldots (1)$$

Where,
The velocity of the input shaft is
$V = r\omega$
Substituting V in equation (1), we get

$$\tau = \tau_y + \mu\left(\frac{r\omega}{h}\right)$$

Step 3 of 6

Consider an area dA on the plate surface.
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$dF = \tau dA$
Substituting τ in dF, we get

$$dF = \left[\tau_y + \mu\left(\frac{r\omega}{h}\right)\right](2\pi r\,dr)$$

The torque generated can write as
$dT = r\,dF$
Integrating on both sides
$\int dT = \int r\,dF$
Substituting dF in dT, we get

$$\int dT = \int_{R_1}^{R_2} r\left[\tau_y + \mu\left(\frac{\omega r}{h}\right)\right](2\pi r\,dr)$$

$$= 2\pi\left(\tau_y \int_{R_1}^{R_2} r^2\,dr + \frac{\mu\omega}{h}\int_{R_1}^{R_2} r^3\,dr\right)$$

Step 4 of 6

Applying limits of integration, we get
The torque generated by a single surface of the plate

$$T = 2\pi\left[\frac{\tau_y}{3}\left(R_2^3 - R_1^3\right) + \frac{\mu\omega}{4h}\left(R_2^4 - R_1^4\right)\right]$$

The torque generated by both surfaces of the plate

$$T = 2\left\{2\pi\left[\frac{\tau_y}{3}\left(R_2^3 - R_1^3\right) + \frac{\mu\omega}{4h}\left(R_2^4 - R_1^4\right)\right]\right\}$$

$$= 4\pi\left[\frac{\tau_y}{3}\left(R_2^3 - R_1^3\right) + \frac{\mu\omega}{4h}\left(R_2^4 - R_1^4\right)\right]$$

For N number of clutch plates, the torque generated is

$$T = 4\pi N\left[\frac{\tau_y}{3}\left(R_2^3 - R_1^3\right) + \frac{\mu\omega}{4h}\left(R_2^4 - R_1^4\right)\right]$$

The torque generated by the clutch when the output shaft is stationary is

$$\boxed{T = 4\pi N\left[\frac{\tau_y}{3}\left(R_2^3 - R_1^3\right) + \frac{\mu\omega}{4h}\left(R_2^4 - R_1^4\right)\right]}$$

Step 5 of 6

(b)
Given data:
The number of clutch plates, $N = 11$
The inner radius of clutch plate, $R_1 = 50$ mm.
The outer radius of clutch plate, $R_2 = 200$ mm.
The speed of the input shaft is, $\dot{n} = 2400$ rpm.
The viscosity of fluid *SAE* 10, $\mu = 0.1\ \text{Pa}\cdot\text{s}$

$$= 4\pi(11)\left[\frac{(2.5\times 10^{3})}{3}\left(0.2^3 - 0.05^3\right) + \frac{(0.1)(251.32)}{4(0.0012)}\left(0.2^4 - 0.05^4\right)\right]$$

$$= 4\pi(11)[6.5625 + 8.3446]$$

$$= 2060.61\ \text{N}\cdot\text{m}$$

The torque for an ER clutch is $\boxed{T = 2060.61\ \text{N}\cdot\text{m}}$

Question: 10.027P

The viscosity of some fluids, called magnetorheological (MR) fluids, changes when a magnetic field is applied. Such fluids involve micron-sized magnetizable particles suspended in an appropriate carrier liquid, and are suitable for use in controllable hydraulic clutches. See Fig. P10–26. The MR fluids can have much higher viscosities than the ER fluids, and they often exhibit shear-thinning behavior in which the viscosity of the fluid decreases as the applied shear force increases. This behavior is also known as pseudoplastic behavior, and can be successfully represented by Herschel-Bulkley constitutive model expressed as $\tau = \tau y + K(du/dy)m$. Here τ is the shear stress applied, τv is the yield stress, K is the consistency index, and m is the power index. For a Herschel-Bulkley fluid with τy = 900 Pa, K = 58 Pa·sm, and m = 0.82, (*a*) find a relationship for the torque transmitted by an MR clutch for N plates attached to the input shaft when the input shaft is rotating at an angular speed of ω while the output shaft is stationary and (*b*) calculate the torque transmitted by such a clutch with N = 11 plates for $R1$ = 50 mm, $R2$ = 200 mm, $\dot{n} = 3000$ rpm, and h = 1.5 mm.

Step 1 of 8

Given data:

The fluid used in the clutches is MR fluid.

The yield stress, $\tau_y = 900\,\text{Pa}$

The consistency index, $K = 58\,\text{Pa}\cdot s^m$

The power index, $m = 0.82$

The shear stress used for The Herschel-Bulkley constitutive model is

$$\tau = \tau_y + K\left(\frac{du}{dy}\right)^m$$

Step 2 of 8

(a)

The number of clutch plates attached to input shaft are N.

The angular speed of the input shaft is ω.

The output shaft is stationary.

We know that velocity gradient

$$\frac{V}{l} = \frac{du}{dy}$$

Here, The fluid thickness $l = h$

$$\frac{V}{h} = \frac{du}{dy}$$

Step 3 of 8

We have,

$$\tau = \tau_y + \mu\left(\frac{du}{dy}\right)^m$$

$$= \tau_y + \mu\left(\frac{V}{h}\right)^{0.82} \qquad \ldots\ldots (1)$$

Where,

The velocity of the input shaft is

$V = r\omega$

Substituting V in equation (1), we get

$$\tau = \tau_y + \mu\left(\frac{r\omega}{h}\right)^{0.82}$$

Step 4 of 8

Consider an area dA on the plate surface.

Then the force can be written as

$dF = \tau dA$

Substituting τ in dF, we get

$$dF = \left[\tau_y + \mu\left(\frac{r\omega}{h}\right)^{0.82}\right](2\pi r dr)$$

The torque generated can write as

$dT = rdF$

Integrating on both sides

$\int dT = \int rdF$

Substituting dF in dT, we get

$$\int dT = \int rdF$$

$$= \int_{R_1}^{R_2} r\left[\tau_y + K(\omega r/h)^{0.82}\right](2\pi r dr)$$

$$= 2\pi\left[\tau_y \int_{R_1}^{R_2} r^2 dr + K\left(\frac{\omega}{h}\right)^{0.82} \int_{R_1}^{R_2} r^{2.82} dr\right]$$

Step 5 of 8

Applying limits of integration, we get

The torque generated by a single surface of the plate

$$T = 2\pi\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]$$

The torque generated by both surfaces of the plate

$$T = (2)\left\{2\pi\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]\right\}$$

$$= 4\pi\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]$$

For N number of clutch plates, the torque generated is

$$T = (N)\left\{4\pi\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]\right\}$$

$$= 4\pi N\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]$$

The torque generated by the MR clutch when the output shaft is stationary is

$$\boxed{T = 4\pi N\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]}$$

Step 6 of 8

(b)

Given data:

The number of clutch plates, $N = 11$

The inner radius of clutch plate, $R_1 = 50$ mm

The outer radius of clutch plate, $R_2 = 200$ mm

The speed of the input shaft is, $\dot{n} = 3000$ rpm.

The fluid thickness, $h = 1.5$ mm

Step 7 of 8

The angular speed of the shaft is

$$\omega = \frac{2\pi\dot{n}}{60}$$

$$= \frac{2\pi(2400)}{60}$$

$$= 251.32\,\text{rad/s}$$

Step 8 of 8

The torque for an MR clutch is given by

$$T = 4\pi N\left[\tau_y\left(\frac{1}{3}\right)\left(R_2^3 - R_1^3\right) + K\left(\frac{\omega}{h}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(R_2^{3.82} - R_1^{3.82}\right)\right]$$

$$= 4\pi(11)\left[(900)\left(\frac{1}{3}\right)\left(0.2^3 - 0.05^3\right) + (58)\left(\frac{314.15}{0.0015}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(0.2^{3.82} - 0.05^{3.82}\right)\right]$$

$$= 4\pi(11)\left[(900)\left(\frac{1}{3}\right)\left(0.2^3 - 0.05^3\right) + (58)\left(\frac{314.15}{0.0015}\right)^{0.82}\left(\frac{1}{3.82}\right)\left(0.2^{3.82} - 0.05^{3.82}\right)\right]$$

$$= 4\pi(11)[2.3625 + 745.373]$$

$T = 103359.53\,\text{N}\cdot\text{m}$

Torque transmitted by the clutch is $\boxed{T = 103359.53\,\text{N}\cdot\text{m}}$

Question: 10.028P

The viscosity of a fluid is to be measured by a viscometer constructed of two 75-cm-long concentric cylinders. The outer diameter of the inner cylinder is 15 cm, and the gap between the two cylinders is 1 mm. The inner cylinder is rotated at 300 rpm, and the torque is measured to be 0.8 N·m. Determine the viscosity of the fluid.

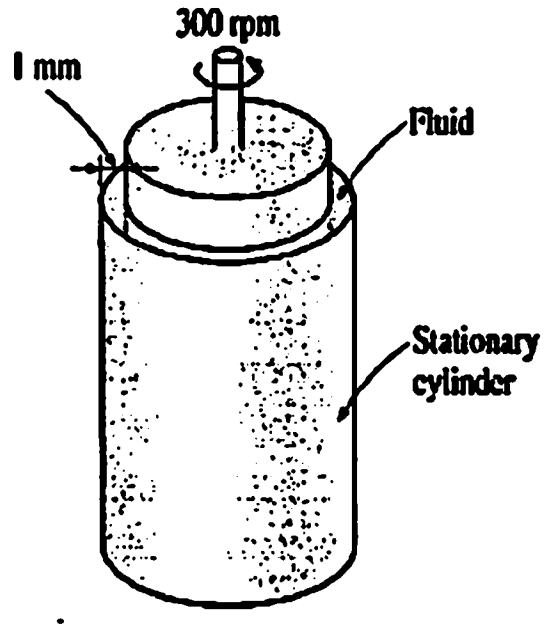


FIGURE P10–28

Step 1 of 3

Given data:

Length of cylinders, $L = 75\text{ cm} = 0.75\text{ m}$

Outer diameter of the inner cylinder, $d = 15\text{ cm} = 0.15\text{ m}$

Therefore the radius, $r = 0.075\text{ m}$

Gap between the cylinders, $dy = 1\text{ mm} = 10^{-3}\text{m}$

Speed of the inner cylinder, $N = 300\text{ rpm}$

Torque, $T = 0.8\text{ N}\cdot\text{m}$

Step 2 of 3

Area is given by

$$A = \pi dL$$
$$= \pi \times 0.15 \times 0.75$$
$$= 0.35325\text{ m}^2$$

Velocity is given by

$$u = \frac{\pi dN}{60}$$
$$= \frac{\pi \times 0.15\text{ m} \times 300\text{ rpm}}{60}$$
$$= 2.356\text{ m/s}$$

Step 3 of 3

Force acting is given by

$$F = \frac{T}{r}$$
$$= \frac{0.8\text{ N}\cdot\text{m}}{0.075\text{ m}}$$
$$= 10.66\text{ N}$$

We know that

$$F = \mu A \frac{du}{dy}$$

$$(10.66\text{ N}) = \frac{\mu \times (0.35325\text{ m}^2) \times (2.356\text{ m/s})}{(10^{-3}\text{ m})}$$

$$\mu = 0.0128\text{kg/m}\cdot\text{s}$$

Therefore the Viscosity of the fluid is $\boxed{\mu = 0.0128\text{kg/m}\cdot\text{s}}$

Question: 10.029P

The viscosity of a fluid is to be measured by a viscometer constructed of two 3-ft-long concentric cylinders. The inner diameter of the outer cylinder is 6 in, and the gap between the two cylinders is 0.05 in. The outer cylinder is rotated at 250 rpm, and the torque is measured to be 1.2 lbf-ft. Determine the viscosity of the fluid.

Step 1 of 3

Given data:

Length of cylinders, $L = 3\,\text{ft}$

Outer diameter of the inner cylinder, $d = 6\text{ in} = 0.5\,\text{ft}$

Therefore the radius, $r = 0.25\,\text{ft}$

Gap between the cylinders, $dy = 0.05\text{in} = 0.00416\,\text{ft}$

Speed of the inner cylinder, $N = 300\,\text{rpm}$

Torque, $T = 1.2\,\text{lbf}\cdot\text{ft}$

Step 2 of 3

Area is given by

$$A = \pi dL$$
$$= \pi \times 0.5 \times 3$$
$$= 4.71\,\text{ft}^2$$

Velocity is given by

$$u = \frac{\pi dN}{60}$$
$$= \frac{\pi \times 0.5 \times 250}{60}$$
$$= 6.54\,\text{ft/s}$$

Step 3 of 3

Force acting is given by

$$F = \frac{T}{r}$$
$$= \frac{1.2}{0.25}$$
$$= 4.8\,\text{lbf}$$

We know that

$$F = \mu A \frac{du}{dy}$$
$$4.8 = \frac{\mu \times 4.71 \times 6.54}{0.00416}$$
$$\mu = 0.000648\,\text{lb}\cdot\text{s/ft}$$

Therefore the Viscosity of the fluid is $\boxed{\mu = 0.000648\,\text{lb}\cdot\text{s/ft}}$

Question: 10.030P

In regions far from the entrance, fluid flow through a circular pipe is one-dimensional, and the velocity profile for laminar flow is given by$u(r)$= $u$max(1 −$r2/R2$), where$R$ is the radius of the pipe, r is the radial distance from the center of the pipe, and$u$max is the maximum flow velocity, which occurs at the center. Obtain (*a*) a relation for the drag force applied by the fluid on a section of the pipe of length$L$ and (*b*) the value of the drag force for water flow at 20°C with R = 0.08 m, L = 30 m, $u$max = 3 m/s, and μ = 0.0010 kg/m·s.

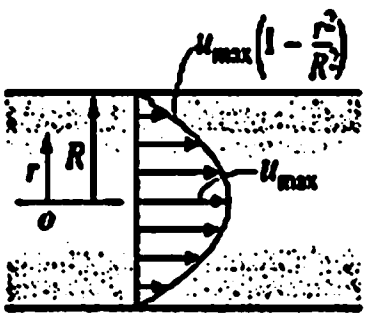


FIGURE P10–30

Step 1 of 5

Given data:

Velocity profile in the pipe, $u(r)=u_{max}\left(1-\frac{r^2}{R^2}\right)$

Step 2 of 5

(a)

We know that

Drag force, $F=\mu A\frac{u}{y}$

Where

The area $A=2\pi Rl$

And

$$\frac{u}{y}=\frac{du}{dy}$$
$$=-\frac{du}{dr}$$
$$=-\frac{d}{dr}\left[u_{max}\left(1-\frac{r^2}{R^2}\right)\right]$$
$$=-u_{max}\left(\frac{-2r^{2-1}}{R^2}\right)_{r=R}$$
$$=\frac{2u_{max}}{R}$$

Step 3 of 5

So drag force becomes;

$$F=\mu(2\pi RL)\left(\frac{2u_{max}}{R}\right)$$
$$=4\pi u_{max}\mu L$$

$$\boxed{F=4\pi u_{max}\mu L}$$

Therefore drag force per unit length is $F=4\pi u_{max}\mu L$

Step 4 of 5

(b) Given data:

The radius of the pipe $R=0.08\,\mathrm{m}$

The length of the pipe $L=30\,\mathrm{m}$

The maximum flow velocity $u_{max}=3\,\mathrm{m/s}$

The dynamic viscosity of fluid $\mu=0.0010\,\mathrm{kg/m\cdot s}$

Step 5 of 5

By substituting the given values:

$$F=4\pi u_{max}\mu L$$
$$=4\pi\times3\times0.001\times30$$
$$=1.131\,\mathrm{N}$$

$$\boxed{F=1.131\,\mathrm{N}}$$

Therefore the value of drag force is $1.131\,\mathrm{N}$

Question: 10.031P

Repeat Prob. 10–30 for *u*max = 7 m/s.

Step 1 of 5

Given data:

Velocity profile in the pipe, $u(r) = u_{max}\left(1-\frac{r^2}{R^2}\right)$

Step 2 of 5

(a)
We know that

Drag force, $F = \mu A\frac{u}{y}$

Where
The area $A = 2\pi Rl$
And

$$\frac{u}{y} = \frac{du}{dy}$$
$$= -\frac{du}{dr}$$
$$= -\frac{d}{dr}\left[u_{max}\left(1-\frac{r^2}{R^2}\right)\right]$$
$$= -u_{max}\left(\frac{-2r^{2-1}}{R^2}\right)_{r=R}$$
$$= \frac{2u_{max}}{R}$$

Step 3 of 5

So drag force becomes;

$$F = \mu(2\pi RL)\left(\frac{2u_{max}}{R}\right)$$
$$= 4\pi u_{max}\mu L$$
$$\boxed{F = 4\pi u_{max}\mu L}$$

Therefore drag force per unit length is $F = 4\pi u_{max}\mu L$

Step 4 of 5

(b) Given data:
The radius of the pipe $R = 0.08\text{m}$
The length of the pipe $L = 30\text{m}$
The maximum flow velocity $u_{max} = 7\text{m/s}$
The dynamic viscosity of fluid $\mu = 0.0010\text{kg/m}\cdot\text{s}$

Step 5 of 5

By substituting the given values:

$$F = 4\pi u_{max}\mu L$$
$$= 4\pi\times 7\times 0.001\times 30$$
$$= 2.64\text{N}$$
$$\boxed{F = 2.64\text{N}}$$

Therefore the value of drag force is 2.64N

Question: 10.032P

What is surface tension? What is it caused by? Why is the surface tension also called surface energy?

Step 1 of 1

Surface tension:

Surface tension is a property of a liquid due to unbalanced molecular cohesive forces at or near the surface.

The main reason for surface tension is cohesion of molecules. The pulling force that causes this tension is due to the attractive forces between the molecules. The unit for surface tension is N/m. It is also written as N-m/m^2 i.e. surface energy per unit area. Surface tension is also defined as work done per unit increase in the surface area.

Question: 10.033P

Consider a soap bubble. Is the pressure inside the bubble higher or lower than the pressure outside?

Step 1 of 1

The air trapped inside a soap bubble will add a pressure to the soap film from the inside and the air surrounding the bubble will add a pressure to the soap film from the outside. This means that the soap bubble is acted by pressure from two sides i.e. from inner and outer.

We know that in a closed area, density of air will be larger than the normal surrounding area. The pressure inside the bubble will be larger than the pressure outside the soap bubble. It is represented as:

$$P_i - P_e = \frac{8\sigma}{d}$$

P_i = Pressure inside the bubble

P_e = Pressure outside the bubble

d = Diameter of soap bubble

σ = Surface tension

Question: 10.034P

What is the capillary effect? What is it caused by? How is it affected by the contact angle?

Step 1 of 1

Capillarity is defined as a phenomenon of rise or fall of a liquid surface in a small tube relative to the adjacent general level of liquid when the tube is held vertically in the liquid.

The property of rise or fall of a liquid in capillary (small diameter tube) inserted in the liquid is known as a capillary effect.
The main cause of capillary effect is due to the adhesive forces (The forces between unlike molecules like glass and water). If the adhesive force dominates the cohesive force then there will be a capillary rise.
If the cohesive force dominates the adhesive forces there will be a capillary fall of the liquid.

The strength of capillary effect is quantified by contact angle ϕ defined as the angle that tangent to the liquid surface makes with the solid surface at the point of contact. If $\phi < 90^\circ$ capillary rise will occur and if $\phi > 90^\circ$ capillary fall will occur.

Wet surface and dry surface depends upon the contact angle.

Question: 10.035P

A small-diameter tube is inserted into a liquid whose contact angle is 110°. Will the level of liquid in the tube be higher or lower than the level of the rest of the liquid? Explain.

Step 1 of 2

Capillary effect:

Capillary effect is a rise or fall of a liquid in a small diameter tube when inserted into a liquid.
It is known that the contact angle is 110°.

Step 2 of 2

If the contact angle is more than 90 degrees, the liquid in the tube is lower than the rest of the liquid.

So, $110^{\circ} > 90^{\circ}$

This means that the level of the liquid in the tube will be lower than the level of the rest of the fluid.

If $\phi > 90^{\circ}$, h is negative value due to capillary drop.

Question: 10.036P

Is the capillary rise greater in small- or large-diameter tubes?

Step 1 of 1

The capillary rise h is given by the relation:

$$h = \frac{2\sigma_s}{\rho g R}\cos\phi$$

Where

R: Radius

ϕ : Capillary rising angle

σ_s : Surface tension

ρ : Density

From the above relation it is clear that

$$h \propto \frac{1}{R}$$

$$h \propto \frac{1}{D}$$

Capillary height is inversely proportional to the diameter of tube.

So, capillary rise will be greater in small diameter tubes and lesser in large diameter tubes.

Question: 10.037P

Consider a 0.2-mm diameter air bubble in a liquid. Determine the pressure difference between the inside and outside of the air bubble if the surface tension at the air-liquid interface is (*a*) 0.08 N/m and (*b*) 0.12 N/m.

Step 1 of 3

Given data:

Diameter of the air bubble, $D = 0.2\,\text{mm}$

$= 0.2\times10^{-3}\,\text{m}$

So the radius of the air bubble, $R = 0.1\times10^{-3}\,\text{m}$

Step 2 of 3

(a)

Surface Tension, $\sigma_s = 0.08\,\text{N/m}$

The pressure difference between inside and outside of the bubble is given as

$$P_i - P_o = \frac{2\sigma_s}{R}$$

$$= \frac{2\times0.08\,\text{N/m}}{0.1\times10^{-3}\,\text{m}}$$

$$= 1600\,\text{N/m}^2$$

Thus the pressure difference of the air bubble, $\boxed{P_i - P_o = 1600\,\text{N/m}^2}$

Step 3 of 3

(b)

Surface Tension, $\sigma_s = 0.12\,\text{N/m}$

$$P_i - P_o = \frac{2\sigma_s}{R}$$

$$= \frac{2\times0.12\,\text{N/m}}{0.1\times10^{-3}\,\text{m}}$$

$$= 2400\,\text{N/m}^2$$

Thus the pressure difference of the air bubble, $\boxed{P_i - P_o = 2400\,\text{N/m}^2}$

Question: 10.038P

A 0.03-in-diameter glass tube is inserted into kerosene at 68°F. The contact angle of kersene with a glass surface is 26°. Determine the capillary rise of kerosene in the tube.

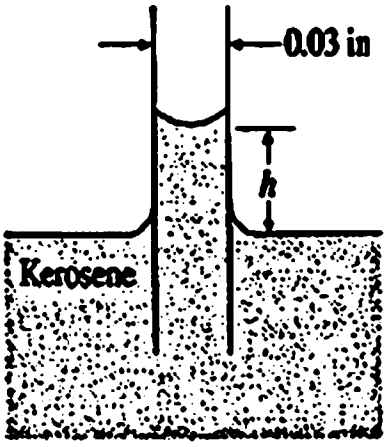


FIGURE P10–38

Step 1 of 2

Given data:

Glass tube diameter, $D = 0.03\text{in.}$

Radius of capillary tube, $R = 0.015\text{in.}$

$= 0.00125\text{ft}$

Kerosene at 20^0 C is the liquid.

Contact angle, $\phi = 26^0$

Step 2 of 2

From properties of liquid tables

Surface tension of kerosene at 60^0 F, $\sigma_s = 1.91856 \times 10^{-3}$ lb/ft

Density of kerosene, $\rho = 51.2\text{lbm/ft}^3$

Capillary raise is given by

$$h = \frac{2\sigma_s}{\rho g R}\cos\phi$$

$$= \frac{2 \times 1.91856 \times 10^{-3} \times \cos 26^\circ}{\frac{51.2}{32.17} \times 32.2 \times 0.00125}$$

$$= 0.05384\text{ft}$$

$$= 0.646\text{in.}$$

Therefore, height of mercury rise is $\boxed{h = 0.65\text{in.}}$

Question: 10.039P

A 1.2-mm-diameter tube is inserted into an unknown liquid whose density is 960 kg/m3, and it is observed that the liquid rises 5 mm in the tube, making a contact angle of 15°. Determine the surface tension of the liquid.

Step 1 of 2

Given data:

Glass tube diameter, $D = 1.2$ mm

Contact angle, $\phi = 15°$

Capillary rise, $h = 5$ mm

$= 0.005$ m

Density of liquid, $\rho = 960$ kg/m^3

Radius of capillary tube, $R = \frac{D}{2}$

$= 1.2/2$

$= 0.6$ mm

$= 0.6 \times 10^{-3}$ m

Step 2 of 2

Capillary raise is given by

$$h = \frac{2\sigma_s}{\rho g R}\cos\phi$$

$$= \frac{2(\sigma_s \text{ N/m})}{(960)\text{ kg/m}^3\,(9.81\text{ m/s}^2)(0.6\times10^{-3}\text{ m})}\cos 15°$$

$$= 0.342\sigma_s$$

$$(0.005\text{ m}) = 0.342\sigma_s$$

$$\sigma_s = 0.01462\text{ N/m}$$

Therefore, Surface Tension $\boxed{\sigma_s = 0.0146\text{ N/m}}$

Question: 10.040P

Determine the gage pressure inside a soap bubble of diameter (*a*) 0.2 cm and (*b*) 5 cm at 20°C.

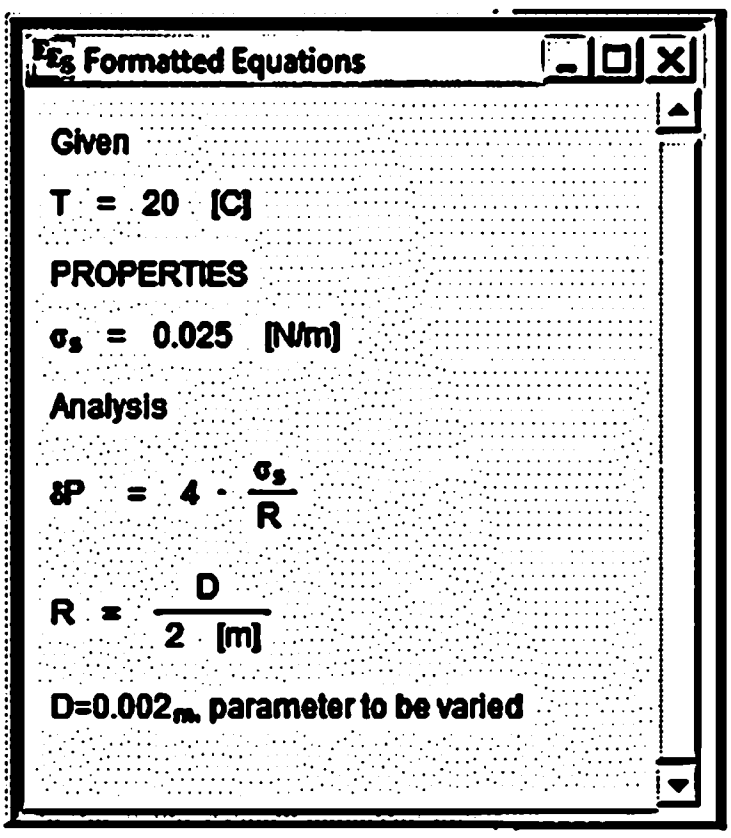
Step 1 of 2

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

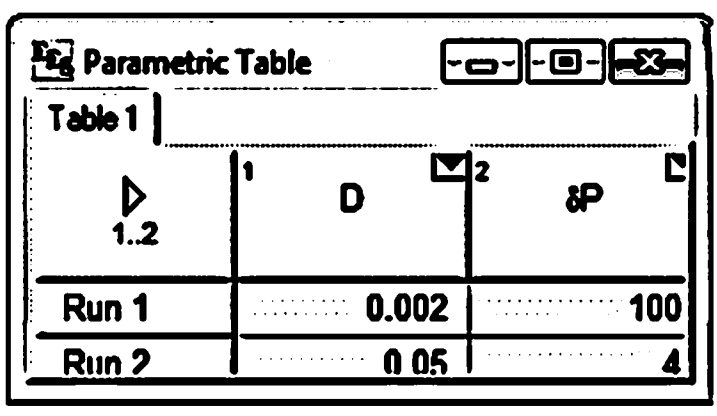
```
"Given"
T=20[C]
"PROPERTIES"
sigma_s=0.025[N/m]
"Analysis"
DeltaP=4*sigma_s/R
R=D/2[m]
"D=0.002[m], parameter to be varied"
```

Now check the mathematical representation of the equations entered in the code by pressing 'Formatted equations' of the 'Windows' tab. The equations will be displayed as shown.



Step 2 of 2

Now to solve the equation for two diameters, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 2
Select variables in equations as D and δP press 'Add' and then press 'ok'
A table will appear and enter the values of diameter under 'D' as 0.002 and 0.05 in the D column.
Then click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below



Thus the gauge pressure inside a 0.2 cm soap bubble is $\boxed{\Delta p = 100\,\mathrm{N/m^2}}$ and

The gauge pressure inside a 5 cm soap bubble is $\boxed{\Delta p = 4\,\mathrm{N/m^2}}$

Question: 10.041P

Nutrients dissolved in water are carried to upper parts of plants by tiny tubes partly because of the capillary effect. Determine how high the water solution will rise in a tree in a 0.002-mm-diameter tube as a result of the capillary effect. Treat the solution as water at 20°C with a contact angle of 15°.

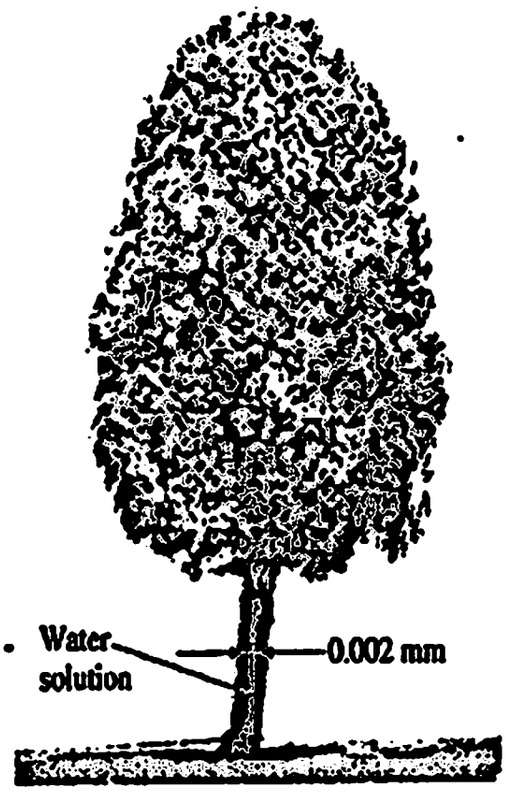


FIGURE P10–41

Step 1 of 2

Given data:

Tube diameter, $D = 0.002\ \text{mm}$

Contact angle, $\phi = 15°$

Density of water, $\rho = 998.21\ \text{kg/m}^3$

Surface tension of water at 20°C, $\sigma_s = 0.073\ \text{N/m}$

Radius of capillary tube, $R = \dfrac{D}{2}$

$$= \frac{0.002}{2}$$

$$= 0.001\ \text{mm}$$

$$= 1\times10^{-6}\ \text{m}$$

Step 2 of 2

Capillary raise is given by

$$h = \frac{2\sigma_s}{\rho g R}\cos\phi$$

$$= \frac{2(0.073\ \text{N/m})}{(998.21\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(1\times10^{-6}\ \text{m})}\cos 15°$$

$$= 14.4\ \text{m}$$

Therefore the capillary rise is $\boxed{h = 14.4\ \text{m}}$.

Question: 10.042P

The surface tension of a liquid is to be measured using a liquid film suspended on a U-shaped wire frame with an 8-cm-long movable side. If the force needed to move the wire is 0.024 N, determine the surface tension of this liquid in air.

Step 1 of 2

Given data:

Movable side length, $b = 8\,\text{cm}$

$= 0.08\,\text{m}$

Force required, $F = 0.024\,\text{N}$

Step 2 of 2

We know that surface tension

$$\sigma_s = \frac{F}{2b}$$

Putting values in the above equation

$$\sigma_s = \frac{0.024\,\text{N}}{2 \times 0.08\,\text{m}}$$

$$= 0.15\,\text{N/m}$$

Thus surface tension $\boxed{\sigma_s = 0.15\,\text{N/m}}$

Question: 10.043P

Contrary to what you might expect, a solid steel ball can float on water due to the surface tension effect. Determine the maximum diameter of a steel ball that would float on water at 20°C. What would your answer be for an aluminum ball? Take the densities of steel and aluminum balls to be 7800 kg/m3 and 2700 kg/m3, respectively.

Step 1 of 3

Surface tension of water at 20°C, $\sigma = 0.073\,\text{N/m}$

When the board has to float, the force should be equal to the weight of the ball.

$$\pi D\sigma_s = mg$$

$$\pi D\sigma_s = \rho Vg$$

$$\pi D\sigma_s = \rho\pi\frac{D^3}{6}g$$

$$\sigma_s = \rho\frac{D^2}{6}g$$

$$D^2 = \frac{6\sigma_s}{\rho g}$$

$$D = \sqrt{\frac{6\sigma_s}{\rho g}}$$

Here σ_s is surface tension, ρ is density of fluid and g is acceleration due to gravity.

Step 2 of 3

Substitute: $\sigma = 0.073\,\text{N/m}$, $\rho_{steel} = 7800\,\text{kg/m}^3$ and $g = 9.81\,\text{m/s}^2$

Diameter of a steel ball:

$$D_{steel} = \sqrt{\frac{6\times 0.073\,\text{N/m}}{7800\,\text{kg/m}^3\times 9.81}}$$

$$= 2.39\times 10^{-3}\,\text{m}$$

Thus diameter of steel ball is $\boxed{2.39\times 10^{-3}\,\text{m}}$

Step 3 of 3

Substitute: $\sigma = 0.073\,\text{N/m}$, $\rho_{Al} = 2700\,\text{kg/m}^3$ and $g = 9.81\,\text{m/s}^2$

Diameter of an aluminum ball:

$$D_{Al} = \sqrt{\frac{6\times 0.073\,\text{N/m}}{2700\,\text{kg/m}^3\times 9.81}}$$

$$= 4.1\times 10^{-3}\,\text{m}$$

Thus diameter of steel ball is $\boxed{4.1\times 10^{-3}\,\text{m}}$

Question:

10.044P

The pressure on the suction side of pumps is typically low, and the surfaces on that side of the pump are susceptible to cavitation, especially at high fluid temperatures. If the minimum pressure on the suction side of a water pump is 0.95 psia absolute, determine the maximum water temperature to avoid the danger of cavitation.

Step 1 of 2

Given data:

Type of pump: Water pump

Operating pressure of the water pump, $P = 0.95\,\text{psia}$

Step 2 of 2

From the table of "saturated water temperature"

At $P = 0.95\,\text{psia}$

Saturation temperature of the water is $T_{sat} = 100°\text{F}$

So we have to avoid cavitations in water pump, we maintained the temperature of the water below saturation temperature of the water.

Therefore, maximum water temperature to avoid cavitations is $\boxed{T_{max} = 100°\text{F}}$.

Question: 10.045P

A closed tank is partially filled with water at 80°C. If the air above the water is completely evacuated, determine the absolute pressure in the evacuated space. Assume the temperature to remain constant.

Step 1 of 2

Given data:

The closed tank is partially filled with water.

All the air is removed from the tank.

The temperature of the water is $T = 80^\circ C$.

The temperature remains constant.

Step 2 of 2

The atmospheric pressure inside the vessel is zero hence the absolute pressure inside the tank will be equal to the vapor pressure of water at $T = 80^\circ C$.

The vapor pressure of water at $T = 80^\circ C$.

$P_w = 355.1$ mm of Hg

Absolute pressure in the evacuated space is $\boxed{P_w = 355.1 \text{ mm of Hg}}$.

Question: 10.046P

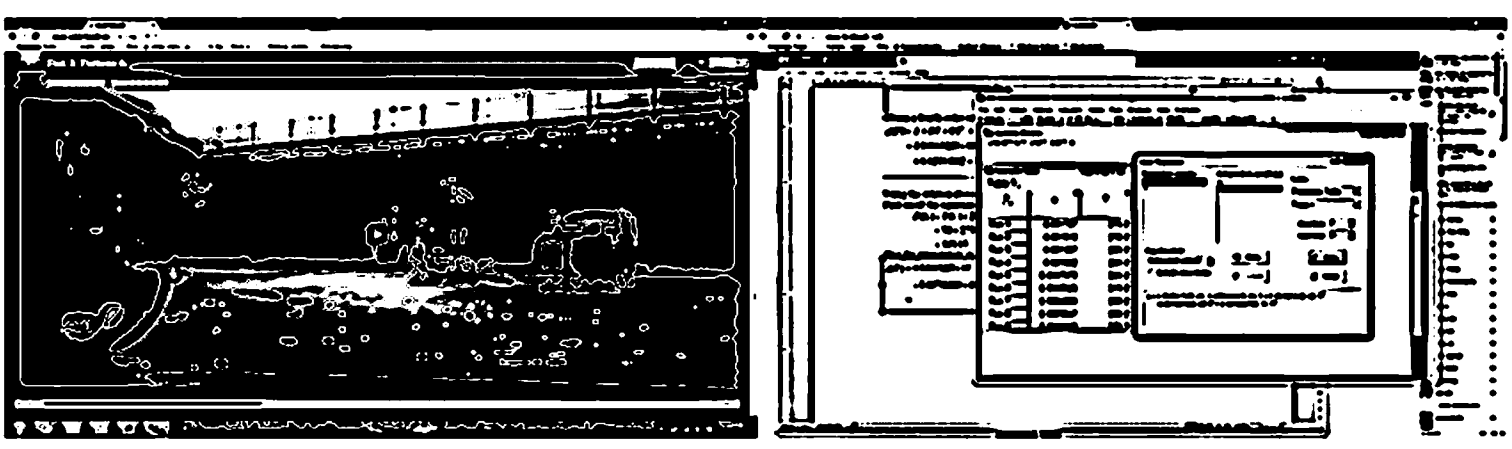
The variation of the dynamic viscosity of water with absolute temperature is given as

T,K	μ, Pa·s
273.15	1.787 × l0−3
278.15	1.519 × 10−3
283.15	1.307 × 10−3
293.15	1.002 × 10−3
303.15	7.975 × 10−4
313.15	6.529 × 10−4
333.15	4.665 × 10−4
353.15	3.547 × 10−4
373.15	2.828 × 10−4

Using tabulated data, develop a relation for viscosity in the form of$\mu = \mu(T) = A + BT + CT2 + DT3 + ET4$ .Using the relation developed, predict the dynamic viscosity of water at 50°C at which the reported value is 5.468 × 10−4 Pa·s. Compare your result with the results of Andrade's equation, which is given in the form of$\mu = D \cdot eB/T$, where$D$ and$B$ are constants whose values are to be determined using the viscosity data given.

Step 1 of 5

Using EES, generate a parametric table for the given information. Then calculate a fourth order linear regression with the dynamic viscosity is the dependent variable and the temperature as the independent variable.



Step 2 of 5

Obtain a fourth order relation for viscosity as a function of temperature.

$$\mu(T) = A + BT + CT^2 + DT^3 + ET^4$$
$$= 4.8291758 \times 10^{-1} - 5.68904387 \times 10^{-3} T + 2.49152104 \times 10^{-5} T^2 - 4.86155745 \times 10^{-8} T^3 + 3.56198079 \times 10^{-11} T^4$$

Using the relation above, calculate the dynamic viscosity at 50°C.

First recall the equation to convert temperature in Celsius to Kelvin.

$$T(\mathrm{K}) = T(\mathrm{C}) + 273.15$$
$$= 50 + 273.15$$
$$= 323.15$$

Plug the temperature in Kelvin into the function for dynamic viscosity.

$$\mu(T) = 4.8291758 \times 10^{-1} - 5.68904387 \times 10^{-3} \times (323.15) + 2.49152104 \times 10^{-5} \times (323.15)^2 - 4.86155745 \times 10^{-8} \times (323.15)^3 + 3.56198079 \times 10^{-11} T^4 \times (323.15)^4$$
$$= 5.529 \times 10^{-4}\ \mathrm{Pa \cdot s}$$

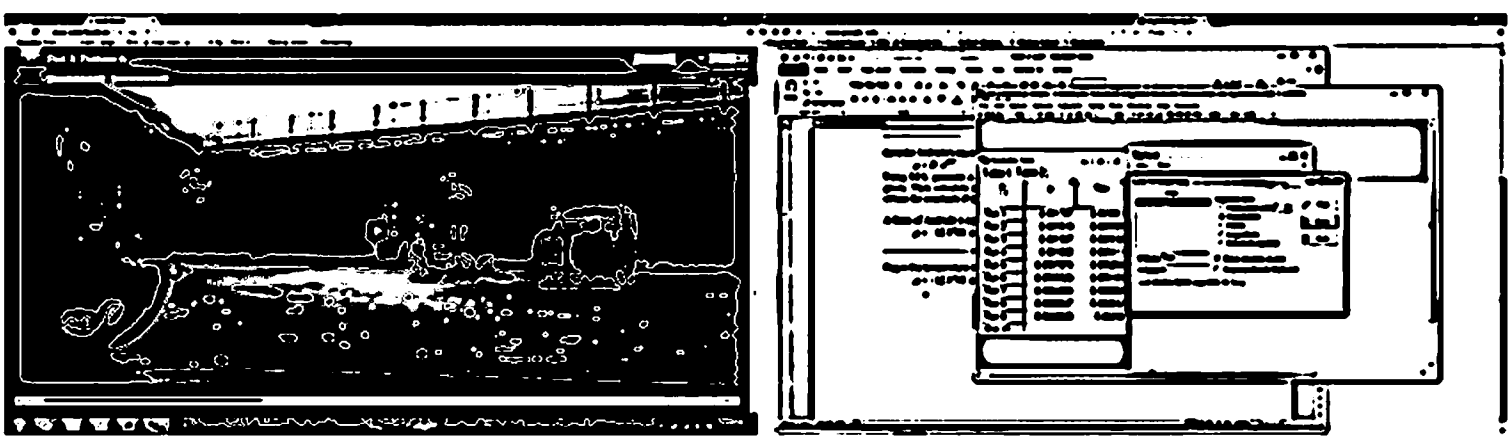
The relation developed predicts a dynamic viscosity of $5.529 \times 10^{-4}\ \mathrm{Pa \cdot s}$ of water at 50°C, which is very close to the report value of $5.468 \times 10^{-4}\ \mathrm{Pa \cdot s}$.

Step 3 of 5

Consider Andrade's equation.

$$\mu = D \cdot e^{B/T}$$

Using EES, generate a plot of dynamic viscosity vs. the inverse of time using the data given. Then calculate an exponential curve fit with the form of Andrade's equation to obtain the constants D and B.



Step 4 of 5

A form of Andrade's equation is obtained.

$$\mu = 0.00000172655 \cdot e^{1876.74/T}$$

Step 5 of 5

Plug in the temperature in Kelvin.

$$\mu = 0.00000172655 \cdot e^{1876.74/323.15}$$
$$= 5.7465 \times 10^{-4}\ \mathrm{Pa \cdot s}$$

The value of dynamic viscosity obtained using Andrade's equation is $5.7465 \times 10^{-4}\ \mathrm{Pa \cdot s}$, which is close to the value obtained using the fourth order relation previously as well as the reported value.

Question: 10.047P

Consider laminar flow of a Newtonian fluid of viscosity μ between two parallel plates. The flow is one-dimensional, and the velocity profile is given as $u(y) = 4u\text{max}\,[y/h - (y/h)2]$, where y is the vertical coordinate from the bottom surface,h is the distance between the two plates, and $u$max is the maximum flow velocity that occurs at midplane. Develop a relation for the drag force exerted on both plates by the fluid in the flow direction per unit area of the plates.

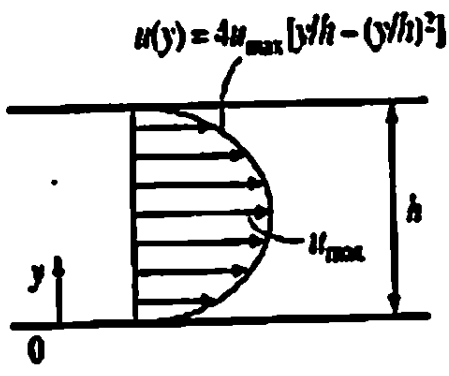


FIGURE P10–47

Step 1 of 4

Given data:

The velocity profile is $u(y) = 4u_{max}\left[(y/h)-(y/h)^2\right]$.

The distance between the plates is h.

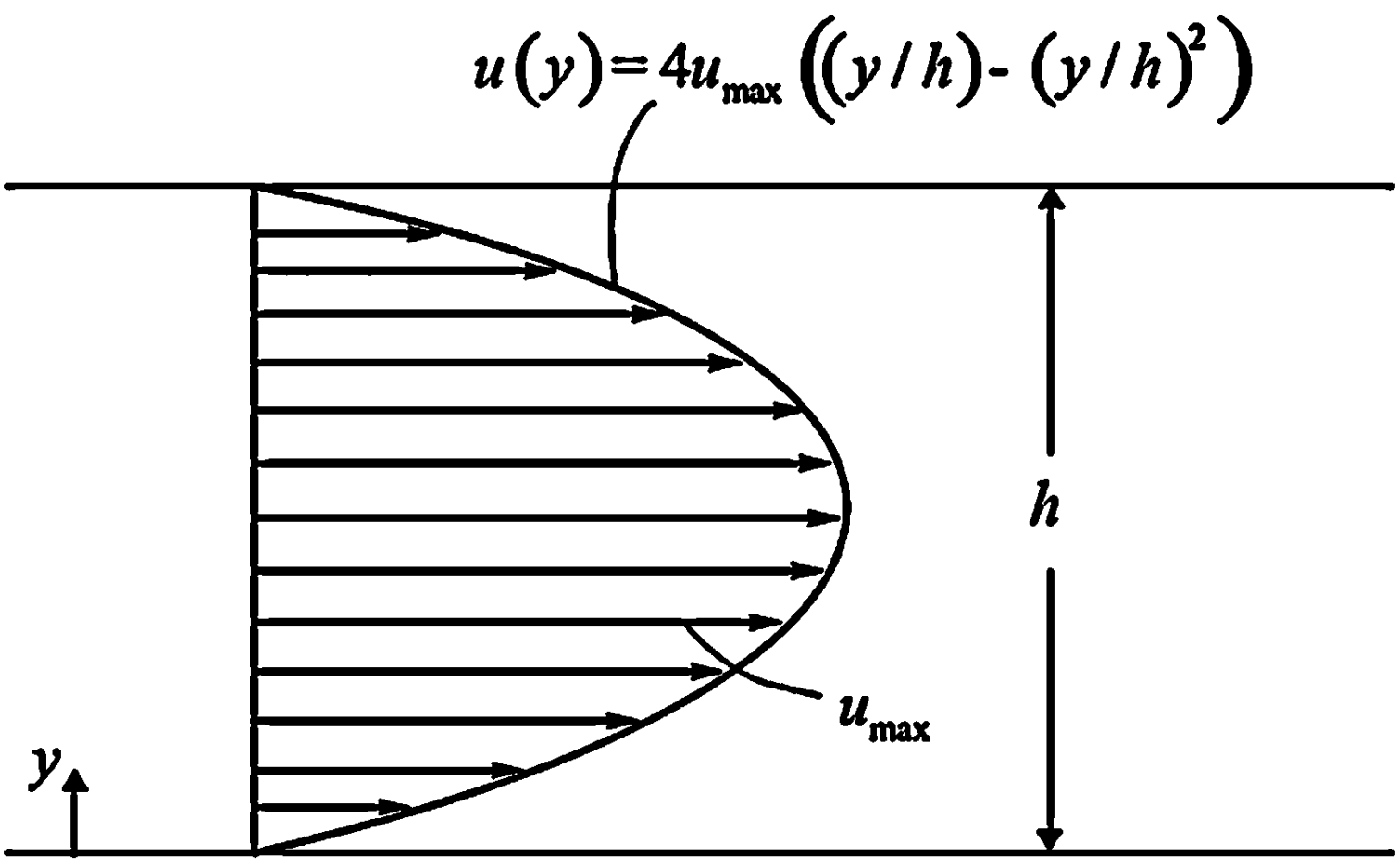
The maximum flow velocity is u_{max}.

The fluid flow is one dimensional.

The fluid is of Newtonian nature.

Step 2 of 4

Given figure:



The velocity profile given in the problem is symmetric. Hence we can assume that the shear stress acting on the bottom plate will be equal to the shear stress acting on the top plate.

Step 3 of 4

The shear stress acting on the bottom of the plate,

$$\tau_{Bottom} = \mu\frac{du}{dy}$$

$$\tau_{Bottom} = \mu\frac{d}{dy}\left(4u_{max}\left[(y/h)-(y/h)^2\right]\right)$$

$$= 4u_{max}\mu\frac{d}{dy}\left[(y/h)-(y/h)^2\right]$$

Differentiating the above equation, we get

$$\tau_{Bottom} = 4u_{max}\left[(1/h)-(2y/h^2)\right]$$

At bottom of the plate $y = 0$.

$$\tau_{Bottom} = 4\mu u_{max}\left[\frac{1}{h}-\frac{2(0)}{h^2}\right]$$

$$= \frac{4\mu u_{max}}{h}$$

Due to symmetry,

Shear stress at top plate is equal to shear stress at bottom plate.

$$\tau_{Top} = \tau_{Bottom}$$

$$= \frac{4\mu u_{max}}{h}$$

Step 4 of 4

The shear force acting on the upper and lower plates can be expressed as

$$F = \left(\tau_{Bottom}A + \tau_{Top}A\right)$$

$$= \frac{4\mu u_{max}}{h} + \frac{4\mu u_{max}}{h}$$

$$F = \frac{8\mu u_{max}A}{h} \quad \text{..... (1)}$$

Where,

A = Area of plate

The friction drag per unit area is

From equation (1)

$$\frac{F}{A} = \frac{8\mu u_{max}}{h}$$

The drag force exerted on both plates by the fluid in the flow direction per unit area is

$$\boxed{\frac{F}{A} = \frac{8\mu u_{max}}{h}}.$$

Question: 10.048P

Some non-Newtonian fluids behave as a Bingham plastic for which shear stress can be expressed as $\tau = \tau_y + \mu(du/dr)$. For laminar flow of a Bingham plastic in a horizontal pipe of radius R, the velocity profile is given as $u(r)=(\Delta P/4\mu L)(r2-R2)+(\tau y/\mu)(r- R)$, where $\Delta P/L$ is the constant pressure drop along the pipe per unit length, μ is the dynamic viscosity, r is the radial distance from the centerline, and τy is the yield stress of Bingham plastic. Determine (*a*) the shear stress at the pipe wall and (*b*) the drag force acting on a pipe section of length L.

Step 1 of 4

Given data:

The shear stress, $\tau = \tau_y + \mu(du/dr)$.

The velocity profile, $u(r) = (\Delta P/4\mu L)(r^2 - R^2) + (\tau_y/\mu)(r - R)$.

The constant pressure drop per unit length is $\Delta P/L$.

The dynamic viscosity is μ.

The radial distance from center line is r.

The yield stress of Bingham plastic is τ_y.

Step 2 of 4

(a)

The fluid considered is non-Newtonian and behaves like Bingham plastic.

The flow is laminar taking place inside a pipe.

The radial distance is along the y-coordinate.

The shear stress at pipe wall can be written as

$$\begin{aligned}\tau_r &= \mu\frac{du}{dr}\\ &= \mu\left[\frac{d}{dr}\left(\frac{\Delta P}{4\mu L}(r^2 - R^2) + \frac{\tau_y}{\mu}(r - R)\right)\right]\\ &= \mu\left[\left(\frac{\Delta P}{4\mu L}\right)\frac{d}{dr}(r^2 - R^2) + \left(\frac{\tau_y}{\mu}\right)\frac{d}{dr}(r - R)\right]\end{aligned}$$

Step 3 of 4

Differentiating the above equation, we get

$$\begin{aligned}\tau_r &= \mu\left(\frac{2R\Delta P}{4\mu L} + \frac{\tau_y}{\mu}\right)\\ &= \mu\left(\frac{R\Delta P}{2\mu L} + \frac{\tau_y}{\mu}\right)\end{aligned}$$

The shear stress acting on the pipe wall can be expressed as

$$\begin{aligned}\tau &= \tau_y + \tau_r\\ &= \tau_y + \mu\left(\frac{r\Delta P}{2\mu L} + \frac{\tau_y}{\mu}\right)\\ &= 2\tau_y + \frac{R\Delta P}{2L}\end{aligned}$$

The shear stress acting on the pipe wall is

$$\boxed{\tau = 2\tau_y + \frac{R\Delta P}{2L}}.$$

Step 4 of 4

(b)

The area of the pipe exposed to the fluid is,

$$A = 2\pi rL$$

The drag force exerted on the pipe surface is

$$\begin{aligned}F_D &= \tau A\\ &= \left(2\tau_y + \frac{R\Delta P}{2L}\right)(2\pi rL)\end{aligned}$$

The drag force exerted on a pipe section of length L is

$$\boxed{F_D = \left(2\tau_y + \frac{R\Delta P}{2L}\right)(2\pi rL)}.$$

Question: 10.049P

A 0.018-in-diameter glass tube is inserted into mercury, which makes a contact angle of 140° with glass. Determine the capillary drop of mercury in the tube at 68°F.

Step 1 of 2

Given data:

Diameter of the glass is tube, $d = 0.018\text{in}$

Radius of the glass is tube, $R = 2.286\text{m}$

Contact angle made by the mercury, $\phi = 140°$

Temperature of the mercury is $t = 68°\text{F} = 20°\text{C}$

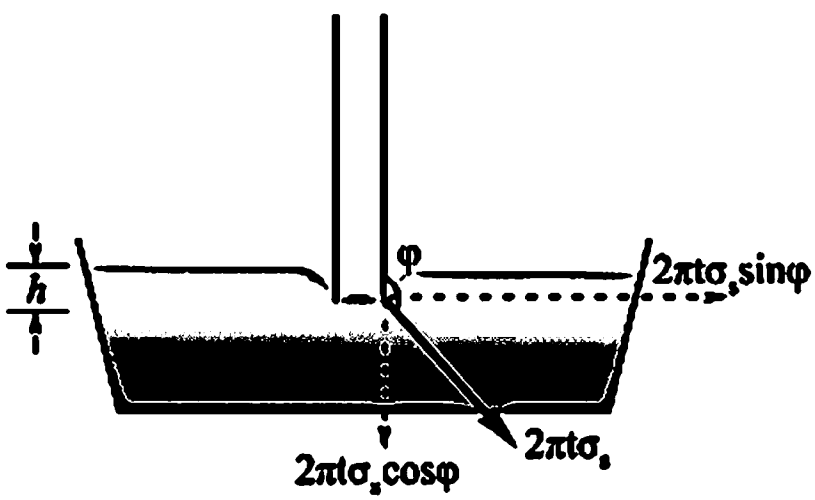
Surface tension of the mercury at 20°C $\sigma_s = 0.44\,\text{N/m}$

Density of the mercury, $\rho = 13600\,\text{kg/m}^3$

Acceleration due to gravity $g = 9.81\,\text{m/s}^2$

Step 2 of 2

Figure showing forces acting near the meniscus of mercury,



We know that capillary drop is given by,

$$h = \frac{2\sigma\cos\phi}{\rho g R}$$

$$= \frac{2\times(0.44\,\text{N/m})\times(\cos 140°)}{(13600\,\text{kg/m}^3)\times(9.81\,\text{m/s}^2)\times(2.286\times10^{-4}\,\text{m})}$$

$$= -0.022\,\text{m}$$

$$= -0.8701\,\text{in.}$$

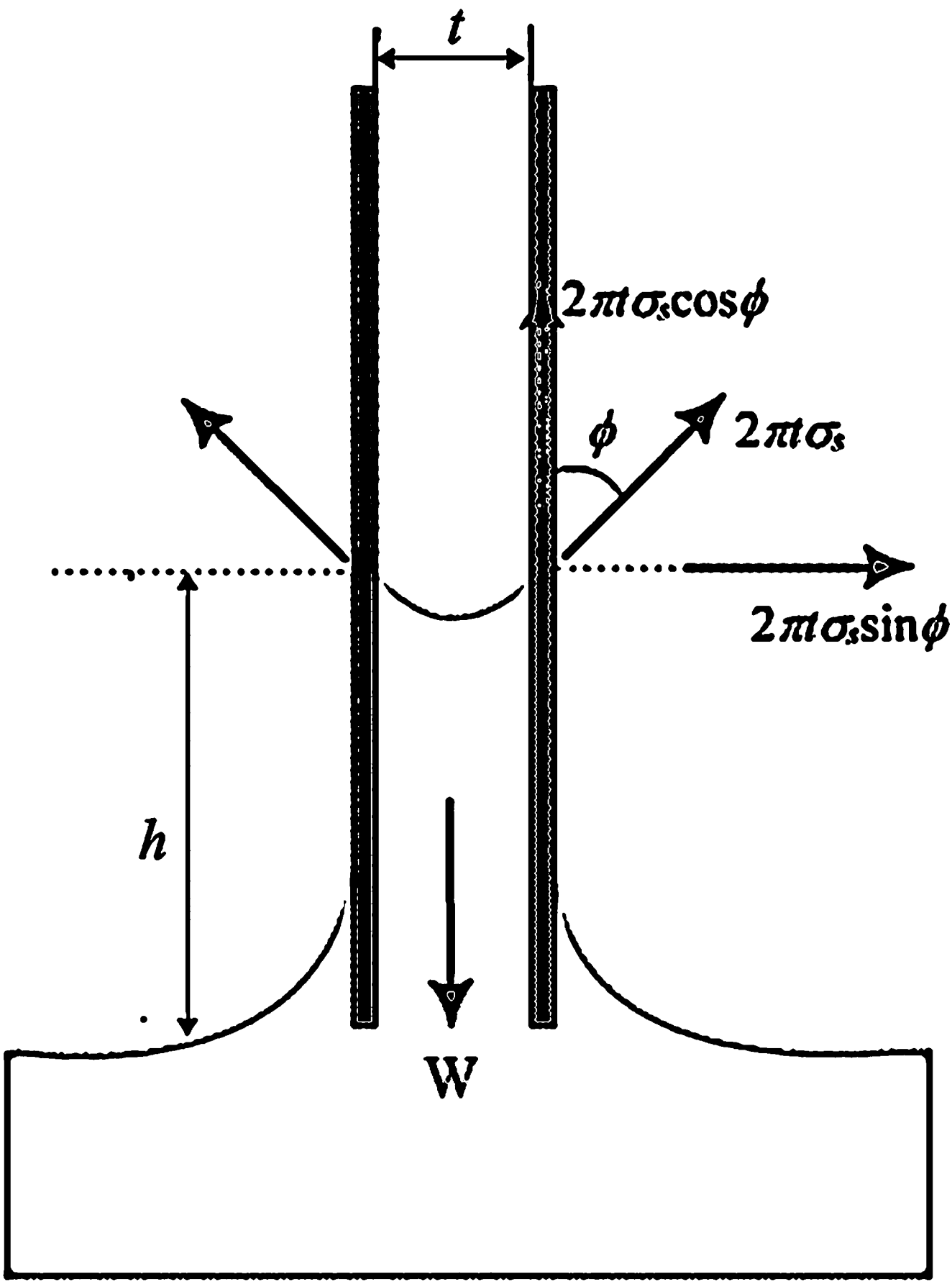
The capillary drop in mercury is obtained as $\boxed{h = -0.8701\,\text{in.}}$.

Question: 10.050P

Derive a relation for the capillary rise of a liquid between two large parallel plates a distance *t* apart inserted into the liquid vertically. Take the contact angle to be ϕ.

Step 1 of 3

Let us draw the diagram:



Step 2 of 3

Let us assume that the water and the plate surfaces are free from impurities and the experiment takes place in atmospheric pressure.

From the figure we know that the bottom of the liquid is at the same level as the free surface of the reservoir where the atmospheric pressure acts. The same pressure will be acting on the top of the liquid column. Thus the two effects cancel each other.

The weight of the liquid column is

$W = mg$ ..... (1)

Where,

m = Mass of the liquid

$= \rho V$

ρ = Density of liquid

V = Volume of the liquid

$= b \times t \times h$

Step 3 of 3

Substituting the values in equation (1)

$$W = (\rho V) g$$
$$= \rho g (b \times t \times h)$$

Let us equate the vertical component of surface tension and weight.

$$W = F_{surface}$$
$$\rho g (b \times t \times h) = 2\pi t \sigma_s \cos\phi$$
$$h = \frac{2\pi\sigma_s}{\rho g b} \cos\phi$$

The required capillary rise is $\boxed{h = \frac{2\pi\sigma_s}{\rho g b} \cos\phi}$

Question: 10.051P

Consider a 40-cm-long journal bearing that is lubricated with oil whose viscosity is 0.1 kg/m·s at 20°C at the beginning of operation and 0.008 kg/m·s at the anticipated steady operating temperature of 80°C. The diameter of the shaft is 8 cm, and the average gap between the shaft and the journal is 0.08 cm. Determine the torque needed to overcome the bearing friction initially and during steady operation when the shaft is rotated at 800 rpm.

Step 1 of 5

Given data:

The length of the journal bearing, $L = 40$ cm

The viscosity of the lubricant at 20°C, $\mu_{20} = 0.1\ \text{kg/m} \cdot \text{s}$

The viscosity of the lubricant at 80°C, $\mu_{80} = 0.008\ \text{kg/m} \cdot \text{s}$

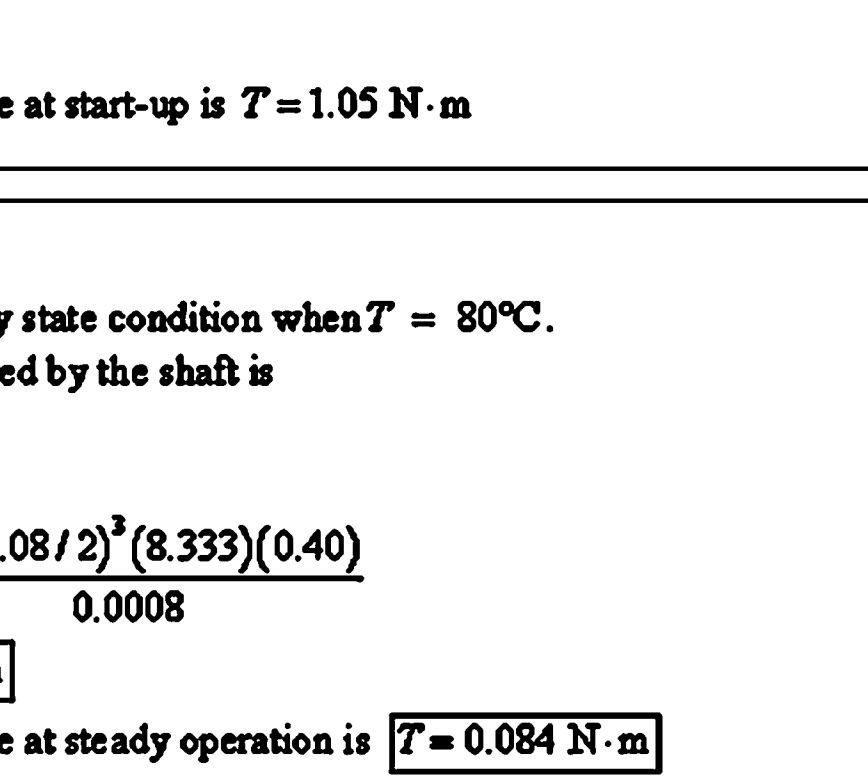
The velocity of the shaft, $n = 800$ rpm

The diameter of the shaft, $d = 0.08$ m

The average gap between the shaft and journal, $l = 0.08$ cm

Step 2 of 5

Let us draw the diagram:



Step 3 of 5

Let us assume that the fluid is Newtonian and the end effects are negligible.

The velocity of the shaft is

$n = 800$ rpm

$$= \left(\frac{800}{60}\right) \text{rps}$$

$$= 8.333 \text{ rps}$$

The torque generated by the shaft can be written as,

$$T = \mu \frac{4\pi^2 R^3 n L}{l}$$

Step 4 of 5

Consider the start-up condition when $T = 20°\text{C}$.

The torque generated by the shaft is

$$T = \mu_{20} \frac{4\pi^2 R^3 n L}{l}$$

$$= (0.1) \frac{4\pi^2 (0.08/2)^3 (8.333)(0.40)}{0.0008}$$

$$\boxed{T = 1.05\ \text{N} \cdot \text{m}}$$

The required torque at start-up is $T = 1.05\ \text{N} \cdot \text{m}$

Step 5 of 5

Consider the steady state condition when $T = 80°\text{C}$.

The torque generated by the shaft is

$$T = \mu_{80} \frac{4\pi^2 R^3 n L}{l}$$

$$= (0.008) \frac{4\pi^2 (0.08/2)^3 (8.333)(0.40)}{0.0008}$$

$$\boxed{T = 0.084\ \text{N} \cdot \text{m}}$$

The required torque at steady operation is $\boxed{T = 0.084\ \text{N} \cdot \text{m}}$

Question: 10.052P

The diameter of one arm of a U-tube is 5 mm while the other arm is large. If the U-tube contains some water, and both surface are exposed to atmospheric pressure, determine the difference between the water levels in the two arms.

Step 1 of 4

Given data:

A U-tube manometer is used.

The diameter of one arm is $D = 5\text{ mm}$

$= 0.005\text{ m}$

The diameter of the other arm is large.

The water surfaces in both arms are exposed to atmospheric pressure.

Step 2 of 4

Let us assume that there are no impurities in water and no contamination on the tube surfaces.

Let us assume the temperature to be $T = 20^\circ\text{C}$

At $T = 20^\circ\text{C}$ the properties of water are

Surface tension, $\sigma_s = 0.073\text{ N/m}$

And the contact angle is approximately $\phi = 0^\circ$

Consider the arm with very large diameter.

Since the diameter is large the capillary rise h will be negligible. Hence we can assume h will be zero in the arm having large diameter.

Then

$h_1 = 0$

Step 3 of 4

Consider the tube of diameter $D_1 = 0.005\text{ m}$.

The capillary rise is,

$$h_2 = \frac{2\sigma_s}{\rho g R}\cos\phi \qquad \ldots\ldots (1)$$

Where,

R = Radius of the tube

$$= \frac{D}{2}$$

$$= \frac{0.005}{2}$$

$$= 2.5\times10^{-3}\text{ m}$$

ρ = Density of water

$= 1000\text{ kg/m}^3$.

g = acceleration due to gravity

$= 9.81\text{ m/s}^2$.

Step 4 of 4

Substituting the above values in equation (1), we get

$$h_2 = \frac{2(0.073\text{ N/m})}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(2.5\times10^{-3}\text{ m})}\cos 0^\circ$$

$$= 0.0593\text{ m}$$

The difference between the water levels in the two arms is

$\Delta h = h_2 - h_1$

$= 0.0593 - 0$

$= 0.0593$

$\boxed{\Delta h = 0.0593\text{ m}}$

The difference between the water levels in the two arms is $\boxed{\Delta h = 0.0593\text{ m}}$.

Question: 10.053P

Some rocks or bricks contain small air pockets in them and have a spongy structure. Assuming the air spaces form columns of an average diameter of 0.004 mm, determine how high water can rise in such a material. Take the surface tension of the air-water interface in that material to be 0.085 N/m.

Step 1 of 3

Given data:

The average diameter of the columns, $D = 0.004$ mm

$= 0.004 \times 10^{-3}$ m

The surface tension of the air water interface, $\sigma_s = 0.085$ N/m

Let us assume the water is not contaminated, the walls of the columns are free from impurities and the pressure in the air spaces is equal to the atmospheric pressure.

Step 2 of 3

The capillary rise is given by

$$h = \frac{2\sigma_s}{\rho g R}\cos\phi \qquad \text{..... (1)}$$

Where,

R = Radius of the tube

$$= \frac{D}{2}$$

$$= \frac{0.004 \times 10^{-3}}{2}$$

$$= 0.002 \times 10^{-3} \text{ m}$$

Density of water, $\rho = 1000$ kg/m^3.

Acceleration due to gravity $g = 9.81$ m/s^2.

Contact angle $\phi = 0^\circ$

Step 3 of 3

Substituting the values in equation (1), we get

$$h = \frac{2(0.085)}{1000 \times 9.81 \times 0.002 \times 10^{-3}}\cos 0^\circ$$

$$= 8.66 \text{ m}$$

The rise of water in the material is $\boxed{h = 8.66 \text{ m}}$.

CHAPTER 11

SOLUTIONS

Question: 11.001P

Define the resultant hydrostatic force acting on a submerged surface, and the center of pressure.

Step 1 of 2

The hydrostatic pressure is defined as

$$P = \rho g h \qquad \ldots\ldots (1)$$

Here ρ is the fluid density. g Is the acceleration due to gravity. And h is the distance from the liquid surface.

As no shear stress exists in a static fluid, all hydrostatic forces on any element of a submerged surface must act in a direction normal to the surface.

Step 2 of 2

The above equation (1) shows that pressure distribution varies linearly over the any liquid surface.

The magnitude of the resultant hydrostatic force of a submerged plane is equal to the product of pressure P_c at the centroid of the surface and area A of the surface.

$$F_R = P_C A$$

The point of application of resultant force on the plane surface is known as centre of pressure.

Question: 11.002P

Someone claims that she can determine the magnitude of the hydrostatic force acting on a plane surface submerged in water regardless of its shape and orientation if she knew the vertical distance of the centroid of the surface from the free surface and the area of the surface. Is this a valid claim? Explain.

Step 1 of 2

For an inclined surface if we know the vertical distance and area of the surface we can determine the magnitude of resultant force.

Yes. It is a valid claim.

Step 2 of 2

Whatever may be the inclination of the submerged plane surface to the free liquid, the magnitude of the resultant hydrostatic force equals the product of area and the pressure at the centroid of the area.

Find the resultant force.

$$F_R = P_c A$$

$$= P_{avg} A$$

Here:

$P_c = P_0 + \rho g h_c$ is the pressure at the centroid of the surface.

This is equivalent to the average pressure P_{avg} on the surface.

$h_c = y_c + \sin\theta$ is the vertical distance of the centroid from the free surface of the liquid.

Question: 11.003P

A submerged horizontal flat plate is suspended in water by a string attached at the centroid of its upper surface. Now the plate is rotated 45° about an axis that passes through its centroid. Discuss the change on the hydrostatic force acting on the top surface of this plate as a result of this rotation. Assume the plate remains submerged at all times.

Step 1 of 1

The magnitude of the resultant hydrostatic force equals the product of area and the pressure at the centroid of the surfaces, i.e.

$F = P_c A$

On rotating of the plate about the axis passing through the centroid of the plate, whatever may be the inclination, the pressure force remains constant.

Therefore there will be no change in the magnitude of hydrostatic force on the surface when it is rotated by 45°.

Question: 11.004P

You may have noticed that dams are much thicker at the bottom. Explain why dams are built that way.

Step 1 of 1

Basic equation for pressure

$P = \gamma h$

Pressure is directly proportional to height. So when height increases pressure also increases.

To withstand high pressure at the bottom of the dam, basements are built much wider than the top. So the bottom of the dam will be thicker.

Question: 11.005P

Consider a submerged curved surface. Explain how you would determine the horizontal component of the hydrostatic force acting on this surface.

Step 1 of 2

The horizontal component of the hydrostatic force acting on a curved surface is equal to the hydrostatic force acting on the vertical projection of the curved surface.

Horizontal component of curved surface,

$F_H = F_x$

$F_H = \rho g A h_c$

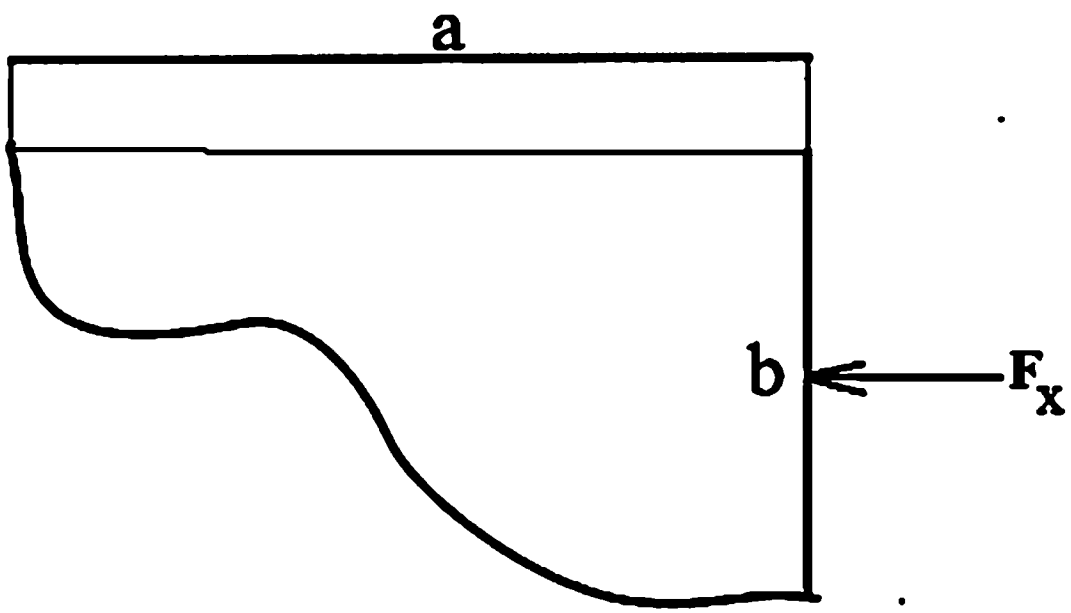
Here

h_c **is the centroidal distance of the surface from the free surface.**

A **is the cross sectional area of the curved surface.**

h_c **and** A **are depends on curved surface of the object.**

Step 2 of 2



In the above figure:

a is the horizontal projection of the curved surface.

b is the vertical projection of the curved surface.

Question: 11.006P

Consider a submerged curved surface. Explain how you would determine the vertical component of the hydrostatic force acting on this surface.

Step 1 of 2

The vertical component of the hydrostatic force acting on a curved surface is equal to the hydrostatic force acting on the horizontal projection of the curved surface, plus weight of the fluid block. (Direction of the weight of the fluid block should be taken into consideration).

Vertical force component of curved surface

$$F_v = F_y \pm W$$

It is a vector addition if both F_y and W act in the same direction, and subtract if they act in opposite directions.

$$F_v = \rho g A h_c \pm W$$

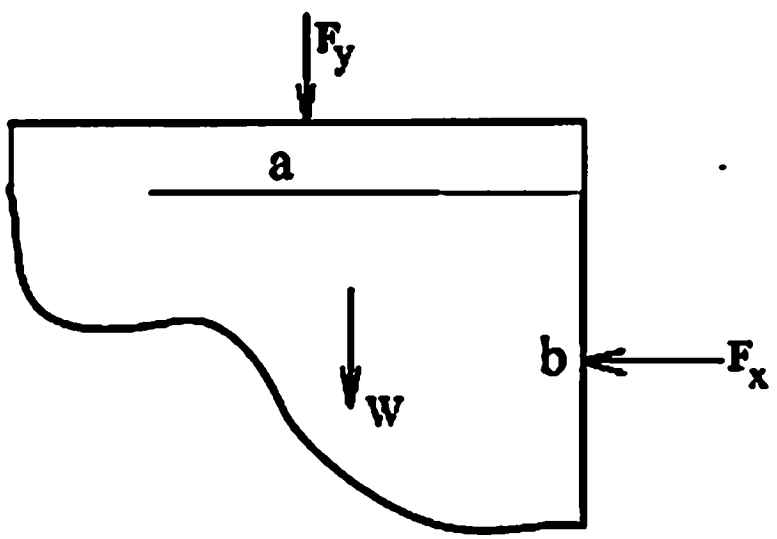
Where

h_c is the centroidal distance of the surface from the free surface.

A is the cross sectional area of the curved surface.

h_c and A are depends on curved surface of the object.

Step 2 of 2



In the above figure:

a is the horizontal projection of the curved surface.

b is the vertical projection of the curved surface.

Question: 11.007P

Consider a circular surface subjected to hydrostatic forces by a constant density liquid. If the magnitudes of the horizontal and vertical components of the resultant hydrostatic force are determined, explain how you would find the line of action of this force.

Step 1 of 2

When the curved surface is circular, the resultant of hydrostatic force acting on the surface always passes through the centre of the circle. This is because, as the pressure forces are normal to the surface, all the lines which are normal to the surface of a circle pass through the center of the circle.

As the pressure forces form a concurrent force system, it can be taken as a single equivalent force at that point.

Step 2 of 2

When magnitudes of the horizontal and vertical components of the resultant hydrostatic forces are given, then the line of action is given as

$$\tan\theta = \frac{F_V}{F_H}$$

Here θ is the angle made by the line of action, F_H is the horizontal force component of hydrostatic force and F_V is the vertical force component of hydrostatic force.

Question: 11.008P

Consider a heavy car submerged in water in a lake with a flat bottom. The driver's side door of the car is 1.1 m high and 0.9 m wide, and the top edge of the door is 10 m below the water surface. Determine the net force acting on the door (normal to its surface) and the location of the pressure center if (*a*) the car is well-sealed and it contains air at atmospheric pressure and (*b*) the car is filled with water.

Step 1 of 5

Given data:

Height of driver's side door, $b = 1.1\,\text{m}$
Width of driver's side door, $w = 0.9\,\text{m}$
Height of the car door top edge from the water surface, $s = 10\,\text{m}$
Density of water, $\rho = 1000\,\text{kg/m}^3$
Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$
The distance of centroid of car door from the surface of water,

$$h_c = s + \frac{b}{2}$$
$$= 10 + \frac{1.1}{2}$$
$$= 10.55\,\text{m}$$

Step 2 of 5

(a)
The door can be considered as a vertical rectangular plate.
The average pressure acting on the car door,

$$P_{avg} = \rho g h_c$$
$$= 1000 \times 9.81 \times 10.55$$
$$= 103495.5\,\text{N/m}^2$$
$$= 103.495\,\text{kN/m}^2$$

Area of the rectangular car door,

$$A = b \times w$$
$$= 1.1 \times 0.9$$
$$= 0.99\,\text{m}^2$$

Step 3 of 5

The resultant hydrostatic force on the car door,

$$F_R = P_{avg} \times A$$
$$= 103.495 \times 0.99$$
$$= 102.46\,\text{kN}$$

Therefore the resultant hydrostatic force on the car door, $\boxed{F_R = 102.46\ \text{kN}}$.
The moment of inertia of rectangle about x-axis,

$$I_{xx} = \frac{wb^3}{12}$$
$$= \frac{0.9 \times 1.1^3}{12}$$
$$= 0.099\,\text{m}^4$$

Step 4 of 5

The perpendicular distance between hydrostatic force and surface of water, $y_c = h_c = 10.55\,\text{m}$
The line of action of the hydrostatic force from the free surface of the water,

$$y_p = y_c + \frac{I_{xx}}{h_c A}$$
$$= 10.55 + \frac{0.099}{10.55 \times 0.99}$$
$$= 10.55 + 9.478 \times 10^{-3}$$
$$= 10.559\,\text{m}$$

Therefore the line of action of the hydrostatic force from the free surface of the water, $\boxed{y_p = 10.559\ \text{m}}$.

Step 5 of 5

(b)
The net force acting on the door is zero when the car is filled with water, because the door is having water on both sides, therefore both inside and outside forces will get neutralized and the net force acing on the door will become zero.

Question: 11.009P pg 1

A long, solid cylinder of radius 2 ft hinged at point *A* is used as an automatic gate, as shown in Fig. P11–9E. When the water level reaches 15 ft, the cylindrical gate opens by turning about the hinge at point *A*. Determine (*a*) the hydrostatic force acting on the cylinder and its line of action when the gate opens and (*b*) the weight of the cylinder per ft length of the cylinder.

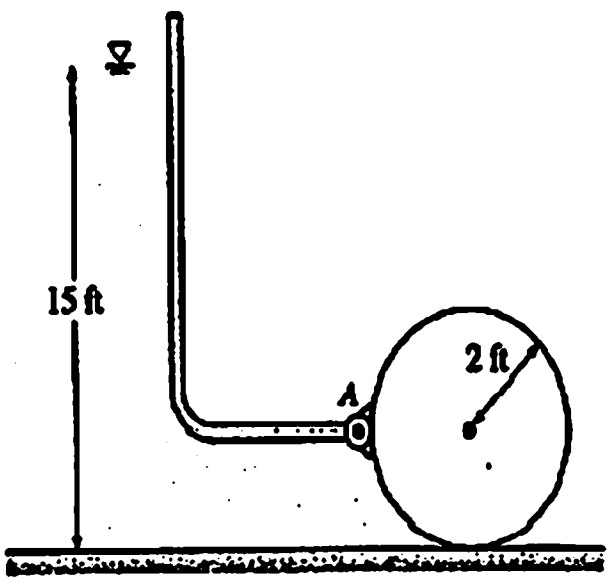
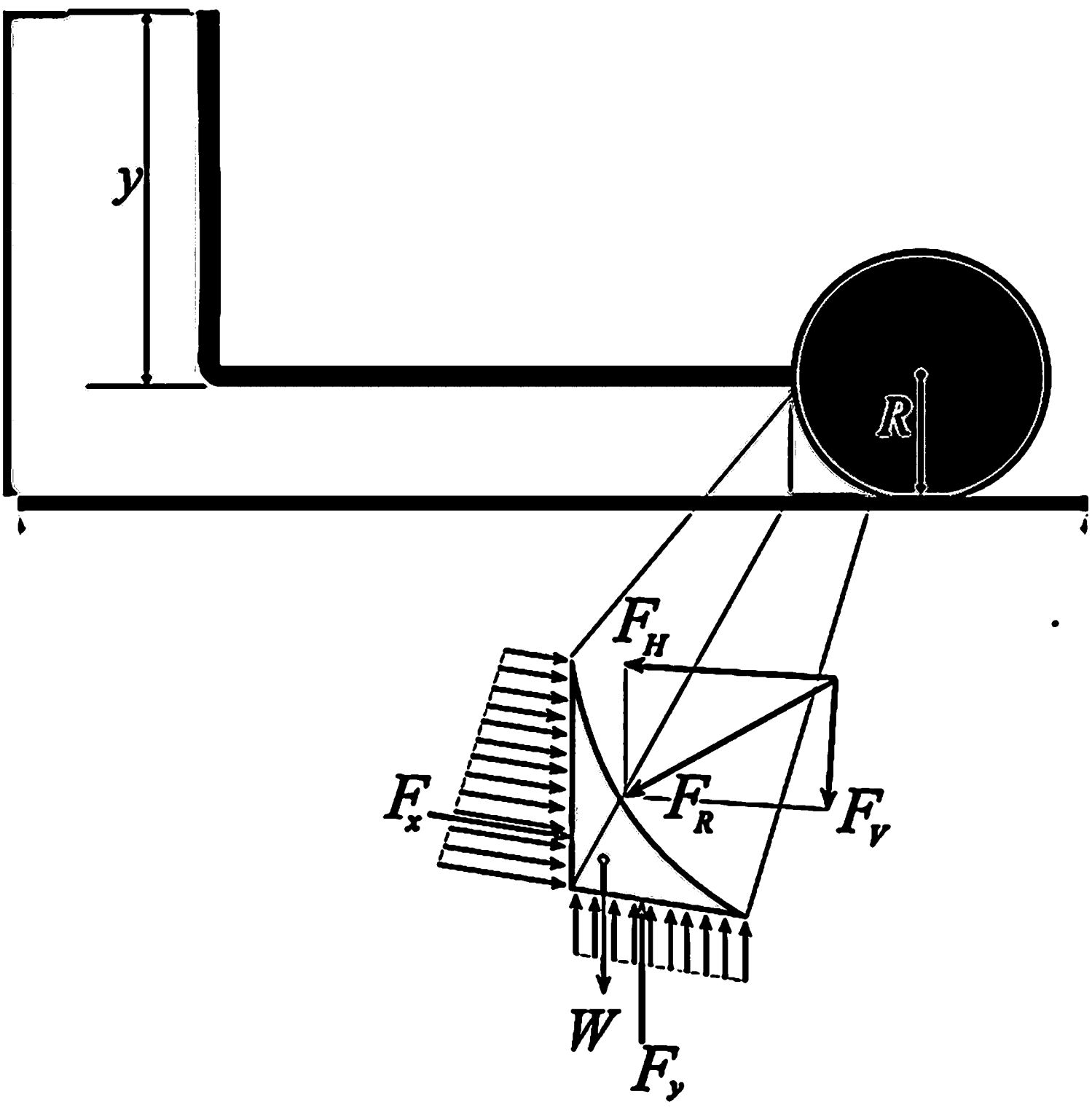


FIGURE P11-9E

Step 1 of 9

Draw the schematic diagram of the automatic gate:



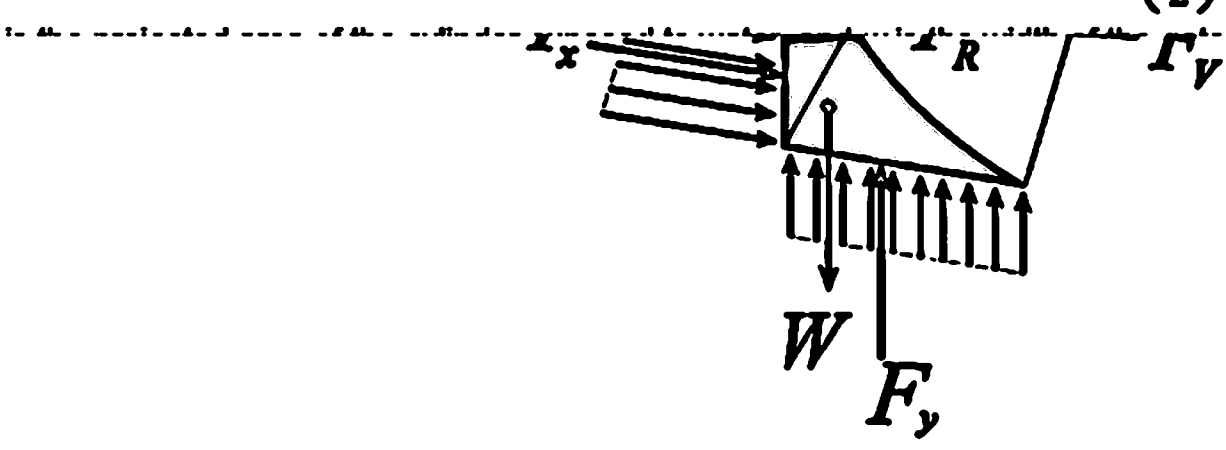
Step 2 of 9

Consider the liquid part enclosed by the cylinder and the vertical projection of the cylinder shown in the figure:

Calculate the horizontal force acting on the projection of enclosed element (on to the vertical surface) $\left(F_H\right)$:

$$F_H = F_x$$
$$= \rho g h_C A$$
$$= \rho g\left(y+\frac{R}{2}\right)(R\times w)$$

Here, ρ is the density of water, *g* is the acceleration due gravity, *R* is the radius of the cylinder, h_C is the distance between the centre of the gravity of vertical projection of the cylinder $\left(\frac{R}{2}\right)$ and the free surface, *A*



Step 2 of 9

Consider the liquid part enclosed by the cylinder and the vertical projection of the cylinder shown in the figure:

Calculate the horizontal force acting on the projection of enclosed element (on to the vertical surface) $\left(F_H\right)$:

$$F_H = F_x$$
$$= \rho g h_C A$$
$$= \rho g\left(y+\frac{R}{2}\right)(R\times w)$$

Here, ρ is the density of water, *g* is the acceleration due gravity, *R* is the radius of the cylinder, h_C is the distance between the centre of the gravity of vertical projection of the cylinder $\left(\frac{R}{2}\right)$ and the free surface, *A*

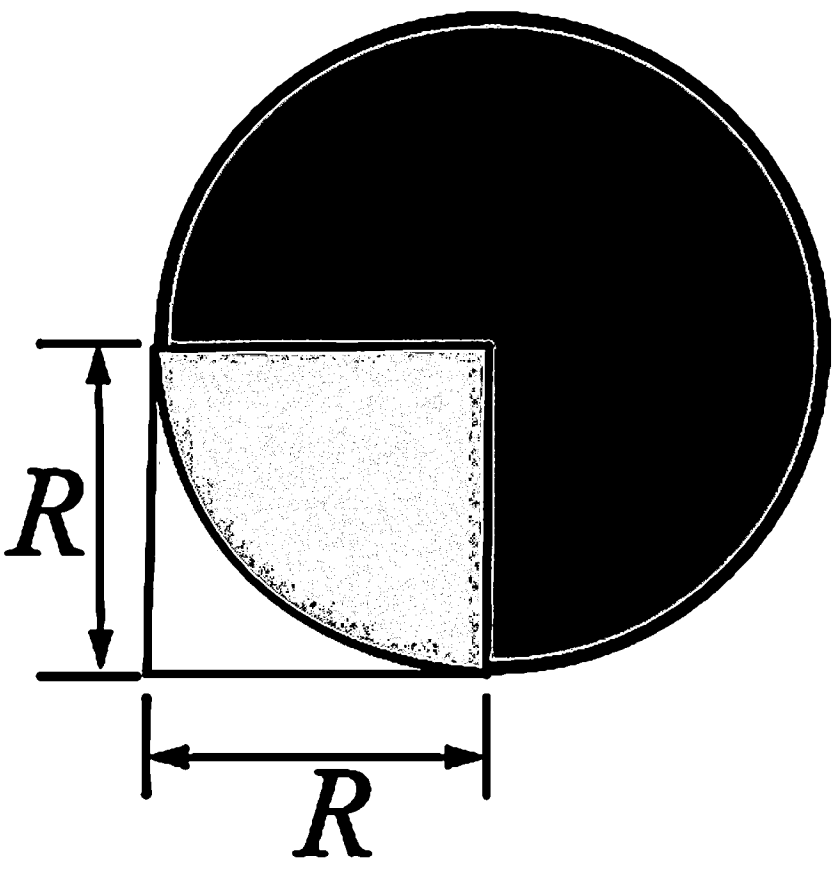
Step 4 of 9

Draw the volume of the enclosed element, (*V*) per unit width using the following figure.



Step 4 of 9

Draw the volume of the enclosed element, (V) per unit width using the following figure.



Step 5 of 9

Calculate the volume of the enclosed element using the following equation.

$$V=\left(R^2-\frac{\pi R^2}{4}\right)w$$

Substitute 1 ft for w.

$$V=\left(2^2-\frac{\pi\times 2^2}{4}\right)\times 1$$
$$=0.854\ \text{ft}^3$$

Step 6 of 9

Calculate the weight of the enclosed element per unit width of the gate.

$$W=mg$$
$$=\rho Vg$$
$$=62.4\times 0.858\times 32.2$$
$$=1723962.24\ \text{lbm}\cdot\text{ft/s}^2\times\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}$$
$$=53.54\ \text{lbf}$$

Calculate the resultant vertical force acting on the enclosed element.

$$F_v=F_y-W$$
$$=1873.51-53.54$$
$$=1819.97\ \text{lbf}$$

Step 7 of 9

Calculate the hydrostatic force acing on the gate, $\left(F_R\right)$ using the following relation.

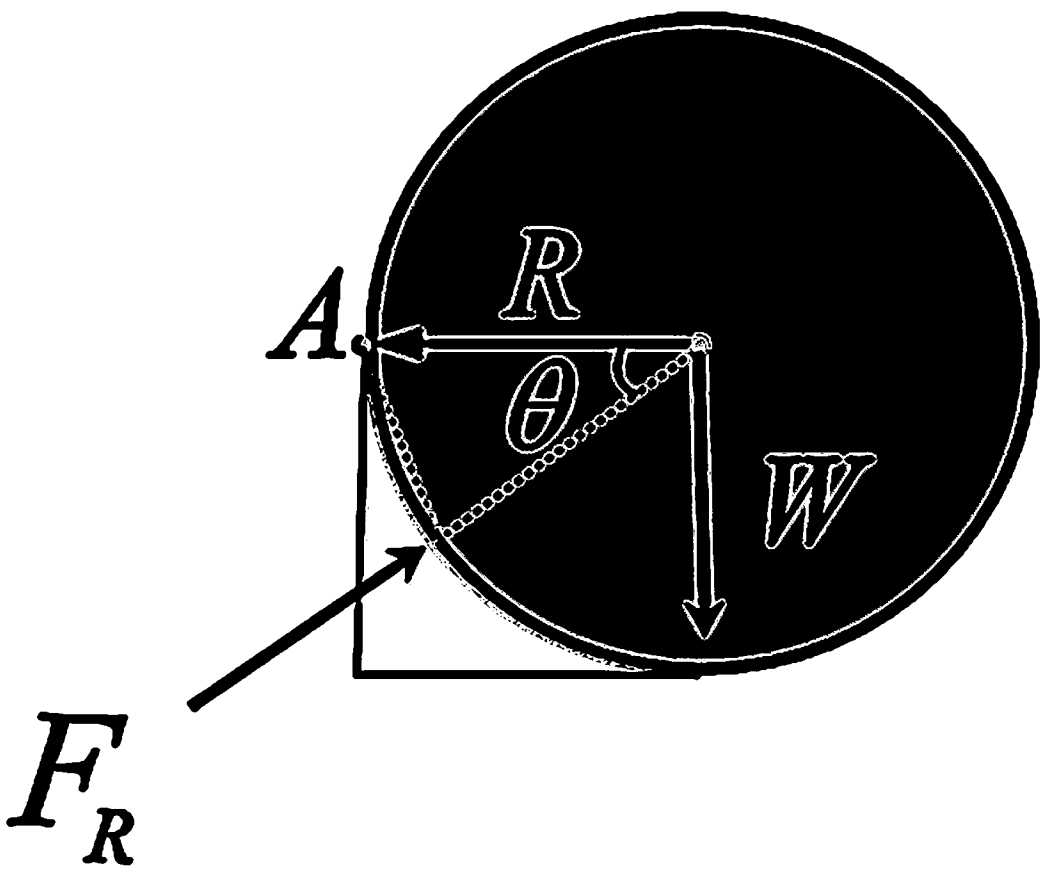
$$F_R=\sqrt{F_H{}^2+F_V{}^2}$$
$$=\sqrt{1748.61^2+1819.97^2}$$
$$=2523.87\ \text{lbf}$$

Calculate the direction of the hydrostatic force using the following equation.

$$\theta=\tan^{-1}\left(\frac{F_V}{F_H}\right)$$
$$=\tan^{-1}\left(\frac{1819.97}{1748.61}\right)$$
$$=46.15^\circ$$

Step 8 of 9

Consider that the gate is about lift:



Step 9 of 9

Take moment about the hinge A:

$$\sum M_{hinge}=0$$
$$F_R R\sin\theta-W_{cylinder}R=0$$
$$\left(2523.87\times 2\times\sin 46.15^\circ\right)-\left(W_{cylinder}\times 2\right)=0$$
$$W_{cylinder}\times 2=3640.20$$
$$W_{cylinder}=1820.1\ \text{lbf}$$

Therefore, the weight of the cylinder per unit width is $\boxed{1820.1\ \text{lbf}}$.

$W_{cylinder} \times 2 = 3640.20$

$W_{cylinder} = 1820.1 \text{ lbf}$

11.009P pg 3

Therefore, the weight of the cylinder per unit width is $\boxed{1820.1 \text{ lbf}}$.

Question: 11.010P

Consider a 4-m-long, 4-m-wide, and 1.5 aboveground swimming pool that is filled with water to their rim. (*a*) Determine the hydrostatic force on each wall and distance of the line of action of this force from the (*b*) If the height of the walls of the pool is doubled and pool is filled, will the hydrostatic force on each wall double or quadruple? Why?

Step 1 of 5

Given:

Length of swimming pool $= 4\text{m}$

Width of swimming pool $= 4\text{m}$

Height of swimming pool $= 1.5\text{m}$

Step 2 of 5

Specific weight of water, $\gamma = 9810\text{N/m}^3$

Area, $A = 4 \times 1.5$

$= 6\text{m}^2$

The distance from free surface of water to centroid of the wall:

$$h_c = \frac{h}{2}$$
$$= \frac{1.5}{2}$$
$$= 0.75\text{m}$$

Step 3 of 5

(a)

Hydrostatic force on the wall:

$$F = \gamma A h_c$$
$$= 9810 \times 6 \times 0.75$$
$$= 44145\text{N}$$
$$= 44.145\text{kN}$$

Thus the hydrostatic force on each wall is $\boxed{F = 44.145\text{kN}}$

Step 4 of 5

Distance of line of action from the ground:

$$y = \frac{h}{3}$$
$$= \frac{1.5}{3}$$
$$= 0.5\text{m}$$

Thus the distance of line of action from the ground is $\boxed{0.5\text{m}}$.

Step 5 of 5

(b)

Hydrostatic force on the wall when the height of the pool is doubled:

$$F = \gamma A h_c$$
$$= \gamma \frac{2h}{2}(2h \times w)$$
$$= 4\left[\gamma \frac{h}{2}(h \times w)\right]$$

Thus the hydrostatic force quadruples.

Question:

11.011P

Consider a 200-ft-high, 1200-ft-wide dam filled capacity. Determine (*a*) the hydrostatic force on the dam and (*b*) the force per unit area of the dam near the top and near the bottom.

Step 1 of 4

Find the area of the dam wall by using the following formula:

$$A = wh$$

Here, w is the width of the dam and h is the height of the dam.

Substitute 1200 ft for w and 200 ft for h.

$$\begin{aligned} A &= (1200)(200) \\ &= 240000\ \text{ft}^2 \end{aligned}$$

Step 2 of 4

Question:

Consider a 200-ft-high, 1200-ft-wide dam filled capacity. Determine (*a*) the hydrostatic force on the dam and (*b*) the force per unit area of the dam near the top and near the bottom.

Step 1 of 4

Find the area of the dam wall by using the following formula:

$$A = wh$$

Here, w is the width of the dam and h is the height of the dam.

Substitute 1200 ft for w and 200 ft for h.

$$\begin{aligned} A &= (1200)(200) \\ &= 240000\ \text{ft}^2 \end{aligned}$$

Step 2 of 4

$F = F_{avg}$

Substitute $6240\ \text{lb/ft}^2$ for γ and $240000\ \text{ft}^2$ for A.

$$\begin{aligned} F &= 6240 \times 240000 \\ &= 1497600000\ \text{lbf} \\ &= 14.976 \times 10^8\ \text{lbf} \end{aligned}$$

Therefore, the hydrostatic force acting on dam is $\boxed{14.976 \times 10^8\ \text{lbf}}$.

Step 4 of 4

b)

The force per unit area at the top of the dam is $\boxed{\text{zero}}$.

Find the force per unit area at the bottom of the dam by using the following formula:

$$\overline{F} = \gamma h$$

Substitute $6240\ \text{lb/ft}^2$ for γ and 200 ft for h.

$$\begin{aligned} \overline{F} &= 6240 \times 200 \\ &= 1248000\ \text{lbf} \end{aligned}$$

Therefore, the force per unit area at the bottom of the dam is $\boxed{1248000\ \text{lbf}}$.

Question: 11.012P

A room in the lower level of a cruise ship has a 30-cm-diameter circular window. If the midpoint of the window is 4 m below the water surface, determine the hydrostatic force acting on the window, and the pressure center. Take the specific gravity of seawater to be 1.025.

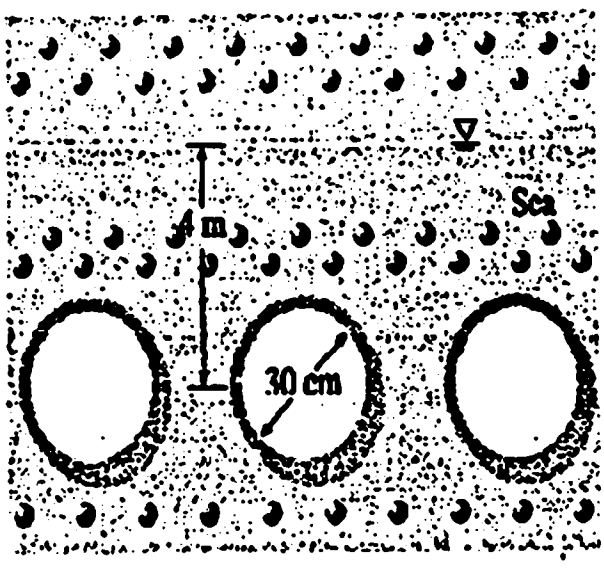


FIGURE P11–12

Step 1 of 4

Given data:

Diameter of the window, $d = 30\,\text{cm} = 0.3\,\text{m}$

The vertical distance between the free surface of water to the centre of window, $h_c = 4\,\text{m}$

Specific gravity of sea water, $SG = 1.025$

Density of sea water,

$$\rho = SG \times \rho_{water}$$
$$= 1.025 \times 1000$$
$$= 1025\,\text{kg/m}^3$$

Step 2 of 4

Area of the window,

$$A = \frac{\pi}{4} d^2$$
$$= \frac{\pi}{4}(0.3)^2$$
$$= 0.07069\,\text{m}^2$$

The average pressure acting on the circular plate,

$$P_{avg} = \rho g h_c$$
$$= 1025 \times 9.81 \times 4$$
$$= 40221\,\text{N/m}^2$$

Step 3 of 4

Hydrostatic force acting on window,

$$F = P_{avg} \times A$$
$$= 40221 \times 0.07069$$
$$= 2843.222\,\text{N}$$

Therefore hydrostatic force acting on window, $\boxed{F = 2843.222\,\text{N}}$.

Moment of inertia of a circular plate about x-axis,

$$I_{xx} = \frac{\pi}{64} d^4$$
$$= \frac{\pi}{64}(0.3)^4$$
$$= 3.976 \times 10^{-4}\,\text{m}^4$$

Step 4 of 4

The perpendicular distance between hydrostatic force and center of window, $y_c = h_c = 4\,\text{m}$

Centre of pressure,

$$y_p = y_c + \frac{I_{xx}}{A y_c}$$
$$= 4 + \frac{3.976 \times 10^{-4}}{0.07069 \times 4}$$
$$= 4 + 1.406 \times 10^{-3}$$
$$= 4.001\,\text{m}$$

Therefore the center of pressure, $\boxed{y_p = 4.001\,\text{m}}$.

Question: 11.013P

The water side of the wall of a 100-m-long dam is a quarter circle with a radius of 10 m. Determine the hydrostatic force on the dam and its line of action when the dam is filled to the rim.

Step 1 of 4

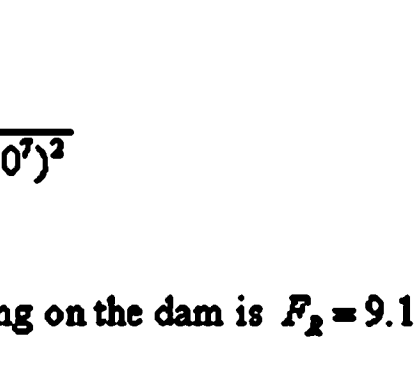
Given data:-

Width of the dam is $w = 100\ \text{m}$

Radius of the quarter circle is $R = 10\ \text{m}$

Step 2 of 4

Consider the free body diagram of the liquid block.



Step 3 of 4

Horizontal force on the vertical surface is

$$F_H = F_x = \rho g h_c A$$

$$= 1000 \times 9.81 \times \frac{10}{2} \times (10 \times 100)$$

$$F_H = 4.905 \times 10^7\ \text{N}$$

Vertical force component on curved surface is

$$F_V = F_y + W$$

Here F_y is zero since it coincides with the free surface of the water.

$$F_y = 0$$

Then weight of the fluid block per meter length is

$$F_V = W = \rho g V$$

$$= 1000 \times 9.81 \times \left(w \times \frac{\pi R^2}{4} \right)$$

$$= 1000 \times 9.81 \times \left(100 \times \frac{\pi \times 10^2}{4} \right)$$

$$F_V = 7.705 \times 10^7\ \text{N}$$

Step 4 of 4

The resultant force on the dam is

$$F_R = \sqrt{F_H^2 + F_V^2}$$

$$= \sqrt{(4.905 \times 10^7)^2 + (7.705 \times 10^7)^2}$$

$$\boxed{F_R = 9.13 \times 10^7\ \text{N}}$$

Therefore the resultant force acting on the dam is $F_R = 9.13 \times 10^7\ \text{N}$

Line of action of the resultant force is

$$\tan\theta = \frac{F_V}{F_H}$$

$$= \frac{7.705 \times 10^7}{4.905 \times 10^7}$$

$$= 1.571$$

$$\boxed{\theta = 57.5^0}$$

Therefore, the line of action of the resultant force is $\theta = 57.5^0$.

Question: 11.014P

A 6-m-high, 5-m-wide rectangular plate blocks the end of a 5-m-deep freshwater channel, as shown in Fig. P11–14. The plate is hinged about a horizontal axis along its upper edge through a point *A* and is restrained from opening by a fixed ridge at point *B*. Determine the force exerted on the plate by the ridge.

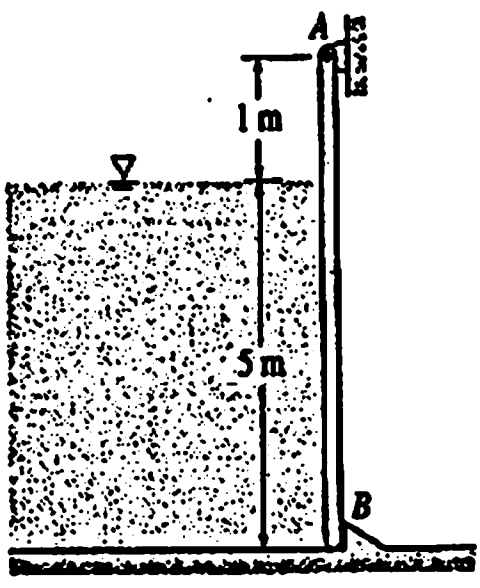


FIGURE P11–14

Step 1 of 6

Given data:

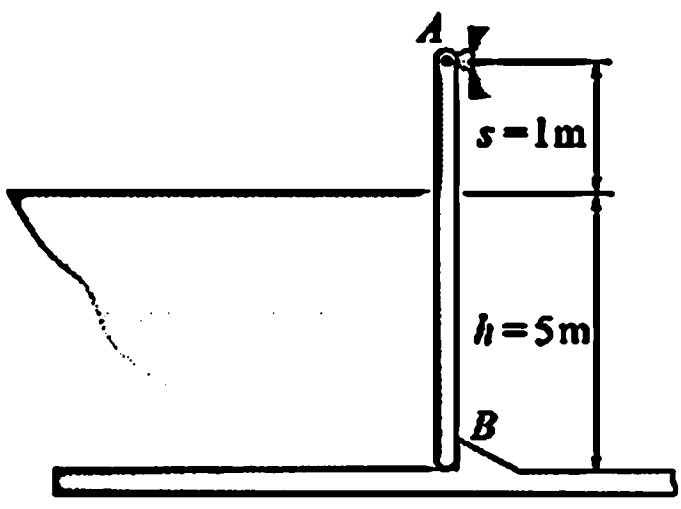
Height of rectangular plate, $l = 6\,\text{m}$

Width of rectangular plate, $b = 5\,\text{m}$

Height of fresh water, $h = 5\,\text{m}$

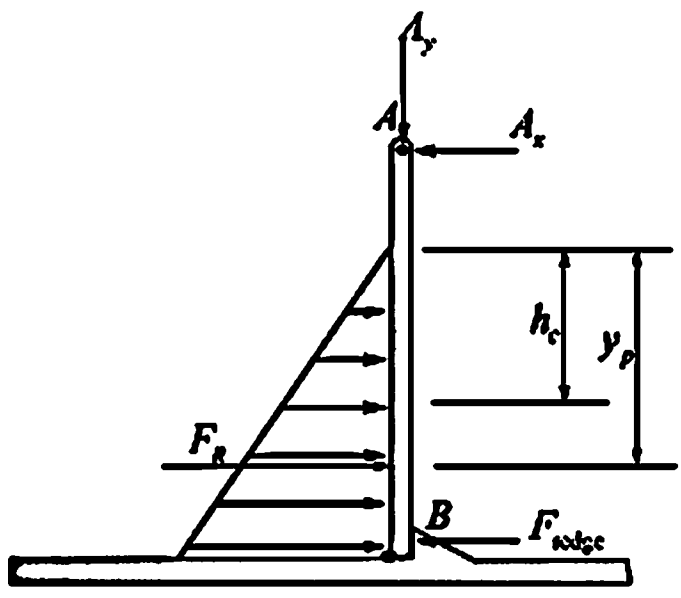
Step 2 of 6

Given figure:



Step 3 of 6

The free body diagram of given figure is as shown in figure-2:



Step 4 of 6

Density of water, $\rho = 1000\,\text{kg/m}^3$

Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$

Area of hydraulic pressure acts on rectangular plate,

$$A = h \times b$$
$$= 5 \times 5$$
$$= 25\,\text{m}^2$$

The centroid of area of hydraulic pressure from the surface of water,

$$h_c = \frac{h}{2}$$
$$= \frac{5}{2}\,\text{m}$$

Step 5 of 6

The average pressure acting on the rectangular plate,

$$P_{avg} = \rho g h_c$$
$$= 1000 \times 9.81 \times \frac{5}{2}$$
$$= 24525\,\text{N/m}^2$$

The hydrostatic force due to water on the plate,

$$F_R = P_{avg} \times A$$
$$= 24525 \times 25$$
$$= 613125\,\text{N}$$
$$= 613.125\,\text{kN}$$

Step 6 of 6

The line of action of the hydrostatic force on rectangular plate,

$$y_p = \frac{2h}{3}$$
$$= \frac{2(5)}{3}$$
$$= 3.33\,\text{m}$$

By taking moment about A, we get $\sum M_A = 0$

$$F_R(s + y_p) - F_{ridge}(\overline{AB}) = 0$$
$$(613.125)(1 + 3.33) - F_{ridge}(6) = 0$$
$$F_{ridge} = 442.47\,\text{kN}$$

Therefore the reaction force at point B, $\boxed{F_{ridge} = 442.47\,\text{kN}}$.

Question: 11.015P

Reconsider Prob. 11–14. Using EES (or other) software, investigate the effect of water depth on the force exerted on the plate by the ridge. Let the water depth vary from 0 to 5 m in increments of 0.5 m. Tabulate and plot your results.

Step 1 of 7

Given data:

Height of rectangular plate above the water surface, $s = 1\text{m}$

Width of rectangular plate, $w = 5\text{m}$

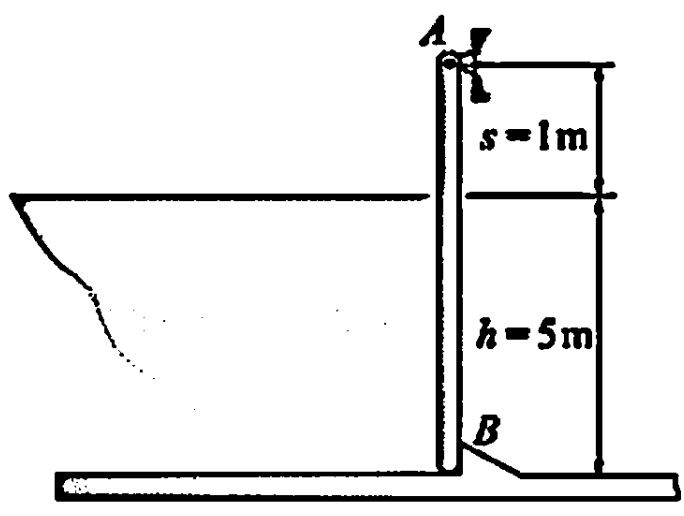
Height of fresh water, $h = 5\text{m}$

Using EES software, investigate the effect of water depth on the force exerted on the plate by the ridge.

The water depth is varying from $0\text{to}5\text{m}$ in increments of 0.5m.

Step 2 of 7

Given figure:



Step 3 of 7

The procedure for developing solution by using EES software:
Open the new equations window EES software and type all the information which is used to solve the problem
For developing table:
1) Click on table followed by new parametric table and add the variables, which are varying variable and required answer variable, then one table will form with that variables
2) Click on the variable which is varying and enter initial value and final value.
3) For getting the required answer variable values corresponding to the varying variable values click on the green color button and then the table will form with required answers
For developing graph:
1) Click on plot followed by new plot window followed by x-y plot.
2) Select the x-axis variable and y-axis variable and then press ok, then the graph will form with the given values.

Step 4 of 7

The equations which are using in EES software for developing the solution are given in equations window is as shown below

Equations Window

```
g=9.81 "m/s^2"
rho=1000 "kg/m^3"
s=1"m"
w=5 "m"
A=w*h
P_ave=rho*g*h/2000 "kPa"
F_R=P_ave*A "kN"
y_p=2*h/3
F_ridge=(s+y_p)*F_R/(s+h)
```

Step 5 of 7

The EES software will convert the equations in equations window to equations in formatting window is as shown below

Formatted Equations

$$g = 9.81 \quad \text{m/s2}$$

$$\rho = 1000 \quad \text{kg/m3}$$

$$s = 1 \quad \text{m}$$

$$w = 5 \quad \text{m}$$

$$A = w \cdot h$$

$$P_{ave} = \rho \cdot g \cdot \frac{h}{2000} \quad \text{kPa}$$

$$F_R = P_{ave} \cdot A \quad \text{kN}$$

$$y_p = 2 \cdot \frac{h}{3}$$

$$F_{ridge} = (s + y_p) \cdot \left[\frac{F_R}{s + h}\right]$$

Step 6 of 7

The EES software is developing the table for F_{ridge} corresponding to the h values is as shown below,

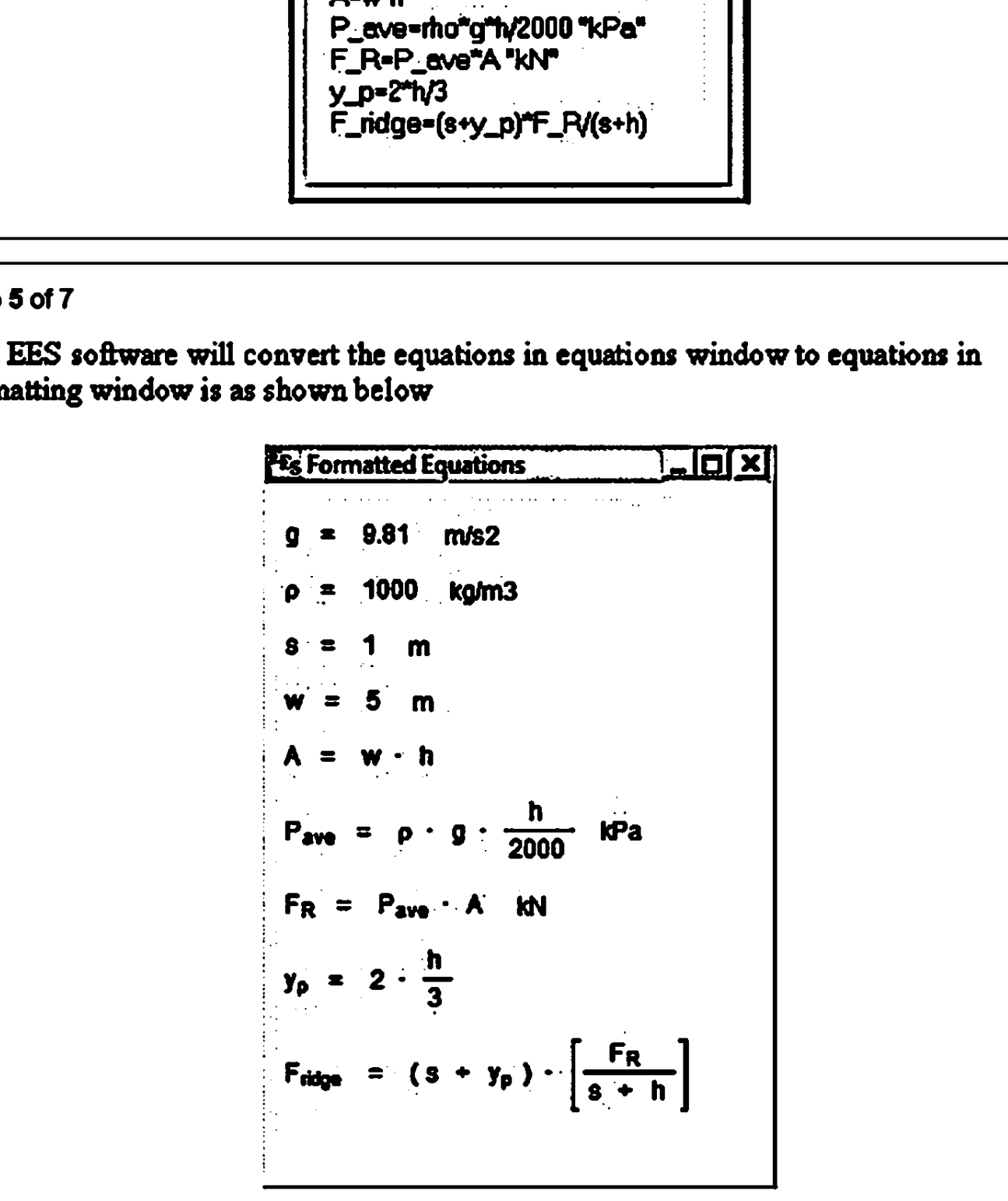
Parametric Table

Table 1

1..11	F_{ridge}	h
Run 1	0	0
Run 2	5.45	0.5
Run 3	20.44	1
Run 4	44.15	1.5
Run 5	76.3	2
Run 6	116.8	2.5
Run 7	165.5	3
Run 8	222.5	3.5
Run 9	287.8	4
Run 10	361.2	4.5
Run 11	442.8	5

Step 7 of 7

The EES software is developing the graph between F_{ridge} and h is as shown below,



Question: 11.016P

The flow of water from a reservoir is controlled by a 5-ft-wide L-shaped gate hinged at point *A*, as shown in Fig. P11–16E. If it is desired that the gate open when the water height is 12 ft, determine the mass of the required weight *W*.

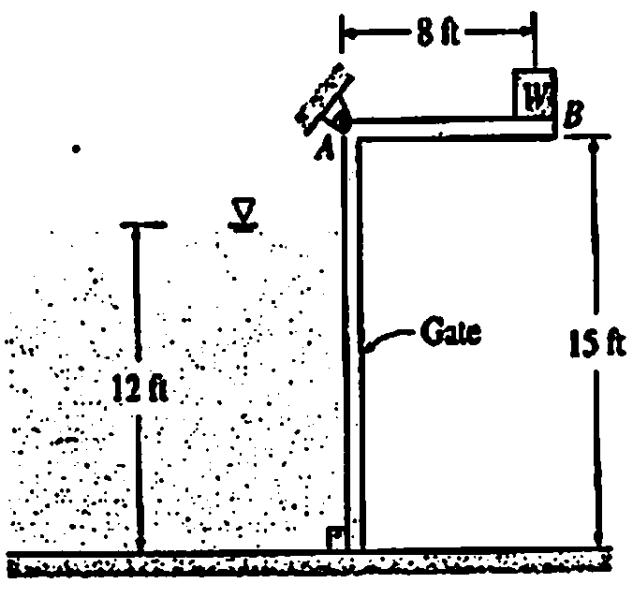


FIGURE P11–16E

Step 1 of 6

Given data:

Width of the gate, $w = 5\text{ft}$

Height of water, $h = 12\text{ft}$

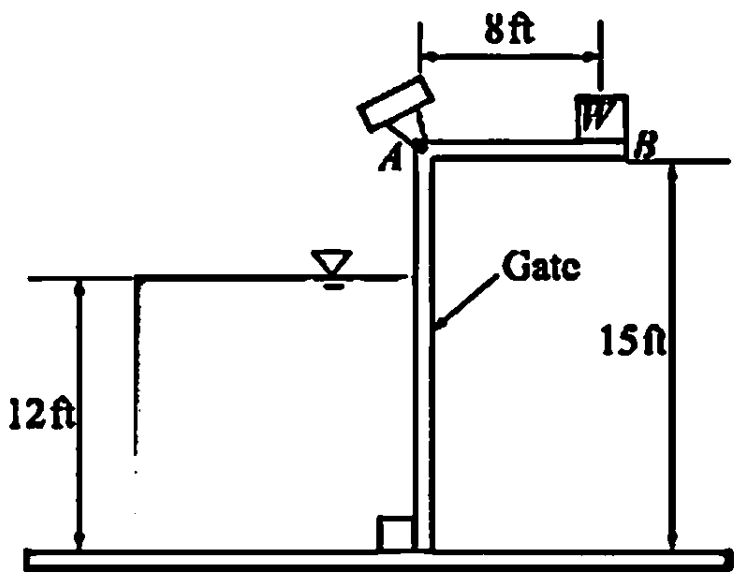
Height of gate above the water surface, $s = 3\text{ft}$

Density of water, $\rho = 62.4\text{lb/ft}^3$

Acceleration due to gravity, $g = 32.174\text{ft/s}^2$

Step 2 of 6

Given figure:



Step 3 of 6

Area of gate up to water level,

$$A = wh$$
$$= 5\times12$$
$$= 60\text{ft}^2$$

The vertical distance from the top of water to centroid of dam wall,

$$h_c = \frac{h}{2}$$
$$= \frac{12}{2}$$
$$= 6\text{ft}$$

Step 4 of 6

The average pressure acting on the gate,

$$P_{avg} = \rho g h_c$$
$$= 62.4\times32.174\times6$$
$$= 12045.945\text{lbf/ft}^2$$

Hydrostatic force acting on the gate,

$$F = P_{avg}\times A$$
$$= 12045.945\times60$$
$$= 722756.7\text{lbf}$$

Step 5 of 6

Moment of inertia about x-axis at the centroid of gate,

$$I_{xc} = \frac{wh^3}{12}$$
$$= \frac{(5)(12)^3}{12}$$
$$= 720\text{ft}^4$$

Centre of pressure acting on the gate,

$$h_p = h_c + \frac{I_{xc}}{Ah_c}$$
$$= 6 + \frac{720}{60\times6}$$
$$= 8\text{ft}$$

Step 6 of 6

Now taking the moment about A, we get

$$\sum M_A = 0$$
$$W\times8 = 722756.7\times(3+8)$$
$$W = 993790.462\text{lbf}$$

The mass of applied weight,

$$m = \frac{W}{g}$$
$$= \frac{993790.462}{32.174}$$
$$= 30887.998\text{lbm}$$

The mass of applied weight, $\boxed{m = 30887.998\text{lbm}}$.

Question: 11.017P

Repeat Prob. 11–16E for a water height of 8 ft.

Step 1 of 6

Given data:

Width of the gate, $w = 5\text{ft}$

Height of water, $h = 8\text{ft}$

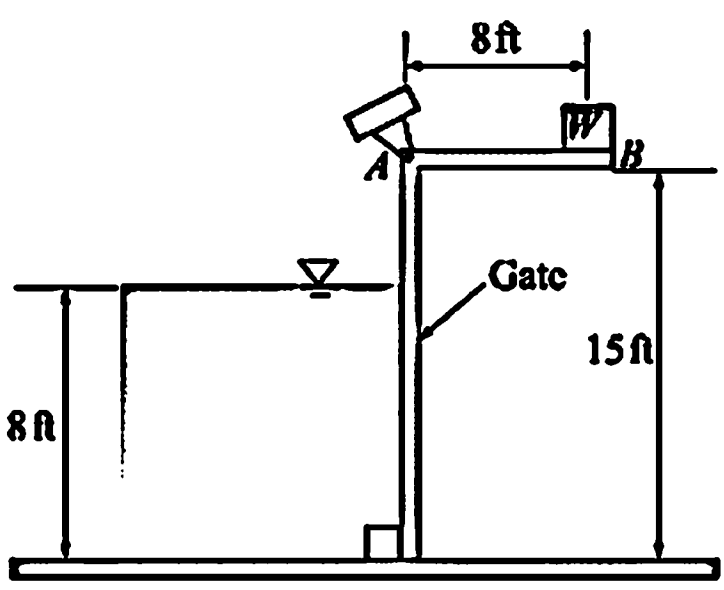
Height of gate above the water surface, $s = 7\text{ft}$

Density of water, $\rho = 62.4\text{lb/ft}^3$

Acceleration due to gravity, $g = 32.174\text{ft/s}^2$

Step 2 of 6

Given figure:



Step 3 of 6

Area of gate up to water level,

$$A = wh$$
$$= 5\times 8$$
$$= 40\text{ft}^2$$

The vertical distance from the top of water to centroid of dam wall,

$$h_c = \frac{h}{2}$$
$$= \frac{8}{2}$$
$$= 4\text{ft}$$

Step 4 of 6

The average pressure acting on the gate,

$$P_{avg} = \rho g h_c$$
$$= 62.4\times 32.174\times 4$$
$$= 8030.6304\text{lbf/ft}^2$$

Hydrostatic force acting on the gate,

$$F = P_{avg}\times A$$
$$= 8030.6304\times 40$$
$$= 321225.216\text{lbf}$$

Step 5 of 6

Moment of inertia about x-axis at the centroid of gate,

$$I_{xc} = \frac{wh^3}{12}$$
$$= \frac{(5)(8)^3}{12}$$
$$= 213.333\text{ft}^4$$

Centre of pressure acting on the gate,

$$h_p = h_c + \frac{I_{xc}}{Ah_c}$$
$$= 4 + \frac{213.333}{40\times 4}$$
$$= 5.333\text{ft}$$

Step 6 of 6

Now taking the moment about A, we get

$$\sum M_A = 0$$
$$W\times 8 = 321225.216\times(3+5.333)$$
$$W = 334596.215\text{lbf}$$

The mass of applied weight,

$$m = \frac{W}{g}$$
$$= \frac{334596.215}{32.174}$$
$$= 10399.5831\text{lbm}$$

The mass of applied weight, $\boxed{m = 10399.5831\text{lbm}}$.

Question: 11.018P

A water trough of semicircular cross section of radius 0.7 m consists of two symmetric parts hinged to each other at the bottom, as shown in Fig. P11–18. The two parts are held together by a cable and turnbuckle placed every 3 m along the length of the trough. Calculate the tension in each cable when the trough is filled to the rim.

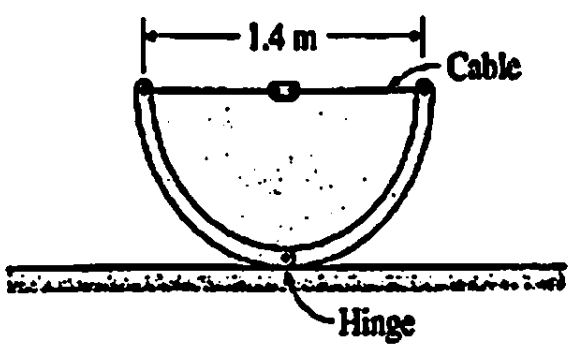
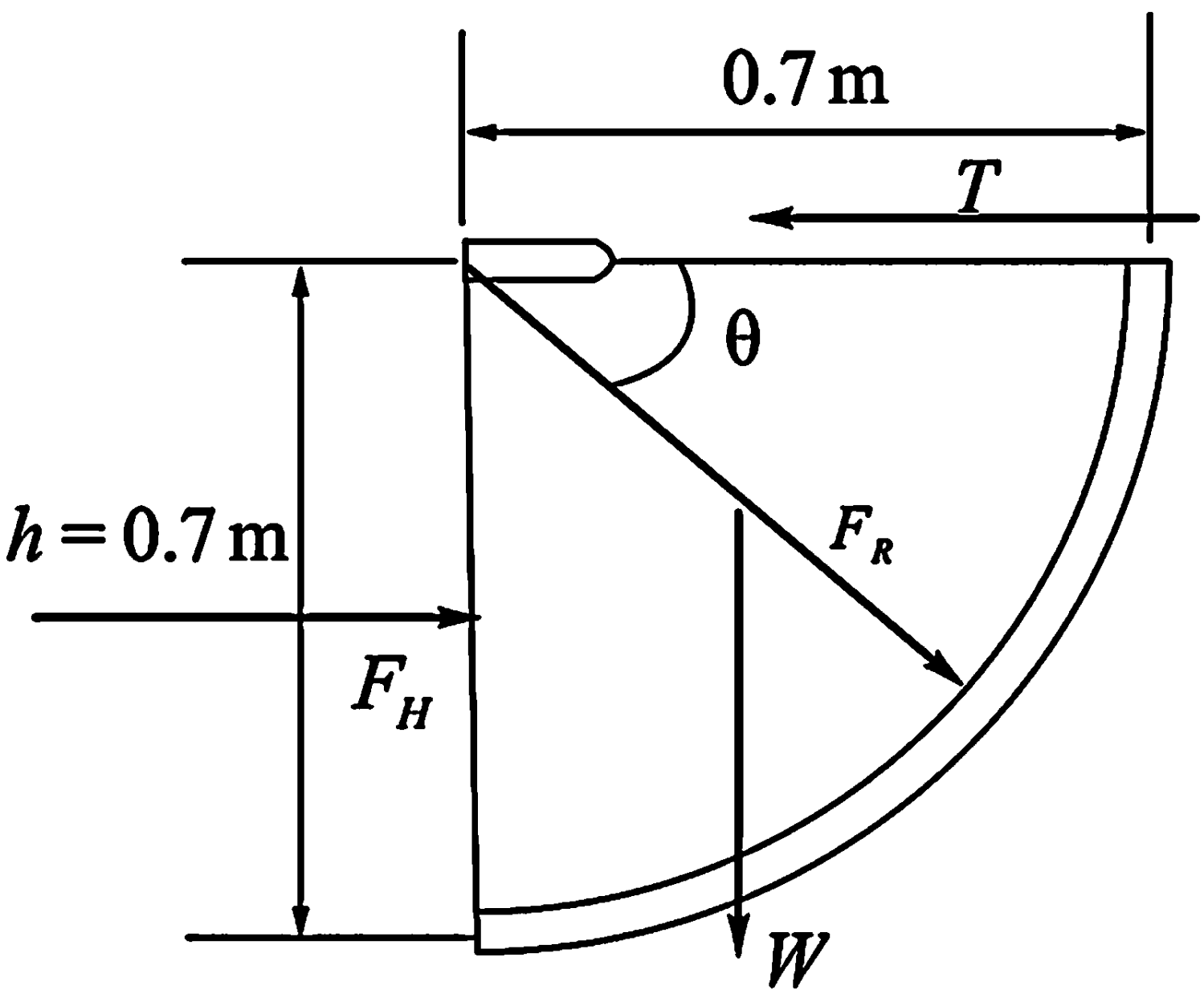


FIGURE P11–18

Step 1 of 4

Consider half of the trough which is in quarter circular shape is as shown in figure:



Step 2 of 4

Find the distance of centroid of rectangle from the surface of water.

$$h_c = \frac{h}{2}$$

Here, h is the height.

Substitute 0.7 m for h.

$$h_c = \frac{0.7}{2}$$
$$= 0.35\,\text{m}$$

Find the cross-sectional area.

$$A_R = rl$$

Here, r is the radius and l is the length.

Substitute 0.7 m for r and 3 m for l.

$$A_R = (0.7)(3)$$
$$= 2.1\,\text{m}^2$$

Find the horizontal hydrostatic force.

$$F_H = \rho g h_c A_R$$

Here, ρ is the density of water and g is the acceleration due to gravity.

Substitute $1000\,\text{kg/m}^3$ for ρ, $9.81\,\text{m/s}^2$ for g, 0.35 m for h_c, and $2.1\,\text{m}^2$ for A_R.

$$F_H = (1000)(9.81)(0.35)(2.1)$$
$$= 7210.35\,\text{N}$$

Step 3 of 4

Find the area of the quarter circle block.

$$A_C = \frac{\pi}{4} r^2$$

Substitute 0.7 m for r.

$$A_C = \frac{\pi}{4}(0.7)^2$$
$$= 0.384845\,\text{m}^2$$

Find the volume of quarter circle block.

$$V = lA_C$$

Substitute 3 m for l and $0.384845\,\text{m}^2$ for A_C.

$$V = (3)(0.384845)$$
$$= 1.1545\,\text{m}^3$$

Find the weight of fluid in the $3\,\text{m}$ block.

$$W = \rho g V$$

Substitute $1000\,\text{kg/m}^3$ for ρ, $9.81\,\text{m/s}^2$ for g and $1.1545\,\text{m}^3$ for V.

$$W = 1000 \times 9.81 \times 1.1545$$
$$= 11325.645\,\text{N}$$

The vertical force is the weight of the fluid in the first quarter circular $3\,\text{m}$ block.

$$F_V = W$$
$$= 11325.645\,\text{N}$$

Step 4 of 4

Find the resultant force acting on the surface of quarter circle.

$$F_R = \sqrt{F_H^2 + F_V^2}$$

Substitute $7210.35\,\text{N}$ for F_H and $11325.645\,\text{N}$ for F_V.

$$F_R = \sqrt{7210.35^2 + 11325.645^2}$$
$$= 13426.07\ \text{N}$$

Find the direction of action of the resultant force.

$$\theta = \tan^{-1}\frac{F_V}{F_H}$$

Substitute $7210.35\,\text{N}$ for F_H and $11325.645\,\text{N}$ for F_V.

$$\theta = \tan^{-1}\left(\frac{11325.645}{7210.35}\right)$$
$$= 57.52°$$

Find the tension in the cable by taking the moments about the point A.

$$\sum M_A = 0$$
$$F_R r \sin(90-\theta)° = Tr$$
$$F_R \sin(90-\theta)° = T$$

Substitute $13426.07\,\text{N}$ for F_R and $57.52°$ for θ.

$$13426.07 \sin(90-57.52)° = T$$
$$T = 7209.87\,\text{N}$$

Therefore, the tension in the cable is $\boxed{7209.87\,\text{N}}$.

Question: 11.019P

The two sides of a V-shaped water trough are hinged to each other at the bottom where they meet, as shown in Fig. P11–19, making an angle of 45° with the ground from both sides. Each side is 0.75 m wide, and the two parts are held together by a cable and turnbuckle placed every 6 m along the length of the trough. Calculate the tension in each cable when the trough is filled to the rim.

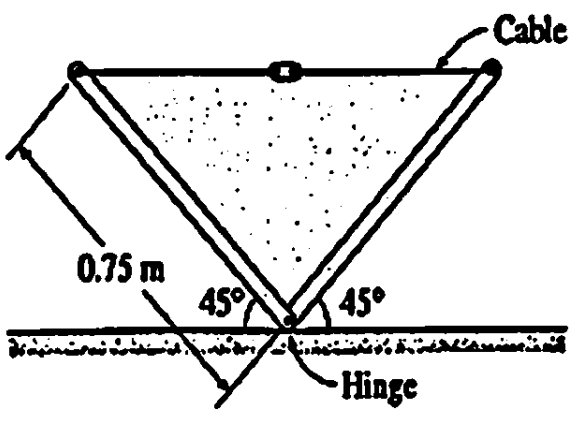
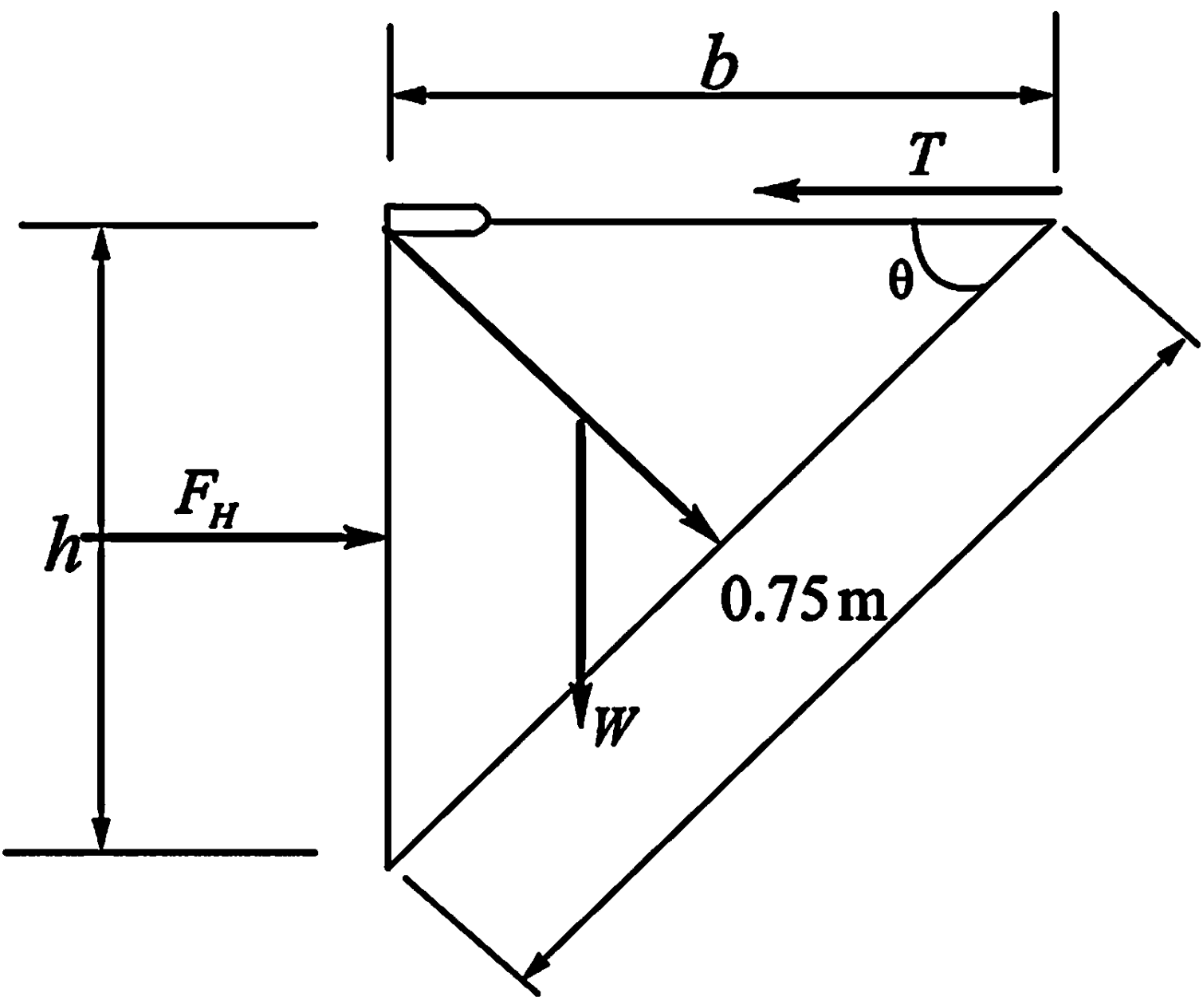


FIGURE P11–19

Step 1 of 8

Consider half of the trough which is in triangular shape is as shown in figure:



Step 2 of 8

Calculate the height of the water with the help of the above figure.

$$h = (0.75)\sin 45°$$
$$= 0.530 \text{ m}$$

Calculate the breadth of the water with the help of the above figure.

$$b = (0.75)\cos 45°$$
$$= 0.530 \text{ m}$$

Calculate the distance of centroid of projected area of water trough; this is a rectangle, from the surface of water.

$$h_c = \frac{h}{2}$$
$$= \frac{0.53}{2}$$
$$= 0.265 \text{ m}$$

Step 3 of 8

Calculate the horizontal hydrostatic force acting on the projected area of the trough.

$$F_H = P_{avg} A_R$$

Here, the average pressure acting the projected area is P_{avg} and the projected area is A_R .

Step 4 of 8

On further continuing, $F_H = (\rho g h_c)(h \times l)$ Here, the density of water is ρ , the acceleration due to gravity is g and the length of first block of trough is l .

Step 5 of 8

Substitute 1000 kg/m^3 for ρ , 9.81 m/s^2 for g and 6 m for l and all the corresponding values in the above equation.

Step 6 of 8

$$F_H = (1000)(9.81)(0.265)(0.530 \times 6)$$
$$= 8266.88 \text{ N}$$

Step 7 of 8

Calculate the vertical force considering the fact that the vertical force is the weight of liquid column itself.

$$F_V = W$$

Here, the weight of the fluid body is W .

Now calculate the weight of fluid body.

$$W = \rho g V$$
$$= (1000)(9.81)\left(6 \times \frac{bh}{2}\right)$$
$$= 9810 \times 6 \times \frac{0.530 \times 0.530}{2}$$
$$= 8266.88 \text{ N}$$

So, the vertical force,

$$F_V = 8266.88 \text{ N}$$

Step 8 of 8

Now, calculate the magnitude of the resultant force.

$$F_R = \sqrt{F_H^2 + F_V^2}$$
$$= \sqrt{8266.8^2 + 8266.8^2}$$
$$= 11691.02 \text{ N}$$

Also, calculate the direction of action of the resultant force,

$$\theta = \tan^{-1}\left(\frac{F_V}{F_H}\right)$$
$$= \tan^{-1}\left(\frac{8266.8}{8266.8}\right)$$
$$= 45°$$

Now take the moments about the point A and equating it to zero we

$$\sum M_A = 0$$
$$T(h) = F_R \cos 45° \times \left(\frac{2h}{3}\right)$$
$$T = \frac{2}{3}(11691.02 \times \cos 45°)$$
$$T = 5511.19 \text{ N}$$

Therefore the tension in each cable, T , is $\boxed{5511.19 \text{ N}}$.

Question: 11.020P

Repeat Prob. 11–19 for the case of a partially filled trough with a water height of 0.4 m directly above the hinge.

Step 1 of 8

Given data:

The trough is partially filled.

The height of the water body from the hinge point is $h = 0.4$ m

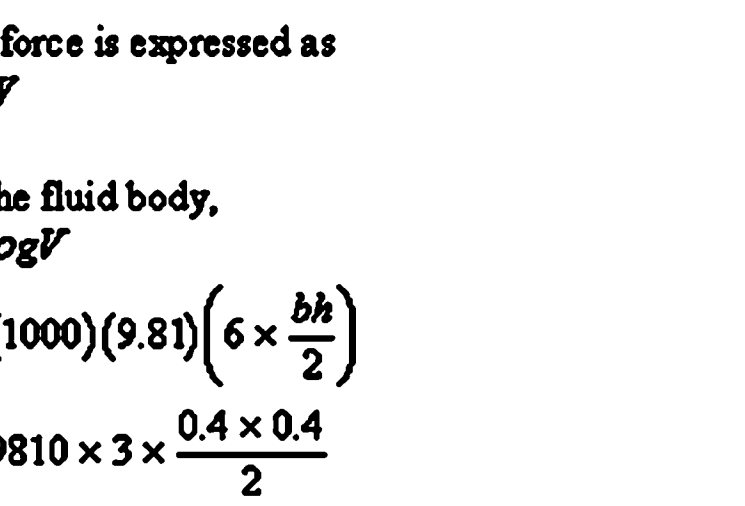
The cable and turnbuckle are placed at 6 m intervals along the trough.

The density of water is $\rho = 1000\ \text{kg/m}^3$

Step 2 of 8

Let us assume that the atmospheric pressure acts on both sides of the trough and gets cancelled. The weight of the trough is negligible.

Let us draw the figure for a section of the triangular trough.



Step 3 of 8

From the figure,

The height of the water body,

$h = 0.4$ m

The breadth of the water body,

$b = 0.4$ m

Step 4 of 8

The average pressure on the vertical surface is the pressure value at the centroid of the surface.

$$P_{avg} = P_C$$
$$= \rho g h_C$$

Consider the horizontal force acting on the vertical surface,

$$F_H = P_{avg} A$$
$$= \rho g h_C A$$
$$= (1000)(9.81)\left(\frac{0.4}{2}\right)(0.4 \times 3)$$
$$= 2354.4\ \text{N}$$

Step 5 of 8

The force acting on the horizontal coincides with the free surface of water hence can be neglected.

Therefore the vertical force is expressed as

$$F_V = W$$

Where the weight of the fluid body,

$$W = \rho g V$$
$$= (1000)(9.81)\left(6 \times \frac{bh}{2}\right)$$
$$= 9810 \times 3 \times \frac{0.4 \times 0.4}{2}$$
$$= 2354.4\ \text{N}$$

Therefore the vertical force is $F_V = 2354.4$ N

Step 6 of 8

The magnitude of the resultant force,

$$F_R = \sqrt{F_H^2 + F_V^2}$$
$$= \sqrt{2354.4^2 + 2354.4^2}$$
$$= 3329.6\ \text{N}$$

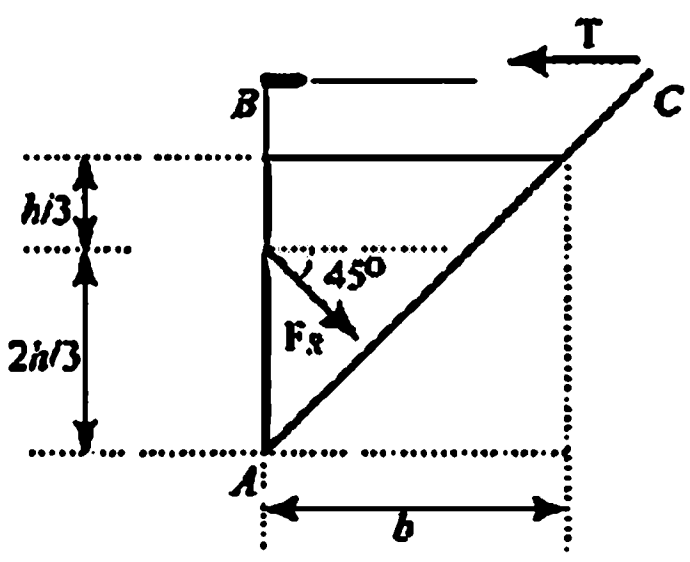
Step 7 of 8

The direction of action of the force,

$$\theta = \tan^{-1}\left(\frac{F_V}{F_H}\right)$$
$$= \tan^{-1}\left(\frac{2354.4}{2354.4}\right)$$
$$= 45°$$

Step 8 of 8

Let us draw the figure



Question: 11.021P

A retaining wall against a mud slide is to be constructed by placing 1.2-m-high and 0.25-m-wide rectangular concrete blocks (ρ = 2700 kg/m3) side by side, as shown in Fig. P11–21. The friction coefficient between the ground and the concrete blocks is f = 0.3, and the density of the mud is about 1800 kg/m3. There is concern that the concrete blocks may slide or tip over the lower left edge as the mud level rises. Determine the mud height at which (*a*) the blocks will overcome friction and start sliding and (*b*) the blocks will tip over.

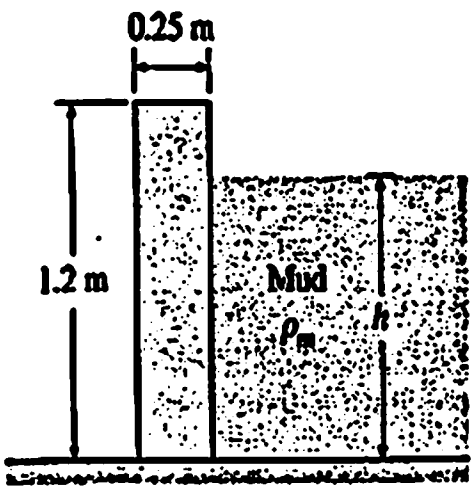


FIGURE P11–21

Step 1 of 6

Given data:

The height of rectangular concrete block, $H = 1.2\,\text{m}$

The width of rectangular concrete block, $w = 0.25\text{m}$

The coefficient of friction between ground and concrete blocks, $f = 0.3$

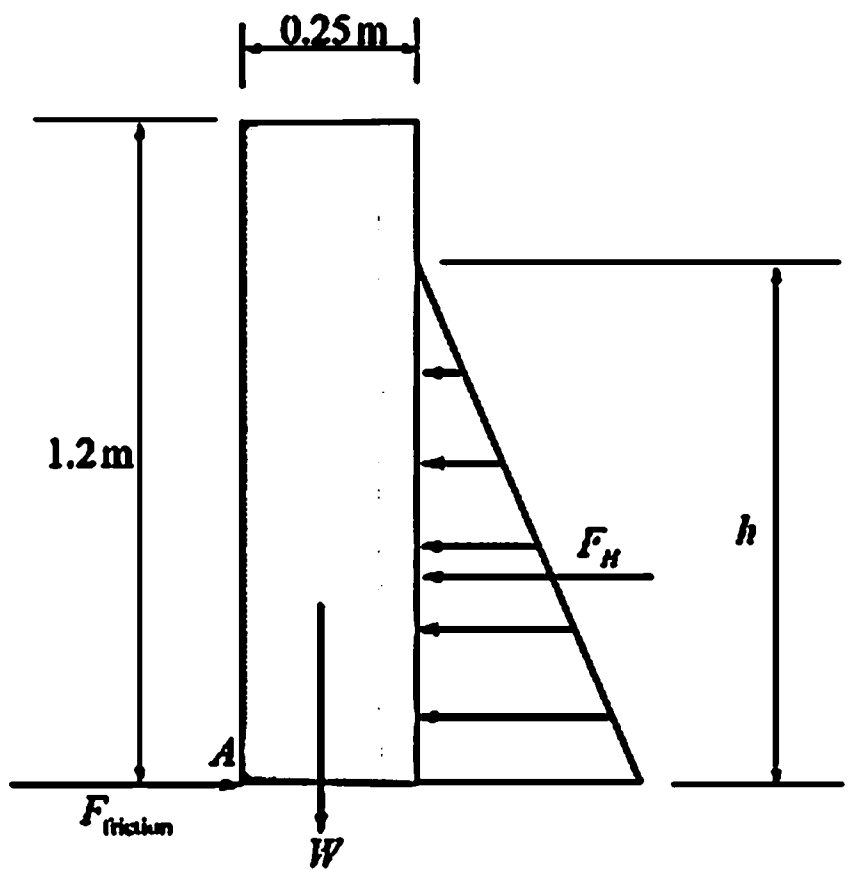
Density of mud is 1800 kg/m^3.

Density of concrete is 2700 kg/m^3.

The height of mud is h

Step 2 of 6

The free body diagram of the given figure is as shown below,



Step 3 of 6

The weight of the concrete wall per unit length

$W = \rho g V$

$= (2700)(9.81)(w \times H \times 1)$

$= (2700)(9.81)(0.25 \times 1.2 \times 1)$

$= 7946.1\text{N}$

Frictional force between the wall and ground

$F_{friction} = fW$

$= (0.3)(7946.1)$

$= 2383.83\text{N}$

Step 4 of 6

The shape of mud at the contact of mud and wall is rectangular shape,

Now the centroid of that rectangle from the surface of mud, $h_c = \dfrac{h}{2}$

Hydrostatic force exerted by mud on the wall

$F_H = (\rho g h_c) A$

$= [(1800\text{kg/m}^3)(9.81\ \text{m/s}^2)(h/2)](h \times 1)$

$= 8829h^2$

Step 5 of 6

a)

For the height of mud to start sliding of block:

When the horizontal hydrostatic force is equal to frictional force then the block will start sliding, therefore,

$F_H = F_{friction}$

$8829h^2 = 2384$

$h = 0.5196\,\text{m}$

Therefore the height of mud to start sliding of block, $\boxed{h = 0.5196\,\text{m}}$

Step 6 of 6

b)

For the height of mud to tip over the block:

The line of action of F_H is $2h/3$ from the mud surface.

Taking moments about point A and equate it to zero.

$\sum M_A = 0$

$W(0.125) = F_H(h/3)$

$(7946.1)(0.125) = (8829h^2)(h/3)$

$h = 0.6962\,\text{m}$

Therefore, the height of mud to tip over the block, $\boxed{h = 0.6962\,\text{m}}$

Question: 11.022P

Repeat Prob. 11–21 for 0.4-m-wide concrete blocks.

Step 1 of 8

Given Data:

Density of mud, $\rho_m = 1800\ \text{kg/m}^3$.

Density of concrete, $\rho_c = 2700\ \text{kg/m}^3$.

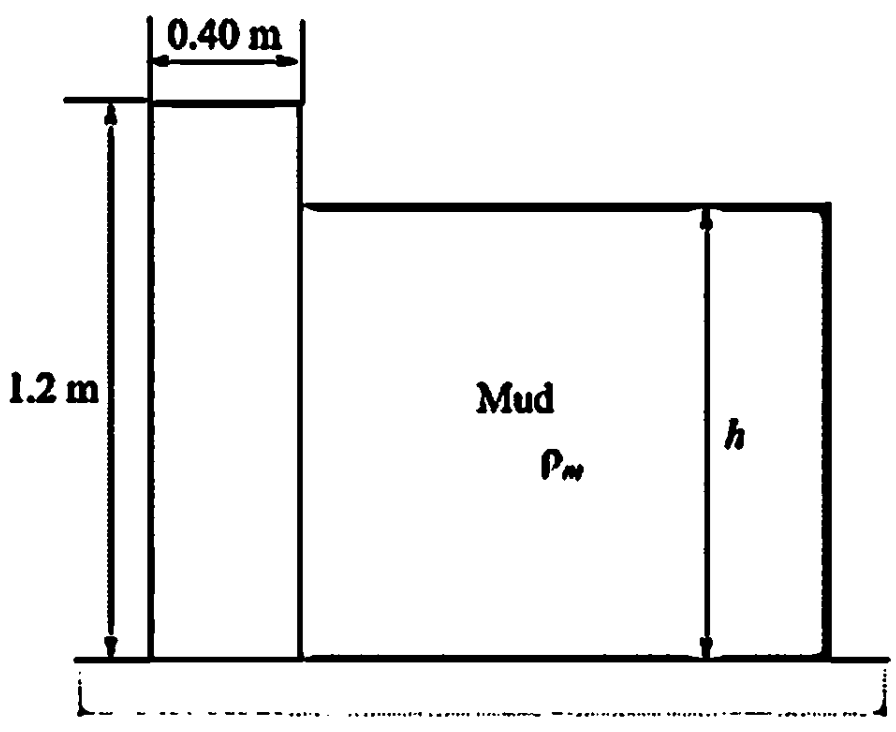
Width of the concrete block, $w = 0.4\ \text{m}$.

Height of the concrete block, $h_c = 1.2\ \text{m}$.

Coefficient of friction between the ground and the concrete block, $f = 0.3$

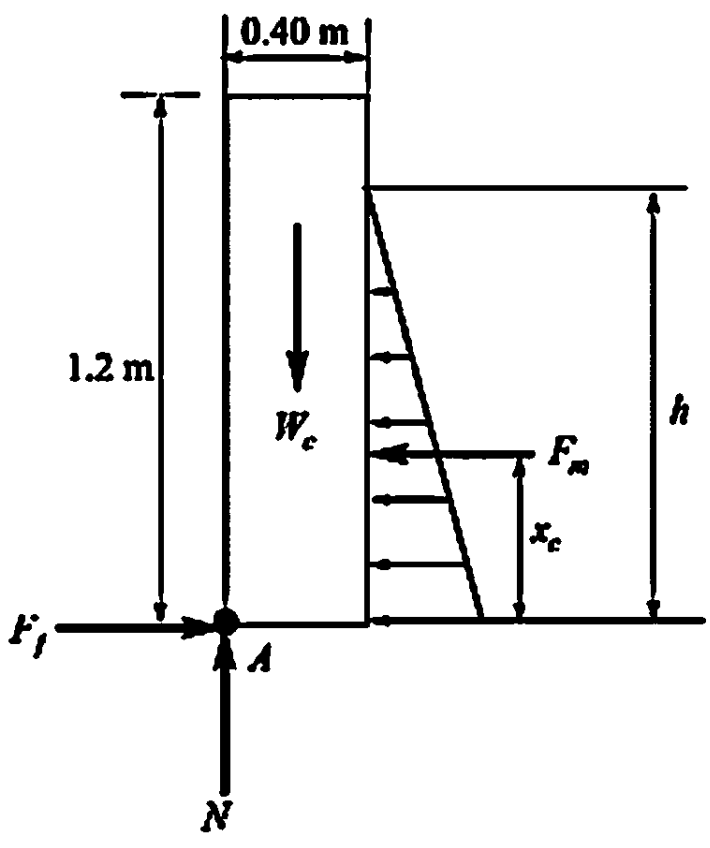
Step 2 of 8

Given figure is shown:



Step 3 of 8

Forces acting on the retaining wall are shown:



Step 4 of 8

Volume of the concrete block per unit length is given by,

$$V_c = l \times h_c \times w$$
$$= 1 \times 1.2 \times 0.4$$
$$= 0.48\ \text{m}^3$$

Weight of the concrete block per unit length is given by,

$$W_c = \rho_c g V_c$$
$$= 2700 \times 9.81 \times 0.48$$
$$= 12713.76\ \text{N}$$

Step 5 of 8

Normal reaction between the wall and the wall is given by,

$$N = W_c$$
$$= 12713.76\ \text{N}$$

Frictional force between the ground and the concrete wall is given by,

$$F_f = f \times N$$
$$= 0.3 \times 12713.76$$
$$= 3814.128\ \text{N}$$

Step 6 of 8

Pressure acting by the hydrostatic mud on the retaining wall is given by,

$$P_m = \frac{1}{2} \times \rho_m g h$$
$$= \frac{1}{2} \times 1800 \times 9.81 \times h$$
$$= 8829h$$

The hydrostatic force acting on the retaining wall due to the mud per unit length is given by,

$$F_m = P_m \times h \times 1$$
$$= 8829h \times h \times 1$$
$$= 8829h^2$$

The height of the hydrostatic force acting on the retaining wall due to the mud from the ground is given by,

$$x_m = h \times \frac{1}{3}$$
$$= \frac{h}{3}$$

Step 7 of 8

(a)

For the block to start sliding:

$$F_f = F_m$$

$$3814.128 = 8829h^2$$

$$h = 0.657\ \text{m}$$

Therefore, the blocks will over come friction and start sliding at height of mud,

$\boxed{h = 0.657\ \text{m}}$

Step 8 of 8

(b)

For the blocks to tip over:

$$\sum M_A = 0$$

$$-W_c \times \left(\frac{0.4}{2}\right) + F_m \times x_m = 0$$

$$-12713.76 \times 0.2 + 8829h^2 \times \frac{h}{3} = 0$$

$$h = 0.952\ \text{m}$$

Therefore, the blocks will tip over at height of mud, $\boxed{h = 0.952\ \text{m}}$

Question: 11.023P

A 4-m-long quarter-circular gate of radius 3 m and of negligible weight is hinged about its upper edge *A*, as shown in Fig. P11–23. The gate controls the flow of water over the ledge at *B*, where the gate is pressed by a spring. Determine the minimum spring force required to keep the gate closed when the water level rises to *A* at the upper edge of the gate.

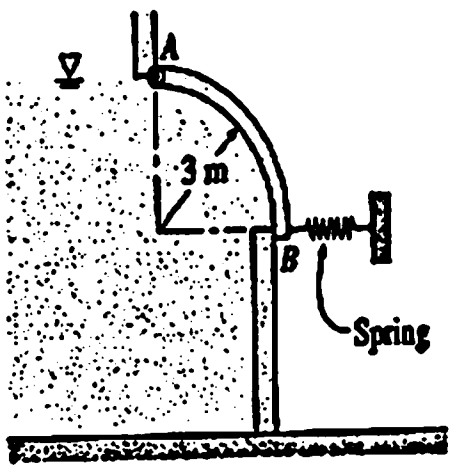


FIGURE P11–23

Step 1 of 5

Given data:

Length of the quarter circular plate=4 m

Radius of the quarter circular plate=3 m

Step 2 of 5

Horizontal force on the vertical surface

$$F_H = \rho g h_c A$$
$$= (1000\text{kg/m}^3)(9.81\text{m/s}^2)(3/2\text{m})(4\text{m}\times 3\text{m})$$
$$= 176.58\text{kN}$$

Vertical force on horizontal surface

$$F_y = \rho g h_{bottom} A$$
$$= (1000\text{kg/m}^3)(9.81\text{m/s}^2)(3\text{m})(4\text{m}\times 3\text{m})$$
$$= 353.2\text{kN}$$

Step 3 of 5

Weight of the fluid block per 4 m length

$$W = \rho g V$$
$$= (1000\text{kg})(9.81\text{m/s}^2)(w\times \pi R^2/4)$$
$$= (1000\text{kg})(9.81\text{m/s}^2)(4\times \pi(3)^2/4)$$
$$= 277.37\text{N}$$

The net upward force

$$F_V = F_y - W$$
$$= 353.2 - 277.37$$
$$= 75.83\text{kN}$$

Step 4 of 5

The resultant force

$$F_R = \sqrt{F_H^2 + F_V^2}$$
$$= \sqrt{(176.58)^2 + (75.83)^2}$$
$$= 192.17\text{kN}$$

$$\tan\theta = \frac{F_V}{F_H}$$
$$= \frac{75.83}{176.58}$$
$$= 0.429$$
$$\theta = 23.24^\circ$$

Step 5 of 5

The minimum spring force is determined by taking the moment about the hinge A.

$$\sum M_A = 0$$
$$F_R R\sin(90-\theta) - F_{spring} R = 0$$
$$F_R R\sin(90-\theta) = F_{spring} R$$
$$192.17\sin(90-23.24) = F_{spring}$$
$$F_{spring} = 176.57\text{kN}$$

Therefore the minimum spring force required is $\boxed{F_{spring} = 176.57\text{kN}}$.

Question: 11.024P

Repeat Prob. 11–23 for a radius of 4 m for the gate.

Step 1 of 4

Given data:-

Radius of the gate is $r = 4$ m

Step 2 of 4

Horizontal force on the vertical surface is

$$F_H = \rho g h_c A$$
$$= 1000 \times 9.81 \times \frac{4}{2} \times (4 \times 4)$$
$$= 314 \text{ kN}$$

Vertical force on horizontal surface is

$$F_V = \rho g h_{bottom} A$$
$$= 1000 \times 9.81 \times 4 \times (4 \times 4)$$
$$= 627.8 \text{ kN}$$

Step 3 of 4

Weight of the fluid block per 4 m length is

$$W = \rho g V$$
$$= 1000 \times 9.81 \times \left(w \times \frac{\pi r^2}{4} \right)$$
$$= 9810 \times \left(4 \times \frac{\pi \times 4^2}{4} \right)$$
$$= 493.104 \text{ kN}$$

The net upward force is

$$F_V = F_y - W$$
$$= 627.8 - 493.104$$
$$= 135 \text{ kN}$$

Step 4 of 4

The resultant force is

$$F_R = \sqrt{F_H^2 + F_V^2}$$
$$= \sqrt{(314)^2 + (135)^2}$$
$$= 341.8 \text{ kN}$$
$$\tan\theta = \frac{F_V}{F_H}$$
$$= \frac{135}{314}$$
$$= 0.429$$
$$\theta = 23.2^0$$

The minimum spring force is determined by taking the moment about the hinge A.

$$\sum M_A = 0$$
$$F_R R \sin(90 - \theta) - F_{spring} R = 0$$
$$F_{spring} = F_R \sin(90 - 23.2)$$
$$F_{spring} = F_R \sin(66.8)$$
$$\boxed{F_{spring} = 314 \text{ kN}}.$$

Therefore the minimum spring force required to keep the gate closed is $F_{spring} = 314$ kN.

Question: 11.025P

What is buoyant force? What causes it? What is the magnitude of the buoyant force acting on a submerged body whose volume is V? What are the direction and the line of action of the buoyant force?

Step 1 of 1

The force exerted by the fluid on an immersed body is known as buoyant force.
The main reason for buoyancy force is when the depth increases, the pressure of water increases. The magnitude of buoyant force on a submerged body whose volume is V is:

$$F_B = \rho_f g V$$

The direction of the buoyancy force is towards upwards and its line of action passes through the centroid of the displaced volume.

Question: 11.026P

Consider two identical spherical balls submerged in water at different depths. Will the buoyant forces acting on these two balls be the same or different? Explain.

Step 1 of 1

The buoyant force on the two spheres is same when submerged at different depths. The magnitude of buoyant force does not depend upon the depth of water as it is clear from the formula:

$F_B = \rho_f g V$

Where F_B is buoyant force

ρ_f is density of fluid

g is gravitational constant

V is volume

Question: 11.027P

Consider two 5-cm-diameter spherical ballsone made of aluminum, the other of iron—submerged in water. Will the buoyant forces acting on these two balls be the same or different? Explain.

Step 1 of 1

The buoyant force acting on the iron and aluminum balls are same. This is because buoyant force depends on the density of fluid, but not on the density of immersed body.

Buoyant force is given as

$$F_B = \rho_f g V$$

where F_B is buoyant force

ρ_f is density of fluid

g is gravitational constant

V is volume

Question: 11.028P

Consider a 3-kg copper cube and a 3-kg copper ball submerged in a liquid. Will the buoyant forces acting on these two bodies be the same or different? Explain.

Step 1 of 1

The buoyant force acting on copper cube and copper ball are same. Buoyant force depends on the volume, but not on the shape of body. Buoyant force is:

$$F_B = \rho_f g V$$

Where F_B is buoyant force

ρ_f is density of fluid

g is gravitational constant

V is volume

Question: 11.029P

Discuss the stability of (*a*) a submerged and (*b*) a floating body whose center of gravity is above the center of buoyancy.

Step 1 of 2

a) A submerged body whose centre of gravity G is above the center of buoyancy B is unstable.

Step 2 of 2

b) For a floating body, when G is above B, it is stable. The centroid of the displaced volume shifts to the side to a point B during a rotational disturbance while the centre of gravity G of the body remains unchanged. Hence the floating body will be stable when G is above B.

Question: 11.030P

The density of a liquid is to be determined by an old 1-cm-diameter cylindrical hydrometer whose division marks are completely wiped out. The hydrometer is first dropped in water, and the water level is marked. The hydrometer is then dropped into the other liquid, and it is observed that the mark for water has risen 0.6 cm above the liquid–air interface (Fig. P11–30). If the height of the original water mark is 13.6 cm, determine the density of the liquid.

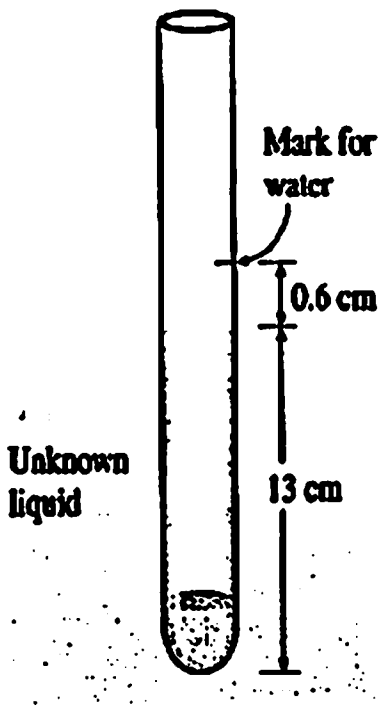


FIGURE P11–30

Step 1 of 4

Given Data:

Diameter of the cylindrical hydrometer, $d = 1$ cm

The height of the water mark when it is dropped in water, $h_{water} = 13.60$ cm

Difference between the height of the water mark and the liquid mark,

$h_{water} - h_{liquid} = 0.6$ cm

Step 2 of 4

Density of water, $\rho_{water} = 1000\ \text{kg/m}^3$

Let the cross-sectional area of the hydrometer be A.

Height of the water mark is given by,

$$h_{liquid} = h_{water} - 0.60$$
$$= 3.60 - 0.60$$
$$= 3.0 \text{ cm}$$

Volume of the hydrometer submerged in water is given by

$$V_{water} = A \times h_{water}$$
$$= A \times 13.6$$

Volume of the hydrometer submerged in liquid is given by

$$V_{liquid} = A \times h_{liquid}$$
$$= A \times 13.0$$

Step 3 of 4

Buoyancy force exerted by the water on the hydrometer is given by

$$F_{B_{water}} = \rho_{water} g V_{water}$$
$$= 1000 \times g \times A \times 13.6$$
$$= 13600 \times g \times A$$

Buoyancy force exerted by the liquid on the hydrometer is given by

$$F_{B_{liquid}} = \rho_{liquid} g V_{liquid}$$
$$= \rho_{liquid} \times g \times A \times 13.0$$
$$= 13.0 \times \rho_{liquid} \times g \times A$$

Step 4 of 4

The buoyancy force exerted by the liquid on the hydrometer must be equal to the buoyancy force exerted by the water on the hydrometer.

$$F_{B_{liquid}} = F_{B_{water}}$$

$$13.0 \times \rho_{liquid} \times g \times A = 13600 \times g \times A$$

$$\rho_{liquid} = \frac{13600}{13.0}$$
$$= 1046.15\ \text{kg/m}^3$$

Therefore, the density of the liquid, $\boxed{\rho_{liquid} = 1046.15\ \text{kg/m}^3}$

Question: 11.031P

A crane is used to lower weights into a lake for an underwater construction project. Determine the tension in rope of the crane due to a 3-ft-diameter spherical steel block(density = 494 lbm/ft3) when it is (*a*) suspended in the air and (*b*) completely immersed in water.

Step 1 of 5

Given Data:

Diameter of the spherical steel block, $d = 3\,\text{ft}$

Density of the spherical steel block, $\rho_s = 494\ \text{lbm/ft}^3$

Volume of the spherical steel block is given by,

$$V = \frac{4}{3}\pi\left(\frac{d}{2}\right)^3$$

$$= \frac{4}{3}\times\pi\times\left(\frac{3}{2}\right)^3$$

$$= 14.1372\,\text{ft}^3$$

Weight of the spherical steel block is given by,

$$W = \rho_s \times g \times V$$

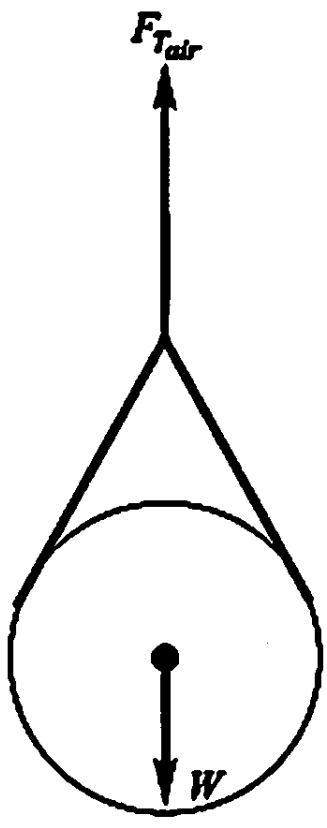
$$= 494\times 32.174\times 14.1372$$

$$= 224696.0348\ \text{lbm}\cdot\text{ft/sec}^2\left|\frac{1\,\text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/sec}^2}\right|$$

$$= 6983.78\,\text{lbf}$$

Step 2 of 5

Free body diagram of the spherical block when it is suspended in water is shown:



Step 3 of 5

(a)

When the spherical steel block is suspended in air:

Since the buoyancy force due to air is negligible. Therefore, the tension in the rope is equal to the weight of the spherical steel block.

The tension in the rope is given by,

$$F_{T_{air}} = W$$

$$= 6983.78\,\text{lbf}$$

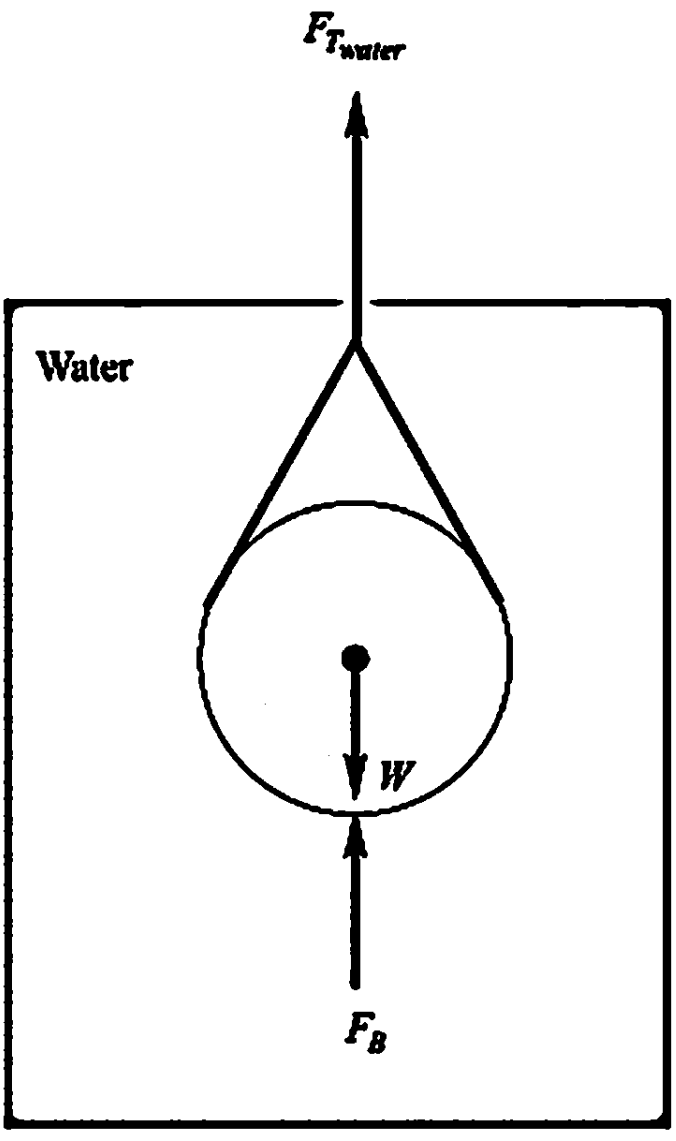
Therefore, the tension in the rope when the spherical block is suspended in air,

$$\boxed{F_{T_{air}} = 6983.78\,\text{lbf}}$$

Step 4 of 5

(b)

Free body diagram of the spherical block when it is suspended in water is shown:



Step 5 of 5

When the spherical steel block is suspended in water:

Buoyancy force acting on the spherical steel block is given by,

$$F_B = \rho_w \times g \times V$$

$$= 62.4\times 32.174\times 14.1372$$

$$= 28382.6570\ \text{lbm}\cdot\text{ft/sec}^2\left|\frac{1\,\text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/sec}^2}\right|$$

$$= 882.27\,\text{lbf}$$

From the free body diagram, the tension in the rope is given by,

$$F_{T_{water}} = W - F_B$$

$$= 6983.78 - 882.27$$

$$= 6101.511\text{bf}$$

Therefore, the tension in the rope when the spherical block is suspended in air,

$$\boxed{F_{T_{water}} = 6101.511\text{bf}}$$

Question: 11.032P

The volume and the average density of an irregularly shaped body are to be determined by using a spring scale. The body weighs 7200 N in air and 4790 N in water. Determine the volume and the density of the body. State your assumptions.

Step 1 of 3

Mass of the body

$$m = \frac{W_{air}}{g}$$

Here W_{air} is the Weight of the body in air; g is acceleration due to gravity.

Substitute $7200\,\text{N}$ for W_{air}, and $9.81\,\text{m/s}^2$ for g

$$m = \frac{7200\,\text{N}}{9.81\,\text{m/s}^2}$$
$$= 733.944\,\text{kg}$$

Step 2 of 3

Buoyancy force is the difference between the weights of the body in water and air.

$$F_B = W_{air} - W_{Water}$$

Here W_{air} is the Weight of the body in air; W_{Water} is the Weight of the body in water.

Substitute $7200\,\text{N}$ for W_{air}, $4790\,\text{N}$ and for W_{Water}.

$$F_B = 7200 - 4790$$
$$= 2410\,\text{N}$$

The Buoyancy force

$$F_B = \rho g V$$

Here ρ is the Density of water; g is acceleration due to gravity, and V is the volume the body.

Substitute $1000\,\text{kg}$ for ρ, $9.81\,\text{m/s}^2$ for g, and 2410 for F_B

$$2410\,\text{N} = (1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)V$$
$$V = 0.2456\,\text{m}^3$$

The volume of a body $\boxed{0.2456\,\text{m}^3}$

Step 3 of 3

Now density of the body

$$\rho_{body} = \frac{m}{V}$$

Here m is the mass of the body, and V is the volume the body.

Substitute $734\,\text{kg}$ for m, and 0.2456 for V

$$\rho_{body} = \frac{734\,\text{kg}}{0.2456\,\text{m}^3}$$
$$= 2988.5\,\text{kg/m}^3$$

Density of the body $\boxed{2988.5\,\text{kg/m}^3}$

Question: 11.033P

Consider a large cubic ice block floating in seawater. The specific gravities of ice and seawater are 0.92 and 1.025, respectively. If a 15-cm-high portion of the ice block extends above the surface of the water, determine the height of the ice block below the surface.

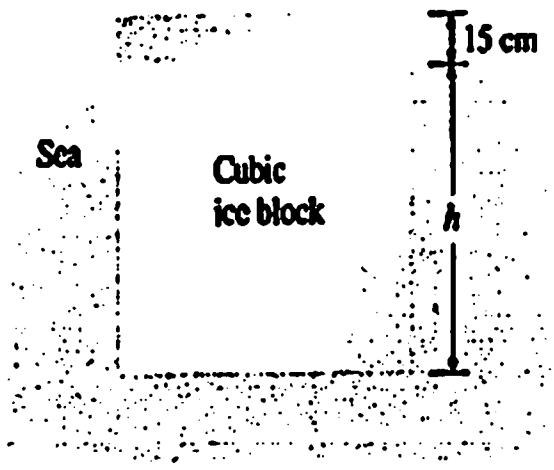


FIGURE P11–33

Step 1 of 4

Given Data:

Specific gravity of ice is $SG_{ice} = 0.92$

Specific gravity of seawater is $SG_{seawater} = 1.025$

Height of the block above the surface, $x = 15\text{cm}$

Step 2 of 4

To find the height of the ice under sea water

The density of sea water is given by,

$$\rho_{seawater} = 1000 \times SG_{seawater}$$
$$= (1000)(1.025)$$
$$= 1025\,\text{kg/m}^3$$

The density of ice is given by,

$$\rho_{ice} = 1000 \times SG_{ice}$$
$$= (1000)(0.92)$$
$$= 920\,\text{kg/m}^3$$

Step 3 of 4

Let the cross-sectional area of the ice block be A.

Total volume of the ice block is given by,

$$V_{total} = A \times (h + x)$$
$$= A \times (h + 15)$$

Volume of the submerged ice block is given by,

$$V_{sub} = A \times h$$

Weight of the ice block is given by,

$$W = \rho_{ice} \times g \times V_{total}$$

Buoyancy force acting on the ice block is given by,

$$F_B = \rho_{seawater} \times g \times V_{sub}$$

Step 4 of 4

The weight of the block is equal to buoyancy force.

$$W = F_B$$
$$\rho_{ice} \times g \times V_{total} = \rho_{seawater} \times g \times V_{sub}$$
$$920 \times g \times A \times (h + 15) = 1025 \times g \times A \times h$$
$$920 \times (h + 15) = 1025 \times h$$
$$105 \times h = 138000$$
$$h = 131.4\ \text{cm}$$
$$= 1.1314\ \text{m}$$

Therefore, the height of the ice block below the surface of water, $\boxed{h = 1.31\text{m}}$

Question: 11.034P

It is estimated that 90 percent of an iceberg's volume is below the surface, while only 10 percent is visible above the surface. For seawater with a density of 1025 kg/m3, estimate the density of the iceberg.

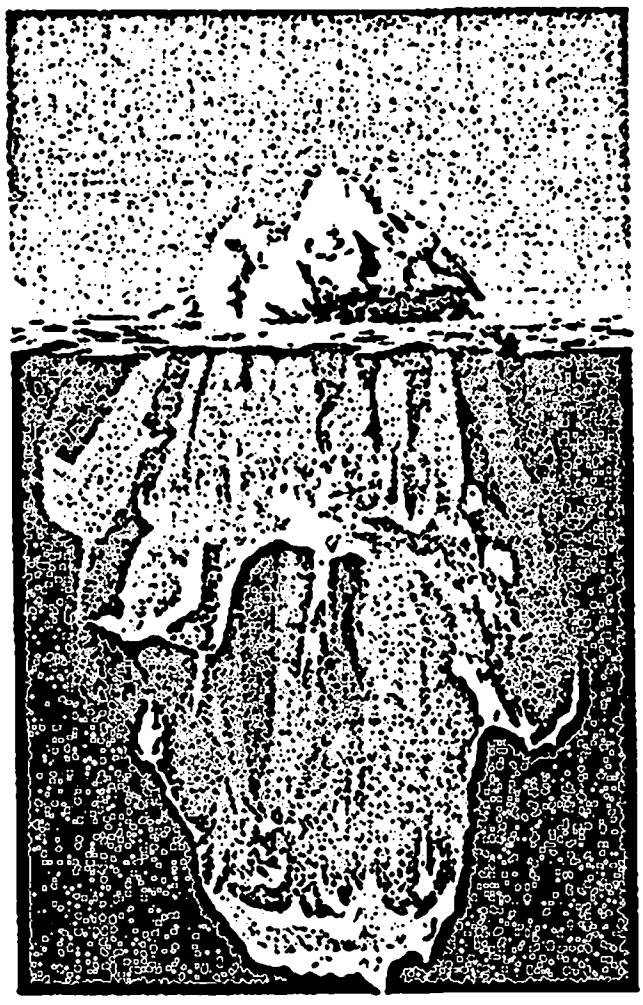


FIGURE P11–34

Step 1 of 3

Given Data:

Percentage of volume of iceberg below the water surface is 90%, $V_{sub} = 0.9V$

Percentage of volume of iceberg above the water surface is 10%, $V_{above} = 0.1V$

Density of sea water, $\rho_{seawater} = 1025 kg/m^3$.

Step 2 of 3

Let the density of the iceberg be $\rho_{iceberg}$

Total weight of the iceberg is given by,

$W = \rho_{iceberg} \times g \times V$

Buoyancy force acting on the iceberg is given by,

$$F_B = \rho_{seawater} \times g \times V_{sub}$$
$$= 1025 \times g \times (0.9V)$$
$$= 922.50 \times g \times V$$

Step 3 of 3

The weight of the block is equal to buoyancy force

$W = F_B$

$\rho_{iceberg} \times g \times V = 922.50 \times g \times V$

$\rho_{iceberg} = 922.50 \ kg/m^3$

Therefore, the density of the iceberg, $\boxed{\rho_{iceberg} = 922.50 kg/m^3}$

Question: 11.035P

A 170-kg granite rock (ρ = 2700 kg/m3) is dropped into a lake. A man dives in and tries to lift the rock. Determine how much force the man needs to apply to lift it from the bottom of the lake. Do you think he can do it?

Step 1 of 3

Weight of the rock in air

$$W = mg$$

Here m is the Mass of the granite; g is acceleration due to gravity.

Substitute $170\,\text{kg}$ for m, and 9.81m/s^2 for g

$$W = (170\,\text{kg})(9.81\text{m/s}^2)$$
$$= 1667.7\text{N}$$

Volume of the rock in air

$$V = \frac{m}{\rho}$$

Here m is the Mass of the granite; ρ_{Granit} is Density of the granite.

Substitute $170\,\text{kg}$ for m, and $2700\,\text{kg/m}^3$ for ρ_{Granit}

$$V = \frac{170\,\text{kg}}{2700\,\text{kg/m}^3}$$
$$= 0.06296\,\text{m}^3$$

Step 2 of 3

The buoyancy force acting on the rock is

$$F_B = \rho_{water} gV$$

Here ρ_{water} is the Density of water; g is acceleration due to gravity, and V is the volume of the rock.

Substitute $1000\,\text{kg}$ for ρ_{water}, 9.81m/s^2 for g, and $0.06296\,\text{m}^3$ for V

$$F_B = (1000\,\text{kg/m}^3)(9.81\text{m/s}^2)(0.06296\,\text{m}^3)$$
$$= 617.6376\text{N}$$

Force required equal to weight of the rock in water

$$W_{Water} = W_{air} - F_B$$
$$= 1667.7 - 617.6376$$
$$= 1050.06\text{N}$$

Weight of the rock in water $\boxed{1050.06\text{N}}$

Step 3 of 3

The force corresponding to mass is

$$m = \frac{W_{water}}{g}$$
$$= \frac{1050.06}{9.81}$$
$$= 107.0397\,\text{kg}$$

The weight of the granite in water is

$$\text{Lift Percentage (\%)} = \frac{1667.7 - 617.6376}{1667.7}$$
$$= 0.6296\% \text{ of water}$$

Mass of the granite in water

$$m_{gra} = 0.6296 \times 170$$
$$= 107.032\,\text{kg}$$

Therefore a person can lift a rock of 107.032kg on earth, so he can lift the rock in water also, as $m_{gra} < m$.

Question: 11.036P

It is said that Archimedes discovered his principle during a bath while thinking about how he could determine if King Hiero's crown was actually made of pure gold. While in the bathtub, he conceived the idea that he could determine the average density of an irregularly shaped object by weighing it in air and also in water. If the crown weighed 3.20 kgf (= 31.4 N) in air and 2.95 kgf (= 28.9 N) in water, determine if the crown is made of pure gold. The density of gold is 19,300 kg/m3. Discuss how you can solve this problem without weighing the crown in water but by using an ordinary bucket with no calibration for volume. You may weigh anything in air.

Step 1 of 4

Given Data:

Weight of the crown in air, $W_a = 31.4\ \text{N}$

Weight of the crown in water, $W_w = 28.9\ \text{N}$

Density of gold, $\rho_g = 19300\ \text{kg/m}^3$

Step 2 of 4

Let us assume that the buoyant force in air is negligible.

The buoyant force acting on the crown by the water is given by,

$$\begin{aligned} F_B &= W_a - W_w \\ &= 31.4 - 28.9 \\ &= 2.5\ \text{N} \end{aligned}$$

Mass of the crown is given by,

$$\begin{aligned} m_c &= \frac{W_a}{g} \\ &= \frac{31.4}{9.81} \\ &= 3.2\ \text{kg} \end{aligned}$$

Step 3 of 4

Buoyant force is equal to the weight of the volume of water displaced.

$$F_B = \rho_{water} \times g \times V$$
$$2.5 = 1000 \times 9.81 \times V$$
$$V = 2.55 \times 10^{-4}\ \text{m}^3$$

The volume of the water displaced is equal to the volume of the crown.

Let the average density of the crown be ρ_{avg}.

Average density of the crown is given by,

$$\begin{aligned} \rho_{avg} &= \frac{m}{V} \\ &= \frac{3.2}{2.55 \times 10^{-4}} \\ &= 12549.02\ \text{kg/m}^3 \end{aligned}$$

Since, $\rho_{avg} < \rho_g$. Therefore, we can conclude that the crown is not made of pure gold.

Step 4 of 4

If the weight of the crown in water is not known:

A bucket which is half filled with water is taken. The weight of the bucket is weighted and be W_1.

Now the crown is completed immersed into the water. After marking the new water level in the bucket, the crown is taken out.

Now, water is filled in the bucket upto the new water mark and is weighed. Let it be W_2.

The volume of the crown is calculated by,

$$V = \frac{W_2 - W_1}{\rho_{water} \times g}.$$

The weight of the crown in air is known as W_a.

The average density of the crown is calculated as follows:

$$\rho_{avg} = \frac{W_a}{V \times g}$$

Question: 11.037P

One of the common procedures in fitness programs is to determine the fat-to-muscle ratio of the body. This is based on the principle that the muscle tissue is denser than the fat tissue, and, thus, the higher the average density of the body, the higher is the fraction of muscle tissue. The average density of the body can be determined by weighing the person in air and also while submerged in water in a tank. Treating all tissues and bones (other than fat) as muscle with an equivalent density of ρmuscle,obtain a relation for the volume fraction of body fat $x$fat.

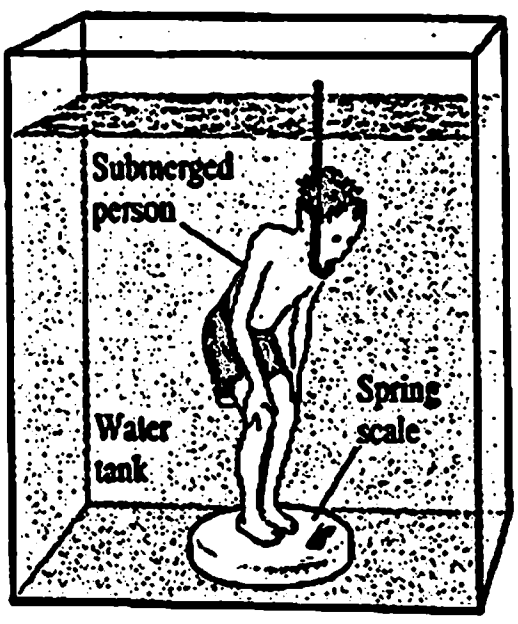


FIGURE P11-37

Step 1 of 3

Let the total volume of the body be

Let the average density of the body be

Let the density of the fat tissue be ρ_{fat}

Volume of the fat tissue is given by,

$$V_{fat} = x_{fat} \times V$$

Volume of the muscle tissue is given by,

$$\begin{aligned} V_{muscle} &= V - V_{fat} \\ &= V - x_{fat} \times V \\ &= V(1 - x_{fat}) \end{aligned}$$

Step 2 of 3

Mass of the fat tissue is given by,

$$\begin{aligned} m_{fat} &= V_{fat} \times \rho_{fat} \\ &= x_{fat} \times V \times \rho_{fat} \end{aligned}$$

Mass of the muscle tissue is given by,

$$\begin{aligned} m_{muscle} &= V_{muscle} \times \rho_{muscle} \\ &= V(1 - x_{fat}) \times \rho_{muscle} \end{aligned}$$

Total mass of the body is given by,

$$m = V \times \rho_{avg}$$

Step 3 of 3

The total mass of the body is equal to the sum of mass of the fat tissue and muscle tissue. Therefore,

$$m = m_{fat} + m_{muscle}$$

$$V \times \rho_{avg} = (x_{fat} \times V \times \rho_{fat}) + (V(1 - x_{fat}) \times \rho_{muscle})$$

$$\rho_{avg} = x_{fat} \times \rho_{fat} + (1 - x_{fat}) \times \rho_{muscle}$$

$$\rho_{avg} = x_{fat} \times \rho_{fat} + \rho_{muscle} - x_{fat} \times \rho_{muscle}$$

$$x_{fat}(\rho_{muscle} - \rho_{fat}) = \rho_{muscle} - \rho_{avg}$$

$$x_{fat} = \frac{\rho_{muscle} - \rho_{avg}}{\rho_{muscle} - \rho_{fat}}$$

Therefore, the volume fraction of the body fat, $\boxed{x_{fat} = \frac{\rho_{muscle} - \rho_{avg}}{\rho_{muscle} - \rho_{fat}}}$.

Question: 11.038P

The hull of a boat has a volume of 180 m3, and the total mass of the boat when empty is 8560 kg. Determine how much load this boat can carry without sinking (*a*) in a lake and (*b*) in seawater with a specific gravity of 1.03.

Step 1 of 4

Given Data:

Volume of boat hull, $V = 180\ \text{m}^3$

Mass of the boat, $m = 8560\ \text{kg}$

The weight of the empty boat is given by,

$$\begin{aligned} W_{boat} &= mg \\ &= (8560\,\text{kg})(9.81\,\text{m/s}^2) \\ &= 83.97\times10^3\ \text{N} \\ &= 83.97\,\text{kN} \end{aligned}$$

Step 2 of 4

(a) To find the load that can carry by the boat without sinking in the lake.

Density of lake water, $\rho_{lake} = 1000\ \text{kg/m}^3$

Buoyancy force acting on the boat in lake is given by,

$$\begin{aligned} F_{B,lake} &= \rho_{lake} g V \\ &= (1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(180) \\ &= 1765.80\times10^3\ \text{N} \\ &= 1765.8\,\text{kN} \end{aligned}$$

The load that can be carried by this boat without sinking in lake is given by,

$$\begin{aligned} W_{load,lake} &= F_{B,lake} - W_{boat} \\ &= 1765.8 - 83.97 \\ &= 1681.83\,\text{kN} \end{aligned}$$

Therefore, the load that can be carried by this boat without sinking in lake,

$$\boxed{W_{load,lake} = 1681.83\ \text{kN}}$$

Step 3 of 4

(b) To find the load that can carry by the boat without sinking in the sea.

Specific gravity of sea water, $SG = 1.03$

Density of sea water is given by,

$$\begin{aligned} \rho_{sea} &= SG \times \rho_{water} \\ &= 1.03\times1000 \\ &= 1030\ \text{kg/m}^3 \end{aligned}$$

Buoyancy force acting on the boat in sea is given by,

$$\begin{aligned} F_{B,sea} &= \rho_{sea} g V \\ &= (1030\,\text{kg/m}^3)(9.81\,\text{m/s}^2)(180) \\ &= 1818.77\times10^3\ \text{N} \\ &= 1818.77\,\text{kN} \end{aligned}$$

Step 4 of 4

The load that can be carried by this boat without sinking in sea is given by,

$$\begin{aligned} W_{load,sea} &= F_{B,sea} - W_{boat} \\ &= 1818.77 - 83.97 \\ &= 1734.80\,\text{kN} \end{aligned}$$

Therefore, the load that can be carried by this boat without sinking in sea,

$$\boxed{W_{load,sea} = 1734.80\ \text{kN}}$$

Question: 11.039P

An air-conditioning system requires a 20-m-long section of 15-cm-diameter ductwork to be laid underwater. Determine the upward force the water will exert on the duct. Take the densities of air and water to be 1.3 kg/m3 and 1000 kg/m3, respectively.

Step 1 of 3

Given data:

Length of the section, $L = 20\ \text{m}$

Diameter of the duct, $D = 15\ \text{cm} = 0.15\ \text{m}$

Density of air, $\rho_{air} = 1.3\ \text{kg/m}^3$

Density of water, $\rho_{water} = 1000\ \text{kg/m}^3$

Step 2 of 3

To find the buoyant force acting on the duct.

Volume of the duct under the water is

$$V = \frac{\pi}{4} D^2 L$$

$$= \frac{\pi}{4} \times 0.15^2 \times 20$$

$$= 0.3534\ \text{m}^3$$

The buoyancy force acting on the duct is given by,

$$F_B = \rho g V$$

$$= 1000 \times 9.81 \times 0.3534$$

$$= 3466.854\ \text{N}$$

Step 3 of 3

Neglecting the air density, the upward force on the duct is equal to the buoyancy force.

Therefore, the upward force acting on the duct is given by,

$$F = F_B$$

$$= 3466.854\ \text{N}$$

Therefore, the upward force that the water will exert on the duct, $\boxed{F = 3466.854\ \text{N}}$.

Question: 11.040P

The density of a floating body can be determined by tying weights to the body until both the body and the weights are completely submerged, and then weighing them separately in air. Consider a wood log that weighs 1540 N in air. If it takes 34 kg of lead (ρ = 11,300 kg/m3) to completely sink the log and the lead in water, determine the average density of the log.

Step 1 of 5

Given data:

The weight of the wooden log, $W_l = 1540$ N

The mass of lead used to submerge the log, $m_d = 34$ kg

The density of lead, $\rho_d = 11{,}300\ \text{kg/m}^3$

Step 2 of 5

Assume

The body is submerged and in a suspended state

We know for a body in suspended state,

$\rho = \rho_f$

Where

ρ is the average density of the body

ρ_f is the density of water

Step 3 of 5

The total mass of the body,

$$\begin{aligned} m &= m_d + m_l \\ &= m_d + \frac{W_l}{g} \\ &= 34 + \frac{1540}{9.81} \\ &= 190.98\ \text{kg} \end{aligned}$$

Step 4 of 5

The mass of the log,

$$\begin{aligned} m_l &= \frac{W_l}{g} \\ &= \frac{1540}{9.81} \\ &= 156.98\ \text{kg} \end{aligned}$$

Step 5 of 5

The volume of lead used,

$$\begin{aligned} \rho_d &= \frac{m_d}{\rho_d} \\ &= \frac{34}{11300}\ \text{m}^3 \end{aligned}$$

Question: 11.041P

The 280-kg, 6-m-wide rectangular gate shown in Fig. P11–41 is hinged at *B* and leans against the floor at *A* making an angle of 45° with the horizontal. The gate is to be opened from its lower edge by applying a normal force at its center. Determine the minimum force *F* required to open the water gate.

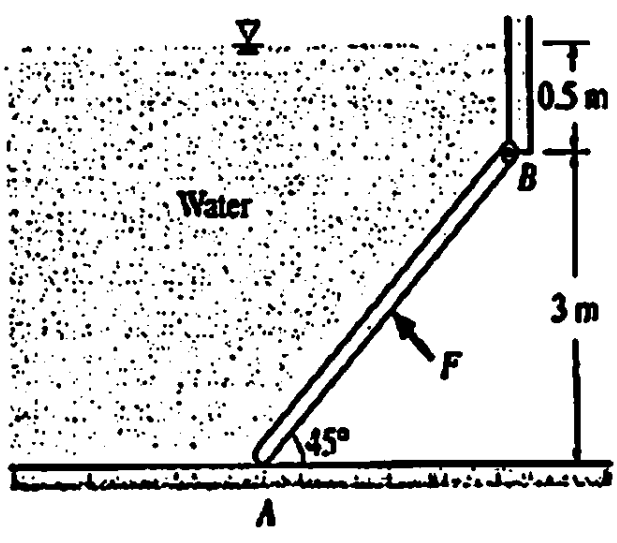
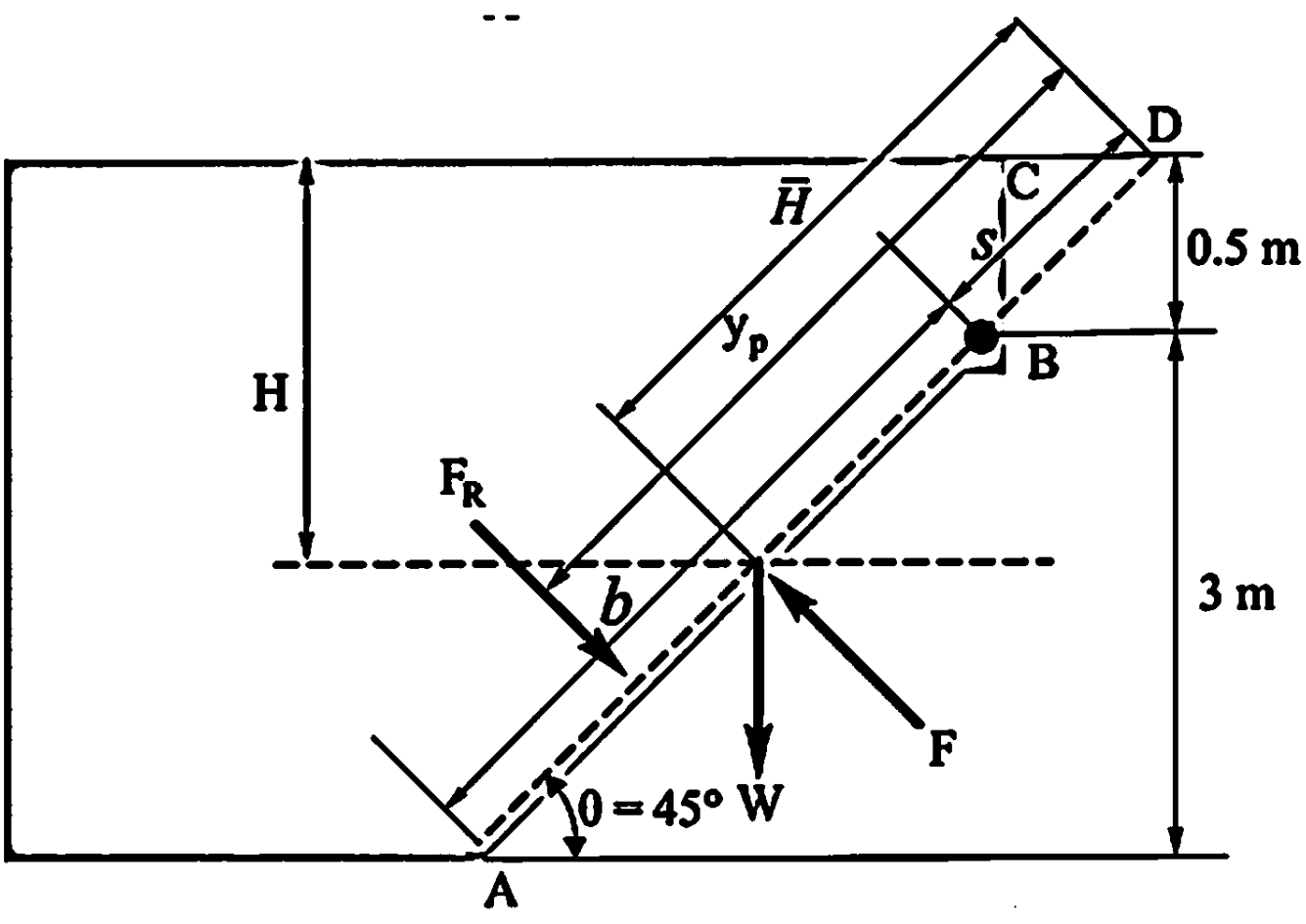


FIGURE P11–41

Step 1 of 6

Show the forces acting on the rectangular gate:



Step 2 of 6

Calculate the point of center of pressure acting on the gate due to the liquid.

$$y_p = \bar{H} + \frac{I}{A\bar{H}}$$

$$= \left(\frac{s+b}{2}\right) + \frac{\frac{b^3 d}{12}}{b \times d\left(\frac{s+b}{2}\right)}$$

$$= \left(\frac{s+b}{2}\right) + \frac{b^2 d}{12\left(\frac{s+b}{2}\right)}$$

Here, s is the length of the gate (AB), $\bar{H}$ is the inclined distance of the centroid from the surface, b is the distance from point B to D, and d is the width of the gate.

Calculate the vertical centroid distance of the gate from the water surface.

$$H = \bar{H}\sin 45°$$

$$= \left(\frac{s+b}{2}\right)\sin 45°$$

Step 3 of 6

Calculate the resultant force acting on the gate due to water.

$$F_R = \rho g H \times A$$

Here ρ is the density of the water, and A is the cross sectional area of the gate.

Take moments about B.

$$F_R\left(y_p - s\right) + W\left(\frac{b}{2}\cos 45°\right) = F \times \frac{b}{2}$$

Here, W is the weight of the gate.

Step 4 of 6

Use EES software to get the solution.

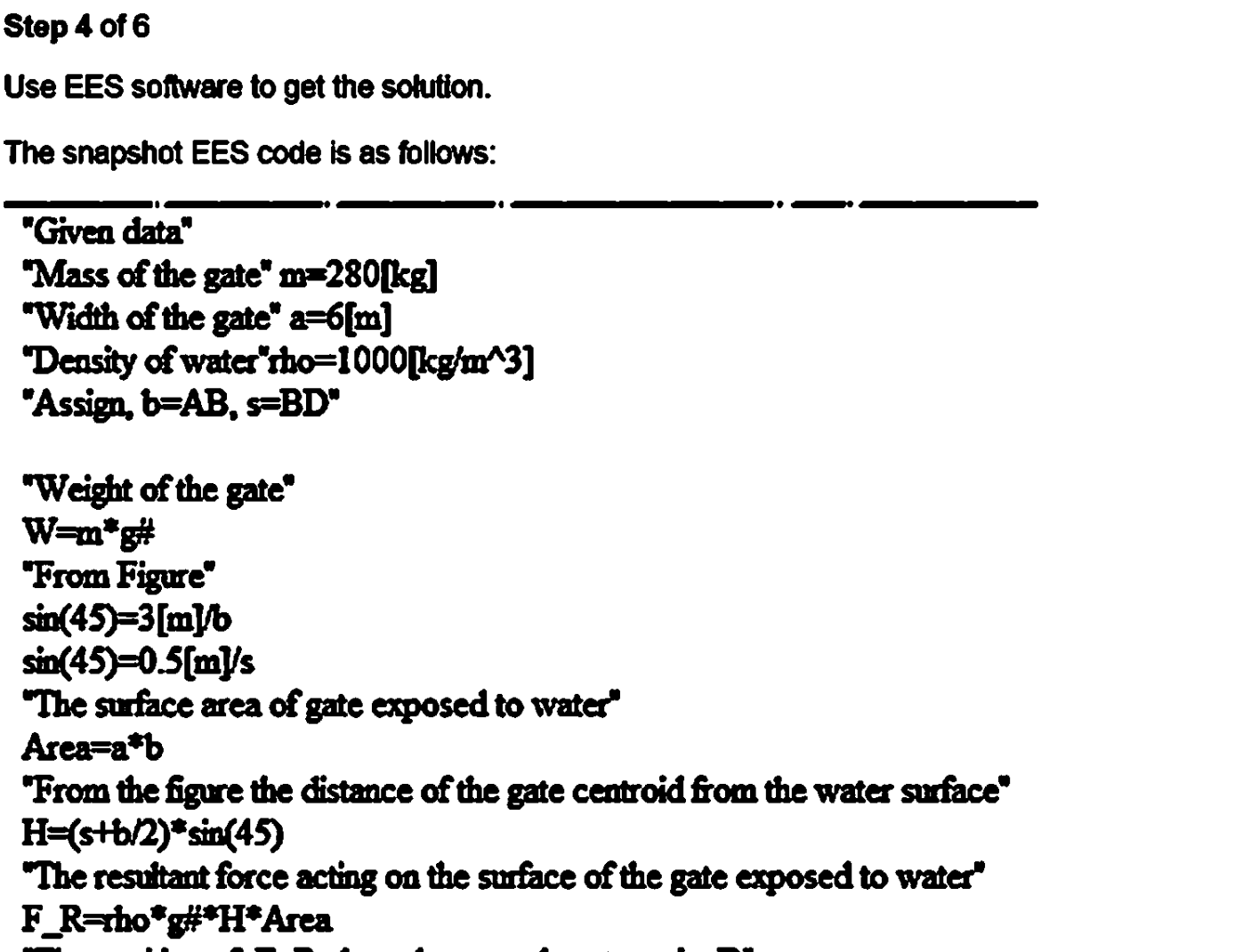
The snapshot EES code is as follows:

```
"Given data"
"Mass of the gate" m=280[kg]
"Width of the gate" a=6[m]
"Density of water"rho=1000[kg/m^3]
"Assign, b=AB, s=BD"

"Weight of the gate"
W=m*g#
"From Figure"
sin(45)=3[m]/b
sin(45)=0.5[m]/s
"The surface area of gate exposed to water"
Area=a*b
"From the figure the distance of the gate centroid from the water surface"
H=(s+b/2)*sin(45)
"The resultant force acting on the surface of the gate exposed to water"
F_R=rho*g#*H*Area
"The position of  F_R along the gate plane to point D"
y_p=s+b/2+b^2/(12*(s+b/2))
"Taking moment about the point B"
F_R*(y_p-s)-F*(b/2)+W*(b*cos(45)/2)=0
```

Step 5 of 6

After writing the EES code, press "F2" key on the keyboard. The solution window pops up. The screen shot of the solution window appears as follows:



Step 6 of 6

From the solution window, the force required to open the gate is as follows:

$$F = 626055\ \text{kg}\cdot\text{m/s}^2$$

$$= 626.055 \times 10^3\ \text{N}$$

$$= 626.055\ \text{kN}$$

Therefore, the minimum force required to open the water gate, is **626.055 kN**.

Question: 11.042P

Repeat Prob. 11–41 for a water height of 1.2 m above the hinge at *B*.

Step 1 of 6

Given Data:

The mass of the gate, $m = 280$ kg

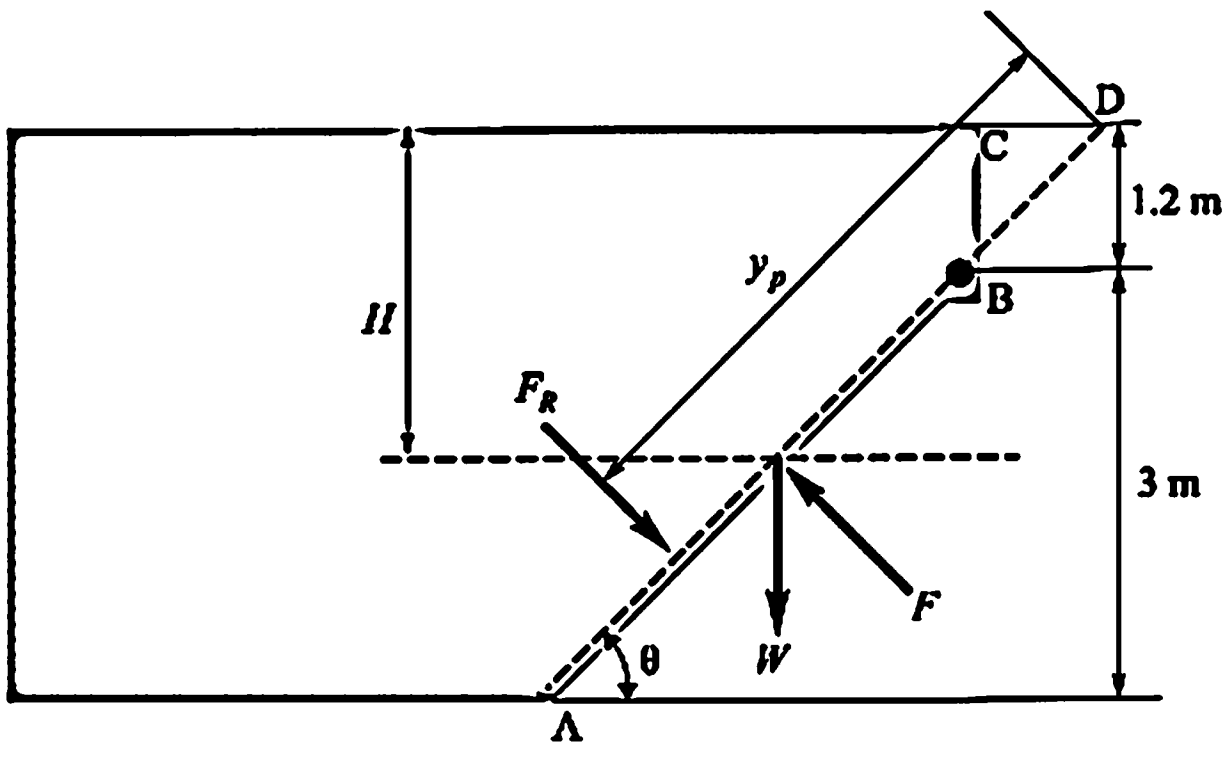
The width of the gate, $a = 6$ m

The angle made by the gate with horizontal, $\theta = 45°$

A normal force is applied at the center of the gate from below.

Step 2 of 6

The forces acting on the gate is shown:



Step 3 of 6

Assign $AB = b$ and $BD = s$

From the figure

$$\sin 45° = \frac{3\text{ m}}{b}$$

$$b = \frac{3}{\sin 45°}$$

$$= 4.2426\text{ m}$$

From the figure,

$$\sin 45° = \frac{1.2\text{ m}}{s}$$

$$s = \frac{1.2}{\sin 45°}$$

$$= 1.697\text{ m}$$

Step 4 of 6

The surface area of the gate exposed to water,

$$A = a \times b$$

$$= 6\text{ m} \times 4.2426\text{ m}$$

$$= 25.4556\text{ m}^2$$

From the figure the distance of the gate centroid from the water surface,

$$H = (s + b/2) \times \sin 45°$$

$$= (1.697 + 4.2426/2) \times \sin 45°$$

$$= 2.699\text{ m}$$

The resultant force acting on the surface of the gate exposed to water,

$$F_R = \rho g H A$$

$$= (1000)(9.81)(2.699)(25.4556)$$

$$= 674.268 \times 10^3\text{ N}$$

Step 5 of 6

The position of F_R along the gate plane to point *D*,

$$y_p = s + (b/2) + \frac{ab^3}{12\left[s + b/2 + P_0/(\rho g \sin\theta)\right]ab}$$

The atmospheric pressure is neglected.

$$y_p = s + (b/2) + \frac{b^2}{12[s + b/2]}$$

$$= 1.697 + (4.2426/2) + \frac{4.2426^2}{1.697 + 4.2426/2}$$

$$= 4.2111\text{ m}$$

Weight of the gate is given by,

$$W = mg$$

$$= 280 \times 9.81$$

$$= 2746.8\text{ N}$$

Step 6 of 6

Taking moment about the point *B*,

$$\sum M_B = 0$$

$$F_R(y_p - s) - F\left(\frac{b}{2}\right) + W\left(\frac{b \times \cos 45°}{2}\right) = 0$$

$$F = \frac{2}{b}\left(F_R(y_p - s) + W\left(\frac{b \times \cos 45°}{2}\right)\right)$$

$$= \frac{2}{4.2426}\left(674.268 \times 10^3 (4.211 - 1.697) + 2746.8\left(\frac{4.2426 \times \cos 45°}{2}\right)\right)$$

$$= 800.71 \times 10^3\text{ N}$$

$$= 800.71\text{ kN}$$

Therefore the applied force, $\boxed{F = 800.71\text{ kN}}$.

Question: 11.043P

A 3-m-high, 6-m-wide rectangular gate is hinged at the top edge at *A* and is restrained by a fixed ridge at *B*. Determine the hydrostatic force exerted on the gate by the 5-m-high water and the location of the pressure center.

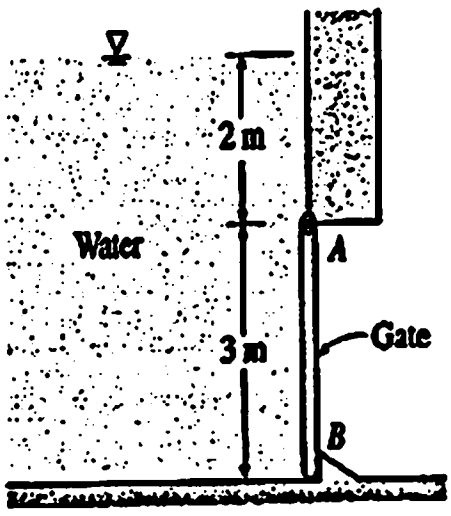
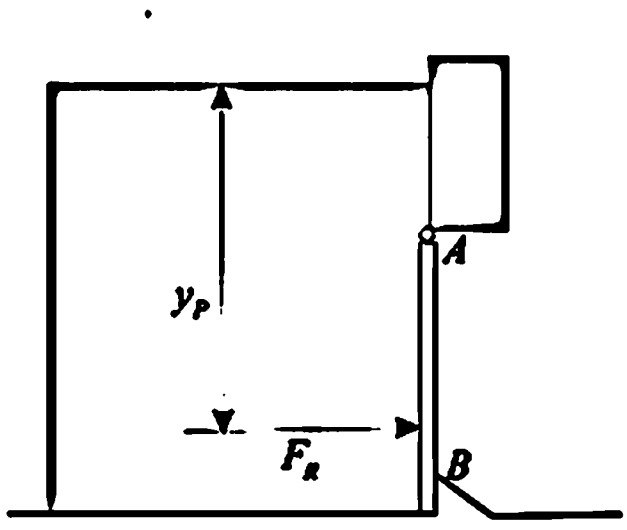


FIGURE P11–43

Step 1 of 3



Step 2 of 3

Center of gravity from the free surface of the water is

$$h_c = 2 + \frac{3}{2}$$
$$= 3.5\,\text{m}$$

Resultant hydrostatic force on the gate is

$$F_R = \rho g h_c A$$

Here ρ is the Density of water; g is acceleration due to gravity, and h_c is the Center of gravity from the free surface of the water.

Substitute 1000kg for ρ, $9.81\,\text{m/s}^2$ for g, and 3.5m for h_c

$$F_R = \left(1000\,\text{kg/m}^3\right)\left(9.81\,\text{m/s}^2\right)(3.5\,\text{m})\left(3\times 6\,\text{m}^2\right)\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)$$
$$= 618\,\text{kN}$$

Resultant hydrostatic force on the gate is $\boxed{618\,\text{kN}}$

Step 3 of 3

The force acting at a vertical distance of $h_p = y_p$ from the free surface directly beneath the centroid of the plate is

$$y_p = s + \frac{b}{2} + \frac{b^2}{12\left(s + \frac{b}{2} + P_0(\rho g)\right)}$$

The free surface of the water is put $P_0 = 0$, then

$$y_p = s + \frac{b}{2} + \frac{b^2}{12\left(s + \frac{b}{2}\right)}$$

Substituting the values, we get the vertical distance of the pressure center from the free surface of water as

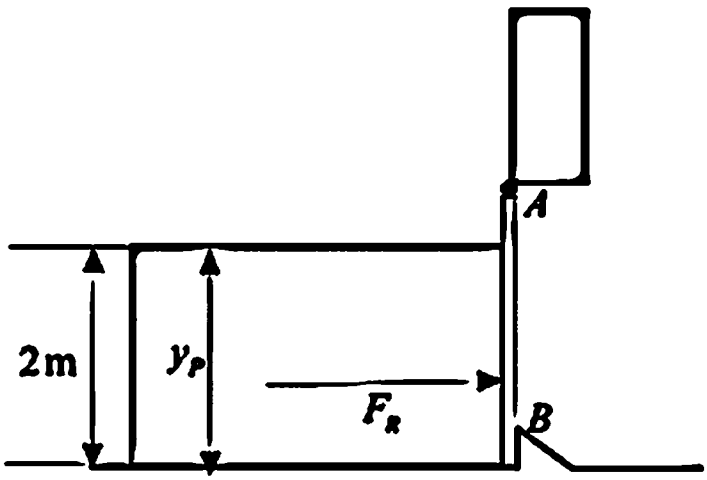
$$y_p = 2 + \frac{3}{2} + \frac{3^2}{12\left(2 + \frac{3}{2}\right)}$$
$$= 3.71\,\text{m}$$

The vertical distance of the pressure center from the free surface of water is $\boxed{3.71\,\text{m}}$

Question: 11.044P

Repeat Prob. 11–43 for a total water height of 2 m.

Step 1 of 3



Step 2 of 3

Center of gravity from the free surface of the water is

$$h_c = \frac{2}{2}$$
$$= 1\,\text{m}$$

Resultant hydrostatic force on the gate is

$$F_R = \rho g h_c A$$

Here ρ is the Density of water; g is acceleration due to gravity, and h_c is the Center of gravity from the free surface of the water.

Substitute 1000kg for ρ, $9.81\,\text{m/s}^2$ for g, and 1m for h_c

$$F_R = \left(1000\,\text{kg/m}^3\right)\left(9.81\,\text{m/s}^2\right)(1\,\text{m})\left(2\times 6\,\text{m}^2\right)\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)$$
$$= 118\,\text{kN}$$

Resultant hydrostatic force on the gate is $\boxed{118\,\text{kN}}$

Step 3 of 3

The vertical distance of the pressure center from the free surface of water is

$$y_p = \frac{2(2\,\text{m})}{3}$$
$$= 1.33\,\text{m}$$

The vertical distance of the pressure center from the free surface of water is $\boxed{1.33\,\text{m}}$

Question: 11.045P

A semicircular 40-ft-diameter tunnel is to be built under a 150-ft-deep, 800-ft-long lake, as shown in Fig. P11–45E. Determine the total hydrostatic force acting on the roof of the tunnel.

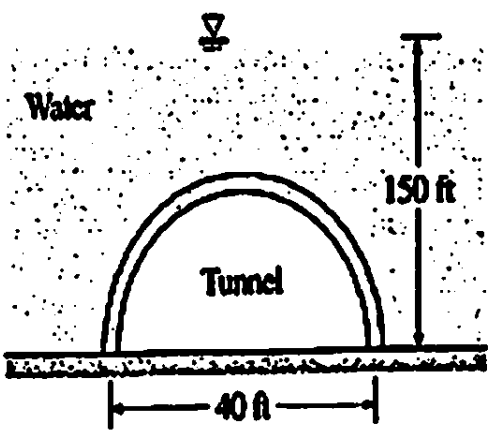


FIGURE P11–45E

Step 1 of 8

Given data:

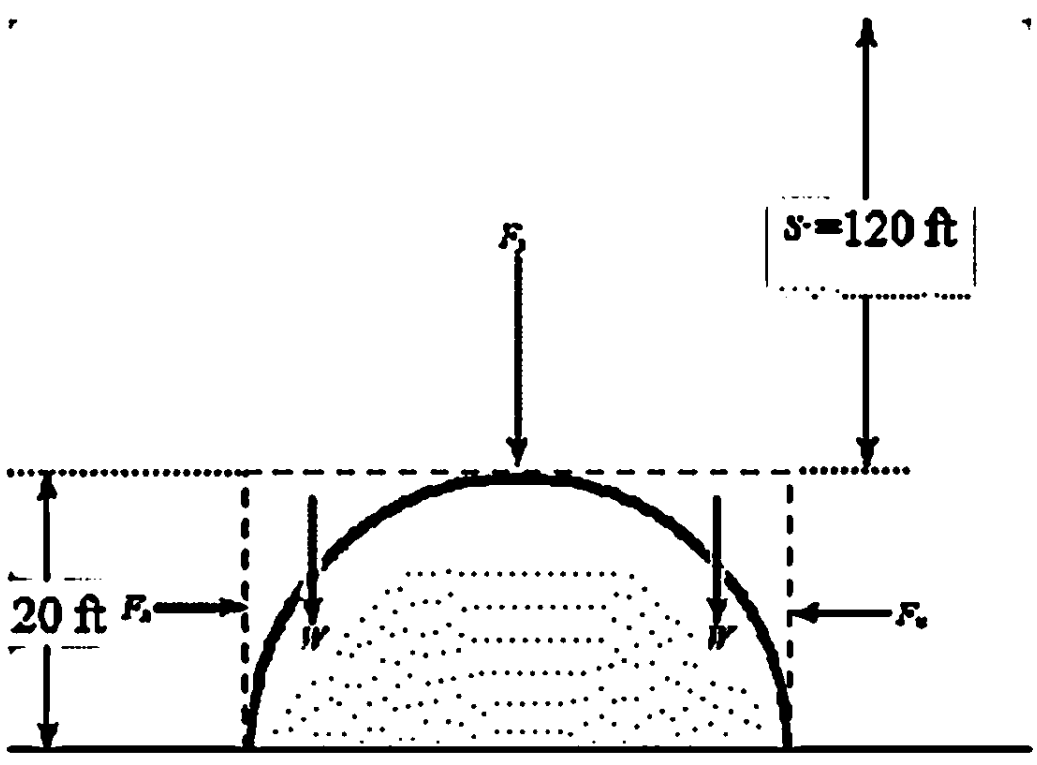
The depth of the lake $h_l = 150\text{ft}$

The length of the lake $l = 800\text{ft}$

The diameter of the tunnel $D = 40\text{ft}$

Step 2 of 8

The free body diagram semicircular tunnel is:



Where

F_y is force acting on horizontal surface

F_x is force acting on each vertical surface

W is the weight of the fluid acting on each side

Step 3 of 8

The tunnel has been enclosed in a rectangular fluid block of height equal to the radius of the tunnel, $R = 20\text{ft}$

Consider the rectangular fluid block with the tunnel.

Consider the horizontal forces on a vertical side.

$$F_x = P_{avg}A$$
$$= \rho g h_C A$$
$$= \rho g\left(s + \frac{R}{2}\right)A$$

Where area A on vertical surface is

$$A = 20\times 800$$
$$= 16000\,\text{ft}^2$$

Step 4 of 8

Substituting the required values:

The horizontal force acting on each vertical side of the tunnel

$$F_x = \left(62.4\frac{\text{lbm}}{\text{ft}^3}\right)\left(32.2\frac{\text{ft}}{\text{sec}^2}\right)\left(130 + \frac{20}{2}\right)\text{ft}\times 16000\,\text{ft}^2\times\frac{1\text{lbf}}{32.2\text{lbm}.\frac{\text{ft}}{\text{sec}^2}}$$

$$= 139.776\times 10^6\,\text{lbf}$$

Since horizontal force acting on the right side and left side of the tunnel is equal and opposite then the net total force acting on the tunnel in horizontal direction is zero.

$$F_X = F_x - F_x$$
$$= 0$$

Step 5 of 8

The vertical force acting on horizontal side of the tunnel is

$$F_y = \rho g h_C A$$
$$= \rho g s\left(D\times l\right)$$
$$= 62.4\frac{\text{lbm}}{\text{ft}^3}\times 32.2\frac{\text{ft}}{\text{sec}^2}\times 130\text{ft}\times(40\times 800)\text{ft}^2\times\frac{1\text{lbf}}{32.2\text{lbm}.\frac{\text{ft}}{\text{sec}^2}}$$
$$= 259.584\times 10^6\,\text{lbf}$$

Step 6 of 8

Weight of the fluid acting on one side of the tunnel is

$$W = \rho g V$$
$$= \rho g\left(R^2 - \frac{\pi R^2}{4}\right)l$$
$$= 62.4\frac{\text{lbm}}{\text{ft}^3}\times 32.2\frac{\text{ft}}{\text{sec}^2}\left(20^2 - \frac{\pi\times 20^2}{4}\right)\text{ft}^2\times 800\text{ft}\times\frac{1\text{lbf}}{32.2\text{lbm}\cdot\frac{\text{ft}}{\text{sec}^2}}$$
$$= 4.285\times 10^6\,\text{lbf}$$

Step 7 of 8

The total vertical force acting on the top of the tunnel

$$F_Y = F_y + W + W$$
$$F_Y = F_y + 2W$$
$$= 259.584\times 10^6 + 2\times 4.285\times 10^6$$
$$= 268.154\times 10^6\,\text{lbf}$$

Step 8 of 8

The total force acting on the tunnel is

$$F = \sqrt{F_X^2 + F_Y^2}$$
$$= \sqrt{F_Y^2}$$
$$= F_Y$$
$$= 268.154\times 10^6$$

Therefore, the total force acting on the tunnel $\boxed{F = 268.154\times 10^6}$

Question: 11.046P

A 50-ton, 6-m-diameter hemispherical dome on a level surface is filled with water, as shown in Fig. P11–46. Someone claims that he can lift this dome by making use of Pascal's law by attaching a long tube to the top and filling it with water. Determine the required height of water in the tube to lift the dome. Disregard the weight of the tube and the water in it.

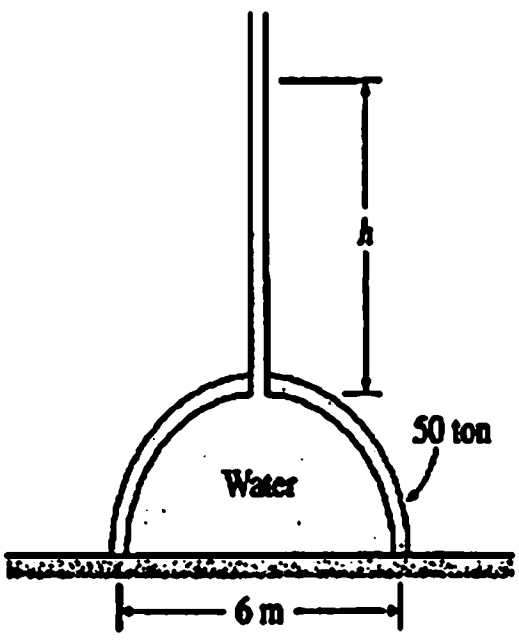


FIGURE P11–46

Step 1 of 4

Given data:

The mass of hemispherical dome, $m_D = 50 \times 10^3 \text{ kg}$

The diameter of the dome, $D = 6 \text{ m}$

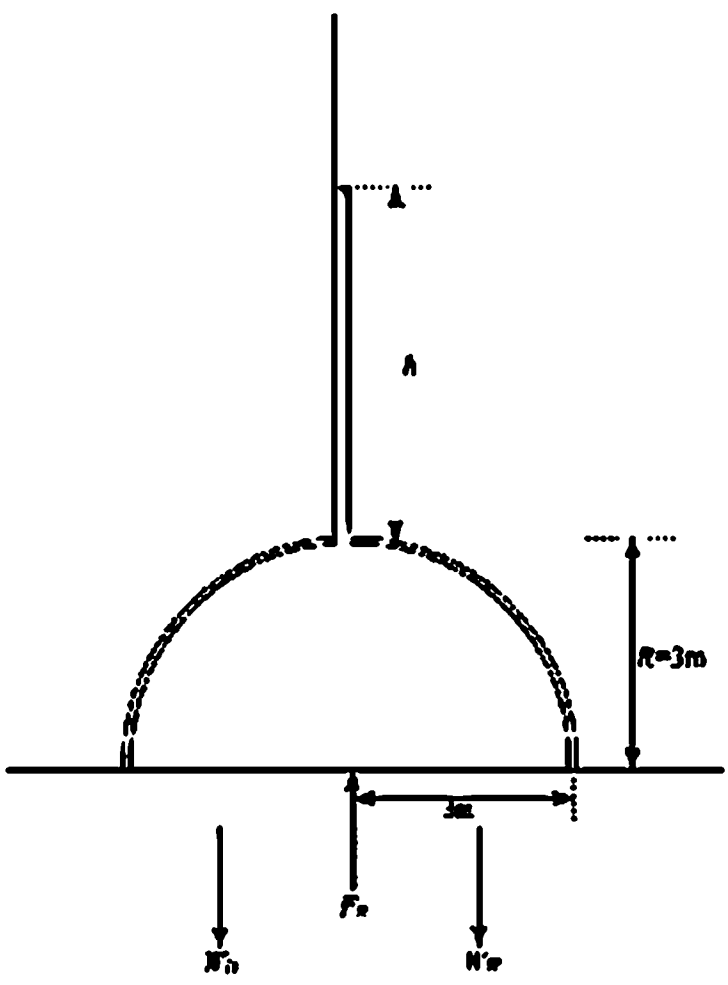
Radius of the dome, $R = \frac{D}{2} = 3\text{m}$

The weight of the tube and the water in it can be neglected.

Assuming atmospheric pressure acts on both sides of the dome and can be neglected.

Step 2 of 4

The hemispherical dome is as shown in figure:



The volume of water inside the hemispherical dome,

$$V_W = \frac{2}{3}\pi R^3$$

Step 3 of 4

Consider the vertical forces.

Applying the equilibrium condition:

$$\sum F_y = 0$$

$$F_R - W_W - W_D = 0$$

Where W_D is the weight of the dome and

W_W is the weight of water inside the dome

F_R is the hydrostatic force acting from bottom

Then

$$F_R = W_W + W_D$$

$$\rho(h+R)\pi R^2 g = m_D g + m_{water} g$$

$$\rho\pi R^2 (h+R) = m_D + m_{water}$$

$$h = \frac{m_D + m_{water}}{\rho\pi R^2} - R$$

Step 4 of 4

Then substituting the known values

$$h = \frac{m_D + \rho_{water} V_W}{\rho\pi R^2} - R$$

$$= \frac{m_D + \rho_{water}\left(\frac{2}{3}\pi R^3\right)}{\rho\pi R^2} - R$$

$$= \frac{\left(50 \times 10^3\right) + \left(1000\right)\left(\frac{2}{3}\pi \times 3^3\right)}{1000\pi \times 3^2} - 3$$

$$= \frac{50000 + 56548.66}{28274.33} - 3$$

$$= 0.768 \text{ m}$$

Therefore the required height of water in the tube is $\boxed{h = 0.77 \text{ m}}$

Question: 11.047P

The water in a 25-m-deep reservoir is kept inside by a 150-m-wide wall whose cross section is an equilateral triangle, as shown in Fig. P11–47. Determine (*a*) the total for * (hydrostatic + atmospheric) acting on the inner surface of the wall and its line of action and (*b*) the magnitude of the? horizontal component of this force. Take *P*atm = 100 kPa.

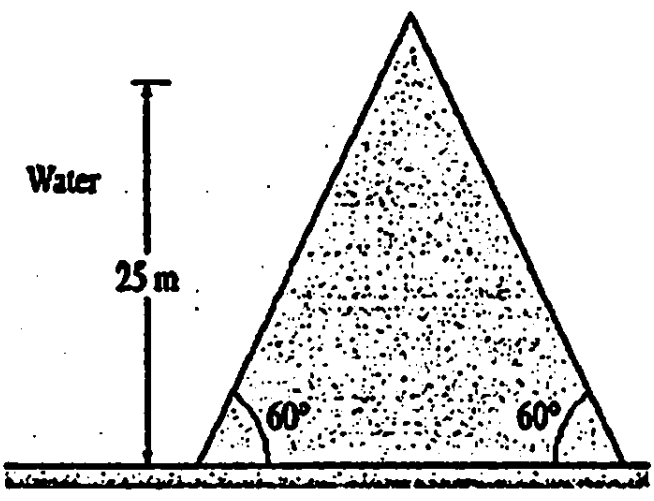


FIGURE P11–47

Step 1 of 7

Given data:

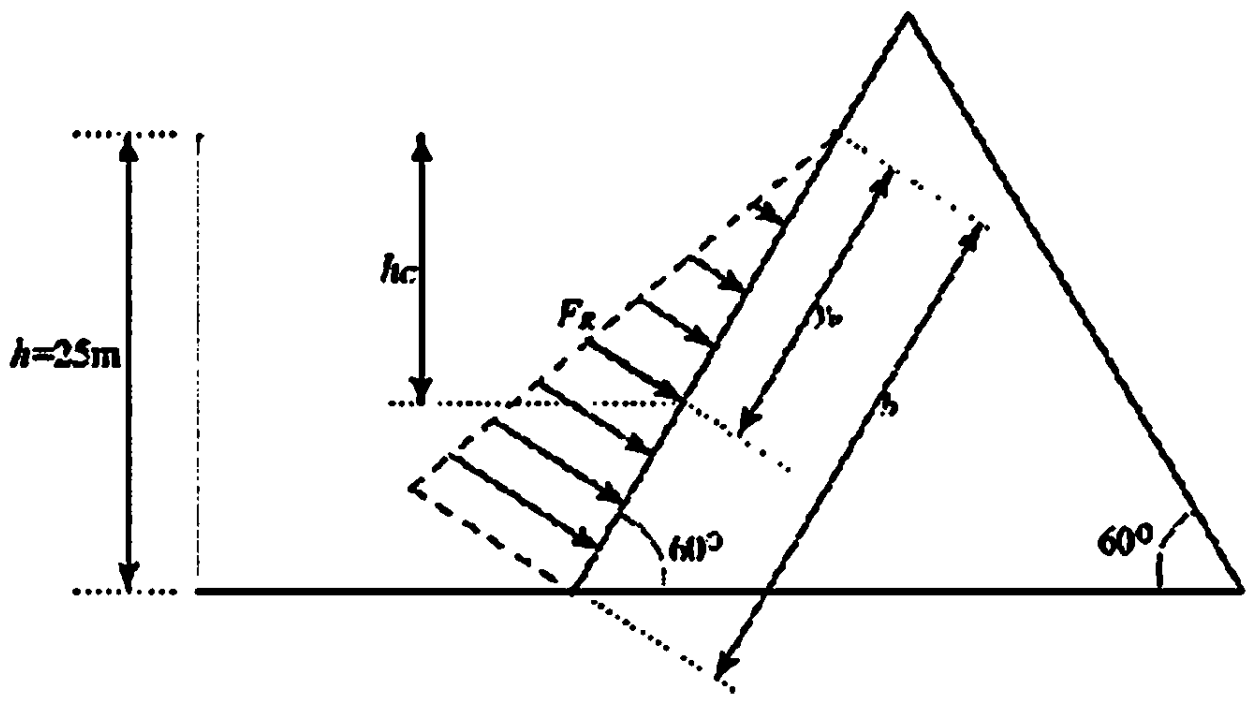
The depth of the reservoir, $d = 25\ \text{m}$

The atmospheric pressure, $P_{atm} = 100\ \text{kPa}$

Width of the wall, $w = 150\ \text{m}$

Step 2 of 7

The figure representing reservoir wall whose cross section is triangle is as shown below:



Step 3 of 7

Assuming:

Atmospheric pressure is acting on both sides of the gate, which cancels each other.

From the above figure,

$$b = \frac{h}{\sin 60^\circ}$$
$$= \frac{25}{\sin 60^\circ}$$
$$= 28.867\ \text{m}$$

Step 4 of 7

The vertical distance of the centroid from the free surface of the liquid is

$$h_C = \frac{h}{2}$$
$$= \frac{25}{2}$$
$$= 12.5\text{m}$$

Step 5 of 7

a)

The hydrostatic force acting on the wall is

$$F_R = P_{avg} A$$
$$= (P_{atm} + \rho g h_C) A$$

Where area of the surface is

$$A = w \times b$$
$$= 150 \times 28.867$$
$$= 4330.05\text{m}^2$$

Substituting the known values in the above equation

$$F_R = \left((100 \times 10^3) + (1000 \times 9.81 \times 12.49) \right) 4330.05$$
$$= 9.635 \times 10^8\text{N}$$

Therefore, the total force acting on the inner surface of the wall $\boxed{F_R = 9.635 \times 10^8\text{N}}$

Step 6 of 7

The line of action of the total force is

$$y_p = s + \frac{b}{2} + \frac{b^2}{12\left(s + \frac{b}{2} + \frac{P_0}{\rho g \sin\theta}\right)}$$
$$= 0 + \frac{28.867}{2} + \frac{28.867^2}{12\left(0 + \frac{28.867}{2} + \frac{100 \times 10^3}{1000 \times 9.81 \times \sin 60^\circ}\right)}$$
$$= 14.4335 + 2.650$$
$$= 17.08\ \text{m}$$

Therefore, line of action of the centre of pressure acting along the wall surface,

$\boxed{y_p = 17.08\ \text{m}}$

Step 7 of 7

b)

The horizontal component of hydrostatic force,

$$F_H = F_R \sin 60^\circ$$
$$= \left(9.635 \times 10^8\right) \sin 60^\circ$$
$$= 8.334 \times 10^8\text{N}$$

Therefore the horizontal component of hydrostatic force $\boxed{F_H = 8.334 \times 10^8\text{N}}$

Question: 11.048P pg 1

An elastic air balloon having a diameter of 30 cm is attached to the base of a container partially filled with water at +4°C, as shown in Fig. P11–48. If the pressure of air above water is gradually increased from 100 kPa to 1.6 MPa, will the force on the cable change? If so, what is the percent change in the force? Assume the pressure on the free surface and the diameter of the balloon are related by $P = CD^n$, where C is a constant and $n = -2$. The weight of the balloon and the air in it is negligible.

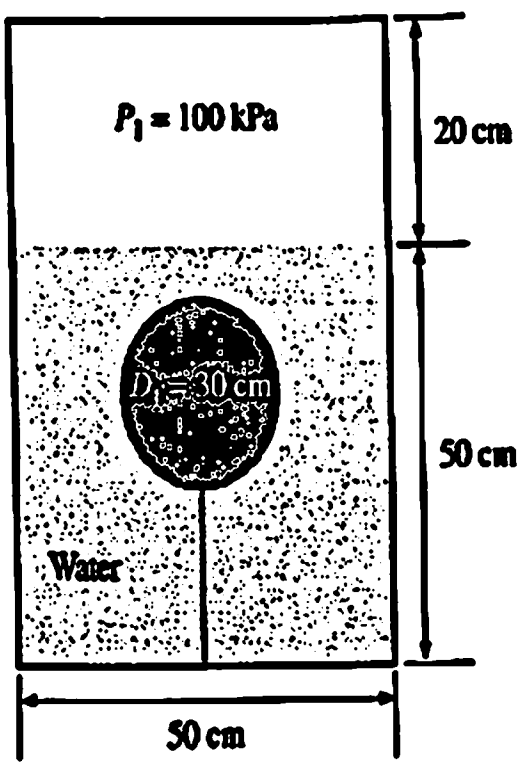


FIGURE P11–48

Step 1 of 9

Given data:

The diameter of the balloon, $D_1 = 0.3$ m.

The initial pressure of air above water $p_1 = 100$ kPa

The final pressure of air above water $p_2 = 1.6$ MPa

The pressure on the surface and diameter of balloon is related by $P = CD^n$.

C is a constant and $n = -2$.

Step 2 of 9

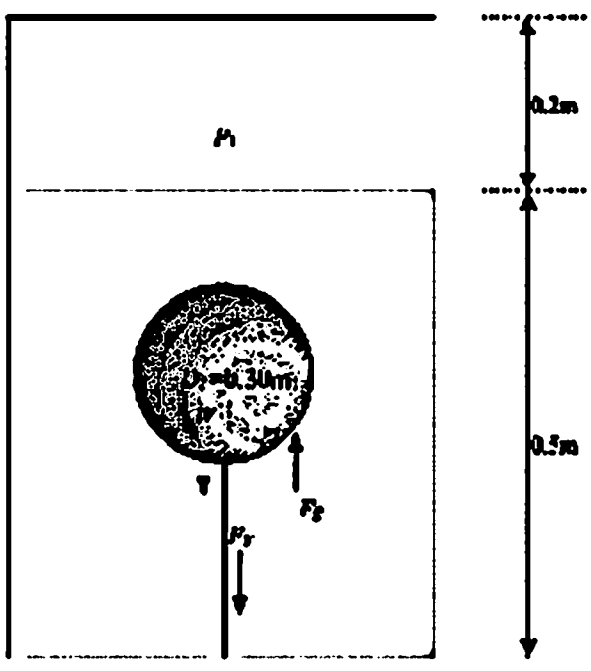
Assuming:

The weight of the balloon and air in it is negligible.

The effect of atmospheric pressure is neglected.

Step 3 of 9

An elastic air balloon is attached to the base of a container is as shown below:



Step 4 of 9

The pressure on the surface and diameter of balloon is related by

$$P = CD^n$$

$$C = \frac{P}{D^n}$$

At initial state 1

$$C = \frac{P_1}{D_1^n}$$

$$= \frac{P_1}{D_1^{-2}}$$

$$C = P_1 D_1^2 \quad \text{..... (1)}$$

And at final state 2

$$C = \frac{P_2}{D_2^n}$$

$$= \frac{P_2}{D_2^{-2}}$$

$$C = P_2 D_2^2 \quad \text{..... (2)}$$

Comparing equations (1) and (2)

$$P_1 D_1^2 = P_2 D_2^2 \quad \text{..... (3)}$$

Step 5 of 9

From equation (3) the final diameter of the balloon when the pressure $P_2 = 1.6$ MPa is

$$P_1 D_1^2 = P_2 D_2^2$$

$$D_2 = D_1 \sqrt{\frac{P_1}{P_2}}$$

$$= (0.3) \sqrt{\frac{100 \times 10^3}{1.6 \times 10^6}}$$

$$= 0.075 \text{ m}$$

Step 6 of 9

From the diagram,

The tension in the cable

$$F_T = F_B - W$$

Where

F_B is the buoyant force on the balloon.

W is the weight of the balloon which is negligible.

Then

$$F_T = F_B$$

For buoyant force,

F_B is the buoyant force on the balloon.
W is the weight of the balloon which is negligible.
Then
$F_T = F_B$
For buoyant force,
$F_B = \rho_{water} g V_{balloon}$

Step 7 of 9

Volume of the balloon when diameter, $D_1 = 0.3$ m

$$V_{balloon} = \frac{4}{3}\pi r_1^3$$
$$= \frac{4}{3}\pi\left(\frac{D_1}{2}\right)^3$$
$$= \frac{\pi D_1^3}{6}$$
$$= \frac{\pi \times 0.3^3}{6}$$
$$= 0.0141\ \text{m}^3$$

Step 8 of 9

Then buoyant force
$$F_{B_1} = (1000)(9.81)(0.0141)$$
$$= 138.32\ \text{N}$$

Volume of the balloon when diameter when the diameter, $D_2 = 0.075$ m

$$V_{balloon} = \frac{\pi D_2^3}{6}$$
$$= \frac{\pi \times 0.075^3}{6}$$
$$= 2.2 \times 10^{-4}\ \text{m}^3$$

Then buoyant force
$$F_{B_2} = (1000)(9.81)\left(2.2 \times 10^{-4}\right)$$
$$= 2.1582\ \text{N}$$

Step 9 of 9

The percentage decrease in cable tension

$$\%\,\text{decrease} = \frac{F_{B_1} - F_{B_2}}{F_{B_1}}$$
$$= \frac{138.32 - 2.1582}{138.32}$$
$$= 0.9843$$
$$= 98.43\%$$

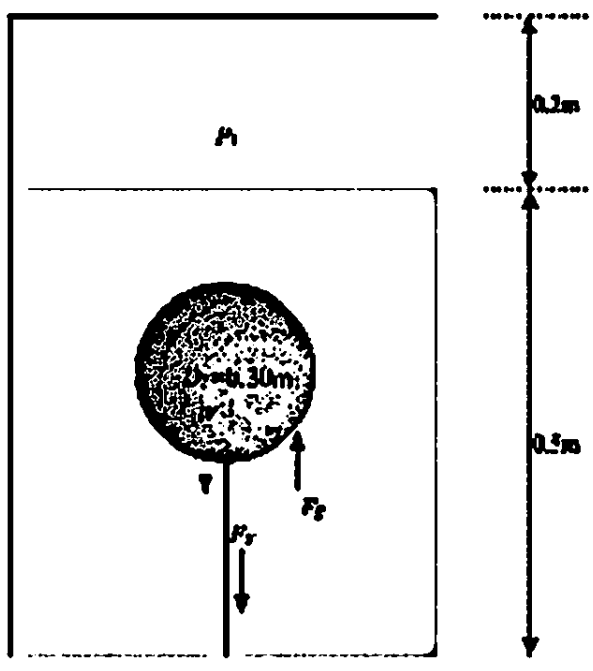
Therefore the cable tension decreases by a margin of $\boxed{98.43\%}$.

Question: 11.049P

Reconsider Prob. 11-48. Using EES (or other) software, investigate the effect of air pressure above water on the cable force. Let this pressure vary from 0.5 MPa to 15 MPa. Plot the cable force versus the air pressure.

Step 1 of 6

An elastic air balloon is attached to the base of a container is as shown below:



Step 2 of 6

The given problem can be solved using EES software by following the below steps.

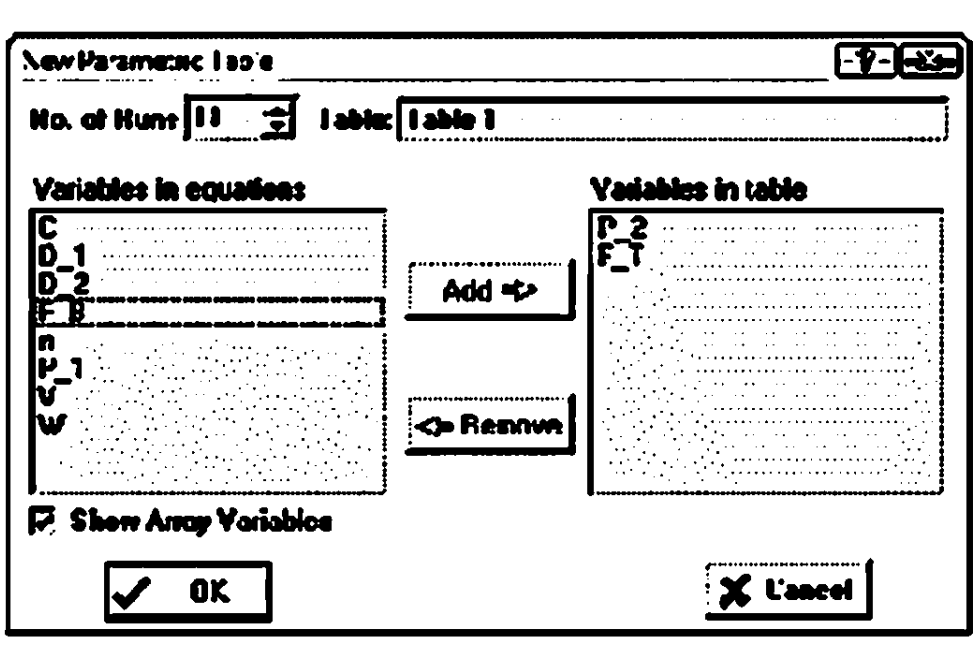
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"Given Data"
D_1=0.30[m]                          "Diameter of ballon"
"P_2 is varying from 0.5[MPa] to 15[MPa]"
W=0                                  "Weight of the ballon is negligible"
P_1=0.1[MPa]
"P=CD^n"
n=-2

P_1=C*D_1^n                          "At pressure P_1"
P_2=C*(D_2^n)                        "At pressure P_2"
V=abs(pi*(D_2)^3/6)                  "Volume of the ballon at diameter D_2"
F_B=1000*g#*V                        "Buoyant force acting on ballon"
F_T=F_B-W                            "Tesnion in the cable"
```

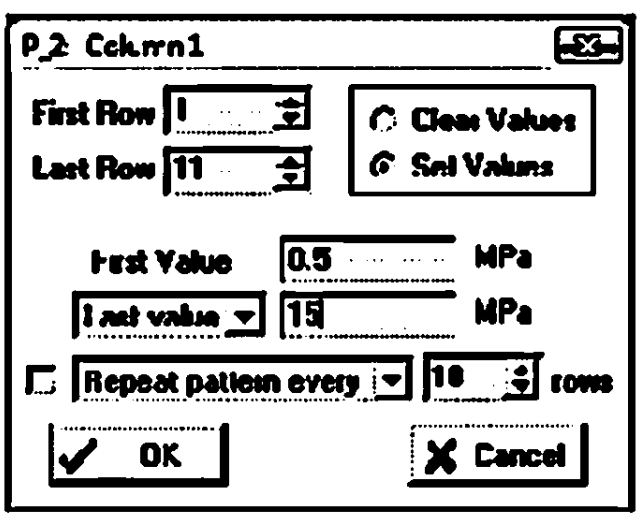
Step 3 of 6

Now to solve the equation for a range of pressure values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "P_2" and "F_T" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 4 of 6

To enter the values in the "P_2" column click on the drop tab in "P_2" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 5 of 6

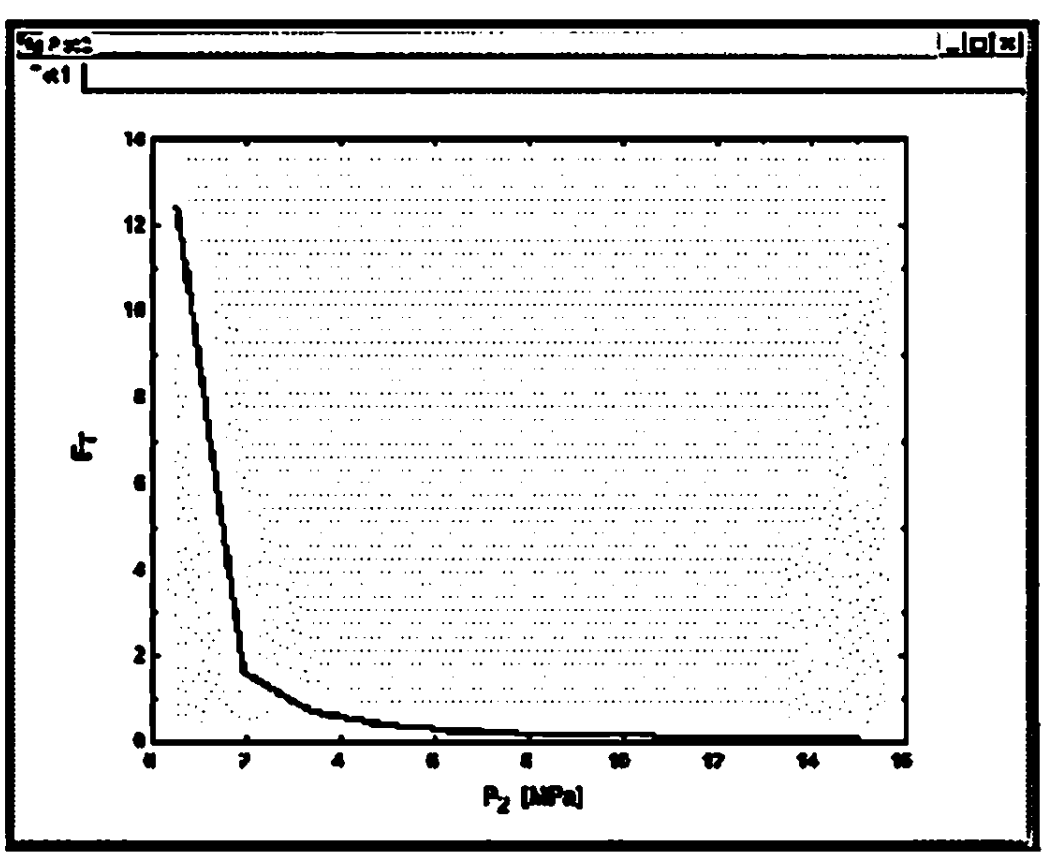
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	1 P_2 [MPa]	2 F_T
Run 1	0.5	12.1
Run 2	1.95	1.61
Run 3	3.4	0.6993
Run 4	4.85	0.4105
Run 5	6.3	0.2773
Run 6	7.75	0.2032
Run 7	9.2	0.1571
Run 8	10.65	0.1261
Run 9	12.1	0.1042
Run 10	13.55	0.0879
Run 11	15	0.07547

Step 6 of 6

Inorder to plot the cable force versus the air pressure, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as "P_2" and Y-axis as "F_T" and press 'OK'
The Graph will be plotted as shown below:



Question: 11.050P

The average density of icebergs is about 917 kg/m3. (*a*) Determine the percentage of the total volume of an iceberg submerged in seawater of density 1042 kg/m3. (*b*) Although icebergs are mostly submerged, they are observed to turn over. Explain how this can happen. (*Hint:*Consider the temperatures of icebergs and seawater.)

Step 1 of 5

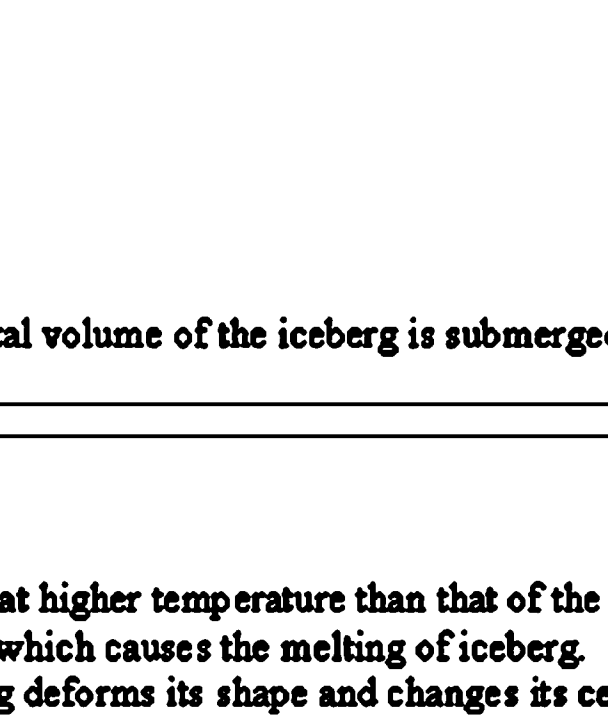
Given data:

The average density of seawater, $\rho_w = 1042\ kg/m^3$

The density of ice berg, $\rho_i = 917\ kg/m^3$

Step 2 of 5

Consider the iceberg in sea water as shown in figure:



Step 3 of 5

a)

Assume W_i is the weight of the iceberg and F_B is the buoyant force which keeps it afloat.

Then by force equilibrium,

$W_i = F_B$

$\rho_i g V_i = \rho_w g V_{sub}$

Where V_i is the volume of the iceberg and

V_{sub} Is the volume of the submerged portion of the iceberg

Step 4 of 5

Then from above equation

$\rho_i g V_i = \rho_w g V_{sub}$

$$\frac{V_i}{V_{sub}} = \frac{\rho_i}{\rho_w}$$

$$= \frac{917}{1042}$$

$$= 0.880$$

$$= 88\%$$

$V_i = 0.88(V_{sub})$

Therefore $\boxed{88\%}$ of the total volume of the iceberg is submerged in sea water.

Step 5 of 5

b)

We know the sea water is at higher temperature than that of the iceberg. This causes heat transfer from water to ice which causes the melting of iceberg.

The melting of the ice berg deforms its shape and changes its centre of mass which causes turning over of the iceberg.

CHAPTER 12
SOLUTIONS

Question: 12.001P

What is streamwise acceleration? How does it differ from normal acceleration? Can a fluid particle accelerate in steady How?

Step 1 of 2

Streamwise acceleration:

The acceleration of a fluid particle along a stream line in a flow can be termed as Streamwise acceleration.

Step 2 of 2

Difference between Streamwise and Normal acceleration

In Streamwise acceleration the fluid particles accelerates along the stream line. But in normal acceleration the fluid particles accelerates in a direction normal to the streamline which is usually created due to change in direction of flow.

Question: 12.002P

Express the Bernoulli equation in three different ways using (*a*) energies, (*b*) pressures, and (*c*) heads.

Step 1 of 3

(a) Bernoulli's equation in terms of energies for unit mass

$$\boxed{\frac{P}{\rho}+\frac{V^2}{2}+gz = \text{constant}}$$

Where,

The flow energy of the fluid is $\frac{P}{\rho}$

The kinetic energy of the fluid is $\frac{V^2}{2}$

The potential energy of the fluid is gz

Step 2 of 3

(b) **Bernoulli's equation in terms of pressures**

We have

$$\frac{P}{\rho}+\frac{V^2}{2}+gz = \text{constant}$$

Divide the equation with density (ρ)

Then we get the Bernoulli's equation in terms of pressures for unit mass as

$$\boxed{P+\rho\frac{V^2}{2}+\rho gz = \text{constant}}$$

Where

The static pressure head is P

The dynamic pressure is $\rho\frac{V^2}{2}$

The hydrostatic pressure is ρgz

Step 3 of 3

(c) **Bernoulli's equation in terms of heads**

We have

$$\frac{P}{\rho}+\frac{V^2}{2}+gz = \text{constant}$$

Divide the equation with acceleration due to gravity, (g)

$$\boxed{\frac{P}{\rho g}+\frac{V^2}{2g}+z = \text{constant}}$$

Where

The pressure head is $\frac{P}{\rho g}$

The velocity head is $\frac{V^2}{2g}$

The elevation head is z

Question: 12.003P

What are the three major assumptions used in the derivation of the Bernoulli equation?

Step 1 of 1

The three major assumptions for the derivation of Bernoulli's Equation

(1) The fluid flow is steady.

(2) The fluid flow is incompressible.

(3) The frictional forces are negligible.

Question: 12.004P

Define static, dynamic, and hydrostatic pressure. Under what conditions is their sum constant for a flow stream?

Step 1 of 2

Static pressure (P):

The static pressure is the actual thermodynamic pressure of the fluid. This pressure does not take into account any dynamic effects and is the same pressure used in property tables.

Dynamic pressure:

The dynamic pressure $\left(\rho V^2/2\right)$ of the fluid can be defined as the pressure developed when the fluid motion is brought to a stop in an isentropic manner

Hydrostatic pressure:

The pressure exerted by a liquid column of height h under the effect of gravity over specified area can be termed as hydrostatic pressure.

It is expressed as

$P_h = \rho g h$

Or

$P_h = \rho g z$

Where z is the elevation

Step 2 of 2

We have the Bernoulli's equation for pressures in term of unit mass,

$$P + \rho\frac{V^2}{2} + \rho g z = \text{constant}$$

The equation represents the sum of static, dynamic and hydrostatic pressures are constant. This is possible when the following conditions are satisfied

(1) The fluid flow is steady.

(2) The fluid flow is incompressible.

(3) The frictional forces are negligible.

Therefore static, dynamic and hydrostatic pressures have been defined and the conditions for the sum of static, dynamic and hydrostatic pressures to be constant have been specified.

Question: 12.005P

What is stagnation pressure? Explain how it can be measured.

Step 1 of 2

Stagnation pressure:
The pressure developed at a point where the fluid has been brought to rest in an isentropic manner can be termed as stagnation pressure.
The stagnation pressure is expressed as

$$P_{stag} = P + \rho\frac{V^2}{2}$$

Where
The static pressure is P

The dynamic pressure is $\rho\frac{V^2}{2}$

Step 2 of 2

The stagnation pressure can be measured by using a Pitot tube. The Pitot tube is a device in the form of a small tube with one end open which is pointed directly into the fluid flow. The moving fluid is brought to rest or stagnates due to the absence of an outlet. The pressure developed can be measured by using this instrument is the stagnation pressure.

Question: 12.006P

Define pressure head, velocity head, and elevation head for a fluid stream and express them for a fluid stream whose pressure is P, velocity is V, and elevation is z.

Step 1 of 3

Pressure head:

Pressure head can be defined as the height of a fluid column that produces a static pressure.

It is represented as $\frac{P}{\rho g}$

Where

The pressure is P

The density of fluid is ρ

The acceleration due to gravity is g

Step 2 of 3

Velocity head:

The elevation that is required for a fluid to attain a velocity V in a frictionless fall can be termed as the velocity head of that fluid.

It is represented as $\frac{V^2}{2g}$

Where

The velocity to be attained is V

Step 3 of 3

Elevation head:

The height at which the stored which also represents the potential energy of the fluid can be termed as elevation head

The elevation head is represented as z.

Consider a fluid under a pressure P, velocity V and elevation z

The pressure head, velocity head and elevation head can be combined to form the Bernoulli equation expressed in terms of head as

$$\frac{P}{\rho g} + \frac{V^2}{2g} + z = \text{constant}$$

Question: 12.007P

What is the hydraulic grade line? How does it differ from the energy grade line? Under what conditions do both lines coincide with the free surface of a liquid?

Step 1 of 3

Hydraulic grade line (HGL):

The line that represents the sum of static pressure head and elevation head can be termed as Hydraulic grade line.

Hydraulic grade line can be represented as

$$\frac{P}{\rho g} + z$$

Where

The static pressure head is $\frac{P}{\rho g}$

The elevation head is z

Step 2 of 3

Energy grade line (EGL):

The energy grade line represents the sum of static pressure, dynamic pressure and elevation pressure heads.

It is represented as

$$\frac{P}{\rho g} + \frac{V^2}{2g} + z$$

Where

The static pressure head is P

The dynamic pressure is $\rho\frac{V^2}{2}$

The elevation head is z

Step 3 of 3

Difference between Hydraulic grade and Energy grade line

The difference between Hydraulic grade and Energy grade line is the presence of the dynamic pressure head term $\left(\frac{V^2}{2g}\right)$ in the Energy grade line.

The energy grade line coincides with the hydraulic grade line when the velocity is zero and the static (gage) pressure is zero.

Then the elevation of the free surface z will represent both HGL and EGL.

Examples: Stationary lakes and reservoirs.

Question: 12.008P

How is the location of the hydraulic grade line determined for open-channel flow? How is it determined at the outlet of a pipe discharging to the atmosphere?

Step 1 of 3

The Hydraulic grade line represents the sum of the static pressure and the elevation head and it is denoted with letter HGL

$$\text{HGL} = \frac{P}{\rho g} + z$$

where P is the pressure at one point

Density of the fluid is ρ

Acceleration due to gravity is g

Elevation head is z

Step 2 of 3

In case of open channel flow the atmospheric pressure acts at the free surface of the liquid. Then the hydraulic grade line, HGL coincides with the free surface of the liquid.

Step 3 of 3

When fluid is discharging from the pipe to the atmosphere, atmospheric pressure acts at the exit of the pipe then the hydraulic grade line coincides with the pipe outlet.

Question: 12.009P

In a certain application, a siphon must go over a high wall. Can water or oil with a specific gravity of 0.8 go over a higher wall? Why?

Step 1 of 5

Given data:

The specific gravity of oil is $SG_O = 0.8$.

The oil and water is to be siphoned over a high wall.

Step 2 of 5

Assumption:

The siphoning is to be carried out under the normal atmospheric pressure of $P_{atm} = 1.013$ bar.

The water used is of density $\rho_w = 1000\ kg/m^3$.

Step 3 of 5

Consider water is being used,

We have

$P_{atm} = \rho_w g h_w$

The maximum height to which water can be siphoned when $P_{atm} = 1.013$ bar

$$h_w = \frac{P_{atm}}{\rho_w g}$$

$$= \frac{1.013 \times 10^5}{(1000)(9.81)}$$

$$= 10.32\ m$$

We can conclude that if the height of the high wall is equal to or below 10.32 m , the water can be siphoned over it.

Step 4 of 5

Consider oil is being used,

The density of oil,

$$\rho_O = SG_O(\rho_w)$$

$$= (0.8)(1000)$$

$$= 800\ kg/m^3$$

Step 5 of 5

We have

$P_{atm} = \rho_O g h_O$

The maximum height to which oil can be siphoned when $P_{atm} = 1.013$ bar

$$h_O = \frac{P_{atm}}{\rho_O g}$$

$$= \frac{1.013 \times 10^5}{(800)(9.81)}$$

$$= 12.9\ m$$

We can conclude that if the height of the high wall is equal to or below 12.9 m , the oil can be siphoned over it.

Question: 12.010P

Explain how and why a siphon works. Someone proposes siphoning cold water over a 7-m-high wall. Is this feasible? Explain.

Step 1 of 3

The difference in elevation and the corresponding pressure difference between the inlet and exit of a tube cause the phenomenon of siphoning.

The exit of a siphon tube is positioned at an elevation below that of the free surface of the liquid, this elevation difference causes the flow of liquid inside the tube.

Step 2 of 3

Write the equation for the atmospheric pressure.

$$P_{atm} = \rho_{water} g h_{water}$$

Here, P_{atm} is the atmospheric pressure, ρ_{water} is the density of water, g is the acceleration due to the gravity, and h_{water} is the height to which water can be siphoned.

Substitute $1.013 \times 10^5 \text{ N/m}^2$ for P_{atm}, 9.81 m/s^2 for g, and 1000 kg/m^3 for ρ_{water}.

$$1.013 \times 10^5 = 1000 \times 9.81 \times h_{water}$$

$$h_{water} = \frac{1.013 \times 10^5}{1000 \times 9.81}$$
$$= 10.326 \text{ m}$$

So, the maximum height to which water can be siphoned when it is exposed to the atmospheric pressure is 10.326 m.

Step 3 of 3

As the proposed height for siphoning cold water is 7 m, which is less than 10.326 m.

Thus, it is **feasible** to siphon cold water over a 7 m high wall.

Question: 12.011P pg 1

A glass manometer with oil as the working fluid is connected to an air duct as shown in Fig. P12—J1C. Will the oil levels in the manometer be as in Fig. P12—1 IC*a* or *b*Explain. What would your response be if the How direction is reversed.

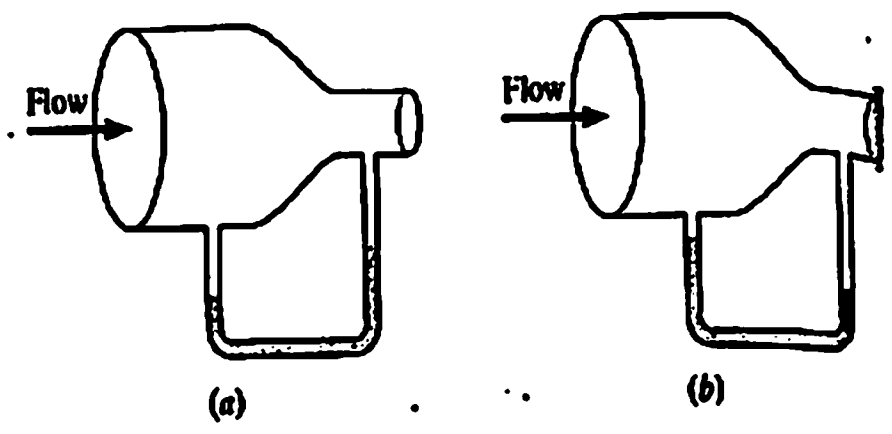
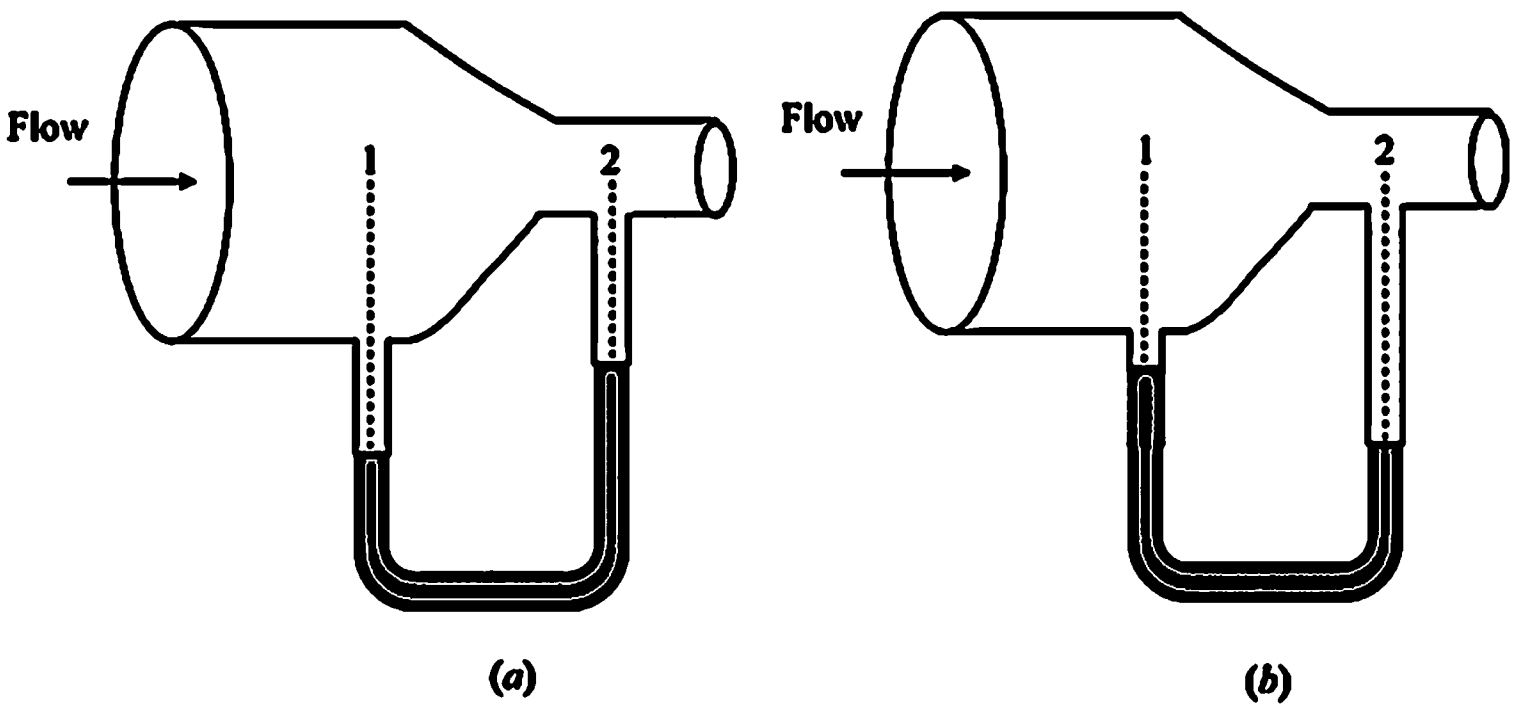


FIGURE P12-11C

Step 1 of 8

Draw the diagram for the given configurations.



Step 2 of 8

Assume the flow is steady, incompressible, and inviscid.

Write the equation for mass flow rate.

$$\dot{m} = \rho V A_c$$

Here, $\dot{m}$ is the mass flow rate, V is the flow velocity, A_c is the area of cross-section, and ρ is the density of the fluid.

As the flow is assumed to be constant, $\dot{m}$ and ρ are constant.

$$V = \frac{\dot{m}}{\rho A_c}$$

$$V \propto \frac{1}{A_c}$$

Therefore, the velocity varies inversely with the area of cross section.

Step 3 of 8

From the above figure, as the flow is from left to right, the area of cross section decreases from section 1 to section 2.

$$A_1 > A_2$$

Here, A_1 is the area at section 1 and A_2 is the area at section 2.

As the velocities are inversely proportional to area of cross section,

$$V_1 < V_2$$

Here, V_1 is the velocity at section 1 and V_2 is the velocity at section 2.

Step 4 of 8

Apply the Bernoulli's equation at section 1 and section 2.

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \text{..... (1)}$$

Here, P_1 is the pressure at section 1, P_2 is the pressure at section 2, V_1 is the velocity at section 1, V_2 is the velocity at the section 2, z_1 is the elevation at section 1, z_2 is the elevation at section 2, ρ is the density of the fluid, and g is the acceleration due to the gravity.

From the figure, section 1 and section 2 are at same levels of elevation.

$$z_1 = z_2$$

Substitute z_2 for z_1 in equation (1).

$$P_1 = P_2 + \frac{1}{2}\rho\left(V_2^2 - V_1^2\right)$$

Step 5 of 8

Find the pressure at the section 2.

$$\frac{P_2}{\rho g} = \frac{P_1}{\rho g} + \frac{V_1^2}{2g} - \frac{V_2^2}{2g}$$

$$P_2 = P_1 + \frac{1}{2}\rho\left(V_1^2 - V_2^2\right)$$

As $V_1 < V_2$, V_1^2 is less than V_2^2.

Hence, the term $V_1^2 - V_2^2$ is negative and $V_2^2 - V_1^2$ is positive.

$$P_2 = P_1 - \frac{1}{2}\rho\left(V_2^2 - V_1^2\right)$$

So, the pressure at section 2 is less than the pressure at section 1.

$$P_2 = P_1 - \frac{1}{2}\rho\left(V_2^2 - V_1^2\right)$$

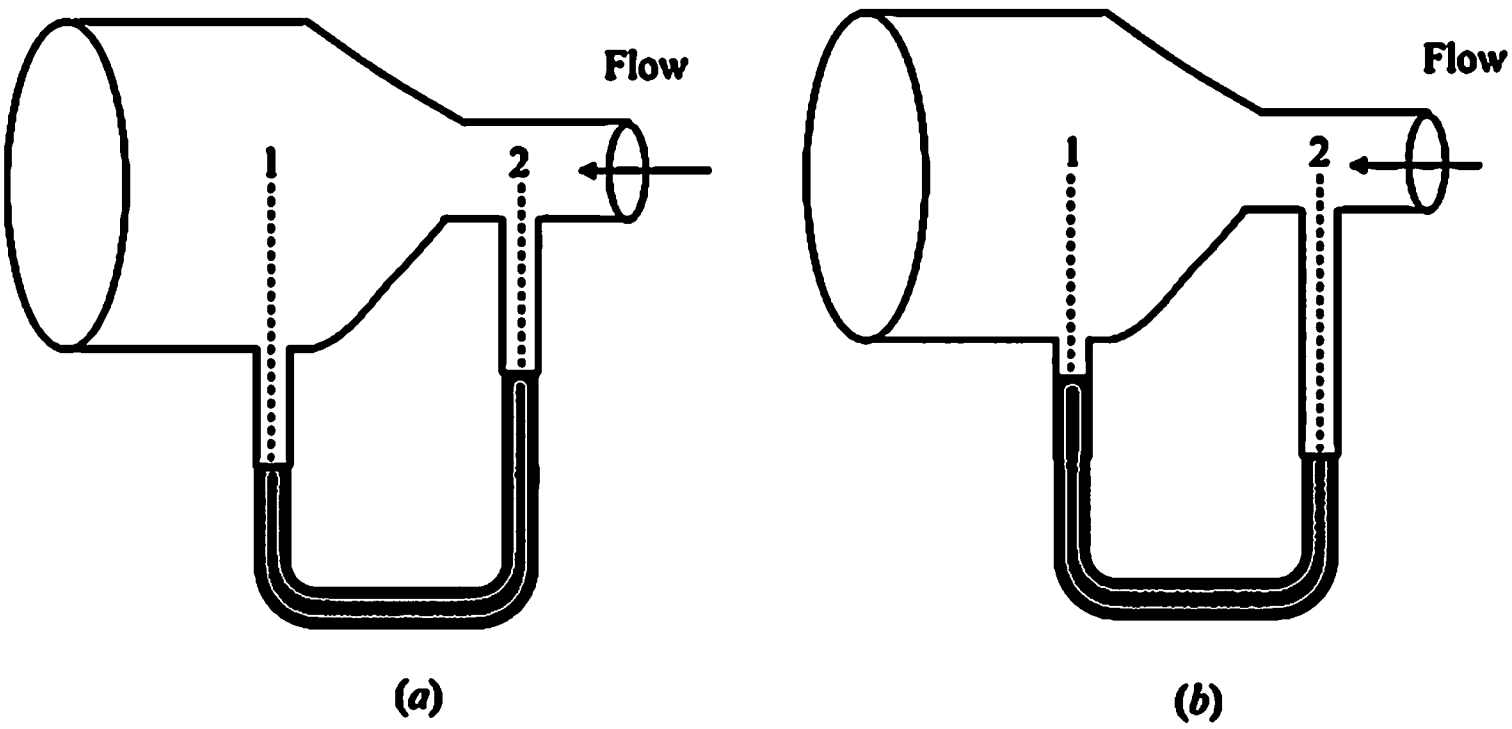
So, the pressure at section 2 is less than the pressure at section 1.

As the pressure in section 2 is less than the pressure in section 1, the fluid will rise in section 2.

Therefore, the correct configuration is figure $\boxed{(a)}$.

Step 6 of 8

Now consider the flow direction to be reversed.



(a) (b)

Step 7 of 8

Here, in the reversed flow configuration the area of cross section increases from section 2 to section 1.

$$A_2 < A_1$$

As the velocities are inversely proportional to area of cross section,

$$V_2 > V_1$$

Step 8 of 8

Apply the Bernoulli's equation at section 1 and section 2.

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

From the figure, section 1 and section 2 are at same levels of elevation.

$$z_1 = z_2$$

Substitute z_2 for z_1 .

$$P_1 = P_2 + \frac{1}{2}\rho\left(V_2^2 - V_1^2\right)$$

As the term $V_2^2 - V_1^2$ is positive, the pressure at 1 is greater than pressure at 2.

So, the fluid at section 1 will depress.

Therefore, the correct configuration for reversed direction of the flow is $\boxed{(a)}$.

Question: 12.012P

The velocity of a fluid flowing in a pipe is to be measured by two different Pito-type mercury manometers shown in Fig. P12-12C. Would you expect both manometers to predict the same velocity for flowing water? If not, which would be more accurate? Explain. What would your response be if air were flowing in the pipe instead of water?

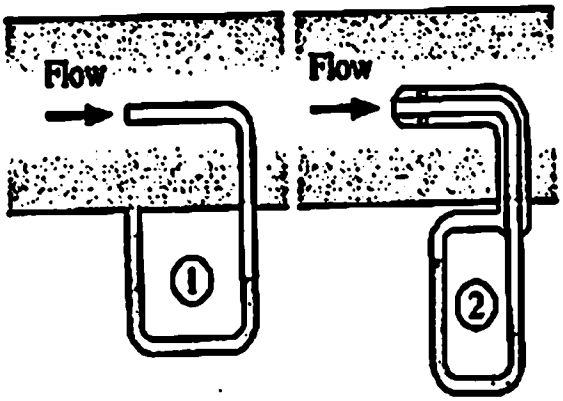


FIGURE P12–12C

Step 1 of 1

Pitot tubes are used to indicate the fluid flow velocity by measuring the difference between the static and dynamic pressures in fluids. From the two different Pitot-type mercury manometers shown, arrangement 2 can be considered to be more accurate because it measures both stagnation pressure and static pressure at same location along the pipe centerline. Whereas in arrangement 1, the stagnation pressure is measured at pipe centerline, along with a static pressure tap that measures static pressure at the bottom of the pipe.

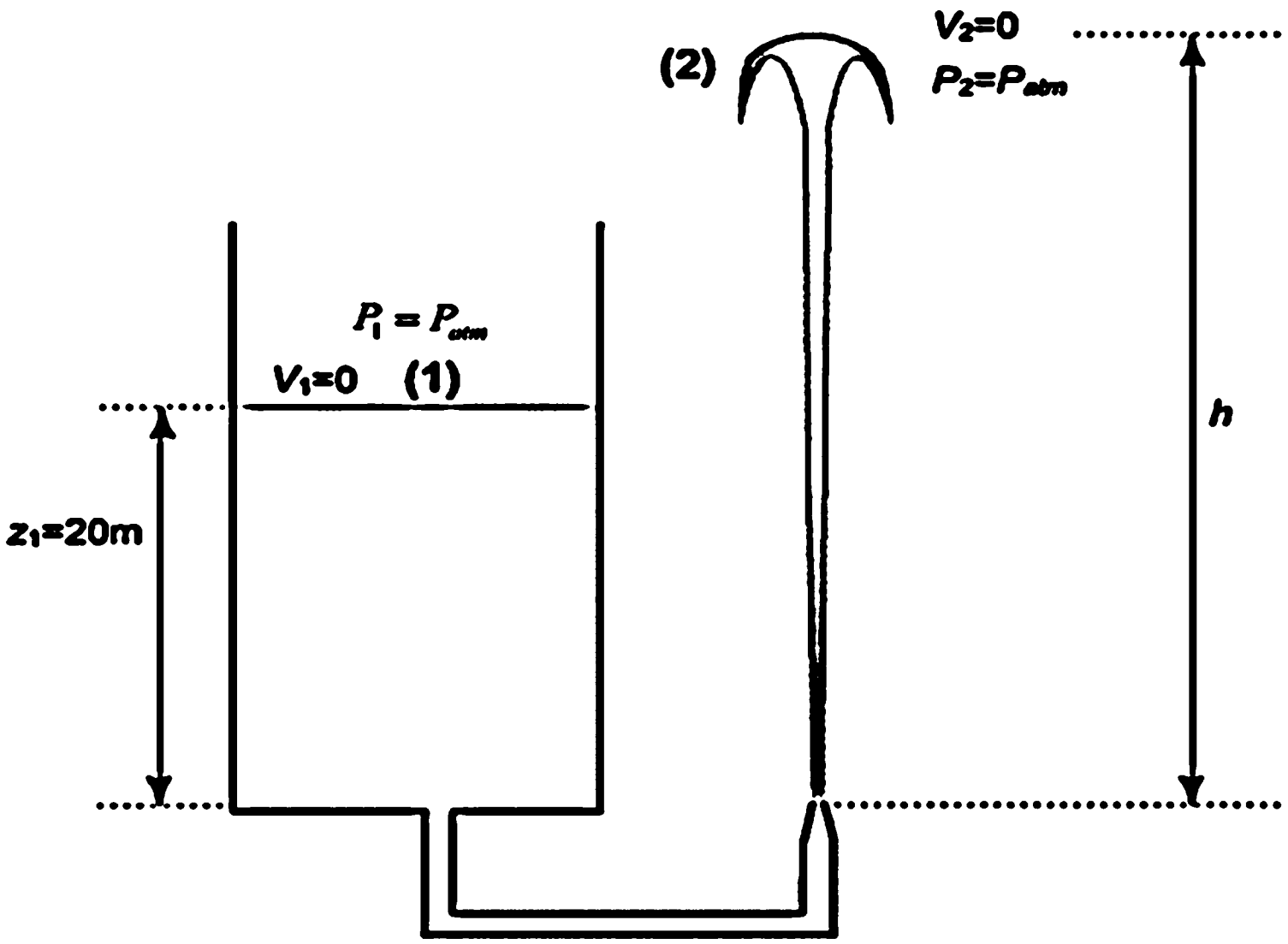
Static pressure in a pipe varies with elevation across the pipe cross section which is similar to hydrostatics (change of pressure with respect to height). Hence we can say that even arrangement is 1 is accurate as the elevation difference between the Pitot probe and static pressure tap is compensated by the change in hydrostatic pressure. Hence there will be no change in the analysis when the water is replaced by air as elevation changes are not important.

Question: 12.013P

The water level of a tank on a building roof is 20 m above the ground. A hose leads from the tank bottom to the ground. The end of the hose has a nozzle, which is pointed straight up. What is the maximum height to which the water could rise? What factors would reduce this height?

Step 1 of 3

Draw the schematic diagram of the tank.



Step 2 of 3

Apply Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

Substitute P_{atm} for P_1, 0 for V_1, $20\,\text{m}$ for z_1, P_{atm} for P_2, 0 for V_2 and 0 for z_2.

$$\frac{P_{atm}}{\rho g}+\frac{0}{2g}+20=\frac{P_{atm}}{\rho g}+\frac{0}{2g}+z_2$$

$$z_2=20\ \text{m}$$

Therefore, the maximum height to which water can rise from the nozzle is $\boxed{z_2=20\ \text{m}}$.

Step 3 of 3

The following are the factors that can reduce the height of the jet:

- Frictional effects being present in the system.
- Lower pressure at the water surface in the tank.

Question: 12.014P

A student siphons water over a 8.5-m-high wall at :sea level. She then climbs to the summit of Mount Shasta (elevation 4390 m, *P*atm = 58.5 kPa) and attempts the same experiment. Comment on her prospects for success.

Step 1 of 3

Given data:

The atmospheric pressure at Mt Shasta, $P_{atm} = 58.5\ \text{kPa}$.

The elevation of Mt Shasta summit, $z = 4390\ \text{m}$.

The height over which water has to be siphoned, $h_s = 8.5\ \text{m}$.

Step 2 of 3

The height to which water can be raised under a pressure of $P_{atm} = 58.5\ \text{kPa}$,

$$h = \frac{P_{atm}}{\rho g}$$

$$= \frac{58.5 \times 10^3}{(1000)(9.81)}$$

$$= 5.963\ \text{m}$$

We get $h < h_s$

Therefore, we can conclude that it is not possible for the person to siphon water over a wall of height $h_s = 8.5$ m at the summit of Mt Shasta.

Step 3 of 3

The pressure required to siphon water over a wall of height $h_s = 8.5$ m at the summit of Mt Shasta,

$$P = \rho g h_s$$

$$= (1000)(9.81)(8.5)$$

$$= 83.385 \times 10^3 \text{Pa}$$

$$= 83.385 \text{kPa}$$

Therefore, a pressure of $\boxed{P = 83.4\ \text{kPa}}$ is required for the siphoning process to occur.

Question: 12.015P

In a hydroelectric power plant, water enters the turbine nozzles at 800 kPa absolute with a low velocity. If the nozzle outlets are exposed to atmospheric pressure of 100 kPa, determine the maximum velocity to which water can be accelerated by the nozzles before striking the turbine blades.

Step 1 of 3

Given data:

Inlet pressure of the nozzle, $P_1 = 800$ kPa

$= 800000$ kPa.

Outlet pressure of the nozzle, $P_2 = 100$ kPa

$= 100000$ kPa.

Step 2 of 3

Let us apply the Bernoulli's equation between the Inlet and outlet

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \qquad \ldots\ldots (1)$$

where

The elevation, $z_1 = z_2 = 0$.

Inlet velocity, $V_1 = 0$.

Density water, $\rho = 1000$ kg/m^3.

Step 3 of 3

By substituting the required values in equation (1), we get

$$\frac{P_1}{\rho g} + 0 + 0 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + (0)$$

$$\frac{V_2^2}{2g} = \frac{P_1}{\rho g} - \frac{P_2}{\rho g}$$

$$V_2^2 = \frac{2(P_1 - P_2)}{\rho}$$

$$= \frac{2(800000 - 100000)}{1000}$$

$$= 1400$$

$$V_2 = \sqrt{1400}$$

$$= 37.42 \text{ m/s}$$

Therefore, the maximum velocity to which water can be accelerated by the nozzle is $\boxed{V_2 = 37.42 \text{ m/s}}$.

Question: 12.016P

A Pilot-static probe is used to measure the speed of an aircraft flying at 3000 m. If the differential pressure reading is 3 kPa, determine the speed of the aircraft.

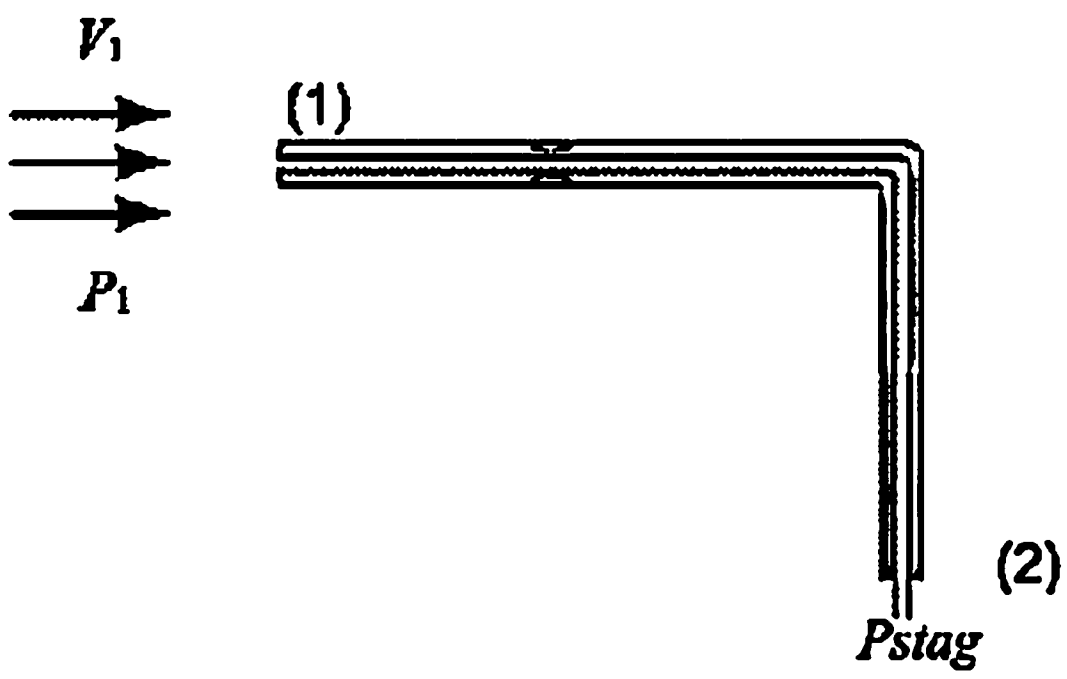
Step 1 of 3

Given data:

Flying height of the aircraft, $h = 3000$ m.

The differential pressure reading, $P_{diff} = 3$ kPa.

Let us draw the figure.



Step 2 of 3

We have,

$$P_{diff} = P_{stag} - P_1$$
$$= 3 \text{ kPa}$$

Let us apply the Bernoulli's equation between the two points,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \qquad \ldots\ldots (1)$$

where

The elevation, $z_1 = z_2$

The velocity at stagnation point (2), $V_2 = 0$

Step 3 of 3

By substituting the above values in equation (1), we get

$$\frac{V_1^2}{2g} = \frac{P_{stag}}{\rho g} - \frac{P_1}{\rho g}$$

$$\frac{V_1^2}{2} = \frac{P_{stag} - P_1}{\rho}$$

Substitute the required values,

$$\frac{V_1^2}{2} = \frac{3 \times 10^3}{0.909}$$

$$V_1 = \sqrt{6600.66}$$
$$= 81.24 \text{ m/s}$$

Therefore, the velocity of the aircraft is $\boxed{V_1 = 81.2 \text{ m/s}}$.

Question: 12.017P

While traveling on a dirt road, the bottom of a car kits a sharp rock and a small hole develops at the bottom of its tank. If the height of the gasoline in the tank is 40 cm, determine the initial velocity of the gasoline at the hole. Discuss how the velocity will change with time and how the flow will be affected if the lid of the tank is closed tightly.

Step 1 of 3

Given data:

The height of the gasoline in the tank, $z_1 = 0.4$ m.

Let us apply the Bernoulli's equation between the surface of gasoline and exit,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \qquad \ldots\ldots (1)$$

Where

The elevation at exit

$z_2 = 0$

Step 2 of 3

By substitute the required values in equation (1), we get

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + (0.40\text{ m}) = \frac{P_{atm}}{\rho g} + \frac{V_2^2}{2g} + (0)$$

$$\frac{V_2^2}{2g} = 0.4$$

$$V_2 = \sqrt{2(9.81)(0.4)}$$

$$= 2.8014\text{ m/s}$$

Therefore, the exit velocity of the gasoline is $\boxed{V_2 = 2.80\text{ m/s}}$.

Step 3 of 3

When the lid of the gasoline tank is tightly sealed, the pressure at the surface will be minimum or much less than the atmospheric pressure. We also know that the exit velocity is proportional to the square root of the height of the fluid.

Then as the height decreases and along with the absence or negligible pressure on the surface the exit velocity will be lower than 2.80 m/s.

Question: 12.018P

The drinking water needs of an office are by large water bottles. One end of a 0.25-in-diameter plastic hose is inserted into the bottle placed on a high stand, while the other end with an on/off valve is maintained 2 ft below the bottom of the bottle. If the water level in the bottle is 1.5 ft when it is full, determine how long it will take at the minimum to fill an 8-oz glass (= 0.00835 ft3) (*a*) when the bottle is first opened and (*b*) when the bottle is almost empty. Neglect frictional losses.

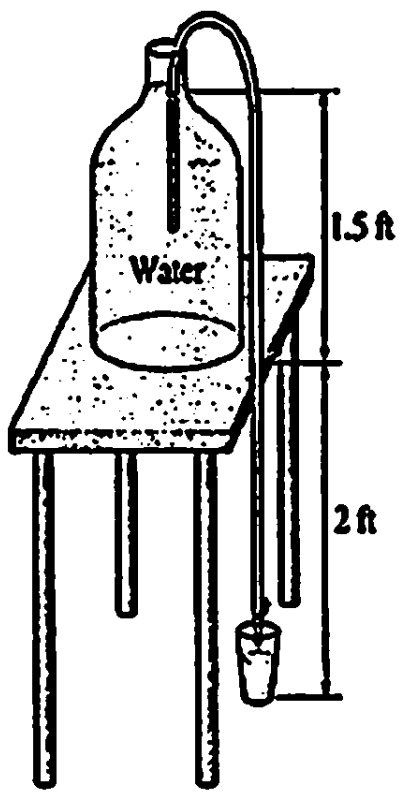


FIGURE P12–18E

Step 1 of 5

Let us apply the Bernoulli's equation,

$$\frac{P_1}{\rho g}+\frac{v_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{v_2^2}{2g}+z_2$$

Where

Inlet open is atmosphere, $p_1 = p_{atm}$

Discharge is also in atmosphere, $p_2 = p_{atm}$

The elevation $z_2 = 0$

The bottle is very large is compare to the tube diameter.

Step 2 of 5

Substituting the required values,

$$\frac{P_{atm}}{\rho g}+0+z_1=\frac{P_{atm}}{\rho g}+\frac{v_2^2}{2g}+0$$

$$v_2^2 = 2gz_1$$

$$v_2 = \sqrt{2gz_1}$$

Step 3 of 5

a)

When the bottle is first opened (Full bottle),

$$z_1 = 3.5\,\text{ft}$$

The flow velocity at outlet,

$$v_2 = \sqrt{2\times 32.2\times 3.5}$$

$$= 15.01\ \text{ft/s}$$

Area of the hose $A=\frac{\pi}{4}d^2$

Here d is the diameter of the hose.

Substitute $d = 0.25$ in. in above equation.

$$A=\frac{\pi}{4}\left(0.25\text{in.}\times\frac{1}{12}\frac{\text{ft}}{\text{in.}}\right)^2$$

$$=0.0003409\,\text{ft}^2$$

Step 4 of 5

Time taken at the minimum to fill the glass,

$$\Delta t=\frac{V}{Q}$$

$$=\frac{V}{Av_2}$$

Here V is the volume of the bottle.

Substitute $V = 0.00835\,\text{ft}^3$, $A = 0.0003409\,\text{ft}^2$ and $v_2 = 15.01\,\text{ft/s}$ in above equation.

$$\Delta t=\frac{0.00835}{0.0003409\times 15.01}$$

$$=1.6318\text{s}$$

Time taken at the minimum to fill the glass is $\boxed{1.63\text{s}}$

Step 5 of 5

b)

When the bottle is almost empty (Full empty)

$$z_1 = 2\,\text{ft}$$

The flow velocity at outlet,

$$v_2 = \sqrt{2\times 32.2\times 2}$$

$$= 11.35\ \text{ft/s}$$

Time taken at the minimum to fill the glass,

$$\Delta t=\frac{V}{Q}$$

$$=\frac{V}{Av_2}$$

Here V is the volume of the bottle.

Substitute $V = 0.00835\,\text{ft}^3$, $A = 0.0003409\,\text{ft}^2$ and $v_2 = 11.35\,\text{ft/s}$ in above equation.

$$\Delta t=\frac{0.00835}{0.0003409\times 11.35}$$

$$=2.158\text{s}$$

Time taken at the minimum to fill the glass is $\boxed{2.16\text{s}}$

Question: 12.019P

A piezometer and a Pilot tube are tapped into a 4-cm-diameter horizontal water pipe, and the height of the water columns are measured to be 26 cm in the piezometer and 35 cm in the Pitot tube (both measured from the top surface of the pipe). Determine the velocity at the center of the pipe.

Step 1 of 7

Given data:

The diameter of the pipe, $D = 4$ cm

$= 0.04$ m.

The height of water column in piezometer, $z_1 = 26$ cm

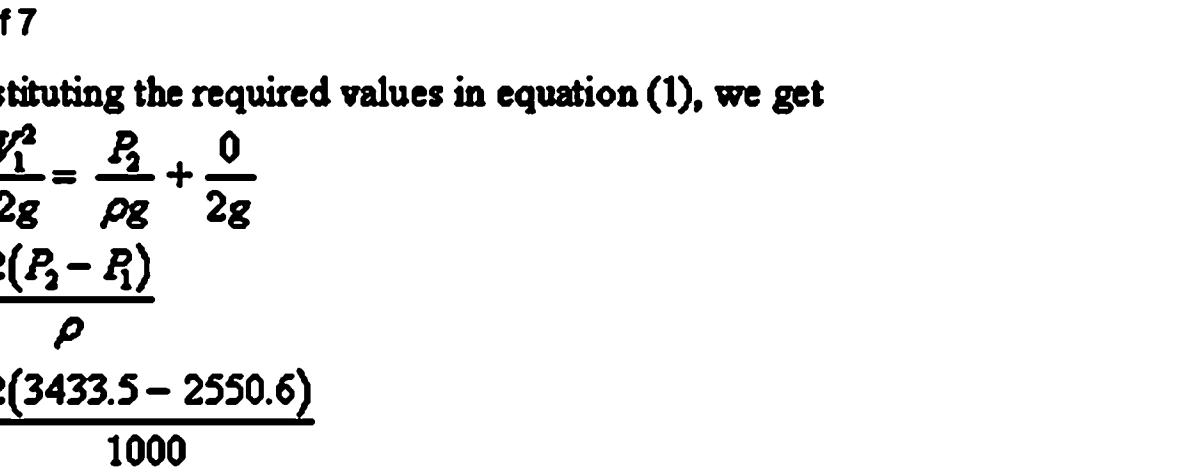
$= 0.26$ m.

The height of water column in Pitot tube, $z_2 = 35$ cm

$= 0.35$ m.

Step 2 of 7

Let us draw the figure.



Step 3 of 7

Consider the piezometer at section (1),

The pressure,

$P_1 = \rho g h_1$

$= (1000)(9.81)(0.26)$

$= 2550.6$ Pa

Step 4 of 7

Consider the Pitot tube at section (2),

$P_2 = \rho g h_2$

$= (1000)(9.81)(0.35)$

$= 3433.5$ Pa

Step 5 of 7

Let us apply the Bernoulli's between the points (1) and (2).

Then,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \qquad \ldots\ldots (1)$$

where

The elevation

$z_1 = z_2$

The velocity at the stagnation point (2)

$V_2 = 0$

Step 6 of 7

By substituting the required values in equation (1), we get

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \frac{0}{2g}$$

$$V_1^2 = \frac{2(P_2 - P_1)}{\rho}$$

$$= \frac{2(3433.5 - 2550.6)}{1000}$$

$= 1.7658$

Step 7 of 7

The flow velocity at the center of the pipe,

$V_1 = \sqrt{1.7658}$

$= 1.32$ m/s

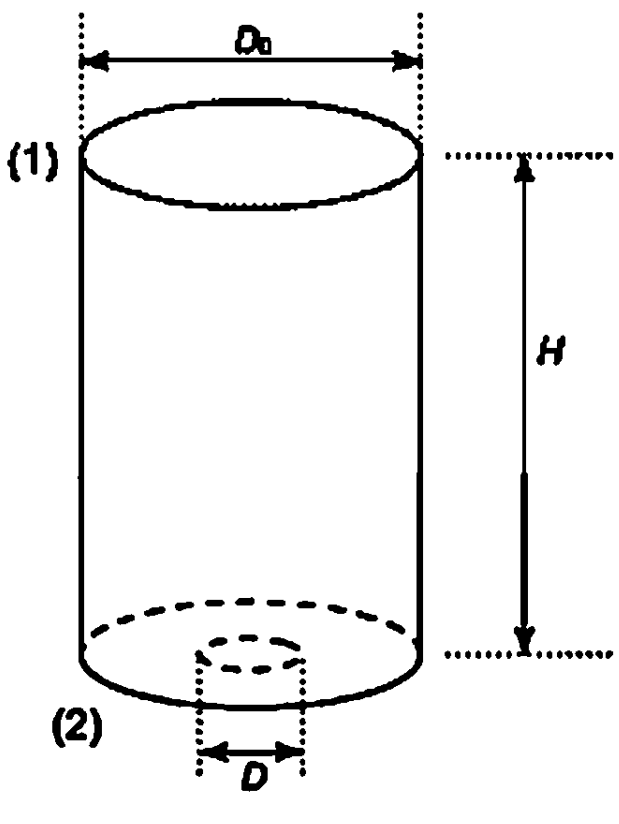
Therefore, the flow velocity is $\boxed{V_1 = 1.32 \text{ m/s}}$.

Question: 12.020P

The diameter of a cylindrical water tank is D_0 and its height is H. The tank is filled with water, which is open to the atmosphere. An orifice of diameter D with a smooth entrance (i.e., negligible losses) is open at the bottom. Develop a relation for the time required for the tank (*a*) to empty halfway and (*b*) to empty completely.

Step 1 of 6

Let us draw the figure



Step 2 of 6

The flow rate,

$$\dot{V} = AV$$

where

A is the area

V is the velocity of fluid

Then flow rate at the orifice,

$$\dot{V} = A_2 V_2$$

$$= \left(\frac{\pi D^2}{4}\right) V_2$$

Step 3 of 6

Let us apply the Bernoulli's equation between (1) and (2).

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

The pressures

$$P_1 = P_2 = P_{atm}$$

P_{atm} is the atmospheric pressure.

The velocity of water at (1),

$$V_1 = 0$$

Then

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_1 = \frac{P_{atm}}{\rho g} + \frac{V_2^2}{2g} + 0$$

$$V_2 = \sqrt{2gz_1}$$

Step 4 of 6

Substituting $V_2 = \sqrt{2gz_1}$ in $\dot{V} = \left(\frac{\pi D^2}{4}\right) V_2$ then

$$\dot{V} = \left(\frac{\pi D^2}{4}\right)\sqrt{2gz_1}$$

We have $\dot{V} = \frac{V}{\Delta t}$

Where

V is the volume of water removed

Δt is the time period.

The volume, $V = \pi \frac{D_0^2}{4} z$

where z is the height of water removed from the cylindrical tank.

Then we can write,

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gz_1} = \frac{V}{\Delta t}$$

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gz_1} = \pi \frac{D_0^2}{4\Delta t} z$$

Step 5 of 6

a)

The water level is half way

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gz_1} = \pi \frac{D_0^2}{4\Delta t} z$$

Where

$$z_1 = H \text{ and } z = \frac{H}{2}$$

Then

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gH} = \pi \frac{D_0^2}{4\Delta t} \frac{H}{2}$$

$$\left(\frac{D_0^2}{D^2}\right)\frac{H}{\sqrt{8gH}} = \Delta t$$

Therefore, the time required is $\boxed{\Delta t = \left(\frac{D_0^2}{D^2}\right)\frac{H}{\sqrt{8gH}}}$.

Step 6 of 6

b)

The tank is completely empty.

We have

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gz_1} = \pi \frac{D_0^2}{4\Delta t} z$$

Where

$$z_1 = H \text{ and } z = H$$

Then

$$\left(\frac{\pi D^2}{4}\right)\sqrt{2gH} = \pi \frac{D_0^2}{4\Delta t} H$$

$$\left(\frac{D_0^2}{D^2}\right)\frac{H}{\sqrt{2gH}} = \Delta t$$

Therefore, the time required is $\boxed{\Delta t = \left(\frac{D_0^2}{D^2}\right)\frac{H}{\sqrt{2gH}}}$.

Question: 12.021P

A siphon pumps water from a large reservoir to a lower tank that is initially empty. The tank also has a rounded orifice 20 ft below the reservoir surface where the water leaves the tank. Both the siphon and the orifice diameters are 2 in. Ignoring frictional losses, determine to what height the water will rise in the tank at equilibrium.

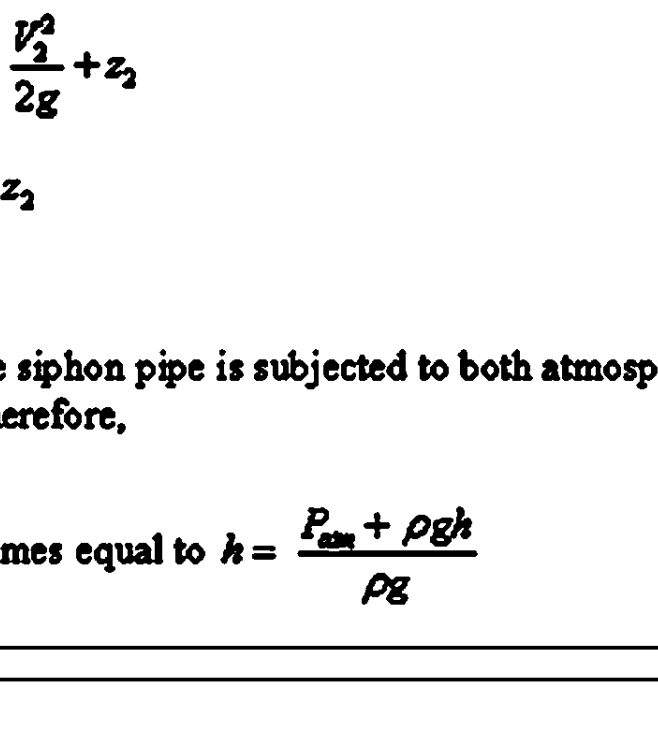
Step 1 of 9

Given data:

Diameters of siphon and orifice, $d = 2\text{in.}$

Step 2 of 9

Schematic of diagram of siphon system:



Step 3 of 9

The pressure at point (1) and (3) is equal to the atmospheric pressure, P_{atm}.

The velocity at point (1) and (3) is negligible or equal to zero.

Apply the Bernoulli's equation between reservoir surface (1) and siphon exit (2).

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

Let us set the orifice point as the reference level.

From the figure

$z_2 = z_4$

Step 4 of 9

Then

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$\frac{P_{atm}}{\rho g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

Where

Since pressure exit of the siphon pipe is subjected to both atmospheric pressure and pressure of the water. Therefore,

$P_2 = P_{atm} + \rho g h$

Then pressure head becomes equal to $h = \frac{P_{atm} + \rho g h}{\rho g}$

Step 5 of 9

Then, simplifying the above expression, we get

$$\frac{P_{atm}}{\rho g} + z_1 = \frac{P_{atm} + \rho g h}{\rho g} + \frac{V_2^2}{2g} + 0$$

$$0 + z_1 = h + \frac{V_2^2}{2g} + 0$$

$$\frac{V_2^2}{2g} = z_1 - h$$

$$V_2 = \sqrt{2g(z_1 - h)}$$

Step 6 of 9

Applying Bernoulli's equation between tank surface (3) and orifice exit (4).

$$\frac{P_3}{\rho g} + \frac{V_3^2}{2g} + z_3 = \frac{P_4}{\rho g} + \frac{V_4^2}{2g} + z_4$$

Since atmospheric pressure acting at surface (3), therefore

$P_3 = P_{atm}$

And set the orifice point as the reference level.

So from the figure

$z_3 = h$

Then, substituting the values in above expression, we get

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_3 = \frac{P_4}{\rho g} + \frac{V_4^2}{2g} + 0$$

$$\frac{P_{atm}}{\rho g} + h = \frac{P_4}{\rho g} + \frac{V_4^2}{2g} + 0$$

Where

$P_4 = P_{atm} + \rho g h$

Step 7 of 9

Then, velocity at exit of orifice is

$$\frac{P_{atm}}{\rho g} + h = \frac{P_{atm} + \rho g h}{\rho g} + \frac{V_4^2}{2g} + 0$$

$$h = \frac{V_4^2}{2g}$$

$$V_4 = \sqrt{2gh}$$

Step 8 of 9

We know that when the diameters of the orifice and the siphon tube are equal to each other, their flow rates will also be equal to each other.

Then

$\dot{V}_2 = \dot{V}_4$

$A_{siphon} V_2 = A_{orifice} V_4$

$V_2 = V_4$

Substitute the known values in above expression, we get

$$\sqrt{2g(z_1 - h)} = \sqrt{2gh}$$

$$2g(z_1 - h) = 2gh$$

$$h = \frac{z_1}{2}$$

Step 9 of 9

The height to which water rises in the tank

$$h = \frac{z_1}{2}$$

$$= \frac{20}{2}$$

$$= 10\text{ft}$$

Therefore, height of the water rises in bottom tank is $\boxed{h = 10\text{ft}}$.

Question: 12.022P pg 1

Water enters a tank of diameter *DT*steadily at a mass flow rate of *m*in An orifice at the bottom with diameter *D*oallows water to escape. The orifice has a rounded entrance, so the frictional losses are negligible. If the tank is initially empty, (*a*) determine the maximum height that the water will reach in the tank and (*b*) obtain a relation for water height *z* as a function of time.

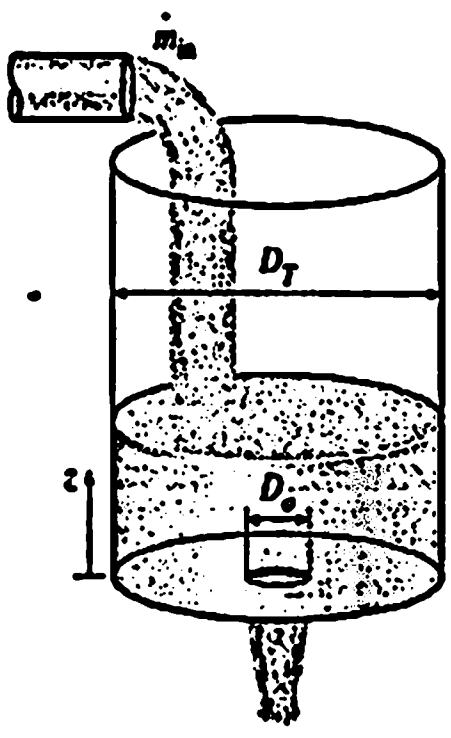
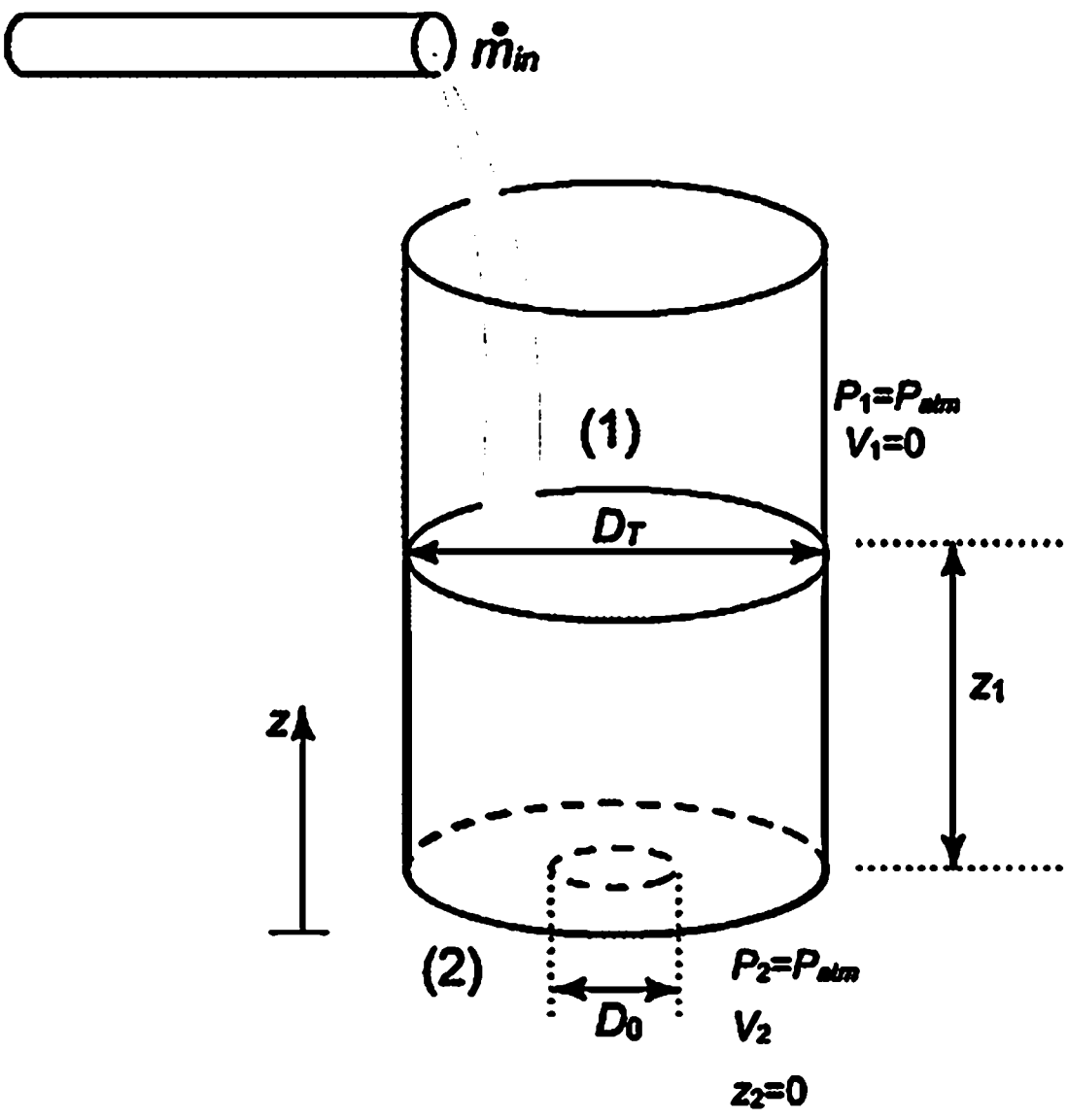


FIGURE P12-22

Step 1 of 7

Draw the schematic diagram of water tank system.



Step 2 of 7

(a)

Apply the Bernoulli's theorem between the points (1) and (2), by setting the bottom of the tank as datum, to find out the expression for the velocity at the exit.

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

Here, the pressure at points 1 and 2 are P_1 and P_2 respectively, velocity of points 1 and 2 are V_1 and V_2 respectively, and the height of points 1 and 2 above datum head is z_1 and z_2 respectively.

Substitute 0 for V_1 and 0 for z_2 .

$$\frac{P_{atm}}{\rho g}+\frac{0}{2g}+z=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g}+0$$

$$\frac{V_2^2}{2g}=z$$

$$V_2=\sqrt{2gz} \quad \text{..... (1)}$$

Step 3 of 7

Calculate the mass flow rate at the point-2 at orifice.

$$\dot{m}_{orifice}=\rho A_{orifice}V_2 \quad \text{..... (2)}$$

Here, the density of the water is ρ , the area of the orifice is $A_{orifice}$, and the velocity at point 2 is V_2 .

Substitute $\left(\frac{\pi D_0^2}{4}\right)$ for $A_{orifice}$ and equation (1) in equation (2).

$$\dot{m}_{orifice}=\rho\left(\frac{\pi D_0^2}{4}\right)\sqrt{2gz}$$

Here, the diameter of the orifice is D_0 .

Square the equation on both sides.

$$\left(\dot{m}_{orifice}\right)^2=\rho^2\left(\frac{\pi D_0^2}{4}\right)^2 2gz$$

$$z=\frac{\left(\dot{m}_{orifice}\right)^2}{\rho^2\left(\frac{\pi D_0^2}{4}\right)^2 2g}$$

When the water is at maximum height, it is equal to $z=z_{max}$ and the flow rate $\dot{m}_{orifice}=\dot{m}_{in}$.

$$z_{max}=\frac{\left(\dot{m}_{in}\right)^2}{\rho^2\left(\frac{\pi D_0^2}{4}\right)^2 2g}$$

$$z_{max} = \frac{(\dot{m}_{in})^2}{\rho^2\left(\frac{\pi D_0^2}{4}\right)^2 2g}$$

Therefore, the maximum height reached by the water in the tank is $\boxed{\frac{(\dot{m}_{in})^2}{\rho^2\left(\frac{\pi D_0^2}{4}\right)^2 2g}}$.

Step 4 of 7

(b)
Apply the principle of conservation of mass for the control volume undergoing a change during period of time dt.

$$dm_{cv} = \dot{m}_{in}dt - \dot{m}_{orifice}dt$$

$$\rho\left(\frac{\pi D_T^2}{4}\right)dz = \dot{m}_{in}dt - \rho\left(\frac{\pi D_0^2}{4}\right)\sqrt{2gz}dt$$

$$dt = \frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{\dot{m}_{in} - \rho\left(\frac{\pi D_0^2}{4}\right)\sqrt{2gz}}dz$$

$$= \frac{1}{\dot{m}_{in}}\left(\frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{1 - \frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2gz}}\right)dz$$

$$= \frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{\dot{m}_{in}}\left(\frac{1}{1 - \left(\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)z^{1/2}}\right)dz$$

Substitute b for $\left(\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)$ and a for $\frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{\dot{m}_{in}}$.

$$dt = a\left(\frac{1}{1 - bz^{1/2}}\right)dz \quad \text{..... (3)}$$

Step 5 of 7

Consider the following expression.

$$z = x^2 \quad \text{..... (4)}$$

$$dz = 2xdx$$

Substitute x^2 for z and $2xdx$ for dz in equation (3).

$$dt = a\left(\frac{1}{1 - bz^{1/2}}\right)dz$$

$$= a\left(\frac{1}{1 - b\left(x^2\right)^{1/2}}\right)(2xdx) \quad \text{..... (5)}$$

$$= a\left(\frac{2xdx}{1 - bx}\right)$$

Dissolve the fraction using the following method.

$$\text{Numerator} = A \times (\text{denominator}) + B \times \frac{d}{dt}(\text{denominator})$$

$$2x = A \times (1 - bx) + B \times \frac{d}{dx} \times (1 - bx) \quad \text{..... (6)}$$

$$2x = A \times (1 - bx) - bB$$

Equate the coefficients of t on the left hand side and right hand side and find out the variables A and B.

$$A = \frac{-2}{b}$$

$$A - bB = 0$$

$$A = bB$$

$$-\frac{2}{b} = bB$$

$$B = -\frac{2}{b^2}$$

Step 6 of 7

Substitute $\left(-\frac{2}{b^2}\right)$ for B and $-\frac{2}{b}$ for A in equation (6).

$$2x = \frac{-2}{b} \times (1 - bx) - \frac{-2}{b^2}b$$

$$2x = \frac{-2}{b} \times (1 - bx) + \frac{2}{b}$$

Substitute $\left(\frac{-2}{b} \times (1 - bx) + \frac{2}{b}\right)$ for $2x$ in equation (5).

$$dt = a\left(\frac{\frac{-2}{b} \times (1 - bx) + \frac{2}{b}}{\quad}\right)dx$$

Substitute $\left(\frac{-2}{b}\times(1-bx)+\frac{2}{b}\right)$ for $2x$ in equation (5).

12 022R.pg 3

$$dt=a\left(\frac{\frac{-2}{b}\times(1-bx)+\frac{2}{b}}{1-bx}\right)dx$$

$$dt=\frac{2a}{b}\left(\frac{1}{1-bx}-1\right)dx$$

Step 7 of 7

Integrate the equation on both sides.

$$\int dt=\int\frac{2a}{b}\left(\frac{1}{1-bx}-1\right)dx$$

$$t=\frac{2a}{b}\left(\ln(1-bx)-x\right)$$

Substitute the equation (4).

$$t=\frac{2a}{b}\left(\ln\left(1-bz^{1/2}\right)-z^{1/2}\right)\quad\ldots\ldots\ (7)$$

Substitute $\left(\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)$ for b and $\frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{\dot{m}_{in}}$ for a in the equation (7).

$$t=\frac{2\frac{\rho\left(\frac{\pi D_T^2}{4}\right)}{\dot{m}_{in}}}{\left(\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)}\left(\ln\left(1-\left(\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)z^{1/2}\right)-z^{1/2}\right)$$

$$=2\frac{D_T^2}{\sqrt{2g}D_0^2}\left(\left(\ln\left(1-\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)z^{1/2}\right)-z^{1/2}\right)$$

Therefore, the relation between the time and height is,

$$\boxed{t=2\frac{D_T^2}{\sqrt{2g}D_0^2}\left(\left(\ln\left(1-\frac{\rho\left(\frac{\pi D_0^2}{4}\right)}{\dot{m}_{in}}\sqrt{2g}\right)z^{1/2}\right)-z^{1/2}\right)}.$$

Question: 12.023P

Water flows through a horizontal pipe at a rate of 1 gal/s. The pipe consists of two sections of diameters 4 in and 2 in with a smooth reducing section. The pressure difference between the two pipe sections is measured by a mercury manometer. Neglecting frictional effects, determine the differential height of mercury between the two pipe sections.

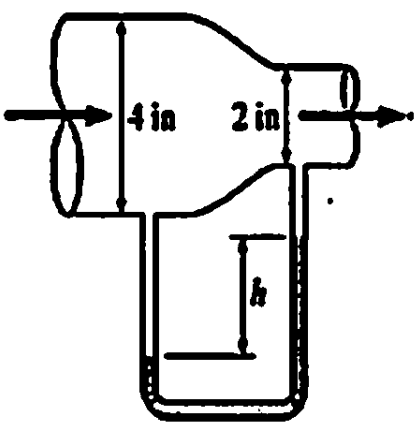


FIGURE P12–23E

Step 1 of 7

Given data:

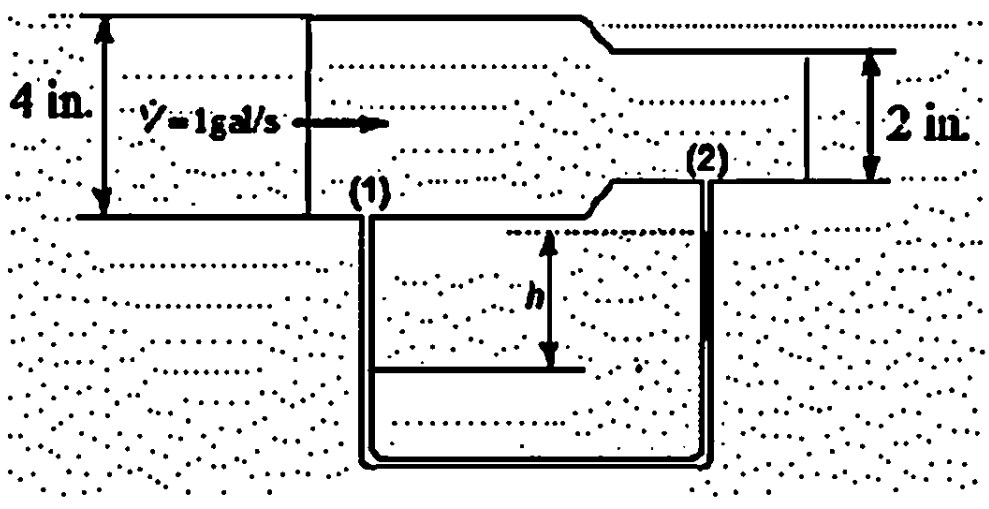
Flow rate of water though a horizontal pipe, $\dot{V} = 1\text{gal/s}$

Diameter of horizontal pipe at section (1), $D_1 = 4\text{in.}$

Diameter of horizontal pipe at section (2), $D_2 = 2\text{in.}$

Step 2 of 7

Schematic diagram of manometer arrangement in a horizontal pipe:



Step 3 of 7

From the continuity equation, rate of flow through a horizontal pipe section is given as

$$\dot{V} = A_1V_1 = A_2V_2$$

Where,

A_1 is the area of the pipe section (1)

A_2 Is the area of the pipe section (2)

V_1 is the velocity of water at section (1)

V_2 is the velocity water at section (2)

Then, velocity of water in a pipe section (1) is

$$\dot{V} = A_1V_1$$

$$V_1 = \frac{\dot{V}}{A_1}$$

$$= \frac{(1\text{gal/s})\left(\frac{231\text{in}^3}{1\text{gal}}\right)}{\frac{\pi}{4}(4\text{in.})^2}$$

$$= 18.38\text{in./s}$$

$$= 18.38\text{in./s} \times 1\text{ft}/12\text{in.}$$

$$= 1.53\text{ft/s}$$

Step 4 of 7

Similarly, velocity of water in a pipe section (2) is

$$\dot{V} = A_2V_2$$

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{(1\text{gal/s})\left(\frac{231\text{in}^3}{1\text{gal}}\right)}{\frac{\pi}{4}(2\text{in.})^2}$$

$$= 73.52\text{in./s}$$

$$= 73.52\text{in./s} \times 1\text{ft}/12\text{in.}$$

$$= 6.126\text{ft/s}$$

Step 5 of 7

Applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{P_1}{\rho_w} + \frac{V_1^2}{2} + gz_1 = \frac{P_2}{\rho_w} + \frac{V_2^2}{2} + gz_2$$

$$P_1 - P_2 = \rho_w\left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right)$$

Since pipe placed along the horizontal direction, therefore datum head becomes equal to zero.

Pressure difference between inlet and exit of the pipe is equivalent to pressure of water.

$$P_1 - P_2 = \gamma_{mercury}h$$

Then we can write,

$$\gamma_{mercury}h = \rho_w\left(\frac{V_2^2 - V_2^2}{2g}\right)$$

Step 6 of 7

From the table of "ideal gas, specific heats of various fluids"

For mercury,

Specific weight $\gamma_{mercury} = 62.41\text{lbm/ft}^3$

Pressure head of mercury in terms of pressure head of water is determined by multiplying to 12.6

Step 7 of 7

Then, The height of water column

$$\gamma_{mercury}h = \rho_w\frac{\left(V_2^2 - V_2^2\right)}{2g}$$

$$\left(62.41\text{lbm/ft}^3 \times 12.6\right)h = 62.41\text{lbm/ft}^3\left(\frac{\left((6.126\text{ft/s})^2 - (1.53\text{ft/s})^2\right)}{2\left(32.2\text{ft/s}^2\right)}\right)$$

$$= 0.0433\text{ft}$$

$$= 0.0433\text{ft} \times \frac{12\text{in.}}{1\text{ft}}$$

$$= 0.52\text{in.}$$

Therefore, the differential height of the water column is $\boxed{h = 0.52\text{in.}}$

Question: 12.024P

An airplane is flying at an altitude of 12,000 m. Determine the gage pressure at the stagnation point on the nose of the plane if the speed of the plane is 300 km/h. How would you solve this problem if the speed were. 1050 km/h? Explain.

Step 1 of 6

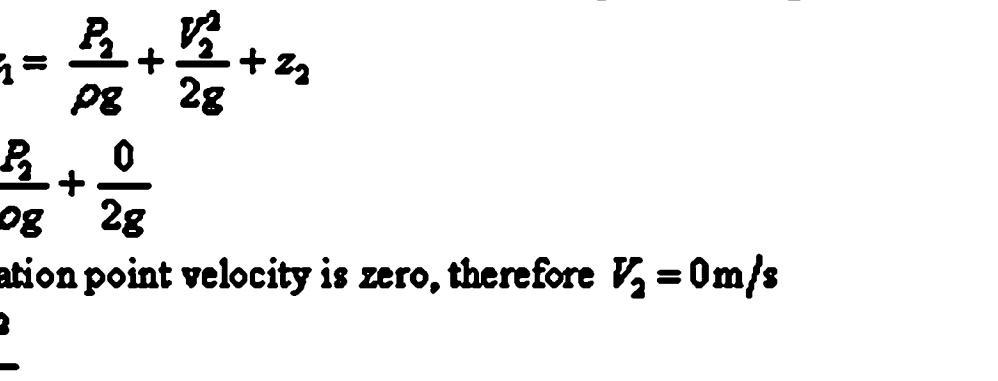
Given data:

The altitude of the airplane, $h = 12000$ m

The speed of the aircraft, $V = 300$ km/h

Step 2 of 6

Schematic diagram of airplane nose:



Step 3 of 6

The aircraft moves at a speed of $V = 300$ km/h through the atmosphere. Then we can assume that the airflow over the aircraft will be having a velocity of 300 km/h .

The stagnation point at the tip of the nose cone is point (2) and point where air has a velocity of 300 km/h before stagnation is represented as point (1).

Applying Bernoulli's equation between the points (1) and (2)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

The points are at the same level, therefore $z_1 = z_2$

From the table of "properties of air table"

The density of air at 12000 m is $\rho = 0.312$ kg/m^3 .

The velocity at point (1),

$$V_1 = 300 \text{ km/h}$$

$$= 300 \text{ km/h} \times \frac{5}{18} \text{m/s}$$

$$= 83.33 \text{ m/s}$$

Step 4 of 6

Substituting the known values in the Bernoulli's equation, we get,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \frac{0}{2g}$$

Since at stagnation point velocity is zero, therefore $V_2 = 0$ m/s

$$P_2 - P_1 = \frac{\rho V_1^2}{2}$$

$$P_2 - P_1 = \frac{(0.312)(83.33)^2}{2}$$

$$= 1083.24 \text{ Pa}$$

Then the gage pressure at stagnation point,

$$P_{gage} = P_2 - P_1$$

$$= 1083.24 \text{ Pa}$$

Therefore, the gage pressure at the stagnation point is $\boxed{1083.24 \text{ Pa}}$.

Step 5 of 6

The aircraft is at a speed of 1050 km/h .

We can assume that the airflow speed over the aircraft is 1050 km/h .

Then,

$$V_1 = 1050 \text{ km/h}$$

$$= 1050 \text{ km/h} \times \frac{\frac{5}{18} \text{m/s}}{1 \text{km/h}}$$

$$= 291.66 \text{ m/s}$$

Step 6 of 6

The mach number of the aircraft,

$$Ma = \frac{V_1}{c}$$

Where,

c be taken is the velocity of sound approximated to 340 m/s

Then

$$Ma = \frac{291.66}{340}$$

$$= 0.85$$

We know that for the flow to be considered incompressible when the Mach number $Ma \leq 0.3$. But our value for Mach number of the aircraft is $Ma = 0.85$ which is greater than 0.3

Therefore the flow in this case is compressible. So the problem should be solved using the Bernoulli's equation for steady, compressible, isentropic flow.

Question: 12.025P

The air velocity in the duct of a heating system is to be measured by a Pitot-static probe inserted into the duet parallel to the flow. If the differential height between the water columns connected to the two outlets of the probe is 2.4 cm, determine (*a*) the flow velocity and (*b*) the pressure rise at the tip of the probe. The air temperature and pressure in the duct are 45°C and 98 kPa, respectively.

Step 1 of 6

Given data:

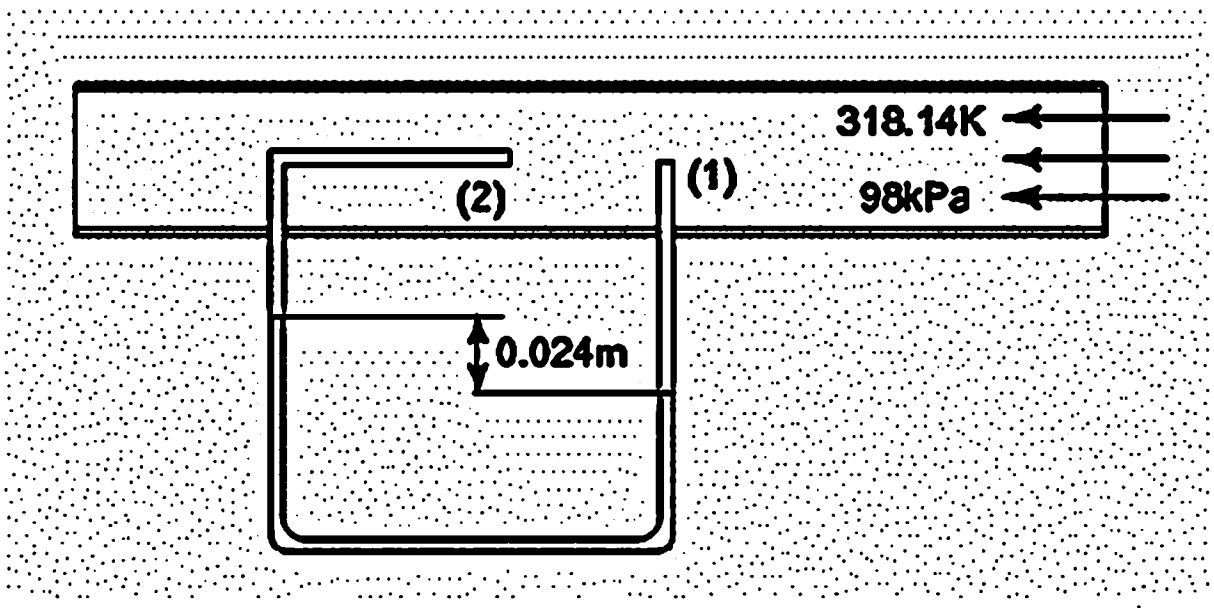
The air temperature in the duct $T = 45°C = 318.15K$

The air pressure in the duct $P_{air} = 98\ kPa$

The differential height between the water columns $\Delta h = 2.4\ cm = 0.024\ m$

Step 2 of 6

Schematic diagram of pitot-static probe:



The points (1) and (2) are at same elevation, $z_1 = z_2$

The velocity at stagnation point (2), $V_2 = 0$

Step 3 of 6

From the ideal gas equation, we have

$P = \rho_{air} RT$

Where

The pressure $P_{air} = 98\ kPa$

The temperature $T = 318.15\ K$

The gas constant $R = 0.287\ kJ/kg \cdot K$

Then the density of air in duct,

$$\rho_{air} = \frac{(98 \times 10^3\ Pa)}{(0.287 \times 10^3\ kJ/kg \cdot K)(318.14\ K)}$$

$$= 1.073\ kg/m^3$$

Step 4 of 6

Applying Bernoulli's equation between the points (1) and (2)

$$\frac{P_1}{\rho_{air} g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g} + z_2$$

Substituting the required values and simplifying the above relation

$$\frac{V_1^2}{2} = \frac{P_2}{\rho_{air}} - \frac{P_1}{\rho_{air}}$$

$$V_1 = \sqrt{\frac{2(P_2 - P_1)}{\rho_{air}}}$$

Step 5 of 6

(b)

The pressure difference can be expressed as,

$$P_2 - P_1 = \rho_{water} g h$$

$$= (1000)(9.81)(0.024)$$

$$= 235.44\ Pa$$

Therefore the pressure rise, $\boxed{P_2 - P_1 = 235.44\ Pa}$

Step 6 of 6

a)

The flow velocity,

$$V_1 = \sqrt{\frac{2(P_2 - P_1)}{\rho_{air}}}$$

$$= \sqrt{\frac{2(235.44)}{1.073}}$$

$$= 20.94\ m/s$$

Therefore, the flow velocity is $\boxed{V_1 = 20.94\ m/s}$

Question: 12.026P

The water in a 8-m-diameter, 3-m-high aboveground swimming pool is to be emptied by unplugging a 3-cm-diameter, 25-m-long horizontal pipe attached to the bottom of the pool. Determine the maximum discharge rate of water through the pipe. Also, explain why the actual flow rate will be less.

Step 1 of 4

Given data:

The diameter of the swimming pool $D = 8$ m

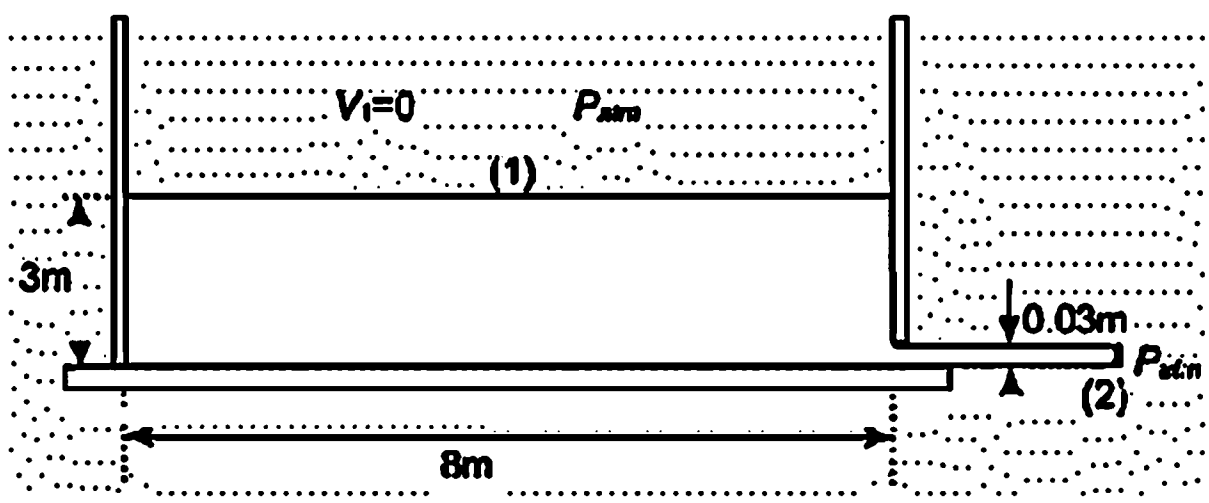
The pool is at an elevation from ground level $h = 3$ m

The diameter of the exit pipe $d = 3\text{cm} = 0.03\text{m}$

Length of pipe $L = 25\text{m}$

Step 2 of 4

Schematic diagram of the swimming pool:



Step 3 of 4

Let us set point (2) as the reference point.

Applying Bernoulli's equation between points (1) and (2)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

Substitute the known values in above expression, we get

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_1 = \frac{P_{atm}}{\rho g} + \frac{V_2^2}{2g} + 0$$

$$V_2^2 = 2gz_1$$

$$V_2 = \sqrt{2gz_1}$$

From the above equation it is evident that the exit velocity is maximum, when the head z_1 is maximum.

Step 4 of 4

The discharge of water through orifice is

$$\dot{V} = A_{orifice} V_2$$

$A_{orifice}$ Is the area of the orifice.

The maximum discharge is when $z_1 = 3$ m

Then,

$$\dot{V}_{max} = \left(\frac{\pi}{4} D_{orifice}^2\right)\sqrt{2gz_1}$$

$$= \left(\frac{\pi}{4} \times 0.03^2\right)\sqrt{2(9.81)(3)}$$

$$= 5.423 \times 10^{-3}\ \text{m}^3/\text{s}$$

Therefore, the maximum discharge rate of water through the pipe

$$\boxed{\dot{V}_{max} = 5.42 \times 10^{-3}\ \text{m}^3/\text{s}}$$

The actual flow rate is less than the theoretical flow rate because we have not taken frictional effects into account. Presence of frictional effects always reduces the flow rate in a system.

Question: 12.027P

Reconsider Prob. 12–26. Determine how long it will take to empty the swimming pool completely.

Step 1 of 7

Given data:

The diameter of the swimming pool $D = 8$ m

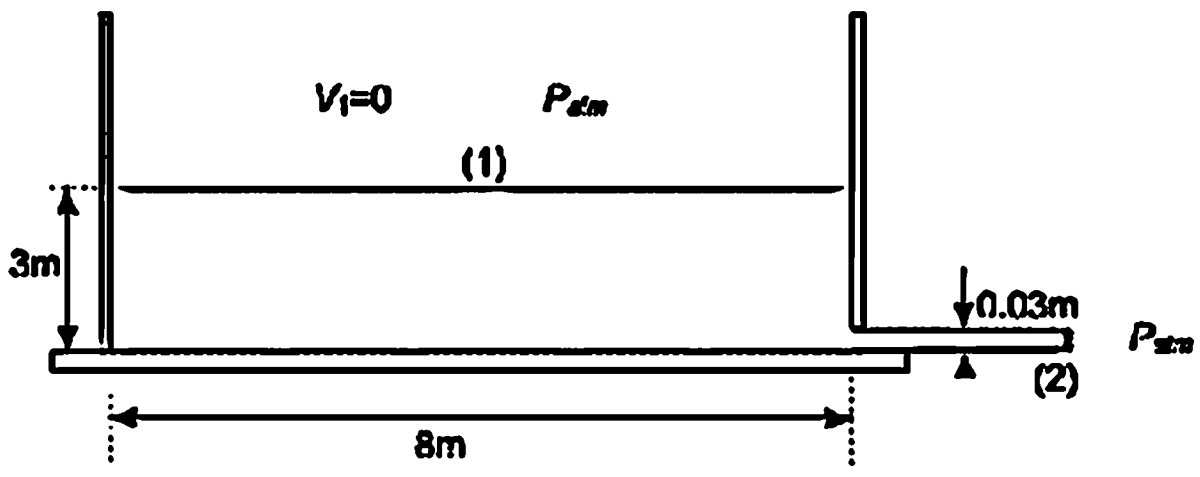
The pool is at an elevation from ground level $h = 3$ m

The diameter of the exit pipe $d = 3\text{cm} = 0.03\text{m}$

Length of pipe, $L = 25\text{m}$

Step 2 of 7

Schematic diagram of the swimming pool:



Step 3 of 7

Let us set point (2) as the reference point.

Applying Bernoulli's equation between points (1) and (2)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

Substituting the known values

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_1 = \frac{P_{atm}}{\rho g} + \frac{V_2^2}{2g} + 0$$

$$V_2^2 = 2gz_1$$

$$V_2 = \sqrt{2gz_1}$$

Step 4 of 7

The discharge,

$$\dot{V} = A_{orifice} V_2$$

$A_{orifice}$ is the area of the orifice

The discharge of water during a time period dt is,

$$\dot{V} = \frac{dV}{dt}$$

$$dV = \dot{V} dt$$

$$dV = AV_2 dt$$

$$= \frac{\pi D^2}{4}\sqrt{2gz_1}dt$$

Where z_1 varies from h to 0 when the tank is emptied completely.

Step 5 of 7

Applying the principle of conservation of mass.

Then the discharge dV must be equal to the volume of water removed from the cylinder during the period dt

Then

$$dV = A_{tank}(-dz)$$

Where

'dz'is the height change and negative sign represents the decrease in elevation

A_{tank} is the area of the tank

Step 6 of 7

Then

$$A_{tank}(-dz) = \frac{\pi D^2}{4}\sqrt{2gz_1}dt$$

$$\left(\frac{\pi}{4}D_0^2\right)(-dz) = \frac{\pi D^2}{4}\sqrt{2gz_1}dt$$

$$dt = -\left(\frac{D_0^2}{D^2}\right)\frac{(dz)}{\sqrt{2gz_1}}$$

Integrating the above equation we get

$$t = -\int_h^0 \left(\frac{D_0^2}{D^2}\right)\frac{(dz)}{\sqrt{2gz_1}}$$

$$= -\frac{1}{\sqrt{2g}}\left(\frac{D_0^2}{D^2}\right)\int_h^0 (z_1)^{-1/2} dz$$

$$= -\frac{1}{\sqrt{2g}}\left(\frac{D_0^2}{D^2}\right)\left(z_1^{1/2}\right)_h^0$$

$$= \frac{1}{\sqrt{2g}}\left(\frac{D_0^2}{D^2}\right)\left(\frac{h^{1/2}}{1/2}\right)$$

Step 7 of 7

Then the time required draining the tank,

$$t = \frac{1}{\sqrt{2g}}\left(\frac{D_0^2}{D^2}\right)\left(h^{1/2}\right)$$

$$= \left(\frac{8^2}{0.03^2}\right)\left(\sqrt{\frac{2h}{g}}\right)$$

$$= \left(\frac{8^2}{0.03^2}\right)\left(\sqrt{\frac{2\times 3}{9.81}}\right)$$

$$= 55613.289\ s$$

$$= 15.44\text{hr}$$

Therefore, the time required to empty the tank $\boxed{t = 15.44\text{hr}}$

Question: 12.028P

Reconsider Prob. 12–27. Using EES (or other) software, investigate the effect of the discharge pipe diameter on the time required to empty the pool completely. Let the diameter vary from 1 to 10 cm in increments of 1 cm. Tabulate and plot the results.

Step 1 of 4

Given data:

The diameter of the swimming pool $D = 8$ m

The pool is at an elevation from ground level $h = 3$ m

The diameter of the exit pipe $d = 3\text{cm} = 0.03\text{m}$

Length of pipe, $L = 25\text{m}$

Step 2 of 4

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.

File Edit Search Options Calculate Tables

Equations Window

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
h=3 "m"
D=d_pipe/100 "m"
D_pool=8 "m"
V_initial=SQRT(2*g*h) "m/s"
Ac=pi*D^2/4
V_dot=Ac*V_initial*1000 "m3/s"
t=(D_pool/D)^2*SQRT(2*h/g)/3600 "hour"
```

Step 3 of 4

After writing the equation in equation window, then select table on menu bar, then click new parameter table.

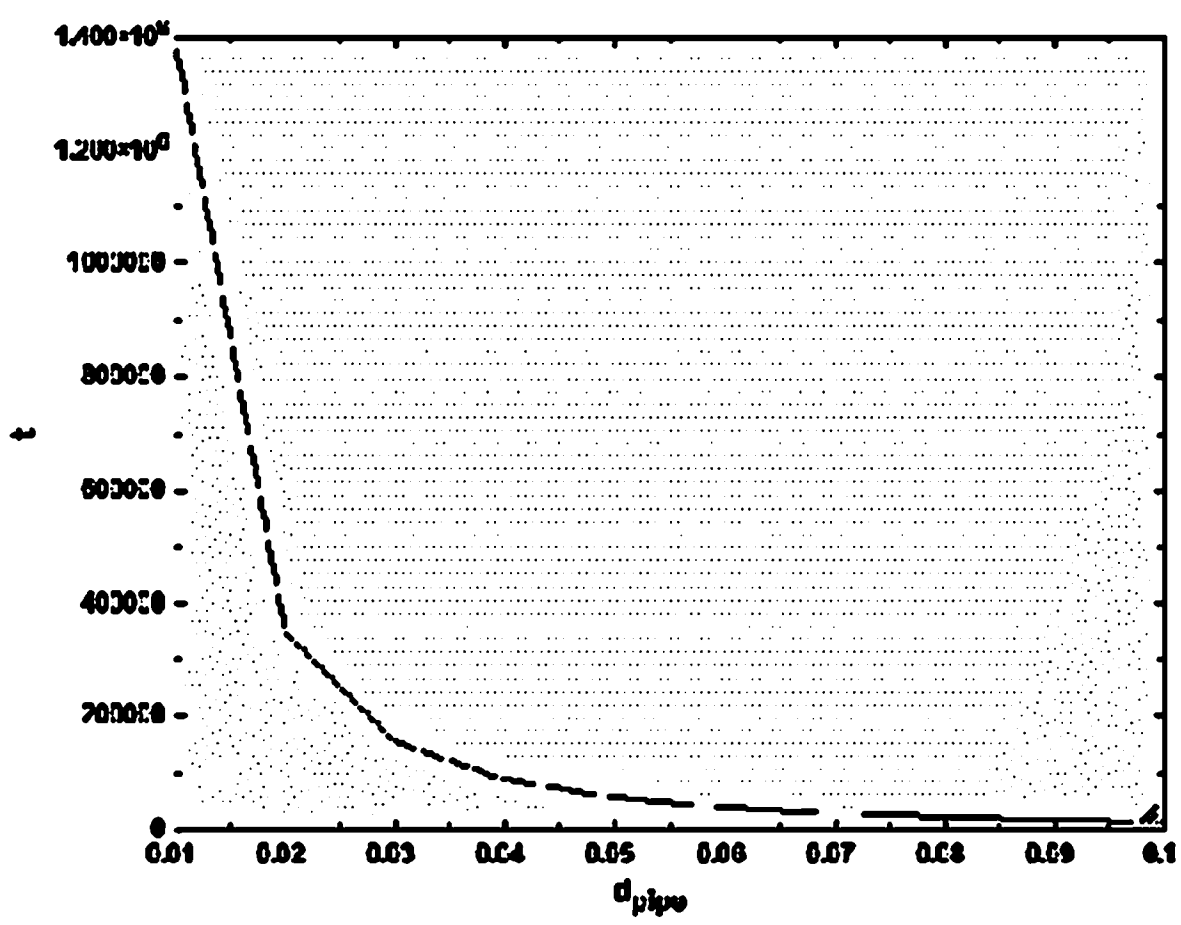
The dialog box will be appeared, then select the diameter of pipe and time, then click ADD, then click OK button.

Then the following results are displayed

1..10	d_{pipe}	t
Run 1	0.01	1.390E+06
Run 2	0.02	347583
Run 3	0.03	154481
Run 4	0.04	86896
Run 5	0.05	55613
Run 6	0.06	38620
Run 7	0.07	28374
Run 8	0.08	21724
Run 9	0.09	17165
Run 10	0.1	13903

Step 4 of 4

Select the plot on menu bar, then click x-y plot .Dialog box will appeared, then select variables to draw graph, then click ok button.



Question: 12.029P

Air at 110 kPa and 50°C flows upward through a 6-cm-diameter inclined duct at a rate of 45 L/s. The duct diameter is then reduced to 4 cm through a reducer. The pressure change across the reducer is measured by a water manometer. The elevation difference between the two points on the pipe where the two arms of the manometer are attached is 0.20 m. Determine the differential height between the fluid levels of the two arms of the manometer.

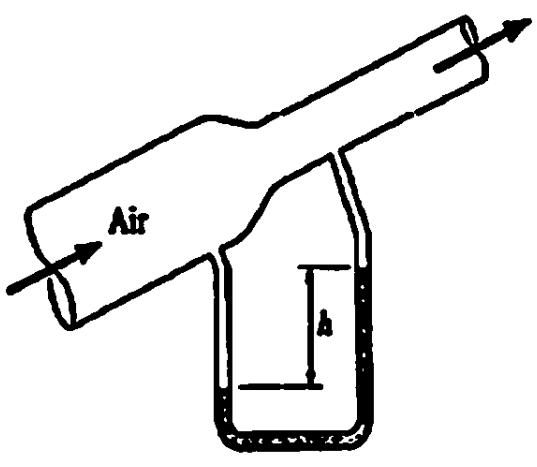
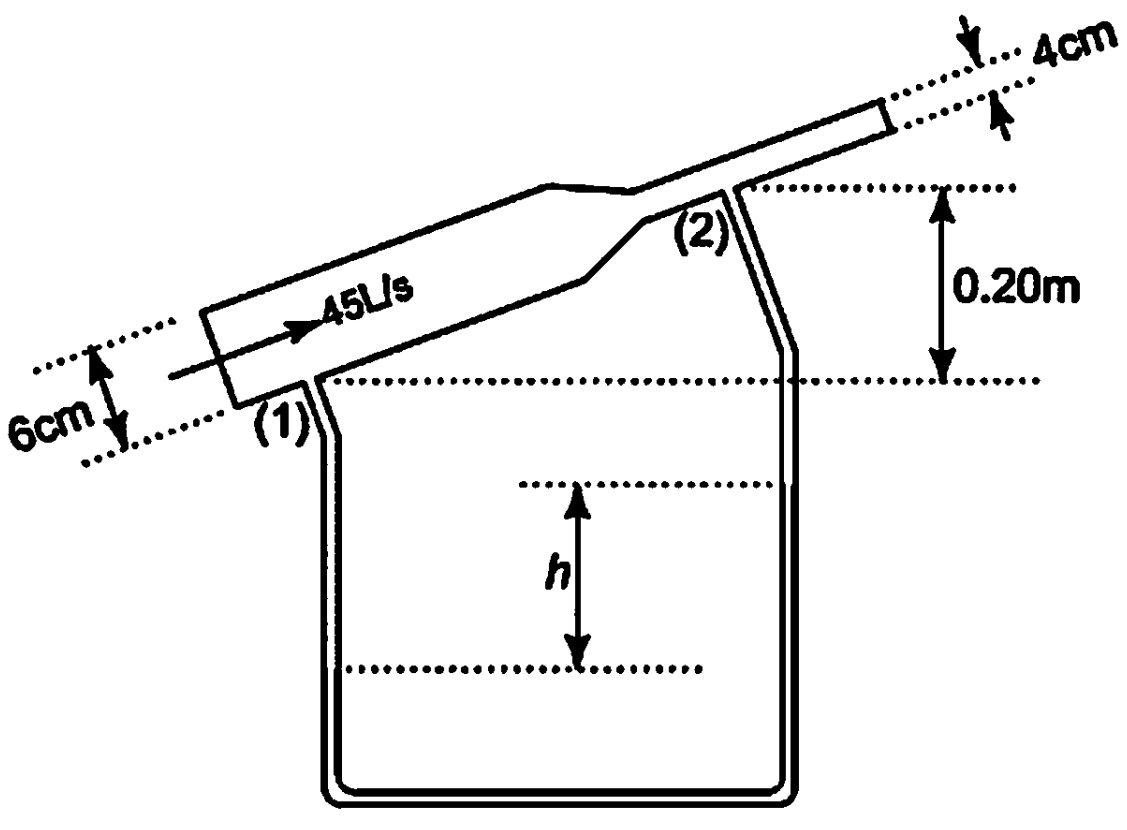


FIGURE P12–29

Step 1 of 5

Free body diagram of pipe and manometer system is as shown below:



Step 2 of 5

Use the ideal gas equation to find out the density of air.

$$P = \rho RT$$

Here, the gas constant is R, density of air is ρ_{air}, the temperature of air is T and the pressure of air is P.

Substitute $0.287\ \text{kJ/kg.K}$ for R, $323.15\ \text{K}$ for T and $110\ \text{kPa}$ for P in the above equation.

$$110 = \rho_{air}(0.287)(323.15)$$
$$\rho_{air} = 1.186\ \text{kg/m}^3$$

Step 3 of 5

Use the continuity equation, to calculate the velocity of flow at point-1.

$$\dot{V} = A_1 v_1$$
$$v_1 = \frac{\dot{V}}{A_1}$$
$$= \frac{\dot{V}}{\frac{\pi}{4} \times d_1^2}$$

Here, the velocity at the point-1 is v_1, the volumetric flow rate is $\dot{V}$ and the area of the duct at point-1 is A_1 and the diameter of the pipe at section-1 is d_1.

Substitute $0.045\ \text{m}^3/\text{s}$ for $\dot{V}$ and $0.06\ \text{m}$ for d_1 in the above equation.

$$v_1 = \frac{0.045}{\frac{\pi}{4}(0.06)^2}$$
$$= 15.91\ \text{m/s}$$

Step 4 of 5

Now calculate the velocity at point (2).

$$v_2 = \frac{\dot{V}}{A_2}$$
$$= \frac{\dot{V}}{\frac{\pi}{4} \times d_2^2}$$

Here, the velocity at the point-2 is v_2, the area of cross-section at point-2 is A_2 and the diameter of the point-2 is d_2.

Substitute $0.04\ \text{m}$ for d_2 and all the corresponding values in the above equation.

$$v_2 = \frac{0.045\ \text{m}^3/\text{s}}{\frac{\pi}{4}(0.04)^2}$$
$$= 35.809\ \text{m/s}$$

Step 5 of 5

Apply the Bernoulli's equation between points (1) and (2), keeping point-1 as the reference point.

$$\frac{P_1}{\rho_{air} g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g} + z_2$$

Here, the pressure at point-1 is P_1, the pressure at point-2 is P_2, the height above the datum head at point-1 and 2 are z_1 and z_2 respectively and the acceleration due to gravity is g.

Substitute 0 for z_1 and $0.20\ \text{m}$ for z_2 and all the corresponding values in the above equation.

$$\frac{P_1}{\rho_{air}} + \frac{V_1^2}{2} + 0 = \frac{P_2}{\rho_{air}} + \frac{V_2^2}{2} + 0.20$$
$$\frac{P_1 - P_2}{\rho_{air}} = \frac{V_2^2 - V_1^2}{2} + 0.20$$

$$P_1 - P_2 = \rho_{air}\left(\frac{V_2^2 - V_1^2}{2} + 0.20\right)$$
$$= 1.186\left(\frac{35.809^2 - 15.91^2}{2} + 0.20\right)$$
$$= 610.526\ \text{Pa}$$

The pressure difference between the points is $P_1 - P_2 = 610.526\ \text{Pa}$

Calculate the differential height taking into account that the pressure difference in duct is equal to differential height of manometer.

$$P_2 - P_1 = \rho_w g h_w$$

Here the density of water is ρ

Substitute $1000\ \text{kg/m}^3$ for ρ and all the corresponding values in the above equation.

$$h = \frac{P_2 - P_1}{\rho_w g}$$
$$= \frac{610.526}{(1000)(9.81)}$$
$$= 0.0622\ \text{m}$$

Therefore, the differential height of the manometer, h, is $\boxed{0.0622\ \text{m}}$

Question: 12.030P

Air is flowing through a venturi meter whose diameter is 2.6 in at the entrance part (location 1) and 1.8 in at the throat (location 2). The gage pressure is measured to be 12.2 psia at the entrance and 11.8 psia at the throat. Neglecting frictional effects, show that the volume flow rate can be expressed as

$$\dot{V} = A_2\sqrt{\frac{2(P_1 - P_2)}{\rho(1 - A_2^2/A_1^2)}}$$

and determine the flow rate of air. Take the air density to be 0.075 lbm/ft3.

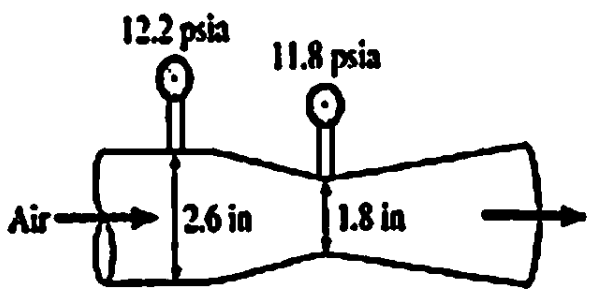


FIGURE P12–30E

Step 1 of 7

Given data:

Inlet diameter of the venturi meter, $D_1 = 2.6\text{in.}$

Throat diameter, $D_2 = 1.8\text{in}$

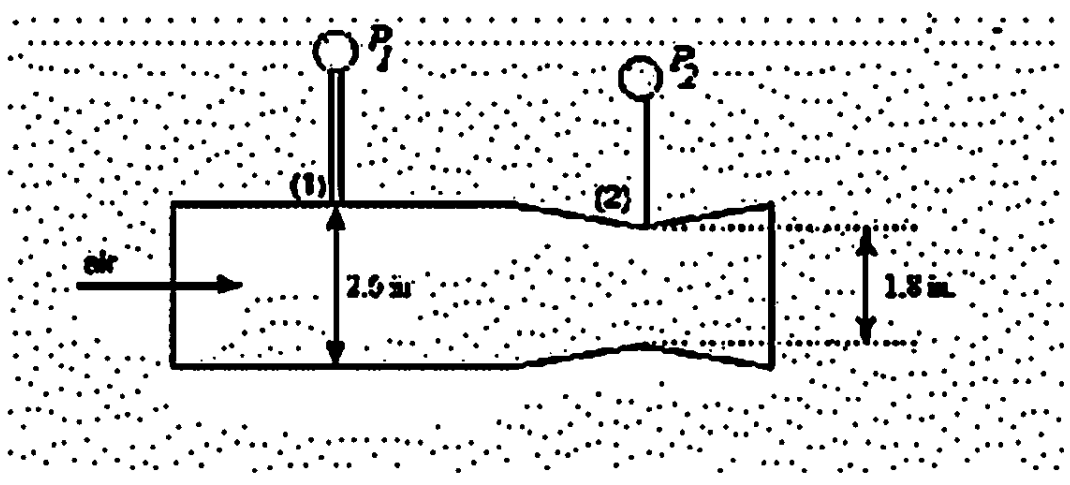
Inlet pressure of the venturi meter, $P_1 = 12.2\text{psia}$

Throat pressure, $P_2 = 11.8\text{psia}$

Density of the air, $\rho = 0.075\text{lbm/ft}^3$

Step 2 of 7

Schematic diagram of venturi meter:



Step 3 of 7

The flow is steady, incompressible and irrotational.
The frictional effects can be neglected.
The points (1) and (2) are at the same elevation.
By apply the Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho_{air}g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{air}g} + \frac{V_2^2}{2g} + z_2$$

Substitute the required values and simplify.

$$\frac{P_1}{\rho} + \frac{V_1^2}{2} + 0 = \frac{P_2}{\rho} + \frac{V_2^2}{2} + 0$$

$$P_1 - P_2 = \frac{\rho}{2}\left(V_2^2 - V_1^2\right)$$

Step 4 of 7

From the continuity equation,
Flow velocities at (1) and (2),

$$V_1 = \frac{\dot{V}}{A_1} \text{ And } V_2 = \frac{\dot{V}}{A_2}$$

Where

$\dot{V}$ Is the flow rate and A is the area at respective position.

Step 5 of 7

$$A_2 = \frac{\pi}{4}D_2^2$$

$$= \frac{\pi}{4}(1.8)^2$$

$$= 2.54\text{in.}^2$$

Step 7 of 7

Then, volume flow rate though a venturi meter is given as

$$\dot{V} = A_2\sqrt{\frac{2(P_1 - P_2)}{\rho\left(1 - \frac{A_2^2}{A_1^2}\right)}}$$

Substituting the all known values above expression, we get

$$\dot{V} = 2.54\sqrt{\frac{2(12.2 - 11.8)}{\left(0.075\text{lbm/ft}^3 \times \left(\frac{1\text{ft}}{12\text{in.}}\right)^3\right)\left(1 - \frac{2.54^2}{5.30^2}\right)}}$$

$$= 392.90\text{in}^3/\text{s}$$

$$= 0.227\text{ft}^3/\text{s}$$

Therefore, volume flow rate though a venturi meter is $\boxed{\dot{V} = 0.227\text{ft}^3/\text{s}}$

Question: 12.031P

The water pressure in the mains of a city at a particular location is 350 kPa gage. Determine if this main can serve water to neighborhoods that are 50 m above this location

Step 1 of 3

Given data:

The gage pressure of water in the mains, $P_{gage} = 350$ kPa.

The height to which water is to be served, $h = 50$ m.

Step 2 of 3

From the properties of various fluids table

For water, the density of water is $\rho = 1000\ \text{kg/m}^3$.

From the expression of gauge pressure

$P_{gage} = \rho g h_{water}$

Where,

h_{water} is the height of water column corresponding to the pressure P_{gage}.

Step 3 of 3

By substitute the known values in above equation, we get

$350 \times 10^3 \text{Pa} = \left(1000\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right) h_{water}$

$h_{water} = 35.677$ m

Here $h_{water} < h$

Therefore we can conclude that the pressure in the mains is insufficient to supply the water to a height of 50 m.

Question: 12.032P

A pressurized tank of water has a 10-cm-diameter orifice at the bottom, where water discharges to the atmosphere. The water level is 2.5 m above the outlet. The lank air 1 pressure above the water level is 250 kPa (absolute) white the atmospheric pressure is 100 kPa. Neglecting frictional effects, determine the initial discharge rate of water from the tank.

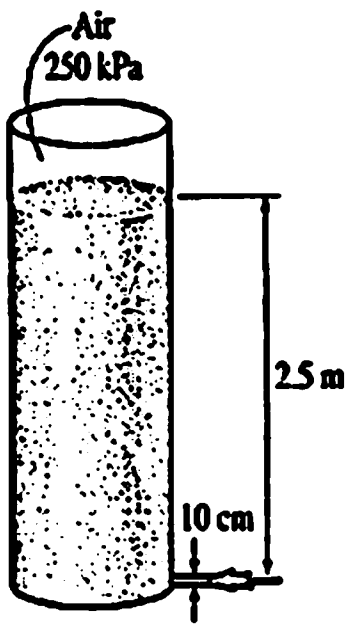


FIGURE P12-32

Step 1 of 4

Given data:-

The diameter of the orifice, $D_0 = 10\,\text{cm} = 0.1\,\text{m}$.

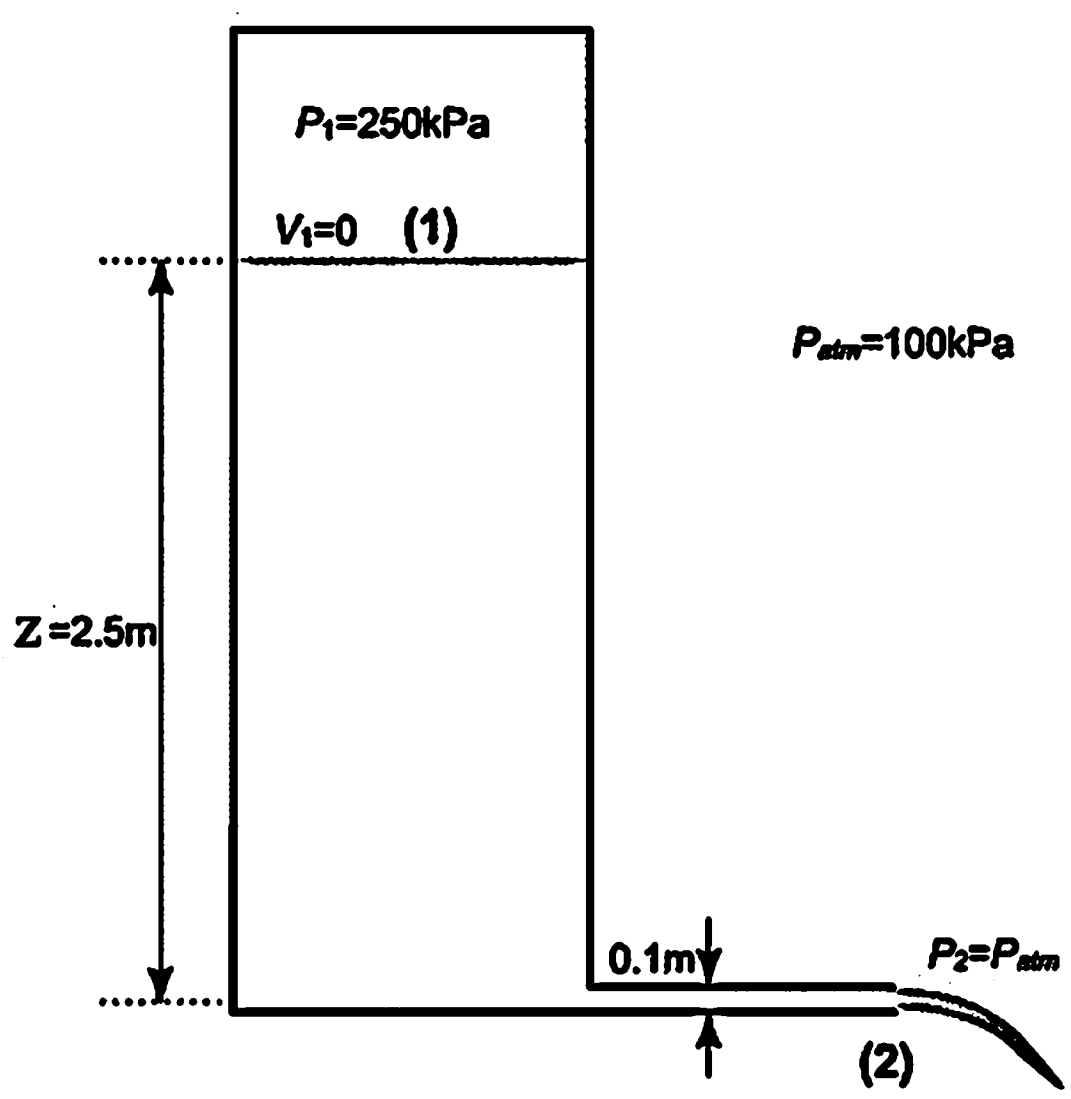
The absolute pressure at the top of the water surface, $P_1 = 250\,\text{kPa}$.

The atmospheric pressure, $P_{atm} = 100\,\text{kPa}$.

The height of water column, $z_1 = 2.5\,\text{m}$.

Step 2 of 4

Schematic diagram of tank system:



Point (2) is set as the reference point.

Step 3 of 4

By apply the Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

Substitute the known values and simplify.

$$\frac{250\times10^3}{1000\times9.81}+\frac{0}{2\times9.81}+z_1=\frac{100\times10^3}{1000\times9.81}+\frac{V_2^2}{2\times9.81}+z_2$$

$$\frac{250\times10^3}{1000\times9.81}+\frac{0}{2\times9.81}+2.5=\frac{100\times10^3}{1000\times9.81}+\frac{V_2^2}{2\times9.81}+0$$

$$\frac{V_2^2}{2\times9.81}=17.79$$

$$V_2=\sqrt{349.05}$$

$$=18.682\,\text{m/s}$$

Step 4 of 4

The flow rate at point (2),

$$\dot{V}=A_{orifice}V_2$$

$$=\left(\frac{\pi}{4}D_0^2\right)(18.682)$$

$$=\left(\frac{\pi}{4}0.1^2\right)(18.682)$$

$$=0.1467\,\text{m}^3/\text{s}$$

Therefore the flow rate through the orifice is $\boxed{\dot{V}=0.1467\,\text{m}^3/\text{s}}$.

Question: 12.033P

Reconsider Prob. 12–32. Using EES (or other) software, investigate the effect of water height in the tank on the discharge velocity. .Let the water height vary from 0 to 5 m in increments of 0.5 m. Tabulate and plot the results.

Step 1 of 4

Given data:

The diameter of the orifice, $D_0 = 10\ \text{cm} = 0.1\ \text{m}$.

The absolute pressure at the top of the water surface, $P_1 = 250\ \text{kPa}$.

The atmospheric pressure, $P_{atm} = 100\ \text{kPa}$.

The height of water column, $z_1 = 2.5\ \text{m}$.

Step 2 of 4

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.

```
g=9.814 " m/s2"
rho=1000 "kg/m3'
"d=0.10 "m"
P1=250 "kPa"
P_atm=100 "kPa"
V=SQRT(2*(P1-P_atm)*1000/rho+2*g*h)
Ac=pi*D^2/4
V_dot=Ac*V
```

Step 3 of 4

After writing the equation in equation window, then select table on menu bar, then click new parameter table.

The dialog box will be appeared, then select the diameter of pipe and time, then click ADD, then click OK button.

The following results are displayed

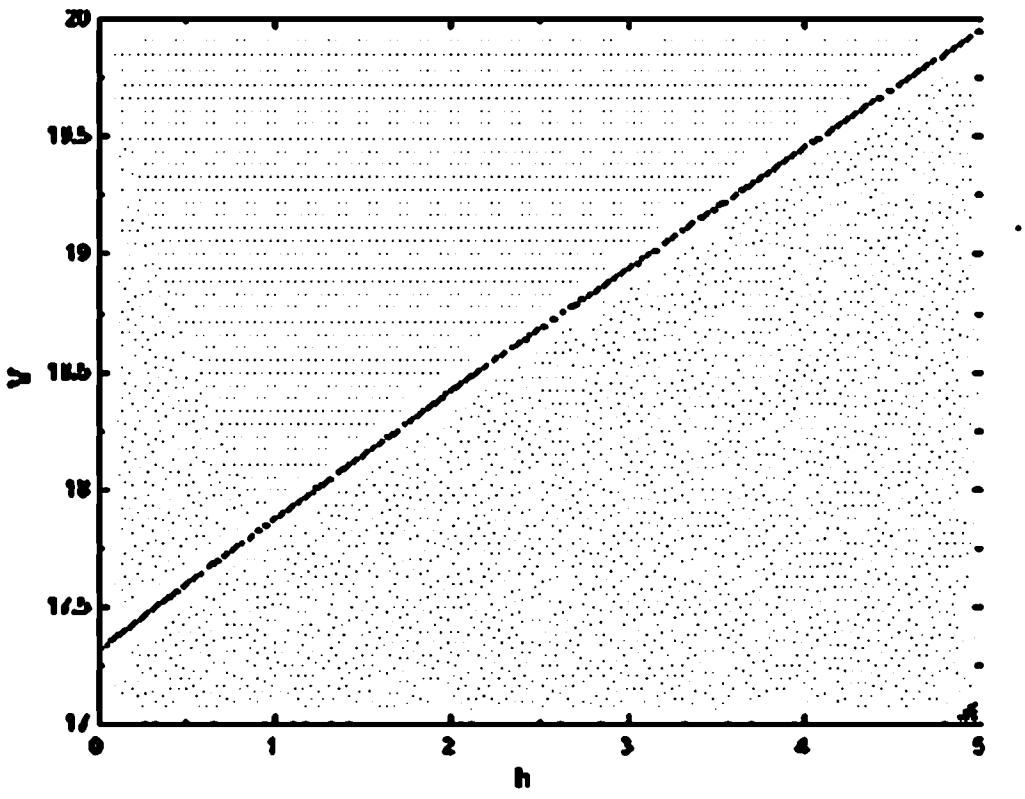
1..11	h	V̇	V
Run 1	0	0.136	17.32
Run 2	0.5	0.1382	17.6
Run 3	1	0.1404	17.88
Run 4	1.5	0.1426	18.15
Run 5	2	0.1447	18.42
Run 6	2.5	0.1467	18.68
Run 7	3	0.1488	18.94
Run 8	3.5	0.1508	19.2
Run 9	4	0.1528	19.46
Run 10	4.5	0.1548	19.71
Run 11	5	0.1567	19.95

Step 4 of 4

Select the plot on menu bar, then click x-y plot .Dialog box will appeared, then select variables to draw graph, then click ok button.

The graph will appeared

By plotting the graph between velocity and height:



Question:

12.034P

A hand held bicycle pump can be used as an atomizer to generate a fine mist of paint or pesticide by forcing air at a high velocity through a small hole and placing a short tube between the liquid reservoir and the high-speed air jet. The pressure across a subsonic jet exposed to the atmosphere is nearly atmospheric, and the surface of the liquid in the reservoir is also open to atmospheric pressure. In light of this, explain how the liquid is sucked up the tube. *Hint:* Read Sec. 12–1 carefully.

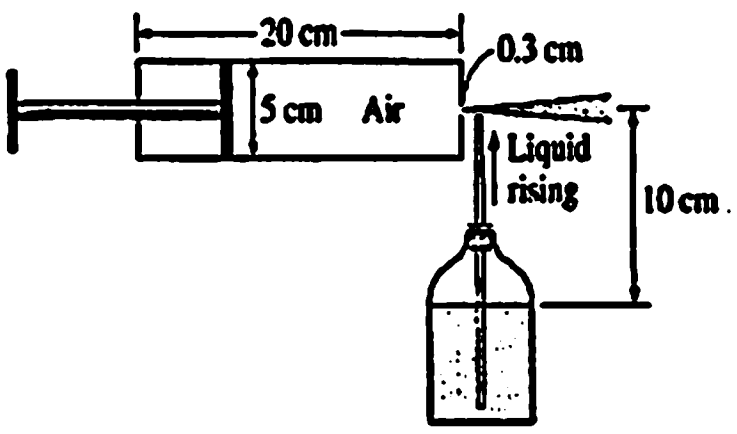
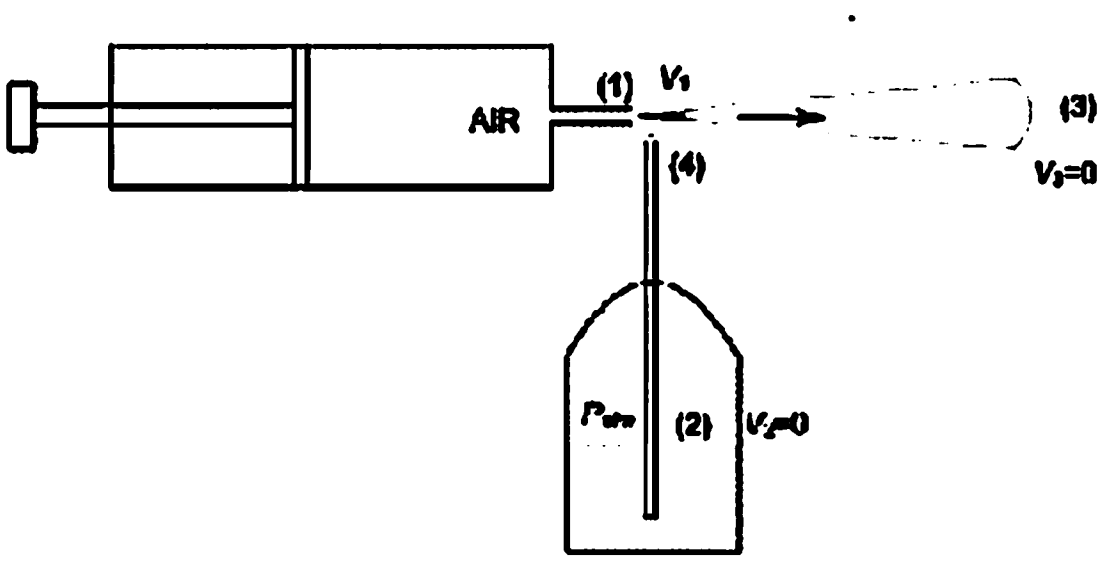


FIGURE P12–34

Step 1 of 3

Let us draw the figure:



Step 2 of 3

The flow is steady, incompressible and irrotational.
The frictional effects in the entire system are negligible.
The elevation difference between the points (1), (3) and (4) are negligible.

By considering points (2) and (4):
By applying the Bernoulli's flow equations between the two points:

$$\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2=\frac{P_4}{\rho g}+\frac{V_4^2}{2g}+z_4$$

$$\frac{P_{atm}}{\rho_{water} g}+\frac{0}{2g}+h=\frac{P_4}{\rho_{water} g}+\frac{0}{2g}+0$$

$$\frac{P_4}{\rho_{water} g}=\frac{P_{atm}}{\rho_{water} g}-h$$

$$P_4=P_{atm}-\rho_{water} g h$$

Hence, the pressure at point (4) must be less than the atmospheric pressure P_{atm} for the water to move up the tube.

Step 3 of 3

The low pressure at point (4) is created due to the high velocity air moving out from the orifice at point (1). Points (1) and (4) are at close proximity hence their pressures will be also equal to create the upward flow of water into the air flow
Therefore

$$P_1=P_4$$

$$=P_{atm}-\rho_{water} g h$$

Therefore, we can conclude that the presence of below atmospheric pressure at point (1) causes the liquid to be sucked up the tube.

Question: 12.035P

The water level in a tank is 15 m above the ground. A hose is connected to the bottom of the tank, and the nozzle at the end of the hose is pointed straight up. The tank cover is airtight, and the air pressure above the water surface is 3 atm gage. The system is at sea level. Determine the maximum height to which the water stream could rise.

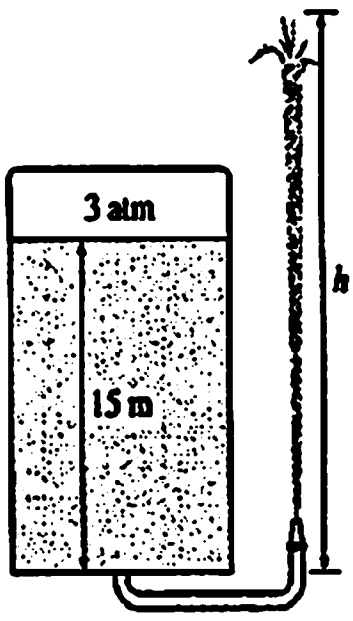


FIGURE P12–35

Step 1 of 4

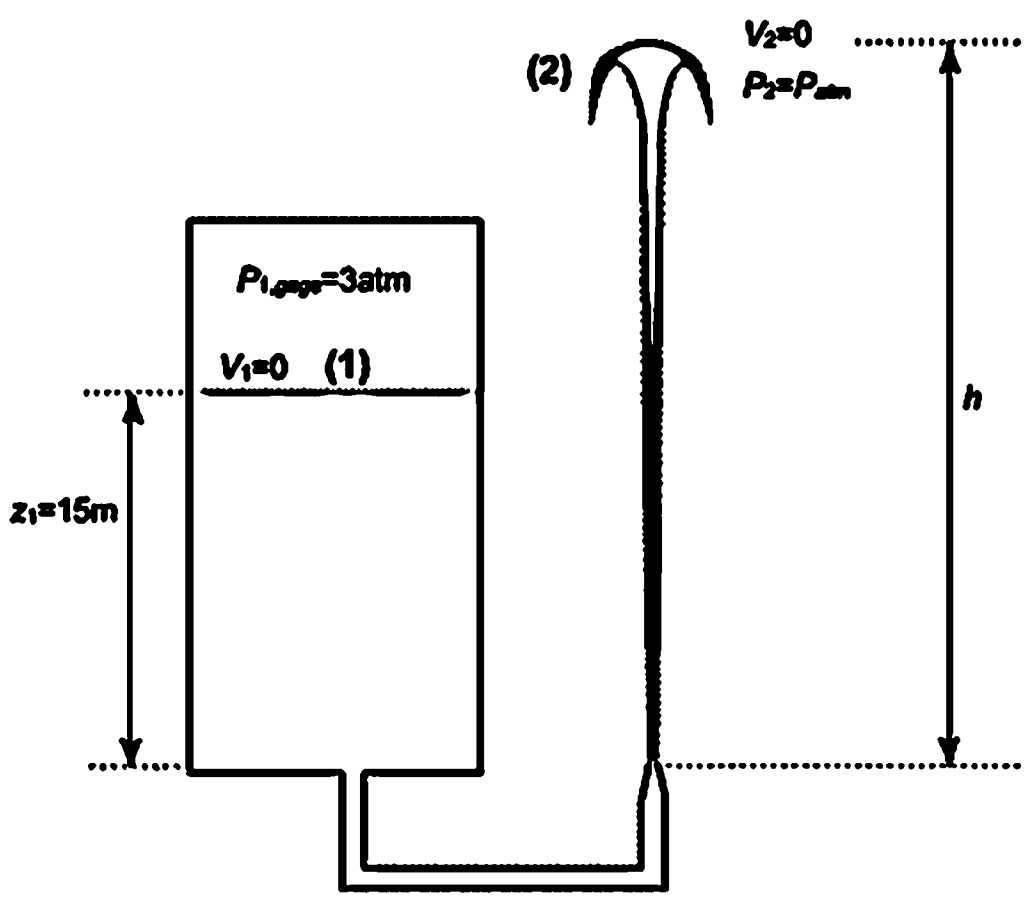
Given data:-

The height of water level in the tank, $h_1 = z_1 = 15$ m.

The pressure at the water surface in the tank, $P_{1,gage} = 3$ atm.

Step 2 of 4

Let us draw the figure.



Step 3 of 4

The pressure at point (1),

$P_1 = 3$ atm

$= 3.039\times10^5$ Pa

The atmospheric pressure,

$P_{atm} = 1.013\times10^5$ Pa

By applying the Bernoulli's equation between the points (1) and (2), we get

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

$$\frac{P_1}{\rho g}+\frac{0}{2g}+15=\frac{P_{atm}}{\rho g}+\frac{0}{2g}+h$$

Step 4 of 4

Then, maximum height of water rise to which the water stream could rise is

$$h=\left[\frac{1}{(1000)(9.81)}(P_1-P_{atm})\right]+15$$

$$=\left[\frac{1}{(1000)(9.81)}(P_{1,gage})\right]+15$$

$$=\left[\frac{1}{(1000)(9.81)}(3.039\times10^5)\right]+15$$

$$= 45.97 \text{ m}$$

Therefore, the height to which water rises is $\boxed{h = 45.97 \text{ m}}$.

Question: 12.036P

A Pitot-static probe connected to a water manometer is used to measure the velocity of air. If the deflection (the vertical distance between the fluid levels in the two arms) is 7.3 cm, determine the air velocity. Take the density of air to be 1.25 kg/m3.

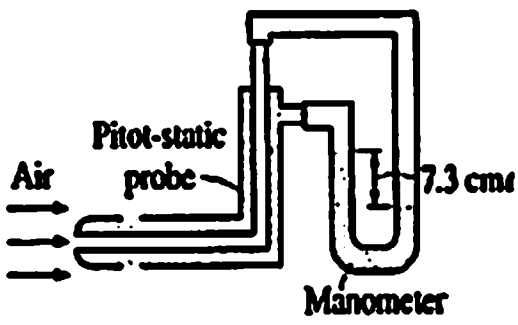


FIGURE P12-36

Step 1 of 4

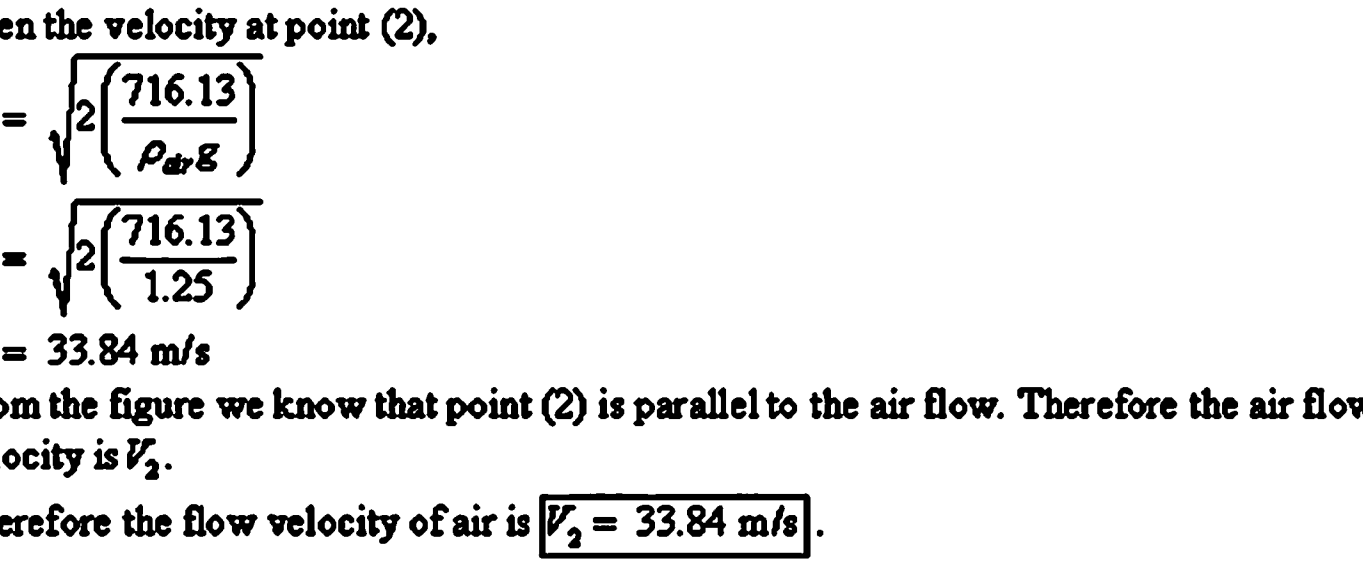
Given data:-

The density of air, $\rho_{air} = 1.25\ \text{kg/m}^3$.

The differential height of fluid in the probe, $h = 0.073\ \text{m}$.

Step 2 of 4

Let us draw the figure



Step 3 of 4

Consider point (1) which is the stagnation point.

The velocity, $V_1 = 0$

Apply the Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho_{air} g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g} + z_2$$

Substitute the required values and simplify.

$$\frac{P_1}{\rho_{air} g} + \frac{0}{2g} = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g}$$

$$V_2 = \sqrt{2\left(\frac{P_1 - P_2}{\rho_{air}}\right)}$$

The pressure difference between the two points can be expressed as,

$$\begin{aligned} P_1 - P_2 &= \rho_{water} g h \\ &= 1000(9.81)(0.073) \\ &= 716.13\ \text{Pa} \end{aligned}$$

Step 4 of 4

Then the velocity at point (2),

$$\begin{aligned} V_2 &= \sqrt{2\left(\frac{716.13}{\rho_{air} g}\right)} \\ &= \sqrt{2\left(\frac{716.13}{1.25}\right)} \\ &= 33.84\ \text{m/s} \end{aligned}$$

From the figure we know that point (2) is parallel to the air flow. Therefore the air flow velocity is V_2.

Therefore the flow velocity of air is $\boxed{V_2 = 33.84\ \text{m/s}}$.

Question: 12.037P

The air velocity in a duct is measured by a Pitot-static probe connected to a differential pressure gage. If the air is at 13.4 psia absolute and 70°F and the reading of the differential pressure gage is 0.15 psi, determine the air velocity.

Step 1 of 4

Given data:

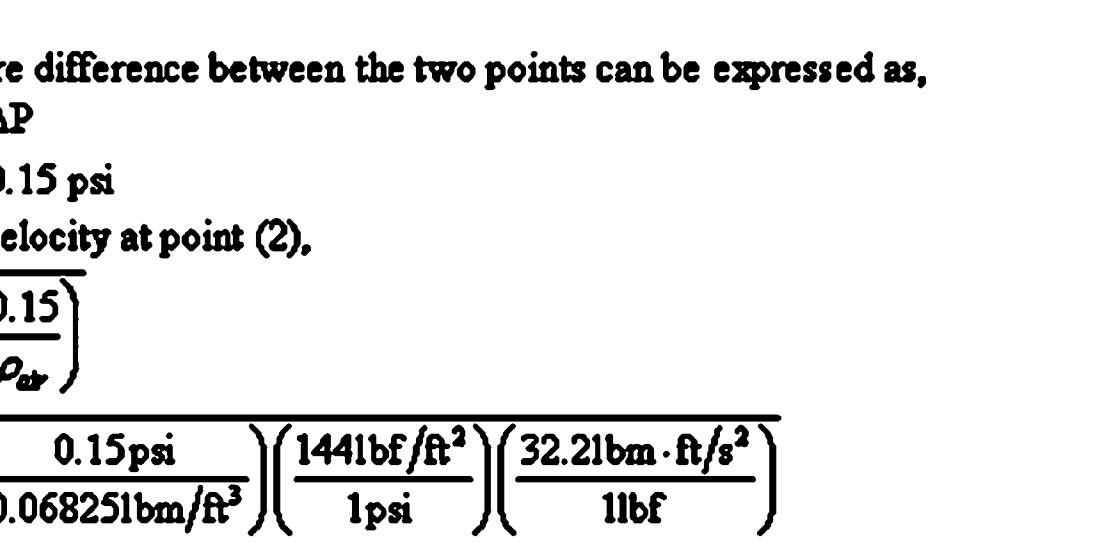
The absolute pressure of air, $P_{abs} = 13.4$ psia.

The air is at a temperature, $T = 70°F$.

The differential pressure gage reading, $\Delta P = 0.15$ psia.

Step 2 of 4

Let us draw the figure:



Step 3 of 4

We have,

$$P_{abs} = \rho_{air} RT$$

$$13.4 = \rho_{air}(0.3704)(70+460\text{R})$$

$$\rho_{air} = 0.06825 \text{ lbm/ft}^3$$

Consider point (1) which is the stagnation point.

The velocity, $V_1 = 0$

Apply the Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho_{air} g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g} + z_2$$

Substitute the required values and simplify.

$$\frac{P_1}{\rho_{air} g} + \frac{0}{2g} = \frac{P_2}{\rho_{air} g} + \frac{V_2^2}{2g}$$

$$V_2 = \sqrt{2\left(\frac{P_1 - P_2}{\rho_{air}}\right)}$$

Step 4 of 4

The pressure difference between the two points can be expressed as,

$$P_1 - P_2 = \Delta P$$

$$= 0.15 \text{ psi}$$

Then, the velocity at point (2),

$$V_2 = \sqrt{2\left(\frac{0.15}{\rho_{air}}\right)}$$

$$= \sqrt{2\left(\frac{0.15\text{psi}}{0.06825\text{lbm/ft}^3}\right)\left(\frac{144\text{lbf/ft}^2}{1\text{psi}}\right)\left(\frac{32.2\text{lbm}\cdot\text{ft/s}^2}{1\text{lbf}}\right)}$$

$$= 143 \text{ ft/s}$$

From the figure we know that point (2) is parallel to the air flow.

Therefore, the air flow velocity is $\boxed{V_2 = 143\text{ft/s}}$.

Question: 12.038P

In cold climates, water pipes may freeze and burst if proper precautions are not taken. In such an occurrence, the exposed part of a pipe on the ground ruptures, and water shoots up to 42 m. Estimate the gage pressure of water in the pipe. State your assumptions and discuss if the actual pressure is more or less than the value you predicted.

Step 1 of 2

Given data:-

The maximum height to which water shoots up, $h = 42\ \text{m}$.

The density of water is at maximum at 4°C or 277.14 K is beyond which it becomes less dense leading to the solidification into ice at 0°C with corresponding expansion. This expansion during this process is the cause of pipe rupture in cold climates.

Step 2 of 2

The density of water at temperature 273.14 K,

$\rho_{water} = 999.8395\ \text{kg/m}^3$

The water rises to a height of 42 m due to the pressure in the line.

Then we have,

$$P_{line} = \rho_{water} g h$$
$$= 999.9395 \times 9.81 \times 42$$
$$= 411.953\ \text{kPa}$$

Therefore, the gage pressure in the line is $\boxed{P_{line} = 411.953\ \text{kPa}}$.

Question: 12.039P

What is irreversible head loss? How is it related to the mechanical energy loss?

Step 1 of 2

Irreversible head loss (h_L) is the loss in head due to the effect of irreversibilities, like friction in piping, and it does not include the losses that occur within the pump or turbine.

Mechanical energy loss is the loss of head due to irreversibilities such as friction in pump or turbine. Mechanical energy loss is dissipated in the form of thermal energy.

Step 2 of 2

The relation between Irreversible head loss (h_L) and mechanical energy loss is expressed by the relation,

$$h_L = \frac{e_{mech\ loss}}{g}$$
$$= \frac{\dot{E}_{mech\ loss}}{\dot{m}g}$$

Here, $e_{mech\ loss}$ is the total mechanical energy loss, $\dot{E}_{mech\ loss}$ is the total mechanical power loss, and $\dot{m}$ is the mass flow rate.

Question: 12.040P

What is useful pump head? How is it related to the power input to the pump?

Step 1 of 1

Useful pump head

The useful power given as input to a pump, which expresses as fluid head can be termed as useful pump head.

Useful pump head and power input can be related by the following equation,

$$h_{pump} = \frac{w_{pump,u}}{g} = \frac{\dot{W}_{pump,u}}{\dot{m}g}$$

Where,

$h_{pump,u}$ is the useful pump head.

$w_{pump,u}$ is the useful work input.

$\dot{W}_{pump,u}$ is the useful power input.

Question: 12.041P

What is the kinetic energy correction factor? Is it significant?

Step 1 of 1

Kinetic energy correction factor (α)

The kinetic energy obtained from average velocity and that obtained from actual velocity is not equal to each other. To correct this error the kinetic energy term $(V^2/2)$ in the energy equation is replaced by the term $\alpha V_{avg}^2/2$.

Where,

α is the kinetic energy correction factor.

V_{avg} is the average velocity.

The energy equation when the kinetic energy correction factor is included,

For a pump turbine system

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$

Where,

$h_{pump,u}$ is the useful pump head.

$h_{turbine,e}$ is the extracted head removed from the fluid by the turbine.

h_L is the irreversible head loss.

Question: 12.042P

The water level in a tank is 20 m above the ground. A hose is connected to the bottom of the tank, and the nozzle at the end of the hose is pointed straight up. The water stream from the nozzle is observed to rise 25 m above the ground. Explain what may cause the water from the hose to rise above the tank level.

Step 1 of 2

Given data:-

The water surface is at a height of 20 m from the ground level.

The water stream from the nozzle rises to a height of 25 m from the ground level.

Step 2 of 2

We know that the maximum height to which water can rise from the nozzle of the hose must be equal to the height of the water surface which is 20 m

In the present situation the water rises to 25 m . This is possible only if the water is pressurized above atmospheric pressure at some point.

This can be achieved by

- Pressurizing at the water surface inside the tank.
- Installing a pumping unit along the flow before the exit which will pressurize the water flow

Thus we can conclude that the difference in height between the water jet rise from the hose and the tank water level surface is due to pressurization.

Question: 12.043P

A person is filling a knee-high bucket with water using a garden hose and holding it such that water discharges from the hose at the level of his waist. Someone suggests that the bucket will fill faster if the hose is lowered such that water discharges from the hose at the knee level. Do you agree with this suggestion? Explain. Disregard any frictional effects.

Step 1 of 2

It is given that the frictional effects are neglected.
We have the Bernoulli's equation

$$\frac{P_1}{\rho_{water} g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho_{water} g} + \frac{V_2^2}{2g} + z_2 \qquad \text{..... (1)}$$

Let us set the ground level as the reference position, then $z_1 = 0$

Let the pressure at exit P_2 be equal to the atmospheric pressure.

Then re-arranging the equation (1) for exit velocity

$$\frac{P_1}{\rho_{water} g} + \frac{V_1^2}{2g} + 0 = \frac{P_2}{\rho_{water} g} + \frac{V_2^2}{2g} + z_2$$

$$\frac{V_2^2}{2} = \frac{P_1 - P_{atm}}{\rho_{water}} + \frac{V_1^2}{2} - z_2$$

$$V_2 = \sqrt{2\left(\frac{P_1 - P_{atm}}{\rho_{water}} + \frac{V_1^2}{2} - z_2\right)} \qquad \text{..... (2)}$$

Step 2 of 2

From the above expression (2) for exit velocity it is evident that the velocity will decrease with the increase in the elevation z_2.

We know that mass low rate,

$\dot{m} = A_{pipe} V_2$

Where A_{pipe} is the area of the pipe

Thus it is evident that the mass flow decreases with increasing elevation.

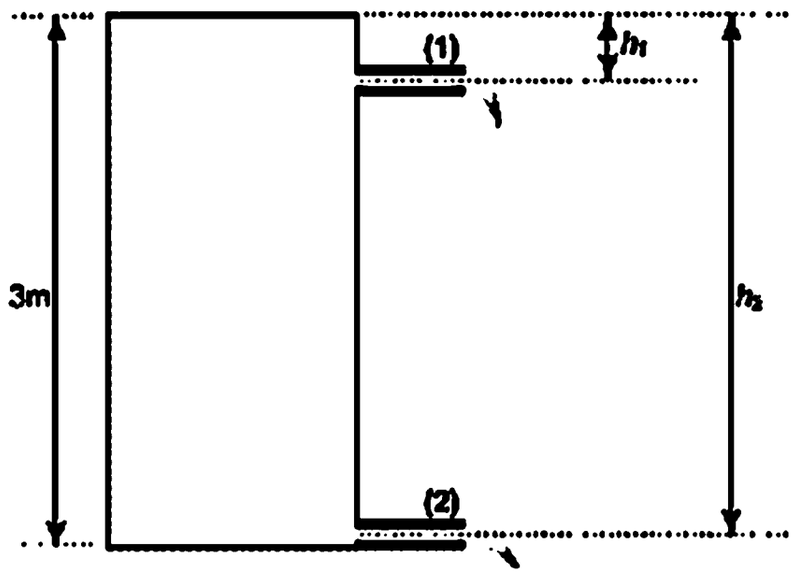
Therefore if the hose is held at knee level rather than at the waist level of the person the bucket will fill faster due to increased discharge.

Question: 12.044P

A 3-m-high tank filled with water has a discharge valve near the bottom and another near the top. (*a*) If these two valves are opened, will there be any difference between the discharge velocities of the two water streams? (*b*) If a hose whose discharge end is left open on the ground is first connected to the lower valve and then to the higher valve, will there be any difference between the discharge rates of water for the two cases? Disregard any frictional effects.

Step 1 of 3

Figure representing the 3 m high tank



Step 2 of 3

(a)

The relation for pressure is expressed as

$P = \rho g h$

Where h is height of the water column above the point considered

Since we know that with increase in applied pressure the flow rate through an orifice increases, we can conclude that as the height h decreases the flow rate will also decrease

Both the valves at (1) and (2) are open

We know from the figure that $h_1 < h_2$

Then the velocities at the valves is $V_1 < V_2$

Therefore we can say that the discharge velocities will be different at valves (1) and (2).

Step 3 of 3

(b)

The discharge rate is expressed as $\dot{m} = AV$

Where V is the velocity and A is the area

Then the flow rates at the valves will be $\dot{m}_1 < \dot{m}_2$

Because the velocities at the valves are $V_1 < V_2$

The mass flow rate of water through the hose when connected to valve (1) will be less than mass flow rate when connected to valve (2).

Question: 12.045P

In a hydroelectric power plant, water flows from an elevation of 240 ft to a turbine, where electric power is generated. For an overall turbine-generator efficiency of 83 percent, determine the minimum flow rate required to generate 100 kW of electricity.

Step 1 of 3

Given data:

Elevation from which the water flows, $H = 240$ ft

Overall turbine-generator efficiency, $\eta = 83\%$

Minimum power to be generated by the plant, $P = 100$ KW

Density of water, $\rho = 62.2$ lb/ft^3

Step 2 of 3

The power to be generated is $P = 100$ KW

From the conversion table we get 1 KW $= 737.56$ lbf.ft/s

Thus using the conversion factor we get the power in English units as

$P = 100 \times 737.56$ lbf.ft/s

$= 73756$ lbf.ft/s

Let us first calculate the flow rate of water using the formula for efficiency of a turbine and then determine the mass flow rate. Thus the relation for efficiency of a turbine is

$$\eta_t = \frac{P}{WQH}$$

Re-arranging for Q we get

$$Q = \frac{P}{WH\eta_t} \qquad \text{..... (1)}$$

Step 3 of 3

Now substituting the given values in the equation (1) we get

$$Q = \frac{P}{WH\eta_t}$$

$$= \frac{73756}{62.2 \times 240 \times 0.83}$$

$$= \frac{73756}{12390.24}$$

$$= 5.95 \text{ ft}^3/\text{s}$$

The mass flow rate is expressed as

$$\dot{m} = \rho AV$$

$$= \rho Q$$

$$= 62.2 \times 5.95$$

$$= 370.26 \text{ lbm/s}$$

Therefore the mass flow rate of water is $\boxed{\dot{m} = 370.26 \text{ lbm/s}}$

Question: 12.046P

Reconsider Prob. 12-45E. Determine the flow rate of water if the irreversible head loss of the piping system between the free surfaces of the source and the sink is 36 ft.

Step 1 of 4

Elevation from which the water flows, $H = 240$ ft
Overall turbine-generator efficiency, $\eta = 83\%$
Minimum power to be generated by the plant, $P = 100$ KW
Density of water, $\rho = 62.2$ lb/ft^3
Head loss of the system from source to the sink, $h_L = 36$ ft

Step 2 of 4

Applying the steady head form of the energy equation we get

$$\frac{P_1}{\rho g}+\alpha_1\left(\frac{V_1^2}{2g}\right)+z_1-h_{turbine}=\frac{P_2}{\rho g}+\alpha_2\left(\frac{V_2^2}{2g}\right)+z_2+h_L$$

Since the water level at the source and the sink remain constant so $V_1 = V_2 \approx 0$ and also the pressure at the source and sink are equal to the atmospheric pressure $P_1 = P_2 = P_{atm}$.

Hence the above equation reduces to

$$\frac{P_{atm}}{\rho g}+\alpha_1(0)+z_1-h_{turbine}=\frac{P_{atm}}{\rho g}+\alpha_2(0)+z_2+h_L$$

$$h_{turbine} = (z_1 - z_2) - h_L$$
$$= 240 - 36$$
$$= 204 \text{ ft}$$

Step 3 of 4

The power to be generated is $P = 100$ KW
From the conversion table we get 1 KW = 737.56 lbf.ft/s
Thus using the conversion factor we get the power in English units as
$P = 100 \times 737.56$ lbf.ft/s
$= 73756$ lbf.ft/s

Let us first calculate the flow rate of water using the formula for efficiency of a turbine and then determine the mass flow rate. Thus the relation for efficiency of a turbine is

$$\eta_t = \frac{P}{WQh_{turbine}}$$

Re-arranging for Q we get

$$Q = \frac{P}{Wh_{turbine}\eta_t} \qquad \text{..... (1)}$$

Step 4 of 4

Now substituting the given values in the equation (1) we get

$$Q = \frac{P}{Wh_{turbine}\eta_t}$$
$$= \frac{73756}{62.2 \times 204 \times 0.83}$$
$$= \frac{73756}{10531.704}$$
$$= 7 \text{ ft}^3/\text{s}$$

The mass flow rate is expressed as

$$\dot{m} = \rho AV$$
$$= \rho Q$$
$$= 62.2 \times 7$$
$$= 435.6 \text{ lbm/s}$$

Therefore the mass flow rate of water is $\boxed{\dot{m} = 435.6 \text{ lbm/s}}$

Question: 12.047P

A fan is to be selected to ventilate a bathroom whose dimensions are 2 m × 3 m × 3 m. The air velocity is not to exceed 8 m/s to minimize vibration and noise. The combined efficiency of the fan-motor unit to be used can be taken to be 50 percent. If the fan is to replace the entire volume of air in 10 min, determine (*a*) the wattage of the fan-motor unit to be purchased, (*b*) the diameter of the fan casing, and (*c*) the pressure difference across the fan. Take the air density to be 1.25 kg/m3 and disregard the effect of the kinetic energy correction factors.

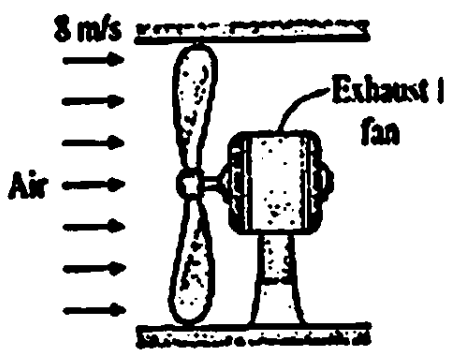


FIGURE P12–47

Step 1 of 4

(a)

The total volume of air (V) in the bathroom is as follows

$$V = 2\text{ m} \times 3\text{ m} \times 3\text{ m}$$
$$= 18\text{ m}^3$$

Use this volume to calculate the volume flow rate ($\dot{V}$) using the time it takes to replace the entire volume of air (t).

$$\dot{V} = \frac{V}{t}$$

Substitute $18\,\text{m}^3$ for V and $10\,\text{min}$ for t in the above equation and obtain the flow rate as follows

$$\dot{V} = \frac{18\text{ m}^3}{10\text{ min}} \times \frac{1\text{ min}}{60\text{ s}}$$
$$= 0.03\text{ m}^3/\text{s}$$

Next calculate the mass flow rate ($\dot{m}$) using the density of air as follows

$$\dot{m} = \rho\dot{V}$$

Substitute $1.25\,\text{kg/m}^3$ for ρ and $0.03\,\text{m}^3/\text{s}$ for $\dot{V}$ in the above relation

$$\dot{m} = 1.25\text{ kg/m}^3 \times 0.03\text{ m}^3/\text{s}$$
$$= 0.0375\text{ kg/s}$$

Step 2 of 4

Apply the energy equation for steady, incompressible flow relating pressure (P), velocity (V), and height (z) between the inlet side of the fan (point 1) and the exit side of the fan (point 2).

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + z_1 g\right) + \dot{W}_{pump} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + z_2 g\right) + \dot{W}_{turbine} + \dot{E}_{mech,loss}$$

Because both points are open to the atmosphere both P_1 and P_2 are equal to the atmospheric pressure. Notice that both points are at the same height, so $z_1 = z_2$.

Also, consider point 1 far enough from the fan so that the flow velocity is zero. The effective power of the fan is equal to the difference between the pump and the mechanical loss.

$$\dot{W}_{pump} - \dot{E}_{mech,loss} = \dot{m}\alpha_2\frac{V_2^2}{2}$$

$$\dot{W}_{fan,eff} = \dot{m}\alpha_2\frac{V_2^2}{2}$$

Recall that the electric power rating of the fan is equal to the effective power divided by the power rating.

$$\dot{W}_{fan,elect} = \frac{1}{\eta}\dot{m}\alpha_2\frac{V_2^2}{2}$$

Substitute $0.0375\,\text{m}^3/\text{s}$ for $\dot{m}$, $8\,\text{m/s}$ for V_2 and 0.5 for η in the above equation. Recall that, since the effect of the kinetic energy correction factors is negligible, $\alpha_2 = 1$

$$\dot{W}_{fan,elect} = \frac{1}{0.5} \times 0.0375\text{ kg/s} \times 1 \times \frac{(8\text{ m/s})^2}{2}$$
$$= 2.4\text{ W}$$

Therefore, the electric power rating of the fan is $2.4\,\text{W}$.

Step 3 of 4

(b)

Define the volume flow rate as calculated previously with the diameter of the fan (D) and the velocity flow (V) as follows

$$\dot{V} = A_2 V_2$$
$$= \pi\left(\frac{D}{2}\right)^2 V_2$$

Plug in the given velocity and the calculated volume flow rate. Solve for the diameter using the relation below

$$D = \sqrt{\frac{4\dot{V}}{\pi V_2}}$$

Substitute $0.03\,\text{m}^3/\text{s}$ for $\dot{V}$ and $8\,\text{m/s}$ for V_2 in the above relation

$$D = \sqrt{\frac{4 \times 0.03\text{ m}^3/s}{\pi \times 8\text{ m/s}}}$$
$$= 6.9\text{ cm}$$

In order for the mean velocity of air to remain below the specified value, the diameter of the fan casing can only be at most $6.9\,\text{cm}$.

Step 4 of 4

(c)

Consider now a point 3 immediately before the fan and a point 4 immediate after the fan. Apply the energy equation as follows

$$\dot{m}\left(\frac{P_3}{\rho} + \alpha_3\frac{V_3^2}{2} + z_3 g\right) + \dot{W}_{pump} = \dot{m}\left(\frac{P_4}{\rho} + \alpha_4\frac{V_4^2}{2} + z_4 g\right) + \dot{W}_{turbine} + \dot{E}_{mech,loss}$$

Note that both points are at the same height and have the same flow velocity. Solve for the difference in pressures.

$$\dot{m}\frac{P_3}{\rho} + \dot{W}_{fan,eff} = \dot{m}\frac{P_4}{\rho}$$

$$P_4 - P_3 = \frac{\dot{W}_{fan,eff}}{\dot{m}/\rho}$$
$$= \frac{\dot{W}_{fan,eff}}{\dot{V}}$$

Plug in the calculated volume flow rate and effective fan power.

$$P_4 - P_3 = \frac{1.2\text{ W}}{0.03\text{ m}^3/\text{s}}$$
$$= 40\text{ Pa}$$

Therefore, the pressure difference across the fan is $40\,\text{Pa}$.

Question: 12.048P

Water is being pumped from a large lake to a reservoir 25 m above at a rate of 25 L/s by a 10-kW (shaft) pump.If the irreversible head loss of the piping system is 5 m,determine the mechanical efficiency of the pump.

Step 1 of 3

Given data:-

Height of the tank from the reservoir, $z_2 = 25\ \text{m}$

The flow rate, $\dot{V} = 25\ \text{L/s} = 0.025\ \text{m}^3/\text{s}$

The power input for the pump, $\dot{W}_{shaft,in} = 10\ \text{kW}$

The irreversible head loss for the piping system, $h_L = 5\ \text{m}$

Step 2 of 3

The mass flow rate of water is determined by using the relation

$$\begin{aligned}\dot{m} &= \rho\dot{V}\\ &= \left(1000\ \text{kg/m}^3\right)\left(25\times10^{-3}\ \text{m}^3/\text{s}\right)\\ &= 25\ \text{kg/s}\end{aligned}$$

The relation for mass flow rate can be expressed as

$$\dot{m} = \frac{\dot{W}_{pump,in}}{gz_2 + gh_L}$$

$$\dot{W}_{pump,in} = \left(gz_2 + gh_L\right)\dot{m}$$

Substituting the values of $z_2 = 25\ \text{m}$, $h_L = 5\ \text{m}$ and $\dot{m} = 25\ \text{kg/s}$ in the above relation we get

$$\begin{aligned}\dot{W}_{pump,in} &= g(25+5)25\\ &= 9.81(25+5)25\\ &= 7357.5\ \text{W}\end{aligned}$$

Step 3 of 3

Thus the efficiency of the pump is determined by

$$\begin{aligned}\eta_{pump} &= \frac{\dot{W}_{pump,in}}{\dot{W}_{shaft,in}}\\ &= \frac{7357.5}{10\times10^3}\\ &= 0.73575\\ &= 73.57\ \%\end{aligned}$$

Therefore the efficiency of the pump is $\boxed{\eta_{pump} = 73.57\ \%}$

Question: 12.049P

Reconsider Prob. 12–48. Using EES (or other) software, investigate the effect of irreversible head loss on the mechanical efficiency of the pump. Let the head loss vary from 0 to 15 m in increments of 1 m. Plot the results, and discuss them.

Step 1 of 4

Given data:

Height of the tank from the reservoir, $z_2 = 25\ \text{m}$

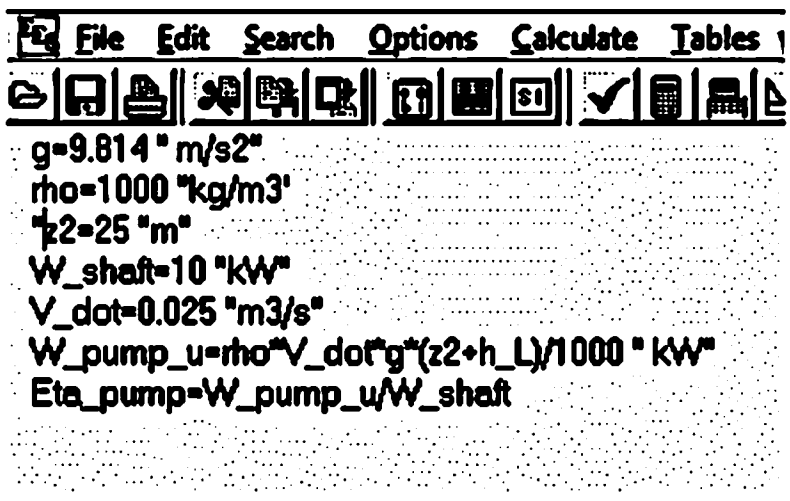
The flow rate, $\dot{V} = 25\ \text{L/s} = 0.025\ \text{m}^3/\text{s}$

The power input for the pump, $\dot{W}_{shaft,in} = 10\ \text{kW}$

The irreversible head loss for the piping system, $h_L = 5\ \text{m}$

Step 2 of 4

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.



Step 3 of 4

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed;

Parametric Table

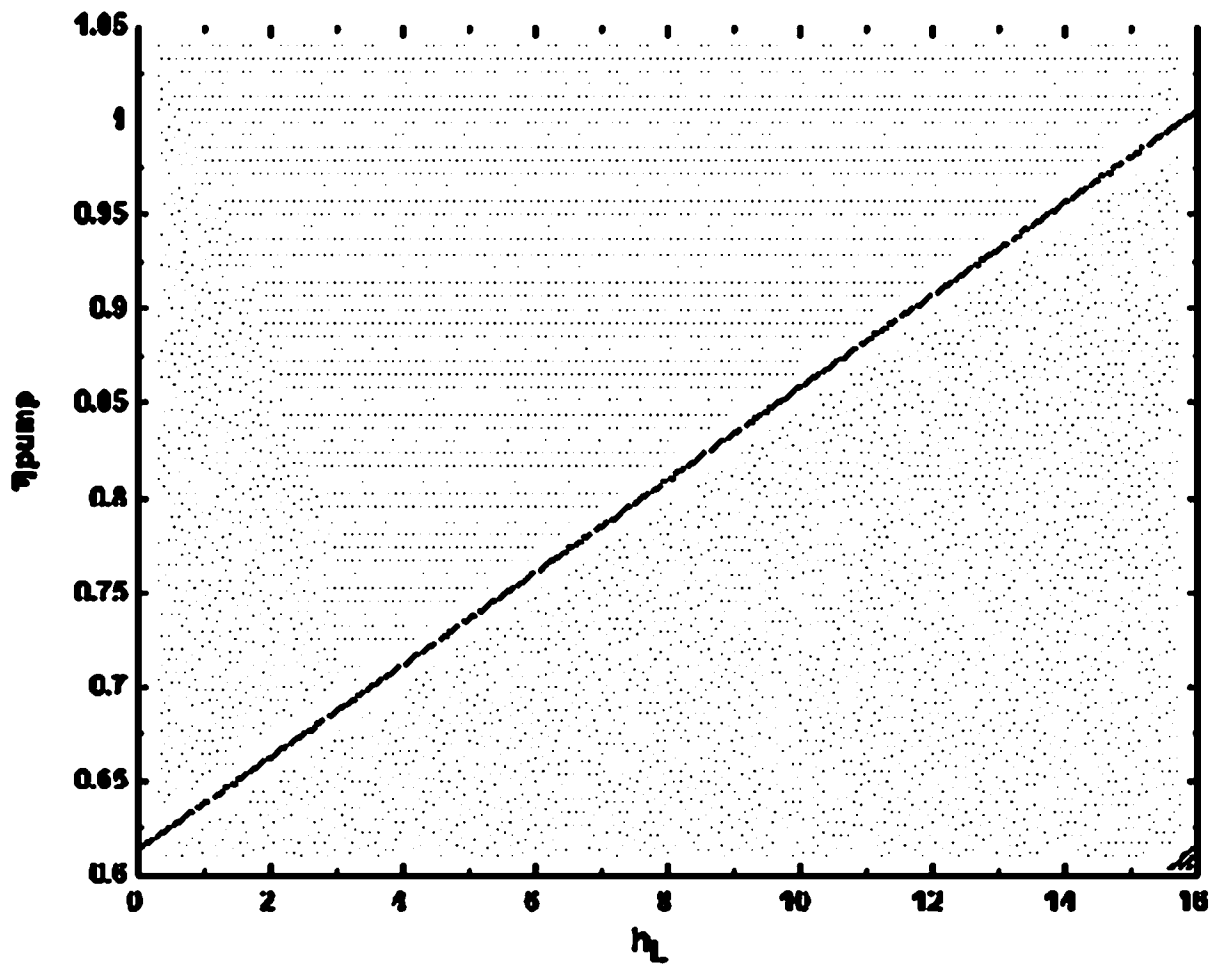
Table 1 | Table 2 | Table 3 | Table 4

1..16	η_{pump}	h_L	$W_{pump,u}$
Run 1	0.6134	0	6.134
Run 2	0.6395	1.067	6.395
Run 3	0.6657	2.133	6.657
Run 4	0.6919	3.2	6.919
Run 5	0.7181	4.267	7.181
Run 6	0.7442	5.333	7.442
Run 7	0.7704	6.4	7.704
Run 8	0.7966	7.467	7.966
Run 9	0.8227	8.533	8.227
Run 10	0.8489	9.6	8.489
Run 11	0.8751	10.67	8.751
Run 12	0.9013	11.73	9.013
Run 13	0.9274	12.8	9.274
Run 14	0.9536	13.87	9.536
Run 15	0.9798	14.93	9.798
Run 16	1.006	16	10.06

Step 4 of 4

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

By plotting the graph between efficiency of pump and head loss:



Question: 12.050P

A 7-hp (shaft) pump is used to raise water to a 15-m higher elevation. If the mechanical efficiency of the pump is 82 percent, determine the maximum volume flow rate of water.

Step 1 of 2

Convert the units of the input shat power,

$$\begin{aligned}\dot{W}_{shaft,in} &= 7\ \text{hp} \\ &= 7\,\text{hp} \times \left(\frac{745.69\,\text{W}}{1\,\text{hp}}\right) \\ &= 5219.83\,\text{W}\end{aligned}$$

Consider the relation for efficiency of the pump.

$$\eta_{pump} = \frac{\dot{W}_{pump,u}}{\dot{W}_{shaft,in}}$$

$$\dot{W}_{pump,u} = \eta_{pump}\left(\dot{W}_{shaft,in}\right)$$

Substitute 0.82 for η_{pump}, and $5219.83\,\text{W}$ for $\dot{W}_{shaft,in}$.

$$\begin{aligned}\dot{W}_{pump,u} &= (0.82)(5219.83) \\ &= 4280.26\,\text{W}\end{aligned}$$

Step 2 of 2

Calculate the mass flow rate

$$\dot{m} = \frac{\dot{W}_{pump,u}}{gz_2}$$

Here, g is acceleration due to gravity and z_2 is the elevation.

Substitute $15\,\text{m}$ for z_2 and $4280.26\,\text{W}$ for $\dot{W}_{pump,u}$.

$$\begin{aligned}\dot{m} &= \frac{4280.26}{9.81(15)} \\ &= 29.08\,\text{kg/s}\end{aligned}$$

Calculate the volume flow rate.

$$\dot{V} = \frac{\dot{m}}{\rho}$$

Substitute $29.08\,\text{kg/s}$ for $\dot{m}$ and $1000\,\text{kg/m}^3$ for ρ.

$$\begin{aligned}\dot{V} &= \frac{29.08}{1000} \\ &= 0.02908\ \text{m}^3/\text{s}\end{aligned}$$

Therefore the flow rate is $\boxed{0.02908\ \text{m}^3/\text{s}}$.

Question: 12.051P

Water flows at a rate of 0.035 m3/s in a horizontal pipe whose diameter is reduced from 15 cm to 8 cm by a reducer. If the pressure at the centerline is measured to be 480 kPa and 445 kPa before and after the reducer, respectively, determine the irreversible head loss in the reducer. Take the kinetic energy correction factors to be 1.05.

Step 1 of 3

Given data:

The flow rate, $\dot{V} = 0.035\ m^3/s$

The diameter of the reducer varies from 15 cm to 8cm

The pressure after the reducer, $P_2 = 480$ kPa

The pressure before the reducer is $P_1 = 445$ kPa

The kinetic energy correction factor, $\alpha = 1.05$

Let the flow velocity be V

The area be A

Step 2 of 3

We know that flow velocity

$$V = \frac{\dot{V}}{A}$$

So before the reducer the velocity where area is A_1 is

$$V_1 = \frac{\dot{V}}{A_1}$$

$$= \frac{0.035}{\frac{\pi}{4}(0.15)^2}$$

$$= 1.980\ m/s$$

After the reducer the velocity where area is A_2

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.035}{\frac{\pi}{4}(0.08)^2}$$

$$= 6.96\ m/s$$

Step 3 of 3

Let us apply the Bernoulli's equation for steady incompressible flow between the entrance and exit of the reducer. We get

$$\frac{P_1}{\rho g} + \alpha \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha \frac{V_2^2}{2g} + z_2 + h_L$$

We know that flow velocity

$$V = \frac{\dot{V}}{A}$$

So before the reducer the velocity where area is A_1 is

$$V_1 = \frac{\dot{V}}{A_1}$$

$$= \frac{0.035}{\frac{\pi}{4}(0.15)^2}$$

$$= 1.980\ m/s$$

After the reducer the velocity where area is A_2

$$V_2 = \frac{\dot{V}}{A_2}$$

Question: 12.052P

The water level in a tank is 20 m above the ground. A hose is connected to the bottom of the tank, and the nozzle at the end of the hose is pointed straight up. The tank is at sea level, and the water surface is open to the atmosphere. In the line leading from the tank to the nozzle is a pump, which increases the pressure of water. If the water jet rises to a height of 27 m from the ground, determine the minimum pressure rise supplied by the pump to the water line.

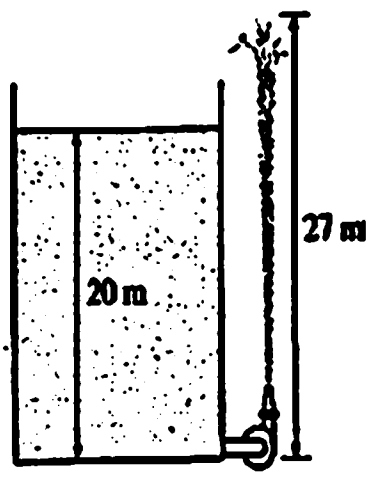


FIGURE P12-52

Step 1 of 5

Given data:

The height to which the water jet rises from the nozzle, $z_j = 27$ m.

The height of water surface in the tank from nozzle, $z_s = 20$ m.

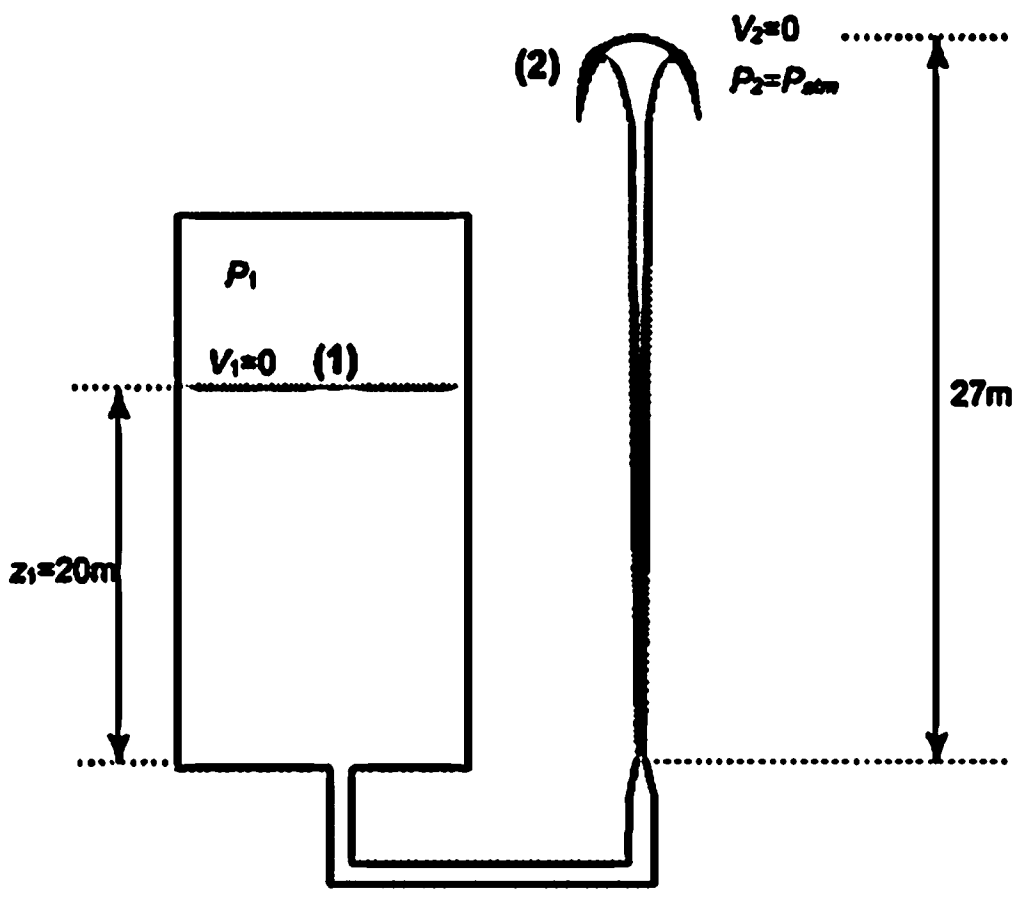
Step 2 of 5

Assumptions are

- The flow is steady, incompressible and irrotational.
- The frictional effects are neglected.
- The pressure outside is atmospheric pressure, P_{atm}.

Step 3 of 5

Figure showing a water tank with nozzle fitted at its bottom,



Step 4 of 5

The atmospheric pressure,

$$P_{atm} = P_2$$
$$= 1.013 \times 10^5 \text{Pa}$$

Applying the Bernoulli's equation between the points (1) and (2):

Then

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$\frac{P_1}{\rho g} + \frac{0}{2g} + 20 = \frac{P_{atm}}{\rho g} + \frac{0}{2g} + 27$$

Step 5 of 5

Then

$$\frac{1}{(1000)(9.81)}(P_1 - P_{atm}) = 27 - 20$$

$$\frac{1}{(1000)(9.81)}(P_{1,gage}) = 27 - 20$$

$$P_{1,gage} = 68670 \text{ Pa}$$

Therefore the gage pressure required to raise the water jet to a height of 27 m is

$\boxed{P_{1,gage} = 68670 \text{ Pa}}$.

$$\frac{1}{(1000)(9.81)}(P_1 - P_{atm}) = 27 - 20$$

$$\frac{1}{(1000)(9.81)}(P_{1,gage}) = 27 - 20$$

$$P_{1,gage} = 68670 \text{ Pa}$$

Therefore the gage pressure required to raise the water jet to a height of 27 m is

$\boxed{P_{1,gage} = 68670 \text{ Pa}}$.

Question: 12.053P

A hydraulic turbine has 85 m of head available at a flow rate of 0.25 m3/s. and its overall turbine-generator efficiency is 78 percent. Determine the electric power outputof this turbine.

Step 1 of 3

Given data:

The available head for the hydraulic turbine, $h_{turbine} = 85$ m

The flow rate is $\dot{V} = 0.25\ m^3/s$

The overall turbine generator efficiency, $\eta_{turbine-gen} = 78\ \%$

Step 2 of 3

We know that efficiency is given by

$$\eta_{turbine-gen} = \frac{\dot{W}_{electric}}{\dot{W}_{turbine,e}} \qquad \text{..... (1)}$$

Also the Electric power output,

$$\dot{W}_{electric} = \eta_{turbine-gen}\dot{W}_{turbine,e}$$

We have,

$$\eta_{turbine} = \frac{\dot{W}_{turbine,e}}{\dot{m}g}$$

$$\dot{W}_{turbine,e} = \eta_{turbine}\dot{m}g$$

$$= \eta_{turbine}\left(\dot{V}\rho\right)g$$

$$= (85)(0.25 \times 1000)(9.81)$$

$$= 208.46 \times 10^3\ W$$

Step 3 of 3

Substituting these values in equation (1) we have,

$$\dot{W}_{electric} = \eta_{turbine-gen}\dot{W}_{turbine,e}$$

$$= 0.78\left(208.46 \times 10^3\right)$$

$$= 162.60 \times 10^3\ W$$

Therefore the electric power output of the turbine is $\boxed{\dot{W}_{electric} = 162.6 \times 10^3\ W}$.

Question: 12.054P pg 1

An oil pump is drawing 25 kW of electric power while pumping oil with ρ =860 kg/m3 at a rate of 0.1m3sThe inlet and outlet diameters of the pipe are 8 cm and 12 cm respectively. If the pressure rise of oil in the pump is mea sured to be 250 kPa and the motor efficiency is 90 percent; determine the mechanical efficiency of the pump. Take the kinetic energy correction factor to be 1.05.

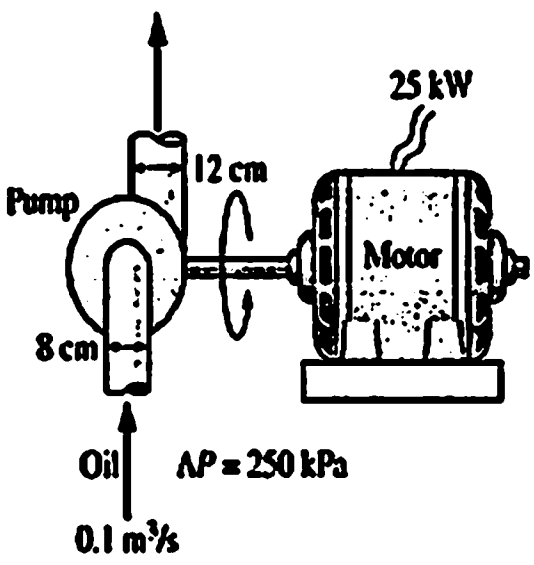


FIGURE P12–54

Step 1 of 6

Given data:

The input electric power, $\dot{W}_{electric,in} = 25\ \text{kW}$

The motor efficiency, $\eta_{motor} = 90\ \%$

The flow rate, $\dot{V} = 0.1\ \text{m}^3/\text{s}$

The pressure rise in the pump, $\Delta P = 250\ \text{kPa}$

The kinetic energy correction factor, $\alpha = 1.05$

The inlet diameter, $D_1 = 0.08\ \text{m}$

The outlet diameter, $D_2 = 0.12\ \text{m}$

Step 2 of 6

Calculation of mechanical efficiency of the pump:

In order to calculate the mechanical efficiency, let us calculate the change in mechanical energy and shaft work.

Let us set the inlet zone as point (1) and exit zone of the pump as point (2).

The inlet area,

Step 1 of 6

Given data:

The input electric power, $\dot{W}_{electric,in} = 25\ \text{kW}$

The motor efficiency, $\eta_{motor} = 90\ \%$

The flow rate, $\dot{V} = 0.1\ \text{m}^3/\text{s}$

The pressure rise in the pump, $\Delta P = 250\ \text{kPa}$

The kinetic energy correction factor, $\alpha = 1.05$

The inlet diameter, $D_1 = 0.08\ \text{m}$

The outlet diameter, $D_2 = 0.12\ \text{m}$

Step 2 of 6

Calculation of mechanical efficiency of the pump:

In order to calculate the mechanical efficiency, let us calculate the change in mechanical energy and shaft work.

Let us set the inlet zone as point (1) and exit zone of the pump as point (2).

The inlet area,

$$A_1 = \frac{\pi}{4} D_1^2$$

$$= \frac{\pi}{4} \times 0.08^2$$

$$= 5.026 \times 10^{-3}\ \text{m}^2$$

The exit area,

$$A_2 = \frac{\pi}{4} D_2^2$$

$$= \frac{\pi}{4} \times 0.12^2$$

$$= 11.30 \times 10^{-3}\ \text{m}^2$$

Step 3 of 6

We have,

$$\dot{V} = A_2 V_2$$

$$0.1 = \left(11.30 \times 10^{-3}\right) \times V_2$$

$$V_2 = 8.849\ \text{m/s}$$

And

$$\dot{V} = A_1 V_1$$

$$0.1 = \left(5.026 \times 10^{-3}\right) V_2$$

$$V_1 = 19.896\ \text{m/s}$$

Step 4 of 6

Let us apply the energy equation between the two points.

Step 4 of 6

Let us apply the energy equation between the two points.

Then,

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{turbine,e}+h_L$$

$$\alpha_1\frac{V_1^2}{2g}-\alpha_2\frac{V_2^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}-\frac{P_1}{\rho g}+z_2+h_{turbine,e}+h_L$$

The difference in elevation between the inlet (1) and exit (2) is negligible,

$$z_1-z_2=0.$$

And the head loss due to turbine is also negligible.

Now substitute the required values,

$$\alpha\left(\frac{V_1^2}{2g}-\frac{V_2^2}{2g}\right)+h_{pump,u}=\frac{1}{\rho g}(P_2-P_1)$$

$$\frac{1.05}{2(9.81)}\left(19.896^2-8.849^2\right)+h_{pump,u}=\frac{1}{860(9.81)}\left(250\times10^3\right)$$

$$h_{pump,u}=12.63\text{ m}$$

Step 5 of 6

We know that,

The efficiency of a motor,

$$\eta_{motor}=\frac{\dot{W}_{shaft}}{\dot{W}_{electric,in}}$$

$$\dot{W}_{shaft}=\eta_{motor}\dot{W}_{electric,in}$$

$$=0.90\left(25\times10^3\right)$$

$$=22.5\times10^3\text{ W}$$

The change in mechanical energy,

$$\Delta E=\dot{m}h_{pump,u}$$

The mechanical efficiency of the pump,

$$\eta_{pump}=\frac{\Delta\dot{E}_{mech}}{\dot{W}_{shaft}}$$

$$=\frac{1086.18}{\left(22.5\times10^3\right)}$$

$$=0.048$$

$$=4.8\%$$

Therefore, the mechanical efficiency of the pump is, $\boxed{\eta_{pump}=4.8\%}$

Question: 12.055P

Water flows at a rate of 20 L/s through a horizontal pipe whose diameter is constant at 3 cm. The pressure drop across a valve in the pipe is measured to be 2 kPa, as shown in Fig PI2-55. Determine the irreversible head loss of the valve, and the useful pumping power needed to overcome the resulting pressure drop.

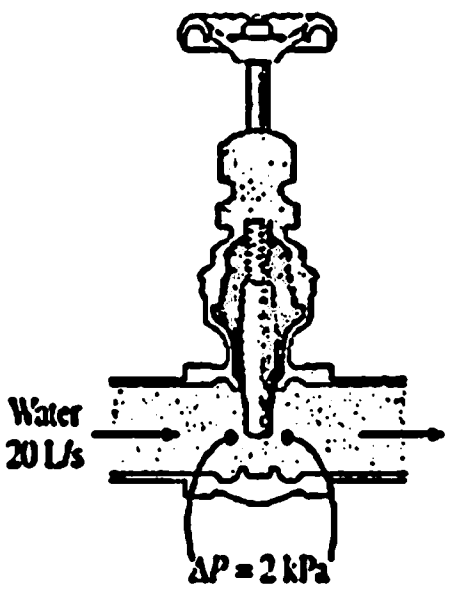
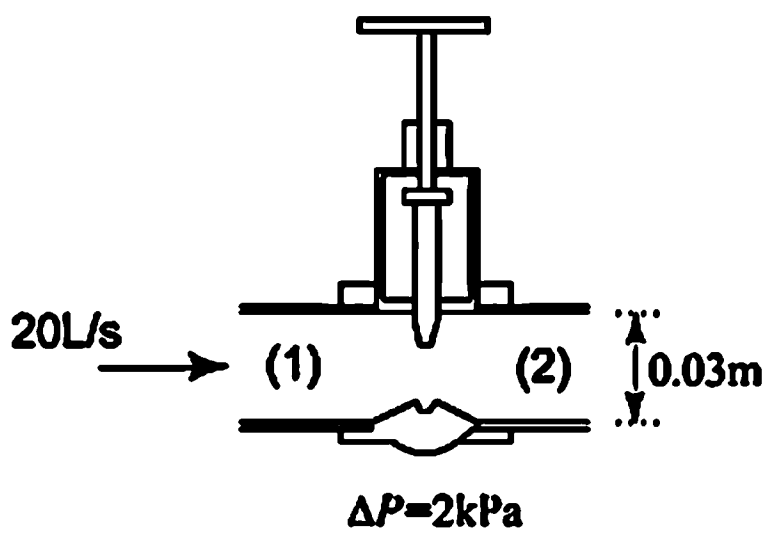


FIGURE P12–55

Step 1 of 5

Figure showing the flow of fluid through the pipe,



Step 2 of 5

Apply Bernoulli's equation for steady incompressible flow between the points (1) and (2):

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{turbine,e}+h_L$$

Here, head loss in pump is $h_{pump,u}$, head loss in turbine is $h_{turbine,e}$, the irreversible head loss h_L, the pressure at section 1 and 2 are P_1, P_2 respectively, velocity at section 1 and 2 are V_1, V_2 respectively and , height of the flow from datum at section 1 and 2 are z_1, z_2 respectively.

Step 3 of 5

Since the head loss in pump and turbine are absent,

$$h_{turbine,e}=h_{pump,u}=0$$

There is no change in height between sections (1) and (2) and the flow occurs in a horizontal path hence,

$$z_1=z_2$$

There is no change in velocity at section 1 and 2, since the diameter of the pipe remains constant.

$$V_1=V_2$$

From the mentioned conditions the Bernoulli equation modifies as,

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{turbine,e}+h_L$$

$$\frac{P_1}{\rho g}=\frac{P_2}{\rho g}+h_L$$

$$h_L=\frac{1}{\rho g}(P_1-P_2)$$

$$h_L=\frac{1}{\rho g}(\Delta P) \qquad \text{..... (1)}$$

Step 4 of 5

Substitute 2 kPa for ΔP, 9.81 m/s^2 for g and 1000 kg/m^3 for ρ in the equation (1),

$$h_L=\frac{1}{\rho g}(\Delta P)$$

$$=\frac{1}{(1000)(9.81)}(2\times10^3)$$

$$=0.2038 \text{ m}$$

Therefore the irreversible head loss h_L is $\boxed{0.2038 \text{ m}}$.

Step 5 of 5

Find the power required to overcome this pressure loss using the formula,

$$W_{pump}=\dot{m}\rho h_L$$

Here, mass flow rate is denoted by $\dot{m}$, density of water by ρ and, head loss by h_L.

Substitute 20 L/s for $\dot{m}$, 1000 kg/m^3 for ρ and 0.2038 m for h_L in the equation,

$$W_{pump}=\dot{m}\rho g h_L$$

$$=20 \text{ (L/s)}\times1000 \text{ kg/m}^3\times9.81 \text{ m/s}^2\times0.2038 \text{ m}$$

$$=\frac{20}{1000}\text{ (m}^3\text{/s)}\times1000 \text{ kg/m}^3\times9.81 \text{ m/s}^2\times0.2038 \text{ m}\times\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

$$=0.03998 \text{ kW}$$

$$=39.98 \text{ W}$$

Therefore, the power required to overcome the pressure loss is $\boxed{39.98 \text{ W}}$.

Question: 12.056P

The water level in a tank is 66 ft above the ground. a hose is connected to the bottom of the tank at the ground level and the nozzle at the end of the hose is pointed straight up The tank cover is airtight, but the pressure over the water surface is unknown. Determine the minimum tank air pressure (gage) that will cause a water stream from the nozzle to rise 90 ft from the ground.

Step 1 of 3

Given data:

The height of the water in the tank from ground level, $z_s = 66\text{ft}$

The height to which the water jet rises, $z_j = 90\text{ft}$

The pressure at tank surface and maximum height of the jet is atmospheric, P_{atm}

Step 2 of 3

Consider Bernoulli's equation for steady incompressible flow between the tank surface and point of maximum height of the water jet, we get

$$\frac{P_s}{\rho g} + \alpha_1 \frac{V_s^2}{2g} + z_s + h_{pump,u} = \frac{P_j}{\rho g} + \alpha_2 \frac{V_j^2}{2g} + z_j + h_{turbine,e} + h_L$$

We know that

$P_s = P_j = P_{atm}$

And the velocity of fluid at the surface and maximum height is zero,

$V_s = V_j = 0$

Step 3 of 3

The useful pump head is

$$\begin{aligned} h_{pump,u} &= z_j - z_s \\ &= 90 - 66 \\ &= 24\text{ft} \end{aligned}$$

Then the pressure rise can be expressed as

$$\begin{aligned} \Delta P &= \rho g h_{pump,u} \\ &= (62.22) \times (32.174) \times (24) \\ &= 480044.79 \\ &= 48.00\text{ksf} \end{aligned}$$

or

$$= 333.64\text{psi}$$

Therefore the minimum pressure rise required is $\boxed{\Delta P = 48.00\text{ksf}}$.

Question: 12.057P

A large tank is initially filled with water 5 m above the center of a sharp-edged 10-cm-diameter orifice. The tank water surface is open to the atmosphere, and the orifice drains to the atmosphere. If the total irreversible head loss in the system is 0.3 m, determine the initial discharge velocity of water from the tank. Take the kinetic energy correction factor at the orifice to be 1.2.

Step 1 of 4

Given data:

The diameter of the orifice, $D_2 = 0.1\,\text{m}$.

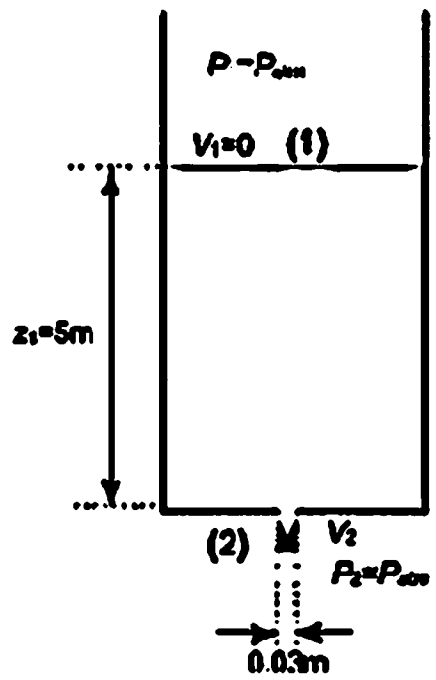
The irreversible head loss, $h_L = 0.3\,\text{m}$.

The free surface of water in the tank is at a height of from the orifice is $z_1 = 5\,\text{m}$.

The kinetic energy correction factor at orifice is $\alpha_2 = 1.2$.

Step 2 of 4

Figure showing a tank fitted with orifice,



Step 3 of 4

The exit at the orifice is set as the reference point. Applying the Bernoulli's equation for steady incompressible flow between the points (1) and (2):

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine} + h_L$$

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L$$

Step 4 of 4

Substituting the required values and on simplifying, we get

$$\frac{P_{atm}}{\rho g} + 0 + 5 = \frac{P_{atm}}{\rho g} + (1.2)\frac{V_2^2}{2g} + 0 + 0.3$$

$$V_2^2 = \frac{(2 \times 9.81)(5 - 0.3)}{1.2}$$

$$V_2 = \sqrt{76.845}$$

$$= 8.76\ \text{m/s}$$

Therefore the initial discharge velocity is $\boxed{V_2 = 8.76\ \text{m/s}}$.

Question: 12.058P

Water enters a hydraulic turbine through a 30-cm-diameter pipe at a rate of 0.6 m3s and exits through a 25-cm-diameter pipe. The pressure drop in the turbine is measured by a mercury manometer to be 1.2 m. For a combined turbine-generator efficiency of 83 percent, determine the net electric power output. Disregard the effect of the kinetic energy correction factors.

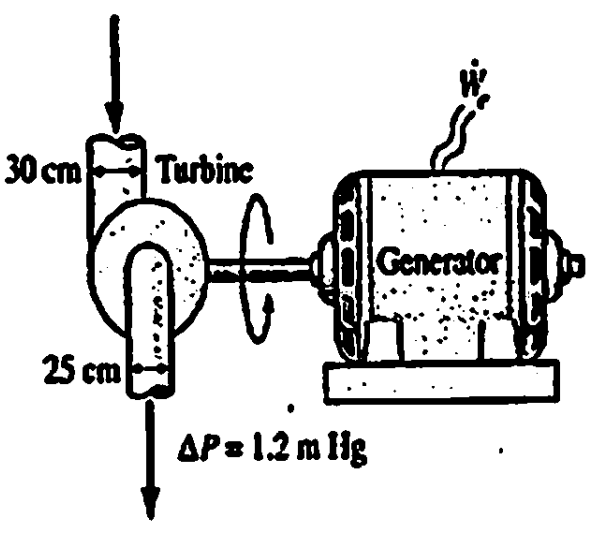
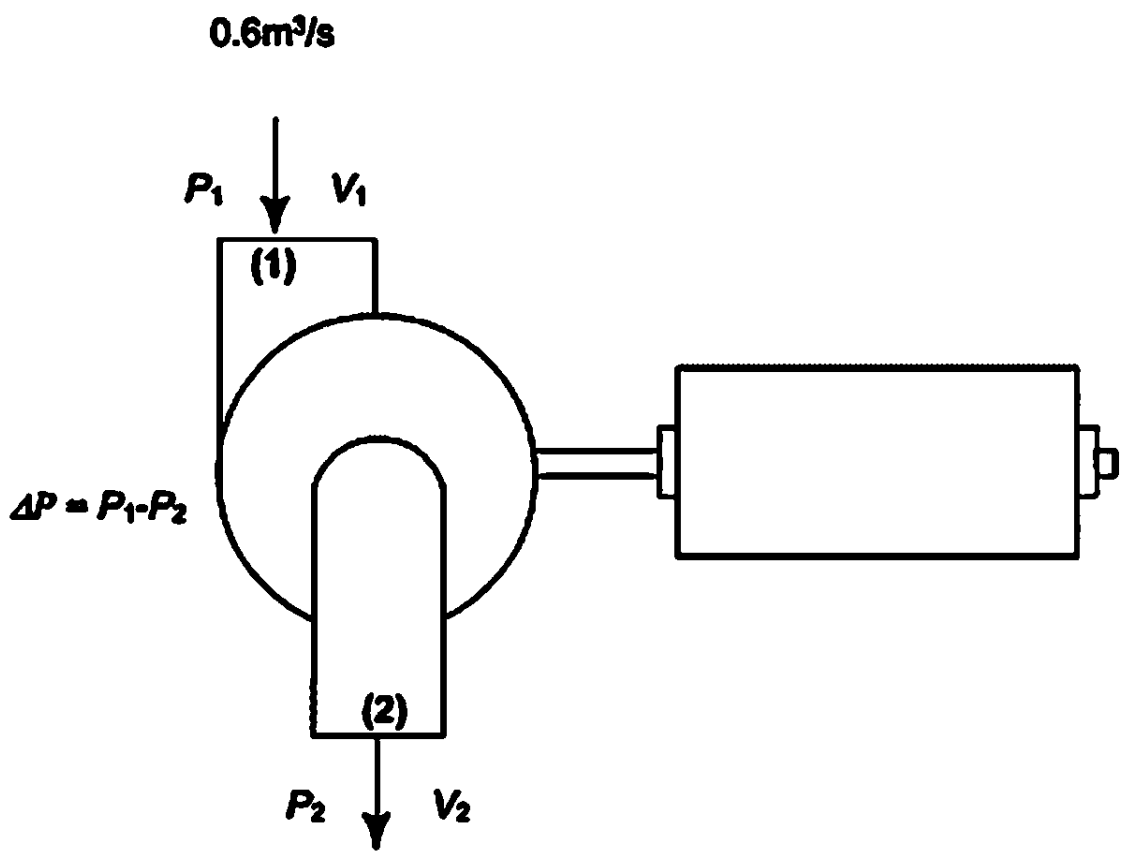


FIGURE P12–58

Step 1 of 5

Sketch the schematic diagram of the pump.



Step 2 of 5

Calculate the mass flow rate of the pump using the following relation

$$\dot{V} = A_1 V_1$$
$$= \frac{\dot{V}}{A_1}$$
$$= \frac{\dot{V}}{\left(\frac{\pi}{4} D_1^2\right)}$$

Here, A is the area of flow, D_1 is the diameter at the inlet and V_1 is the inlet velocity of flow.

Substitute $0.6\ \text{m}^3/\text{s}$ for $\dot{V}$ and 0.3 m for D_1

$$V_1 = \frac{0.6}{\frac{\pi}{4}(0.3)^2}$$
$$= \frac{0.6 \times 4}{\pi \times (0.3)^2}$$
$$= 8.48\ \text{m/s}$$

Step 3 of 5

Calculate the exit velocity using the following relation

$$V_2 = \frac{\dot{V}}{A_2}$$
$$= \frac{\dot{V}}{\frac{\pi}{4} D_2^2}$$

Here, D_2 is the diameter at the outlet.

Substitute 0.25 m for D_2

$$V_2 = \frac{0.6}{\frac{\pi}{4}(0.25)^2}$$
$$= \frac{0.6 \times 4}{\pi (0.25)^2}$$
$$= 12.22\ \text{m/s}$$

Step 4 of 5

Calculate the pressure drop using the following relation

$$\Delta P = \rho_{Hg} g h_{Hg}$$

Here, ρ_{Hg} are the densities of mercury and water respectively and h_{Hg} is the drop in pressure head.

Substitute $13{,}600\ \text{kg/m}^3$ for ρ_{Hg} , $9.81\ \text{m/s}^2$ for g and 1.2 for h_{Hg}

$$\Delta P = 13{,}600 \times 9.81 \times (1.2)$$
$$= 160 \times 10^3\ \text{Pa}$$

Step 5 of 5

Applying the Bernoulli's flow equation between (1) and (2):

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$
$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + 0 + 0 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + 0 + h_{turbine,e} + 0$$
$$h_{turbine,e} = \frac{P_1}{\rho g} - \frac{P_2}{\rho g} + \alpha\left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right)$$
$$= \frac{\Delta P}{\rho g} + \alpha\left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right)$$

Here, P is the pressure, ρ is the density of water, g is the acceleration due to gravity, α is the correction factor, $h_{turbine,e}$ is the turbine head and $h_{pump,u}$ is the pump head.

Substitute 1 for α and $1000\ \text{kg/m}^3$ for ρ

$$h_{turbine,e} = \frac{160 \times 10^3}{1000 \times 9.81} + (1.0)\left(\frac{8.48^2 - 12.22^2}{2(9.81)}\right)$$
$$= 12.36\ \text{m}$$

Calculate the electric power output using the following equation

$$\dot{W}_{turbine} = \eta_{turb-gen} \times \rho \dot{V} g h_{turbine,e}$$

Here, $\eta_{turb-gen}$ is the turbine – generator efficiency.

Substitute 0.83 for $\eta_{turb-gen}$

$$\dot{W}_{turbine} = 0.83 \times 1000 \times 0.6 \times 9.81 \times 12.36$$
$$= 60.383 \times 10^3\ \text{W}$$

Therefore, the net electric power output of the turbine is $\boxed{60.383\ \text{kW}}$.

Question: 12.059P

The velocity profile for turbulent flow in a circular pipe is approximated as $u(r)=umax(1-r/R)1/n$ where $n=7$. Determine the kinetic energy correction factor for this flow.

Step 1 of 8

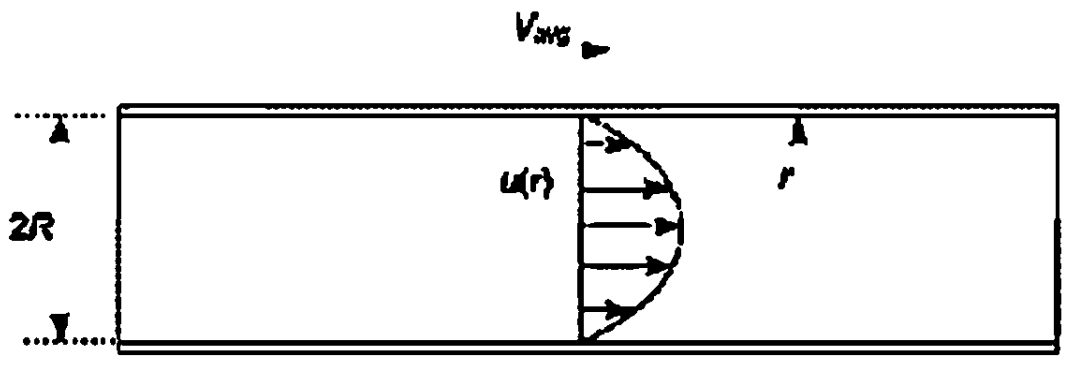
Given data:

The velocity profile of the fluid flow, $u(r) = u_{max}(1 - r/R)^{1/n}$.

Where $n = 7$

Step 2 of 8

Let us draw the figure.



Step 3 of 8

We have

$$\alpha = \frac{1}{A}\int_A \left[\frac{V(r)}{V_{avg}}\right]^3 dA$$

$$= \frac{1}{\pi R^2}\int_A \left[\frac{u(r)}{V_{avg}}\right]^3 dA$$

$$= \frac{u_{max}^3}{\pi R^2 V_{avg}^3}\int_0^R \left[(1 - r/R)^{1/n}\right]^3 (2\pi r)dr$$

$$= \frac{2u_{max}^3}{R^2 V_{avg}^3}\int_0^R \left[(1 - r/R)^{1/n}\right]^3 (r)dr$$

Step 4 of 8

We have for average velocity,

$$V_{avg} = \frac{1}{A}\int_A V(r)dA$$

$$= \frac{1}{A}\int_A u(r)dA$$

$$= \frac{1}{\pi R^2}\int_0^R u_{max}(1 - r/R)^{1/n} 2\pi r dr$$

$$= \frac{2u_{max}}{R^2}\int_0^R (1 - r/R)^{1/n} r dr$$

Step 5 of 8

Where

$$\int_0^R \left(1 - \frac{r}{R}\right)^{1/n} r dr = \left\{\frac{\left(1 - \frac{r}{R}\right)^{(1/n)+2}}{\frac{1}{R^2}\left[(1/n)+2\right]} - \frac{\left(1 - \frac{r}{R}\right)^{(1/n)+1}}{\frac{1}{R^2}\left[(1/n)+1\right]}\right\}_0^R$$

$$= \frac{\left(1 - \frac{R}{R}\right)^{(1/n)+2}}{\frac{1}{R^2}\left[(1/n)+2\right]} - \frac{\left(1 - \frac{r}{R}\right)^{(1/n)+1}}{\frac{1}{R^2}\left[(1/n)+1\right]}$$

$$= \frac{n^2 R^2}{(n+1)(2n+1)}$$

Step 6 of 8

Then

$$V_{avg} = \frac{2u_{max}}{R^2}\frac{n^2 R^2}{(n+1)(2n+1)}$$

We have $n = 7$

$$V_{avg} = \frac{2u_{max}}{R^2}\frac{7^2 R^2}{(7+1)(2\times 7+1)}$$

$$= 0.81667u_{max}$$

Step 7 of 8

We have

$$\alpha = \frac{2u_{max}^3}{R^2 V_{avg}^3}\int_0^R \left[(1 - r/R)^{1/n}\right]^3 r dr$$

$$= \frac{2u_{max}^3}{R^2(0.81667u_{max})^3}\int_0^R \left[(1 - r/R)^{1/n}\right]^3 r dr$$

Where

$$\int_A \left[(1 - r/R)^{1/n}\right]^3 r dr = \left[\frac{\left(1 - \frac{r}{R}\right)^{(3/n)+2}}{(1/R^2)\left(\frac{3}{n}+2\right)} - \frac{\left(1 - \frac{r}{R}\right)^{(3/n)+1}}{(1/R^2)\left(\frac{3}{n}+1\right)}\right]_0^R$$

$$= \frac{\left(1 - \frac{R}{R}\right)^{(3/n)+2}}{(1/R^2)\left(\frac{3}{n}+2\right)} - \frac{\left(1 - \frac{R}{R}\right)^{(3/n)+1}}{(1/R^2)\left(\frac{3}{n}+1\right)}$$

$$= \frac{n^2 R^2}{(n+3)(2n+3)}$$

Step 8 of 8

Then

$$\alpha = \frac{2u_{max}^3}{R^2(0.81667u_{max})^3}\frac{n^2 R^2}{(n+3)(2n+3)}$$

$$= \frac{2u_{max}^3}{R^2(0.81667u_{max})^3}\frac{7^2 R^2}{(7+3)(2\times 7+3)}$$

$$= 1.058$$

Therefore the kinetic energy correction factor is $\boxed{\alpha = 1.058}$.

Question: 12.060P

Water is pumped from a lower reservoir to a higher reservoir by a pump that provides 20 kW of useful mechanical power to the water. The free surface of the upper reservoir is 45 m higher than the surface of the lower reservoir. If the flow rate of water is measured to be 0.03 m3/s, determine the irreversible head loss of the system and the lost mechanical power during this process.

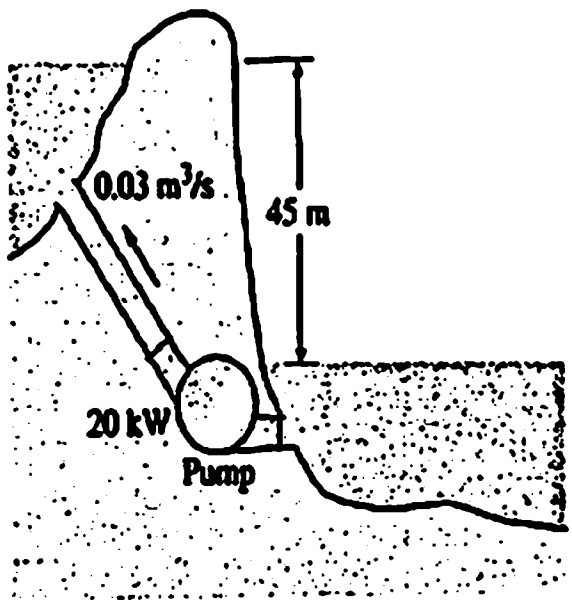


FIGURE P12–60

Step 1 of 5

Given data:

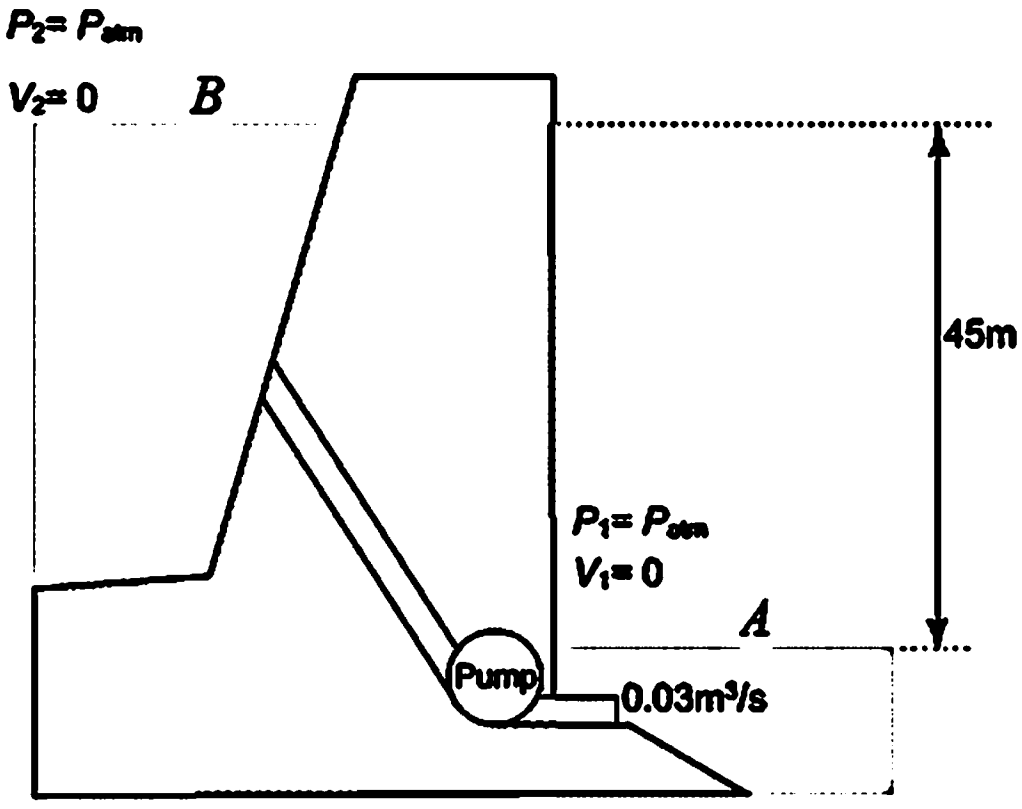
The useful mechanical power given by the pump, $\dot{W}_{pump,u} = 20\ \text{kW}$.

The free surface of upper reservoir is $45\ \text{m}$ above the lower reservoir.

The flow rate, $\dot{V} = 0.03\ \text{m}^3/\text{s}$.

Step 2 of 5

Figure showing different levels reservoir;



Step 3 of 5

We know that mass flow rate,

$$\dot{m} = \rho\dot{V}$$
$$= (1000)0.03$$
$$= 30\ \text{kg/s}$$

Step 4 of 5

Applying the Bernoulli's flow equation between the points *A* and *B*:

$$\dot{m}\left(\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + gz_1\right) + \dot{W}_{pump} = \dot{m}\left(\frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + gz_2\right) + \dot{W}_{turbine} + \dot{E}_{mech,loss}$$

$$\dot{m}\left(\frac{P_1}{\rho g} + gz_1\right) + \dot{W}_{pump} = \dot{m}\left(\frac{P_2}{\rho g} + gz_2\right) + \dot{E}_{mech,loss}$$

$$(300)\left(\frac{P_{atm}}{\rho g} + g\times 0\right) + \left(20\times 10^3\right) = (30)\left(\frac{P_{atm}}{\rho g} + g\times 45\right) + \dot{E}_{mech,loss}$$

$$\dot{E}_{mech,loss} = 6756.5\ \text{W}$$

Therefore, the mechanical power loss during the process is $\boxed{\dot{E}_{mech,loss} = 6756.5\ \text{W}}$

Step 5 of 5

Then the irreversible head loss,

$$h_L = \frac{\dot{E}_{mech,loss}}{\dot{m}g}$$
$$= \frac{6756.5}{(30)(9.81)}$$
$$= 22.95\ \text{m}$$

Therefore, the irreversible head loss is $\boxed{h_L = 22.72\ \text{m}}$.

Question: 12.061P

Water in a partially filled large tank is to be supplied to the roof top, which is 8 m above the water level in the tank, through a 2.5-cm-internal-diameter pipe by maintaining a constant air pressure of 300 kPa (gage) in the tank. If the head loss in the piping is 2 m of water, determine the discharge rate of the supply of water to the roof top.

Step 1 of 4

Given data:

The water has to be supplied to a height, $h = 8$ m

The internal diameter of the pipe, $d = 2.5$ cm

The gage pressure in the tank, $P_{gage} = 300$ kPa

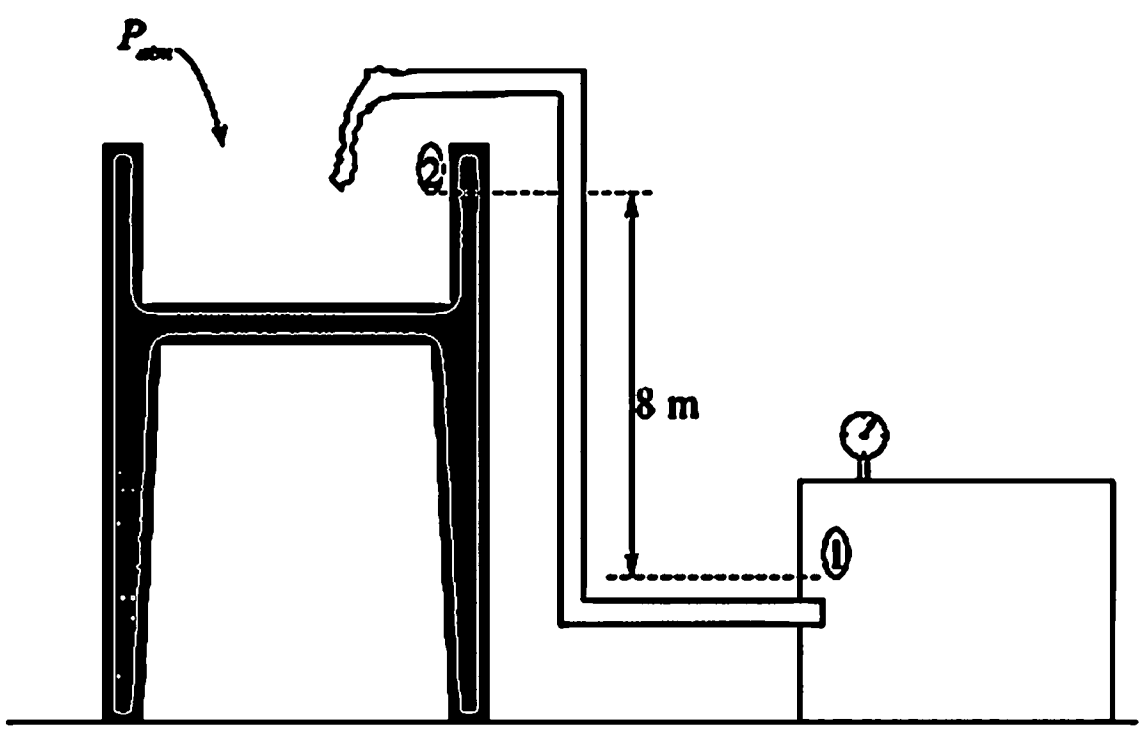
The head loss in the pipe, $h_L = 2$ m

Initial velocity of the water in the tank, $V_1 = 0$

Density of the water is taken as $\rho = 1000\ \text{kg/m}^3$

Step 2 of 4

Water supplied from partially filled tank to the top of the roof as shown below.



Step 3 of 4

Applying the Bernoulli's flow equation between the points (1) and (2) we get

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{turbine}+h_L$$

Substituting the required values and simplify:

$$\frac{P_{abs}}{\rho g}+0+z_1+0=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g}+z_2+h_L$$

$$\frac{V_2^2}{2g}=\left(\frac{P_{abs}}{\rho g}-\frac{P_{atm}}{\rho g}\right)+z_1-z_2-h_L$$

$$=\frac{P_{gage}}{\rho g}+z_1-z_2-h_L$$

$$=\frac{300\times10^3}{(1000)(9.81)}+0-8-2$$

$$=20.581$$

$$V_2=\sqrt{2g\times20.581}$$

$$=20.10\ \text{m/s}$$

Velocity of the water discharging on to the top of the roof is $V_2 = 20.10$ m/s

Step 4 of 4

Area of the pipe is given by

$$A_2=\frac{\pi}{4}\times d^2$$

$$=\frac{\pi}{4}(0.025)^2$$

$$=4.909\times10^{-4}\ \text{m}^2$$

The discharge rate of the supply of water to the roof top,

$$\dot{V}=AV_2$$

$$=4.909\times10^{-4}\times20.10$$

$$=9.869\times10^{-3}\ \text{m}^3/\text{s}$$

Therefore the discharge rate of the supply of water to the roof top is $\boxed{\dot{V} = 9.869\times10^{-3}\text{m}^3/\text{s}}$

Question: 12.062P pg 1

Underground water is to be pumped by a 78 percent efficient 5-kW submerged pump to a pool whose free surface is 30 m above the underground water level. The diameter of the pipe is 7 cm on the intake side and 5 cm on the discharge side. Determine (*a*) the maximum flow rate of water and (*b*) the pressure difference across the pump. Assume the elevation difference between the pump inlet and the outlet and the effect of the kinetic energy correction factors to be negligible.

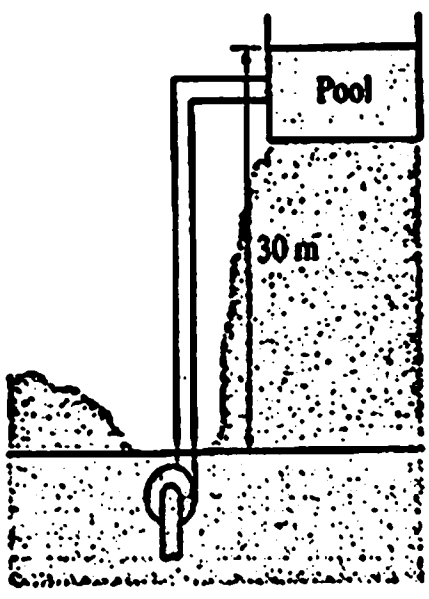


FIGURE P12–62

Step 1 of 8

Given data:

The pump efficiency, $\eta_{pump} = 78\%$

The power delivered by the submerged pump, $\dot{W}_{pump} = 5\ \text{kW}$

Height of the pool from the under water surface, $h = 30\ \text{m}$

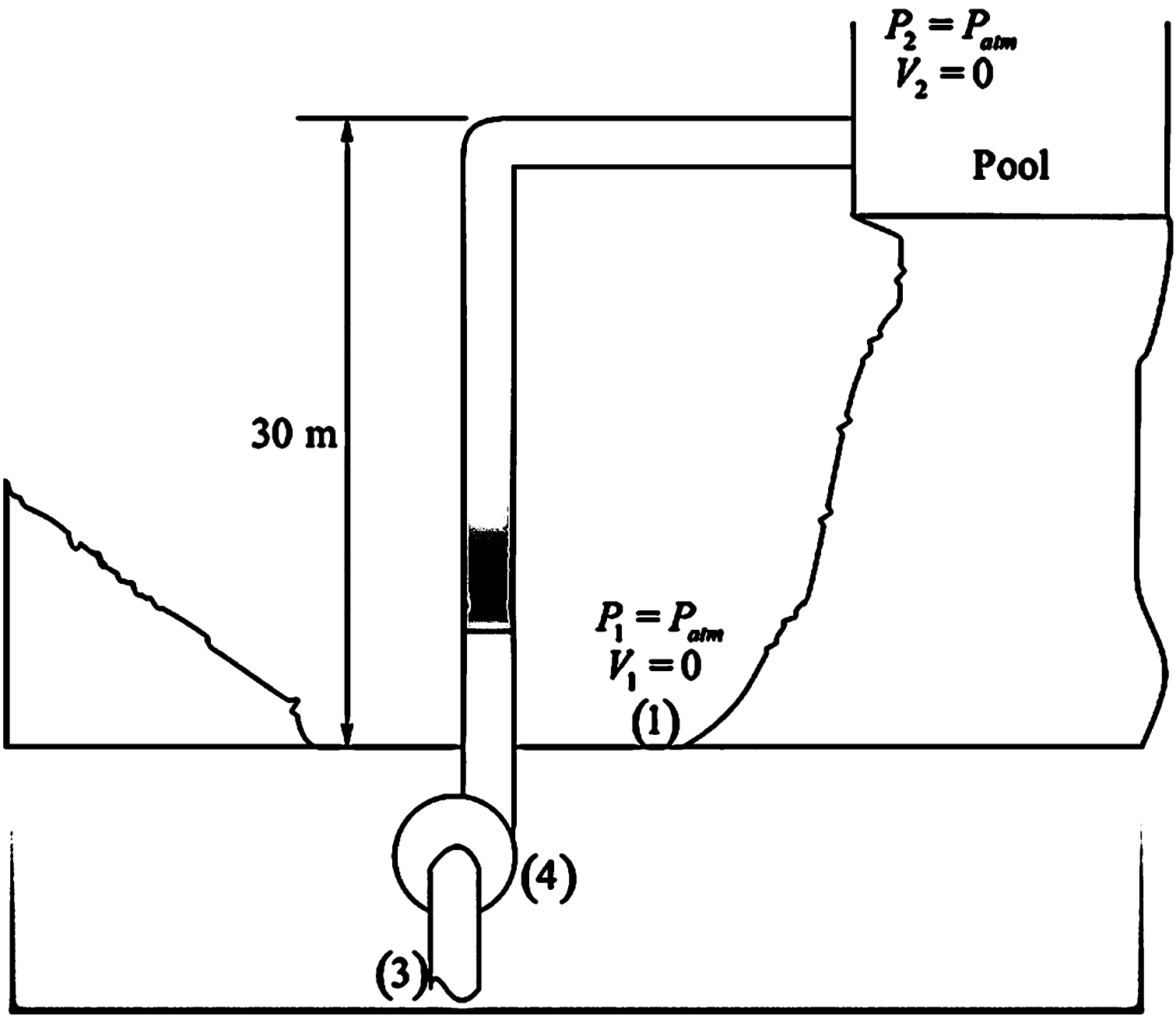
Diameter of the pipe on the intake side, $d_3 = 0.07\ \text{m}$

Diameter of the pipe on the discharge side, $d_4 = 0.05\ \text{m}$

Assume the elevation difference between the pump inlet and the outlet and the effect of the kinetic energy correction factor to be negligible.

Step 2 of 8

Under ground water is supplied by a pump to the pool as shown in the figure below.



Step 3 of 8

The mechanical efficiency of the pump is given by

$$\eta_{pump} = \frac{\dot{W}_{pump,u}}{\dot{W}_{pump}}$$

$$\dot{W}_{pump,u} = \eta_{pump}\dot{W}_{pump}$$
$$= 0.78(5)$$
$$= 3.9\ \text{kW}$$

Step 4 of 8

Applying the Bernoulli's equation for steady flow between the points (1) and (2), we get

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + gz_1\right) + \dot{W}_{pump,u} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + gz_2\right) + \dot{W}_{turbine,e} + E_{mech,loss}$$

Since the work done by the turbine and mechanical losses is zero then the equation is given by

$$\dot{m}\frac{P_{atm}}{\rho} + \dot{m}g(0) + \dot{W}_{pump,u} = \dot{m}\frac{P_{atm}}{\rho} + \dot{m}gz_2 \quad \text{..... (1)}$$

From the equation the mass flow rate of the water through the pump is given by

$$\dot{W}_{pump,u} = \dot{m}gz_2$$
$$\dot{m} = \frac{3.9\times10^3}{(9.81)(30)}$$
$$\dot{m} = 13.25\ \text{kg/s}$$

Step 5 of 8

a)

The maximum flow rate of the water through the pump is given by

$$\dot{m} = \rho\dot{V}$$

a)

The maximum flow rate of the water through the pump is given by

$$\dot{m} = \rho \dot{V}$$

$$\dot{V} = \frac{\dot{m}}{\rho}$$

$$= \frac{13.25}{1000}$$

$$= 0.01325 \text{ m}^3/\text{s}$$

Therefore, the maximum flow rate of the water through the pump is $\boxed{\dot{V} = 0.01325 \text{ m}^3/\text{s}}$

Step 6 of 8

Area of the pipe at the intake side of the water is given by

$$A_3 = \frac{\pi d_3^2}{4}$$

$$= \frac{\pi (0.07)^2}{4}$$

$$= 3.848 \times 10^{-3} \text{ m}^2$$

Discharge of the water through the pipe at the intake side is given by

$$V_3 = \frac{\dot{V}}{A_3}$$

$$= \frac{0.01325}{3.848 \times 10^{-3}}$$

$$= 3.443 \text{ m/s}$$

The velocity of the water at the intake side of the pipe is $V_3 = 3.443$ m/s

Area of the pipe at the discharge side of the water is given by

$$A_4 = \frac{\pi d_4^2}{4}$$

$$= \frac{\pi (0.05)^2}{4}$$

$$= 1.963 \times 10^{-3} \text{ m}^2$$

Discharge of the water through the pipe at the discharge side is given by

$$V_4 = \frac{\dot{V}}{A_4}$$

$$= \frac{0.01325}{1.963 \times 10^{-3}}$$

$$= 6.749 \text{ m/s}$$

The velocity of the water at the discharge side of the pipe is $V_4 = 6.749$ m/s

Step 7 of 8

Applying the Bernoulli's equation for steady flow between the points (3) and (4):

$$\frac{P_3}{\rho g} + \alpha_1 \frac{V_3^2}{2g} + z_3 + h_{pump,u} = \frac{P_4}{\rho g} + \alpha_2 \frac{V_4^2}{2g} + z_4 + h_{turbine,u} + h_L$$

$$\frac{P_3}{\rho g} + \frac{V_3^2}{2g} + h_{pump,u} = \frac{P_4}{\rho g} + \frac{V_4^2}{2g}$$

$$\frac{P_3}{\rho g} - \frac{P_4}{\rho g} = \frac{1}{2g}\left(V_4^2 - V_3^2\right) - h_{pump,u}$$

$$P_4 - P_3 = \rho\left[\frac{\dot{W}_{pump,u}}{\dot{m}g} - \frac{1}{2}\left(V_4^2 - V_3^2\right)\right]$$

Step 8 of 8

b)

Pressure difference across the pump is given by

$$P_4 - P_3 = \rho\left[\frac{\dot{W}_{pump,u}}{\dot{m}g} - \frac{1}{2}\left(V_4^2 - V_3^2\right)\right]$$

$$= (1000)\left[\frac{3.9 \times 10^3}{13.25 \times 9.81} - \frac{1}{2}\left(6.749^2 - 3.443^2\right)\right]$$

$$= 1000[30.004 - 16.85]$$

$$= 13154.04 \text{ Pa}$$

Therefore, the pressure difference across the pump is $\boxed{P_4 - P_3 = 13154.04 \text{ Pa}}$

Question: 12.063P

Reconsider Prob. 12–62. Determine the flow rate of water and the pressure difference across the pump if the irreversible head loss of the piping system is 4 m.

Step 1 of 9

Given data:

The pump efficiency, $\eta_{pump} = 78\%$

The power delivered by the submerged pump, $\dot{W}_{pump} = 5$ kW

Height of the pool from the under water surface, $h = 30$ m

Diameter of the pipe on the intake side, $d_3 = 0.07$ m

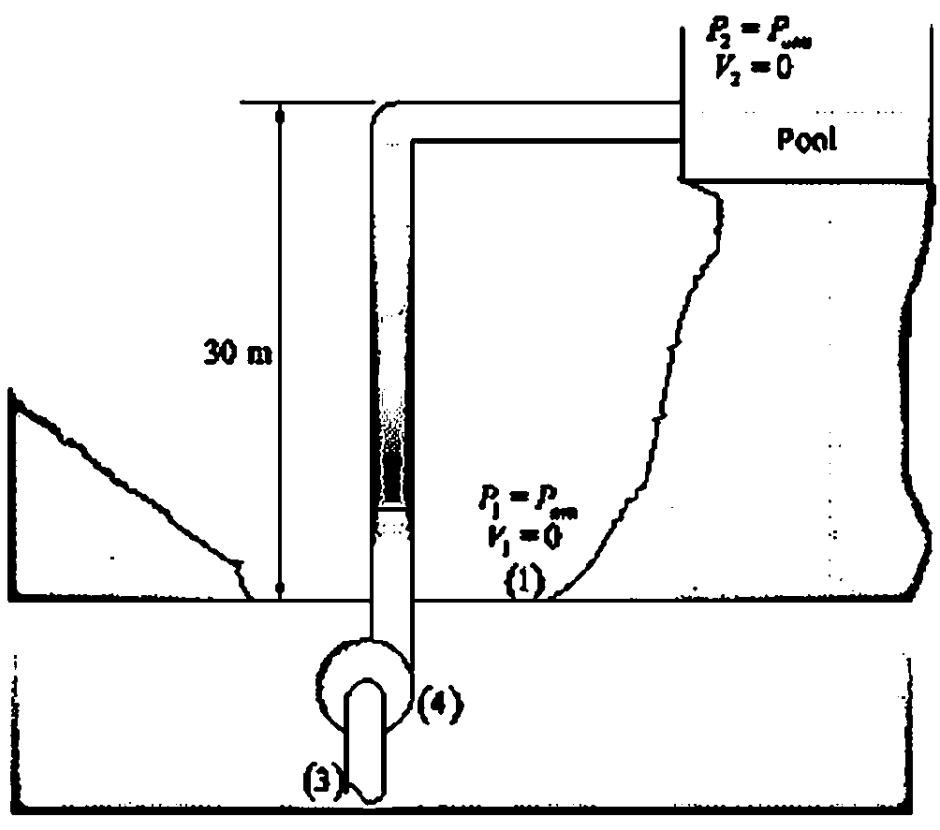
Diameter of the pipe on the discharge side, $d_4 = 0.05$ m

Irreversible head loss in the pipe system, $h_L = 4$ m

Assume the elevation difference between the pump inlet and the outlet and the effect of the kinetic energy correction factor to be negligible.

Step 2 of 9

Under ground water is supplied by a pump to the pool as shown in the figure below.



Step 3 of 9

The mechanical efficiency of the pump is given by

$$\eta_{pump} = \frac{\dot{W}_{pump,u}}{\dot{W}_{pump}}$$

$$\dot{W}_{pump,u} = \eta_{pump}\dot{W}_{pump}$$
$$= 0.78(5)$$
$$= 3.9 \text{ kW}$$

Step 4 of 9

Applying the Bernoulli's equation for steady flow between the points (1) and (2), we get

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + gz_1\right) + \dot{W}_{pump,u} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + gz_2\right) + \dot{W}_{turbine,e} + \dot{E}_{mech,loss}$$

Since the work done by the turbine and mechanical losses is zero then the equation is given by

$$\dot{m}\frac{P_{atm}}{\rho} + \dot{m}g(0) + \dot{W}_{pump,u} + \dot{E}_{mech,loss} = \dot{m}\frac{P_{atm}}{\rho} + \dot{m}gz_2 \qquad \ldots\ldots (1)$$

From the equation (1), the mass flow rate of the water through the pump is given by

$$\dot{W}_{pump,u} = \dot{m}gz_2 + \dot{E}_{mech,loss}$$
$$= \dot{m}gz_2 + \dot{m}gh_L$$
$$3.9 \times 10^3 = \dot{m}\left((9.81)(30) + (9.81)(4)\right)$$
$$\dot{m} = 11.69 \text{ kg/s}$$

Step 5 of 9

a)

The maximum flow rate of the water through the pump is given by

$$\dot{m} = \rho\dot{V}$$

$$\dot{V} = \frac{\dot{m}}{\rho}$$
$$= \frac{11.69}{1000}$$
$$= 0.01169 \text{ m}^3/\text{s}$$

Therefore, the maximum flow rate of the water through the pump is $\boxed{\dot{V} = 0.01169 \text{ m}^3/\text{s}}$

Step 6 of 9

Area of the pipe at the intake side of the water is given by

$$A_3 = \frac{\pi d_3^2}{4}$$
$$= \frac{\pi(0.07)^2}{4}$$
$$= 3.848\times10^{-3} \text{ m}^2$$

Discharge of the water through the pipe at the intake side is given by

$$V_3 = \frac{\dot{V}}{A_3}$$
$$= \frac{0.01169}{3.848\times10^{-3}}$$
$$= 3.038 \text{ m/s}$$

The velocity of the water at the intake side of the pipe is $V_3 = 3.038$ m/s

Step 7 of 9

Area of the pipe at the discharge side of the water is given by

$$A_4 = \frac{\pi d_4^2}{4}$$
$$= \frac{\pi(0.05)^2}{4}$$
$$= 1.963\times10^{-3} \text{ m}^2$$

Discharge of the water through the pipe at the discharge side is given by

$$V_4 = \frac{\dot{V}}{A_4}$$
$$= \frac{0.01169}{1.963\times10^{-3}}$$
$$= 5.955 \text{ m/s}$$

The velocity of the water at the discharge side of the pipe is $V_4 = 5.955$ m/s

Step 8 of 9

Applying the Bernoulli's equation for steady flow between the points (3) and (4):

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1\frac{V_1^2}{2} + gz_1\right) + \dot{W}_{pump,u} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2\frac{V_2^2}{2} + gz_2\right) + \dot{W}_{turbine,e} + \dot{E}_{mech,loss}$$

We can neglect frictional effects due to piping when we consider the pump only, $\dot{E}_{mech,loss} = 0$.

Then

$$\left(\frac{P_3}{\rho} + \frac{V_3^2}{2}\right) + \frac{\dot{W}_{pump,u}}{\dot{m}} = \left(\frac{P_4}{\rho} + \frac{V_4^2}{2}\right)$$

$$\frac{P_4}{\rho} - \frac{P_3}{\rho} = \frac{\dot{W}_{pump,u}}{\dot{m}} - \left(\frac{V_4^2}{2} - \frac{V_3^2}{2}\right)$$

Step 9 of 9

b)

Pressure difference across the pump is given by

$$P_4 - P_3 = \rho\left[\frac{\dot{W}_{pump,u}}{\dot{m}} - \frac{1}{2}\left(V_4^2 - V_3^2\right)\right]$$
$$= (1000)\left[\frac{3.9\times10^3}{11.69} - \frac{1}{2}\left(5.955^2 - 3.038^2\right)\right]$$
$$= 1000[333.62 - 13.12]$$
$$= 320503.7 \text{ Pa}$$

Therefore, the pressure difference across the pump is $\boxed{P_4 - P_3 = 320504 \text{ Pa}}$

Question: 12.064P

A fireboal is to fight fires at coastal areas by drawing seawater with a density of 1030 kg/m3 through a 20-cm-diameter pipe at a rate of 0.1 m3/s and discharging it through a hose nozzle with an exit diameter of 5 cm. The total irreversible head loss of the system is 3 m, and the position of the nozzle is 3 m above sea level. For a pump efficiency of 70 percent, determine the required shaft power input to the pump and the water discharge velocity.

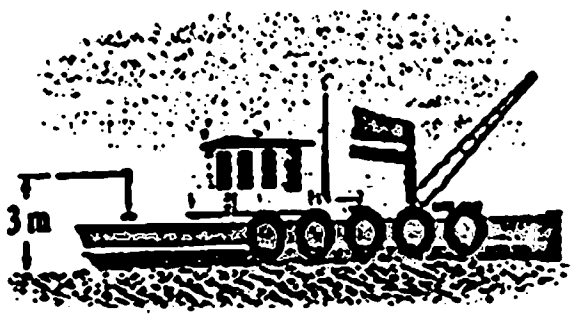


FIGURE P12–64

Step 1 of 6

Given data:

The density of seawater, $\rho = 1030\ \text{kg/m}^3$

The pipe diameter, $D_p = 0.20\ \text{m}$

The flow rate through the pipe, $\dot{V} = 0.1\ \text{m}^3/\text{s}$

The exit diameter of the nozzle, $D_{out} = 0.05\ \text{m}$

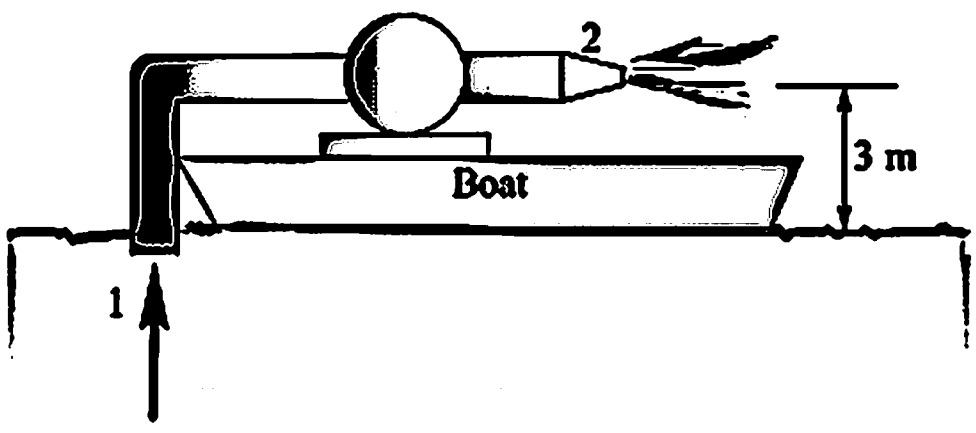
The irreversible head loss of the system, $h_L = 3\ \text{m}$

Height the nozzle above the sea level is $z_2 = 3\ \text{m}$

The mechanical efficiency of the pump is $\eta_{pump} = 70\%$

Step 2 of 6

A fire boat is to fight fires at the coastal areas from the nozzle as shown in the figure below.



Step 3 of 6

Let us set the suction point at sea level as point (1) and the discharge point at the nozzle as point (2).

Consider point (1) where the suction taking place.

Velocity of the water at point (1) is given by

$$\dot{V} = A_1 V_1$$

$$V_1 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.1}{\frac{\pi}{4}(0.2^2)}$$

$$= 3.183\ \text{m/s}$$

Step 4 of 6

And the velocity of the water discharging at point (2) is given by

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.1}{\frac{\pi}{4}(0.05^2)}$$

$$= 50.93\ \text{m/s}$$

Therefore, the discharge velocity of the water is $\boxed{V_2 = 50.93\ \text{m/s}}$

Step 5 of 6

Applying the Bernoulli's equation for steady incompressible flow between the points (1) and (2):

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$

$$\frac{P_{atm}}{\rho g} + (1)\frac{3.183^2}{2g} + 0 + h_{pump,u} = \frac{P_{atm}}{\rho g} + (1)\frac{50.93^2}{2g} + 3 + 0 + 3$$

$$h_{pump,u} = 137.69\ \text{m}$$

Step 6 of 6

The efficiency of the pump is given by

$$\eta_{pump} = \frac{\dot{W}_{pump,u}}{\dot{W}_{shaft,in}}$$

$$\dot{W}_{shaft,in} = \frac{\left(\rho \dot{V} g h_{pump,u}\right)}{\eta_{pump}}$$

$$= \frac{1030(0.1)(9.81)(137.69)}{0.7}$$

$$= 198.75 \times 10^3\ \text{W}$$

$$= 199\ \text{kW}$$

Therefore, the required shaft power input to the pump is $\boxed{\dot{W}_{shaft,in} = 199\ \text{kW}}$

Question: 12.065P

Air flows through a pipe at a rate of 170 L/s. The pipe consists of two sections of diameters 18 cm and 10 cm with a smooth reducing section that connects them. The pressure difference between the two pipe sections is measured by a water manometer. Neglecting frictional effects, determine the differential height of water between the two pipe sections. Take the air density to be 1.20 kg/m3.

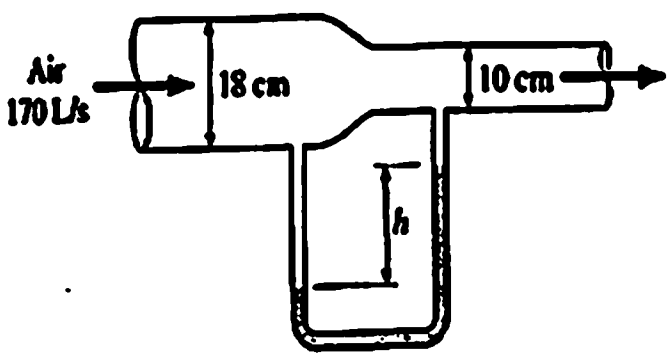


FIGURE P12-65

Step 1 of 7

Given data:

The flow rate of the air through the pipe,

$\dot{V} = 170\ \text{L/s}$

$= 0.17\ \text{m}^3/\text{s}$

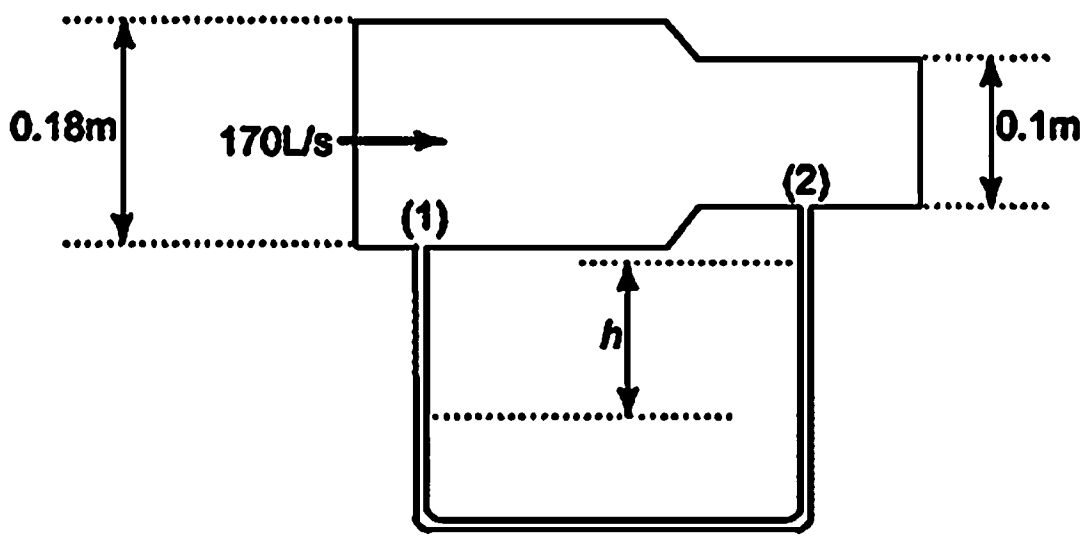
The inlet diameter of the pipe, $D_1 = 0.18\ \text{m}$

The exit diameter of the pipe, $D_2 = 0.1\ \text{m}$

The density of the air, $\rho_a = 1.2\ \text{kg/m}^3$

Step 2 of 7

Air flows through the pipe consisting of two different sections and a water manometer as shown below.



Step 3 of 7

Discharge through the pipe at the two sections is given by

$\dot{V} = A_1V_1 = A_2V_2$

Where A is the area of the pipe and V is the velocity of the water

Then the velocity of the water at the inlet of the pipe is given by

$$V_1 = \frac{\dot{V}}{A_1}$$

$$= \frac{0.17}{\frac{\pi}{4}(0.18)^2}$$

$$= 6.6805\ \text{m/s}$$

Step 4 of 7

Velocity of the water at the exist of the pipe is given by

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.17}{\frac{\pi}{4}(0.10)^2}$$

$$= 21.645\ \text{m/s}$$

Step 5 of 7

Applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{P_1}{\rho_a} + \frac{V_1^2}{2} + gz_1 = \frac{P_2}{\rho_a} + \frac{V_2^2}{2} + gz_2$$

$$P_1 - P_2 = \rho_a\left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right) \qquad \ldots\ldots (1)$$

Step 6 of 7

The pressure force acting in the water manometer is given by

$$P_1 - P_2 = \rho_{water}gh \qquad \ldots\ldots (2)$$

By substituting the equation (2) in (1), we get

$$\rho_{water}gh = \rho_a\frac{V_2^2 - V_1^2}{2}$$

Step 7 of 7

The differential height of water between the two pipes is given by

$$h = \rho_a\frac{V_2^2 - V_1^2}{2g\rho_{water}}$$

$$= (1.20)\frac{\left(21.645^2 - 6.6805^2\right)}{2(9.81)(1000)}$$

$$= 0.026\ \text{m}$$

$$= 2.6\ \text{cm}$$

Therefore, the differential height of the water between the two pipes is $\boxed{h = 2.6\ \text{cm}}$.

Question: 12.066P

Air at 100 kPa and 25°C flows in a horizontal duct of variable cross section. The water column in the manometer that measures the difference between two sectiqns has a vertical displacement of 8 cm. If the velocity in the first section is low and the friction is negligible, determine the velocity at the second section. Also, if the manometer reading has a possible error of ±2 mm, conduct an error analysis to estimate the range of validity for the velocity found.

Step 1 of 7

Given data:

Air flowing through the horizontal duct at a pressure, $P = 100\ \text{kPa}$.

The difference in the heights of the water column manometers, $h = 0.08\ \text{m}$.

The density of the water, $\rho_w = 1000\ \text{kg/m}^3$.

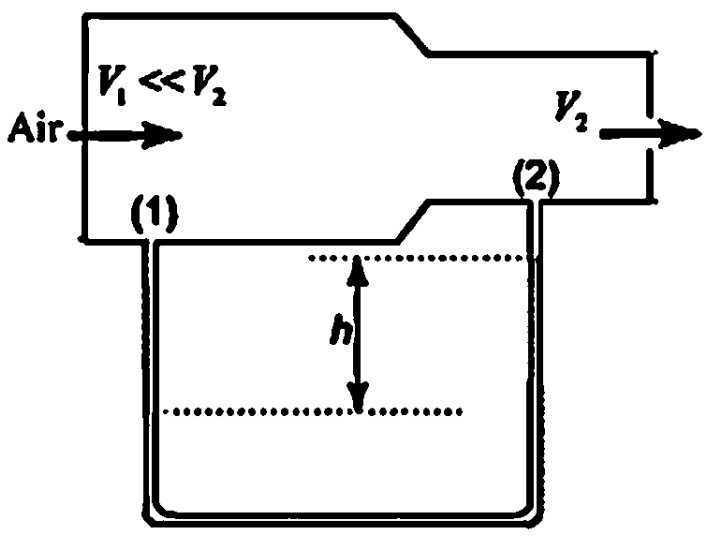
Possible error in the manometer, $h_e = \pm 0.02\ \text{m}$.

The universal gas constant, $R = 0.2870\ \text{kJ/kg}\cdot\text{K}$.

Temperature of the air flowing in the duct, $T = 25\ ^\circ\text{C}$.

Step 2 of 7

Air flows through the horizontal duct of variable cross section with a water column manometer as shown below.



Step 3 of 7

The density of the air in the tank is given by

$$\rho_a = \frac{P}{RT}$$

$$= \frac{100 \times 10^3}{(0.287 \times 10^3)(298)}$$

$$= 1.17\ \text{kg/m}^3$$

Step 4 of 7

Applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{P_1}{\rho_a} + \frac{V_1^2}{2} + gz_1 = \frac{P_2}{\rho_a} + \frac{V_2^2}{2} + gz_2 \qquad \ldots\ldots (1)$$

The effect of elevation can be neglected between the two points because they lie on the same ground and the initial velocity of the air is very small it can be negligible.Then from equation (1)

$$\frac{P_1}{\rho_a} + 0 + 0 = \frac{P_2}{\rho_a} + \frac{V_2^2}{2} + 0$$

$$V_2^2 = \frac{2(P_1 - P_2)}{\rho_a}$$

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_a}} \qquad \ldots\ldots (2)$$

Step 5 of 7

The pressure force acting in the water manometer is given by

$$P_1 - P_2 = \rho_{water} g h \qquad \ldots\ldots (3)$$

By substituting the equation (3) in (2), we get

$$V_2 = \sqrt{\frac{2\rho_{water} g h}{\rho_a}}$$

The velocity of the air at the exit is given by

$$V_2 = \sqrt{\frac{2\rho_{water} g h}{\rho_a}}$$

$$= \sqrt{\frac{2 \times 1000 \times 9.81 \times 0.08}{1.17}}$$

$$= 36.63\ \text{m/s}$$

Therefore, the velocity of the air at the second section is $\boxed{V_2 = 36.63\ \text{m/s}}$.

Step 6 of 7

Error analysis

The velocity is proportional to the square root of the differential height of the manometer fluid. That is, $V_2 = k\sqrt{h}$

Differentiating on both sides we get

$$dV_2 = \frac{1}{2} k \frac{dh}{\sqrt{h}}$$

Now dividing both sides by V_2 we get

$$\frac{dV_2}{V_2} = \frac{\frac{1}{2} k \frac{dh}{\sqrt{h}}}{V_2}$$

$$\frac{dV_2}{V_2} = \frac{\frac{1}{2} k \frac{dh}{\sqrt{h}}}{k\sqrt{h}}$$

$$\frac{dV_2}{V_2} = \frac{dh}{2h}$$

$$= \frac{\pm 0.02}{2 \times 0.08}$$

$$= \pm 0.013$$

Step 7 of 7

$$dV_2 = \pm(0.013 \times V_2)$$

$$= \pm(0.013 \times 36.6)$$

$$= \pm 0.4758\ \text{m/s}$$

$$= \pm 0.5\ \text{m/s}$$

Therefore, after conducted the error analysis the estimated validity of the velocity is expressed as $\boxed{V_2 = 36.6 \pm 0.5\ \text{m/s}}$

Question: 12.067P

A very large tank contains air at 102 kPa at a location where the atmospheric air is at 100 kPa and 20°C Now a 2-cm-diamcter tap is opened. Determine the maximum flow rate of air through the hole. What would your response be r| air is discharged through a 2-m-long. 4-cm-diameter tube with 2-cm-diameter nozzle? Would you solve the problem same way if the pressure in the storage tank were 300 kPa|

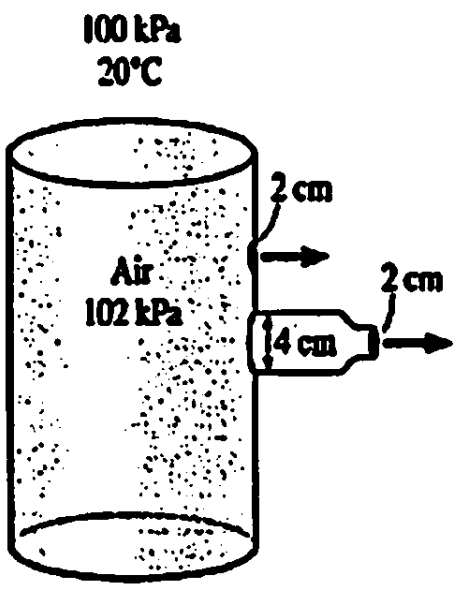


FIGURE P12–67

Step 1 of 6

Given data:

The pressure of the air inside the tank, $P_a = P_1 = 102\ \text{kPa}$.

The atmospheric pressure outside the tank, $P_{atm} = P_2 = 100\ \text{kPa}$.

The diameter of the tap, $D_t = 0.02\ \text{m}$.

Temperature of the air in the tank, $T = 20\ ^\circ\text{C}$.

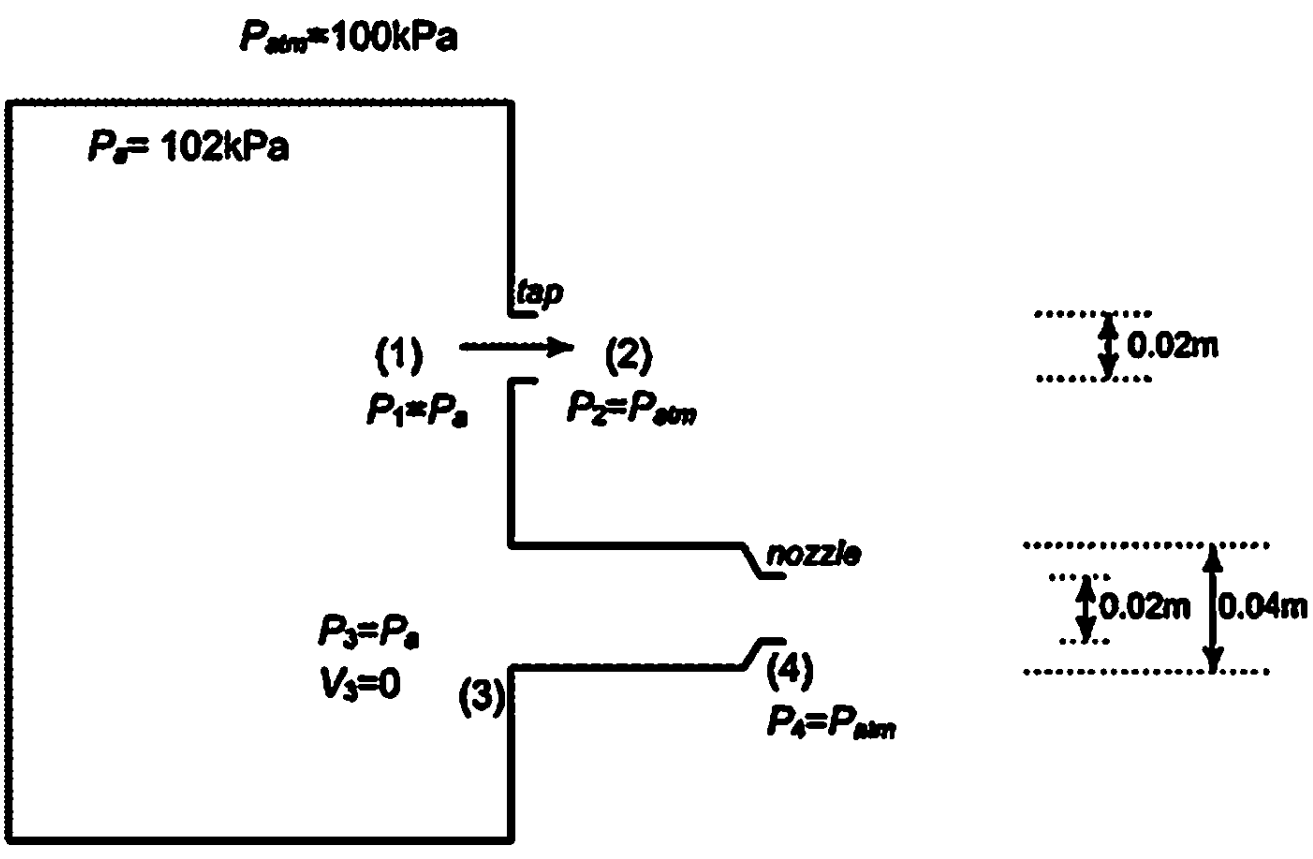
The universal gas constant $R = 0.2870\ \text{kJ/kg}\cdot\text{K}$

Air escapes through a hose of diameter, $d_{hose} = 0.04\ \text{m}$

Diameter of the nozzle end, $d_{noz} = 0.02\ \text{m}$

Step 2 of 6

Let us draw the figure.



Step 3 of 6

Applying the Bernoulli's flow equation between the tank and the tap, points (1) and (2):

$$\frac{P_1}{\rho_a} + \frac{V_1^2}{2} + gz_1 = \frac{P_2}{\rho_a} + \frac{V_2^2}{2} + gz_2 \quad \text{..... (1)}$$

The effect of elevation can be neglected between the two points because they lie on the same ground and the velocity of the air in the tank is zero

Then from equation (1)

$$\frac{P_1}{\rho_a} + 0 + 0 = \frac{P_2}{\rho_a} + \frac{V_2^2}{2} + 0$$

$$V_2^2 = \frac{2(P_1 - P_2)}{\rho_a}$$

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_a}} \quad \text{..... (2)}$$

Step 4 of 6

The density of the air in the tank is given by

$$\rho_a = \frac{P}{RT}$$

$$= \frac{102 \times 10^3}{(0.287 \times 10^3)(293)}$$

$$= 1.2129\ \text{kg/m}^3$$

Step 5 of 6

From equation (2), the velocity of the air exerting out from the tap to the atmosphere is given by

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_a}}$$

$$= \sqrt{\frac{2(102 - 100)}{1.2129}}$$

$$= \sqrt{3.298}$$

$$= 1.816\ \text{m/s}$$

Step 6 of 6

The discharge rate of the air through the tap is given by

$$\dot{V}_{tap} = A_{tap} V_2$$

$$= \left(\frac{\pi}{4} \times 0.02^2\right)(1.816)$$

$$= 5.705 \times 10^{-4}\ \text{m}^3/\text{s}$$

The maximum flow rate is $\boxed{\dot{V}_{tap} = 5.705 \times 10^{-4}\ \text{m}^3/\text{s}}$

The air escapes through a hose of diameter 0.04m, length 2m with an end nozzle of diameter 0.02cm.

Let us assume the flow is steady, incompressible and irrotational having the frictional effects inside the hose and at nozzle are negligible.

The discharge rate through the hose shall not vary due to the increased size and length because as the area increases there will be corresponding decrease in pressure and the frictional effects are neglected throughout the entire length of the hose.

This can be expressed as

$$\dot{V}_{hose} = A_3 V_3 = A_4 V_4$$

Where we can express based on our knowledge

$$\dot{V}_{tap} = \dot{V}_{hose}$$

When the pressure in the storage tank is $P = 300\ \text{kPa}$ then there occur pressure change between the tank and atmosphere. Since the air density is no longer remains same as in the tank and the atmosphere the flow is not incompressible. Thus it can be solved using the theory of compressible flow.

Question: 12.068P

Water is flowing through a Venturi meter whose 1 diameter is 7 cm at the entrance part and 4 cm at the throat 1 The pressure is measured to be 380 kPa at the entrance and J 150 kPa at the throat. Neglecting frictional effects, determine 1 the flow rate of water.

Step 1 of 4

Given data:

The inlet diameter of the Venturi meter, $D_1 = 0.07$ m .

The throat diameter of the Venturi meter, $D_2 = 0.04$ m .

The inlet pressure, $P_1 = 380$ kPa .

The throat pressure, $P_2 = 150$ kPa .

The frictional effects can be neglected.

Step 2 of 4

Let us assume the flow is steady, incompressible and irrotational.

Let the inlet section be point (1) and throat section is point (2).

Both the points are at the same elevation $z_1 = z_2 = 0$.

Flow rate of the water is given by

$$\dot{V} = A_1V_1 = A_2V_2$$

Then we can express

$$V_1 = \frac{\dot{V}}{A_1} \qquad \ldots\ldots (1)$$

And

$$V_2 = \frac{\dot{V}}{A_2} \qquad \ldots\ldots (2)$$

Step 3 of 4

Applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1 = \frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2$$

$$\frac{P_1 - P_2}{\rho} = \frac{V_2^2 - V_1^2}{2}$$

Now by substituting equation (1) and (2) in (3), we get

$$\frac{P_1 - P_2}{\rho} = \frac{1}{2}\left[\left(\frac{\dot{V}}{A_2}\right)^2 - \left(\frac{\dot{V}}{A_1}\right)^2\right]$$

Step 4 of 4

The flow rate of the water is given by

$$\dot{V} = \frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1 - P_2)}{\rho\left[1 - \left(\frac{A_2}{A_1}\right)^2\right]}}$$

$$= \frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1 - P_2)}{\rho\left[1 - \left(\frac{D_2}{D_1}\right)^4\right]}}$$

$$= \frac{\pi(0.04)^2}{4}\sqrt{\frac{2(380 - 150)}{1000\left[1 - \left(\frac{4}{7}\right)^4\right]}\left(\frac{1000}{1}\right)}$$

$$= \frac{\pi(0.04)^2}{4} \times 22.69$$

$$= 0.0285 \text{ m}^3/\text{s}$$

Therefore the flow rate of the water is $\boxed{\dot{V} = 0.0285 \text{ m}^3/\text{s}}$.

Question: 12.069P

Water flows at a rate of 0.025 m3/s in a horizontal pipe whose diameter increases from 6 to 11 cm by an 1 enlargement section. If the head loss across the enlargement section is 0.45 m and the kinetic energy correction factor at 1 both the inlet and the outlet is 1.05. determine the pressure 1 change.

Step 1 of 4

Given data:

The flow rate of the water, $\dot{V} = 0.025\ \text{m}^3/\text{s}$.

The initial diameter of the pipe section, $D_1 = 0.06\ \text{m}$.

The final diameter of the pipe section, $D_2 = 0.11\ \text{m}$.

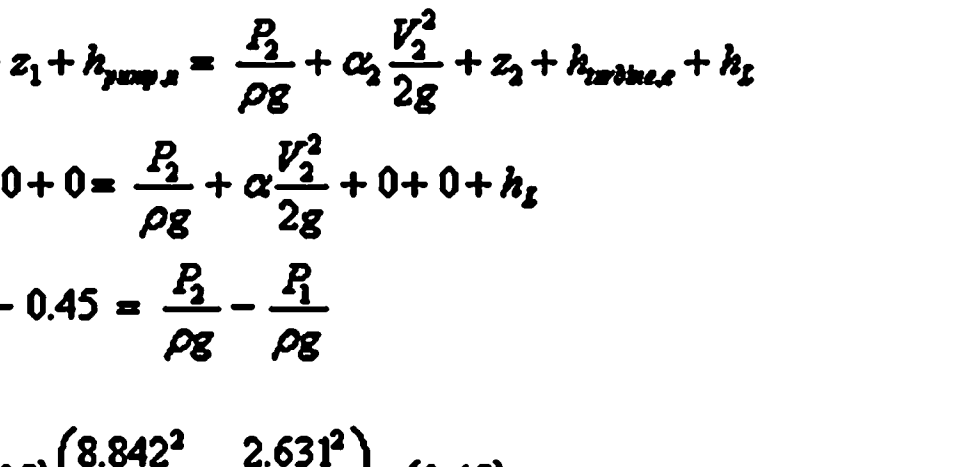
The head loss, $h_L = 0.45\ \text{m}$.

The kinetic energy correction factor at inlet and outlet, $\alpha_1 = \alpha_2 = 1.05$.

The frictional effects are neglected.

Step 2 of 4

Water flows in a pipe whose diameter is varying as shown in the figure below.



Step 3 of 4

Let us assume the flow is steady and incompressible and the inlet and exit are at the same elevation.

Velocity of the water at point (1) is given by

$$V_1 = \frac{\dot{V}}{A_1}$$

$$= \frac{0.025}{\frac{\pi}{4}(0.06^2)}$$

$$= 8.842\ \text{m/s}$$

And

Velocity of the water at point (2) is given by

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{0.025}{\frac{\pi}{4}(0.11)^2}$$

$$= 2.631\ \text{m/s}$$

Step 4 of 4

Applying the Bernoulli's equation for steady flow between the points (1) and (2):

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine} + h_L$$

$$\frac{P_1}{\rho g} + \alpha \frac{V_1^2}{2g} + 0 + 0 = \frac{P_2}{\rho g} + \alpha \frac{V_2^2}{2g} + 0 + 0 + h_L$$

$$\alpha \frac{V_1^2}{2g} - \alpha \frac{V_2^2}{2g} - 0.45 = \frac{P_2}{\rho g} - \frac{P_1}{\rho g}$$

Then,

$$P_2 - P_1 = \rho(1.05)\left(\frac{8.842^2}{2} - \frac{2.631^2}{2}\right) - (0.45)\rho g$$

$$= 1000(1.05)\left(\frac{8.842^2}{2} - \frac{2.631^2}{2}\right) - 0.45(1000)(9.81)$$

$$= 32.996 \times 10^3\ \text{Pa}$$

$$= 32.996\ \text{kPa}$$

Therefore the pressure change in the pipe is $\boxed{\Delta P = 32.996\ \text{kPa}}$.

Question: 12.070P

A 3-m-high large tank is initially filled with water. The tank water surface is open to the atmosphere, and a sharp-edged 10-cm-diameter orifice at the bottom drains to the atmosphere through a horizontal 80-m-long pipe. If the total irreversible head loss of the system is determined to he 1.5 m. determine the initial velocity of the water from the tank. Disregard the effect of the kinetic energy correction factors.

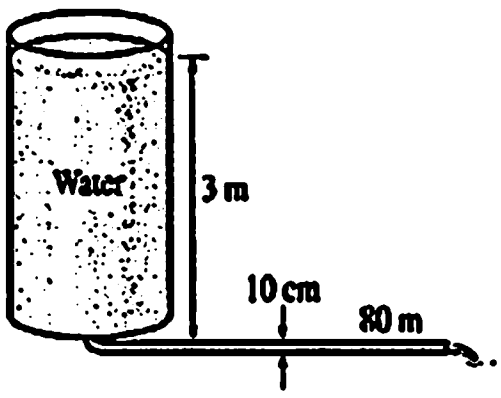


FIGURE P12–70

Step 1 of 5

Given data:

The total irreversible head loss, $h_L = 1.5$ m.

Height of the water in the tank from the surface, $Z_1 = 3$ m.

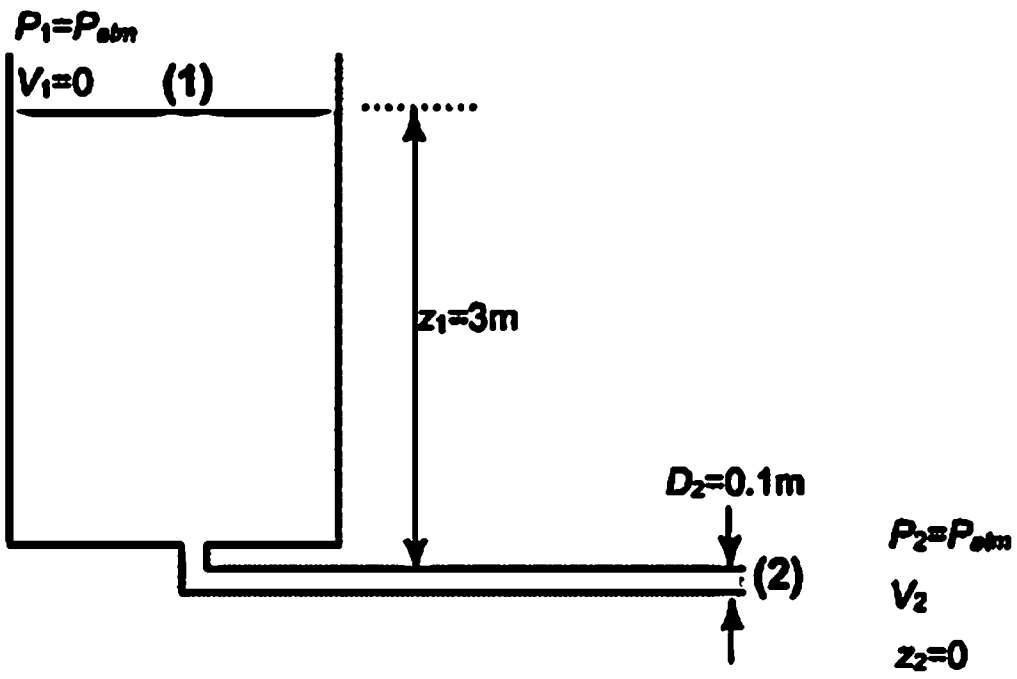
The tank has the orifice at the bottom of the pipe having diameter, $d = 0.1$ m.

The length of the pipe, $L = 80$ m

The kinetic energy corrections can be neglected.

Step 2 of 5

A tank filled of water is orifice at the bottom of the tank as shown in the figure below.



Step 3 of 5

Let us assume

1. The fluid is incompressible and the flow is steady.
2. The frictional effects can be neglected.
3. The kinetic energy correction factors can be neglected, $\alpha_1 = \alpha_2 = 1$.

Step 4 of 5

Applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,s} + h_L$$

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + 3 + 0 = \frac{P_{atm}}{\rho g} + (1)\frac{V_2^2}{2g} + 0 + 0 + 1.5$$

$$\frac{V_2^2}{2g} = 1.5$$

Step 5 of 5

Then the outlet velocity of the water is given by

$$V_2 = \sqrt{1.5(2g)}$$

$$= \sqrt{1.5(2 \times 9.81)}$$

$$= 5.425 \text{ m/s}$$

Therefore the outlet velocity of the water $\boxed{V_2 = 5.425 \text{ m/s}}$

Question: 12.071P

Reconsider Prob. 12-70. Using EES (or other) software, investigate the effect of the tank : - height on the initial discharge velocity of water from the complexly filled tank. Let the tank height vary from 2 to 15 m in increments of 1 m, and assume the irreversible head loss to remain constant. Tabulate and plot the results.

Step 1 of 8

Given data:

The total irreversible head loss, $h_L = 1.5\ \text{m}$.

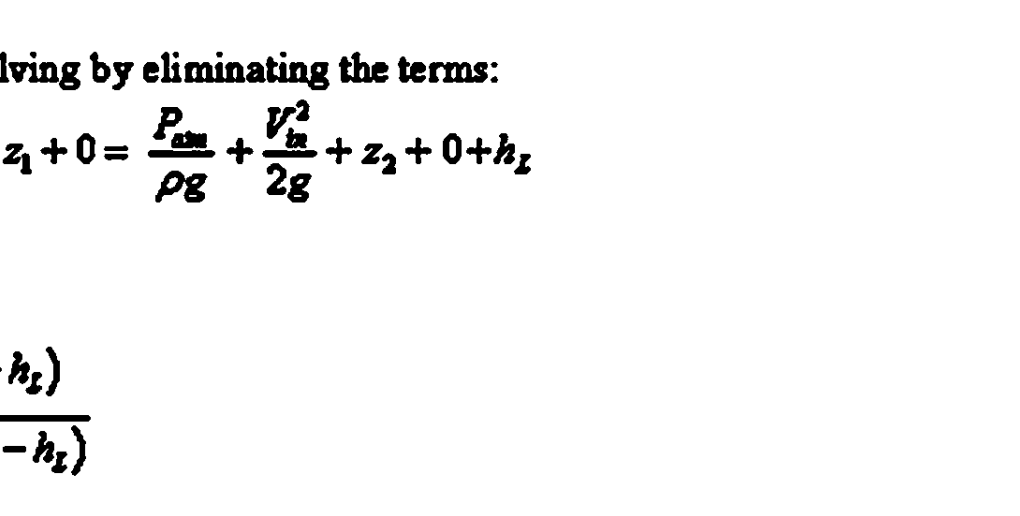
Acceleration due to gravity, $g = 9.81\ \text{m/s}^2$.

Density of water, $\rho = 1000\ \text{kg/m}^3$

Diameter of orifice, $D = 0.1\ \text{m}$

Step 2 of 8

Free body diagram of the tank:



Step 3 of 8

By applying the Bernoulli's flow equation between the points (1) and (2).

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$

where, p is pressure at points (1) and (2).

ρ is density of fluid.

g is acceleration due to gravity

V is velocity at points (1) and (2).

z is elevation at points (1) and (2).

h is head at pump, turbine and total head.

Step 4 of 8

On further solving by eliminating the terms:

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_1 + 0 = \frac{P_{atm}}{\rho g} + \frac{V_{in}^2}{2g} + z_2 + 0 + h_L$$

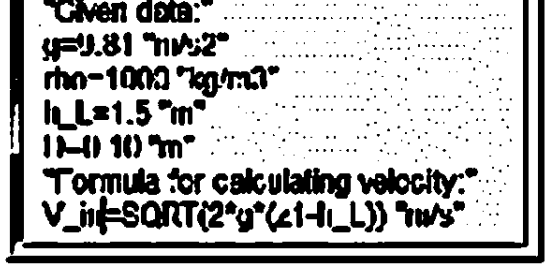
$$\frac{V_{in}^2}{2g} = z_1 - h_L$$

$$V_{in}^2 = 2g(z_1 - h_L)$$

$$V_{in} = \sqrt{2g(z_1 - h_L)}$$

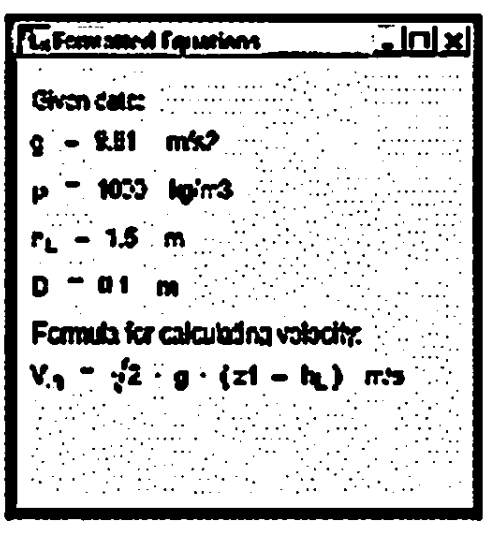
Step 5 of 8

Using the above formula and by EES software, we give the inputs in equation window as shown below:



Step 6 of 8

The equations in the formatted window are shown below:



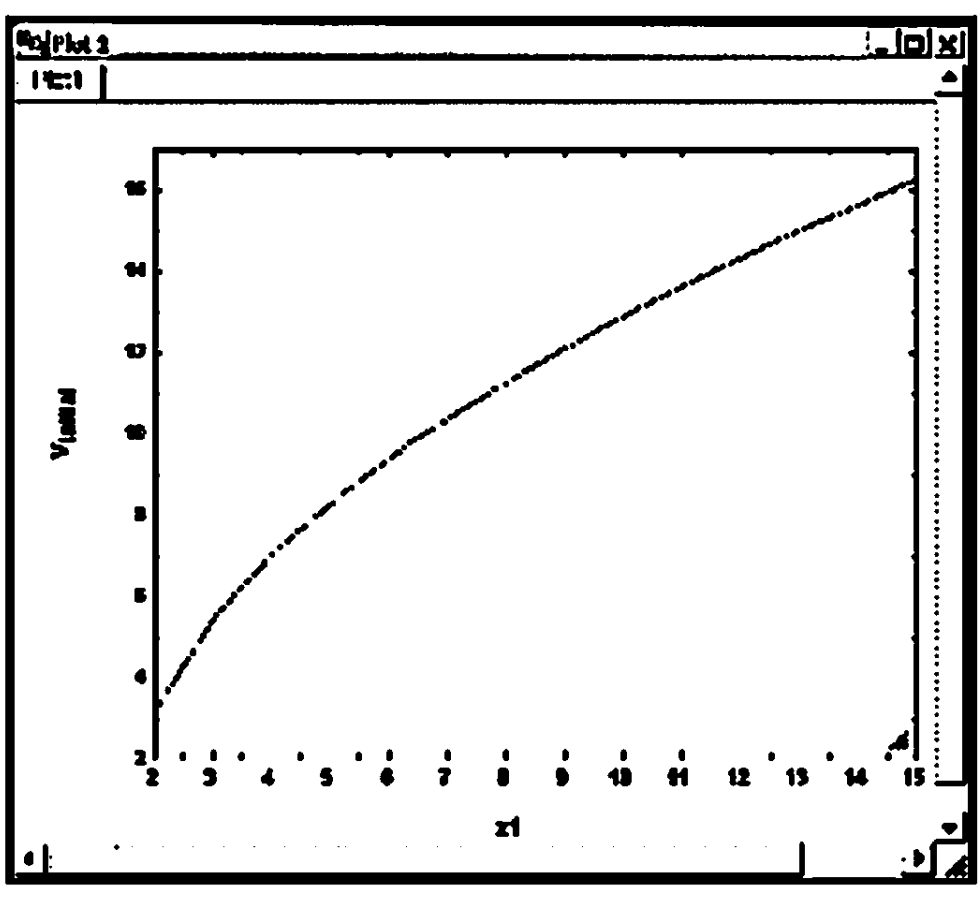
Step 7 of 8

The tabulated values are shown below as in EES varying the tank height from 2m to 15m with increments of 1m:

	z1	V_{inmm}	h_L
Run 1	2	3.132	1.5
Run 2	3.083	5.574	1.5
Run 3	4.167	7.233	1.5
Run 4	5.25	8.578	1.5
Run 5	6.333	9.738	1.5
Run 6	7.417	10.77	1.5
Run 7	8.5	11.72	1.5
Run 8	9.583	12.59	1.5
Run 9	10.67	13.41	1.5
Run 10	11.75	14.18	1.5
Run 11	12.83	14.91	1.5
Run 12	13.92	15.61	1.5
Run 13	15	16.27	1.5

Step 8 of 8

The plot obtained from the above table is shown as:

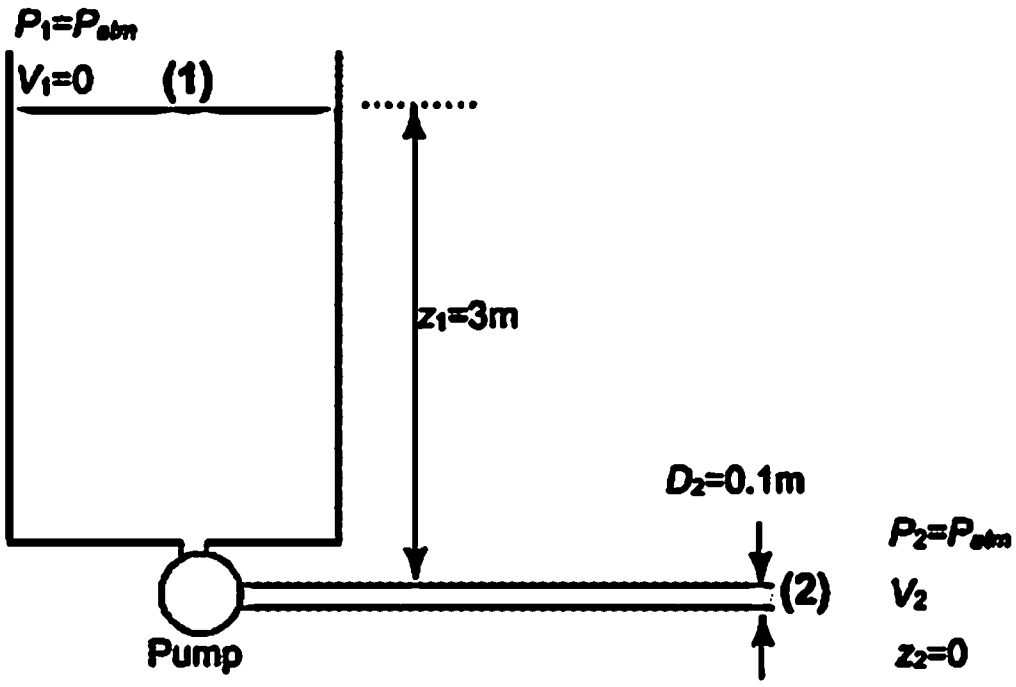


Question: 12.072P

Reconsider Prob. 12–70. In order to drain the tank faster, a pump is installed near the tank exit. Determine the pump head input necessary to establish an average water velocity of 6.5 m/s when the tank is full.

Step 1 of 3

Free body diagram of the tank:



Step 2 of 3

Given data:

The total irreversible head loss, $h_L = 1.5\ \text{m}$.

The exit velocity, $V_2 = 6.5\ \text{m/s}$.

By applying the Bernoulli's flow equation between the points (1) and (2):

$$\frac{p_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{p_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine} + h_L \qquad \ldots\ldots (1)$$

Where, p is pressure at points (1) and (2).

ρ is density of fluid.

g is acceleration due to gravity.

V is velocity at points (1) and (2).

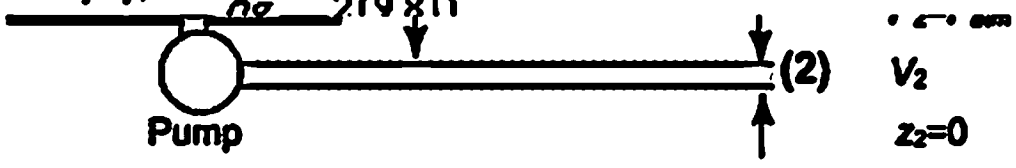
z is elevation at points (1) and (2).

h is head at pump, turbine and total head.

Step 3 of 3

By substituting the given values in (1):

$$\frac{P_{atm}}{\rho g} + \frac{0}{2g} + 3 + h_{pump,u} = \frac{P_{atm}}{\rho g} + \frac{(6.5)^2}{2(9.81)} + 0 + 0 + 1.5$$



Step 2 of 3

Given data:

The total irreversible head loss, $h_L = 1.5\ \text{m}$.

The exit velocity, $V_2 = 6.5\ \text{m/s}$.

Question: 12.073P

The water level in a tank is 120 ft above the ground. A hose is connected to the bottom of the tank, and the nozzle at the end of the hose is pointed straight up. The tank is at sea level, and the water surface is open to the atmosphere. In the line leading from the tank to the nozzle is a pump, which increases the water pressure by 10 psia. Determine the maximum height to which the water stream could rise.

Step 1 of 4

Given data:

The height of the water in the tank from ground level, $z_s = 120\ \text{ft}$.

The pressure rise, $\Delta P = 10\ \text{psia} = 1440\ \text{lbf/ft}^2$.

The pressure at tank surface and maximum height of the jet is atmospheric, P_{atm}.

Step 2 of 4

Density of water,

$$\rho = 62.4\ \text{lbm/ft}^3$$

$$= \frac{62.4}{32.2}\ \text{lbf.s}^2/\text{ft}^4$$

Acceleration due to gravity, $g = 32.2\ \text{ft/s}^2$

The pressure rise is expressed as,

$$\Delta P = \rho g h_{pump,u}$$

Rewriting the above equation:

$$h_{pump,u} = \frac{\Delta P}{\rho g}$$

$$= \frac{1440}{\frac{62.4}{32.2} \times 32.2}$$

$$= 23.08\ \text{ft}$$

Step 3 of 4

By applying the Bernoulli's equation for steady incompressible flow between the tank surface and point of maximum height of the water jet,

$$\frac{P_s}{\rho g} + \alpha_1 \frac{V_s^2}{2g} + z_s + h_{pump,u} = \frac{P_j}{\rho g} + \alpha_2 \frac{V_j^2}{2g} + z_j + h_{minor,e} + h_L$$

Where, the pressures

$$P_s = P_j = P_{atm}$$

The velocity at the surface and maximum height is zero,

$$V_s = V_j = 0$$

Step 4 of 4

The useful pump head:

$$h_{pump,u} = z_j - z_s$$

By substituting values in the above equation:

$$23.08 = z_j - 120$$

$$z_j = 23.08 + 120$$

$$= 143.08$$

Therefore the maximum height to which the water rises is $\boxed{z_j = 143.08\ \text{ft}}$.

Question: 12.074P

A wind tunnel draws atmospheric air at 20°C and 101.3 kPa by a large fan located near the exit of the tunnel. If the air velocity in the tunnel is 80 m/s, determine the pressure in the tunnel.

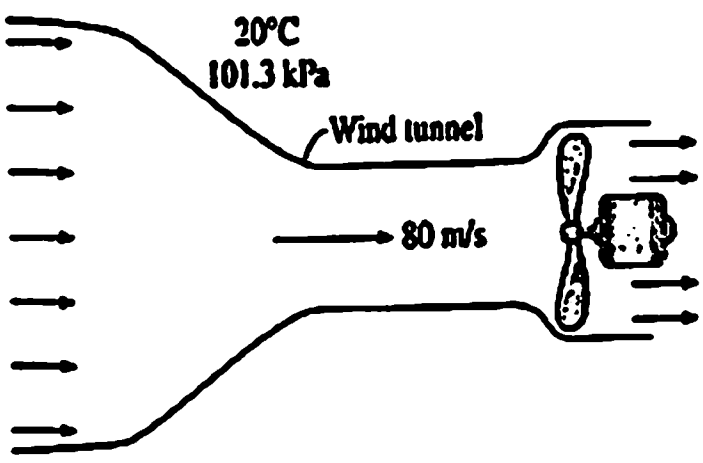
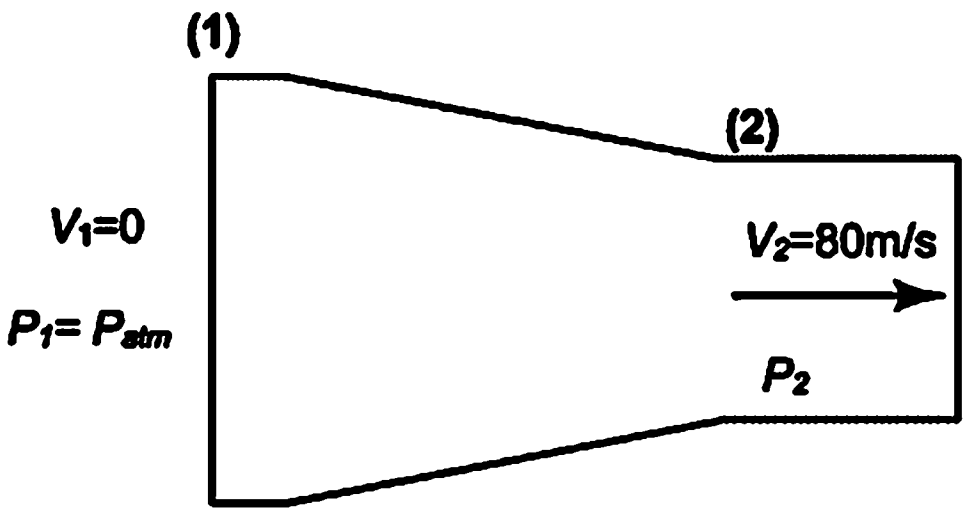


FIGURE P12–74

Step 1 of 3

Draw the wind tunnel showing the velocities and pressures at state 1 and state2.



Step 2 of 3

Find the density of air by using ideal gas equation.

$$\rho=\frac{P_1}{RT_1}$$

Here, P_1 is atmospheric air at state 1, R is gas constant of air, and T_1 is temperature of air at state 1.

Substitute $101.3\,\text{kPa}$ for P_1, $0.287\,\text{kJ/kg.K}$ for R, and $(20+273.14)\,\text{K}$ for T_1

$$\rho_1=\frac{101.3}{(0.287)(293.15)}$$
$$=1.204\,\text{kg/m}^3$$

Step 3 of 3

Apply Bernoulli's equation between the points (1) and (2).

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

Here, $\frac{P}{\rho g}$ indicates pressure head, $\frac{V^2}{2g}$ is velocity head, and z is datum head.

Assume there is no change in datum head at section 1 and 2. Then the modified Bernoulli's equation becomes,

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}$$
$$\frac{P_1}{\rho}+\frac{V_1^2}{2}=\frac{P_2}{\rho}+\frac{V_2^2}{2}$$

Substitute 101.3kPa for P_1, 0 m/s for V_1, 80 m/s for V_2, and $9.81\,\text{m/s}^2$ for g.

$$\frac{101.3\times10^3}{1.204}+0=\frac{P_2}{1.204}+\frac{(80)^2}{2}$$
$$84.136\times10^3=\frac{P_2}{1.204}+3200$$
$$P_2=97446.94\,\text{Pa}$$
$$P_2=97.446\,\text{kPa}$$

Therefore, the pressure inside the wind tunnel is $\boxed{97.446\,\text{kPa}}$

CHAPTER 13
SOLUTIONS

Question: 13.001P

Is momentum a vector? If so, in what direction does it point?

Step 1 of 1

The momentum of a body is defined as the product of the mass and the velocity of a body.
Le the mass of a body is m
The velocity of a body is V
Then the momentum of a body is mV

The velocity is a vector quantity. As the momentum is a constant multiple of velocity, momentum of a body is also a vector and the direction of the momentum is in the direction of velocity of that body.

Question: 13.002P

Express the conservation of momentum principle. What can you say about the momentum of a body if the net force acting on it is zero?

Step 1 of 1

The conservation of momentum principle states that for a system, the total momentum remains constant provided no external resultant force acts on the system. Thus the moment of an isolated system remains constant.

Total initial momentum = Total final momentum.
If no external force is acting on a body then the momentum of the body remains constant.

Question: 13.003P

Express Newton's first, second, and third laws.

Step 1 of 1

Newton's first law states that every object in a state of rest or uniform motion remains in that state unless an external force is applied on it. Newton's first law is also called as the law of inertia.

Newton's second law states that the rate of change of momentum of an object is directly proportional to the resultant force acting on it

For a rigid body of mass m, Newton's second law is expressed as

$$\vec{F} = m\vec{a} = m\frac{d\vec{V}}{dt} = \frac{d\left(m\vec{V}\right)}{dt}$$

Where $\vec{F}$ is the net force acting on the body and $\vec{a}$ is the acceleration of the body under the influence of $\vec{F}$

Newton's third law states that for every action, there is an equal and opposite reaction.

Question: 13.004P

Express Newton's second law of motion for rotating bodies. What can you say about the angular velocity and angular momentum of a rotating nonrigid body of constant mass if the net torque acting on it is zero?

Step 1 of 1

Angular momentum is the product of the moment of inertia and angular velocity. The angular momentum remains constant if the net external torque acting on the system is zero. Newton's second law of motion for rotating bodies states that the rate of change of the angular momentum of the body is equal to the net torque acting it.

If the net torque acting on a rotating a non-rigid body with constant mass is zero then the angular momentum of the body remains constant and the angular velocity will change.

Question: 13.005P

Describe body forces and surface forces, and explain how the net force acting on a control volume is determined. Is fluid weight a body force or surface force? How about pressure?

Step 1 of 1

Body forces are forces that act throughout the entire body of the control volume. Some examples of body forces are gravitational force, magnetic force etc. Forces which act on the point of contact are called surface forces. Some examples of surface forces are pressure, reaction force etc.

In a control volume the net force is the sum of body forces and surface forces.

Fluid weight is a body force, which acts throughout the entire body. Force acting per unit area is pressure. Thus pressure is a surface force.

Question: 13.006P

How do surface forces arise in the momentum analysis of a control volume? How can we minimize the number of surface forces exposed during analysis?

Step 1 of 1

Surface forces are generated when the control volume is isolated from its surroundings for the purpose of analysis.

When the control volume is detached the effect of any detached object from the volume is accounted by a force at that location.

When choosing a control volume, care should be taken to expose only the forces such as reaction forces that are required for analysis and minimum number of other forces.

Question: 13.007P

Explain the importance of the Reynolds transport theorem in fluid mechanics, and describe how the linear momentum equation is obtained from it.

Step 1 of 1

Reynolds transport theorem the relationship between the time rates of change of an extensive property for a system and a control volume. The linear momentum equation is obtained from Reynolds transport theorem by setting $b = \vec{V}$, thus $B = m\vec{V}$ in the Reynolds transport theorem.

Question: 13.008P

What is the importance of the momentum-flux correction factor in the momentum analysis of flow systems? For which type(s) of flow is it significant and must it be considered in analysis: laminar flow, turbulent flow, or jet flow?

Step 1 of 1

The momentum flux can be expressed in terms of the mass flow rate and average flow velocity by using the momentum-flux correction factor (◊).

$$\beta = \frac{1}{A_c}\int_{A_c}\left(\frac{V}{V_{avg}}\right)^2 dA_c$$

For fully developed laminar pipe flows the value of ◊◊ is about 1.33. Thus ◊ cannot be ignored and is significant and should be considered in laminar flow. But for fully developed turbulent flow, the value of ◊◊ is very close to unity so it can be neglected. Also for jet flow the value of ◊ is unity. So for jet flows the momentum flux correction factor can be neglected.

Question: 13.009P

Write the momentum equation for steady one- dimensional flow for the case of no external forces and explain the physical significance of its terms.

Step 1 of 2

The momentum equation for steady one dimensional flow is expressed as,

$$\sum \vec{F} = \sum_{in} \beta \dot{m} \vec{V} - \sum_{out} \beta \dot{m} \vec{V} \quad \text{..... (1)}$$

Here, $\vec{F}$ is the net force acting on the control volume, $\sum_{in} \beta \dot{m} \vec{V}$ represents the incoming momentum flux, $\sum_{out} \beta \dot{m} \vec{V}$ represents the net outgoing momentum flow flux, β is the momentum correction factor, $\dot{m}$ is the mass flow rate, and $\vec{V}$ is the velocity of flow.

Step 2 of 2

As the flow is of steady, one-dimensional, and of no external forces,

$$\sum \vec{F} = 0$$

Substitute 0 for $\sum \vec{F}$ in equation (1).

$$\sum_{in} \beta \dot{m} \vec{V} - \sum_{out} \beta \dot{m} \vec{V} = 0$$

Therefore, the momentum equation for steady, one-dimensional flow for the case of no external forces is $\boxed{\sum_{in} \beta \dot{m} \vec{V} - \sum_{out} \beta \dot{m} \vec{V} = 0}$.

As it is a case of general momentum equation, since there are no external forces acting,

Conservation of momentum holds good.

Question: 13.010P

In the application of the momentum equation, explain why we can usually disregard the atmospheric pressure and work with gage pressures only.

Step 1 of 1

Atmospheric pressure tends to act in all directions. As a result, individual atmospheric forces tend to cancel out each other when taken into analysis.

Due to the above reason we disregard atmospheric pressure and consider the gage pressures only during the application of momentum equation.

Question: 13.011P

A rocket in space (no friction or resistance to motion) can expel gases relative to itself at some high velocity *V*. Is *V* the upper limit to the rocket's ultimate velocity?

Step 1 of 1

When there is no friction the velocity of the rocket will continue to increase. Thus *V* is not the upper limit to the rocket's ultimate velocity. So if there is no resistance to motion, then the rocket will accelerate continuously.

Question: 13.012P

Describe in terms of momentum and airflow how a helicopter is able to hover.

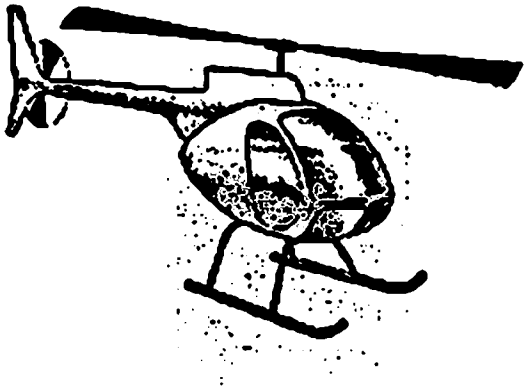


FIGURE P13–12C

Step 1 of 1

The propeller blades of a helicopter spins, hence it creates a downdraft of air. This pushes the helicopter. The air beneath the blades thus, has greater pressure than the air above them. This momentum in the air creates a lift. To hover a helicopter the momentum created by the flow of air must be balanced by the lift force of the helicopter.

Question: 13.013P

Does it take more, equal, or less power for a helicopter to hover at the top of a high mountain than it does at sea level? Explain.

Step 1 of 1

At higher altitudes the density of the air decreases, so more air must be forced into the downdraft by the helicopter blades in order to provide the same lift force as of the sea level. Thus, it requires more power for a helicopter to hover on the top of a high mountain than it does at sea level.

Question: 13.014P

In a given location, would a helicopter require more energy in summer or winter to achieve a specified performance? Explain.

Step 1 of 1

In summer season the density of the air is less. So in order to create lift, more air must be forced into the downdraft by the helicopter blades. In winter the density of the air is high compared to that of summer. Therefore, less air must be forced into the downdraft by the helicopter blades to provide the same lift.

Thus a helicopter requires more energy in summer than in winter to achieve a specified performance.

Question: 13.015P

A horizontal water jet from a nozzle of constant exit cross section impinges normally on a stationary vertical flat plate. A certain force F is required to hold the plate against the water stream. If the water velocity is doubled, will the necessary holding force also be doubled? Explain.

Step 1 of 3

The following assumptions are made for explanation:

(1) The flow of water at the nozzle outlet is steady.
(2) The water splatters in direction normal to the approach of the water jet.
(3) The water jet is exposed to atmosphere, thus the pressure of water jet and splattered water is atmospheric which is neglected.
(4) The vertical forces and momentum fluxes are not considered since they have no effect on horizontal reaction force.
(5) The effect of momentum flux correction factor is negligible, $\beta \cong 1$

Step 2 of 3

For the total reaction force, we have

$$\begin{aligned} F_R &= \beta \dot{m} V \\ &= \dot{m} V \\ &= (\rho A V) V \\ &= \rho A V^2 \end{aligned}$$

Step 3 of 3

When the velocity is doubled,

$$\begin{aligned} F_R &= \rho A (2V)^2 \\ &= 4 \rho A V^2 \end{aligned}$$

From the above result we can conclude that when the velocity is doubled the resultant force increases by a factor of four rather than being doubled.

Question: 13.016P

Two firefighters are fighting a fire with identical water hoses and nozzles, except that one is holding the hose straight so that the water leaves the nozzle in the same direction it comes, while the other holds it backward so that the water makes a U-turn before being discharged. Which firefighter will experience a greater reaction force?

Step 1 of 3

Given data:

Fire fighter (1) holds the hose straight so that water leaves in the same direction as it comes.

Fire fighter (2) holds the hose straight backwards so that water makes a U-turn before leaving the nozzle.

Step 2 of 3

Consider the case of fire fighter (2).

The water is discharged after undergoing a turn of 180°. This is similar to the discharge of water from a reversing elbow.

We know the reaction force in this case,

$$F_{Rx} = -\beta \dot{m}(V_2 + V_1) - P_{1,gage} A_1 \qquad \ldots\ldots (3)$$

Step 3 of 3

Consider the case of fire fighter (1).

The water is discharged from the nozzle along the direction of flow. That is in the positive x-direction.

We know the reaction force in this case,

$$F_{Rx} = \beta \dot{m}(V_2 + V_1) - P_{1,gage} A_1 \qquad \ldots\ldots (4)$$

On comparing the equations (3) and (4),

We can conclude that the fire fighter (2) will be experiencing greater reaction force than fire fighter (1).

Question: 13.017P

A constant-velocity horizontal water jet from a stationary nozzle impinges normally on a vertical flat plate that rides on a nearly frictionless track. As the water jet hits the plate, it begins to move due to the water force. Will the acceleration of the plate remain constant or change? Explain.

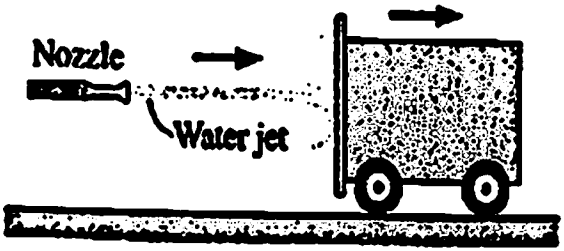


FIGURE P13–17C

Step 1 of 1

The impulse force exerted by the water on the plate, $F = \dot{m}V$

Where V is the relative velocity between water and the plate

We know mass flow rate, $\dot{m} = \rho AV$

Thus the impulse force becomes

$$F = (\rho AV)V$$

$$= \rho AV^2$$

Now, as the water jet hits the plate, it moves. Thus the relative velocity will decrease as the plate moves. As the relative velocity decreases the impulse force will also decrease.

The acceleration of the plate

$$a = \frac{F}{m}$$

Since the force decrease the acceleration of the plate will also decrease.

Therefore the acceleration of the plate is not constant.

Question: 13.018P

A horizontal water jet of constant velocity V from a stationary nozzle impinges normally on a vertical flat plate that rides on a nearly frictionless track. As the water jet hits the plate, it begins to move due to the water force. What is the highest velocity the plate can attain? Explain.

Step 1 of 2

Write the equation for impulse force exerted by the water jet on the plate.

$$F = \dot{m}(V-u)$$

Here, $(V-u)$ is the relative velocity between water and the plate, V is the velocity of jet and u is the velocity of plate.

Write the equation for mass flow rate.

$$\dot{m} = \rho A(V-u)$$

Here, ρ is the density and A is the cross sectional area.

Substitute $\rho A(V-u)$ for $\dot{m}$.

$$\begin{aligned} F &= \dot{m}(V-u) \\ &= (\rho A(V-u))(V-u) \\ &= \rho A(V-u)^2 \end{aligned}$$

Now, as the water jet hits the plate, it moves. Thus the relative velocity will decrease as the plate moves. As the relative velocity decreases the impulse force will also decrease.

Step 2 of 2

The maximum force exerted by the water jet on the plate can be obtained at the velocity of plate is zero. As the velocity of plate increases the relative velocity decreases, thereby the force exerted on the plate decreases. The minimum force on the plate can be obtained at the velocity of plate is equal to the velocity of jet. If the velocity of plate is more than the velocity of jet, then the force exerted on the plate becomes negative, which is impossible case.

Therefore, the maximum velocity of the plate is equal to the velocity of jet.

Question: 13.019P

A horizontal water jet of constant velocity V impinges normally on a vertical flat plate and splashes off the sides in the vertical plane. The plate is moving toward the oncoming water jet with velocity 1/2 V. If a force F is required to maintain the plate stationary, how much force is required to move the plate toward the water jet?

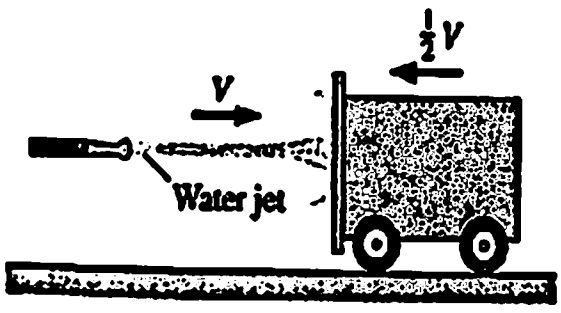


FIGURE P13–19

Step 1 of 4

The Schematic representation of the given system is shown below.

CV

Question:

A horizontal water jet of constant velocity V impinges normally on a vertical flat plate and splashes off the sides in the vertical plane. The plate is moving toward the oncoming water jet with velocity 1/2 V. If a force F is required to maintain the plate stationary, how much force is required to move the plate toward the water jet?

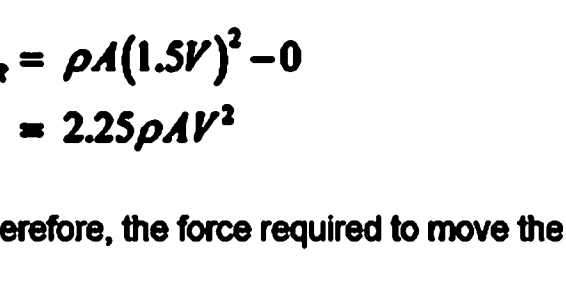


FIGURE P13–19

Step 1 of 4

The Schematic representation of the given system is shown below.

The effect of momentum flux correction factor is negligible.

CV

Step 3 of 4

When the plate is moving with a velocity $0.5V$ against the jet of velocity V, the relative velocity of the jet with respect to the plate can be represented as

$$\begin{aligned} V_{in} &= V-(-0.5V) \\ &= 1.5V \end{aligned}$$

Calculate the reaction force by applying momentum equation as shown below.

$$\begin{aligned} \sum \overline{F_R} &= \sum_{in} \beta \dot{m} \overline{V} - \sum_{out} \beta \dot{m} \overline{V} \\ &= \dot{m}V_{in} - \dot{m}V_{out} \\ &= (\rho A V_{in})(V_{in}) - (\rho A V_{out})(V_{out}) \\ &= \left[\rho A (V_{in})^2\right] - \left[\rho A (V_{out})^2\right] \end{aligned}$$

Step 4 of 4

Substitute $1.5V$ for V_{in} and 0 for V_{out}.

$$\begin{aligned} F_R &= \rho A (1.5V)^2 - 0 \\ &= 2.25\rho A V^2 \end{aligned}$$

Therefore, the force required to move the plate towards the jet is $\boxed{2.25\rho A V^2}$.

Question: 13.020P

A 90° elbow in a horizontal pipe is used to direct water flow upward at a rate of 30 kg/s. The diameter of the entire elbow is 10 cm. The elbow discharges water into the atmosphere, and thus the pressure at the exit is the local atmospheric pressure. The elevation difference between the centers of the exit and the inlet of the elbow is 40 cm. The weight of the elbow and the water in it is considered to be negligible. Determine (*a*) the gage pressure at the center of the inlet of the elbow and (b) the anchoring force needed to hold the elbow in place. Take the momentum-flux correction factor to be 1.03 at both the inlet and the outlet.

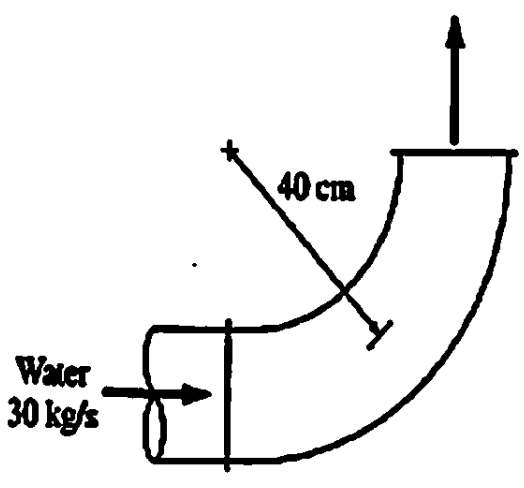


FIGURE P13–20

Step 1 of 8

Given data:

The upward flow rate, $\dot{m}_1 = 30\ \text{kg/s}$

The diameter of the elbow, $D = 10\text{cm} = 0.1\text{m}$

The momentum-flux correction factor for each inlet and outlet, $\beta = 1.03$

Step 2 of 8

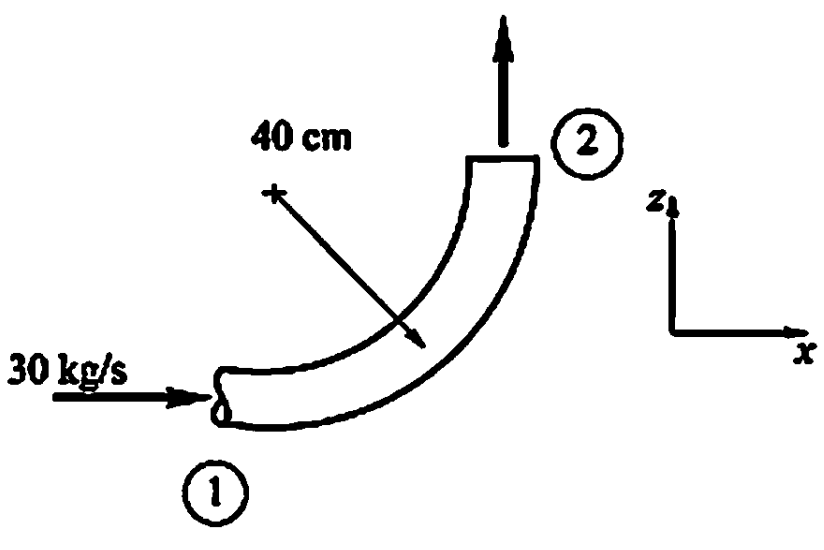
Assumptions:

1. The flow is steady, frictionless, incompressible, and irrotational.
2. Water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.
3. The weight of the elbow and the water in it is negligible.

Properties: Take the density of water as 1000 kg/m^3.

Step 3 of 8

A 90^0 elbow is as shown in figure:



Step 4 of 8

(a)

Consider elbow as the control volume.

Now, the inlet is at point 1 and the outlet at point 2.

Since the flow is steady, we have

$\dot{m}_1 = \dot{m}_2 = \dot{m} = 30\ \text{kg/s}$

Mass flow rate,

$\dot{m} = \rho V A$

$$V = \frac{\dot{m}}{\rho\left(\frac{\pi D^2}{4}\right)}$$

$$= \frac{30\ \text{kg/s}}{(1000\ \text{kg/m}^3)\frac{\pi(0.1\ \text{m})^2}{4}}$$

$V = 3.82\ \text{m/s}$

Step 5 of 8

The mean inlet and outlet velocities of water, $V = V_1 = V_2 = 3.82\ \text{m/s}$

From Bernoulli's equation, we have

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$P_1 - P_2 = \rho g(z_2 - z_1)$$

Hence gage pressure at section 1, we have

$$P_{1\text{gage}} = \rho g(z_2 - z_1)$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.4\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 3.924\ \text{kN/m}^2$$

$$= 3.924\ \text{kPa}$$

Thus, the gage pressure, $\boxed{P_{1\text{gage}} = 3.92\ \text{kPa}}$

Step 6 of 8

(b)

Let the anchoring force of the elbow in x direction be F_{Rx} and anchoring force of the elbow in z direction be F_{Rz}

The momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Now, the components of the equation in x direction,

$$F_{Rx} + P_{1\text{gage}} A_1 = 0 - \beta \dot{m}(+V_1)$$

$$= -\beta \dot{m}(+V_1)$$

On substituting the known values in the above equation, we have

$$F_{Rx} = -\beta \dot{m}(+V_1) - P_{1\text{gage}} A_1$$

$$= -1.03(30\ \text{kg/s})(3.82\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) - (3924\ \text{N/m}^2)\frac{\pi(0.1\ \text{m})^2}{4}$$

$$= -118.038\ \text{N} - 30.819\ \text{N}$$

$$= -148.86\ \text{N}$$

Step 7 of 8

The components of the equation in z direction,

$$F_{Rz} = \beta \dot{m}(+V_2)$$

$$= \beta \dot{m} V_2$$

Substituting the known values in the above equation, we have

$$F_{Rz} = \beta \dot{m} V_2$$

$$= 1.03(30\ \text{kg/s})(3.81\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 117.729\ \text{N}$$

Step 8 of 8

The resultant force of the given system, we have

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2}$$

$$= \sqrt{(-148.86)^2 + 117.729^2}$$

$$= 189.78\ \text{N}$$

The angle for the force acting in the positive x direction,

$$\theta = \tan^{-1}\left(\frac{F_{Rz}}{F_{Rx}}\right)$$

$$= \tan^{-1}\left(\frac{117.729}{-148.86}\right)$$

$$= -38.34^\circ$$

Therefore, the anchoring force, $\boxed{F_R = 189.78\ \text{N}}$

The force is acting at an angle $\boxed{141.66^\circ}$ from the positive x direction.

Question: 13.021P

Repeat Prob. 13-20 for the case of another (identical) elbow attached to the existing elbow so that the fluid makes a U-turn.

Step 1 of 7

Given data:

The upward flow rate, $\dot{m}_1 = 30\ \text{kg/s}$

The diameter of the elbow, $D = 10\text{cm} = 0.1\text{m}$

The momentum-flux correction factor for each inlet and outlet, $\beta = 1.03$

Density of water is taken as $1000\ \text{kg/m}^3$

Step 2 of 7

Assumptions:

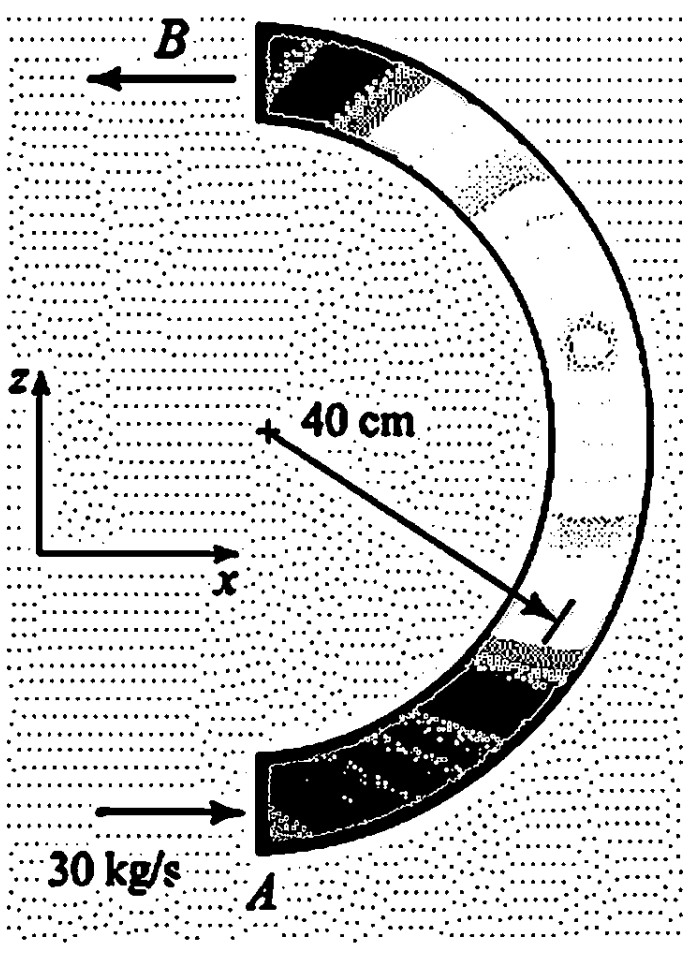
A. The flow is steady, frictionless, incompressible, irrotational and one dimensional.

B. Water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.

C. The weight of the elbow and the water in it is negligible.

Step 3 of 7

Figure showing the elbow bend,



Step 4 of 7

a) Let us consider the elbow as a control volume and water enters at point A and the outlet at point B.

Now, $\dot{m}_1 = \dot{m}_2 = \dot{m} = 30\ \text{kg/s}$, since the flow is steady.

And mass flow rate is given by, $\dot{m} = \rho VA$

Calculating volume;

$$V = \frac{\dot{m}}{\rho\left(\frac{\pi D^2}{4}\right)}$$

$$= \frac{30\ \text{kg/s}}{\left(1000\ \text{kg/m}^3\right)\frac{\pi(0.1\ \text{m})^2}{4}}$$

The volume, $V = 3.82\ \text{m/s}$

Step 5 of 7

The mean inlet and outlet velocities of water are $V = V_1 = V_2 = 3.81\ \text{m/s}$.

Using the Bernoulli's equation,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$P_1 - P_2 = \rho g(z_2 - z_1)$$

Thus,

$$P_{\text{gage}} = \rho g(z_2 - z_1)$$

$$= \left(1000\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)(0.8\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 7.848\ \text{kN/m}^2$$

$$= 7.848\ \text{kPa}$$

Thus, the gage pressure is $\boxed{P_{\text{gage}} = 7.85\ \text{kPa}}$.

Step 6 of 7

b)

Let the anchoring force of the elbow in x direction be F_{Rx} and anchoring force of the elbow in z direction be F_{Rz}. The momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m}\vec{V} - \sum_{in} \beta \dot{m}\vec{V}.$$

Step 7 of 7

The components of the equation in x direction is

$$F_{Rx} + P_{\text{gage}}A_1 = \beta \dot{m}(-V_2) - \beta \dot{m}(+V_1)$$

$$= -2\beta \dot{m}V$$

$$F_{Rx} = -2\beta \dot{m}V - P_{\text{gage}}A_1$$

$$= -2(1.03)(30\ \text{kg/s})(3.82\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) - \left(7848\ \text{N/m}^2\right)\frac{\pi(0.1\ \text{m})^2}{4}$$

$$= -236.058\ \text{N} - 61.63\ \text{N}$$

$$= -297.688\ \text{N}$$

The components of the equation in z direction is zero thus, $F_{Rz} = 0$. Thus

$$F_R = F_{Rx} = -297.688\ \text{N}$$

Therefore, the anchoring force is $\boxed{F_R = 298\ \text{N}}$. The force is acting at negative x direction.

Question: 13.022P

A horizontal water jet impinges against a vertical flat plate at 25 ft/s and splashes off the sides in the vertical plane. If a horizontal force of 350 lbf is required to hold the plate against the water stream, determine the volume flow rate of the water.

Step 1 of 3

Given data:

The mass flow rate of water, $\dot{m} = 25\,\text{ft/s}$

The horizontal force required to hold the plate against the water stream,

$$\begin{aligned} F_{Rx} &= 350\,\text{lbf} \\ &= 350 \times 32.2 \quad \left(\text{since}, 1\,\text{lbf} = 32.2\,\text{lbm}\cdot\text{ft/s}^2\right) \\ &= 11270\,\text{lbm}\cdot\text{ft/s}^2 \end{aligned}$$

The density of water, $\rho = 62.4\,\text{lbm/ft}^3$

Step 2 of 3

By using moment equation for steady one-dimensional flow in x -direction, we get

$$\sum F = \sum_{\text{out}} \beta \dot{m} V - \sum_{\text{in}} \beta \dot{m} V$$

$$-F_{Rx} = -\dot{m} V_1$$

$$F_{Rx} = \dot{m} V_1$$

$$\dot{m} = \frac{F_{Rx}}{V_1}$$

$$= \frac{11270}{25}$$

$$= 450.8\,\text{ft/s}$$

Step 3 of 3

By using the formula for volume flow rate, we get

$$\dot{V} = \frac{\dot{m}}{\rho}$$

$$= \frac{450.8}{62.4}$$

$$= 7.224\,\text{ft}^3/\text{s}$$

Therefore, the volume flow rate, $\boxed{\dot{V} = 7.224\,\text{ft}^3/\text{s}}$.

Question: 13.023P

A reducing elbow in a horizontal pipe is used to deflect water flow by an angle $\theta = 45°$ from the flow direction while accelerating it. The elbow discharges water into the atmosphere. The cross-sectional area of the elbow is 150 cm2 at the inlet and 25 cm2 at the exit. The elevation difference between the centers of the exit and the inlet is 40 cm. The mass of the elbow and the water in it is 50 kg. Determine the anchoring force needed to hold the elbow in place. Take the momentum- flux correction factor to be 1.03 at both the inlet and outlet.

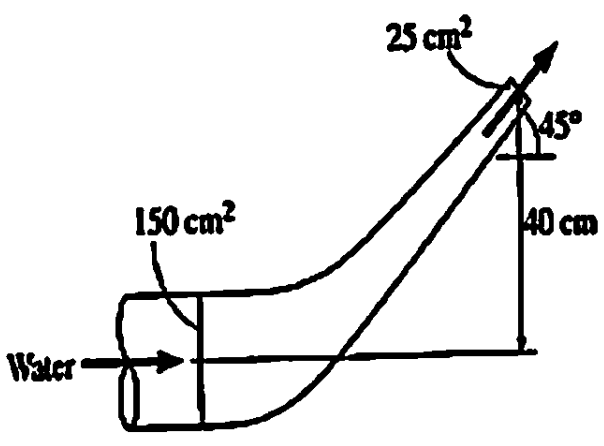


FIGURE P13–23

Step 1 of 8

Given data:-

The cross sectional area at inlet is $A_1 = 150\ \text{cm}^2 = 0.015\text{m}^2$.

The cross sectional area at exit is $A_1 = 25\ \text{cm}^2 = 0.0025\text{m}^2$.

The elevation difference is $z_2 - z_1 = 40\,\text{cm} = 0.4\,\text{m}$.

The angle is $\theta = 45°$.

The mass of elbow and the water in it is $m = 50\ \text{kg}$.

The momentum-flux correction factor for each inlet and outlet is $\beta = 1.03$.

Step 2 of 8

Assumptions

1. The flow is steady, frictionless, incompressible, and irrotational.
2. Water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.
3. Let the mass flow rate be $\dot{m} = 30\ \text{kg/s}$

Properties: Take the density of water as $1000\ \text{kg/m}^3$.

Step 3 of 8

Take the elbow as the control volume. Now, the inlet is at point 1 and the outlet at point 2.

Now, $\dot{m}_1 = \dot{m}_2 = \dot{m} = 30\ \text{kg/s}$,

Since the flow is steady. Also, $\dot{m} = \rho VA$.

Now, the inlet velocity is

$$V_1 = \frac{\dot{m}}{\rho A_1}$$

$$= \frac{30\ \text{kg/s}}{(1000\ \text{kg/m}^3)(0.015\ \text{m}^2)}$$

$$= 2\ \text{m/s}$$

Now, the outlet velocity is

$$V_2 = \frac{\dot{m}}{\rho A_2}$$

$$= \frac{30\ \text{kg/s}}{(1000\ \text{kg/m}^3)(0.0025\ \text{m}^2)}$$

$$= 12\ \text{m/s}$$

Step 4 of 8

Now, use Bernoulli's equation,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$P_1 - P_2 = \rho g\left[\frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1)\right]$$

Thus,

$$P_{1,\text{gage}} = \rho g\left[\frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1)\right]$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)\left[\frac{(12\ \text{m/s})^2 - (2\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} + 0.4\right]\left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 73.92\ \text{kN/m}^2$$

$$= 73.92\ \text{kPa}$$

Step 5 of 8

Let the anchoring force of the elbow in x direction be F_{Rx} and anchoring force of the elbow in z direction be F_{Rz}. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m}\vec{V} - \sum_{in} \beta \dot{m}\vec{V}.$$

Now, the components of the equation in x direction is

$$F_{Rx} + P_{1,\text{gage}} A_1 = \beta \dot{m} V_2 \cos\theta - \beta \dot{m} V_1$$

$$F_{Rx} = \beta \dot{m}(V_2 \cos\theta - V_1) - P_{1,\text{gage}} A_1$$

$$= 1.03(30\ \text{kg/s})\left[(12\ \text{m/s})\cos 45° - 2\ \text{m/s}\right]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) - (73920\ \text{N/m}^2)(0.015)$$

$$= 200.40\ \text{N} - 1108.8\ \text{N}$$

$$= -908.4\ \text{N}$$

Step 6 of 8

Now, the components of the equation in z direction is

$$F_{Rz} - W = \beta \dot{m} V_2 \sin\theta$$

$$F_{Rz} = \beta \dot{m} V_2 \sin\theta + W$$

Now, weight of the elbow and the water in it is

$$W = mg$$

$$= (50\ \text{kg})(9.81\ \text{m/s}^2)$$

$$= 490.5\ \text{N}$$

Step 7 of 8

Now, substitute the known values and solve F_{Rx}.

$$F_{Rx} = 1.03(30\ \text{kg/s})\left[(12\ \text{m/s})\sin 45°\right]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) + (490.5\ \text{N})$$

$$= 752.69\ \text{N}$$

Now,

The resultant force

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2}$$

$$= \sqrt{(-908.4)^2 + (752.69)^2}$$

$$= 1179.72\ \text{N}$$

Step 8 of 8

Now, the angle,

$$\theta = \tan^{-1}\left(\frac{F_{Rz}}{F_{Rx}}\right)$$

$$= \tan^{-1}\left(\frac{752.69}{-908.4}\right)$$

$$= -39.64°$$

Therefore, the anchoring force is $\boxed{F_R = 1179.72\ \text{N}}$. The force is acting at an angle $\boxed{140.36°}$ from the positive x direction.

Question: 13.024P

Repeat Prob. 13-23 for the case of $\theta = 110°$.

Step 1 of 8

Given data:-

The cross sectional area at inlet is $A_1 = 150\ \text{cm}^2 = 0.015\,\text{m}^2$.

The cross sectional area at exit is $A_1 = 25\ \text{cm}^2 = 0.0025\,\text{m}^2$.

The elevation difference is $z_2 - z_1 = 40\,\text{cm} = 0.4\,\text{m}$.

The angle is $\theta = 110°$.

The mass of elbow and the water in it is $m = 50\ \text{kg}$.

The momentum-flux correction factor for each inlet and outlet is $\beta = 1.03$.

Step 2 of 8

Assumptions

1. The flow is steady, frictionless, incompressible, and irrotational.
2. Water is discharged to the atmosphere, and thus the gage pressure at the outlet is zero.
3. Let the mass flow rate be $\dot{m} = 30\ \text{kg/s}$

Properties: Take the density of water as $1000\ \text{kg/m}^3$.

Step 3 of 8

Take the elbow as the control volume. Now, the inlet is at point 1 and the outlet at point 2.

Now, $\dot{m}_1 = \dot{m}_2 = \dot{m} = 30\ \text{kg/s}$, since the flow is steady. Also, $\dot{m} = \rho V A$.

Now, the inlet velocity is

$$V_1 = \frac{\dot{m}}{\rho A_1}$$

$$= \frac{30\ \text{kg/s}}{(1000\ \text{kg/m}^3)(0.015\ \text{m}^2)}$$

$$= 2\ \text{m/s}$$

Now, the outlet velocity is

$$V_2 = \frac{\dot{m}}{\rho A_2}$$

$$= \frac{30\ \text{kg/s}}{(1000\ \text{kg/m}^3)(0.0025\ \text{m}^2)}$$

$$= 12\ \text{m/s}$$

Step 4 of 8

Now, use Bernoulli's equation,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$P_1 - P_2 = \rho g\left[\frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1)\right]$$

Thus,

$$P_{1,\text{gage}} = \rho g\left[\frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1)\right]$$

$$= (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)\left[\frac{(12\ \text{m/s})^2 - (2\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} + 0.4\right]\left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 73.92\ \text{kN/m}^2$$

$$= 73.92\ \text{kPa}$$

Step 5 of 8

Let the anchoring force of the elbow in x direction be F_{Rx} and anchoring force of the elbow in z direction be F_{Rz}. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Now, the components of the equation in x direction is

$$F_{Rx} + P_{1,\text{gage}} A_1 = \beta \dot{m} V_2 \cos\theta - \beta \dot{m} V_1$$

$$F_{Rx} = \beta \dot{m}(V_2 \cos\theta - V_1) - P_{1,\text{gage}} A_1$$

$$= 1.03(30\ \text{kg/s})\left[(12\ \text{m/s})\cos 110° - 2\ \text{m/s}\right]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) - (73920\ \text{N/m}^2)(0.015)$$

$$= -188.62\ \text{N} - 1108.8\ \text{N}$$

$$= -1297.42\ \text{N}$$

Step 6 of 8

Now, the components of the equation in z direction is

$$F_{Rz} - W = \beta \dot{m} V_2 \sin\theta$$

$$F_{Rz} = \beta \dot{m} V_2 \sin\theta + W$$

Now, weight of the elbow and the water in it is

$$W = mg$$

$$= (50\ \text{kg})(9.81\ \text{m/s}^2)$$

$$= 490.5\ \text{N}$$

Step 7 of 8

Now, substitute the known values and solve F_{Rz}.

$$F_{Rz} = 1.03(30\ \text{kg/s})\left[(12\ \text{m/s})\sin 110°\right]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) + (490.5\ \text{N})$$

$$= 838.94\ \text{N}$$

Now,

The resultant force

$$F_R = \sqrt{F_{Rx}^2 + F_{Rz}^2}$$

$$= \sqrt{(-1297.42)^2 + (838.94)^2}$$

$$= 1545.03\ \text{N}$$

Step 8 of 8

Now, the angle,

$$\theta = \tan^{-1}\left(\frac{F_{Rz}}{F_{Rx}}\right)$$

$$= \tan^{-1}\left(\frac{838.94}{-1297.42}\right)$$

$$= -32.89°$$

Therefore, the anchoring force is $\boxed{F_R = 1545\ \text{N}}$. The force is acting at an angle $\boxed{147°}$ from the positive x direction.

Question: 13.025P

Water accelerated by a nozzle to 20 m/s strikes the vertical back surface of a cart moving horizontally at a constant velocity of 5 m/s in the flow direction. The mass flow rate of water through the stationary nozzle is 30 kg/s. After the strike, the water stream splatters off in all directions in the plane of the back surface, (*a*) Determine the force that needs to be applied by the brakes of the cart to prevent it from accelerating. (*b*) If this force were used to generate power instead of wasting it on the brakes, determine the maximum amount of power that could ideally be generated.

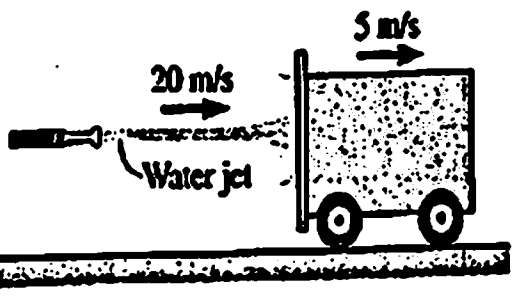


FIGURE P13–25

Step 1 of 5

Given data:-

The velocity of water, $v_{jet} = 20\ \text{m/s}^2$

The velocity of the cart, $v_{cart} = 5\ \text{m/s}^2$

The mass flow rate, $\dot{m}_{jet} = 30\ \text{kg/s}$

Step 2 of 5

Assumptions are

1. The flow is steady, frictionless and incompressible.
2. Water is discharged to the atmosphere, and thus it is not considered.
3. The flow is jet flow so the momentum-flux correction factor is unity.
4. Neglect the friction.

Step 3 of 5

Consider the cart as the control volume.

The relative velocity between cart and the water jet,

$$\begin{aligned} V_r &= V_{jet} - V_{cart} \\ &= (20\ \text{m/s}) - (5\ \text{m/s}) \\ &= 15\ \text{m/s} \end{aligned}$$

The mass flow rate with respect to the relative velocity is

$$\begin{aligned} \dot{m}_r &= \frac{V_r}{V_{jet}} \dot{m}_{jet} \\ &= \frac{15\ \text{m/s}}{20\ \text{m/s}} (30\ \text{kg/s}) \\ &= 22.5\ \text{kg/s} \end{aligned}$$

Step 4 of 5

Let F_{Rx} be in positive x direction. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Thus, $F_{Rx} = \dot{m} V$. Now, the braking force is

$$\begin{aligned} F_{brake} &= -\dot{m}_r V_r \\ &= -(22.5\ \text{kg/s})(15\ \text{m/s}) \left(\frac{1\ \text{N}}{1\ \text{kg} \cdot \text{m/s}^2} \right) \\ &= -337.5\ \text{N} \end{aligned}$$

Therefore, the braking force is $\boxed{F_{brake} = -338\ \text{N}}$

Step 5 of 5

Power dissipated by the brake is

$$\begin{aligned} P &= F_{brake} V_{cart} \\ &= (337.5\ \text{N})(5\ \text{m/s}) \\ &= 1687.5\ \text{N} \cdot \text{m/s} \\ &= 1687.5\ \text{W} \end{aligned}$$

Therefore, the maximum power that could be generated is $\boxed{P = 1.69\ \text{kW}}$

Question: 13.026P

Reconsider Prob.13-25 .If the mass of the cart is 400 kg and the brakes fail, determine the acceleration of the cart when the water first strikes it.Assume the mass of water that wets the back surface is negligible.

Step 1 of 4

Given data:

The velocity of water, $v_{jet} = 20\ \text{m/s}^2$

The velocity of the cart, $v_{cart} = 5\ \text{m/s}^2$

The mass flow rate, $\dot{m}_{jet} = 30\ \text{kg/s}$

The mass of the cart, $m = 400\ \text{kg}$

Step 2 of 4

Assumptions are

- The flow is steady, frictionless and incompressible.
- Water is discharged to the atmosphere, and thus the pressure is not considered.
- The flow is jet flow so the momentum-flux correction factor is unity.
- Neglect the friction.

Step 3 of 4

Consider the cart as the control volume.

Now, the relative velocity between cart and the water jet,

$$\begin{aligned} V_r &= V_{jet} - V_{cart} \\ &= (20\ \text{m/s}) - (5\ \text{m/s}) \\ &= 15\ \text{m/s} \end{aligned}$$

Now, the mass flow rate with respect to the relative velocity is

$$\begin{aligned} \dot{m}_r &= \frac{V_r}{V_{jet}} \dot{m}_{jet} \\ &= \frac{15\ \text{m/s}}{20\ \text{m/s}} (30\ \text{kg/s}) \\ &= 22.5\ \text{kg/s} \end{aligned}$$

Step 4 of 4

Let F_{Rx} be in positive x direction. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Thus,

$$\begin{aligned} F_{Rx} &= \dot{m}_r V_r \\ &= (22.5\ \text{kg/s})(15\ \text{m/s}) \left(\frac{1\ \text{N}}{1\ \text{kg} \cdot \text{m/s}^2} \right) \\ &= 337.5\ \text{N} \end{aligned}$$

Now, as the break fails the force of the water jet will accelerate the cart. Thus, the acceleration is

$$\begin{aligned} a &= \frac{F}{m_{cart}} \\ &= \frac{337.5\ \text{N}}{400\ \text{kg}} \\ &= 0.8437\ \text{m/s}^2 \end{aligned}$$

Therefore, the acceleration of the cart is $\boxed{a = 0.844\ \text{m/s}^2}$.

Question: 13.027P pg 1

A 100-ft3/s water jet is moving in the positive *x*-direction at 18 ft/s. The stream hits a stationary splitter such that half of the flow is diverted upward at 45° and the other half is directed downward, and both streams have a final speed of 18 ft/s. Disregarding gravitational effects, determine the *x*- and *z*-components of the force required to hold the splitter in place against the water force.

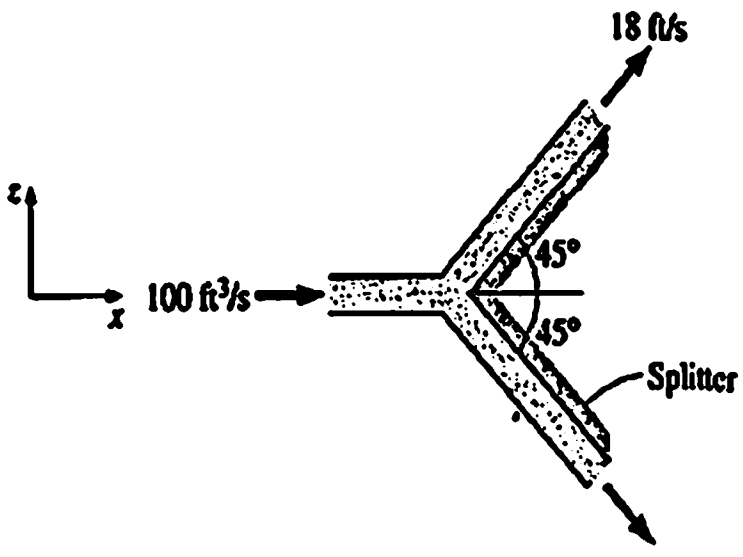
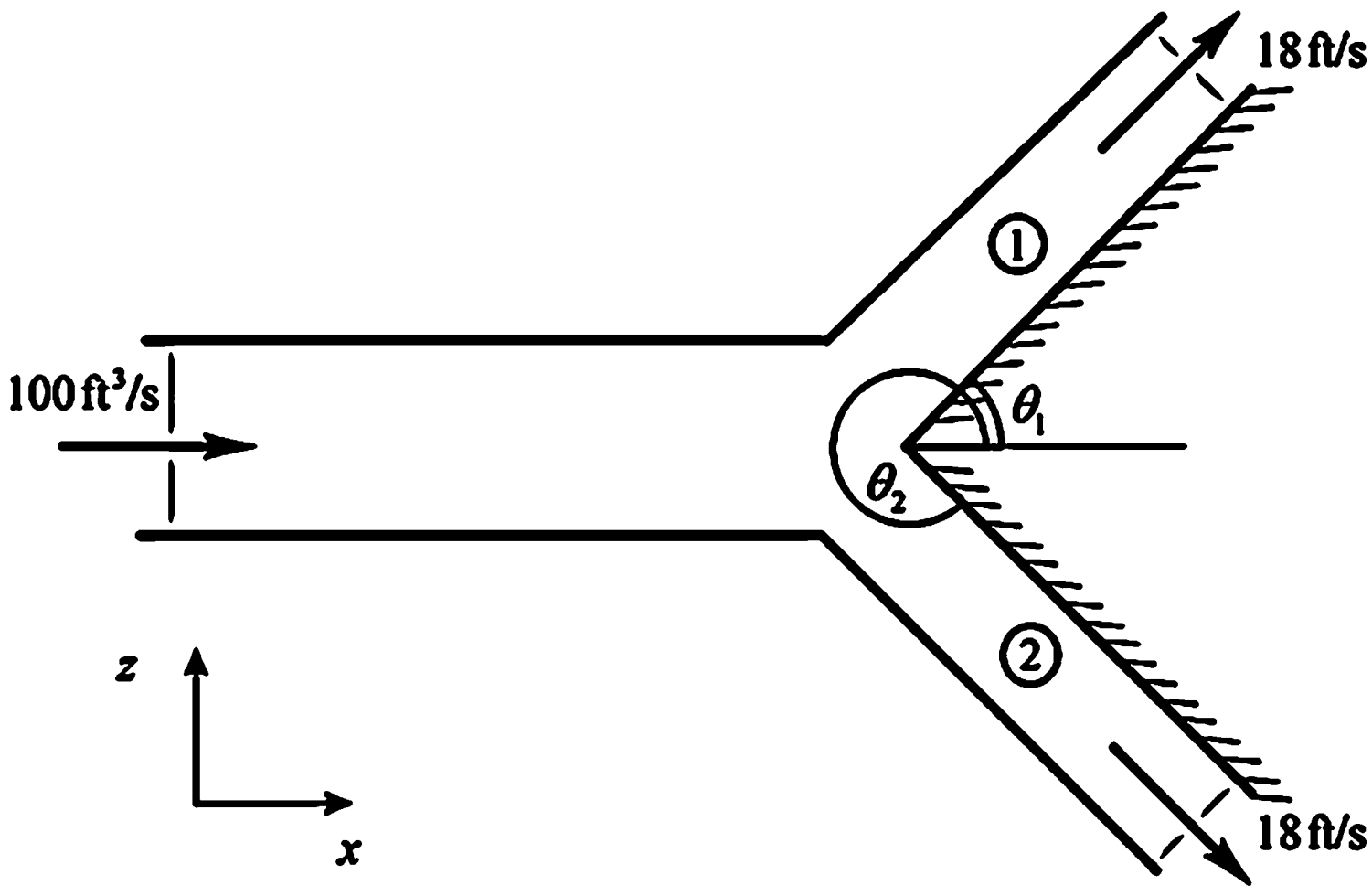


FIGURE P13–27E

Step 1 of 7

Draw the Figure that showing flow of water through the channel,



Step 2 of 7

Find the mass flow rate at the entrance of the channel,

$\dot{m} = \rho \dot{V}$

Here, density of water is denoted as ρ and, velocity of flow as $\dot{v}$.

Substitute $62.37\,\text{lb/ft}^3$ for ρ and, $100\,\text{ft}^3/\text{s}$ for $\dot{v}$ in the equation,

$$\begin{aligned}\dot{m} &= \rho \dot{V} \\ &= 62.37 \times 100 \\ &= 62.37 \times 10^2\ \text{lb/s}\end{aligned}$$

Step 3 of 7

Since the bifurcated channels are identical the mass flow rates are equal in both the sections.

$\dot{m}_1 = \dot{m}_2$

Here, mass flow rate of the section 1 is $\dot{m}_1$ and, mass flow rate of the section 2 is $\dot{m}_2$.

The flow is considered to be steady state, frictionless and the fluid to be incompressible. Applying the law of conservation of mass to the flow,

$$\begin{aligned}\dot{m} &= \dot{m}_1 + \dot{m}_2 \\ &= 2\dot{m}_1 \qquad (\text{since, } \dot{m}_1 = \dot{m}_2) \\ \dot{m}_1 &= \frac{1}{2}\dot{m}\end{aligned}$$

Step 4 of 7

Apply the momentum equation for steady one-dimensional flow,

$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}$.

Consider the momentum equations along the $x-$ axis,

$F_{Rx} = \dot{m}_1 V_1 \cos\theta_1 + \dot{m}_2 V_2 \cos\theta_2 - \dot{m} V$

Here, mass flow rate before hitting the splitter is $\dot{m}$, velocity of flow before hitting the splitter is V , mass flow rate channel 1 is $\dot{m}_1$, velocity of flow channel 1 is V_1 , angle my the split channel 1 with horizontal is θ_1 , mass flow rate channel 2 is $\dot{m}_2$, velocity of flow channel 2 is V_2 and, angle my the split channel 2 with horizontal is θ_2 .

Step 5 of 7

On simplification of above equation,

$$\begin{aligned}F_{Rx} &= \dot{m}_1 V_1 \cos\theta_1 + \dot{m}_2 V_2 \cos\theta_2 - \dot{m} V \\ &= \frac{\dot{m}}{2} V_1 \cos\theta_1 + \frac{\dot{m}}{2} V_2 \cos\theta_2 - \dot{m} V \qquad \left(\text{since, } \dot{m}_1 = \dot{m}_2 = \frac{\dot{m}}{2}\right) \\ &= \left(\frac{1}{2}\dot{m}\right) V \left(\cos\theta_1 + \cos\theta_2\right) - \dot{m} V \qquad (\text{also, } V_1 = V_2 = V)\end{aligned}$$

$$F_{Rx} = \dot{m} V \left[\frac{\left(\cos\theta_1 + \cos\theta_2\right)}{2} - 1\right] \quad \ldots\ldots (1)$$

Step 6 of 7

$F_{Rx} = \dot{m}V\left[\frac{(\cos\theta_1+\cos\theta_2)}{2}-1\right]$ 13(1027P pg 2

Step 6 of 7

Substitute 62.37×10^2 lb/s for $\dot{m}$, 45° for θ_1, 315° for θ_2 and, 18 ft/s^2 and V in the equation (1),

$$F_{Rx} = \dot{m}V_1\left[\frac{(\cos\theta_1+\cos\theta_2)}{2}-1\right]$$
$$= 62.37\times10^2\times18\times\left[\frac{(\cos45°+\cos315°)}{2}-1\right]$$
$$= 62.37\times10^2\times18\times[0.707-1]$$
$$= -32{,}893.938 \text{ lb}$$

Therefore, the force required in *x*-direction to keep the splitter in position is $\boxed{32{,}893.938 \text{ lb}}$ (the opposite direction of the water jet)

Apply the momentum equation for steady one-dimensional flow,

$$\sum\vec{F} = \sum_{out}\beta\dot{m}\vec{V} - \sum_{in}\beta\dot{m}\vec{V}.$$

Consider the momentum equations along the z– axis,

$$F_{Rz} = (\dot{m}_1V_1\sin\theta_1 + \dot{m}_2V_2\sin\theta_2) - 0$$
$$= \frac{\dot{m}}{2}V_1\sin\theta_1 + \frac{\dot{m}}{2}V_2\sin\theta_2 \qquad \left(\text{since}, \dot{m}_1 = \dot{m}_2 = \frac{\dot{m}}{2}\right)$$
$$= \frac{\dot{m}}{2}V_1\sin\theta_1 + \frac{\dot{m}}{2}V_1\sin\theta_2 \qquad (\text{also}, V_1 = V_2)$$

$$= \frac{\dot{m}}{2}V_1(\sin\theta_1+\sin\theta_2) \quad \text{..... (2)}$$

Here, mass flow rate channel 1 is $\dot{m}_1$, velocity of flow channel 1 is V_1, angle my the split channel 1 with horizontal is θ_1, mass flow rate channel 2 is $\dot{m}_2$, velocity of flow channel 2 is V_2 and, angle my the split channel 2 with horizontal is θ_2.

Step 7 of 7

Substitute 62.37×10^2 lb/s for $\dot{m}$, 45° for θ_1, 315° for θ_2 and, 18 ft/s for V_1 in the equation(2).

$$F_{Rz} = \frac{\dot{m}}{2}V_1(\sin\theta_1+\sin\theta_2)$$
$$= \frac{62.37\times10^2}{2}\times18\times(\sin45°+\sin315°)$$
$$= \frac{62.37\times10^2}{2}\times18\times(0.7071-0.7071)$$
$$= 0$$

There is no force required in *z*-axis direction. Since, the bifurcated forces are equal and opposite in direction to each other.

Question: 13.028P

Reconsider Prob. 13-27E. Using EES (or other) software, investigate the effect of the splitter angle on the force exerted on the splitter in the incoming flow direction. Let the half splitter angle vary from 0° to 180° in increments of 10°. Tabulate and plot your results, and draw some conclusions.

Step 1 of 7

Given data:

The velocity of water, $V_1 = 18\ \text{ft/s}^2$

The velocity of water in the splitter, $V_2 = 18\ \text{ft/s}^2$

The angle varies form, $\theta = 0°\text{to}180°$

The discharge, $\dot{V} = 100\text{ft}^3/s$

The density of water, $\rho = 62.37\text{lb/ft}^3$.

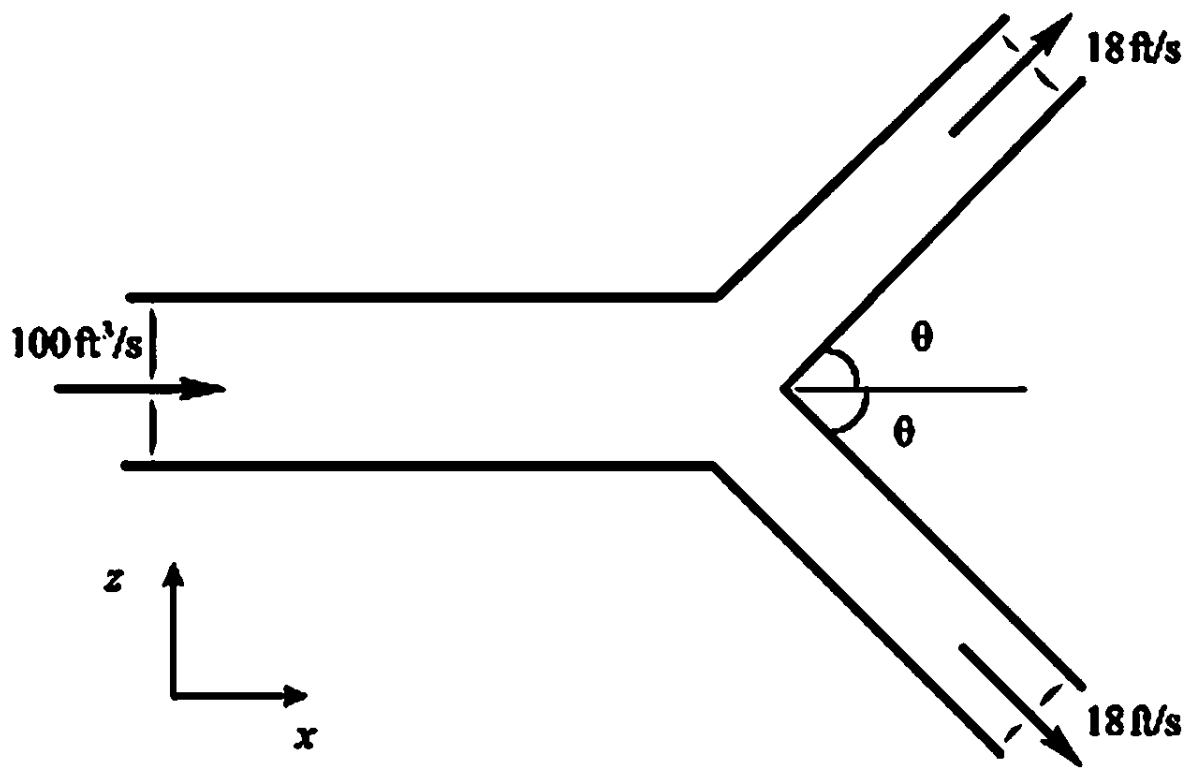
Step 2 of 7

Assumptions are

1. The flow is steady, frictionless and incompressible.
2. Water is discharged to the atmosphere, and thus the pressure is not considered.
3. The flow is jet flow so the momentum-flux correction factor is unity.

Step 3 of 7

Given figure is,



Step 4 of 7

Splitter is taken as the control volume. We have,

The mass flow rate is given by, $\dot{m}_2 = \frac{1}{2}\dot{m}$

Now, the mass flow rate of water,

$$\dot{m} = \rho\dot{V}$$
$$= 62.37 \times 1000$$
$$= 62.37 \times 10^3\ \text{lb/s}$$

Step 5 of 7

Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta\dot{m}\vec{V} - \sum_{in} \beta\dot{m}\vec{V}.$$

Thus, the momentum equations along the x - axis is

$$F_{Rx} = 2\dot{m}_2 V_2 \cos\theta - \dot{m}V_1$$
$$= 2\left(\frac{1}{2}\dot{m}\right)V_2 \cos\theta - \dot{m}V_1$$
$$= \dot{m}(V_2 \cos\theta - V_1)$$

Step 6 of 7

Now, substituting the known values, we get,

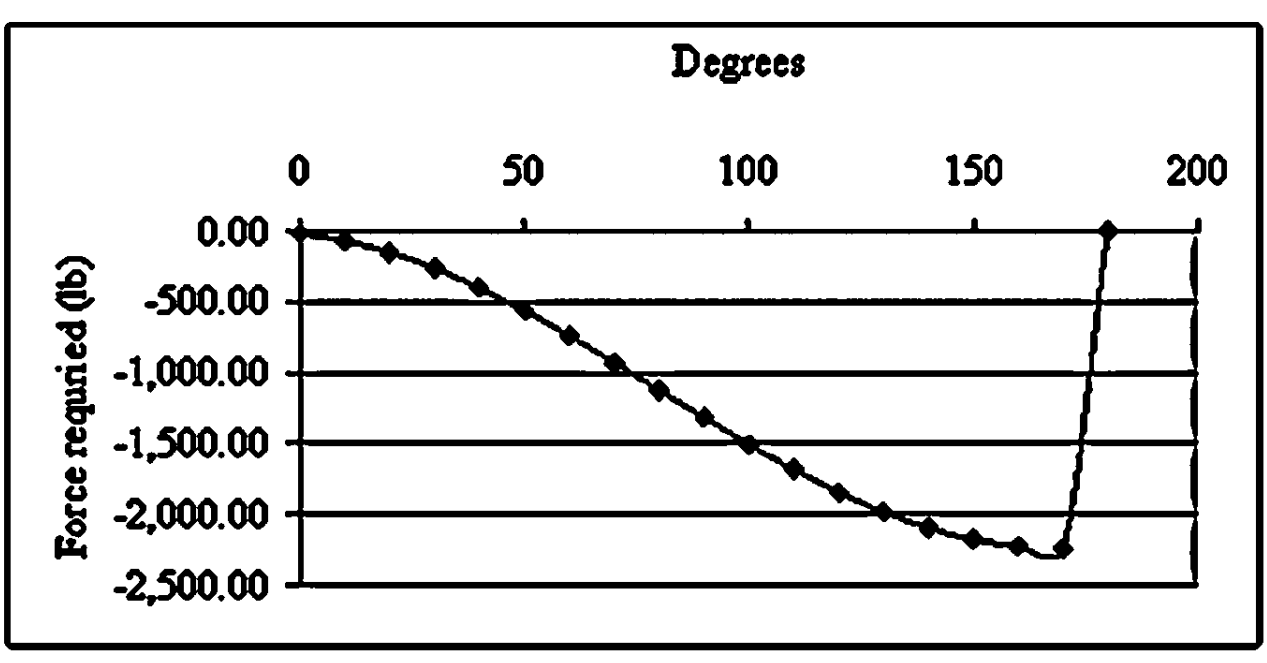
$$F_{Rx} = 62.37 \times 10^3 (18\cos\theta° - 18)$$

For various values of splitter angle the force required is tabulated below;

Splitter angle (degrees)	Force required (lb)
0	-17,069.43
10	-67,758.64
20	-150,526.24
30	-262,855.36
40	-401,330.18
50	-561,739.85
60	-739,206.48
70	-928,333.51
80	-1,123,369.80
90	-1,318,384.50
100	-1,507,447.43
110	-1,684,809.40
120	-1,845,077.03
130	-1,983,376.76
140	-2,095,503.04
150	-2,178,046.23
160	-2,228,496.30
170	-2,245,319.10
180	0.00

Step 7 of 7

The Splitter angle versus Force required for holding the splitter is plotted on the graph show below:



The graph shows us the force required to apply on the splitter increases with the increase in angle.

Question: 13.029P

A horizontal 5-cm-diameter water jet with a velocity of 18 m/s impinges normally upon a vertical plate of mass 1000 kg. The plate rides on a nearly frictionless track and is initially stationary. When the jet strikes the plate, the plate begins to move in the direction of the jet. The water always splatters in the plane of the retreating plate. Determine (*a*) the acceleration of the plate when the jet first strikes it (time = 0), (*b*) the time it takes for the plate to reach a velocity of 9 m/s, and (*c*) the plate velocity 20 s after the jet first strikes the plate. For simplicity, assume the velocity of the jet is increased as the cart moves such that the impulse force exerted by the water jet on the plate remains constant.

Step 1 of 7

Given data: The diameter of the jet is $d = 5$ cm

The velocity of water is $V = 18$ m/s

The mass of the plate is $m_{plate} = 1000$ kg

The density of water is $1000\ \text{kg/m}^3$

Step 2 of 7

Assumptions are

- The flow is steady, frictionless and incompressible.
- Water is discharged to the atmosphere, and thus the pressure is not considered.
- The flow is jet flow so the momentum-flux correction factor is unity.
- The motion of water jet and the cart are horizontal.
- The relative velocity remains constant.

Step 3 of 7

a)

Take the vertical plate on the frictionless

Track as the control volume.

Now, the mass flow rate of water,

$$\dot{m} = \rho A V$$

$$= (1000\ \text{kg/m}^3)(18\ \text{m/s})\frac{\pi(0.05\ \text{m})^2}{4}$$

$$= 35.34\ \text{kg/s}$$

Step 4 of 7

Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Thus, the momentum equations along the x - axis is

$$F_{flx} = -\dot{m}_i V_i$$

$$= -(35.34\ \text{kg/s})(18\ \text{m/s})$$

$$= -636.12\ \text{kg} \cdot \text{m/s}^2$$

$$= -636.12\ \text{N}$$

Step 5 of 7

Thus, the force on the plate is $F = -F_{flx} = 636.12$ N. Now, the initial acceleration of the plate,

$$a = \frac{F}{m_{plate}}$$

$$= \frac{636.12\ \text{kg} \cdot \text{m/s}^2}{1000\ \text{kg}}$$

$$= 0.636\ \text{m/s}^2$$

Therefore, the acceleration is $\boxed{a = 0.636\ \text{m/s}^2}$.

Step 6 of 7

b)

We know, $a = \frac{\Delta V}{\Delta t}$. Also,

The final velocity is given as 9 m/s. So,

$$\Delta t = \frac{(9\ \text{m/s} - 0)}{0.636\ \text{m/s}^2}$$

$$= 14.15\ \text{s}$$

Therefore, the time taken to reach the velocity of 9 m/s is $\boxed{t = 14.15\ \text{s}}$.

Step 7 of 7

c)

We know, $V_{final} = V_{initial} + at$. Also,

The time is given as 20 s.

Thus,

$$V_{final} = 0 + (0.636\ \text{m/s})(20\ \text{s})$$

$$= 12.7\ \text{m/s}$$

Therefore, the velocity of the plate after 20 s is $\boxed{V_{plate} = 12.7\ \text{m/s}}$.

Question: 13.030P

A 4-in-diameter horizontal water jet having a velocity of 120 ft/s strikes a curved plate, which deflects the water 180° at the same speed. Ignoring the frictional effects, determine the force required to hold the plate against the water stream.

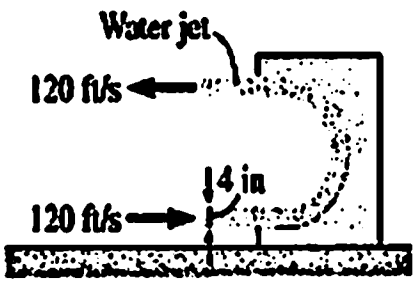
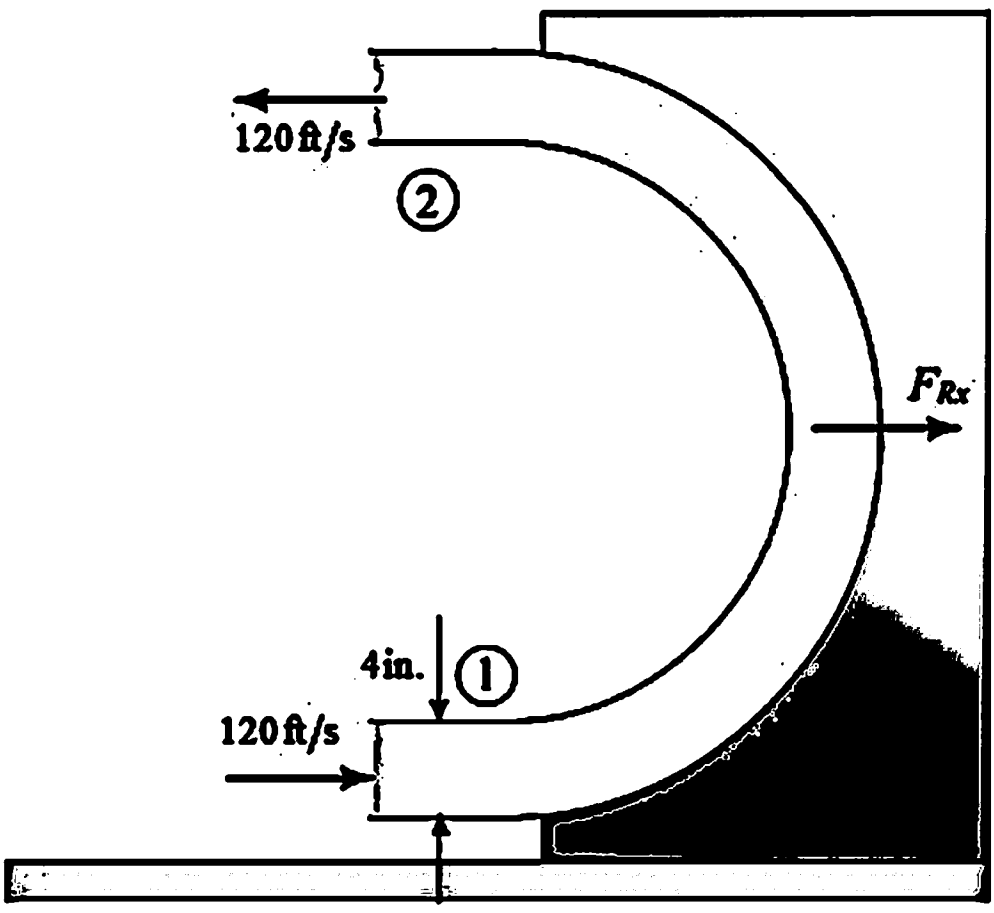


FIGURE P13–30E

Step 1 of 3

Draw the diagram that represents the horizontal water jet strikes on the curved plate.



Here, F_{Rx} is the force exerted by water on the curved plate.

Step 2 of 3

Assume mass flow rate remains constant at the entrance and leaving of the curved plate.

$$\dot{m} = \dot{m}_1 = \dot{m}_2$$

Find the mass flow rate of water by using the following equation:

$$\dot{m} = \rho V A$$

$$= \rho V \frac{\pi}{4} D^2$$

Here, ρ is density of water, V is velocity of water jet, A is the cross sectional area of the jet, and D is diameter of the jet.

Substitute $62.4\,\text{lbm/ft}^3$ for ρ , 120 ft/s for V, and $\frac{4}{12}$ ft for D.

$$\dot{m} = (62.4) \times (120) \times \frac{\pi \left(\frac{4}{12}\right)^2}{4}$$

$$= 653.45\,\text{lb/s}$$

Step 3 of 3

Apply the momentum equation for steady one-dimensional flow.

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m} \vec{V} - \sum_{\text{in}} \beta \dot{m} \vec{V}$$

$$F_{Rx} = \beta \dot{m}(-V_2) - \beta \dot{m}(+V_1)$$

Here, β is the momentum correction factor, V_1 is the velocity of water jet at inlet in positive x direction, and V_2 is the velocity of water jet at outlet in negative x direction,

Substitute 1 for β, $653.45\,\text{lb/s}$ for $\dot{m}$, $120\,\text{ft/s}$ for V_1, V_2

$$F_{Rx} = \beta \dot{m}(-V_2) - \beta \dot{m}(V_1)$$

$$= \left(1 \times 653.45 \frac{\text{lbm}}{\text{s}} \times (-120) \frac{\text{ft}}{\text{s}} - 1 \times 653.45 \frac{\text{lbm}}{\text{s}} \times (120) \frac{\text{ft}}{\text{s}}\right) \times \frac{1\,\text{lbf}}{32.2\,\text{lbm.ft/s}^2}$$

$$= 4870.434\,\text{lbf}$$

Therefore, the force applied in the negative x direction to hold the curved plate is **4870.434 lbf**

Question: 13.031P

A 3-in-diameter horizontal jet of water, with velocity 140 ft/s, strikes a bent plate, which deflects the water by 135° from its original direction. How much force is required to hold the plate against the water stream and what is its direction? Disregard frictional and gravitational effects.

Step 1 of 5

Diameter of the water jet $D = 3\text{in.}$

Velocity of water $V = 140\text{ft/s}$

Water is deflected from its initial position is $\theta = 135°$

Step 2 of 5

The mass flow rate of water can be calculated as,

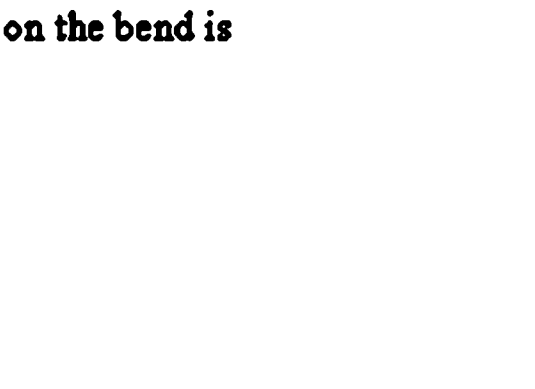
$$\dot{m} = \rho AV$$

$$= \left(62.4\text{lbm/ft}^3\right) \times \frac{\pi\left(\frac{3}{12}\right)^2}{4}\text{ft}^2 \times 140\text{ft/s}$$

$$= 428.827\text{lbm/s}$$

Step 3 of 5

The velocity triangle at the exit is as shown in figure:



Step 4 of 5

The momentum equation for steady flow in x direction is

$$\sum \vec{F}_x = \dot{m}\left(V_{out} - V_{in}\right)_x$$

$$= \dot{m} \times \left(-V\cos 45° - V\right)$$

$$= -\dot{m}V\left(\cos 45° + 1\right)$$

$$= -428.827 \times 140 \times (1.707)$$

$$= -102481.07\text{lbf}$$

The momentum equation for steady flow in y direction is

$$\sum \vec{F}_y = \dot{m}\left(V_{out} - V_{in}\right)_y$$

$$= \dot{m}\left(V\sin 45° - 0\right)$$

$$= 428.827 \times 140 \times \sin 45°$$

$$= 42451.707\text{lbf}$$

Step 5 of 5

Now the resultant force acting on the bend is

$$F = \sqrt{F_x^2 + F_y^2}$$

$$= \sqrt{(-102481.07)^2 + 42451.707^2}$$

$$= 110925.72\text{lbf}$$

Therefore, the force required to hold the plate in place is $\boxed{110925.72\text{lbf}}$

Direction of the force acting on the bend is

$$\alpha = \tan^{-1}\left(\frac{F_y}{F_x}\right)$$

$$= \tan^{-1}\left(\frac{42451.707}{-102481.07}\right)$$

$$= -22.501°$$

$$\boxed{\alpha = -22.501°}$$

Question: 13.032P

Firefighters are holding a nozzle at the end of a hose while trying to extinguish a fire. If the nozzle exit diameter is 6 cm and the water flow rate is 5 m3/min, determine (*a*) the average water exit velocity and (*b*) the horizontal resistance force required of the firefighters to hold the nozzle.

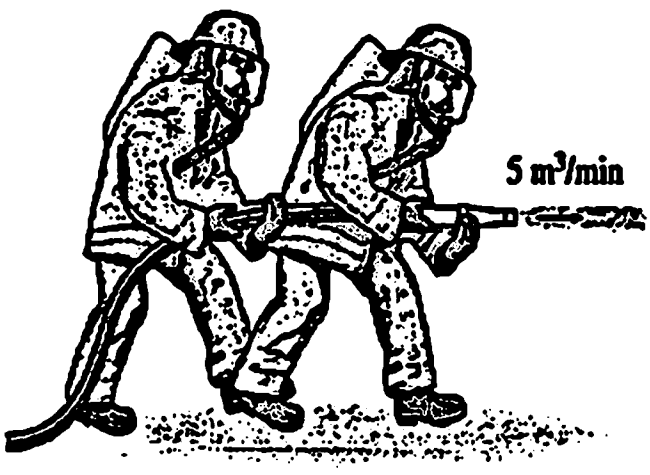


FIGURE P13–32

Step 1 of 4

Given data:-

The exit diameter of the nozzle is $D = 6\,\text{cm}$.

The water flow rate is $\dot{V} = 5\,\text{m}^3/\text{min}$.

Step 2 of 4

Assumptions

1. The flow is steady, frictionless and incompressible.
2. Water is discharged to the atmosphere, and thus the pressure is not considered.
3. The flow is jet flow so the momentum-flux correction factor is unity.
4. The gravitational force is not considered.

Step 3 of 4

a) The density of water is $1000\,\text{kg/m}^3$. Take the nozzle and horizontal portion of the hose as the control volume.

$$V = \frac{\dot{V}}{A}$$

$$= \frac{5\,\text{m}^3/\text{min}}{\frac{\pi(0.06\,\text{m})^2}{4}}\left(\frac{1}{60\,\text{s}}\right)$$

$$= 29.47\,\text{m/s}$$

Therefore, the average exit velocity of water is $\boxed{V = 29.5\,\text{m/s}}$.

Step 4 of 4

b) Now, the mass flow rate,

$$\dot{m} = \rho\dot{V}$$

$$= \left(1000\,\text{kg/m}^3\right)\left(5\,\text{m}^3/\text{min}\right)\left(\frac{1}{60\,\text{s}}\right)$$

$$= 83.33\,\text{kg/s}$$

Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta\dot{m}\vec{V} - \sum_{in} \beta\dot{m}\vec{V}.$$

Now, the momentum equations along the x - axis is

$$F_{in} = \dot{m}V_{out}$$

$$= (83.33\,\text{kg/s})(29.47\,\text{m/s})$$

$$= 2455.83$$

Therefore, the horizontal force required to hold the nozzle is $\boxed{F = 2456\,\text{N}}$.

Question: 13.033P

A 5-cm-diameter horizontal jet of water with a velocity of 40 m/s relative to the ground strikes a flat plate that is moving in the same direction as the jet at a velocity of 10 m/s. The water splatters in all directions in the plane of the plate. How much force does the water stream exert on the plate?

Step 1 of 4

The relative velocity of water $V_{jet} = 40$ m/s

The velocity of the plate $V_{plate} = 10$ m/s

The diameter of the jet $D = 5$ cm

Step 2 of 4

Assumptions

1. The flow is steady, frictionless and incompressible.
2. Water is discharged to the atmosphere, and thus it is not considered.
3. The flow is jet flow so the momentum-flux correction factor is unity.
4. Neglect the friction.
5. The vertical forces are not considered since they have no effect on the horizontal force exerted on the plate.

Step 3 of 4

Consider the plate as the control volume.

Now, the relative velocity between plate and the water jet,

$$\begin{aligned} V_r &= V_{jet} - V_{plate} \\ &= (40 \text{ m/s}) - (10 \text{ m/s}) \\ &= 30 \text{ m/s} \end{aligned}$$

Now, the mass flow rate with respect to the relative velocity is

$$\begin{aligned} \dot{m}_r &= \rho A V_r \\ &= \rho \times \frac{\pi}{4} D^2 \times V_r \\ &= (1000 \text{ kg/m}^3) \times \frac{\pi (0.05 \text{ m})^2}{4} \times (30 \text{ m/s}) \\ &= 58.9 \text{ kg/s} \end{aligned}$$

Step 4 of 4

Let F_{Rx} be in positive x direction. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Thus, $F_{Rx} = \dot{m} V_1$,

$$\begin{aligned} F_{Rx} &= \dot{m}_r V_r \\ &= (58.9 \text{ kg/s})(30 \text{ m/s}) \\ &= 1767 \text{ kg} \cdot \text{m/s} \\ &= 1767 \text{ N} \end{aligned}$$

Therefore, the force exerted by the water on the plate is $\boxed{F = 1767 \text{ N}}$

Question: 13.034P

Reconsider Prob. 13-33. Using EES (or other) software, investigate the effect of the plate velocity on the force exerted on the plate. Let the plate velocity vary from 0 to 30 m/s, in increments of 3 m/s. Tabulate and plot your results.

Step 1 of 5

The relative velocity of water $V_{jet} = 40$ m/s

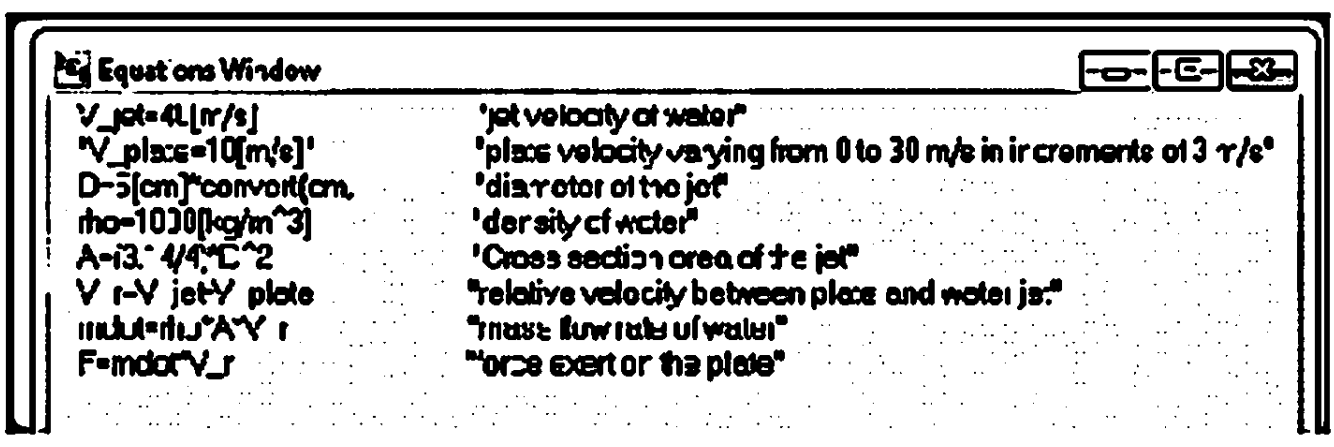
The velocity of the plate $V_{plate} = 10$ m/s

The diameter of the jet $D = 5$ cm

Step 2 of 5

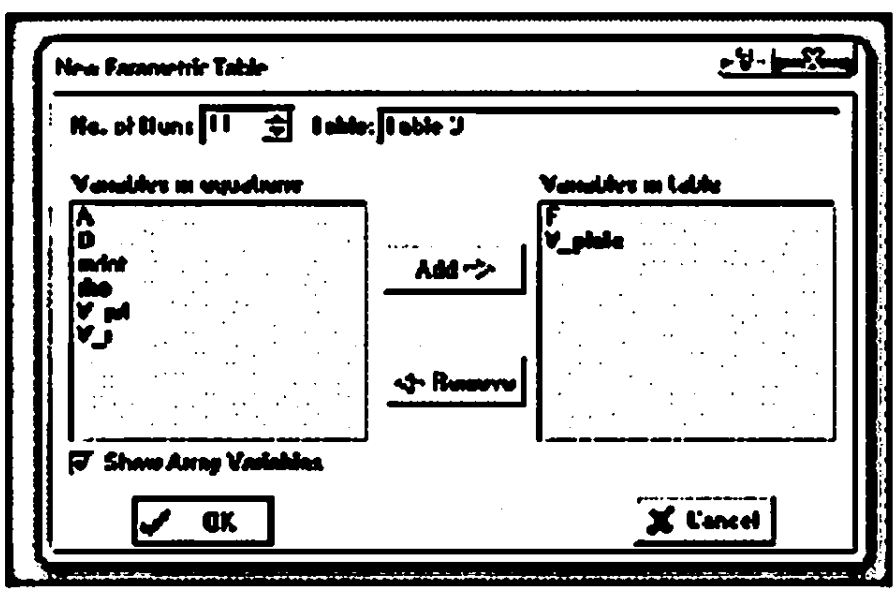
The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

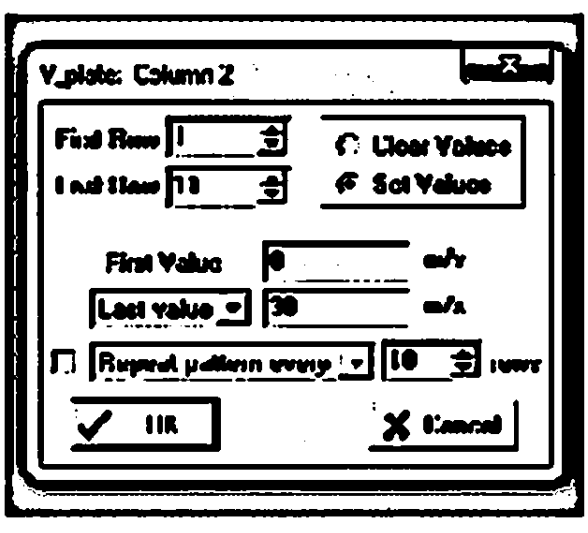


Step 3 of 5

Now to solve the equation for a range of plate velocity,use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "V_plate", "F", press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:

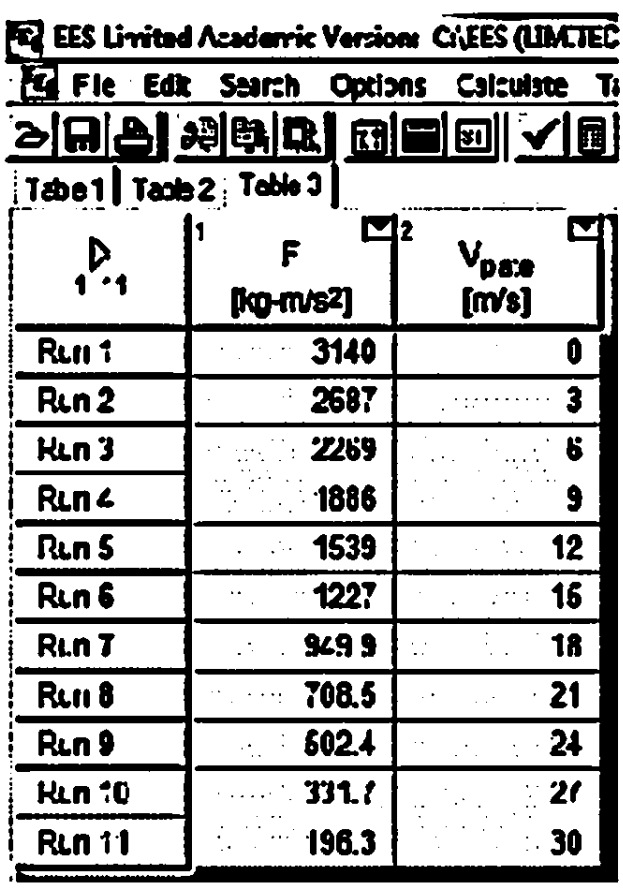


To enter the values in the "V_plate" column, click on the drop tab in "V_plate" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

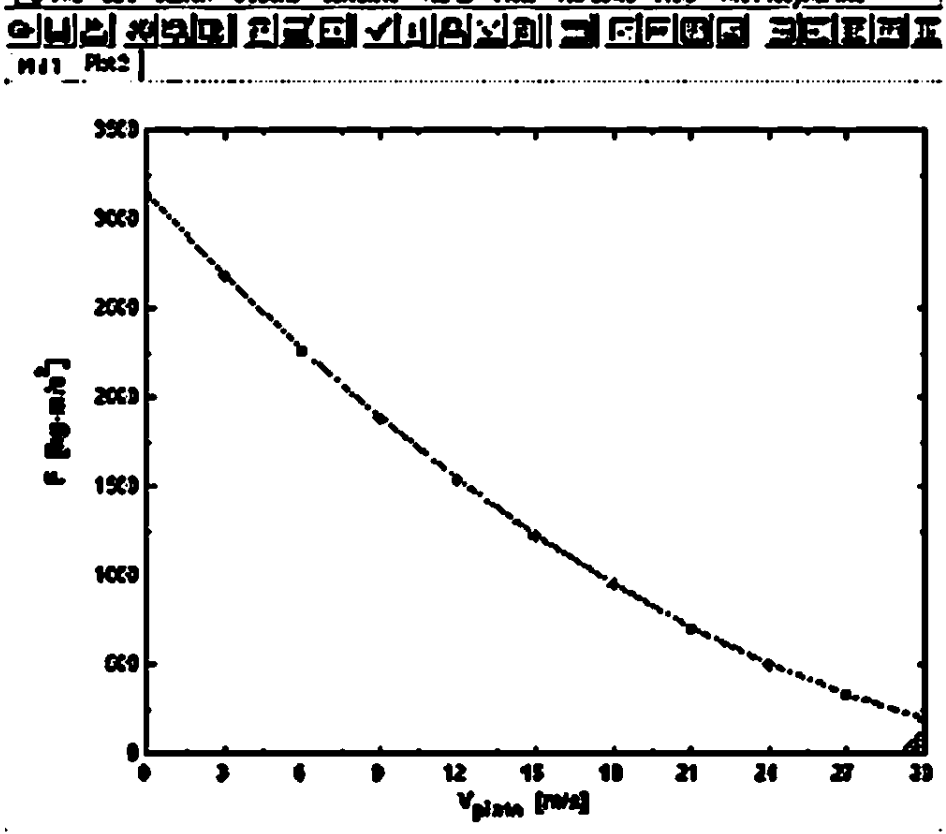
Now click on 'Solve' of 'Calculate' menu.
The Solution will appear as shown below:



	1 F [kg-m/s2]	2 V_{plate} [m/s]
Run 1	3140	0
Run 2	2687	3
Run 3	2269	6
Run 4	1886	9
Run 5	1539	12
Run 6	1227	15
Run 7	949.9	18
Run 8	708.5	21
Run 9	502.4	24
Run 10	331.7	27
Run 11	196.3	30

Inorder to plot the plate velocity and force exerted on plate, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot showing the force exerted on plate and plate velocity is shown:

Step 5 of 5



Question: 13.035P

A fan with 24-in-diameter blades moves 2000 cfm (cubic feet per minute) of air at 70°F at sea level. Determine (*a*) the force required to hold the fan and (*b*) the minimum power input required for the fan. Choose a control volume sufficiently large to contain the fan, with the inlet sufficiently far upstream so that the gage pressure at the inlet is nearly zero. Assume air approaches the fan through a large area with negligible velocity and air exits the fan with a uniform velocity at atmospheric pressure through an imaginary cylinder whose diameter is the fan blade diameter.

Step 1 of 10

The diameter of the fan blade, $D = 24\text{in.}$

The flow rate of air, $\dot{V} = 2000\,\text{cfm}$

The temperature of air, $T = 70°\text{F} = 530°\text{R}$

Step 2 of 10

Assumptions:

The flow is steady and incompressible.

The frictional effects are negligible.

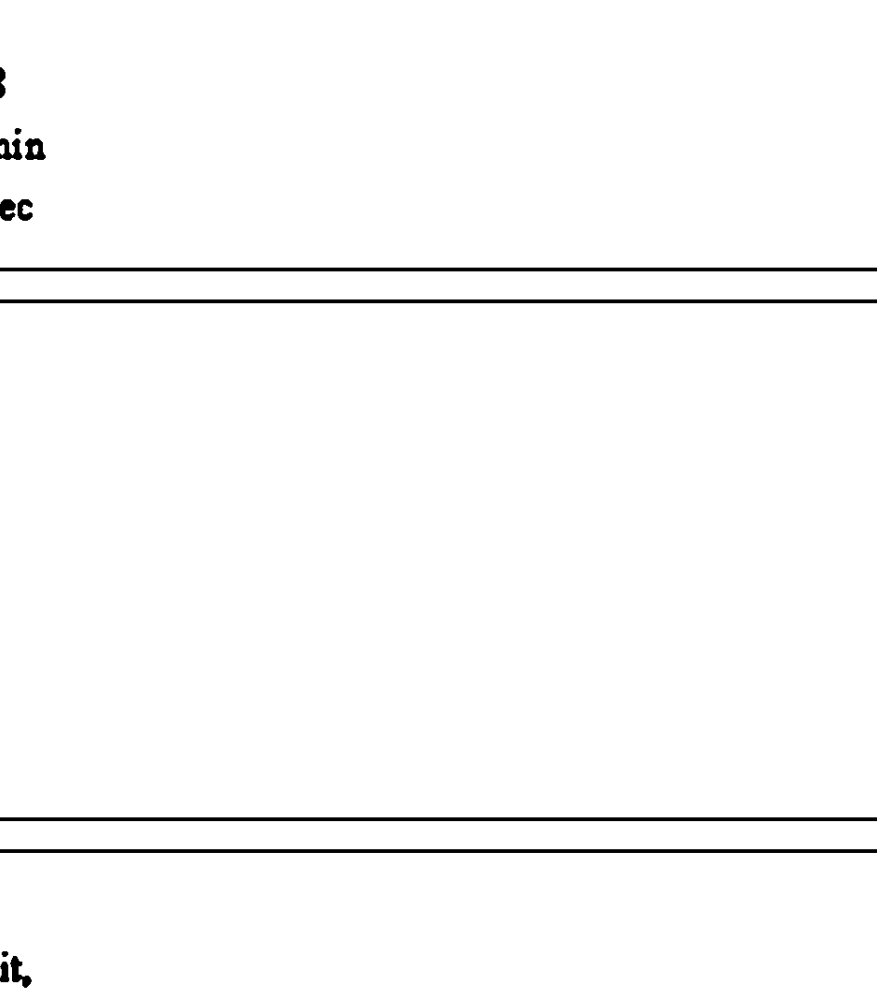
Air approaches the fan through a large area with negligible velocity.

The pressures at inlet and exit are atmospheric.

The wind flow is uniform upstream and downstream, $\beta \cong 1$.

Step 3 of 10

A fan with 24 in. diameter blades is shown in figure,



Step 4 of 10

Density of air can be calculated by using ideal gas equation,

$$\rho = \frac{P}{RT}$$

$$= \frac{(14.7\times144)}{(53.35)(530)}$$

$$= 0.0748\,\text{lb/ft}^3$$

Step 5 of 10

The mass flow rate of air,

$$\dot{m} = \rho\dot{V}$$

$$= 62.4\times0.0748$$

$$= 4.667\,\text{slug/min}$$

$$= 0.0777\,\text{slug/sec}$$

Step 6 of 10

The exit area,

$$A_2 = \frac{\pi D^2}{4}$$

$$= \frac{\pi\left(\frac{24}{12}\right)^2}{4}$$

$$= 3.1415\text{ft}^2$$

Step 7 of 10

The velocity at exit,

$$V_2 = \frac{\dot{V}}{A_2}$$

$$= \frac{2000\times\frac{1}{60}}{3.1415}$$

$$= 10.61\,\text{ft/s}$$

Step 8 of 10

a) We have,

$$\sum F_R = \sum \beta\dot{m}\vec{V}_{out} - \sum \beta\dot{m}\vec{V}_{in}$$

Then

$$F_R = \sum \beta\dot{m}\vec{V}_{out} - \sum \beta\dot{m}\vec{V}_{in}$$

$$= \dot{m}V_2 - \dot{m}V_1$$

$$= 0.0777\times10.61 - 0$$

$$= 0.8243\text{lbf}$$

Therefore, the force required to hold the fan is $\boxed{F_R = 0.8243\text{lbf}}$

Step 9 of 10

b) Apply the conservation of energy principle for the control volume between the inlet and exit

Then the equation can be expressed as

$$\dot{m}\left(\frac{P_1}{\rho}+\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{fan} = \dot{m}\left(\frac{P_2}{\rho}+\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{turbine}+\dot{E}_{mech,loss}$$

We know,

$P_1 = P_2 = P_{atm}$,

$z_1 = z_2$ and

$V_1 = 0$

Step 10 of 10

Power required is

$$\dot{W}_{fan} = \frac{\dot{m}V_2^2}{2}$$

$$= \frac{0.0777\left(10.61^2\right)}{2}$$

$$= 4.373\frac{\text{lbf}\cdot\text{ft}}{\text{sec}}$$

$$= 4.373\frac{\text{lbf}\cdot\text{ft}}{\text{sec}}\times\frac{1000\,\text{W}}{737.56\frac{\text{lbf}\cdot\text{ft}}{\text{sec}}}$$

$$= 5.92\,\text{W}$$

Therefore, the minimum power required for the fan is $\boxed{\dot{W}_{fan} = 5.92\,\text{W}}$

inlet and exit

Then the equation can be expressed as

Question: 13.036P pg 1

An unloaded helicopter of mass 10,000 kg hovers at sea level while it is being loaded. In the unloaded hover mode, the blades rotate at 400 rpm. The horizontal blades above the helicopter cause a 18-m-diameter air mass to move downward at an average velocity proportional to the overhead blade rotational velocity (rpm). A load of 14,000 kg is loaded onto the helicopter, and the helicopter slowly rises. Determine (*a*) the volumetric airflow rate downdraft that the helicopter generates during unloaded hover and the required power input and (*b*) the rpm of the helicopter blades to hover with the 14,000-kg load and the required power input. Take the density of atmospheric air to be 1.18 kg/m3. Assume air approaches the blades from the top through a large area with negligible velocity and air is forced by the blades to move down with a uniform velocity through an imaginary cylinder whose base is the blade span area.

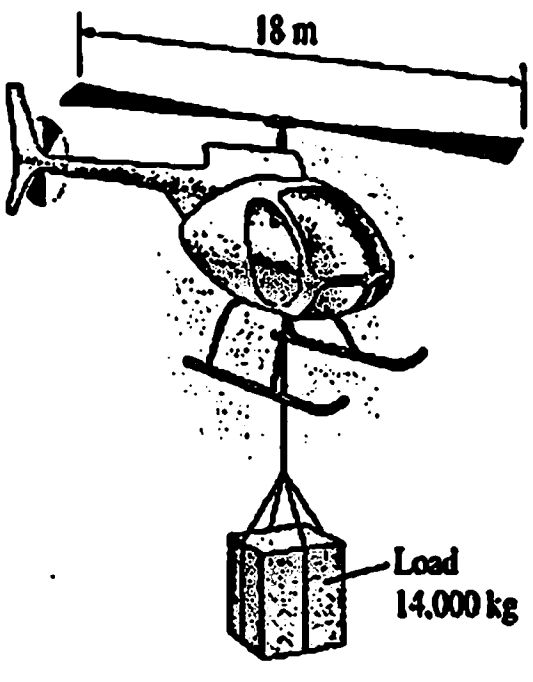


FIGURE P13–36

Step 1 of 17

The diameter of the rotor blade, $D = 18$ m

The mass of the helicopter alone, $m_h = 10000$ kg

The mass of the load alone, $m_l = 14000$ kg

The angular velocity of rotor during unloaded hover mode, $n_{UL} = 400$ rpm.

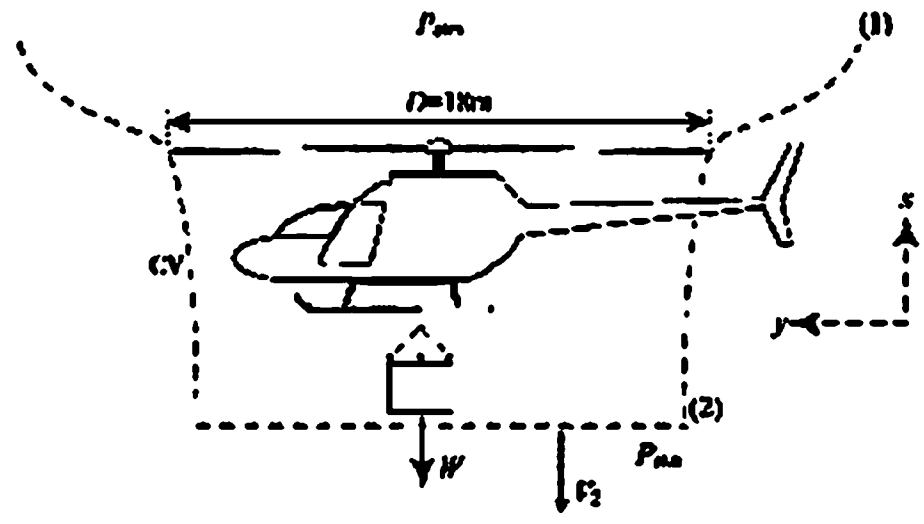
The density of air, $\rho = 1.18$ kg/m^3

Step 2 of 17

Assumptions:
The flow is steady and incompressible.
The frictional effects are negligible.
Air approaches the rotor through a large area with negligible velocity.
The pressures at inlet and exit are atmospheric.
The wind flow is uniform upstream and downstream, $\beta \cong 1$.

Step 3 of 17

The control volume of the helicopter and a load is as shown in figure:



Step 4 of 17

The weight of the helicopter alone,

$$W_h = m_h g$$
$$= (10000)(9.81)$$
$$= 98100 \text{ N}$$

Step 5 of 17

The weight of the load alone,

$$W_l = m_l g$$
$$= (14000)(9.81)$$
$$= 137340 \text{ N}$$

Step 6 of 17

a)
Consider the unloaded hover mode:

The total force acting on the helicopter is equal to the weight of the helicopter.
Then

$$\sum F_{hx} = -W$$
$$= -W_h$$

We have,

$$\sum F_{hx} = \sum \beta \dot{m} \vec{V}_{out} - \sum \beta \dot{m} \vec{V}_{in}$$

Where,

$\dot{m} = \rho A V$,

$V_{in} = 0$ and

$\beta \cong 1$

Then

$$-W_h = -(\rho A V_2) V_2$$
$$W_h = \rho A V_2^2$$

Step 7 of 17

The exit velocity can be expressed as

$$V_2 = \sqrt{\frac{W_h}{\rho A}}$$
$$= \sqrt{\frac{W_h}{\rho\left(\frac{\pi D^2}{4}\right)}}$$
$$= \sqrt{\frac{98100}{}}$$

$$= \sqrt{\frac{W_h}{\rho\left(\frac{\pi D^2}{4}\right)}}$$

$$= \sqrt{\frac{98100}{1.18\left(\frac{\pi \times 18^2}{4}\right)}}$$

$$= 18.0749 \text{ m/s}$$

Step 8 of 17

The mass flow rate of air

$$\dot{m} = \rho A V_2$$

$$= 1.18\left(\frac{\pi \times 18^2}{4}\right)18.0749$$

$$= 5427.412 \text{ kg/s}$$

Step 9 of 17

Volumetric flow rate of air can be calculated as

$$\dot{V} = \frac{\dot{m}}{\rho}$$

$$= \frac{5427.412}{1.18}$$

$$= 4599.50 \text{ m}^3/\text{s}$$

Therefore, the volumetric down flow rate is $\boxed{\dot{V} = 4599.50 \text{ m}^3/\text{s}}$

Step 10 of 17

Apply the conservation of energy principle between the inlet and exit of control volume
We have

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{rotor} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{turbine} + \dot{E}_{mech,loss}$$

Let us assume the elevation difference between the two points is negligible.
We have,

$P_1 = P_2 = P_{atm}$, $z_1 = z_2$ and $V_1 = 0$

Then

$$\dot{W}_{rotor} = \frac{\dot{m}V_2^2}{2}$$

$$= \frac{5427.414\left(18.0749^2\right)}{2}$$

$$= 886.573 \times 10^3 \text{ W}$$

Therefore, the required power input is $\boxed{\dot{W}_{rotor} = 886.573 \times 10^3 \text{ W}}$

Step 11 of 17

b)
Consider the loaded hover mode of the helicopter.
The total weight of the helicopter,

$$W = W_h + W_l$$

$$= 98100 + 137340$$

$$= 235440 \text{ N}$$

The total force acting on the helicopter is equal to the weight of the helicopter and the load.
Then

$$\sum F_{in} = -W$$

We have,

$$\sum F_{in} = \sum \beta \dot{m} \vec{V}_{out} - \sum \beta \dot{m} \vec{V}_{in}$$

Where,

$\dot{m} = \rho A V$,

$V_{in} = 0$ and

$\beta \cong 1$

Then

$$-W = -(\rho A V_2) V_{2L}$$

$$W = \rho A V_{2L}^2$$

Step 12 of 17

The exit velocity during loaded mode can be expressed as

$$V_{2L} = \sqrt{\frac{W}{\rho A}}$$

$$= \sqrt{\frac{W}{\rho\left(\frac{\pi D^2}{4}\right)}}$$

$$= \sqrt{\frac{235440}{1.18\left(\frac{\pi \times 18^2}{4}\right)}}$$

$$= 28.00 \text{ m/s}$$

Step 13 of 17

The mass flow rate,

$$\dot{m}_L = \rho A V_{2L}$$

$$= 1.18\left(\frac{\pi \times 18^2}{4}\right)28$$

$$= 8407.655 \text{ kg/s}$$

Step 14 of 17

Apply the conservation of energy principle between the inlet and exit of control volume
We have

$$\dot{m}_L\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{rotor,L} = \dot{m}_L\left(\frac{P_2}{\rho} + \frac{V_{2L}^2}{2} + gz_2\right) + \dot{W}_{turbine,L} + \dot{E}_{mechloss,L}$$

Let us assume the elevation difference between the two points is negligible.
We have,

We have

$$\dot{m}_L\left(\frac{P_1}{\rho}+\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{rotor,L}=\dot{m}_L\left(\frac{P_2}{\rho}+\frac{V_{2L}^2}{2}+gz_2\right)+\dot{W}_{turbine,L}+\dot{E}_{mech\,loss,L}$$

Let us assume the elevation difference between the two points is negligible.
We have,

$P_1 = P_2 = P_{atm}$,

$z_1 = z_2$ and

$V_1 = 0$

Step 15 of 17

Then

$$\dot{W}_{rotor,L}=\frac{\dot{m}_L V_{2L}^2}{2}$$

$$=\frac{8407.655(28^2)}{2}$$

$$=3295.8\times10^3\,\text{W}$$

Therefore, the required power input is $\boxed{\dot{W}_{rotor,L}=3295.8\times10^3\,\text{W}}$.

Step 16 of 17

We know that the velocity is proportional to the overhead rotor rpm.

Then we can express for unloaded mode as,

$$\dot{m}_L\left(\frac{P_1}{\rho}+\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{rotor,L}=\dot{m}_L\left(\frac{P_2}{\rho}+\frac{V_{2L}^2}{2}+gz_2\right)+\dot{W}_{turbine,L}+\dot{E}_{mech\,loss,L}$$

Let us assume the elevation difference between the two points is negligible.
We have,

$P_1 = P_2 = P_{atm}$,

$z_1 = z_2$ and

$V_1 = 0$

Step 15 of 17

Then

$$\dot{W}_{rotor,L}=\frac{\dot{m}_L V_{2L}^2}{2}$$

$$=\frac{8407.655(28^2)}{2}$$

$$=3295.8\times10^3\,\text{W}$$

Therefore, the required power input is $\boxed{\dot{W}_{rotor,L}=3295.8\times10^3\,\text{W}}$.

Step 16 of 17

We know that the velocity is proportional to the overhead rotor rpm.

Question: 13.037P pg 1

Reconsider the helicopter in Prob. 13-36, except that it is hovering on top of a 2800-m-high mountain where the air density is 0.928 kg/m3. Noting that the unloaded helicopter blades must rotate at 400 rpm to hover at sea level, determine the blade rotational velocity to hover at the higher altitude. Also determine the percent increase in the required power input to hover at 3000-m altitude relative to that at sea level.

Step 1 of 15

The diameter of the rotor blade, $D = 18$ m

The mass of the helicopter, $m = 10000$ kg

The altitude of the helicopter at mountain top, $z_{mnt} = 2800$ m

The density of air at mountain top, $\rho_{mnt} = 0.928\ \text{kg/m}^3$

The final altitude of the helicopter, $z_2 = 3000$ m

The rotational velocity of rotor at sea level, $n_{sea} = 400$ rpm

The exit velocity is proportional to the rotational velocity of blade.

Step 2 of 15

Assumptions:

The flow is steady and incompressible.

The frictional effects are negligible.

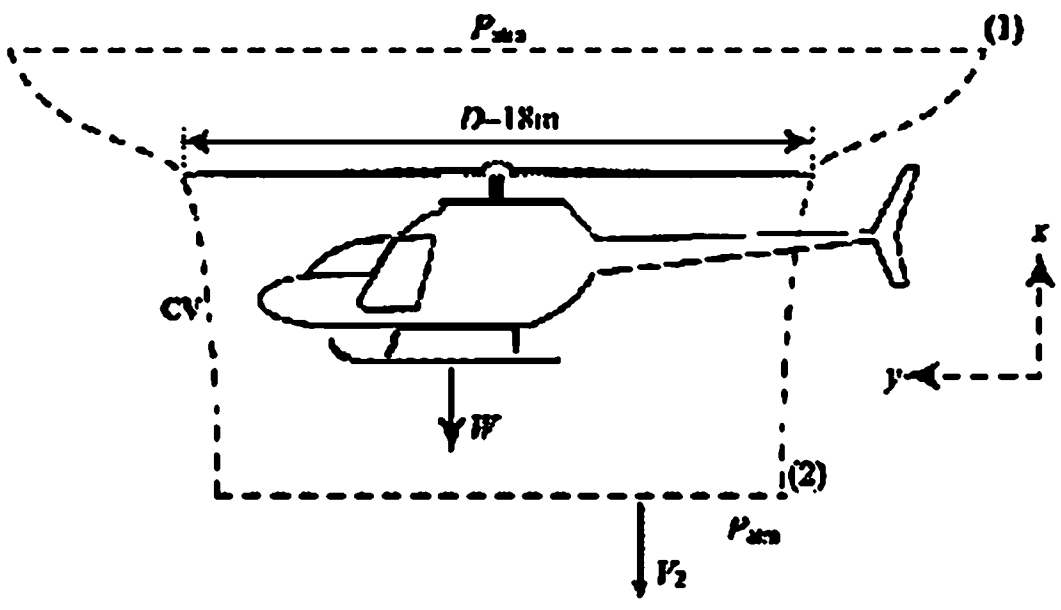
Air approaches the rotor through a large area with negligible velocity.

The pressures at inlet and exit are atmospheric.

The wind flow is uniform upstream and downstream, $\beta \cong 1$.

Step 3 of 15

The control volume of the helicopter and a load is as shown in figure:



Step 4 of 15

The weight of the helicopter,

$W = mg$

$= (10000)(9.81)$

$= 98100$ N

Step 5 of 15

The total force acting on the helicopter is equal to the weight of the helicopter.

Then

$\sum F_{fy} = -W$

We have,

$\sum F_{fx} = \sum \beta \dot{m} \vec{V}_{out} - \sum \beta \dot{m} \vec{V}_{in}$

Where,

$\dot{m} = \rho A V$,

$V_{in} = 0$ and

$\beta \cong 1$

Then

$-W = -(\rho A V_2) V_2$

$W = \rho A V_2^2$

Step 6 of 15

The exit velocity can be expressed as

$$V_2 = \sqrt{\frac{W}{\rho A}}$$

Consider the unloaded hover mode at sea level,

$$V_{2,sea} = \sqrt{\frac{W}{\rho_{sea} A}}$$

Consider the unloaded hover mode at the mountain top.

$$V_{2,mnt} = \sqrt{\frac{W}{\rho_{mnt} A}}$$

Step 7 of 15

Then

$$\frac{V_{2,mnt}}{V_{2,sea}} = \frac{\sqrt{\rho_{sea}}}{\sqrt{\rho_{mnt}}}$$

$$= \sqrt{\frac{1.18}{0.928}}$$

$$= 1.127$$

Step 8 of 15

We know the exit velocity is proportional to blade velocity.

Then

$n_{mnt} \propto V_{2mnt}$

$= aV_{2mnt}$

And

$n_{sea} \propto V_{2sea}$

$= aV_{2sea}$

Step 9 of 15

Then

$$\frac{n_{mnt}}{n_{sea}} = \frac{V_{2mnt}}{V_{2sea}}$$

Step 9 of 15

Then

$$\frac{n_{mtn}}{n_{sea}} = \frac{V_{2mtn}}{V_{2sea}}$$
$$= 1.1276$$

Step 10 of 15

Then

$$n_{mtn} = 1.1276(400)$$
$$= 451 \text{ rpm}$$

The rotor velocity at mountain top, $\boxed{n_{mtn} = 451.04 \text{ rpm}}$

Step 11 of 15

The mass flow rate,

$$\dot{m} = \rho A V_{2,sea}$$
$$= 0.928\left(\frac{\pi \times 18^2}{4}\right)20.381$$
$$= 4812.91 \text{ kg/s}$$

Step 12 of 15

Apply the conservation of energy principle between the inlet and exit of control volume at sea level

We have

$$\dot{m}\left(\frac{P_1}{\rho} + \frac{V_{1sea}^2}{2} + gz_1\right) + \dot{W}_{rotor,sea} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_{2sea}^2}{2} + gz_2\right) + \dot{W}_{turbine,sea} + \dot{E}_{mechloss,sea}$$

Let us assume the elevation difference between the two points is negligible.

We have,

$P_1 = P_2 = P_{atm}$,

$z_1 = z_2$ and

$V_1 = 0$

Then

$$\dot{W}_{rotor,sea} = \frac{\dot{m}V_{2sea}^2}{2}$$
$$= \frac{\rho_{sea} A V_{2sea} V_{2sea}^2}{2}$$
$$= \frac{\rho_{sea} A}{2}\left(\sqrt{\frac{W}{A\rho_{sea}}}\right)^3$$
$$= \frac{\rho_{sea}}{2\sqrt{\rho_{sea}}^3}\left(\sqrt{\frac{W}{A}}\right)^3$$

Step 13 of 15

Similarly for the altitude of 3000 m

$$\dot{W}_{rotor,a} = \frac{\dot{m}V_{2a}^2}{2}$$
$$= \frac{\rho_a A V_{2a} V_{2a}^2}{2}$$
$$= \frac{\rho_a A}{2}\left(\sqrt{\frac{W}{A\rho_a}}\right)^3$$
$$= \frac{\rho_a}{2\sqrt{\rho_a}^3}\left(\sqrt{\frac{W}{A}}\right)^3$$

Step 14 of 15

Then

$$\frac{\dot{W}_{rotor,sea}}{\dot{W}_{rotor,a}} = \frac{\frac{\rho_{sea}}{2\sqrt{\rho_{sea}}^3}\left(\sqrt{\frac{W}{A}}\right)^3}{\frac{\rho_a}{2\sqrt{\rho_a}^3}\left(\sqrt{\frac{W}{A}}\right)^3}$$
$$= \frac{\sqrt{\rho_a}}{\sqrt{\rho_{sea}}}$$
$$= \frac{\sqrt{0.897}}{\sqrt{1.18}}$$
$$= 0.8718$$

That is

$$\dot{W}_{rotor,sea} = 0.8718\dot{W}_{rotor,a}$$

Step 15 of 15

Then

$$\left(\Delta\dot{W}\right) = \dot{W}_{rotor,a} - \dot{W}_{rotor,sea}$$
$$= 1 - 0.8718$$
$$= 0.128$$
$$= 12.8\%$$

Therefore, the power required for the helicopter at an altitude of 3000 m is $\boxed{12.8\%}$ greater than the power required at an altitude at sea level.

Question: 13.038P

A sluice gate, which controls flow rate in a channel by simply raising or lowering a vertical plate, is commonly used in irrigation systems. A force is exerted on the gate due to the difference between the water heights *y*1, and *y*2 and the flow velocities *V*1 and *V*2 upstream and downstream from the gate, respectively. Take the width of the sluice gate (into the page) to be *w*. Wall shear stresses along the channel walls may be ignored, and for simplicity, we assume steady, uniform flow at locations 1 and 2. Develop a relationship for the force *FR* acting on the sluice gate as a function of depths *y*1, and *y*2, mass flow rate *m*, gravitational constant *g*, gate width *w*, and water density *ρ*.

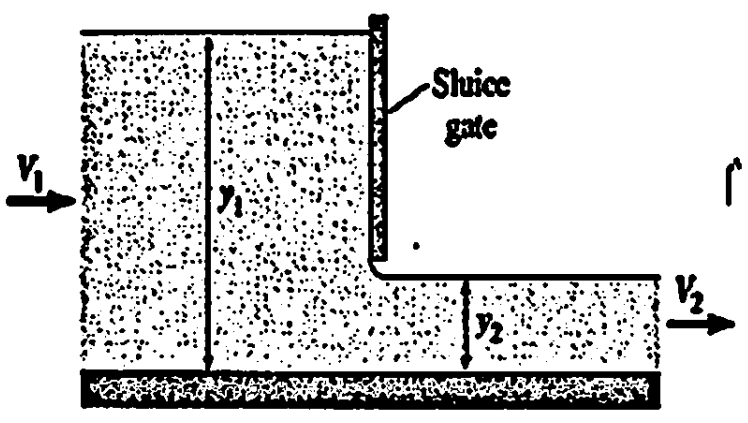


FIGURE P13–38

Step 1 of 4

Apply the Bernoulli's equation between the inlet and exit

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+y_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+y_2$$

Here, P_1, P_2 are pressures at inlet and exit, ρ is the density of water, V_1, V_2 are the velocities at the inlet and exit, and y_1, y_2 are the water heights.

Substitute P_{atm} for P_1, P_2 .

$$\frac{P_{atm}}{\rho g}+\frac{V_1^2}{2g}+y_1=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g}+y_2 \quad \text{..... (1)}$$
$$V_2^2-V_1^2=2g(y_1-y_2)$$

Mass flow rate is constant though out the flow.

$$\dot{m}=\rho A_1 V_1=\rho A_2 V_2$$

Where, A_1 and A_2 is the inlet and exit flow areas.

$$A_1V_1=A_2V_2=\dot{V}$$

Here, $\dot{V}$ is the volumetric flow rate.

Step 2 of 4

Substitute $\frac{\dot{V}}{A_1}$ for V_1 , wy_1 for A_1 , wy_2 for A_2 , and $\frac{\dot{V}}{A_2}$ for V_2 in equation (1).

$$V_2^2-V_1^2=2g(y_1-y_2)$$
$$\left(\frac{\dot{V}}{A_2}\right)^2-\left(\frac{\dot{V}}{A_1}\right)^2=2g(y_1-y_2)$$
$$\left(\frac{\dot{V}}{wy_2}\right)^2-\left(\frac{\dot{V}}{wy_1}\right)^2=2g(y_1-y_2)$$

$$\dot{V}=\sqrt{w^2\frac{y_1y_2}{y_1^2-y_2^2}(y_1-y_2)}$$
$$=w\sqrt{2g\frac{(y_2-y_1)y_1y_2}{(y_2-y_1)(y_2+y_1)}}$$
$$=w\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$$

Step 3 of 4

Find the initial velocity.

$$V_1=\frac{\dot{V}}{A_1}$$
$$=\frac{w}{A_1}\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$$

Substitute $w\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$ for $\dot{V}$ and wy_1 for A_1 .

$$V_1=\frac{w}{A_1}\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$$

Find the final velocity.

$$V_2=\frac{\dot{V}}{A_2}$$
$$=\frac{w}{A_2}\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$$

Substitute $w\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$ for $\dot{V}$ and wy_2 for A_2 .

$$V_2=\frac{w}{A_2}\sqrt{2g\frac{y_1y_2}{(y_2+y_1)}}$$

Step 4 of 4

Write the force balance for the system.

$$\sum F_{fix}=\sum\beta\dot{m}\vec{V}_{out}-\sum\beta\dot{m}\vec{V}_{in}$$
$$\sum F_{fix}=P_1A_1-P_2A_2-F_R$$
$$\dot{m}(V_2-V_1)=\left(\rho g\frac{y_1}{2}A_1-\rho g\frac{y_2}{2}A_2\right)-F_R$$
$$\left(\rho g\frac{y_1}{2}A_1-\rho g\frac{y_2}{2}A_2\right)-F_R=\dot{m}\frac{\dot{V}}{A_2}-\dot{m}\frac{\dot{V}}{A_1}$$
$$\left(\rho g\frac{y_1}{2}A_1-\rho g\frac{y_2}{2}A_2\right)-F_R=\frac{\rho}{A_2}-\frac{\rho}{A_1}$$

$$F_R=\left(\rho g\frac{y_2}{2}A_2-\rho g\frac{y_1}{2}A_1\right)-\left(\frac{\rho}{A_2}-\frac{\rho}{A_1}\right)$$
$$=\left(\rho g\frac{y_2}{2}wy_2-\rho g\frac{y_1}{2}wy_1\right)-\left(\frac{\rho}{A_2}-\frac{\rho}{A_1}\right)$$
$$=\rho gw\left(\frac{y_2^2}{2}-\frac{y_1^2}{2}\right)-\rho\left(\frac{1}{A_2}-\frac{1}{A_1}\right)$$

Therefore, the reaction force can be expressed as $\boxed{\rho gw\left(\frac{y_2^2}{2}-\frac{y_1^2}{2}\right)-\rho\left(\frac{1}{A_2}-\frac{1}{A_1}\right)}$.

Question: 13.039P

Water enters a centrifugal pump axially at atmospheric pressure at a rate of 0.12 m3/s and at a velocity of 7 m/s, and leaves in the normal direction along the pump casing, as shown in Fig. PI3-39. Determine the force acting on the shaft (which is also the force acting on the bearing of the shaft) in the axial direction.

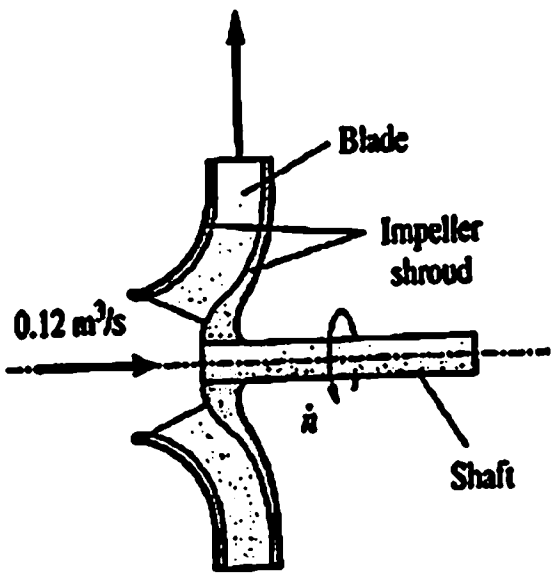


FIGURE P13–39

Step 1 of 5

Given data:-

The velocity of water is $V = 7\ \text{m/s}$.

The water flow rate is $\dot{V} = 0.12\ \text{m}^3/\text{s}$.

Step 2 of 5

Assumptions

1. The flow is steady, frictionless and incompressible.
2. Atmosphere pressure is not considered.
3. The momentum-flux correction is neglected.

Step 3 of 5

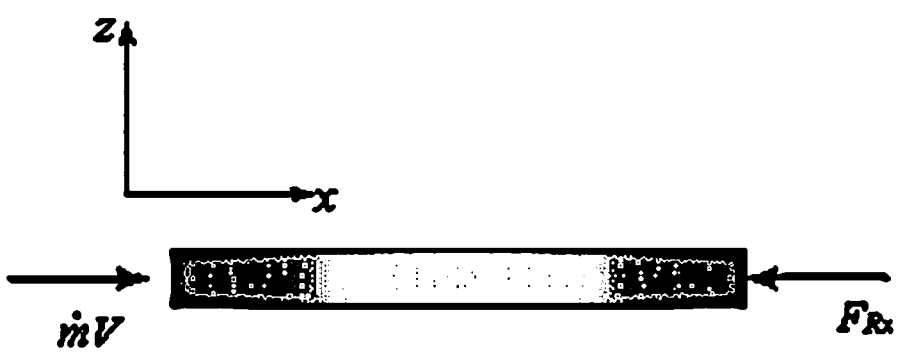
Take the centrifugal pump as the control volume.

Now, the mass flow rate is

$$\begin{aligned}\dot{m} &= \rho\dot{V} \\ &= (1000\ \text{kg/m}^3)(0.12\ \text{m}^3/\text{s}) \\ &= 120\ \text{kg/s}\end{aligned}$$

Step 4 of 5

Draw the free-body diagram of the shaft.



Step 5 of 5

Let F_{Rx} be in positive x direction. Now, the momentum equation for steady one-dimensional flow,

$$\sum \vec{F} = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}.$$

Thus, $-F_{Rx} = -\dot{m}V_i$,

$$\begin{aligned}F_{Rx} &= \dot{m}V_i \\ &= (120\ \text{kg/s})(7\ \text{m/s}) \\ &= 840\ \text{kg}\cdot\text{m/s} \\ &= 840\ \text{N}\end{aligned}$$

Therefore, the force acting on the shaft is $\boxed{F = 840\ \text{N}}$.

Question: 13.040P

An incompressible fluid of density ρ and viscosity μ, flows through a curved duct that turns the flow 180°. The duct cross-sectional area remains constant. The average velocity, momentum flux correction factor, and gage pressure are known at the inlet (1) and outlet (2), as in Fig. P13–40. (a) Write an expression for the horizontal force Fx of the fluid on the walls of the duct in terms of the given variables. (b) Verify your expression by plugging in the following values: ρ = 998.2 kg/m3, μ, = 1.003 × 10-3 kg/m-s, $A1$, = $A2$ = 0.025 m2, $\beta 1$ = 1.01, $\beta 2$ = 1.03, $V1$ = 10 m/s, $P1$,gage= 78.47 kPa, and $P2$,gage = 65.23 kPa.

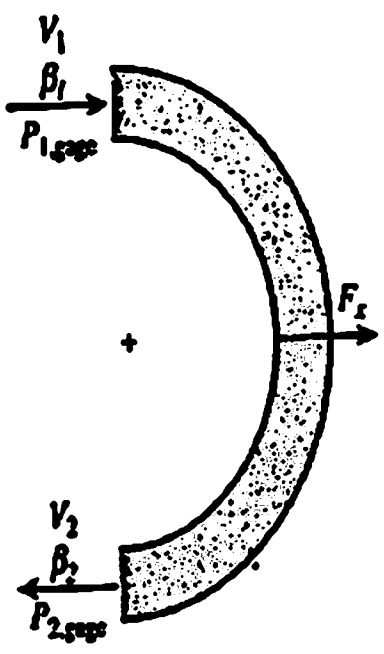


FIGURE P13–40

Step 1 of 5

Density of the fluid ρ

Fluid viscosity μ

The average velocity at inlet is V_1

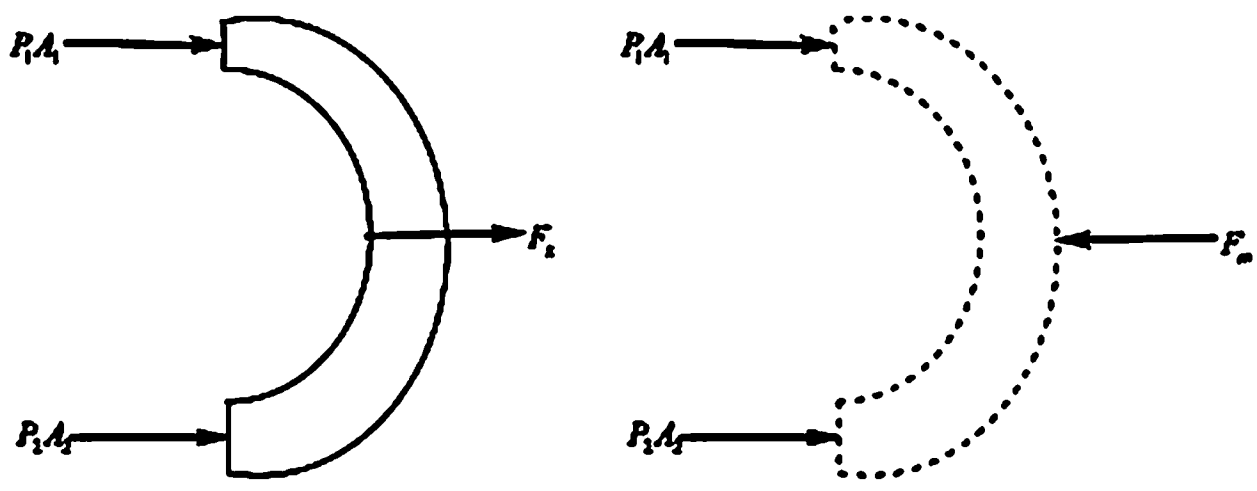
Average velocity at exit is V_2

Momentum flux correction factor at inlet and exit is β_1, β_2

Gage pressure at the inlet and exit is P_1, P_2

Step 2 of 5

The fluid flows through a curved duct that turns the flow 180^0 is shown below:
The curved surface indicates control volume of water



Where

F_x Represents force acting by the fluid on the walls of the duct

F_m Represents reaction force by the walls of the duct on the water

Step 3 of 5

The pressure force acting at inlet is P_1A_1

The pressure force acting at exit is P_2A_2

By applying momentum equation on this control volume the net thrust force acting on the body can be determined as,

$$\sum F = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}$$

$$-F_m + P_1A_1 + P_2A_2 = \beta \dot{m}(-V_{out}) - \beta \dot{m} V_{in}$$

$$-F_m = \dot{m}(-\beta_2 V_{out} - \beta_1 V_{in}) - P_1A_1 - P_2A_2$$

Therefore, the fluid force F_x acting on the walls of the curved duct is

$$F_x = -F_m$$

$$\boxed{F_x = \dot{m}(-\beta_2 V_{out} - \beta_1 V_{in}) - P_1A_1 - P_2A_2}$$

Step 4 of 5

(b)

Density of the fluid $\rho = 998.2 \text{kg/m}^3$

Dynamic viscosity of the fluid $\mu = 1.003 \times 10^{-3} \text{kg/m} \cdot \text{s}$

Cross sectional area of the duct at inlet and exit is $A_1 = A_2 = 0.025 \text{m}^2$

Momentum flux correction factor at inlet $\beta_1 = 1.01$

Momentum flux correction factor at exit $\beta_2 = 1.03$

Velocity at inlet $V_1 = 10 \text{m/s}$

Gage pressure at inlet is $P_{1,gage} = 78.47 \text{kPa}$

Gage pressure at exit is $P_{2,gage} = 65.23 \text{kPa}$

Step 5 of 5

Since duct area remains constant, there is no change in velocity at inlet and exit

$V_1 = V_2 = 10 \text{m/s}$

Mass flow rate of incompressible fluid is

$$\dot{m} = \rho A_1 V_1$$
$$= 998.2 \times 0.025 \times 10$$
$$= 249.55 \text{kg/sec}$$

Substituting the known values in force F_m

$$F_m = \dot{m}(\beta_2 V_{out} - \beta_1 V_{in}) - P_1A_1 - P_2A_2$$
$$= 249.55(-1.03 \times 10 - 1.01 \times 10) - 78.47 \times 10^3 \times 0.025 - 65.23 \times 10^3 \times 0.025$$
$$= -5090.82 - 1961.75 - 1630.75$$
$$= -8683.32 \text{N}$$

Therefore, the fluid force acting on the walls of the duct is

$$\boxed{F_x = -F_m = 8683.32 \text{N}}$$

Question: 13.041P

Consider the curved duct of Prob. 13-40, except allow the cross-sectional area to vary along the duct ($A1 \neq A2$). (*a*) Write an expression for the horizontal force *Fx* of the fluid on the walls of the duct in terms of the given variables. (*b*) Verify your expression by plugging in the following values: ρ = 998.2 kg/m3, *A*1 = 0.025 m2, *A*2 = 0.015 m2, $\beta 1$ = 1.02, $\beta 2$= 1.04, *V*1 = 20 m/s, *P*1,gage = 88.34 kPa, and *P*2,gage = 67.48 kPa.

Step 1 of 5

Given data:

Inlet velocity of the fluid, $V_1 = 20$ m/s

Momentum flux correction factor at inlet, $\beta_1 = 1.02$

Gage pressure at inlet, $P_{1,gage} = 88.34$ kPa

Inlet area, $A_1 = 0.025$ m^2

Momentum flux correction factor at exit, $\beta_2 = 1.04$

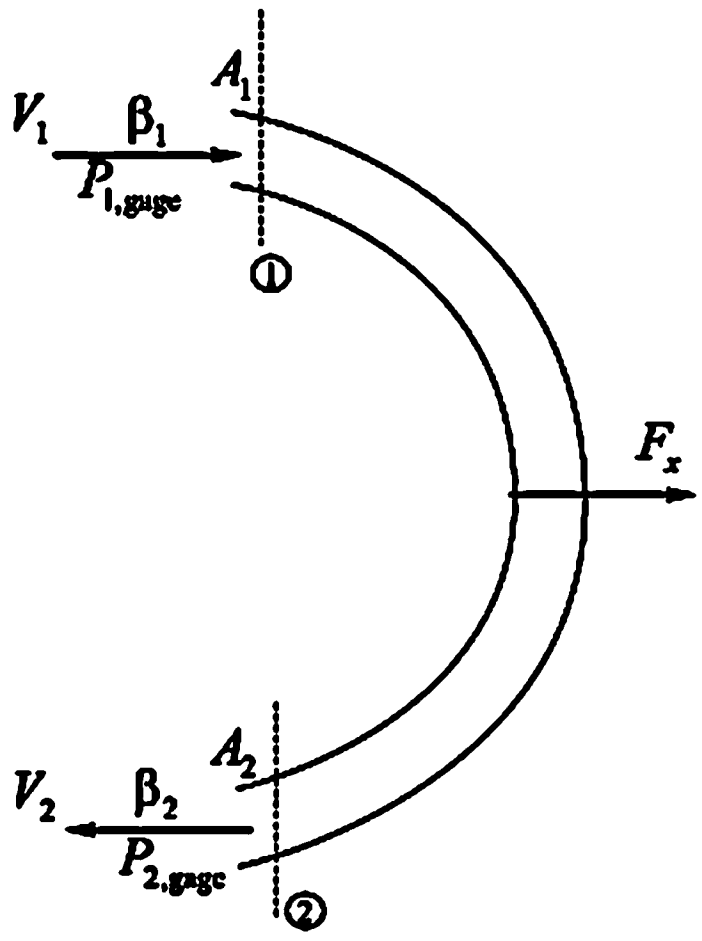
Gage pressure at outlet, $P_{2,gage} = 67.48$ kPa

Outlet area, $A_2 = 0.015$ m^2

Density of the fluid, $\rho = 998.2$ kg/m^3

Step 2 of 5

The curved duct is as shown in figure:



Step 3 of 5

Mass flow rate of the fluid at inlet,

$\dot{m} = \rho A_1 V_1$

$= 998.2 \times 0.025 \times 20$

$= 499.1$ kg/sec

Mass flow is constant throughout the duct,

Therefore,

$\dot{m}_{inlet} = \dot{m}_{outlet}$

$\rho A_1 V_1 = \rho A_2 V_2$

$499.1 = 998.2 \times 0.015 \times V_2$

$V_2 = 33.333$m/sec

Step 4 of 5

(a)

Steady flow momentum equation with external force,

$$\sum F = \sum_{out} \beta \dot{m} V - \sum_{in} \beta \dot{m} V$$

$$F_x + P_{1,gage} A_1 + P_{2,gage} A_2 = \beta_2 \dot{m}(-V_2) - \beta_1 \dot{m} V_1$$

$$F_x = \beta_2 \dot{m}(-V_2) - \beta_1 \dot{m} V_1 - \left(P_{1,gage} A_1 + P_{2,gage} A_2\right)$$

Therefore,

The expression for the horizontal force,

$$\boxed{F_x = \beta_2 \dot{m}(-V_2) - \beta_1 \dot{m} V_1 - \left(P_{1,gage} A_1 + P_{2,gage} A_2\right)}$$

Step 5 of 5

(b)

Steady flow momentum equation with external force,

$$F_x + P_{1,gage} A_1 + P_{2,gage} A_2 = \beta_2 \dot{m}(-V_2) - \beta_1 \dot{m} V_1$$

Therefore, horizontal force,

$$F_x = -\beta_2 \dot{m} V_2 - \beta_1 \dot{m} V_1 - P_{1,gage} A_1 - P_{2,gage} A_2$$

$$= -(1.04 \times 499.1 \times 33.333) - (1.02 \times 499.1 \times 20) - 88.34 \times 10^3 \times 0.025 - 67.48 \times 10^3 \times 0.015$$

$$= -30704.3 \text{ N}$$

Therefore, the horizontal force of the fluid on the walls of the duct, $\boxed{F_x = -30704.3 \text{ N}}$

Question: 13.042P

Water flowing in a horizontal 25-cm-diameter at 8 m/s and 300 kPa gage enters a 90° bend reducing section, which connects to a 15-cm-diameter vertical pipe.The inlet of the bend is 50 cm above the exit. Neglecting any frictional and gravitational effects, determine the net results force exerted on the reducer by the water. Take the momentum -flux correction factor to be 1.04.

Step 1 of 8

Given data:

Inlet velocity of the fluid, $V_1 = 8\ m/s$

Gage pressure at inlet, $P_{1,gage} = 300\ kPa = 300\times10^3\ Pa$

Inlet diameter, $d_1 = 25\ cm = 0.25\ m$

The height of inlet of pipe from datum, $z_1 = 50\ cm = 0.05\ m$

The height of outlet of pipe from datum, $z_2 = 0\ cm = 0\ m$

Outlet diameter, $d_2 = 15\ cm = 0.15\ m$

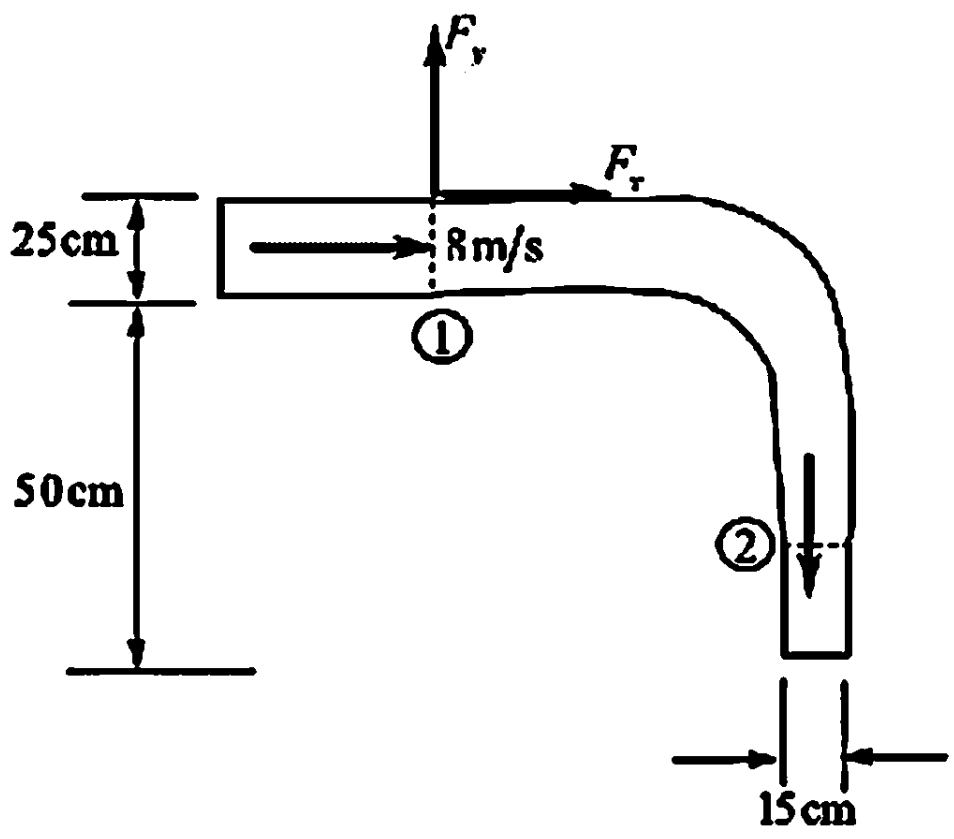
Density of the fluid, $\rho = 1000\ kg/m^3$

Momentum flux correction factor, $\beta = 1.04$

Acceleration due to gravity, $g = 9.81 m/s^2$

Step 2 of 8

The reducer is as shown in figure:



Step 3 of 8

Mass flow rate of the fluid at inlet,

$$\dot{m} = \rho A_1 V_1$$

$$= 1000\times\frac{\pi}{4}\times d_1^2\times 8$$

$$= 1000\times\frac{\pi}{4}\times 0.25^2\times 8$$

$$= 392.699\ m/s$$

Step 4 of 8

Mass flow is constant throughout the duct,

Therefore,

$$\dot{m}_{inlet} = \dot{m}_{outlet}$$

$$\rho A_1 V_1 = \rho A_2 V_2$$

$$392.699 = 1000\times\frac{\pi}{4}\times d_2^2\times V_2$$

$$392.699 = 1000\times\frac{\pi}{4}\times 0.15^2\times V_2$$

$$V_2 = 22.22\ m/s$$

Step 5 of 8

By using Bernoulli equation, we have

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1 = \frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2$$

$$\frac{300\times1000}{1000\times9.81}+\frac{8^2}{2\times9.81}+0.05 = \frac{P_2}{1000\times9.81}+\frac{(22.222)^2}{2\times9.81}+0$$

$$30.581+3.261+0.05 = \frac{P_2}{1000\times9.81}+25.169$$

$$P_2 = 85572.63\ Pa$$

$$= 85.572\ kPa$$

Step 6 of 8

Steady flow momentum equation with external force

$$\sum F = \sum_{ex}\beta\dot{m}V - \sum_{in}\beta\dot{m}V$$

By using the moment equation along x-axis,

$$F_x + P_{1,gage}A_1 = 0 - \beta\dot{m}V_1$$

$$F_x = -\beta\dot{m}V_1 - P_{1,gage}A_1$$

$$F_x = -1.04\times392.699\times8 - 300\times1000\times\frac{\pi}{4}\times0.25^2$$

$$= -17993.471\ N$$

$$= -17.993\ kN$$

Step 7 of 8

By using the moment equation along y-axis,

$$F_y - P_{2,gage}A_2 = \beta\dot{m}(-V_2) - 0$$

$$F_y = -\beta\dot{m}V_2 + P_{2,gage}A_2$$

$$F_y = -1.04\times392.699\times22.222 + 85.572\times1000\times\frac{\pi}{4}\times0.15^2$$

$$F_y = -7563.437\ N$$

$$= -7.563\ kN$$

The net resultant force acting on the reducer by the water,

$$F_R = \sqrt{F_x^2+F_y^2}$$

$$= \sqrt{(-17.993)^2+(-7.563)^2}$$

$$= 19.517\ kN$$

Therefore, the net resultant force acting on the reducer by the water, $\boxed{F_R = 19.517\ kN}$.

Step 8 of 8

The angle of resultant force with positive x-axis,

$$\theta = \tan^{-1}\left(\frac{F_y}{F_x}\right)$$

$$= \tan^{-1}\left(\frac{-7.563}{-17.993}\right)$$

$$= 22.798°$$

Therefore, the angle of resultant force with positive x-axis, $\boxed{\theta = 22.798°}$.

Question: 13.043P

Commercially available large wind turbines have blade span diameters as large as 100 m and generate over 3 MW of electric power at peak design conditions. Consider a wind turbine with a 90-m blade span subjected to 25-km/h steady winds. If the combined turbine-generator efficiency of the wind turbine is 32 percent determine (*a*) the power generated by the turbine and (*b*) the horizontal force exerted by the wind on the supporting mast of the turbine. Take the density of air to be 1.25 kg/m3, and disregard frictional effect.

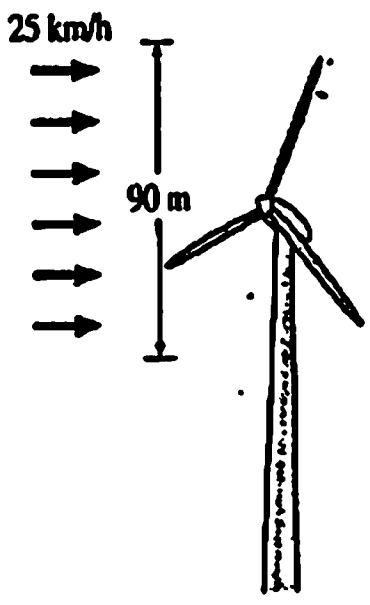


FIGURE P13–43

Step 1 of 5

Diameter of the span, $d = 90\,\text{m}$

Power generated by the electric generator, $W_{act} = 3\,\text{MW}$

Initial velocity of the blade,

$$V_1 = 25\ \text{km/h}$$
$$= 25\times\left(\frac{5}{18}\right)$$
$$= 6.94\ \text{m/s}$$

Generator efficiency,

$$\eta_{gen} = 32\%$$
$$= 0.32$$

Density of air, $\rho = 1.25\ \text{kg/m}^3$

Step 2 of 5

$$\dot{m} = \rho A_1 V_1$$

Mass flow rate,
$$= 1.25\times\frac{\pi(90)^2}{4}\times 6.94$$
$$= 55187.96\ \text{m/s}$$

Step 3 of 5

Turbine efficiency,

$$\eta_{turbine} = \frac{W_{act}}{W_{max}}$$
$$0.32 = \frac{3}{W_{max}}$$
$$W_{max} = \frac{3}{0.32}$$
$$= 9.37\ \text{MW}$$

Therefore, the power developed by the turbine is $\boxed{W_{max} = 5.12\ \text{MW}}$

Step 4 of 5

The exit velocity of blade for wind turbine is determined as,

$$V_2 = V_1\sqrt{(1-\eta_{turbine})}$$
$$= 6.94\sqrt{(1-0.32)}$$
$$= 5.72\ \text{m/s}$$

Step 5 of 5

The moment equation, we have

$$\sum\vec{F} = \sum_{out}\beta\dot{m}\vec{V} - \sum_{in}\beta\dot{m}\vec{V} \quad \text{..... (1)}$$

Where,

Moment flux correction factor for study flow is $\beta = 1$

The equation (1) can be re-written as,

$$\vec{F} = \beta\dot{m}V_2 - \beta\dot{m}V_1$$
$$= \beta\dot{m}(V_2 - V_1)$$
$$= 1\times 55187.96\times(5.72-6.94)$$
$$= -67329.31\ \text{N}$$

Therefore, the horizontal force exerted by the wind is $\boxed{\vec{F} = -67329.31\ \text{N}}$

Question: 13.044P pg 1

Water enters vertically and steadily at a rate of 15 L/s into the sprinkler shown in Fig. P13-44. Both water jets have a diameter of 1.2 cm. Disregarding any frictional effects, determine (*a*) the rotational speed of the sprinkler in rpm and (*b*) the torque required to prevent the sprinkler from rotating.

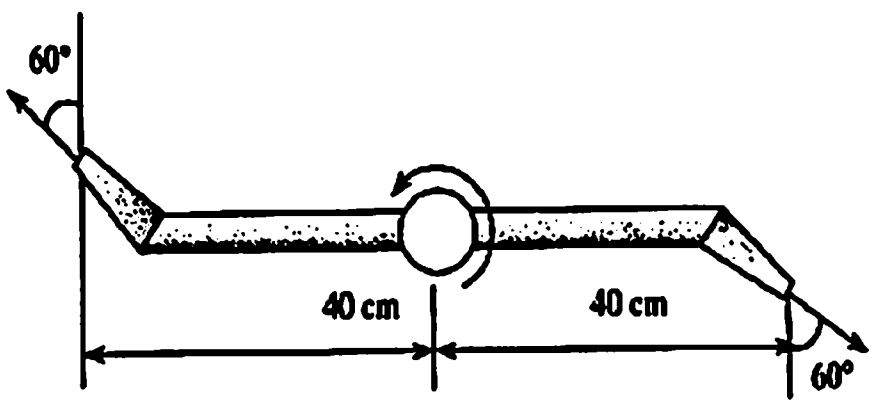


FIGURE P13–44

Step 1 of 9

Given data:

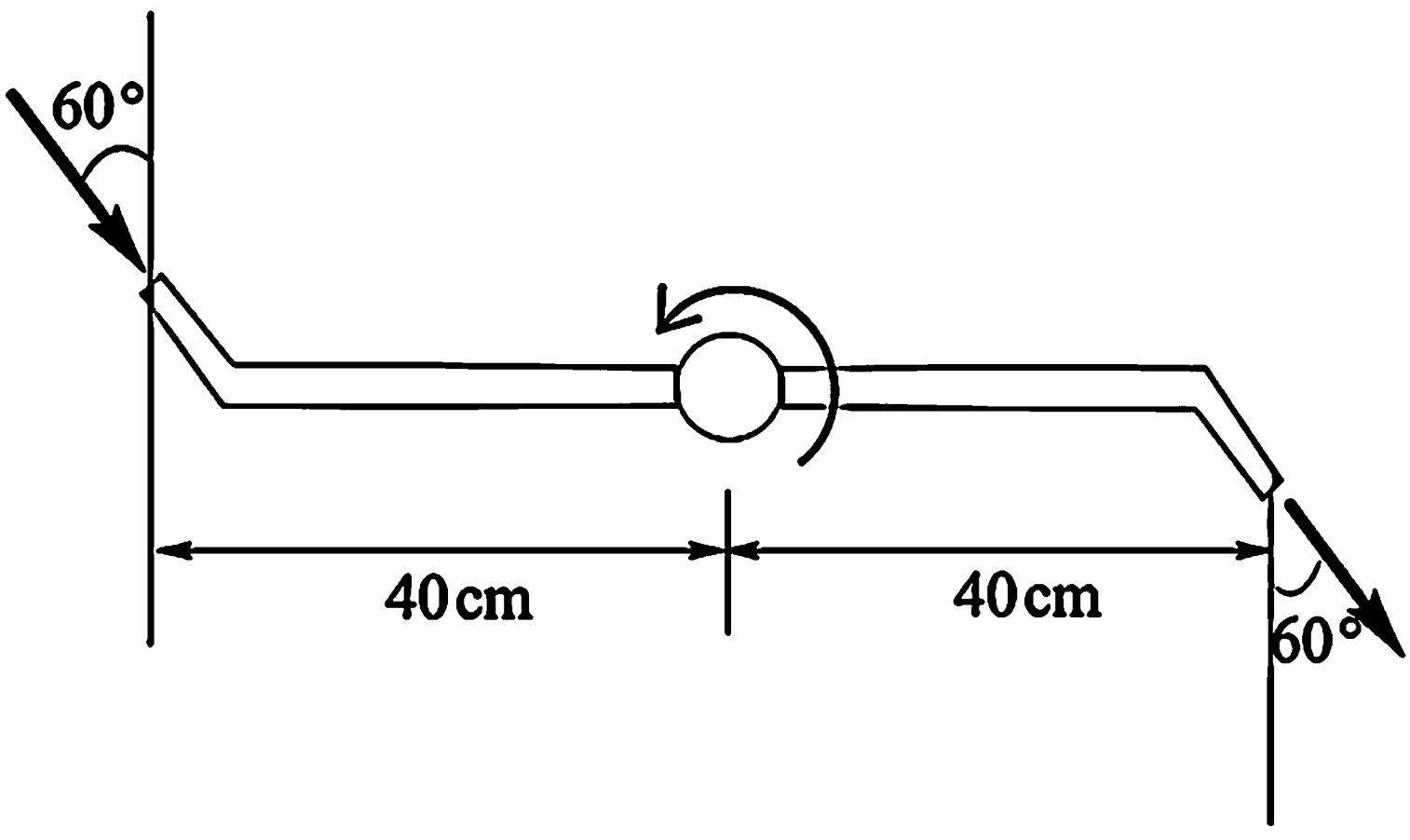
The mass flow rate of water along the vertical pipe,

$$\dot{m}_v = 15 \text{ L/s}$$
$$= 14.4 \text{ kg/s}$$

The diameter of water jets, $D = 1.2 \text{ cm}$

Step 2 of 9

Given figure:



Step 3 of 9

The density of water is $\rho = 1000 \text{ kg/m}^3$.

Since, both the nozzles are identical and at equal distance from the center.

Then,

$$\dot{m}_1 = \dot{m}_2 = \frac{\dot{m}_v}{2}$$

Therefore, mass flow rate,

$$\dot{m} = \frac{\dot{m}_v}{2}$$
$$= \frac{14.4}{2}$$
$$= 7.2 \text{ kg/s}$$

Step 4 of 9

The volumetric flow rate,

$$\dot{V} = \frac{\dot{m}}{\rho}$$
$$= \frac{7.2}{1000}$$
$$= 0.0072 \text{ m}^3/\text{s}$$

Step 5 of 9

The velocity of jet,

$$V_{jet} = \frac{\dot{V}}{A_{jet}}$$
$$= \frac{0.0072}{2\left[\frac{\pi}{4}(0.012^2)\right]}$$
$$= 31.83 \text{ m/s}$$

Step 6 of 9

The angular momentum equation can be expressed as,

$$\sum M = \left(\sum rmV_{out}\right) - \left(\sum rmV_{in}\right)$$

We know that there are no external moments acting on the system. Then the momentum equation about the axis of rotation becomes

$$0 = -2rmV_r \cos\theta$$

Therefore the relative velocity of the jet $V_r = 0$

Step 7 of 9

Therefore the relative velocity of the jet, V_r = 13.044P pg 2

Step 7 of 9

We also know,

$$V_r = V_{jet,s} - V_{nozzle}$$

Then

$$V_{jet,s} - V_{nozzle} = 0$$
$$V_{jet,s} = V_{nozzle}$$
$$V_{jet}\cos 60^\circ = V_{nozzle}$$

$$V_{nozzle} = V_{jet}\cos 60^\circ$$
$$= 31.83 \times \cos 60^\circ$$
$$= 15.915 \text{ m/s}$$

Step 8 of 9

(a)

Calculation of rotational speed of the sprinkler:

We know that,

$$V_{nozzle} = \omega \frac{D}{2}$$

Then, rate of rotation or angular velocity is,

$$\omega = \frac{V_{nozzle}}{(D/2)}$$
$$= \frac{15.915}{(0.4/2)}$$
$$= 79.575 \text{ rad/s}$$

The rotational speed,

$$\dot{n} = \frac{\omega \times 60}{2\pi}$$
$$= \frac{79.575 \times 60}{2\pi}$$
$$= 759.9 \text{ rpm}$$

Therefore, the rotational speed of the system is $\boxed{\dot{n} = 759.9 \text{ rpm}}$

Step 9 of 9

(b)

The torque required to prevent the sprinkler from rotating:

The torque can be expressed as,

$$T_{shaft} = r\dot{m}_v V_{jet}$$
$$= (0.2)(14.4)(31.83)$$
$$= 91.67 \text{ N} \cdot \text{m}$$

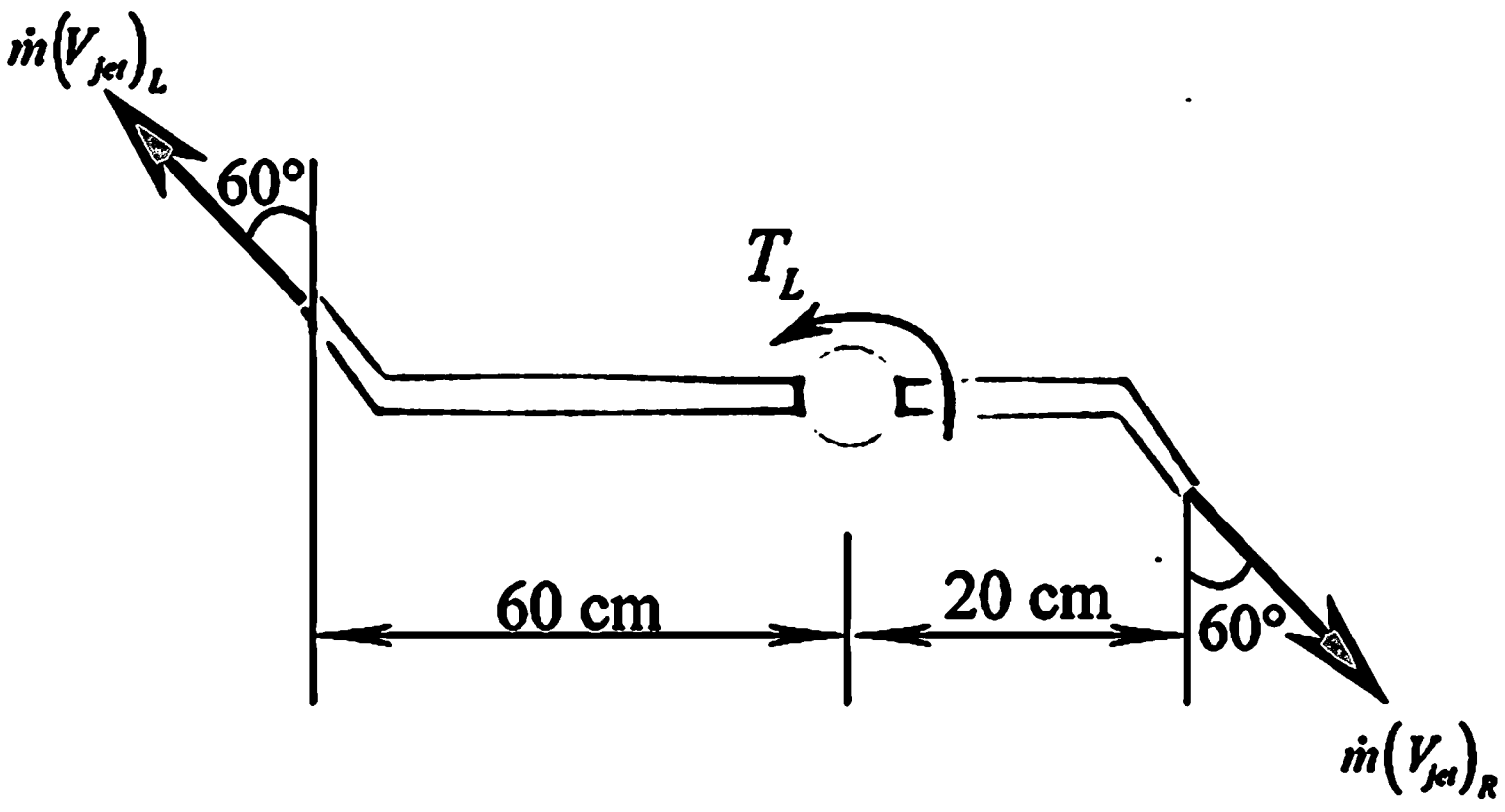
Therefore, the torque required is $\boxed{T_{shaft} = 91.67 \text{ N} \cdot \text{m}}$

Question: 13.045P pg 1

Repeat Prob. 13-44 for the case of unequal arms —the left one being 60 cm and the right one 20 cm from the axis of rotation.

Step 1 of 9

Draw the schematic diagram of the sprinkler.



Step 2 of 9

Convert the units of volume flow rate of water L/s to m^3/s.

$$1\ L/s = 10^{-3}\ m^3/s$$

$$Q = 15\ L/s$$
$$= 15\times10^{-3}\ m^3/s$$

Find the mass flow rate of water by using the equation.

$$\dot{m} = \rho Q$$

Here, Q is volume flow rate of water, and ρ is the density of water.

Substitute $15\times10^{-3}\ m^3/s$ for Q, and $1000\ kg/m^3$ for ρ.

$$\dot{m} = 1000\times15\times10^{-3}$$
$$= 15\ kg/s$$

Step 3 of 9

Find the cross sectional area for left section of the sprinkler using the equation.

$$A_L = \frac{\pi}{4}d^2$$

Here, d is the diameter of the jet.

Substitute 0.012 m for the value d.

$$A_L = \frac{\pi}{4}\times0.012^2$$
$$= 1.13\times10^{-4}\ m^2$$

Find the cross sectional area for right section of the sprinkler using the equation.

$$A_R = \frac{\pi}{4}d^2$$

Substitute 0.012 m for the value d.

$$A_R = \frac{\pi}{4}\times0.012^2$$
$$= 1.13\times10^{-4}\ m^2$$

Step 4 of 9

Find the velocity at left section of the sprinkler using the equation.

$$\frac{Q}{2} = A_L V_L$$

Here, A_L is the cross sectional area at left section of the sprinkler and V_L is the velocity at larger section of the sprinkler.

Substitute $15\times10^{-3}\ m^3/s$ for Q and $1.13\times10^{-4}\ m^2$ for A_L.

$$\frac{15\times10^{-3}}{2} = 1.13\times10^{-4}V_L$$
$$V_L = 66.37\ m/s$$

Find the velocity along the vertical direction at left jet.

$$(V_L)_V = V_L\cos\theta$$

Here, θ is the inclination.

Substitute $66.37\ m/s$ for V_L, and $60°$ for θ.

$$(V_L)_V = 66.37\cos60°$$
$$= 33.18\ m/s$$

Step 5 of 9

Find the velocity at right section of the sprinkler using the equation.

$$\frac{Q}{2} = A_R V_R$$

Here, A_R is the cross sectional area at right section of the sprinkler and V_R is the velocity of the sprinkler.

Substitute $15\times10^{-3}\ m^3/s$ for Q and $1.13\times10^{-4}\ m^2$ for A_R.

2

Here, A_R is the cross sectional area at right section of the sprinkler and V_R is the velocity of the sprinkler.

Substitute $15\times10^{-3}\ \text{m}^3/\text{s}$ for Q and $1.13\times10^{-4}\ \text{m}^2$ for A_R.

$$\frac{Q}{2}=A_RV_R$$
$$\frac{15\times10^{-3}}{2}=1.13\times10^{-4}V_R$$
$$V_R=66.37\ \text{m/s}$$

Step 6 of 9

Find the velocity along the vertical direction at right jet.

$$(V_R)_V=V_R\cos\theta$$

Substitute $66.37\ \text{m/s}$ for V_L, and $60°$ for θ.

$$(V_R)_V=66.37\cos60°$$
$$=33.18\ \text{m/s}$$

Find the velocity of the nozzle with respect to left jet using the equation,

$$(V_{nozzle})_L=r_L\omega$$

Here, r_L is the arc length and ω is the angular speed of sprinkler.

Substitute 0.6 m for r_L.

$$(V_{nozzle})_L=0.6\omega$$

Find the velocity of the nozzle with respect to right jet using the equation.

$$(V_{nozzle})_R=r_R\omega$$

Here, r_R is the arc length.

Substitute 0.2 m for r_R.

$$(V_{nozzle})_R=0.2\omega$$

Step 7 of 9

Find the absolute velocity of water at left jet.

$$(V_{jet})_L=(V_L)_V-(V_{nozzle})_L$$

Substitute $33.18\ \text{m/s}$ for $(V_L)_V$ and (0.6ω) for $(V_{nozzle})_L$.

$$(V_{jet})_L=33.18-0.6\omega$$

Find the absolute velocity water at right jet.

$$(V_{jet})_R=(V_R)_V-(V_{nozzle})_R$$

Substitute $33.18\ \text{m/s}$ for $(V_R)_V$ and (0.2ω) for $(V_{nozzle})_R$.

$$(V_{jet})_R=33.18-0.2\omega$$

Step 8 of 9

(a)

Apply angular momentum equation for no external moments acting on the system.

$$\sum M=\sum rmV_{out}-\sum rmV_{in}$$
$$0=r_L\dot{m}(V_{jet})_R-r_R\dot{m}(V_{jet})_L$$

Substitute $(33.18-0.6\omega)$ for $(V_{jet})_L$, $15\ \text{kg/s}$ for $\dot{m}$, 0.6 m for r_L, 0.2 for r_R and $(33.18-0.2\omega)$ for $(V_{jet})_R$.

$$0=r_L\dot{m}(V_{jet})_R-r_R\dot{m}(V_{jet})_L$$
$$0=0.6\times15\times(33.18-0.6\omega)-0.2\times15\times(33.18-0.2\omega)$$
$$9(33.18-0.6\omega)=3(33.18-0.2\omega)$$
$$199.08=4.8\omega$$
$$\omega=41.47\ \text{rad/s}$$

Find the rotations speed of sprinkler using the equation.

$$n=\frac{\omega\times60}{2\pi}$$

Substitute $41.47\ \text{rad/s}$ for ω.

$$n=\frac{41.47\times60}{2\pi}$$
$$=396\ \text{rpm}$$

Therefore, the rotational speed of sprinkler is $\boxed{396\ \text{rpm}}$.

(b)

Find the absolute velocity of water on the sprinkler due to the left jet.

$$(V_{jet})_L=33.18-0.6\omega$$

Substitute $41.47\ \text{rad/s}$ for ω.

$$(V_{jet})_L=33.18-(0.6\times41.47)$$
$$=8.29\ \text{N}\cdot\text{m}$$

Find the absolute velocity of water on the sprinkler due to the right jet.

$$(V_{jet})_R=33.18-0.2\omega$$

$= 8.29\ \text{N}\cdot\text{m}$

Find the absolute velocity of water on the sprinkler due to the right jet.

$$\left(V_{jet}\right)_R = 33.18 - 0.2\omega$$

Substitute $41.47\ \text{rad/s}$ for ω.

$$\left(V_{jet}\right)_R = 33.18 - (0.2 \times 41.47)$$
$$= 24.88\ \text{N}\cdot\text{m}$$

Step 9 of 9

Find the torque on the sprinkler due to the left jet.

$$T_L = r_L \dot{m} \left(V_{jet}\right)_L$$

Substitute $8.29\ \text{m/s}$ for $\left(V_{jet}\right)_L$, $15\ \text{kg/s}$ for $\dot{m}$ and 0.6 m for r_L.

$$T_L = 0.6 \times 15 \times 8.29$$
$$= 74.61\ \text{N}\cdot\text{m}$$

Find the torque on the sprinkler due to the right jet.

$$T_R = r_R \dot{m} \left(V_{jet}\right)_R$$

Substitute $24.88\ \text{m/s}$ for $\left(V_{jet}\right)_R$, $15\ \text{kg/s}$ for $\dot{m}$ and 0.35 m for r_R.

$$T_R = 0.35 \times 15 \times 24.88$$
$$= 130.62\ \text{N}\cdot\text{m}$$

Find the total torque required to prevent sprinkler from rotating.

$$T = T_L + T_R$$
$$= 74.61 + 130.62$$
$$= 205.23\ \text{N}\cdot\text{m}$$

Therefore, the total torque required to prevent sprinkler from rotating is

$\boxed{205.23\ \text{N}\cdot\text{m}}$.

Question: 13.046P

A 60-kg ice skater is standing on ice with ice skates (negligible friction). She is holding a flexible hose (essentially weightless) that directs a 2-cm-diameter stream of water horizontally parallel to her skates. The water velocity at the hose outlet is 10 m/s relative to the skater. If she is initially standing still, determine (*a*) the velocity of the skater and the distance she travels in 5 s and (*b*) how long it will take to move 5 m and the velocity at that moment.

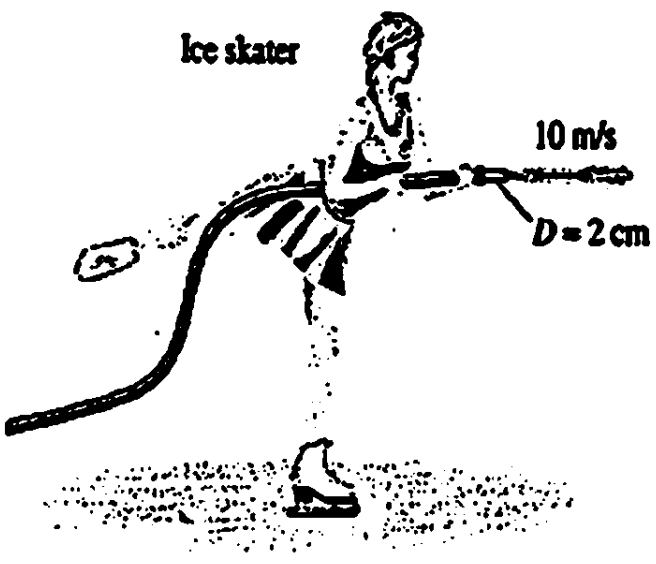


FIGURE P13–46

Step 1 of 4

(a)

Calculate the mass flow rate of water through the hose by using the following equation

$$\dot{m} = \rho A V$$

Here, density of water is ρ , the cross-sectional area of the hose is A and the velocity of the water at the hose outlet V

Substitute $1000\,\text{kg/m}^3$ for ρ, $\frac{\pi(0.02\,\text{m})^2}{4}$ for A and $10\,\text{m/s}$ for V in the equation

$$\dot{m} = (1000\,\text{kg/m}^3)\frac{\pi(0.02\,\text{m})^2}{4}(10\,\text{m/s})$$
$$= 3.14\,\text{kg/s}$$

Step 2 of 4

The force exerted on the skater by the water stream (F) is the momentum flux of the water stream acting opposite to the direction of water flow.

$$F = \dot{m}V$$
$$= (3.14\,\text{kg/s})(10\,\text{m/s})$$
$$= 31.4\,\text{N}$$

(*a*)

Find the acceleration of the skater

$$a = \frac{F}{m}$$

Substitute $31.4\,\text{N}$ for F and $60\,\text{kg}$ for m in the equation

$$a = \frac{F}{m}$$
$$= \frac{31.4\,\text{N}}{60\,\text{kg}}$$
$$= 0.523\,\text{m/s}^2$$

Find the velocity of the skater after 5 seconds

$$V_{skater} = at$$
$$= (0.523\,\text{m/s}^2)(5\,\text{s})$$
$$= 2.62\,\text{m/s}$$

The velocity of the skater after 5 s is $2.62\,\text{m/s}$.

Step 3 of 4

Determine the distance traveled by the skater (x) after 5 s.

$$x = \frac{1}{2}at^2$$
$$= \frac{1}{2}(0.523\,\text{m/s}^2)(5\,\text{s})^2$$
$$= 6.54\,\text{m}$$

The distance traveled by the skater after 5 s is $6.54\,\text{m}$.

Step 4 of 4

(b)

Use kinematics to determine the time it takes to move 5 meters distance.

$$t = \sqrt{\frac{2x}{a}}$$

Here, distance moved is x

Substitute $5\,\text{m}$ for x and $0.523\,\text{m/s}^2$ for a in the equation

$$t = \sqrt{\frac{2(5\,\text{m})}{0.523\,\text{m/s}^2}}$$
$$= 4.4\,\text{s}$$

Therefore, the time taken for the skater to travel 5 m distance is $4.4\,\text{s}$

Use kinematics to determine the velocity of the skater after she has traveled 5 m.

$$V_{skater} = at$$
$$= (0.523\,\text{m/s}^2)(4.4\,\text{s})$$
$$= 2.3\,\text{m/s}$$

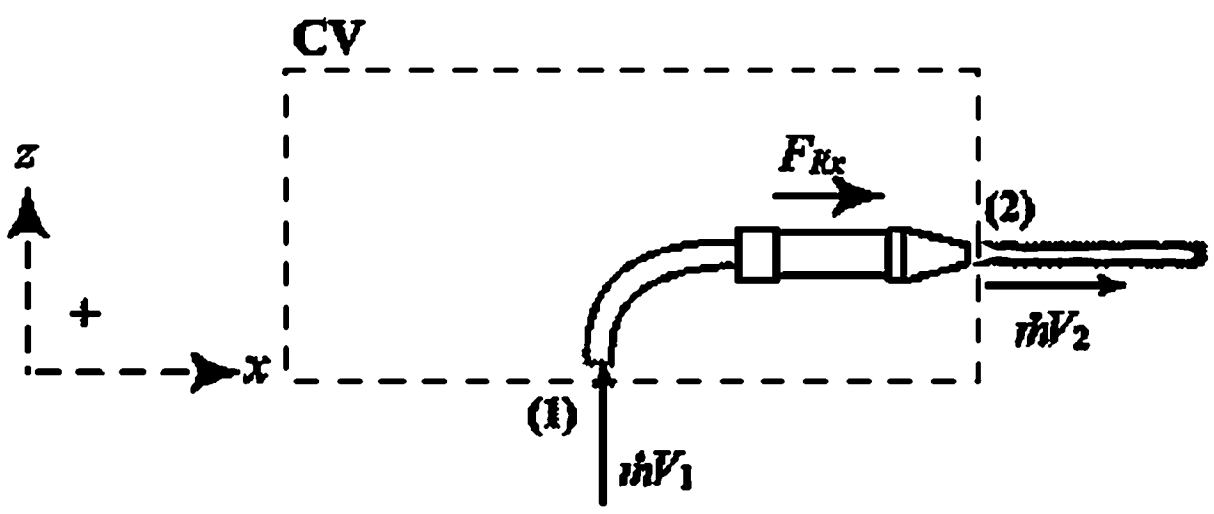
Therefore, the velocity of the skater after she has traveled 5 m is $2.3\,\text{m/s}$.

Question: 13.047P

Show that the force exerted by a liquid jet on a stationary nozzle as it leaves with a velocity *V* is proportional to *V* 2 or, alternatively, to *m*2. Assume the jet stream is perpenticular to the incoming liquid flow line.

Step 1 of 5

Schematic representation of jet, nozzle and direction of forces:



Step 2 of 5

Let us prove that, the force exerted by the jet is proportional to square of velocity of jet:
The momentum equation for steady one dimensional flow in the horizontal direction can be expressed as,

$$\sum F = \sum \beta \dot{m} V_{out} - \sum \beta \dot{m} V_{in}$$

From the figure, the inlet components are in the vertical direction. Since, we can get the reaction force with horizontal components alone, we can neglect the inlet components.

Step 3 of 5

Then the above momentum equation becomes,

$$F_{Rx} = \dot{m}V_2 - 0$$

Since,

$$\dot{m} = \rho A V$$

The momentum equation, $F_{Rx} = \rho A V_2^2$

Thus, the nozzle exerts a reaction force $F_{Rx} = \rho A V_2^2$ against the jet or the jet exerts a force of magnitude equal to the reaction force in the opposite direction.

That is $\boxed{F_{jet} = \rho A V_2^2}$.

Hence proved, that the force exerted by the jet is proportional to V_2^2.

Step 4 of 5

Let us prove that, the force exerted by the jet is proportional to
From Newton's law, we have

$$F_{Rx} = \dot{m}V_2$$

We also know that,

$$V_2 = \frac{\dot{m}}{\rho A}$$

Step 5 of 5

Then, the reaction force,

$$F_{Rx} = \dot{m}\left(\frac{\dot{m}}{\rho A}\right)$$

$$= \frac{\dot{m}^2}{\rho A}$$

The nozzle exerts a reaction force $F_{Rx} = \frac{\dot{m}^2}{\rho A}$ against the jet or the jet exerts a force of magnitude equal to the reaction force in the opposite direction.

$$\boxed{F_{jet} = \frac{\dot{m}^2}{\rho A}}.$$

Hence proved, that the force exerted by the jet is proportional to $\dot{m}^2$.

Question: 13.048P

A tripod holding a nozzle, which directs a 5-cm-diameter stream of water from a hose, is shown in Fig. P13-48. The nozzle mass is 10 kg when filled with water. The tripod is rated to provide 1800 N of holding force. A firefighter was standing 60 cm behind the nozzle and was hit by the nozzle when the tripod suddenly failed and released the nozzle. You have been hired as an accident reconstructionist and, after testing the tripod, have determined that as water flow rate increased, it did collapse at 1800 N. In your final report you must state the water velocity and the flow rate consistent with the failure and the nozzle velocity when it hit the firefighter. For simplicity, ignore pressure and momentum effects in the upstream portion of the hose.

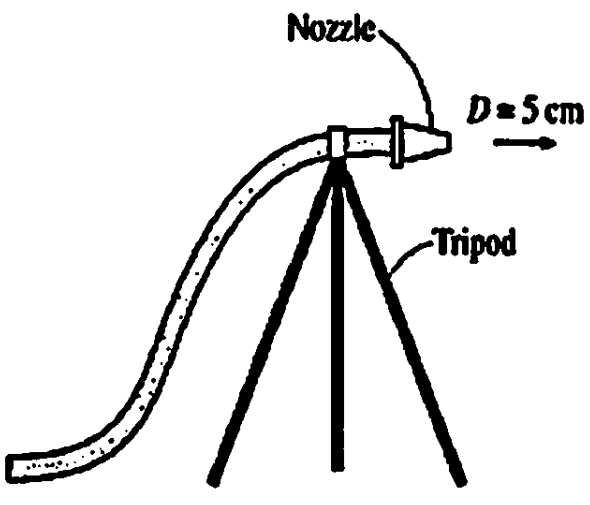


FIGURE P13–48

Step 1 of 7

Given data:

The reaction force offered by the tripod in horizontal direction, $F_{Rx} = 1800$ N

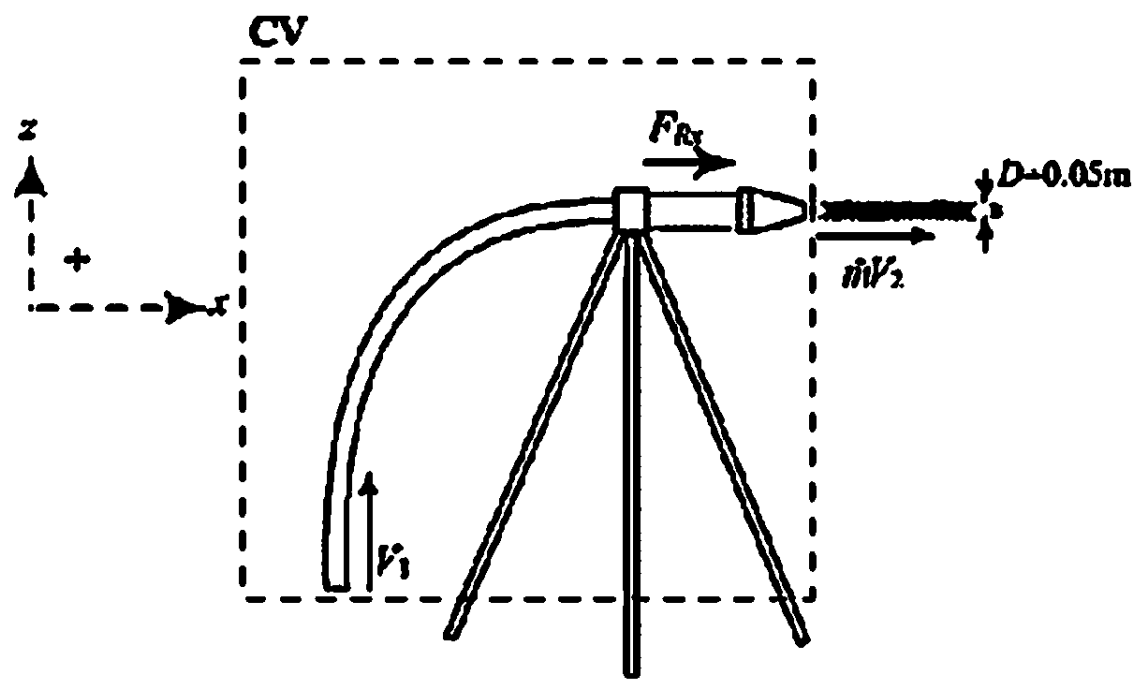
The mass of the nozzle, $m_N = 10$ kg

The distance between the nozzle and firefighter, $d = 0.6$ m

The diameter of water stream at nozzle, $D = 0.05$ m

Step 2 of 7

Given figure:



Step 3 of 7

Calculation of exit velocity of the stream:

The velocity of the stream at exit is,

$$V_2 = \sqrt{\frac{F_{Rx}}{\rho A}} \qquad \ldots\ldots (1)$$

The area of water stream at exit,

$$A = \pi\frac{D^2}{4}$$
$$= \pi\frac{0.05^2}{4}$$
$$= 0.0019634 \text{ m}^2$$

Step 4 of 7

The momentum equation for steady one dimensional flow in the horizontal direction can be expressed as,

$$\sum F = \sum \beta \dot{m} V_{out} - \sum \beta \dot{m} V_{in}$$

Since, the momentum flux correction factor is negligible and $V_1 = 0$

Then,

$$F_{Rx} = \dot{m}V_2 - 0$$
$$F_{Rx} = \rho A V_2^2 \qquad (\text{since, } \dot{m} = \rho A)$$

By substituting the known values in equation (1), we get

$$V_2 = \sqrt{\frac{F_{Rx}}{\rho A}}$$
$$= \sqrt{\frac{1800}{1000 \times 0.0019634}}$$
$$= 30.27 \text{ m/s}$$

Therefore, the exit velocity of the stream is $\boxed{V_2 = 30.27 \text{ m/s}}$

Step 5 of 7

Calculation of flow rate of water:

Flow rate of water can be calculated as,

$$\dot{V} = AV_2$$
$$= (0.0019634)(30.27)$$
$$= 0.05944 \text{ m}^3/\text{s}$$

Therefore, the volumetric flow rate is $\boxed{\dot{V} = 0.05944 \text{ m}^3/\text{s}}$

Step 6 of 7

Calculation of the velocity of the nozzle when it strikes the fire fighter:

From Newton's law we have, the holding force,

$$F_{Rx} = m_N a$$

Then,

$$a = \frac{F_{Rx}}{m_N}$$
$$= \frac{1800}{10}$$
$$= 180 \text{ m/s}^2$$

We have the equation for motion

$$s = V_0 t + \frac{1}{2}at^2$$

For the nozzle,

$$0.6 = 0 + \frac{1}{2}(180)t^2$$

Then

$$t = \sqrt{\frac{2 \times 0.6}{180}}$$
$$= 0.0816 \text{ s}$$

Step 7 of 7

The final velocity of nozzle,

$$V = V_0 + at$$
$$= 0 + (180)(0.0816)$$
$$= 14.7 \text{ m/s}$$

Therefore, the velocity of the nozzle when it strikes the fire fighter is, $\boxed{V = 14.7 \text{ m/s}}$

Question: 13.049P

Consider an airplane with a jet engine attached to the tail section that expels combustion gases at a rate of 18 kg/s with a velocity of $V = 300$ m/s relative to the plane. During landing, a thrust reverser (which serves as a brake for the aircraft and facilitates landing on a short runway) is lowered in the path of the exhaust jet, which deflects the exhaust from rearward to 150°. Determine (*a*) the thrust (forward force) that the engine produces prior to the insertion of the thrust reverser and (*b*) the braking force produced after the thrust reverser is deployed.

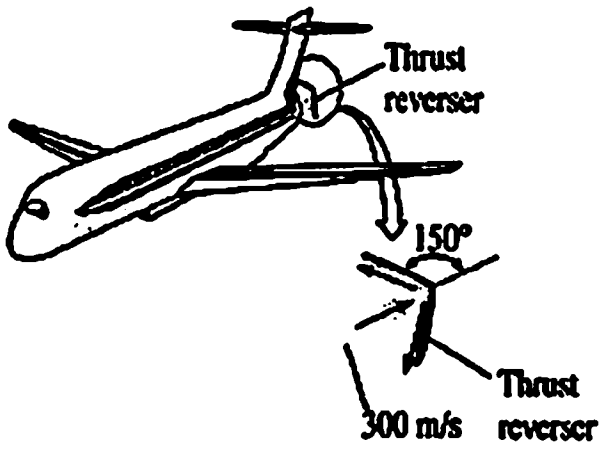


FIGURE P13–49

Step 1 of 3

Given data:

The mass flow rate of combustion gases, $\dot{m} = 18\ \text{kg/s}$

The velocity of exhaust gases, $V = 300\ \text{m/s}$

The angle of exhaust from rearward, $\alpha = 150°$

Step 2 of 3

a)

By using formula for the thrust force exerted on the air plane, we get

$$\begin{aligned} F_{\text{thrust}} &= \dot{m}V \\ &= 18 \times 300 \\ &= 5400\ \text{N} \end{aligned}$$

Therefore, the thrust force exerted on the air plane, $\boxed{F_{\text{thrust}} = 5400\ \text{N}}$

Step 3 of 3

b)

By using the moment equation for steady one-dimensional flow in x -direction, we get

$$\sum F = \sum_{\text{out}} \beta \dot{m} V - \sum_{\text{in}} \beta \dot{m} V$$

$$\begin{aligned} F_{Rx} &= \dot{m}(-V)\cos(180° - \alpha) + \dot{m}(-V)\cos(180° - \alpha) - \dot{m}V \\ &= 18(-300)\cos(180° - \alpha) + 18(-300)\cos(180° - \alpha) - 18 \times 300 \\ &= 18(-300)\cos(180° - 150) + 18(-300)\cos(180° - 150) - 18 \times 300 \\ &= -14753.074\ \text{N} \end{aligned}$$

The braking force acting on the air plane is equal and opposite of the force F_{Rx}.

Therefore the braking force acting on the plane, $\boxed{F_{\text{braking}} = -14753.074\ \text{N}}$.

Question: 13.050P

Reconsider Prob. 13-49. Using EES (or other) software, investigate the effect of / thrust reverser angle on the braking force exerted on the airplane. Let the reverser angle vary from 0° (no reversing) to 180° (full reversing) in increments of 10°. Tabulate and plot your results and draw conclusions.

Step 1 of 5

The procedure for developing solution by using EES software:
Open the new equations window EES software and type all the information which is used to solve the problem
For developing table:
1) Click on table followed by new parametric table and add the variables, which are varying variable and required answer variable, then one table will form with that variables
2) Click on the variable which is varying and enter initial value and final value.
3) For getting the required answer variable values corresponding to the varying variable values click on the green color button and then the table will form with required answers
For developing graph:
1) Click on plot followed by new plot window followed by x-y plot.
2) Select the x-axis variable and y-axis variable and then press ok, then the graph will form with the given values.

Step 2 of 5

The equations which are using in EES software for developing the solution are given in equations window is as shown below

Equations Window

```
V_jet=300 "m/s"
m_dot=18 "Kg/s"
F_Rx=-(1-2*cos(alpha))*m_dot*V_jet "N"
```

Step 3 of 5

The EES software will convert the equations in equations window to equations in formatting window is as shown below

Formatted Equations

$V_{jet} = 300 \ m/s$

$\dot{m} = 18 \ Kg/s$

$F_{Rx} = -(1 - 2 \cdot \cos(\alpha)) \cdot \dot{m} \cdot V_{jet} \ N$

Step 4 of 5

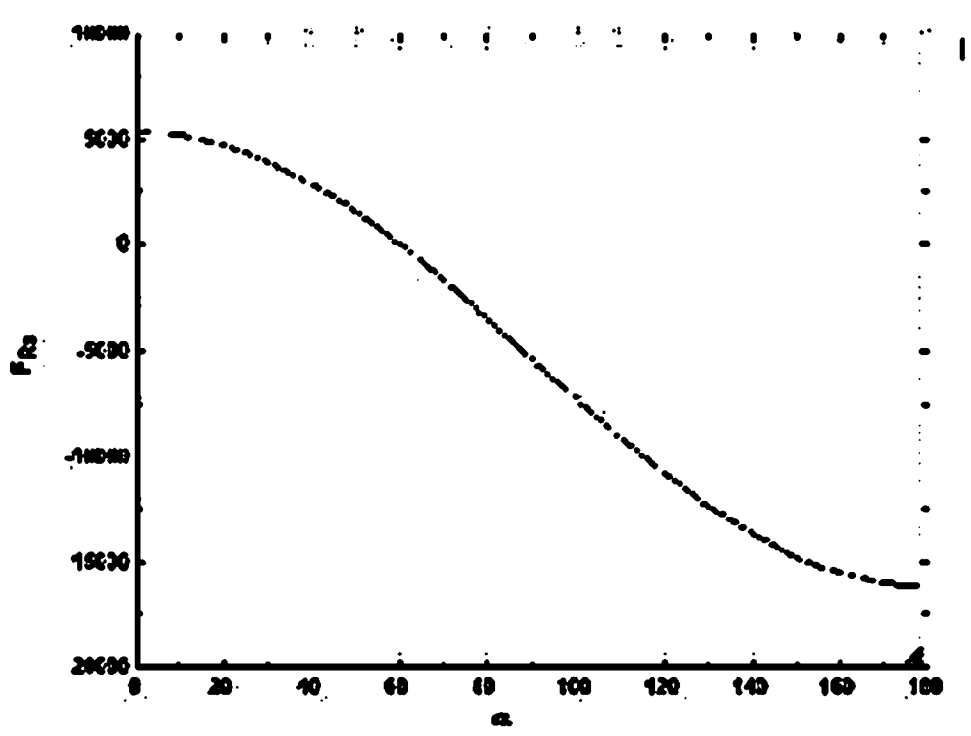
The EES software is developing the table for F_{Rx} corresponding to the α values is as shown below,

Table 1

1..19	1 α	2 F_{Rx}
Run 1	0	5400
Run 2	10	5236
Run 3	20	4749
Run 4	30	3953
Run 5	40	2873
Run 6	50	1542
Run 7	60	-6.446E-10
Run 8	70	-1706
Run 9	80	-3525
Run 10	90	-5400
Run 11	100	-7275
Run 12	110	-9094
Run 13	120	-10800
Run 14	130	-12342
Run 15	140	-13673
Run 16	150	-14753

Step 5 of 5

The EES software is developing the graph between F_{Rx} and α is as shown below,



Question: 13.051P

A spacecraft cruising in space at a constant velocity of 1500 ft/s has a mass of 18,000 lbm. To slow down the spacecraft, a solid fuel rocket is fired, and the combustion gases leave the rocket at a constant rate of 150 lbm/s at a velocity of 5000 ft/s in the same direction as the spacecraft for a period of 5 s. Assuming the mass of the spacecraft remains constant, determine (*a*) the deceleration of the spacecraft during this 5-s period, (*b*) the change of velocity of the spacecraft during this time period, and (c) the thrust exerted on the spacecraft.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 13.052P

A 6-cm-diameter horizontal water jet having a velocity of 25 m/s strikes a vertical stationary flat plate. The water splatters in all directions in the plane of the plate. How much force is required to hold the plate against the water stream?

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 13.053P

A 5-cm-diameter horizontal jet of water, with velocity 30 m/s, strikes the tip of a horizontal cone, which deflects the water by 45° from its original direction. How much force is required to hold the cone against the water stream?

Step 1 of 2

Calculate the mass flow rate by using the following equation

$$\dot{m} = \rho A V$$

Here, density of water is ρ , cross-sectional area is A and the velocity of jet is V.

Substitute $1000\ \text{kg/m}^3$ for ρ, $\frac{\pi(0.05\ \text{m})^2}{4}$ for A and $30\ \text{m/s}$ for V in the equation

$$\dot{m} = (1000\ \text{kg/m}^3)\frac{\pi(0.05\ \text{m})^2}{4}(30\ \text{m/s})$$
$$= 58.90\ \text{kg/s}$$

Step 2 of 2

Consider the momentum flux equation

$$\sum F = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}$$

Here, momentum flux correction factor is β, mass flow rate is $\dot{m}$ and velocity is V

There are two forces acting on the cone, the anchoring forces in each direction (F_{Rx}, F_{Ry}). There is one source of mass into the region, the jet, and one source of mass out of the region, the deflected water.

Now consider the anchoring force in the x direction, the water velocity of the jet is the same as the water velocity of the deflected water.

$$F_{Rx} = \dot{m} V \cos\theta - \dot{m} V$$

Substitute $58.90\ \text{kg/s}$ for $\dot{m}$, $45°$ for θ and $30\ \text{m/s}$ for V in the equation

Plug in the calculated mass flow rate, the given velocity, and the angle of 45°.

$$F_{Rx} = (58.90\ \text{kg/s})(30\ \text{m/s})(\cos 45° - 1)$$
$$= -518\ \text{N}$$

The anchoring force in the y direction is zero because of symmetry

$$F_{Ry} = 0\ \text{N}$$

Therefore a force of $518\ \text{N}$ must be applied to hold the cone against the water stream.

Question: 13.054P

Water is flowing into and discharging from a pipe U-section as shown in Fig. PI3-54. At flange (1), the total absolute pressure is 200 kPa, and 42 kg/s flows into the pipe. At flange (2), the total pressure is 150 kPa. At location (3), 10 kg/s of water discharges to the atmosphere, which is at 100 kPa. Determine the total *x*- and *z*-forces at the two flanges connecting the pipe. Discuss the significance of gravity force for this problem. Take the momentum-flux correction factor to be 1.03 throughout the pipes.

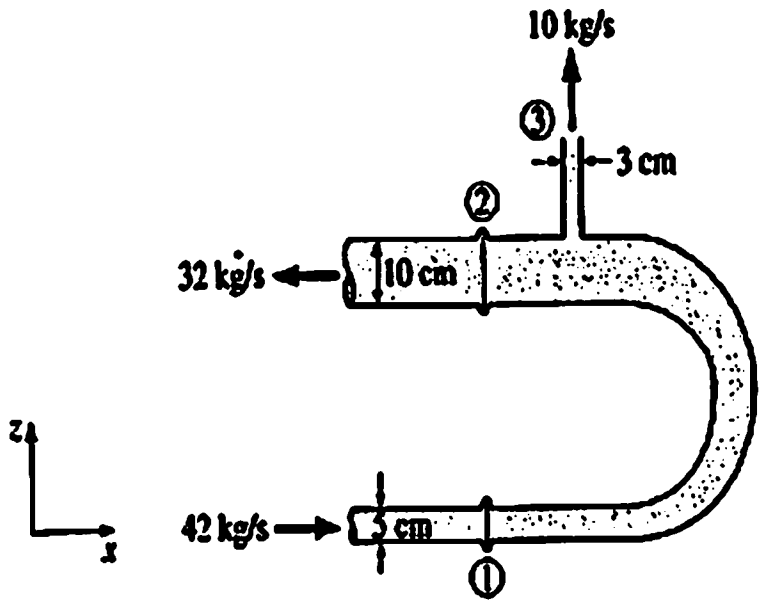


FIGURE P13–54

We don't have the solution to this problem yet.

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Ask an expert

Question: 13.055P

Indiana Jones needs to ascend a 10-m-high building. There is a large hose filled with pressurized water hanging down from the building top. He builds a square platform and mounts four 4-cm-diameter nozzles pointing down at each corner. By connecting hose branches, a water jet with 15-m/s velocity can be produced from each nozzle. Jones, the platform, and the nozzles have a combined mass of 150 kg. Determine (*a*) the minimum water jet velocity needed to raise the system, (*b*) how long it takes for the system to rise 10 m when the water jet velocity is 18 m/s and the velocity of the platform at that moment, and (*c*) how much higher will the momentum raise Jones if he shuts off the water at the moment the platform reaches 10 m above the ground. How much time does he have to jump from the platform to the roof?

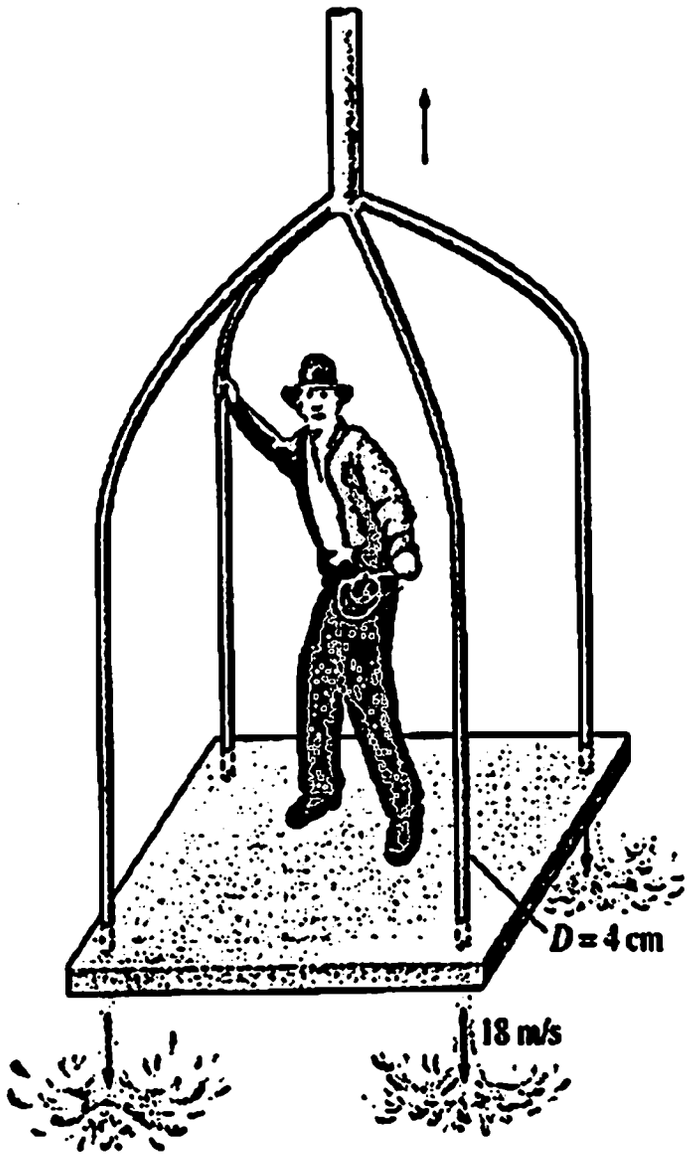


FIGURE P13–55

We don't have the solution to this problem yet.

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Ask an expert

Question: 13.056P

An engineering student considers using a fan as a levitation demonstration. She plans to face the box-enclosed fan so the air blast is directed face down through a 3-ft-diameter blade span area. The system weighs 5 lbf, and the student will secure the system from rotating. By increasing the power to the fan, she plans to increase the blade rpm and air exit velocity until the exhaust provides sufficient upward force to cause the box fan to hover in the air. Determine (*a*) the air exit velocity to produce 5 lbf, (*b*) the volumetric flow rate needed, and (*c*) the minimum mechanical power that must be supplied to the airstream. Take the air density to be 0.078 lbm/ft3.

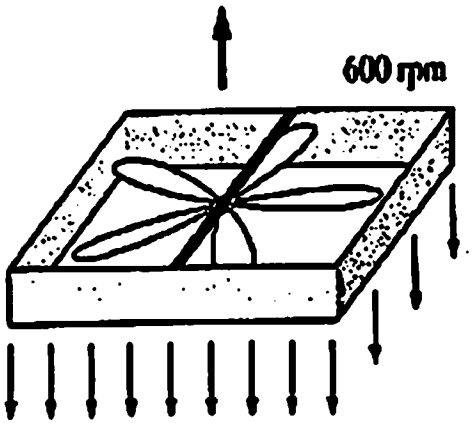


FIGURE P13–56E

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 13.057P

An orbiting satellite has a mass of 5000 kg and is traveling at a constant velocity of *V*0. To alter its orbit, an attached rocket discharges 100 kg of gases from the reaction Of solid fuel at a speed of 3000 m/s relative to the satellite in a direction opposite *V*0. The fuel discharge rate is constant for 2 s. Determine (*a*) the thrust exerted on the satellite, (*b*) the acceleration of the satellite during this 2-s period, and (*c*) the change of velocity of the satellite during this time period.

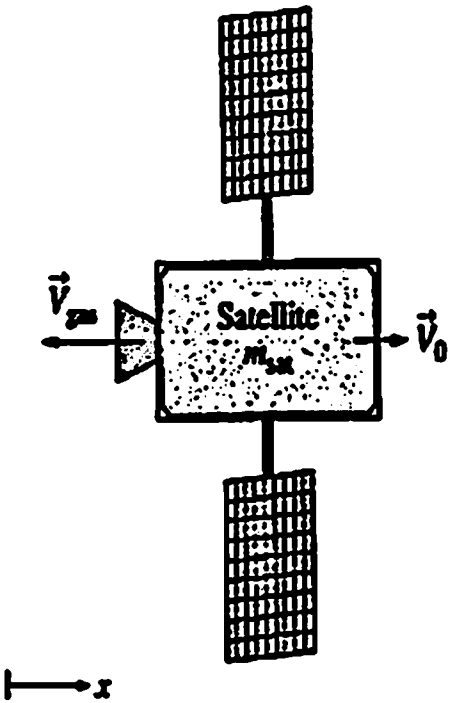


FIGURE P13–57

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 13.058P

A 7-cm diameter vertical water jet is injected upwards by a nozzle at a speed of 15 m/s. Determine the maximum weight of a flat plate that can be supported by this water jet at a height of 2 m from the nozzle.

Step 1 of 5

Find the time to reach height 2m from the nozzle using the equation.

$$s = ut - \frac{1}{2}gt^2$$

Here, u is the initial velocity of the jet, h is the height from the nozzle, and g is the acceleration due to gravity.

Substitute $15\ \text{m/s}$ for u, $9.81\ \text{m/s}^2$ for g, and 2 m for s.

$$s = ut - \frac{1}{2}gt^2$$

$$2 = 15t - \frac{1}{2}\times 9.81 \times t^2$$

$$4.9t^2 - 15t + 2 = 0$$

$$t = 0.139\ \text{s}$$

Step 2 of 5

Find the velocity at height 2m from the nozzle using the equation.

$$v = u - gt$$

Substitute $15\ \text{m/s}$ for u and $9.81\ \text{m/s}^2$ for g.

$$v = u - gt$$
$$= 15 - 9.81\times 0.139$$
$$= 13.63\ \text{m/s}$$

Find the area of the jet using the equation.

$$A = \frac{\pi}{4}d^2$$

Here, d is the diameter of the jet.

Substitute $0.07\ \text{m}$ for d.

$$A = \frac{\pi}{4}d^2$$
$$= \frac{\pi}{4}(0.07)^2$$
$$= 0.00384\ \text{m}^2$$

Step 3 of 5

Find the volume flow rate of water using the continuity equation.

$$Q = Av$$

Here, area of the jet A.

Substitute $0.00384\ \text{m}^2$ for A and $13.63\ \text{m/s}$ for v.

$$Q = Av$$
$$= 0.00384\times 13.63$$
$$= 0.05233\ \text{m}^3/\text{s}$$

Find the mass flow rate of water using the equation.

$$\dot{m} = Q\rho$$

Here, density of the water is ρ.

Substitute $1000\ \text{kg/m}^3$ for ρ.

$$\dot{m} = Q\rho$$
$$= 0.05233\times 1000$$
$$= 52.33\ \text{kg/s}$$

Step 4 of 5

Apply the momentum equation with external force for steady flow,

$$\sum F = \sum_{out} \dot{m}V - \sum_{in} \dot{m}V$$
$$W = \dot{m}(V_{out}) - \dot{m}V_{in}$$

Here, W is the weight of the flat plate, V_{out} is the velocity after striking, and V_{in} is the velocity before striking.

Substitute $13.63\ \text{m/s}$ for $(V_{out} = v)$, 0 for V_{in} and $52.33\ \text{kg/s}$ for $\dot{m}$.

$$W = \dot{m}(V_{out}) - \dot{m}V_{in}$$
$$= 52.33\times 13.63 - 52.33\times 0$$
$$= 52.33\times 13.63$$
$$= 713.2579\ \text{N}$$

Therefore, the weight of the flat plate is $\boxed{713.2579\ \text{N}}$.

Step 5 of 5

Find the mass of the flat plate using the equation.

$$W = mg$$

Here, m is the mass of the flat plate.

Substitute $713.2579\ \text{N}$ for W.

$$W = mg$$
$$713.2579 = m\times 9.81$$
$$m = \frac{713.2579}{9.81}$$
$$= 72.7\ \text{kg}$$

Therefore, the mass of the flat plate is $\boxed{72.7\ \text{kg}}$.

Question: 13.059P

A soldier jumps from a plane and opens his parachute when his velocity reaches the terminal velocity *VT*. The parachute slows him down to his landing velocity of *VF*. After the parachute is deployed, the air resistance is proportional to the velocity squared (i.e., $F = kV2$). The soldier, his parachute, and his gear have a total mass of *m*. Show that $k = mg/V_F^2$ and develop a relation for the soldier's velocity after he opens the parachute at time $t = 0$.

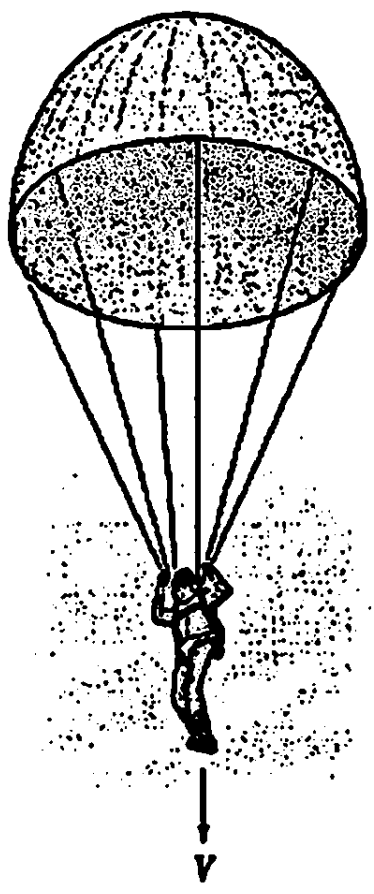


FIGURE P13–59

Step 1 of 5

As the Air resistance force is proportional to the square of the landing velocity when parachute is deployed,

$$F_{\text{air resistance}} = kV_F^2 \quad \text{..... (1)}$$

Here, constant is k, V_F is the landing velocity, and $F_{\text{air resistance}}$ is the force due to the air resistance.

Step 2 of 5

Landing velocity occurs when the air resistance is equal to the weight of the object.

Calculate the air resistance.

As the system is to be in equilibrium,

$$F_{\text{air resistance}} = W \quad \text{..... (2)}$$

Here, $F_{\text{air resistance}}$ is the force due to the air resistance and W is the total weight of the system.

Step 3 of 5

From equations (1) and (2), obtain the value of constant k.

$$mg = kV_F^2$$

$$k = \frac{mg}{V_F^2}$$

Therefore, the value of the constant is $\boxed{k = \frac{mg}{V_F^2}}$.

Step 4 of 5

Calculate the net force developed by parachute when the soldier has a velocity V.

$$F_{net} = W - F_{\text{air resistance}}$$

Here, F_{net} is the net force acting on the system.

Substitute mg for W and kV^2 for $F_{\text{air resistance}}$.

$$F_{net} = mg - kV^2$$

Substitute $\frac{mg}{V_F^2}$ for k.

$$F_{net} = mg - \frac{mg}{V_F^2} \times V^2$$

$$= mg\left(1 - \frac{V^2}{V_F^2}\right)$$

Step 5 of 5

According Newton's 2nd law

$$F_{net} = m\frac{dV}{dt}$$

$$mg\left(1 - \frac{V^2}{V_F^2}\right) = m\frac{dV}{dt}$$

Integrate to calculate the velocity,

Here $V = V_T$ when $t = 0$.

$$\frac{dV}{\left(1 - V^2/V_F^2\right)} = g\,dt$$

$$\int_{V_T}^{V} \frac{dV}{\left(V_F^2 - V^2\right)} = \frac{g}{V_F^2}\int_0^t dt$$

$$\frac{1}{2V_F}\left(\int_{V_T}^{V}\left(\frac{1}{(V_F + V)} + \frac{1}{(V_F - V)}\right)dV\right) = \frac{g\int_0^t dt}{V_F^2}$$

$$\frac{1}{2V_F}\left(\ln(V_F + V) + \ln(V_F - V)\right)_{V_T}^{V} = \frac{g(t)_0^t}{V_F^2}$$

Up on substituting the integrands, obtain the equation for the velocity of the soldier.

Therefore, velocity of soldier as a function of time is $V = V_F \frac{V_T + V_F + (V_T - V_F)e^{-2gt/V_F}}{V_T + V_F - (V_T - V_F)e^{-2gt/V_F}}$.

Question: 13.060P

Nearly frictionless vertical guide rails maintain a plate of mass *mp* in a horizontal position, such that it can slide freely in the vertical direction. A nozzle directs a water stream of area *A* against the plate underside. The water jet splatters in the plate plane, applying an upward force against the plate. The water flow rate *m* (kg/s) can be controlled. Assume that distances are short, so the velocity of the rising jet can be considered constant with height, (*a*) Determine the minimum mass flow rate $\dot{m}_{min}$ necessary to just levitate the plate and obtain a relation for the steady-state velocity of the upward moving plate for $\dot{m} > \dot{m}_{min}$ (*b*) At time *t* = 0, the plate is at rest, and the water jet with $\dot{m} > \dot{m}_{min}$ is suddenly turned on. Apply a force balance to the plate and obtain the integral that relates velocity to time (do not solve).

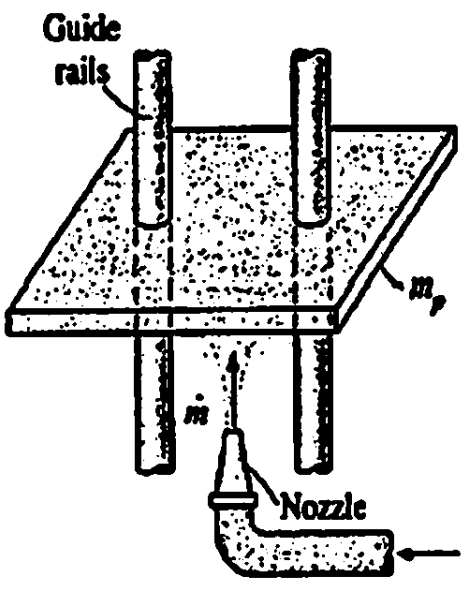


FIGURE P13–60

Step 1 of 3

a) Take the plate as the system. Recall the steady linear momentum equation relating force (*F*), the momentum flux correction factor (ß), mass flow rate ($\dot{m}$), and the velocity (*V*).

$$F = \sum_{out} \beta \dot{m} \vec{V} - \sum_{in} \beta \dot{m} \vec{V}$$

There is only one force acting on the system, the weight of the plate (*W*). There is no mass flow out but there is a source of mass flow in from the water jet.

$$W = \dot{m} V_j$$

The mass flow rate can be defined as the product of the density of water (*?*), the cross-sectional area of the jet (*A*), and the water velocity (V_J).

$$\dot{m} = \rho A V_j$$

$$V_j = \frac{\dot{m}}{\rho A} \quad \text{..... (1)}$$

The weight of the plate is defined as the product of its mass (*m*) and the gravitational constant (*g*).

$$mg = \frac{\dot{m}^2}{\rho A}$$

$$\dot{m} = \sqrt{\rho A m g}$$

Thus, the minimum mass flow rate necessary to just levitate the plate is $\boxed{\sqrt{\rho A m g}}$

Step 2 of 3

Now consider that the plate has some constant velocity. The momentum equation is then changed to reflect the relative velocity.

$$W = \dot{m}(V_j - V)$$

Recall the definitions of the water jet velocity, the mass flow rate, and the weight and rearrange to solve for the velocity of the plate.

$$mg = \rho A (V_j - V)^2$$

$$(V_j - V)^2 = \frac{mg}{\rho A}$$

Apply root on both sides.

$$(V_j - V)^2 = \frac{mg}{\rho A}$$

$$V_j - V = \sqrt{\frac{mg}{\rho A}}$$

Substitute (1) in above equation.

$$V_j - V = \sqrt{\frac{mg}{\rho A}}$$

$$V = \frac{\dot{m}}{\rho A} - \sqrt{\frac{mg}{\rho A}}$$

Thus, the relation for the steady-state velocity of the upward moving plate is $\boxed{\frac{\dot{m}}{\rho A} - \sqrt{\frac{mg}{\rho A}}}$

Step 3 of 3

b) Now consider a force balance on the plate. There are two forces acting on the plate, one from the momentum impulse and one from the weight. These two forces cause an acceleration of the plate (*a*).

$$\dot{m}(V_j - V) - W = ma$$

$$a = \frac{\rho A (V_j - V)^2}{m} - g$$

$$\frac{mdV}{\rho A (V_j - V)^2 - mg} = dt$$

Integrate both sides.

$$\int_0^V \frac{mdV}{\rho A (V_j - V)^2 - mg} = \int_0^t dt$$

$$t = \int_0^V \frac{mdV}{\rho A (V_j - V)^2 - mg}$$

Therefore, the integral that relates velocity to time is $\boxed{t = \int_0^V \frac{mdV}{\rho A (V_j - V)^2 - mg}}$

CHAPTER 14
SOLUTIONS

Question: 14.001P

What is hydraulic diameter? How is it defined? What is it equal to for a circular pipe of diameter D?

Step 1 of 1

Hydraulic diameter is defined as it is ratio of cross sectional area of the pipe to wetted perimeter of the pipe. This parameter useful to determined the Reynolds number for laminar or turbulent flow.

$$D_h = \frac{4A_c}{p}$$

Where,

A_c is the cross sectional area of the pipe

p is the perimeter of the pipe

For circular pipe:

$$D_{hc} = \frac{4A_c}{P}$$

$$= \frac{4\left(\frac{\pi}{4}D_c^2\right)}{\pi D_c}$$

$$= D_c$$

Hence, for circular pipe hydraulic diameter is reduced to diameter of the circular pipe.

Question: 14.002P

How is the hydrodynamic entry length defined for flow in a pipe? Is the entry length longer in laminar or turbulent flow?

Step 1 of 1

The region from the inlet of pipe to the point, at which the velocity profile is fully developed (where the boundary layer merges at the centerline) is called hydrodynamic entry region. The length of the hydrodynamic entry region is called hydrodynamic entry length. It is denoted by L_h

The entry length is longer in laminar flow when compared to the turbulent flow. But if the Reynolds number is very low, the boundary length will also be low.

Question: 14.003P

Why are liquids usually transported in circular pipes?

Step 1 of 1

The pipes with circular cross section can withstand large pressure differences between the inside and the outside without undergoing any distortion. Also the circular shape contains the largest cross sectional area to perimeter ratio, and less frictional losses.

Question: 14.004P

What is the physical significance of the Reynolds number? How is it defined for (*a*) flow in a circular pipe of inner diameter *D* and (*b*) flow in a rectangular duct of cross section *a* × *b*?

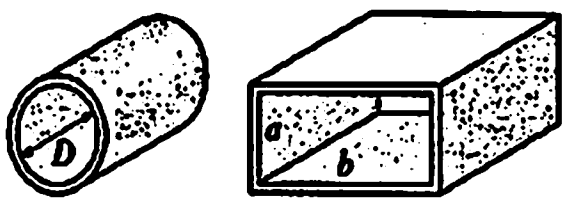


FIGURE P14–4C

Step 1 of 4

Reynolds number is the ratio of the inertial forces to viscous forces in the fluid. The flow regime depends on the Reynolds number.

The Reynolds number is given by $\mathrm{Re} = \dfrac{VD_h}{\nu}$

Where

V is the velocity of fluid

D_h is the hydraulic diameter and

ν is the kinematic viscosity

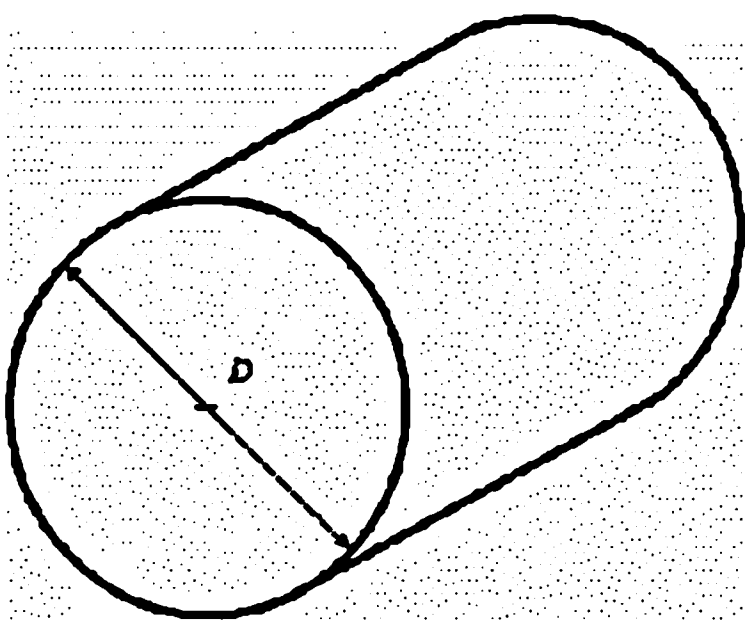
Step 2 of 4

The inertial forces in a fluid are proportional to the fluid density and square of the fluid velocity. For large Reynolds number, the inertial forces are large relative to the viscous forces. So the viscous forces cannot prevent the random and rapid fluctuations of the fluid. Therefore, the flow will be turbulent.

For small or moderate Reynolds number, the viscous forces are larger than the inertial forces. Thus it suppress the fluctuations of the fluid thereby keep the fluid inline. Thus, the flow will be laminar.

Step 3 of 4

a) Consider the flow in a circular pipe of diameter D.

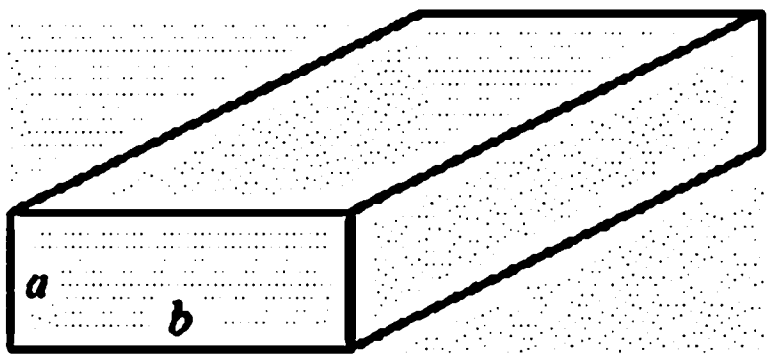


For circular pipe, the hydraulic diameter, $D_h = D$

Then, the Reynolds number, $\boxed{\mathrm{Re} = \dfrac{VD}{\nu}}$

Step 4 of 4

b) Consider the flow in a rectangular duct of cross section $a \times b$



For rectangular duct, the hydraulic diameter,

$$D_h = \frac{2ab}{a+b}$$

Then,

$$\mathrm{Re} = \frac{VD_h}{\nu}$$

$$= \frac{V\left(\dfrac{2ab}{a+b}\right)}{\nu}$$

$$= \frac{V}{\nu}\left(\frac{2ab}{a+b}\right)$$

Therefore, the Reynolds number, $\boxed{\mathrm{Re} = \dfrac{V}{\nu}\left(\dfrac{2ab}{a+b}\right)}$

Question: 14.005P

Consider a person walking first in air and then in water at the same speed. For which motion will the Reynolds number be higher?

Step 1 of 3

Let (1) and (2) denote the inlet and outlet of the manometer respectively.

The area of section (1) is more than the area of section (2).

$$A_1 > A_2$$

Hence, the velocity of section (2) is more than the velocity of section (1).

$$V_2 > V_1$$

Write the Bernoulli's equation.

$$\frac{p}{\rho}+\frac{V^2}{2}+gz=C$$

Here, p is the pressure, ρ is the density of the working fluid, V is the velocity of flow, g is the acceleration due to gravity, z is the elevation and C is a constant.

Step 2 of 3

Write the Bernoulli's equation for the inlet and the outlet.

$$\frac{p_1}{\rho}+\frac{V_1^2}{2}+gz_1=\frac{p_2}{\rho}+\frac{V_2^2}{2}+gz_2=C$$

The elevations are equal for both sections.

$$z_1 = z_2$$

$$\frac{p_1}{\rho}+\frac{V_1^2}{2}=\frac{p_2}{\rho}+\frac{V_2^2}{2}=C$$

Since $V_2 > V_1$, for the constant to hold good:

$$p_1 > p_2$$

Therefore, figure *a* is correct.

Step 3 of 3

If the flow is reversed, the conditions pertaining to the area and the velocity remain the same. Only the direction of flow changes.

Therefore, figure *a* is correct.

Question: 14.006P

Show that the Reynolds number for flow in a circular pipe of diameter D can be expressed as $Re = 4\dot{m}/(\pi D\mu)$.

Step 1 of 3

The diameter of the circular pipe is D.

The cross-sectional flow area $A_c = \frac{\pi}{4}D^2$

The Reynolds number for a circular tube is given by $Re = \frac{\rho VD}{\mu}$ .....(1)

Step 2 of 3

The average velocity is given by $V = \frac{\dot{m}}{\rho A_c}$

Where,

$\dot{m}$ is the mass flow rate and

ρ is the density of the fluid

Substitute the known expressions.

$$V = \frac{\dot{m}}{\rho\left(\frac{\pi}{4}D^2\right)}$$

$$= \frac{4\dot{m}}{\pi\rho D^2} \qquad \text{.....(2)}$$

Step 3 of 3

Now substituting the value of velocity from equation (2) in (1) we get

$$Re = \frac{\rho\left(\frac{4\dot{m}}{\pi\rho D^2}\right)D}{\mu}$$

$$= \frac{4\dot{m}}{\pi D\mu}$$

Therefore, the Reynolds number for flow in a circular pipe is $\boxed{Re = \frac{4\dot{m}}{\pi D\mu}}$

Question: 14.007P

Which fluid at room temperature requires a larger pump to flow at a specified velocity in a given pipe: water or engine oil? Why?

Step 1 of 1

At room temperature, engine oil requires a larger pump than for water to flow at a specified velocity in a given pipe. This is because engine oil has more viscosity than that of water.

Question: 14.008P

Consider the flow of air and water in pipes of the same diameter, at the same temperature, and at the same mean velocity. Which flow is more likely to be turbulent? Why?

Step 1 of 3

Reynolds number is the ratio of inertial force to viscous force in the fluid.

We can write, the Reynolds number, $\mathrm{Re} = \dfrac{\rho V_{avg} D}{\mu} = \dfrac{V_{avg} D}{\nu}$,

Here, V_{avg} is the average flow velocity, D is the characteristic length of the geometry, μ is the dynamic viscosity, ρ is the density of the flowing fluid, and ν is the kinematic viscosity.

Thus, Reynolds number is inversely proportional to the kinematic viscosity, for a fluid with constant velocity and constant diameter.

Step 2 of 3

Consider the conditions at $25°\mathrm{C}$.

From the table of properties of saturated water at $25°\mathrm{C}$,

$$\rho_{water} = 997\ \mathrm{kg/m^3}$$
$$\mu_{water} = 0.891 \times 10^{-3}\ \mathrm{kg/m \cdot s}$$

Calculate the kinematic viscosity for water by using the following relations.

$$\nu_{water} = \frac{\mu_{water}}{\rho_{water}}$$
$$= \frac{0.891 \times 10^{-3}}{997}$$
$$= 8.937 \times 10^{-7}\ \mathrm{m^2/s}$$

Step 3 of 3

And from the table of properties of air at 1 atm pressure and $25°\mathrm{C}$, we get $\nu_{air} = 1.562 \times 10^{-5}\ \mathrm{m^2/s}$.

Here, water has the smallest kinematic viscosity. So, motion of water has the highest Reynolds number. Therefore, the water flow is more likely to be turbulent.

The density and the dynamic viscosity of water are greater than that of air. But the density is much greater when compared to the dynamic viscosity. So, the kinematic viscosity of water will be smaller than that of air.

Question: 14.009P

Consider laminar flow in a circular pipe. Is the wall shear stress *tw* higher near the inlet of the pipe or near the exit? Why? What would your response be if the flow were turbulent?

Step 1 of 1

For laminar uniform flow in a circular pipe, the highest wall shear stress will be at the inlet of the tube, where the boundary layer thickness is almost zero.

Also for the turbulent uniform flow, the highest wall shear stress will be at the inlet of the tube.

Question: 14.010P

How does surface roughness affect the pressure drop in a pipe flow is turbulent? What would your response be if the flow were laminar?

Step 1 of 1

The pressure drop in turbulent flow is

$$\Delta P = f \frac{L}{D} \frac{\rho V_{avg}^2}{2}$$

$\Delta P \propto f$

Where,

V_{avg} is the average flow velocity,

D is the characteristic length of the geometry,

ρ is the density of the flowing fluid and

L is the length of the pipe

f is friction factor

Here pressure drop is directly proportional to the friction factor and friction factor is more for rough surfaced pipes when compared to smooth surface.
Therefore, when the surface roughness is more the pressure drop will also be more.

In case of laminar flow effect of surface roughness on the friction factor is negligible and thus the pressure drop is also negligible.

Question: 14.011P

Shown here is a cool picture of water being released at 300,000 gallons per second in the spring of 2008. This was part of a revitalization effort for the ecosystem of the Grand Canyon and the Colorado River. Estimate the Reynolds number of the pipe flow. Is it laminar or turbulent? (Hint: For a length scale, approximate the height of the man directly above the pipe to be 6 ft.)

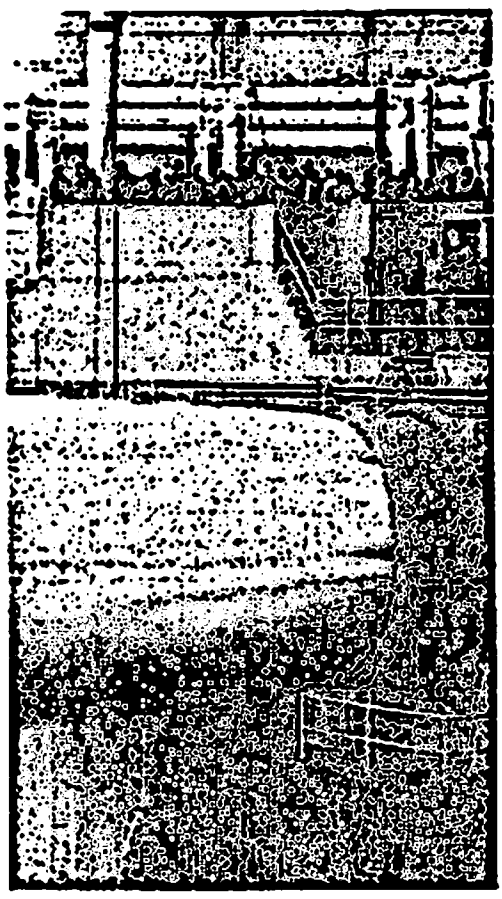


FIGURE P14–11E

Photo courtesy of Don Becker, U.S. Geological Survey.

Step 1 of 4

Given data:

The flow rate of the water in pipe, $\dot{V} = 300{,}000$ gal/s

The height of the man, $h = 6$ ft

Step 2 of 4

The diameter of the pipe is nearly 2.5 times the height of man. Hence

$$D = 1.5h$$
$$= 1.5 \times 6$$
$$= 9 \text{ ft}$$

The area of the pipe is

$$A = \frac{\pi}{4} D^2$$
$$= \frac{\pi}{4} (9 \text{ ft})^2$$
$$= 63.62 \text{ ft}^2$$

Step 3 of 4

The flow rate of the water in pipe,

$$\dot{V} = 300{,}000 \text{ gal/s}$$
$$= 300{,}000 \text{ gal/s} \left(\frac{0.13368 \text{ ft}^3/\text{s}}{1 \text{ gal/s}} \right)$$
$$= 40104 \text{ ft}^3/\text{s}$$

The velocity of the flow in pipe is

$$V = \frac{\dot{V}}{A}$$
$$= \frac{300{,}000 \text{ gal/s}}{63.62 \text{ ft}^2}$$
$$= \frac{40104 \text{ ft}^3/\text{s}}{63.62 \text{ ft}^2}$$
$$= 630.37 \text{ ft/s}$$

Step 4 of 4

From the properties table of water,

$$\rho = 62.4 \text{ lbm/ft}^3$$
$$\mu = 8.781 \times 10^{-4} \text{ lbm/ft} \cdot \text{s}$$

The Reynolds number of the given flow is

$$\text{Re} = \frac{\rho V D}{\mu}$$
$$= \frac{62.4 \times 630.37 \times 9}{8.781 \times 10^{-4}}$$
$$= 403 \times 10^6$$

Hence the estimated Reynolds number is $\boxed{\text{Re} = 403 \times 10^6}$

As the Reynolds number is greater than 4000, the flow is turbulent.

Question: 14.012P

Someone claims that the shear stress at the center of a circular pipe during fully developed laminar flow is zero. Do you agree with this claim? Explain.

Step 1 of 1

The shear stress is proportional to the velocity gradient. At the center of the pipe, the velocity gradient is zero. Thus, the shear stress will be zero at the center of the pipe. Therefore, the claim that "the shear stress at the center of a circular pipe during fully developed laminar flow is zero" is correct.

Question: 14.013P

Someone claims that in fully developed turbulent flow in a pipe, the shear stress is a maximum at the pipe wall. Do you agree with this claim? Explain.

Step 1 of 1

The shear stress is proportional to the velocity gradient. At the pipe wall, the velocity gradient is maximum. Thus, the shear stress will maximum at the pipe wall. Therefore, the claim that " in fully developed turbulent flow in a pipe, the shear stress is maximum at the pipe wall" is zero correct.

Question: 14.014P

How does the wall shear stress *tw* vary along the flow direction in the fully developed region in (*a*) laminar flow and (*b*) turbulent flow?

Step 1 of 2

a) For laminar flow, the wall shear stress τ_w remains constant along the flow direction in the fully developed region.

Step 2 of 2

b) For turbulent flow, the wall shear stress τ_w remains constant along the flow direction in the fully developed region.

Question: 14.015P

In the fully developed region of flow in a circular pipe, does the velocity profile change in the flow direction?

Step 1 of 1

The thickness of the boundary layer inside a pipe increases in the flow direction until it reaches the pipe center and it fills the entire pipe. A little further downstream the velocity becomes fully developed and remains constant. It occurs when the normalized temperature profile remains unchanged as the pipe is not heated or cooled during the flow process.

Therefore in the fully developed region of flow in a circular pipe, the velocity profile does not change in the flow direction.

Question: 14.016P

How is the friction factor for flow in a pipe related to the pressure loss? How is the pressure loss related to the pumping power requirement for a given mass flow rate?

Step 1 of 2

The pressure loss for fully developed internal flow is given by $\Delta P_L = f \frac{L}{D} \frac{\rho V_{avg}^2}{2}$,

Where,
f is the Darcy friction factor,
L is the length of the pipe,
D is the diameter of the pipe,
ρ is the density of the flowing fluid,
V_{avg} is the average velocity and
$\frac{\rho V_{avg}^2}{2}$ is called the dynamic pressure

Thus, the friction factor is directly proportional to the pressure loss, for a specified fluid flowing in a particular velocity through a pipe having constant length and diameter.

Step 2 of 2

The pumping power required for the internal flow is given by $\dot{W}_{pump} = \dot{V} \Delta P_L$,

The volume flow rate is given by $\dot{V} = \frac{\dot{m}}{\rho}$,

Where, $\dot{V}$ is the volume flow rate
Where, $\dot{m}$ is the mass flow rate

Therefore, the pumping power required, $\dot{W}_{pump} = \frac{\dot{m} \Delta P_L}{\rho}$

Thus, the pressure loss is directly proportional to the pumping power required.

Question: 14.017P

Discuss whether fully developed pipe flow is one-, two-, or three-dimensional.

Step 1 of 1

In a fully developed laminar flow, the fluid particle moves along a streamline at constant axial velocity. And, there is no motion in the radial direction. Thus, we can say that the fully developed fluid flow is one dimensional.

Question:

14.018P

Consider fully developed flow in a circular pipe with negligible entrance effects. If the length of the pipe is doubled, the head loss will (*a*) double, (*b*) more than double, (*c*) less than double, (*d*) reduce by half, or (*e*) remain constant.

Step 1 of 2

The head loss for fully developed internal flow is given by $\Delta h_L = f\dfrac{L}{D}\dfrac{V^2}{2g}$

Where,
f is the Darcy friction factor,
L is the length of the pipe,
D is the diameter of the pipe,
V is the average velocity and
g is the gravitational acceleration.

Step 2 of 2

The new head loss, when the length of the pipe is doubled $(L = 2L)$

$$\begin{aligned}\Delta h_L' &= f\frac{(2L)}{D}\frac{V^2}{2g}\\ &= 2\left(f\frac{L}{D}\frac{V^2}{2g}\right)\\ &= 2\Delta h_L\end{aligned}$$

Therefore, when the length of the pipe is doubled, the head loss is also doubled.

The new head loss, when the length of the pipe is doubled $(L = 2L)$

$$\begin{aligned}\Delta h_L' &= f\frac{(2L)}{D}\frac{V^2}{2g}\\ &= 2\left(f\frac{L}{D}\frac{V^2}{2g}\right)\\ &= 2\Delta h_L\end{aligned}$$

Therefore, when the length of the pipe is doubled, the head loss is also doubled.

Question: 14.019P

Someone claims that the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in the fully developed region, multiplying it by the cross-sectional area, and dividing the result by 2. Do you agree? Explain.

Step 1 of 2

We know, the volume flow rate,

$\dot{V} = V_{avg} A_c$,

where, V_{avg} is the average flow velocity and

A_c is the cross-sectional area of the pipe.

But, the average flow velocity,

$V_{avg} = \frac{u_{max}}{2}$,

where, u_{max} is the maximum velocity of flow which is through the centerline in the fully developed region.

Step 2 of 2

Then, the volume flow rate,

$\dot{V} = \frac{u_{max}}{2} A_c$.

Therefore, the claim that "the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in a fully developed region, multiplying it by the cross sectional area, and dividing the result by 2" is correct.

Step 2 of 2

Then, the volume flow rate,

$\dot{V} = \frac{u_{max}}{2} A_c$.

Therefore, the claim that "the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in a fully developed region, multiplying it by the cross sectional area, and dividing the result by 2" is correct.

Question: 14.020P

Someone claims that the average velocity in a circular pipe in fully developed laminar flow can be determined by simply measuring the velocity at $R/2$ (midway between the wall surface and the centerline). Do you agree? Explain.

Step 1 of 2

The velocity profile in fully developed laminar flow in circular pipe is

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right),$$

where, u_{max} is the maximum velocity of flow which is through the centerline in the fully developed region.

When $r = R/2$,

$$u(R/2) = u_{max}\left(1 - \frac{(R/2)^2}{R^2}\right)$$

$$= u_{max}\left(1 - \frac{1}{4}\right)$$

$$= \frac{3}{4}u_{max}$$

Step 2 of 2

The average velocity,

$$V_{avg} = \frac{u_{max}}{2}$$

Now, $u(R/2) \neq V_{avg}$.

Therefore, the claim that "the average velocity in a circular pipe in fully developed laminar flow can be determined by simply measuring the velocity at $R/2$" is wrong.

Question: 14.021P

Consider fully developed laminar flow in a circular pipe. If the diameter of the pipe is reduced by half while the flow rate and the pipe length are held constant, the head loss will (*a*) double, (*b*) triple, (*c*) quadruple, (*d*) increase by a factor of 8, or (*e*) increase by a factor of 16.

Step 1 of 6

Given data:
The flow through the circular channel is fully developed and laminar.
The flow rate and pipe length are held constant.

Step 2 of 6

The head loss for fully developed internal flow is given by $\Delta h_L = f\frac{L}{D}\frac{V^2}{2g}$

Where,
f is the Darcy friction factor,
L is the length of the pipe,
D is the diameter of the pipe,
V is the average velocity and
g is the gravitational acceleration.

Step 3 of 6

The Reynolds number is given by $\mathrm{Re} = \frac{VD}{\nu}$,
Where, ν is the kinematic viscosity of the flowing fluid
The friction factor,

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{(VD/\nu)} = \frac{64\nu}{VD}$$

Step 4 of 6

The cross sectional flow area, $A_c = \frac{\pi}{4}D^2$
The flow rate is given by $\dot{V} = VA_c$
Then,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\left(\frac{\pi}{4}D^2\right)} = \frac{4\dot{V}}{\pi D^2}$$

Step 5 of 6

Thus, the head loss can be written as

$$\Delta h_L = \left(\frac{64\nu}{VD}\right)\frac{L}{D}\frac{V^2}{2g} = \frac{64\nu L}{D^2}\frac{V}{2g} = \frac{64\nu L}{D^2}\frac{\left(4\dot{V}/\pi D^2\right)}{2g} = \frac{128\nu L\dot{V}}{\pi D^4 g}$$

Thus, we get that the head loss is inversely proportional to the fourth power of the diameter of the pipe.

Step 6 of 6

When the diameter is reduced by half,

$$\Delta h_L' = \frac{128\nu L\dot{V}}{\pi(D/2)^4 g} = 16\left(\frac{128\nu L\dot{V}}{\pi D^4 g}\right) = 16h_L$$

Therefore, when the diameter is reduced by half the head loss will increase by a factor of 16.

Question: 14.022P

What is the physical mechanism that causes the friction factor to be higher in turbulent flow?

Step 1 of 1

The fluid in turbulent flow is subjected to enhanced mixing due to swirling eddies. This results in increased mass transfer or quicker transport of mass to all regions. All this would increase the friction and hence the turbulent flow is associated with much higher values of friction factor.

Question: 14.023P

What is turbulent viscosity? What causes it?

Step 1 of 1

Turbulent viscosity is the viscosity due to the momentum transport by the turbulent eddies. It is the property of the flow and is denoted by μ_t

The relation between turbulent shear stress (τ_{turb}) and the turbulent viscosity (μ_t) is

$$\tau_{turb} = -\rho\overline{u'v'}$$

$$= \mu_t \frac{\partial \bar{u}}{\partial y}$$

where, u' and v' are the fluctuating components of velocity,

ρ is the density and

$\bar{u}$ is the mean velocity in the direction of flow.

Question: 14.024P

Consider fully developed laminar flow in a circular pipe. If the viscosity of the fluid is reduced by half by heating while the flow rate is held constant, how does the head loss change?

Step 1 of 4

Obtain the relation for the pressure drop for a fully developed laminar flow in a circular pipe.

$$\Delta P = \frac{8\mu L V_{avg}}{R^2} = \frac{32\mu L V_{avg}}{D^2} \quad \text{..... (1)}$$

Here, L is the length is the pipe, μ is the dynamic viscosity, V_m is the mean velocity, and D is the diameter.

Since the area of the pipe remains constant, the length and diameter of the pipe can be considered as constants in the equation (1).

$$L = D = C$$

Step 2 of 4

Write the equation for flow rate.

$$Q = AV_{avg}$$

Here, A is the area of the pipe.

Since area of the pipe remains same, the flow arte equation becomes as follows:

$$Q \propto V_{avg}$$

Since, flow rate is kept constant, thus the mean velocity becomes constant. As the flow rate is directly proportional to mean velocity. Thus,

$$V_{avg} = C$$

Step 3 of 4

Hence pressure is directly proportional to viscosity.

$$\Delta P \propto \mu$$

Write the equation for head loss as follows:

$$h_L = \frac{\Delta P}{\rho g} \quad \text{..... (2)}$$

Here, ρ is the density and g is the acceleration due to gravity.

From the relation it is clear that, head loss is proportional to viscosity.

$$h_L \propto \mu$$

Hence if the viscosity is reduced by half then head loss is also reduced by half. It can be proved as follows:

Substitute equation (1) for ΔP in equation (2).

$$h_L = \frac{\Delta P}{\rho g} = \frac{\left(\frac{32\mu L V_{avg}}{D^2}\right)}{\rho g} \quad \text{..... (2)} = \frac{32\mu L V_{avg}}{\rho g D^2}$$

Step 4 of 4

When the viscosity of the fluid is reduced by half, obtain the relation for new head loss as follows:

Substitute $\left(\frac{\mu}{2}\right)$ for μ in equation (2).

$$\Delta h_L' = \frac{32(\mu/2) L V_{avg}}{\rho g D^2} = \frac{1}{2}\left(\frac{32\mu L V_{avg}}{\rho g D^2}\right)$$

Substitute h_L for $\left(\frac{32\mu L V_{avg}}{\rho g D^2}\right)$.

$$\Delta h_L' = \frac{1}{2} h_L$$

Therefore, when the viscosity of the fluid is reduced by half, the head loss will be reduced by half.

Question: 14.025P

How is head loss related to pressure loss? For a given fluid, explain how you would convert head loss to pressure loss.

Step 1 of 2

The relation between the head loss (Δh_L) and the pressure loss (ΔP_L) is given by

$$\Delta h_L = \frac{\Delta P_L}{\rho g},$$

Where,

ρ is the density of the flowing fluid and

g is the gravitational acceleration.

Step 2 of 2

Then, the pressure loss,

$$\Delta P_L = \rho g \Delta h_L$$

Therefore, the pressure loss is obtained by multiplying the head loss with the density of the fluid and the gravitational acceleration.

Question: 14.026P

Consider laminar flow of air in a circular pipe with perfectly smooth surfaces. Do you think the friction factor for this flow is zero? Explain.

Step 1 of 1

For laminar flow in a circular pipe, the friction factor is given by the formula

$$f = \frac{64\mu}{\rho D V_{avg}} \qquad \ldots\ldots (1)$$

The relation for Reynolds number is given by the formula

$$\mathrm{Re} = \frac{\rho D V_{avg}}{\mu} \qquad \ldots\ldots (2)$$

Hence from equations (1) and (2) , we get the relation for friction factor as below.

$$f = \frac{64}{\mathrm{Re}}$$

Thus, the friction factor is independent of the surface roughness of the pipe. And hence the friction factor for perfectly smooth surfaces is not zero.

Question: 14.027P

Explain why the friction factor is independent of the Reynolds number at very large Reynolds numbers.

Step 1 of 1

Friction factor depends on Reynolds number to some extent only, but for large values of Reynolds number, it is not true. This can be explained as follows:

Friction factor is a function of Reynolds number and relative roughness.

$$f = \text{func}\left(\text{Re}, \frac{\varepsilon}{D}\right)$$

From FIGURE 14-22, at very large Reynolds numbers, the friction factor curves in the Moody chart are all almost horizontal. Friction factor here depends only on the relative roughness but not the Reynolds numbers.

Therefore, it can be concluded that friction factor is independent of Reynolds number for larger value of Reynolds numbers.

Question: 14.028P

Water at 10°C (ρ = 999.7 kg/m3 and μ = 1.307 × 10-3 kg/m-s) is flowing steadily in a 0.20-cm-diameter, 15-long pipe at an average velocity of 1.2 m/s. Determine (*a*) the pressure drop, (*b*) the head loss, and (*c*) the pumping power requirement to overcome this pressure drop.

Step 1 of 6

Given data:

Water is flowing at 10°C through a pipe.

The density, $\rho = 999.7\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 1.307 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$

The length of the pipe, $L = 15\ \text{m}$

The diameter of pipe, $D = 0.20\ \text{cm} = 2 \times 10^{-3}\ \text{m}$

The average velocity, $V = 1.2\ \text{m/s}$

Step 2 of 6

The flow is assumed to be steady, fully developed and incompressible.

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(999.7\ \text{kg/m}^3)(1.2\ \text{m/s})(2 \times 10^{-3}\ \text{m})}{1.307 \times 10^{-3}\ \text{kg/m} \cdot \text{s}}$$

$$= 1835.7$$

Here, Re < 2300 Therefore, the flow is laminar.

Step 3 of 6

The friction factor,

$$f = \frac{64}{\text{Re}}$$

$$= \frac{64}{1835.7}$$

$$= 0.03486$$

Step 4 of 6

a) The pressure drop,

$$\Delta P_L = f \frac{L}{D} \frac{\rho V^2}{2}$$

$$= 0.03486 \left[\frac{15\ \text{m}}{2 \times 10^{-3}\ \text{m}}\right]\left[\frac{(999.7\ \text{kg/m}^3)(1.2\ \text{m/s})^2}{2}\right]$$

$$= 188.187 \times 10^3\ \text{Pa}$$

$$= 188.187\ \text{kPa}$$

Therefore, the pressure drop is $\boxed{188.187\ \text{kPa}}$

Step 5 of 6

b) The head loss is given by $h_L = \frac{\Delta P_L}{\rho g}$,

Where, g is the gravitational acceleration.

Substituting the known values.

$$h_L = \frac{188.187 \times 10^3\ \text{Pa}}{(999.7\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}$$

$$= 19.19\ \text{m}$$

Therefore, the head loss, $\boxed{h_L = 19.2\ \text{m}}$

Step 6 of 6

c) The cross sectional flow area,

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(2 \times 10^{-3}\ \text{m})^2$$

$$= 3.1416 \times 10^{-6}\ \text{m}^2$$

The flow rate,

$$\dot{V} = V A_c$$

$$= (1.2\ \text{m/s})(3.1416 \times 10^{-6}\ \text{m}^2)$$

$$= 3.77 \times 10^{-6}\ \text{m}^3\text{/s}$$

Question: 14.029P

Water at 15°C (ρ = 999.1 kg/m3 and μ = 1.138 X 10-3 kg/m-s) is flowing steadily in a 30-m-long and 5-cm-diameter horizontal pipe made of stainless steel at a rate of 9 L/s. Determine (*a*) the pressure drop, (*b*) the head loss, and (*c*) the pumping power requirement to overcome this pressure drop.

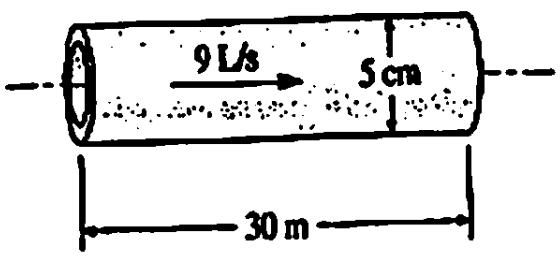


FIGURE P14–29

Step 1 of 7

Given data:

The temperature of the water in pipe $T = 15^{\circ}\text{C}$

The density of water $\rho = 999.1\,\text{kg/m}^3$

The dynamic viscosity of water $\mu = 1.138 \times 10^{-3}\,\text{kg/m}\cdot\text{s}$

The length of the pipe $L = 30\,\text{m}$

The diameter of pipe $D = 5\,\text{cm} = 5 \times 10^{-2}\,\text{m}$

The volumetric flow rate of water $\dot{V} = 9\,\text{L/s} = 9 \times 10^{-3}\,\text{m}^3/\text{s}$

Step 2 of 7

The cross sectional area of pipe

$$A_c = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}\left(5 \times 10^{-2}\,\text{m}\right)^2$$

$$= 1.963 \times 10^{-3}\,\text{m}^2$$

The average flow velocity,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{9 \times 10^{-3}\,\text{m}^3/\text{s}}{1.963 \times 10^{-3}\,\text{m}^2}$$

$$= 4.585\,\text{m/s}$$

Step 3 of 7

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{\left(999.1\,\text{kg/m}^3\right)\left(4.585\,\text{m/s}\right)\left(5 \times 10^{-2}\,\text{m}\right)}{1.138 \times 10^{-3}\,\text{kg/m}\cdot\text{s}}$$

$$= 2.0127 \times 10^5$$

Here, Re > 4000. Therefore, the flow is turbulent.

Step 4 of 7

For turbulent flow, the Colebrook's equation is given by

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

where, f is the friction factor.

ε is the surface roughness.

For stainless steel, the surface roughness is $\varepsilon = 0.002\,\text{mm} = 2 \times 10^{-6}\,\text{m}$

By substituting the known values in Colebrook's equation

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2 \times 10^{-6}\,\text{m}/5 \times 10^{-2}\,\text{m}}{3.7} + \frac{2.51}{\left(2.0127 \times 10^5\right)\sqrt{f}}\right)$$

$$= -2.0 \log\left(1.0811 \times 10^{-5} + \frac{1.2421 \times 10^{-5}}{\sqrt{f}}\right)$$

By solving the equation for f, we get $f = 0.0159$

Step 5 of 7

(a) The pressure drop in flow

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2}$$

$$= 0.0159\left[\frac{30\,\text{m}}{5 \times 10^{-2}\,\text{m}}\right]\left[\frac{\left(999.1\,\text{kg/m}^3\right)\left(4.585\,\text{m/s}\right)^2}{2}\right]$$

$$= 100.185 \times 10^3\,\text{Pa}$$

$$= 100.185\,\text{kPa}$$

Therefore, the pressure drop ΔP_L is $\boxed{100.185\,\text{kPa}}$

Step 6 of 7

(b) The head loss in the flow

$$h_L = \frac{\Delta P_L}{\rho g}$$

$$= \frac{100.185 \times 10^3\,\text{Pa}}{\left(999.1\,\text{kg/m}^3\right)\left(9.81\,\text{m/s}^2\right)}$$

$$= 10.22\,\text{m}$$

Therefore, the head loss h_L in pipe is $\boxed{10.22\,\text{m}}$

Step 7 of 7

(c) The required pumping power

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P_L$$

$$= \left(9 \times 10^{-3}\,\text{m}^3/\text{s}\right)\left(100.185 \times 10^3\,\text{Pa}\right)$$

$$= 901.665\,\text{W}$$

Therefore, the power $\dot{W}_{\text{pump}}$ required to overcome the pressure loss is $\boxed{901.665\,\text{W}}$

Question: 14.030P

Heated air at 1 atm and 100°F is to be transported in a 400-ft-long circular plastic duct at a rate of 12 ft3/s. If the head loss in the pipe is not to exceed 50 ft, determine the minimum diameter of the duct.

Step 1 of 2

Write the velocity profile equation for fully developed laminar flow.

$$u(r)=4\left(1-\frac{r^2}{R^2}\right)$$

Here, r is the element radius and R is the radius of profile.

Compare the profile equation with fully developed laminar flow profile equation (8.17).

$$2V_{avg}=4$$

$$V_{avg}=\frac{4}{2}$$

$$=2\,\text{m/s}$$

Therefore, the average velocity is $\boxed{2\,\text{m/s}}$.

Step 2 of 2

Find the maximum velocity by using the following formula.

$$u_{max}=2V_{avg}$$

Substitute 2 m/s for V_{avg} .

$$u_{max}=2(2)$$

$$=4\,\text{m/s}$$

Therefore, the maximum velocity is $\boxed{4\,\text{m/s}}$.

Find the volume flow rate by using the following formula.

$$\dot{V}=V_{avg}A_c$$

$$=V_{avg}\left(\pi R^2\right)$$

Substitute 2 m/s for V_{avg} and 2 cm for R.

$$\dot{V}=(2)\left(\pi\left(2\,\text{cm}\times10^{-2}\,\frac{\text{m}}{\text{cm}}\right)^2\right)$$

$$=0.0025143\,\text{m}^3/\text{s}$$

Therefore, the volume flow rate is $\boxed{0.0025143\,\text{m}^3/\text{s}}$.

Question: 14.031P

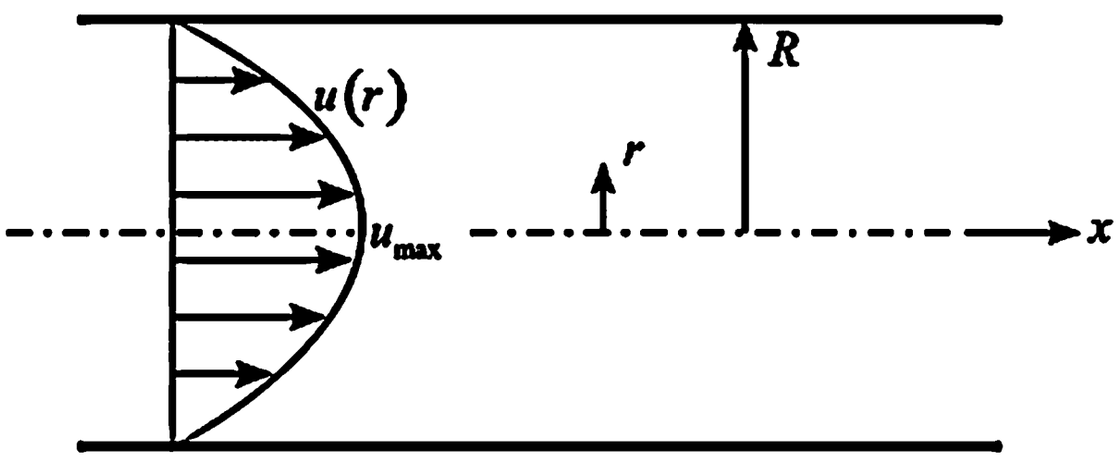
In fully developed laminar flow in a circular pipe, the velocity at $R/2$ (midway between the wall surface and the centerline) is measured to be 8 m/s. Determine the velocity at the center of the pipe.

Step 1 of 3

Given data:

The velocity at a position $R/2$, $u(R/2) = 8\ \text{m/s}$.

Velocity profile of the flow is as shown below.



Step 2 of 3

The velocity profile of the fully developed laminar flow for a circular channel is given by

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right) \qquad \ldots\ldots (1)$$

where, r is the radius of the differential fluid element $r = (R/2)$

R is the radius of the pipe and

u_{max} is the maximum velocity, at the center of the pipe.

Step 3 of 3

By substituting the known values in equation (1), we get

$$u(r) = u_{max}\left(1 - \frac{(R/2)^2}{R^2}\right)$$

$$u(R/2) = u_{max}\left(1 - \frac{(R/2)^2}{R^2}\right)$$

$$8\ \text{m/s} = u_{max}\left(1 - \frac{(R/2)^2}{R^2}\right)$$

$$8\ \text{m/s} = \frac{3}{4}(u_{max})$$

$$u_{max} = \frac{4}{3}(8\ \text{m/s})$$

$$= 10.7\ \text{m/s}$$

Therefore, the velocity at the center of the pipe is $\boxed{u_{max} = 10.7\ \text{m/s}}$

Question: 14.032P

The velocity profile in fully developed laminar flow in a circular pipe of inner radius $R = 2$ cm, in m/s, is given by $u(r) = 4(1 - r2/R2)$. Determine the average and maximum velocities in the pipe and the volume flow rate.

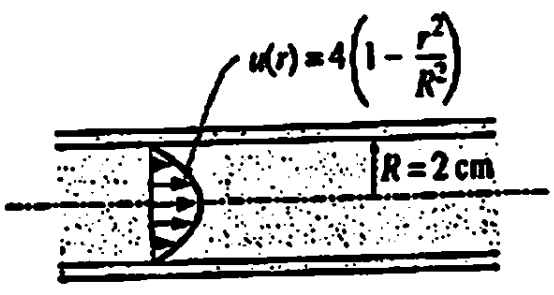


FIGURE P14–32

Step 1 of 4

Obtain the properties of air at 1 atm pressure from the "Properties of air table".

The density of air at $45°C$, $\rho = 1.109\ \text{kg/m}^3$

The dynamic viscosity at $45°C$, $\mu = 1.941 \times 10^{-5}\ \text{kg/m}\cdot\text{s}$

The kinematic viscosity at $45°C$, $\nu = 1.750 \times 10^{-5}\ \text{m}^2/\text{s}$

For glass, the roughness, $\varepsilon = 0$

Find the cross-sectional flow area between glass cover using the following formula:

$$A_c = bl$$

Here, b is the width of the solar collector and $0.03\ \text{m}$ for l.

Substitute 1 m for b and 0.03 m for b.

$$\begin{aligned} A_c &= (1\ \text{m})(0.03\ \text{m}) \\ &= 0.03\ \text{m}^2 \end{aligned}$$

Find the perimeter using the following formula:

$$p = 2(b + l)$$

Substitute 1 m for b and 0.03 m for b.

$$\begin{aligned} p &= 2(1\ \text{m} + 0.03\ \text{m}) \\ &= 2.06\ \text{m} \end{aligned}$$

Step 2 of 4

Find the hydraulic diameter using the following formula:

$$D_h = \frac{4A_c}{p}$$

Substitute $0.03\ \text{m}^2$ for A_c and 2.06 m for p.

$$\begin{aligned} D_h &= \frac{4(0.03\ \text{m}^2)}{2.06\ \text{m}} \\ &= 0.05825\ \text{m} \end{aligned}$$

Find the average velocity of air using the following formula:

$$V = \frac{\dot{V}}{A_c}$$

Here, $\dot{V}$ is rate of discharge of air.

Substitute $0.03\ \text{m}^2$ for A_c and $0.15\ \text{m}^3/\text{s}$ for $\dot{V}$.

$$\begin{aligned} V &= \frac{0.15}{0.03} \\ &= 5\ \text{m/s} \end{aligned}$$

Step 3 of 4

Find the Reynolds number using following formula:

$$\text{Re} = \frac{VD_h}{\nu}$$

Substitute 5 m/s for V, $0.05825\ \text{m}$ for D_h and $1.750 \times 10^{-5}\ \text{m}^2/\text{s}$ for ν.

$$\begin{aligned} \text{Re} &= \frac{(5)(0.05825)}{1.750 \times 10^{-5}} \\ &= 16642.86 \end{aligned}$$

$\text{Re} > 4000$, the flow is turbulent

Find the friction factor using the Colebrook equation.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Substitute 0 for ε and 16642.86 for Re.

$$\begin{aligned} \frac{1}{\sqrt{f}} &= -2.0\log\left(\frac{0/D}{3.7} + \frac{2.51}{(16642.86)\sqrt{f}}\right) \\ &= -2.0\log\left(\frac{0.1508 \times 10^{-3}}{\sqrt{f}}\right) \\ f &= 0.0271 \end{aligned}$$

Step 4 of 4

Find the pressure drop using the following formula:

$$\Delta P_L = f\frac{L}{D_h}\frac{\rho V^2}{2}$$

Substitute 0.0271 for f, 5 m for L, $0.05825\ \text{m}$ for D_h, $1.109\ \text{kg/m}^3$ for ρ, and 5 m/s for V.

$$\begin{aligned} \Delta P_L &= 0.0271\left[\frac{5}{0.05825}\right]\left[\frac{(1.109)(5)^2}{2}\right] \\ &= 32.25\ \text{Pa} \end{aligned}$$

Therefore, the pressure drop in the collector is $\boxed{32.25\ \text{Pa}}$.

Question: 14.033P

Repeat Prob. 14-32 for a pipe of inner radius 7 cm.

Step 1 of 5

Given data:

The inner radius of the circular pipe, $R = 7\text{ cm}$

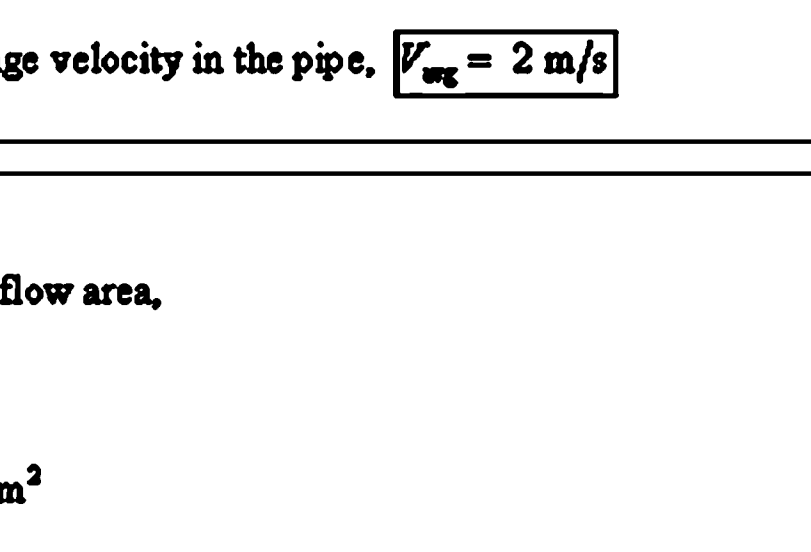
$= 0.07\text{ m}$

The velocity profile of the fully developed laminar flow for the given circular channel,

$$u(r) = 4\left(1 - \frac{r^2}{R^2}\right)$$

Step 2 of 5

Let us draw the velocity profile.



Step 3 of 5

The velocity profile of the fully developed laminar flow for a circular channel is given by

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right)$$

Where, r is the radius of the differential fluid element.

Compare the given expression and the general expression of the velocity profile of the fully developed laminar flow,

$$u_{max}\left(1 - \frac{r^2}{R^2}\right) = 4\left(1 - \frac{r^2}{R^2}\right)$$

$$u_{max} = 4\text{ m/s}$$

Therefore, the maximum velocity in the pipe, $\boxed{u_{max} = 4\text{ m/s}}$

Step 4 of 5

The average velocity,

$$V_{avg} = \frac{u_{max}}{2}$$

$$= \frac{4\text{ m/s}}{2}$$

$$= 2\text{ m/s}$$

Therefore, the average velocity in the pipe, $\boxed{V_{avg} = 2\text{ m/s}}$

Step 5 of 5

The cross-sectional flow area,

$$A_c = \pi R^2$$

$$= \pi(0.07\text{ m})^2$$

$$= 15.393 \times 10^{-3}\text{ m}^2$$

The volume flow rate,

$$\dot{V} = V_{avg} A_c$$

$$= (2\text{ m/s})(15.393 \times 10^{-3}\text{ m}^2)$$

$$= 30.786 \times 10^{-3}\text{ m}^3/\text{s}$$

Therefore, the volume flow rate, $\boxed{\dot{V} = 30.786 \times 10^{-3}\text{ m}^3/\text{s}}$

Question: 14.034P

Consider an air solar collector that is 1 m wide and 5 m long and has a constant spacing of 3 cm between the glass cover and the collector plate. Air flows at an average temperature of 45°C at a rate of 0.15 m3/s through the 1-m-wide edge of the collector along the 5-m-long passageway. Disregarding the entrance and roughness effects and the 90° bend, determine the pressure drop in the collector.

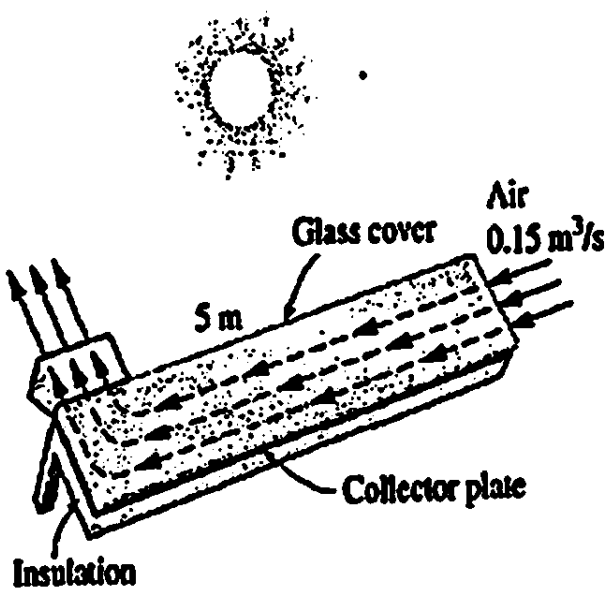


FIGURE P14–34

Step 1 of 7

Given data:

The width of the solar collector, $b = 1\text{ m}$

The length of the solar collector, $L = 5\text{ m}$

The constant spacing between the glass cover, $l = 3\text{ cm} = 0.03\text{ m}$

The temperature of the air, $T = 45°\text{C}$

The rate of discharge of air, $\dot{V} = 0.15\text{ m}^3/\text{s}$

Step 2 of 7

From the table of properties of air at 1 atm pressure,

The density of air at 45°C, $\rho = 1.109\text{ kg/m}^3$

The dynamic viscosity at 45°C, $\mu = 1.941 \times 10^{-5}\text{ kg/m}\cdot\text{s}$

The kinematic viscosity at 45°C, $\nu = 1.750 \times 10^{-5}\text{ m}^2/\text{s}$

Step 3 of 7

For glass, the roughness, $\varepsilon = 0$

The cross-sectional flow area between the glass cover,

$$A_c = bl$$
$$= (1\text{ m})(0.03\text{ m})$$
$$= 0.03\text{ m}^2$$

The perimeter,

$$p = 2(b + l)$$
$$= 2(1\text{ m} + 0.03\text{ m})$$
$$= 2.06\text{ m}$$

Step 4 of 7

The hydraulic diameter,

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4(0.03\text{ m}^2)}{2.06\text{ m}}$$
$$= 0.0582\text{ m}$$

The average velocity,

$$V = \frac{\dot{V}}{A_c}$$
$$= \frac{0.15\text{ m}^3/\text{s}}{0.03\text{ m}^2}$$
$$= 5\text{ m/s}$$

Step 5 of 7

The Reynolds number,

$$\text{Re} = \frac{VD_h}{\nu}$$
$$= \frac{(5\text{ m/s})(0.0582\text{ m})}{1.750 \times 10^{-5}\text{ m}^2/\text{s}}$$
$$= 16.629 \times 10^3$$

Since Re > 4000, the flow is turbulent

Step 6 of 7

For turbulent flow, the Colebrook equation is given by $\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$,

Where, f is the friction factor.

Substituting the known values in Colebrook equation and solving the equation for f we get,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0/D}{3.7} + \frac{2.51}{(16.629 \times 10^3)\sqrt{f}}\right)$$
$$= -2.0\log\left(\frac{0.1509 \times 10^{-3}}{\sqrt{f}}\right)$$
$$f = 0.0271$$

Step 7 of 7

The pressure drop,

$$\Delta P_L = f\frac{L}{D_h}\frac{\rho V^2}{2}$$
$$= 0.0271\left[\frac{5\text{ m}}{0.0582\text{ m}}\right]\left[\frac{(1.109\text{ kg/m}^3)(5\text{ m/s})^2}{2}\right]$$
$$= 32.27\text{ Pa}$$

Therefore, the pressure drop in the collector, $\boxed{\Delta P_L = 32.3\text{ Pa}}$

Question: 14.035P

Consider laminar flow of a fluid through a square channel with smooth surfaces. Now the average velocity of the fluid is doubled. Determine the change in the head loss of the fluid. Assume the flow regime remains unchanged.

Step 1 of 4

Write the expression for Reynolds number.

$$\mathrm{Re} = \frac{\rho V D}{\mu}$$

Here, ρ is the density of the flowing fluid, V is the average flow velocity, D is the diameter of the pipeline and μ is the dynamic viscosity

Write the expression for friction factor.

$$f = 0.184\,\mathrm{Re}^{-0.2}$$

Substitute $\frac{\rho V D}{\mu}$ for Re.

$$f = 0.184\left(\frac{\rho V D}{\mu}\right)^{-0.2}$$

Step 2 of 4

Find the head loss using the following formula:

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

Here, L is the length of the pipeline and g is the gravitational acceleration.

Substitute $0.184\left(\frac{\rho V D}{\mu}\right)^{-0.2}$ for f.

$$h_L = \left[0.184\left(\frac{\rho V D}{\mu}\right)^{-0.2}\right]\frac{L}{D}\frac{V^2}{2g} \quad \text{..... (1)}$$

$$= \frac{0.092L}{gD^{1.8}}\left(\frac{\rho}{\mu}\right)^{-0.2} V^{1.8}$$

Step 3 of 4

Find the new head loss when the velocity is doubled.

Substitute $2V$ for V in equation (1).

$$h_L' = \frac{0.092L}{gD^{1.8}}\left(\frac{\rho}{\mu}\right)^{-0.2}(2V)^{1.8}$$

$$= 3.48\left[\frac{0.092L}{gD^{1.8}}\left(\frac{\rho}{\mu}\right)^{-0.2} V^{1.8}\right]$$

$$= 3.48h_L$$

Therefore, the new head loss when the velocity of fluid velocity is doubled is $\boxed{3.48h_L}$.

Step 4 of 4

For fully turbulent flow in a rough pipe, the friction factor is independent of the Reynolds number.

Write the expression for head loss of the pipe.

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

Find the new head loss when the velocity is doubled.

Substitute $2V$ for V.

$$h_L' = f\frac{L}{D}\frac{(2V)^2}{2g}$$

$$= 4\left[f\frac{L}{D}\frac{V^2}{2g}\right]$$

$$= 4h_L$$

Therefore, the new head loss for turbulent flow when the velocity of fluid velocity is doubled is $\boxed{4h_L}$.

14.036P

Question:

Repeat Prob. 14-35 for turbulent flow in smooth pipes for which the friction factor is given as $f = 0.184Re^{-0.2}$. What would your answer be for fully turbulent flow in a rough pipe?

Step 1 of 6

Given data:

The friction factor, $f = 0.184\,\mathrm{Re}^{-0.2}$

It is assumed that the flow is turbulent through out the pipe. The inner surface is assumed to be smooth.

The Reynolds number, $\mathrm{Re} = \dfrac{\rho VD}{\mu}$

Where, ρ is the density of the flowing fluid

V is the average flow velocity

D is the diameter of the pipeline and

μ is the dynamic viscosity

Step 2 of 6

Question:

Repeat Prob. 14-35 for turbulent flow in smooth pipes for which the friction factor is given as $f = 0.184Re^{-0.2}$. What would your answer be for fully turbulent flow in a rough pipe?

Step 1 of 6

Given data:

The friction factor, $f = 0.184\,\mathrm{Re}^{-0.2}$

It is assumed that the flow is turbulent through out the pipe. The inner surface is assumed to be smooth.

The Reynolds number, $\mathrm{Re} = \dfrac{\rho VD}{\mu}$

Where, ρ is the density of the flowing fluid

V is the average flow velocity

D is the diameter of the pipeline and

μ is the dynamic viscosity

Step 2 of 6

$$h_L = \frac{gD^{1.8}}{}\left(\frac{\mu}{}\right)^{\cdot\cdot\cdot}$$

$$= 3.48\left[\frac{0.092L}{gD^{1.8}}\left(\frac{\rho}{\mu}\right)^{-0.2}V^{1.8}\right]$$

$$= 3.48h_L$$

Therefore, when the velocity is doubled, the head loss is increased by 3.48 times.

Step 5 of 6

For fully turbulent flow in a rough pipe, the friction factor is independent of the Reynolds number.

Therefore, the flow velocity

Thus, in the equation of head loss, $h_L = f\dfrac{L}{D}\dfrac{V^2}{2g}$

The friction factor, length of pipeline, diameter of the pipeline and the gravitational acceleration are constant.

Step 6 of 6

When the velocity is doubled $(V = 2V)$, the new head loss,

$$h_L' = f\frac{L}{D}\frac{(2V)^2}{2g}$$

$$= 4\left[f\frac{L}{D}\frac{V^2}{2g}\right]$$

$$h_L = \frac{gD^{1.8}}{}\left(\frac{\mu}{}\right)^{\cdot\cdot\cdot}$$

$$= 3.48\left[\frac{0.092L}{gD^{1.8}}\left(\frac{\rho}{\mu}\right)^{-0.2}V^{1.8}\right]$$

$$= 3.48h_L$$

Therefore, when the velocity is doubled, the head loss is increased by 3.48 times.

Question: 14.039P

Oil with ρ = 876 kg/m3 and μ = 0.24 kg/m-s is flowing through a 1.5-cm-diameter pipe that discharges into the atmosphere at 88 kPa. The absolute pressure 15 m before the exit is measured to be 135 kPa. Determine the flow rate of oil through the pipe if the pipe is (*a*) horizontal, (*b*) inclined 8° upward from the horizontal, and (*c*) inclined 8° downward from the horizontal.

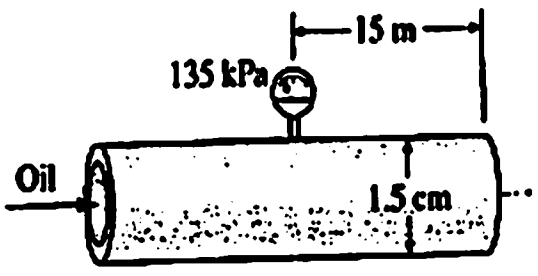


FIGURE P14–39

Step 1 of 5

Given data:

The density of oil, $\rho = 876\ \text{kg/m}^3$

The dynamic viscosity of oil, $\mu = 0.24\ \text{kg/m}\cdot\text{s}$

The diameter of pipe, $D = 1.5\ \text{cm} = 1.5\times10^{-2}\ \text{m}$

The exit pressure, $P_2 = 88\ \text{kPa}$

The absolute pressure 15 m before exit, $P_1 = 135\ \text{kPa}$

Step 2 of 5

The flow is assumed to be steady, fully developed, incompressible and laminar.

The length, $L = 15\ \text{m}$

The cross sectional flow area,

$$A_c = \frac{\pi}{4}D^2$$
$$= \frac{\pi}{4}\left(1.5\times10^{-2}\ \text{m}\right)^2$$
$$= 1.767\times10^{-4}\ \text{m}^2$$

Step 3 of 5

The pressure drop,

$$\Delta P = P_1 - P_2$$
$$= 135\ \text{kPa} - 88\ \text{kPa}$$
$$= 47\ \text{kPa}$$
$$= 47\times10^3\ \text{Pa}$$

Step 4 of 5

a) The pipe is horizontal.

The volume flow rate for horizontal pipe is given by $\dot{V} = \dfrac{\Delta P \pi D^4}{128\mu L}$

Substituting the known values, we get

$$\dot{V} = \frac{\left(47\times10^3\ \text{Pa}\right)\pi\left(1.5\times10^{-2}\ \text{m}\right)^4}{128\left(0.24\ \text{kg/m}\cdot\text{s}\right)\left(15\ \text{m}\right)}$$
$$= 1.62\times10^{-5}\ \text{m}^3/\text{s}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 1.62\times10^{-5}\ \text{m}^3/\text{s}}$

Step 5 of 5

b) The pipe is inclined 8° upward from the horizontal. So, $\theta = 8°$

The volume flow rate for inclined pipe is given by $\dot{V} = \dfrac{\left(\Delta P - \rho g L \sin\theta\right)\pi D^4}{128\mu L}$,

Substituting the known values, we get

$$\dot{V} = \frac{\left[\Delta P - \rho g L \sin\theta\right]\pi D^4}{128\mu L}$$
$$= \frac{\left[47\times10^3\ \text{Pa} - \left(876\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)\left(15\ \text{m}\right)\sin\left(8°\right)\right]\pi\left(1.5\times10^{-2}\ \text{m}\right)^4}{128\left(0.24\ \text{kg/m}\cdot\text{s}\right)\left(15\ \text{m}\right)}$$
$$= 1.00\times10^{-5}\ \text{m}^3/\text{s}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 1.00\times10^{-5}\ \text{m}^3/\text{s}}$

Question: 14.040P

Glycerin at 40°C with ρ = 1252 kg/m3 and μ = 0.27 kg/m-s is flowing through a 2-cm-diameter, 25-m- long pipe that discharges into the atmosphere at 100 kPa. The flow rate through the pipe is 0.048 L/s. (a) Determine the absolute pressure 25 m before the pipe exit. (*b*) At what angle θ must the pipe be inclined downward from the horizontal for the pressure in the entire pipe to be atmospheric pressure and the flow rate to be maintained the same?

Step 1 of 8

Given data:

The density of glycerin at 40°C, $\rho = 1252\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 0.27\ \text{kg/m}\cdot\text{s}$

The diameter of pipe, $D = 2\ \text{cm}$

$= 2\times10^{-2}\ \text{m}$

The length of the pipe, $L = 25\ \text{m}$

The exit pressure, $P_2 = 100\ \text{kPa}$

The flow rate through the pipe, $\dot{V} = 0.048\ \text{lit/s}$

$= 0.048\times10^{-3}\ \text{m}^3/\text{s}$

Step 2 of 8

The flow is assumed to be steady, fully developed and incompressible.

The volume flow rate for inclined pipe is given by $\dot{V} = \dfrac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L}$,

Where, ΔP is the pressure drop,

θ is the angle between the pipe and horizontal and

g is the gravitational acceleration.

Step 3 of 8

The cross sectional flow area,

$$A_c = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}\left(2\times10^{-2}\ \text{m}\right)^2$$

$$= 3.1416\times10^{-4}\ \text{m}^2$$

Therefore, the velocity of flow,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{0.048\times10^{-3}\ \text{m}^3/\text{s}}{3.1416\times10^{-4}\ \text{m}^2}$$

$$= 0.153\ \text{m/s}$$

Step 4 of 8

The Reynolds number,

$$\text{Re} = \frac{\rho VD}{\mu}$$

$$= \frac{\left(1252\ \text{kg/m}^3\right)\left(0.153\ \text{m/s}\right)\left(2\times10^{-2}\ \text{m}\right)}{0.27\ \text{kg/m}\cdot\text{s}}$$

$$= 14.189$$

Here, Re < 2300. Therefore, the flow is laminar.

Step 5 of 8

a) Here, the pipe is horizontal.

So, $\theta = 0$

Then,

$$\dot{V} = \frac{\left(\Delta P - \rho g L(\sin 0)\right)\pi D^4}{128\mu L}$$

$$= \frac{\Delta P\pi D^4}{128\mu L}$$

On rearranging,

$$\Delta P = \frac{128\mu L\dot{V}}{\pi D^4}$$

$$= \frac{128(0.27\ \text{kg/m}\cdot\text{s})(25\ \text{m})\left(0.048\times10^{-3}\ \text{m}^3/\text{s}\right)}{\pi\left(2\times10^{-2}\ \text{m}\right)^4}$$

$$= 82.506\times10^3\ \text{N/m}^2$$

$$= 82.506\ \text{kN/m}^2$$

$$= 82.506\ \text{kPa}$$

Step 6 of 8

The pressure drop is given by $\Delta P = P_1 - P_2$,

Where, P_1 is the initial pressure

Then,

$$P_1 = \Delta P + P_2$$

$$= 82.506\ \text{kPa} + 100\ \text{kPa}$$

$$= 182.506\ \text{kPa}$$

Therefore, the pressure 25 m before exit, $\boxed{P_1 = 182.506\ \text{kPa}}$

Step 7 of 8

b) The pressure in the pipe is equal to atmospheric pressure.

Now, $\Delta P = P_1 - P_2 = 0$

The flow rate, $\dot{V} = 0.048\times10^{-3}\ \text{m}^3/\text{s}$

The volume flow rate for inclined pipe,

$$\dot{V} = \frac{\left[\Delta P - \rho g L\sin\theta\right]\pi D^4}{128\mu L}$$

$$0.048\times10^{-3}\ \text{m}^3/\text{s} = \frac{\left[0 - \left(1252\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)(25\ \text{m})\sin\theta\right]\pi\left(2\times10^{-2}\ \text{m}\right)^4}{128(0.27\ \text{kg/m}\cdot\text{s})(25\ \text{m})}$$

$$= -\left(1.786\times10^{-4}\ \text{m}^3/\text{s}\right)\sin\theta$$

Step 8 of 8

Now calculating for θ, we get

$$0.048\times10^{-3} = -1.786\times10^{-4}(\sin\theta)$$

$$\sin\theta = -\frac{0.048\times10^{-3}}{1.786\times10^{-4}}$$

$$\sin\theta = -0.269$$

$$\theta = -15.6°$$

Therefore, the pipe must be inclined downwards at an angle $\boxed{\theta = 15.6°}$

Question: 14.041P

Oil at 80°F (ρ = 56.8 lbm/ft3 and μ = 0.0278 lbm/ft-s) jS flowing steadily in a 0.5-in-diameter, 120-ft-long pipe. During the flow, the pressure at the pipe inlet and exit is measured to be 120 psi and 14 psi, respectively. Determine the flow rate of oil through the pipe assuming the pipe is (*a*) horizontal, (*b*) inclined 20° upward, and (*c*) inclined 20° downward.

Step 1 of 5

Given data:

Temperature of the oil, $T = 80°\text{F}$

Density of the oil at temperature $T = 80°\text{F}$, $\rho = 56.8\,\text{lbm/ft}^3$

Dynamic viscosity $\mu = 0.0278\,\text{lbm/ft}\cdot\text{scc}$

Pressure at inlet of the pipe $P_1 = 120\,\text{psi}$

Pressure at exit of the pipe $P_2 = 14\,\text{psi}$

Diameter of the pipe $D = 0.5\,\text{in.}$

Length of the pipe $L = 120\,\text{ft}$

Step 2 of 5

Cross section area of the oil flow

$$\begin{aligned} A_c &= \frac{\pi}{4}D^2 \\ &= \frac{\pi}{4}(0.5)^2 \\ &= 0.196\,\text{in.}^2 \end{aligned}$$

Pressure drop in the pipe

$$\begin{aligned} \Delta P &= P_1 - P_2 \\ &= 120\,\text{psi} - 14\,\text{psi} \\ &= 106\,\text{psi} \\ &= 15264\,\text{lb/ft}^2 \end{aligned}$$

Step 3 of 5

a) The pipe is horizontal $(\theta = 0°)$

The volume flow rate for horizontal pipe is given by $\dot{V} = \dfrac{\Delta P \pi D^4}{128\mu L}$

Substituting the known values, we get

$$\begin{aligned} \dot{V} &= \frac{\left(106\frac{\text{lbf}}{\text{in.}^2}\right)\pi(0.5)^4\,\text{in.}^4}{128\left(0.0278\frac{\text{lbm}}{\text{ft}\cdot\text{sec}}\right)(120\,\text{ft})} \\ &\quad \frac{\left(106\times32.2\times12\frac{\text{lbm}\cdot\text{in}}{\text{sec}^2}\right)\pi(0.5)^4\,\text{in.}^2}{128\left(0.0278\frac{\text{lbm}}{\text{ft}\cdot\text{scc}}\right)(120\,\text{ft})} \\ &= 28.43\,\text{in.}^3/\text{sec} \end{aligned}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 28.43\,\text{in.}^3/\text{s}}$

Step 4 of 5

b) The pipe is inclined in $20°$ upward from the horizontal. So, $\theta = 20°$

The volume flow rate for inclined pipe is given by $\dot{V} = \dfrac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$,

Substituting the known values, we get

$$\begin{aligned} \dot{V} &= \frac{[\Delta P - \rho g L \sin\theta]\pi D^4}{128\mu L} \\ &= \frac{\left[(15264\times32.2)\frac{\text{lbm}\cdot\text{ft}}{\text{ft}^2\cdot\text{sec}^2} - \left(56.8\frac{\text{lbm}}{\text{ft}^3}\times32.2\ \text{ft/s}^2\right)(120\,\text{ft})\sin(20°)\right]\pi\left(\frac{0.5}{12}\right)^4\text{ft}^4}{128(0.0278\,\text{lbm/ft}\cdot\text{sec})(120\,\text{ft})} \\ &= 9.23\times10^{-3}\,\text{ft}^3/\text{s} \end{aligned}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 9.23\times10^{-3}\,\text{ft}^3/\text{s}}$

Step 5 of 5

c) The pipe is inclined $8°$ downward from the horizontal. So, $\theta = -8°$

The volume flow rate for inclined pipe, $\dot{V} = \dfrac{(\Delta P - \rho g L \sin\theta)\pi D^4}{128\mu L}$

Substituting the known values, we get

$$\begin{aligned} \dot{V} &= \frac{[\Delta P - \rho g L \sin\theta]\pi D^4}{128\mu L} \\ &= \frac{\left[(15264\times32.2)\frac{\text{lbm}\cdot\text{ft}}{\text{ft}^2\cdot\text{sec}^2} - \left(56.8\frac{\text{lbm}}{\text{ft}^3}\times32.2\ \text{ft/s}^2\right)(120\,\text{ft})\sin(-20°)\right]\pi\left(\frac{0.5}{12}\right)^4\text{ft}^4}{128(0.0278\,\text{lbm/ft}\cdot\text{sec})(120\,\text{ft})} \\ &= 0.0125\,\text{ft}^3/\text{s} \end{aligned}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 0.0125\,\text{ft}^3/\text{s}}$

Question: 14.042P

Oil with a density of 850 kg/m3 and kinematic viscosity of 0.00062 m2/s is being discharged by a 8-mm-diameter, 40-m-long horizontal pipe from a storage tank open to the atmosphere. The height of the liquid level above the center of the pipe is 4 m. Disregarding the minor losses, determine the flow rate of oil through the pipe.

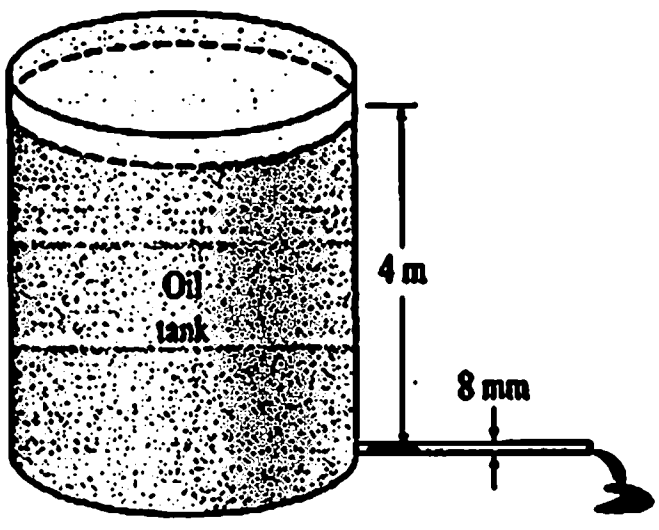


FIGURE P14–42

Step 1 of 5

Given data:

Density of the oil, $\rho = 850\ \text{kg/m}^3$

Kinetic viscosity of the oil $\nu = 0.00062\ \text{m}^2/\text{s}$

Diameter of pipe, $D = 8\ \text{mm}$

$= 8 \times 10^{-3}\ \text{m}$

The length of the pipe, $L = 40\ \text{m}$

The height of the liquid level above the centre of the pipe, $h = 4\ \text{m}$

Step 2 of 5

The volume flow rate for horizontal pipe is given by $\dot{V} = \dfrac{\Delta P \pi D^4}{128 \mu L}$,

Where, ΔP is the pressure drop and
μ is the dynamic viscosity

The pressure at the bottom open of the oil tank,

$$P_{bottom} = \rho g h$$
$$= (850\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(4\ \text{m})$$
$$= 33.354 \times 10^3\ \text{N/m}^2$$

Now, the change in pressure,

$$\Delta P = P_{bottom} - P_{top}$$
$$= (3.354 \times 10^3) - 0$$
$$= 33.354 \times 10^3\ \text{N/m}^2$$

Step 3 of 5

The dynamic viscosity,

$$\mu = \rho \nu$$
$$= (850\ \text{kg/m}^3)(0.00062\ \text{m}^2/\text{s})$$
$$= 0.527\ \text{kg/m} \cdot \text{s}$$

Substitute the known values in the equation for flow rate,

$$\dot{V} = \frac{(33.354 \times 10^3\ \text{N/m}^2)\pi(8 \times 10^{-3}\ \text{m})^4}{128(0.527\ \text{kg/m} \cdot \text{s})(40\ \text{m})}$$
$$= 1.59 \times 10^{-7}\ \text{m}^3/\text{s}$$

Therefore, the flow rate of oil through horizontal pipe, $\boxed{\dot{V} = 1.59 \times 10^{-7}\ \text{m}^3/\text{s}}$

Step 4 of 5

The cross sectional flow area,

$$A_t = \frac{\pi}{4} D^2$$
$$= \frac{\pi}{4}(8 \times 10^{-3}\ \text{m})^2$$
$$= 5.026 \times 10^{-5}\ \text{m}^2$$

The velocity of flow,

$$V = \frac{\dot{V}}{A_t}$$
$$= \frac{1.59 \times 10^{-7}\ \text{m}^3/\text{s}}{5.026 \times 10^{-5}\ \text{m}^2}$$
$$= 3.16 \times 10^{-3}\ \text{m/s}$$

Step 5 of 5

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$
$$= \frac{(850\ \text{kg/m}^3)(3.16 \times 10^{-3}\ \text{m/s})(8 \times 10^{-3}\ \text{m})}{0.527\ \text{kg/m} \cdot \text{s}}$$
$$= 0.0408$$

Here, Reynolds number Re <2300

Therefore, the flow is laminar and the assumption and analysis is correct.

Question: 14.043P

In an air heating system, heated air at 40°C and 105 kPa absolute is distributed through a 0.2 m × 0.3 m rectangular duct made of commercial steel at a rate of 0.5 m3/s. Determine the pressure drop and head loss through a 40-m- long section of the duct.

Step 1 of 6

Assume the flow is steady, incompressible and fully developed.

Obtain the equivalent roughness factor for commercial steel pipe from "Table 8.1 Equivalent roughness for new pipes" in the textbook.

$$\varepsilon = 0.045\,\text{mm}$$

Obtain the dynamic viscosity of air at $40°\text{C}$ from "Ideal gas properties of air" in the textbook.

$$\mu = 1.918 \times 10^{-5}\,\text{kg/m}\cdot\text{s}$$

Find the density of air.

$$\rho = \frac{P}{RT}$$

Here, P is the pressure of air, R is the characteristic gas constant of air, T is the temperature of air.

Substitute $105\,\text{kPa}$ for P, $0.287\,\text{kJ/kg-K}$ for R, and $313\,\text{K}$ for T.

$$\rho = \frac{105}{(0.287\ \text{kJ/kg}\cdot\text{K})(313\ \text{K})}$$
$$= 1.169\ \text{kg/m}^3$$

Step 2 of 6

Find the cross sectional flow area.

$$A_c = 0.2\ \text{m} \times 0.3\ \text{m}$$
$$= 0.06\ \text{m}^2$$

Find the wetted perimeter.

$$p = 2(0.2\ \text{m} + 0.3\ \text{m})$$
$$= 1\ \text{m}$$

Find the hydraulic diameter.

$$D_h = \frac{4A_c}{p}$$

Substitute $0.06\,\text{m}^2$ for A_c and $0.24\,\text{m}$ for p.

$$D_h = \frac{4(0.06)}{1}$$
$$= 0.24\ \text{m}$$

Step 3 of 6

Find the velocity of flow.

$$V = \frac{\dot{V}}{A_c}$$

Here, $\dot{V}$ is the flow rate of air.

Substitute $0.5\,\text{m}^3/\text{s}$ for $\dot{V}$ and $0.06\,\text{m}^2$ for A_c.

$$V = \frac{0.5}{0.06}$$
$$= 8.33\ \text{m/s}$$

Find the Reynolds number.

$$\text{Re} = \frac{\rho V D_h}{\mu}$$

Substitute $1.169\ \text{kg/m}^3$ for ρ, $8.33\ \text{m/s}$ for V, $0.24\ \text{m}$ for D_h, and $1.918 \times 10^{-5}\ \text{kg/m}\cdot\text{s}$ for μ.

$$\text{Re} = \frac{(1.169)(8.33)(0.24)}{1.918 \times 10^{-5}}$$
$$= 121.85 \times 10^3$$

Since $\text{Re} > 4000$, the flow is turbulent.

Step 4 of 6

Use Colebrook equation to find friction factor.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Substitute $0.045\ \text{mm}$ for ε, $0.24\ \text{m}$ for D_h, and 121.85×10^3 for Re.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0.045\,\text{mm} \times 10^{-3}\,\frac{\text{m}}{\text{mm}} \Big/ 0.24}{3.7} + \frac{2.51}{(121.85 \times 10^3)\sqrt{f}}\right)$$
$$= -2.0 \log\left(5.067 \times 10^{-5} + \frac{2.06 \times 10^{-5}}{\sqrt{f}}\right)$$

Solve the equation for f.

$$f = 0.0183$$

Step 5 of 6

Find the pressure drop through the duct.

$$\Delta P_L = f\frac{L}{D_h}\frac{\rho V^2}{2}$$

Substitute 0.0183 for f, $40\,\text{m}$ for L, $0.24\ \text{m}$ for D_h, $1.169\ \text{kg/m}^3$ for ρ, and $8.33\ \text{m/s}$ for V.

$$\Delta P_L = f\frac{L}{D_h}\frac{\rho V^2}{2}$$
$$= 0.0183\left[\frac{40}{0.24}\right]\left[\frac{(1.169)(8.33)^2}{2}\right]$$
$$= 123.7\ \text{Pa}$$

Step 6 of 6

Find the head loss in the duct.

$$h_L = \frac{\Delta P_L}{\rho g}$$

Substitute $123.7\,\text{Pa}$ for ΔP_L, $1.169\ \text{kg/m}^3$ for ρ, and $9.81\,\text{m/s}^2$ for g.

$$h_L = \frac{123.7\ \text{Pa}}{(1.169\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}$$
$$= 10.79\ \text{m}$$

Therefore, the pressure drop and the head loss in the duct is $\boxed{124\,\text{Pa}, 10.79\,\text{m}}$.

Question: 14.044P

Glycerin at 40°C with ρ = 1252 kg/m3 and μ = 0.27 kg/m-s is flowing through a 4-cm-diameter horizontal smooth pipe with an average velocity of 3.5 m/s. Determine the pressure drop per 10 m of the pipe.

Step 1 of 4

Given data:

The density of glycerin at 40°C, $\rho = 1252\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 0.27\ \text{kg/m}\cdot\text{s}$

The diameter of pipe, $D = 4\ \text{cm}$

$= 4 \times 10^{-2}\ \text{m}$

The average velocity of the glycerin, $V = 3.5\ \text{m/s}$

The length of the pipe, $L = 10\ \text{m}$

Step 2 of 4

The flow is assumed to be steady, fully developed and incompressible.

The cross sectional flow area,

$$A_c = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}\left(4 \times 10^{-2}\ \text{m}\right)^2$$

$$= 1.2566 \times 10^{-3}\ \text{m}^2$$

The flow rate of the fluid,

$$\dot{V} = VA_c$$

$$= \left(3.5\ \text{m/s}\right)\left(1.2566 \times 10^{-3}\ \text{m}^2\right)$$

$$= 4.3981 \times 10^{-3}\ \text{m}^3/\text{s}$$

Step 3 of 4

The Reynolds number,

$$\text{Re} = \frac{\rho VD}{\mu}$$

$$= \frac{\left(1252\ \text{kg/m}^3\right)\left(3.5\ \text{m/s}\right)\left(4 \times 10^{-2}\ \text{m}\right)}{0.27\ \text{kg/m}\cdot\text{s}}$$

$$= 649.185$$

Here, Re < 2300. Therefore, the flow is laminar.

Step 4 of 4

The friction factor,

$$f = \frac{64}{\text{Re}}$$

$$= \frac{64}{649.185}$$

$$= 0.0986$$

The pressure drop in the pipe,

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2}$$

$$= 0.0986\left[\frac{10\ \text{m}}{4 \times 10^{-2}\ \text{m}}\right]\left[\frac{\left(1252\ \text{kg/m}^3\right)\left(3.5\ \text{m/s}\right)^2}{2}\right]$$

$$= 189.028 \times 10^3\ \text{Pa}$$

$$= 189.028\ \text{kPa}$$

Therefore, the pressure drop per 10 m is $\boxed{189.03\ \text{kPa}}$

Question: 14.045P

Reconsider Prob. 14-44. Using EES (or other) software, investigate the effect of the pipe diameter on the pressure drop for the same constant flow rate. Let the pipe diameter vary from 1 to 10 cm in increments of 1 cm. Tabulate and plot the results, and draw conclusions.

Step 1 of 3

The EES code for solving the given problem is as below:

```
Vdot=3.5*pi*(0.05)^2/4          "Volumetric flow rate"
Ac=pi*D^2/4                     "Cross sectional area"
rho=1252                        "density"
nu=mu/rho                       "Kinematic viscosity"
mu=0.27                         "Dynamic viscosity"
L=10                            "Length"
V=Vdot/Ac                       "Velocity"
Re=V*D/nu                       "Reynolds number"
f=64/Re                         "friction"
DeltaP=f*(L/D)*rho*V^2/2000     "Change in pressure"
W=Vdot*DeltaP                   "Work"
```

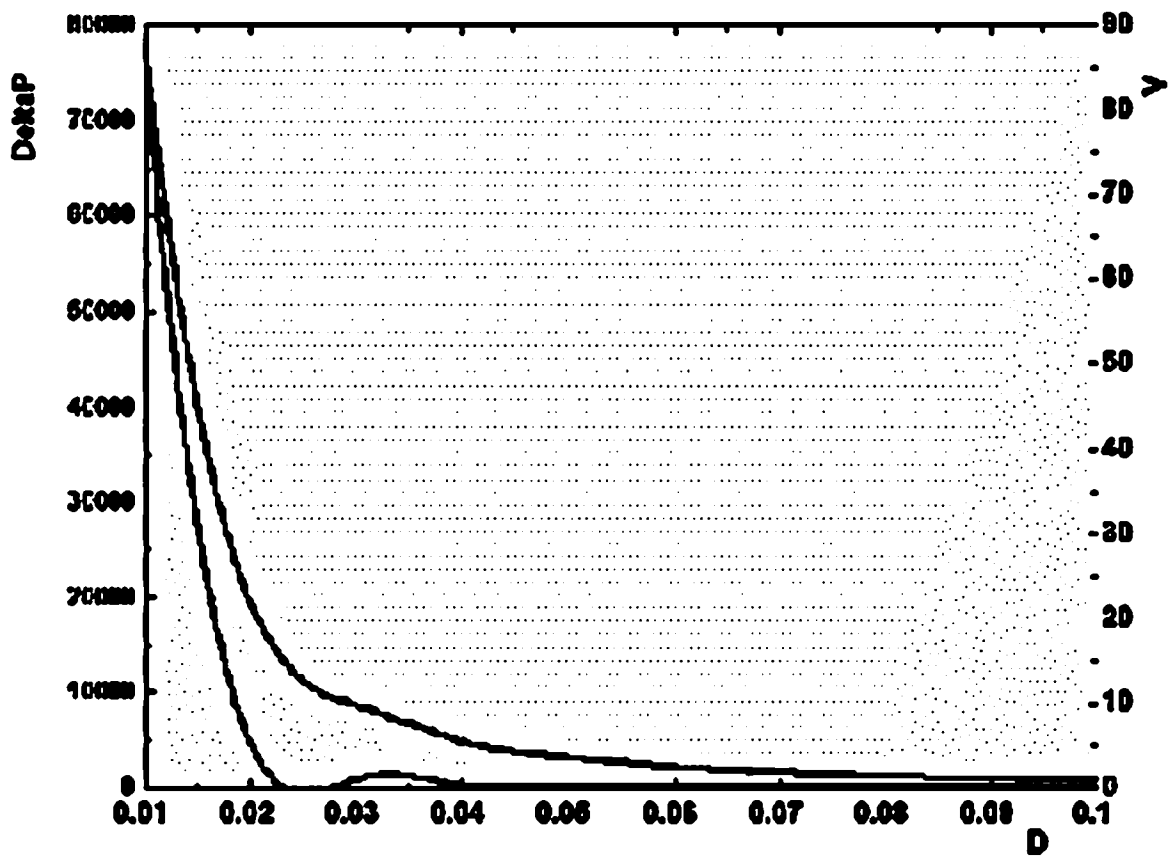
Step 2 of 3

Now enter the values of diameter by selecting the

▷ 1..10	1 D	2 δP	3 Re	4 V
Run 1	0.01	75600	4057	87.5
Run 2	0.02	4725	2029	21.88
Run 3	0.03	933.3	1352	9.722
Run 4	0.04	295.3	1014	5.469
Run 5	0.05	121	811.5	3.5
Run 6	0.06	58.33	676.2	2.431
Run 7	0.07	31.49	579.6	1.786
Run 8	0.08	18.46	507.2	1.367
Run 9	0.09	11.52	450.8	1.08
Run 10	0.1	7.56	405.7	0.875

Step 3 of 3

The plot showing the variation of 'DeltaP' and 'V' with diameter is as below:



Question: 14.046P

Air at 1 atm and 60°F is flowing through a 1 ft × 1 ft square duct made of commercial steel at a rate of 1600 cfm. Determine the pressure drop and head loss per ft of the duct.

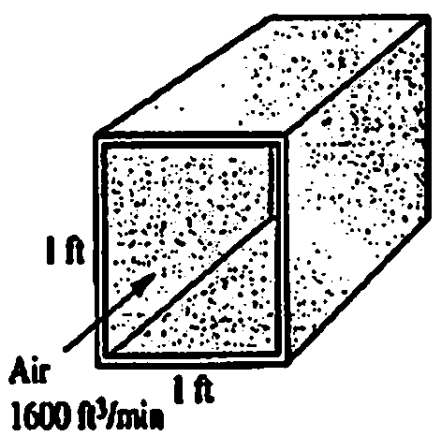


FIGURE P14–46E

Step 1 of 5

Convert the discharge rate of air from ft^3/min to ft^3/s.

Discharge of air passing through the duct,

$$\dot{V} = 1600\frac{\text{ft}^3}{\text{min}} \times \frac{1\ \text{min}}{60\ \text{sec}}$$
$$= 26.66\ \text{ft}^3/\text{s}$$

Calculate the cross sectional area of the duct.

$$A = 1 \times 1$$
$$= 1\ \text{ft}^2$$

Calculate the velocity of flow.

$$V = \dot{V} / A$$

Substitute $26.66\ \text{ft}^3/\text{s}$ for $\dot{V}$, and $1\ \text{ft}^2$ for A.

$$V = 26.66 / 1$$
$$= 26.66\ \text{ft/s}$$

Step 2 of 5

Characteristic length for a square duct,

$$D_h = a$$
$$= 1\ \text{ft}$$

Here, a is the length of a side of the square duct.

Obtain the following properties of air from table A-22E, "Properties of air at 1 atm pressure", from the textbook at 1 atm and $60°\text{F}$.

Density of air, $\rho = 0.0764\ \text{lbm/ft}^3$

Dynamic viscosity of air, $\mu = 1.135 \times 10^{-5}\ \text{lbm/ft} \cdot \text{s}$

Calculate the Reynolds number.

$$\text{Re} = \frac{\rho V D_h}{\mu}$$

Substitute $0.0764\ \text{lbm/ft}^3$ for ρ, $26.66\ \text{ft/s}$ for V, 1 ft for a, and $1.135 \times 10^{-5}\ \text{lbm/ft} \cdot \text{s}$ for μ.

$$\text{Re} = \frac{0.0764 \times 26.66 \times 1}{1.135 \times 10^{-5}}$$
$$= 179455.85$$

Since the obtained Reynolds number is more than 4000 flow is in turbulent nature.

Step 3 of 5

Equivalent roughness for commercial steel from table "Equivalent roughness values for new commercial pipes" is considered to be $\varepsilon = 0.00015$

The ducts relative roughness is given by,

$$\frac{\varepsilon}{D} = \frac{0.00015}{1}$$
$$= 0.00015$$

From Colebrook equation friction factor can be calculated as,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$
$$= -2.0 \log\left(\frac{0.00015}{3.7} + \frac{2.51}{179455.85\sqrt{f}}\right)$$
$$= -2.0 \log\left(4.05 \times 10^{-5} + \frac{1.398 \times 10^{-5}}{\sqrt{f}}\right)$$
$$f = 0.01704$$

Step 4 of 5

Pressure drop in the duct for turbulent flow is given by,

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2}$$

But pressure loss per unit length is given by,

$$\frac{\Delta P}{L} = f \frac{\rho V^2}{2D}$$
$$= 0.01704 \times \frac{0.0764 \times 26.66^2}{2 \times 1} \times \left(\frac{1}{32.2}\right)$$
$$= 0.0143\ \text{lbf/ft}^3$$
$$= 1 \times 10^{-4}\ \text{psi/ft}$$

Step 5 of 5

Head loss per feet is given by,

$$h_l = \frac{\frac{\Delta P}{L}}{\rho g}$$

Substitute $1 \times 10^{-4}\ \text{psi/ft}$ for $\Delta P / L$, and $0.0764\ \text{lbm/ft}^3$ for ρg.

$$h_l = \frac{1 \times 10^{-4}\ \frac{\text{psi}}{\text{ft}} \times \frac{144\ \text{in}^2}{1\ \text{ft}^2}}{0.0764\ \frac{\text{lbm}}{\text{ft}^3}}$$
$$= 0.188\ \text{ft/ft}$$

Therefore the pressure loss is obtained as $\boxed{1 \times 10^{-4}\ \text{psi/ft}}$ and the head loss is obtained as $\boxed{0.188\ \text{ft/ft}}$.

Question: 14.047P

Liquid ammonia at -20°C is flowing through a 30-m-long section of a 5-mm-diameter copper tube at a rate of 0.15 kg/s. Determine the pressure drop, the head loss, and the pumping power required to overcome the frictional losses in the tube.

Step 1 of 9

Given data:

Temperature at which liquid ammonia is flowing through a copper tube is $-20°C$.

The length of the pipe, $L = 30\,\text{m}$.

The diameter of pipe,

$$D = 5\,\text{mm} = 5\times10^{-3}\,\text{m}$$

The mass flow rate, $\dot{m} = 0.15\,\text{kg/s}$.

Step 2 of 9

The flow is assumed to be steady, fully developed and incompressible.

From the table of properties of saturated ammonia, corresponding to $-20°C$,

The density, $\rho = 665.1\,\text{kg/m}^3$.

The dynamic viscosity, $\mu = 2.361\times10^{-4}\,\text{kg/m}\cdot\text{s}$.

For copper tube, the roughness,

$$\varepsilon = 0.0015\,\text{mm} = 1.5\times10^{-6}\,\text{m}$$

Step 3 of 9

Area of the cross sectional flow,

$$A_c = \frac{\pi}{4}D^2 = \frac{\pi}{4}\left(5\times10^{-3}\right)^2 = 19.635\times10^{-6}\,\text{m}^2$$

Velocity,

$$V = \frac{\dot{m}}{\rho A_c} = \frac{0.15}{(665.1)\left(19.635\times10^{-6}\right)} = 11.486\,\text{m/s}$$

Step 4 of 9

The flow rate,

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.15\,\text{kg/s}}{665.1\,\text{kg/m}^3} = 2.255\times10^{-4}\,\text{m}^3/\text{s}$$

Step 5 of 9

The Reynolds number,

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(665.1)(11.486)\left(5\times10^{-3}\right)}{2.361\times10^{-4}} = 161.782\times10^3$$

Here, $\text{Re} > 4000$. Therefore, the flow is turbulent.

Step 6 of 9

For turbulent flow, the Colebrook equation is given by

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right),$$

Where f is the friction factor.

Substitute the known values.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\left(1.5\times10^{-6}/5\times10^{-3}\right)}{3.7} + \frac{2.51}{\left(161.782\times10^3\right)\sqrt{f}}\right)$$

$$= -2.0\log\left(8.108\times10^{-5} + \left(\frac{1.55\times10^{-5}}{\sqrt{f}}\right)\right)$$

Solve the equation for f, we get

$$f = 0.01818$$

Step 7 of 9

The pressure drop,

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.01818\left[\frac{30}{5\times10^{-3}}\right]\left[\frac{(665.1)(11.486)^2}{2}\right]$$

$$= 4785.6\times10^3\,\text{Pa}$$

$$\boxed{\Delta P_L = 4785.6\,\text{kPa}}$$

Therefore, the pressure drop is $\Delta P_L = 4785.6\,\text{kPa}$.

Step 8 of 9

The head loss is given by $h_L = \frac{\Delta P_L}{\rho g}$,

Where g is the gravitational acceleration.

Substitute the known values.

$$h_L = \frac{4785.6\times10^3}{665.1\times9.81}$$

$$\boxed{h_L = 733.5\,\text{m}}$$

Therefore, the head loss, $h_L = 733.5\,\text{m}$.

Step 9 of 9

The pumping power required,

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P_L = \left(2.255\times10^{-4}\right)\times\left(4785.6\times10^3\right)$$

$$= 1.079\times10^3\,\text{W}$$

$$\boxed{\dot{W}_{\text{pump}} = 1.079\,\text{kW}}$$

Therefore, the pumping power required to overcome the pressure loss, $\dot{W}_{\text{pump}} = 1.079\,\text{kW}$.

Question: 14.048P

What is minor loss in pipe flow? How is the minor loss coefficient *KL* defined?

Step 1 of 1

In typical pipe systems in addition to straight sections, there may be various fittings, valves, bends, elbows, tees, inlets, exits, expansions and contractions. These components disturb the smooth flow of the fluid, results in additional losses due to the influence of flow separation and mixing etc. These losses are called minor losses. Minor losses are expressed in terms of the loss coefficient.

The loss coefficient is given by, $K_L = \dfrac{h_L}{\left(V^2/2g\right)}$,

Where, h_L is the additional irreversible head loss,

V is the average flow velocity and

g is the gravitational acceleration.

Question: 14.049P

Define equivalent length for minor loss in pipe flow. How is it related to the minor loss coefficient?

Step 1 of 1

The presence of various fittings, valves, bends, elbows, tees, inlets, exits, expansions and contractions in a piping system cause disturbance in the smooth flow of the fluid, results in additional losses due to the influence of flow separation and mixing etc. These losses are called minor losses.

The length of a straight pipe which would give the same head loss as the minor loss component is known as the equivalent length.

The equivalent length is given by $L_{equiv} = \frac{D}{f} K_L$,

where, K_L is the loss coefficient,

D is the diameter of the pipe that contains the components and

f is the friction factor.

Question:

14.050P

The effect of rounding of a pipe inlet on the loss coefficient is (*a*) negligible, (*b*) somewhat significant, or (*c*) very significant.

Step 1 of 1

The head loss at the pipe inlet strongly depends on the geometry of the inlet. The minor loss coefficient (K_L) of reentrant pipe inlet is 0.8, sharp-edged pipe inlet is 0.5 and well-rounded pipe inlet is 0.03.

Thus, the head loss at the rounded pipe inlet is almost negligible. Therefore, the effect of rounding of the pipe inlet on the loss coefficient is very significant.

Question: 14.051P

Which has a greater minor loss coefficient during pipe flow: gradual expansion or gradual contraction? Why?

Step 1 of 1

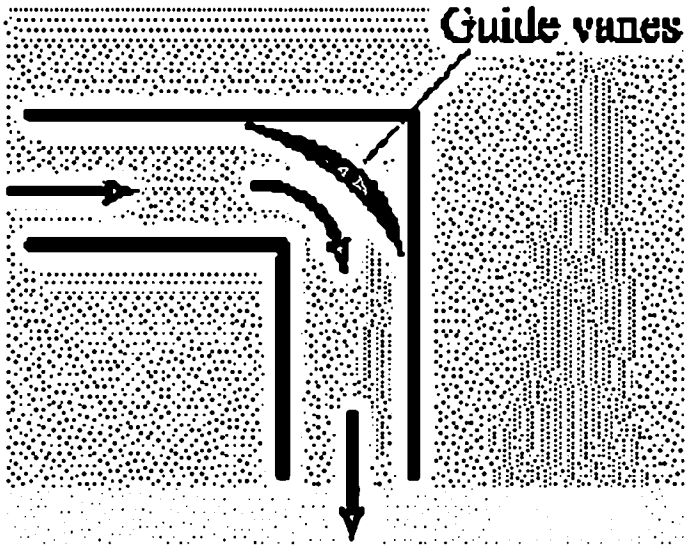
The gradual expansion has a greater minor loss coefficient than a gradual contraction in pipe flow. This is because, in gradual expansion flow, a flow separation is occurred due to the adverse pressure gradient in the boundary layer.

Question: 14.052P

A piping system involves sharp turns, and thus large minor head losses. One way of reducing the head loss is to replace the sharp turns by circular elbows. What is another way?

Step 1 of 1

In addition to circular elbows, the head losses can be reduced by using properly placed guide vanes in sharp turns.



Question: 14.053P

During a retrofitting project of a fluid flow system to reduce the pumping power, it is proposed to install vanes into the miter elbows or to replace the sharp turns in 90° miter elbows by smooth curved bends. Which approach will result in a greater reduction in pumping power requirements?

Step 1 of 1

The loss coefficient for miter elbow with vanes, $K_L \approx 0.2$

The loss coefficient for smooth curved bend, $K_L \approx 0.9$

Since the loss coefficient is less for the miter elbow with vanes, the losses associated with it will be less. Therefore, the miter elbow with vanes will help to reduce the pumping power requirements.

Question: 14.054P

Water is to be withdrawn from a 5-m-high water reservoir by drilling a 1.5-cm-diameter hole at the bottom surface. Disregarding the effect of the kinetic energy correction factor, determine the flow rate of water through the hole if (*a*) the entrance of the hole is well-rounded and (*b*) the entrance is sharp-edged.

Step 1 of 8

Given data:

Height of the reservoir, $H = 5\ \text{m}$

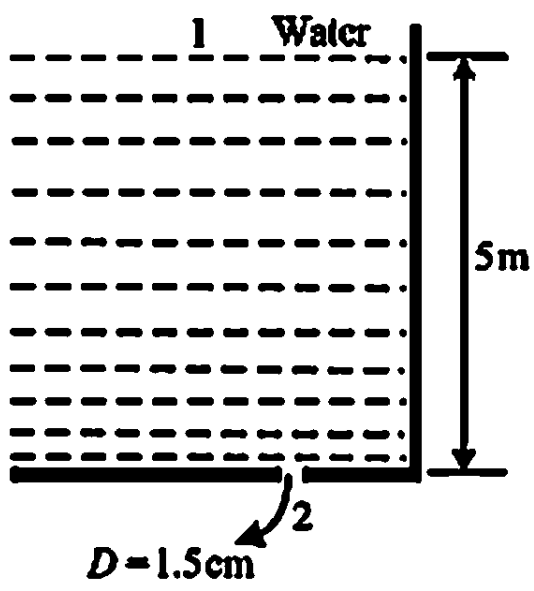
The diameter of the bottom hole,

$D = 1.5\ \text{cm}$

$= 1.5 \times 10^{-2}\ \text{m}$

Step 2 of 8

Free body diagram of the water reservoir with a hole at the bottom:



Step 3 of 8

The subscript 1 denotes the values at the free surface and subscript 2 denotes the values at the exit of the hole.

The energy equation for the two points in terms of heads,

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L$$

Step 4 of 8

Take the exit of the hole as the reference level.

Then,

$z_1 = 5\ \text{m}$

$z_2 = 0$

The pressure at both the points is equal to the atmospheric pressure,

$P_1 = P_2 = P_{\text{atm}}$

Here, the flow section does not involve any work devices such as pump or turbine. Thus,

$h_{\text{pump},u} = h_{\text{turbine},e} = 0$

The flow velocity at the free surface,

$V_1 = 0$

The effect of the kinetic energy correction factor is disregarded. So,

$\alpha_1 = \alpha_2 = 1$

Step 5 of 8

The head loss of the flow, $h_L = K_L \frac{V^2}{2g}$

Here K_L is the loss coefficient

On substituting the known values in the energy equation, we get

$$\frac{P_{\text{atm}}}{\rho g} + (1)\frac{(0)^2}{2g} + z_1 + 0 = \frac{P_{\text{atm}}}{\rho g} + (1)\frac{V_2^2}{2g} + 0 + 0 + K_L \frac{V_2^2}{2g}$$

$$z_1 = \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g}$$

$$= \frac{V_2^2}{2g}(1 + K_L)$$

$$V_2 = \sqrt{\frac{2gz_1}{1 + K_L}}$$

Step 6 of 8

The cross sectional flow area through the hole,

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}\left(1.5 \times 10^{-2}\ \text{m}\right)^2$$

$$= 1.767 \times 10^{-4}\ \text{m}^2$$

The flow rate through the hole,

$$\dot{V} = V_2 A_c$$

$$= \sqrt{\frac{2gz_1}{1 + K_L}} A_c$$

Step 7 of 8

(a)

For well-rounded entrance, $K_L = 0.03$

On substituting the known values in the equation for flow rate, we get

$$\dot{V} = \sqrt{\frac{2\left(9.81\ \text{m/s}^2\right)(5\ \text{m})}{1 + 0.03}}\left(1.767 \times 10^{-4}\ \text{m}^2\right)$$

$$= 9.759 \times 1.767 \times 10^{-4}$$

$$= 1.724 \times 10^{-3}\ \text{m}^3/\text{s}$$

Therefore, the flow rate through the well-rounded entrance, $\boxed{\dot{V} = 1.724 \times 10^{-3}\ \text{m}^3/\text{s}}$

(b)

Step 8 of 8

For sharp edged entrance, $K_L = 0.5$

On substituting the known values in the equation for flow rate, we get

$$\dot{V} = \sqrt{\frac{2\left(9.81\ \text{m/s}^2\right)(5\ \text{m})}{1 + 0.5}}\left(1.767 \times 10^{-4}\ \text{m}^2\right)$$

$$= 8.087 \times 1.767 \times 10^{-4}$$

$$= 1.429 \times 10^{-3}\ \text{m}^3/\text{s}$$

Therefore, the flow rate through the sharp edged entrance, $\boxed{\dot{V} = 1.429 \times 10^{-3}\ \text{m}^3/\text{s}}$

Question: 14.055P pg 1

A horizontal pipe has an abrupt expansion from $D1$ = 8 cm to $D2$ = 16 cm. The water velocity in the smaller section is 10 m/s and the flow is turbulent. The pressure in the smaller section is $P1$ = 410 kPa. Taking the kinetic energy correction factor to be 1.06 at both the inlet and the outlet, determine the downstream pressure $P2$. and estimate the error that would have occurred if Bernoulli's equation had been used.

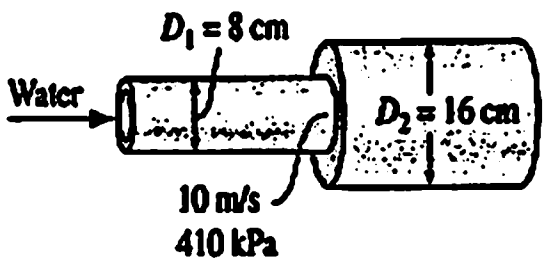


FIGURE P14–55

Step 1 of 8

Given data:

The diameter of pipe 1,

$$D_1 = 8 \text{ cm}$$
$$= 8 \times 10^{-2} \text{ m}$$

The diameter of pipe 2,

$$D_2 = 16 \text{ cm}$$
$$= 16 \times 10^{-2} \text{ m}$$

The velocity in the pipe 1, $V_1 = 10 \text{ m/s}$

The pressure in the pipe 1,

$$P_1 = 410 \text{ kPa}$$
$$= 410 \times 10^3 \text{ Pa}$$

The kinetic energy correction factor for both inlet and outlet, $\alpha_1 = \alpha_2 = 1.06$

Step 2 of 8

The flow is assumed to be steady, fully developed and incompressible.

The cross sectional flow area of pipe 1,

$$A_{C1} = \frac{\pi}{4} D_1^2$$
$$= \frac{\pi}{4}\left(8 \times 10^{-2}\right)^2$$
$$= 5.026 \times 10^{-3} \text{ m}^2$$

The cross sectional flow area of pipe 2,

$$A_{C2} = \frac{\pi}{4} D_2^2$$
$$= \frac{\pi}{4}\left(16 \times 10^{-2}\right)^2$$
$$= 20.106 \times 10^{-3} \text{ m}^2$$

Step 3 of 8

Applying the equation conservation of mass flow, we get

$$\rho V_1 A_{C1} = \rho V_2 A_{C2}$$
$$V_2 = \frac{A_{C1}}{A_{C2}} V_1$$
$$= \frac{5.026 \times 10^{-3}}{20.106 \times 10^{-3}}(10)$$
$$= 2.5 \text{ m/s}$$

The loss coefficient for sudden expansion,

$$K_L = \alpha\left(1 - \frac{A_{small}}{A_{big}}\right)^2$$
$$= 1.06\left(1 - \frac{5.026 \times 10^{-3} \text{ m}^2}{20.106 \times 10^{-3} \text{ m}^2}\right)^2$$
$$= 0.5963$$

Step 4 of 8

The head loss of the given system,

$$h_L = K_L \frac{V^2}{2g}$$

Where, g is the gravitational acceleration.

Hence the head loss,

$$h_L = K_L \frac{V_1^2}{2g}$$
$$= 0.5963\left(\frac{(10)^2}{2(9.81)}\right)$$
$$= 3.04 \text{ m}$$

Step 5 of 8

The energy equation for the expansion section in terms of heads,

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$

The pipe is horizontal,

$$z_1 = z_2$$

Here, the flow section does not involve any work devices such as pump or turbine.

The pipe is horizontal,

$$z_1 = z_2$$

Here, the flow section does not involve any work devices such as pump or turbine.

Thus,

$$h_{pump,u} = h_{turbine,e} = 0$$

The density of water, $\rho = 1000\ \text{kg/m}^3$

Step 6 of 8

Substitute the known values in the energy equation, we get

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + 0 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_1 + 0 + h_L$$

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + h_L$$

$$\frac{P_2}{\rho g} = \frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} - \alpha_2 \frac{V_2^2}{2g} - h_L$$

$$P_2 = P_1 + \rho\left(\frac{\alpha_1 V_1^2}{2} - \frac{\alpha_2 V_2^2}{2} - g h_L\right)$$

On substitute the known values in the above energy equation, we get

$$P_2 = (410\times10^3) + (1000)\left(\frac{(1.06\times10^2)}{2} - \frac{(1.06\times2.5^2)}{2} - (9.81\times3.04)\right)$$

$$= 429.865\times10^3\ \text{Pa}$$

$$= 429.865\ \text{kpa}$$

$$\approx 430\ \text{kpa}$$

Therefore, the downstream pressure, $\boxed{P_2 = 430\ \text{kPa}}$

Step 7 of 8

The actual Bernoulli's equation for the given system, we have

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

$$\frac{P_2}{\rho g} = \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 - \frac{V_2^2}{2g} - z_2$$

$$= \frac{P_1}{\rho g} + \frac{V_1^2}{2g} - \frac{V_2^2}{2g}$$

$$P_2 = P_1 + \rho\left(\frac{V_1^2}{2} - \frac{V_2^2}{2}\right)$$

Step 8 of 8

On substitute the known values in the above Bernoulli's equation, we get

$$(P_2)_{\text{Bernoulli}} = (410\times10^3) + (1000)\left(\frac{10^2}{2} - \frac{2.5^2}{2}\right)$$

$$= 456.875\times10^3\ \text{Pa}$$

$$= 456.875\ \text{kPa}$$

The error in Bernoulli's equation,

$$\text{Error} = (P_2)_{\text{Bernoulli}} - P_2$$

$$= 456.875 - 429.865$$

$$= 27\ \text{kPa}$$

Therefore the error in the Bernoulli's equation, $\boxed{\text{Error} = 27\ \text{kPa}}$

Question: 14.056P

Consider flow from a water reservoir-through a circular hole of diameter D at the side wall at a vertical distance H from the free surface. The flow rate through an actual hole with a sharp-edged entrance (KL = 0.5) is considerably less than the flow rate calculated assuming "frictionless" flow and thus zero loss for the hole. Disregarding the effect of the kinetic energy correction factor, obtain a relation for the "equivalent diameter" of the sharp-edged hole for use in frictionless flow relations.

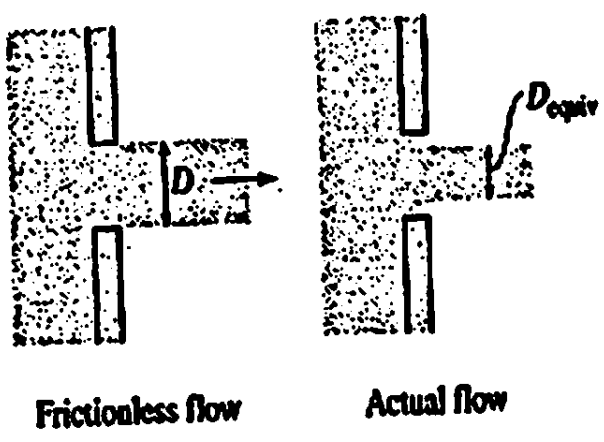


FIGURE P14–56

Step 1 of 9

Given data:

The loss coefficient, $K_L = 0.5$

The vertical distance from the hole to the free surface is H.

The diameter of the hole is D.

Step 2 of 9

The flow is assumed to be steady and incompressible.

The subscript 1 denotes the values at the free surface and subscript 2 denotes the values at the exit of the hole.

The energy equation for the two points in terms of heads is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L,$$

Step 3 of 9

Take the exit of the hole as the reference level.

Then,

$z_1 = H$

$z_2 = 0$

The pressure at both the points is equal to the atmospheric pressure. So,

$P_1 = P_2 = P_{atm}$

Step 4 of 9

Here, the flow section does not involve any work devices such as pump or turbine. Thus,

$h_{pump,s} = h_{turbine,e} = 0$

The flow velocity at the free surface,

$V_1 = 0$

The effect of the kinetic energy correction factor is disregarded. So,

$\alpha_1 = \alpha_2 = 1$

The head loss is given by $h_L = K_L \frac{V^2}{2g}$.

Step 5 of 9

Substitute the known values in the energy equation.

$$\frac{P_{atm}}{\rho g} + (1)\frac{(0)^2}{2g} + H + 0 = \frac{P_{atm}}{\rho g} + (1)\frac{V_2^2}{2g} + 0 + 0 + K_L \frac{V_2^2}{2g}$$

$$H = \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g}$$

$$= \frac{V_2^2}{2g}(1 + K_L)$$

Thus,

$$V_2 = \sqrt{\frac{2gH}{1 + K_L}}$$

Step 6 of 9

The cross sectional flow area through the hole of diameter D,

$$A_C = \frac{\pi}{4} D^2$$

The flow rate through the hole,

$$\dot{V} = V_2 A_C$$

$$= \sqrt{\frac{2gH}{1 + K_L}} \cdot \left(\frac{\pi}{4} D^2\right)$$

Step 7 of 9

The cross sectional flow area through the hole of equivalent diameter D_{equiv},

$$A_{C,equiv} = \frac{\pi}{4} D_{equiv}^2$$

For frictionless flow, $K_L = 0$.

Then, the corresponding flow velocity,

$$V_2 = \sqrt{\frac{2gH}{1 + 0}}$$

$$= \sqrt{2gH}$$

Step 8 of 9

The flow rate through the hole of equivalent diameter, D_{equiv}

$$\dot{V}_{equiv} = V_2 A_{c,equiv}$$

$$= \sqrt{2gH} \cdot \left(\frac{\pi}{4} D_{equiv}^2\right)$$

Now, the flow rate remains constant. So,

$$\dot{V} = \dot{V}_{equiv}$$

$$\sqrt{\frac{2gH}{1 + K_L}} \cdot \left(\frac{\pi}{4} D^2\right) = \sqrt{2gH} \cdot \frac{\pi}{4} D_{equiv}^2$$

Step 9 of 9

On simplifying,

$$D_{equiv}^2 = \frac{D^2}{\sqrt{1 + K_L}}$$

$$D_{equiv} = \frac{D}{(1 + K_L)^{1/4}}$$

$$= \frac{D}{(1 + 0.5)^{1/4}}$$

$$= 0.904D$$

$$\boxed{D_{equiv} = 0.904D}$$

Therefore, the relation for the equivalent diameter is $D_{equiv} = 0.904D$.

Question: 14.057P

Repeat Prob. 14-56 for a slightly rounded entrance (KL = 0.12).

Step 1 of 4

The subscript 1 denotes the values at the free surface and subscript 2 denotes the values at the exit of the hole.

Write the energy equation for steady and incompressible flow between points 1 and 2.

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \quad \text{..... (1)}$$

Here, P_1 and P_2 are the pressures at points 1 and 2 respectively, α_1 and α_2 are the kinetic energy correction factors, V_1 and V_2 are the flow velocities, z_1 and z_2 are the distances from the reference level, ρ is the density of the fluid, g is the gravitational acceleration,

$h_{pump,u}$ is the useful pump head delivered, $h_{turbine,e}$ is the turbine head extracted from the fluid and h_L is the irreversible head loss between the two points.

Step 2 of 4

Take the exit of the hole as the reference level.

Such that:

$$z_1 = H$$
$$z_2 = 0$$

The pressure at both the points is equal to the atmospheric pressure.

$$P_1 = P_2 = P_{atm}$$

Here, the flow section does not involve any work devices such as pump or turbine.

$$h_{pump,u} = h_{turbine,e} = 0$$

The flow velocity at the free surface is zero.

$$V_1 = 0$$

The effect of the kinetic energy correction factor is disregarded.

$$\alpha_1 = \alpha_2 = 1$$

State the formula for head loss.

$$h_L = K_L \frac{V^2}{2g}$$

Step 3 of 4

Substitute P_{atm} for P_1 and P_2, 1 for α_1 and α_2, 0 for V_1, H for z_1, 0 for z_2, and 0 for $h_{pump,u}$ and $h_{turbine,e}$ in the equation (1).

$$\frac{P_{atm}}{\rho g} + (1)\frac{(0)^2}{2g} + H + 0 = \frac{P_{atm}}{\rho g} + (1)\frac{V_2^2}{2g} + 0 + 0 + K_L \frac{V_2^2}{2g}$$

$$H = \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g}$$
$$= \frac{V_2^2}{2g}(1 + K_L)$$

Rewrite the equation to derive the velocity at point 2.

$$V_2 = \sqrt{\frac{2gH}{1 + K_L}} \quad \text{..... (2)}$$

State the formula for cross sectional flow area through the hole of diameter, D.

$$A_c = \frac{\pi}{4} D^2$$

State the expression for the flow rate through the hole.

$$\dot{V} = V_2 A_c$$

$$\dot{V} = \sqrt{\frac{2gH}{1 + K_L}} \cdot \frac{\pi}{4} D^2$$

Step 4 of 4

State the formula for cross sectional flow area through the hole of equivalent diameter, D_{equiv}.

$$A_{c,equiv} = \frac{\pi}{4} D_{equiv}^2$$

State the condition for frictionless flow.

$$K_L = 0.$$

Substitute 0 for K_L in the equation (2).

$$V_2 = \sqrt{\frac{2gH}{1 + 0}}$$
$$= \sqrt{2gH}$$

State the flow rate through the hole of equivalent diameter, D_{equiv}.

$$\dot{V}_{equiv} = V_2 A_{c,equiv}$$

Substitute $\sqrt{2gH}$ for V_2, $\frac{\pi}{4} D_{equiv}^2$ for $A_{c,equiv}$.

$$\dot{V}_{equiv} = \sqrt{2gH} \cdot \frac{\pi}{4} D_{equiv}^2$$

Now, equate the flow rates as it remains constant in both cases (frictionless flow and actual flow).

$$\dot{V} = \dot{V}_{equiv}$$
$$\sqrt{\frac{2gH}{1 + K_L}} \cdot \frac{\pi}{4} D^2 = \sqrt{2gH} \cdot \frac{\pi}{4} D_{equiv}^2$$

Simplify the equation to determine equivalent diameter.

$$D_{equiv}^2 = \frac{D^2}{\sqrt{1 + K_L}}$$

$$D_{equiv} = \frac{D}{(1 + K_L)^{1/4}}$$

Substitute 0.12 for K_L.

$$D_{equiv} = \frac{D}{(1 + 0.12)^{1/4}}$$
$$= 0.972D$$

Therefore, the relation for the equivalent diameter, D_{equiv} is $\boxed{0.972D}$.

Question: 14.058P

A person filling a bucket with water using a garden hose suddenly remembers that attaching a nozzle to the hose increases the discharge velocity of water and wonders if this increased velocity would decrease the filling time of the bucket. What would happen to the filling time if a nozzle were attached to the hose: increase it, decrease it, or have no effect? Why?

Step 1 of 1

From the problem, we can say that, the nozzle is connected in series to the hose. For series connection, the flow rate is constant regardless of the diameters of the individual pipes.

The flow rate is given by, $\dot{V} = VA_c$

Where, V is the flow velocity

And A_c is the cross-sectional flow area

From the above relation the velocity and the area are inversely proportional. So, even if the velocity is increased, the cross-sectional area is reduced. So the flow rate remains the same. Therefore, there is no effect in filling time when a nozzle is connected to the hose.

Question:

14.059P

Consider two identical 2-m-high open tanks filled with water on top of a 1-m-high table. The discharge valve of one of the tanks is connected to a hose whose other end is left open on the ground while the other tank does not have a hose connected to its discharge valve. Now the discharge valves of both tanks are opened. Disregarding any frictional loses in the hose, which tank do you think empties completely first? Why?

Step 1 of 4

Given data:

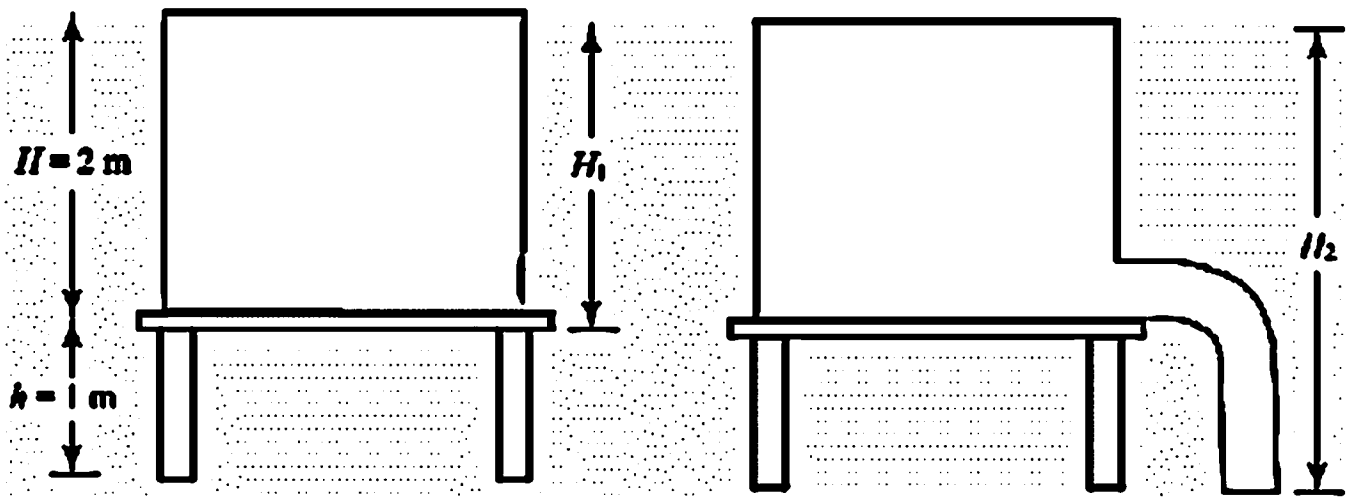
The height of the identical tanks, $H = 2\text{ m}$

The height of the table, $h = 1\text{ m}$

The frictional loss in the hose is neglected.

Step 2 of 4

The two identical tanks filled with water is shown below



Step 3 of 4

In order to find out which tank will empty first, let us calculate velocity of water of both the tanks.

From the figure,

The height of water level in first tank, $H_1 = H = 2\text{ m}$

The height of water level in the second tank,

$$H_2 = H + h$$
$$= 2\text{ m} + 1\text{ m}$$
$$= 3\text{ m}$$

Step 4 of 4

The exit velocity of water of the first tank,

$$V_{\text{tank 1}} = \sqrt{2gH_1}$$
$$= \sqrt{2\left(9.81\text{ m/s}^2\right)(2\text{ m})}$$
$$= 6.26\text{ m/s}$$

The exit velocity of water of the second tank,

$$V_{\text{tank 2}} = \sqrt{2gH_2}$$
$$= \sqrt{2\left(9.81\text{ m/s}^2\right)(3\text{ m})}$$
$$= 7.67\text{ m/s}$$

Since the exit velocity of the second tank is more than the first, the second one will empty first.

Question: 14.060P

A piping system involves two pipes of different diameters (but of identical length, material, and roughness) connected in series. How would you compare the (*a*) flow rates and (*b*) pressure drops in these two pipes?

Step 1 of 2

a) For a piping system with two pipes of different diameters connected in series, the flow rate will be constant as the pipes are of identical length, material and roughness.

Step 2 of 2

b) For a piping system with two pipes of different diameters connected in series, the pressure drop will be more for the smaller pipe compared to the bigger pipe. As the losses are more for pipe with bigger diameter.

Question: 14.061P

A piping system involves two pipes of identical diameters but of different lengths connected in parallel. How would you compare the pressure drops in these two pipes?

Step 1 of 1

For the pipes involving parallel connections, the difference in the length causes no change in the pressure drop between the pipes.

The pressure drop remains same and it does not depend on the length, diameter and roughness.

Therefore, for a piping system that involves two pipes of identical diameters and different lengths connected in parallel, the pressure drop through both pipes is the same.

Question: 14.062P

Water is pumped from a large lower reservoir to a higher reservoir. Someone claims that if the head loss is negligible, the required pump head is equal to the elevation difference between the free surfaces of the two reservoirs. Do you agree?

Step 1 of 1

Here, water is pumped from a lower reservoir to a higher reservoir and the head loss is negligible. So that the required pump head is equal to the elevation difference between the free surfaces of the two reservoirs is the correct statement.

Question: 14.063P

A piping system equipped with a pump is operating steadily. Explain how the operating point (the flow rate and the head loss) is established.

Step 1 of 1

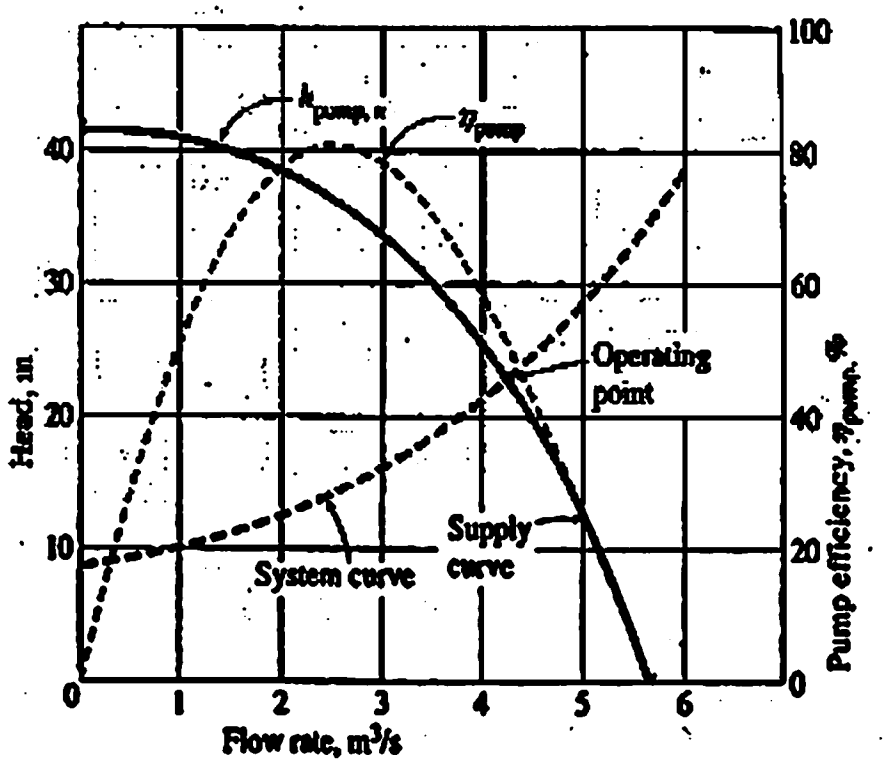
A plot of required useful pump head as a function of flow rate is called system curve. A plot of experimentally determined useful pump head as a function of flow rate is called characteristic curve. The pump installed in a piping system will operate at the point of intersection of the system curve and the characteristic curve. This point of intersection is called the operating point.

Question: 14.064P

For a piping system, define the system curve, the characteristic curve, and the operating point on a head versus flow rate chart.

Step 1 of 1

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System curve: A plot of required useful pump head as a function of flow rate is called system curve.

Characteristic curve: A plot of experimentally determined useful pump head as a function of flow rate is called characteristic curve.

Operating point: The pump installed in a piping system will operate at a point of intersection of the system curve and the characteristic curve which is called the operating point.

Question: 14.065P

The water needs of a small farm are to be met by pumping water from a well that can supply water continuously at a rate of 4 L/s. The water level in the well is 20 m f below the ground level, and water is to be pumped to a large tank on a hill, which is 58 m above the ground level of the well, using 5-cm internal diameter plastic pipes. The require length of piping is measured to be 420 m, and the total minor loss coefficient due to the use of elbows, vanes, etc. is estimated to be 12. Taking the efficiency of the pump to be 75 percent, determine the rated power of the pump that needs to be purchased, in kW. The density and viscosity of water at anticipated operation conditions are taken to be 1000 kg/m3 and 0.00131 kg/m-s, respectively. Is it wise to purchase a suitable pump that meets the total power requirements, or is it necessary to also pay particular attention to the large elevation head in this case? Explain.

Step 1 of 8

Given data:

The volume flow rate, $\dot{V} = 4\ \text{L/s} = 4 \times 10^{-3}\ \text{m}^3/\text{s}$

The diameter of the plastic pipe, $d = 5\ \text{cm} = 0.05\ \text{m}$

The water level in the well is 20 m below the ground level.

The water level in the tank is 58 m above the ground level.

The length of the piping, $L = 420\ \text{m}$

The total minor loss coefficient, $\sum K_L = 12$

The efficiency of the pump, $\eta_{\text{pump}} = 75\%$

The density of water, $\rho = 1000\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 0.00131\ \text{kg/m} \cdot \text{s}$

Step 2 of 8

In order to calculate the total power, let us calculate the pump power as follows:

The energy equation is given by,

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L \qquad \ldots\ldots (1)$$

Here, the free surface of the well is taken as the point 1 and exit of the pipe is taken as the point 2.

Let the point 1 be reference level. So, $z_1 = 0$

Then,

$z_2 = 20 + 58$

$= 78\ \text{m}$

The kinetic energy correction factor is neglected. So, $\alpha_1 = \alpha_2 = 1$

The fluid is open to atmosphere at both points 1 and 2. So, $P_1 = P_2 = P_{\text{atm}}$

At the surface of the tank and well, the velocity is negligible. Then, $V_1 = V_2 = 0$

Here, there is no turbine. So, $h_{\text{turbine},e} = 0$

Step 3 of 8

The velocity,

$$V_2 = \frac{\dot{V}}{A_c}$$

$$= \frac{4 \times 10^{-3}}{\frac{\pi}{4}(0.05)^2}$$

$$= 2.0372\ \text{m/s}$$

Step 4 of 8

The Reynolds's number,

$$\text{Re} = \frac{\rho V_2 d}{\mu}$$

$$= \frac{1000 \times 2.0372 \times 0.05}{0.00131}$$

$$= 77{,}755.725$$

Since Re > 4000, the flow is turbulent.

Step 5 of 8

For turbulent flow, the Colebrook equation is given by

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right),$$

Where,

f is the friction factor.

For plastic tube, the roughness, $\varepsilon = 0$

Substitute the known values.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0/0.05\ \text{m}}{3.7} + \frac{2.51}{(77{,}755.725)\sqrt{f}}\right)$$

$$= -2.0 \log\left(\frac{3.228 \times 10^{-5}}{\sqrt{f}}\right)$$

Solving the above equation we get,

$f = 0.01897$

Step 6 of 8

We know that, the head loss,

$$h_L = \left(f\frac{L}{d} + \sum K_L\right)\frac{V_2^2}{2g}$$

$$= \left[0.01897\left(\frac{420}{0.05}\right) + 12\right]\frac{(2.0372)^2}{2(9.81)}$$

$$= 36.245\ \text{m}$$

Step 7 of 8

By substituting these known values in equation (1), we get,

$$\frac{P_{\text{atm}}}{\rho g} + (1)\frac{0^2}{2\times 9.81} + 0 + h_{\text{pump},s} = \frac{P_{\text{atm}}}{\rho g} + (1)\frac{2.0372^2}{2\times 9.81} + 78 + 0 + 36.245$$

$$h_{\text{pump},s} = 0.212 + 78 + 36.245$$

$$= 114.457\ \text{m}$$

Step 8 of 8

Now, the pump power required,

$$\dot{W}_{\text{pump},s} = \dot{V}\rho g h_{\text{pump},s}$$

$$= \left(4 \times 10^{-3}\right) \times 1000 \times 9.81 \times 114.457$$

$$= 4491.29\ \text{W}$$

The rated power of the pump,

$$\dot{W}_{\text{pump}} = \frac{\dot{W}_{\text{pump},s}}{\eta_{\text{pump}}}$$

$$= \frac{4491.29}{0.75}$$

$$= 5988.4\ \text{W}$$

$$= 5.988\ \text{kW}$$

Therefore, the required rated power of the pump, $\boxed{\dot{W}_{\text{pump}} = 5.988\ \text{kW}}$

It is wise to buy a pump which can be used for large elevation head, so that there is no need to change the pump even if the head increases due to some reason.

Question: 14.066P

Water at 70°F flows by gravity from a large reservoir at a high elevation to a smaller one through a 90-ft-long, 2-in-diameter cast iron piping system that includes four standard flanged elbows, a well-rounded entrance, a sharp-edged exit, and a fully open gate valve. Taking the free surface of the lower reservoir as the reference level, determine the elevation z1 of the higher reservoir for a flow rate of 10 ft3/min.

Step 1 of 6

The length of the pipe, $L = 90$ ft

The diameter of the pipe, $D = 2$ in. $= 0.167$ ft

The flow rate, $\dot{V} = 10\ \text{ft}^3/\text{min} = 0.167\ \text{ft}^3/\text{s}$

Water is flowing through cast iron pipe at 70°F

Step 2 of 6

The roughness of cast iron is $\varepsilon = 0.00085$ ft

From the properties table of saturated water, at 70°F

The density of water is $\rho = 62.3\ \text{lbm/ft}^3$

The dynamic viscosity is $\mu = 6.556 \times 10^{-6}\ \text{lbm/ft-s}$

The cross-sectional flow area between the glass cover,

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.167)^2$$

$$= 0.0219\ \text{ft}^2$$

Step 3 of 6

The average velocity,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{0.167}{0.0219}$$

$$= 7.6256\ \text{ft/s}$$

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(62.3)(7.6256)(0.167)}{6.556 \times 10^{-6}}$$

$$= 121 \times 10^5$$

Since Re > 4000, the flow is turbulent.

Step 4 of 6

For turbulent flow, the Colebrook equation for friction factor (f) is

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0.00085/0.167}{3.7} + \frac{2.51}{(121 \times 10^5)\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(1.3756 \times 10^{-3} + \frac{2.07 \times 10^{-7}}{\sqrt{f}}\right)$$

Solving the equation for f:

$f = 0.03053$

Step 5 of 6

For standard flanged elbows, the loss coefficient $K_{L,\,\text{elbow}} = 0.3$

For well rounded entrance, the loss coefficient $K_{L,\,\text{entrance}} = 0.03$

For sharp edged exit, the loss coefficient $K_{L,\,\text{exit}} = 1.05$

For fully opened gate valve, the loss coefficient $K_{L,\,\text{gate}} = 0.2$

Then, total loss coefficient,

$$\sum K_L = 4K_{L,\,\text{elbow}} + K_{L,\,\text{entrance}} + K_{L,\,\text{exit}} + K_{L,\,\text{gate}}$$

$$= 4(0.3) + 0.03 + 1.05 + 0.2$$

$$= 2.48$$

The total head loss in the pipe is,

$$h_L = h_{L,\,\text{major}} + h_{L,\,\text{minor}}$$

$$= \left[f\frac{L}{D} + \sum K_L\right]\frac{V^2}{2g}$$

$$= \left[(0.03053)\frac{90}{0.167} + 2.48\right]\frac{(7.6256)^2}{2(32.2)}$$

$$= 17.1\ \text{ft}$$

Step 6 of 6

Here, the points 1 and 2 are at the free surfaces of the higher and lower reservoirs respectively. Thus, the energy equation between the two points is given by

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_L \qquad \text{..... (1)}$$

The fluid at both points is open to the atmosphere. So,

$P_1 = P_2 = P_{\text{atm}}$

The fluid velocities at both points are nearly zero.

$V_1 \approx V_2 \approx 0$

The free surface of the lower reservoir is taken as the reference point. So,

$z_2 = 0$

Thus equation (1) becomes,

$$\frac{P_{\text{atm}}}{\rho g} + \alpha_1\frac{(0)^2}{2g} + z_1 = \frac{P_{\text{atm}}}{\rho g} + \alpha_2\frac{(0)^2}{2g} + 0 + h_L$$

On simplifying,

$$z_1 = h_L$$

$$= 17.1\ \text{ft}$$

Therefore, the elevation of the higher reservoir is $\boxed{z_1 = 17.1\ \text{ft}}$

Question: 14.067P

A 2.4-m-diameter tank is initially filled with water 4 m above the center of a sharp-edged 10-cm-diameter orifice. The tank water surface is open to the atmosphere, and the orifice drains to the atmosphere. Neglecting the effect of the kinetic energy correction factor, calculate (*a*) the initial velocity from the tank and (*b*) the time required to empty the tank. Does the loss coefficient of the orifice cause a significant increase in the draining time of the tank?

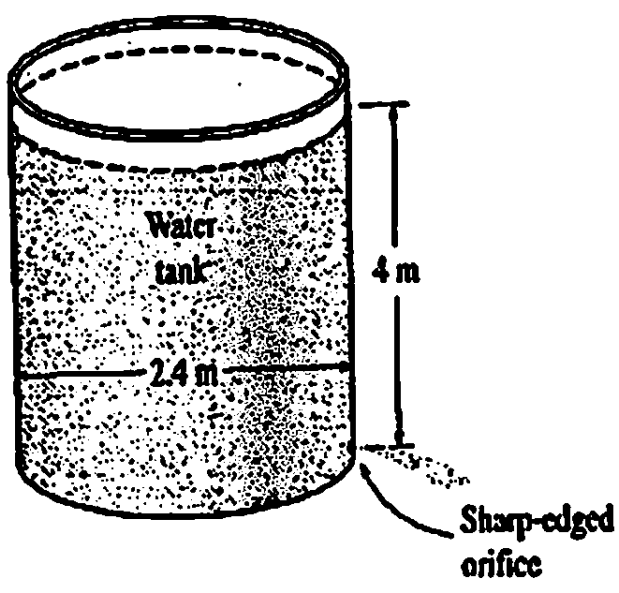


FIGURE P14–67

Step 1 of 10

Given data:

The diameter of the tank, $D = 2.4\ \text{m}$.

The initial height of the tank, $h = 4\ \text{m}$.

The diameter of the sharp edged orifice, $d = 10\ \text{cm} = 0.1\ \text{m}$.

The kinetic energy correction factor is neglected. So, $\alpha_1 = \alpha_2 = 1$.

Step 2 of 10

a)

The energy equation is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L$$

Here, free surface of the tank is taken as the point 1 and exit of the orifice is taken as the point 2.

Let the point 2 be reference level. So, $z_2 = 0$.

Step 3 of 10

The fluid is open to atmosphere at both points 1 and 2. So, $P_1 = P_2 = P_{\text{atm}}$.

At the surface of the tank, the velocity is negligible. Hence $V_1 = 0$.

Here, there is no turbine or pump. So, $h_{\text{pump},s} = h_{\text{turbine},e} = 0$.

The head loss,

$$h_L = K_L \frac{V_2^2}{2g}$$

Step 4 of 10

Then, the energy equation becomes

$$\frac{P_{\text{atm}}}{\rho g} + \frac{0^2}{2g} + z_1 + 0 = \frac{P_{\text{atm}}}{\rho g} + \frac{V_2^2}{2g} + 0 + 0 + K_L \frac{V_2^2}{2g}$$

$$z_1 = \frac{V_2^2}{2g} + K_L \frac{V_2^2}{2g}$$

$$z_1 = \frac{V_2^2}{2g}(1 + K_L)$$

$$V_2 = \sqrt{\frac{2gz_1}{1 + K_L}}$$

Step 5 of 10

Initially, the height $z_1 = h = 4\ \text{m}$.

The loss coefficient for sharp edged entrance, $K_L = 0.5$

Substitute the known values and solve for the velocity V_2.

$$V_2 = \sqrt{\frac{2(9.81\ \text{m/s}^2)(4\ \text{m})}{1 + 0.5}}$$

$$= 7.233\ \text{m/s}$$

Therefore, the initial velocity from the tank, $\boxed{V_2 = 7.233\ \text{m/s}}$.

Step 6 of 10

b)

For any time t, the velocity is given by $V_2 = \sqrt{\frac{2gz}{1 + K_L}}$.

Then, the flow rate,

$$\dot{V} = V_2 A_{\text{c. office}}$$

$$= \sqrt{\frac{2gz}{1 + K_L}} \cdot \frac{\pi}{4} d^2$$

The volume of water in tank is given by

$$dV = A_{\text{tank}}(-dz)$$

$$= -\frac{\pi D^2}{4} dz$$

The negative sign is given because the volume is decreasing when time increases.

Step 7 of 10

Then, for a differential time interval dt, the amount of flow

$$dV = \dot{V} dt$$

$$= \frac{\pi}{4} d^2 \sqrt{\frac{2gz}{1 + K_L}} \cdot dt$$

Substituting the known values

$$-\frac{\pi D^2}{4} dz = \frac{\pi}{4} d^2 \sqrt{\frac{2gz}{1 + K_L}} \cdot dt$$

$$-D^2 dz = d^2 \sqrt{\frac{2gz}{1 + K_L}} \cdot dt$$

$$dt = -\frac{D^2}{d^2} \sqrt{\frac{1 + K_L}{2gz}} \cdot dz$$

Step 8 of 10

Now, integrate the expression.

$$\int_0^t dt = \int_{z_1}^0 -\frac{D^2}{d^2} \sqrt{\frac{1 + K_L}{2gz}}\, dz$$

$$[t]_0^t = -\frac{D^2}{d^2} \sqrt{\frac{1 + K_L}{2g}} \left[\frac{z^{1/2}}{1/2}\right]_{z_1}^0$$

$$t = -\frac{D^2}{d^2} \sqrt{\frac{1 + K_L}{2g}} \left[\frac{0}{1/2} - \frac{z_1^{1/2}}{1/2}\right]$$

$$= \frac{2D^2}{d^2} \sqrt{\frac{(1 + K_L) z_1}{2g}}$$

Step 9 of 10

Then, the time required to empty the tank,

$$t = \frac{2(2.4\ \text{m})^2}{(0.1\ \text{m})^2} \sqrt{\frac{(1 + 0.5)(4\ \text{m})}{2(9.81\ \text{m/s}^2)}}$$

$$= 637.06\ \text{s}$$

$$= 10.62\ \text{min}$$

Therefore, the time required to empty the tank is $\boxed{t = 10.62\ \text{min}}$.

Step 10 of 10

If loss coefficient is neglected, the time required,

$$t = \frac{2(2.4\ \text{m})^2}{(0.1\ \text{m})^2} \sqrt{\frac{(1 + 0)(4\ \text{m})}{2(9.81\ \text{m/s}^2)}}$$

$$= 520.16\ \text{s}$$

$$= 8.67\ \text{min}$$

Thus, we got that the time required to empty the tank is less when the loss coefficient is neglected. So, the loss coefficient is significant.

Question:

A 3-m-diameter tank is initially filled with water 2 m above the center of a sharp-edged 10-cm-diameter orifice. The tank water surface is open to the atmosphere, and the orifice drains to the atmosphere through a 100-m-long pipe. The friction coefficient of the pipe is taken to be 0.015 and the effect 0f the kinetic energy correction factor can be neglected. Determine (*a*) the initial velocity from the tank and (*b*) the time required to empty the tank.

Step 1 of 9

Given data:

The diameter of the tank, $D = 3\,\text{m}$.

The initial height of the tank, $h = 2\,\text{m}$.

The diameter of the sharp edged orifice and pipe, $d = 10\,\text{cm} = 0.1\,\text{m}$.

The length of the pipe, $L = 100\,\text{m}$.

The friction coefficient of the pipe, $f = 0.015$

The kinetic energy correction factor is neglected. So, $\alpha_1 = \alpha_2 = 1$.

Step 2 of 9

a)

The energy equation is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L$$

Here, free surface of the tank is taken as the point 1 and exit of the pipe is taken as the point 2.

Let the point 2 be reference level. So, $z_2 = 0$.

Step 3 of 9

The fluid is open to atmosphere at both points 1 and 2. So, $P_1 = P_2 = P_{\text{atm}}$.

At the surface of the tank, the velocity $V_1 = 0$.

Here, there is no turbine or pump. So, $h_{\text{pump},u} = h_{\text{turbine},e} = 0$.

The head loss,

$$h_L = \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

Step 4 of 9

Then, the energy equation becomes

$$\frac{P_{\text{atm}}}{\rho g} + (1)\frac{(0)^2}{2g} + z_1 + (0) = \frac{P_{\text{atm}}}{\rho g} + (1)\frac{V_2^2}{2g} + (0) + (0) + \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

$$z_1 = \frac{V_2^2}{2g} + \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

$$z_1 = \frac{V_2^2}{2g}\left(1 + f\frac{L}{d} + K_L\right)$$

$$V_2 = \sqrt{\frac{2gz_1}{1 + f\frac{L}{d} + K_L}}$$

Step 5 of 9

Initially, the height $z_1 = h = 2\,\text{m}$.

The loss coefficient for sharp edged entrance, $K_L = 0.5$

Substitute the known values and solve for the velocity V_2.

$$V_2 = \sqrt{\frac{2(9.81\,\text{m/s}^2)(2\,\text{m})}{1 + 0.015\left(\frac{100\,\text{m}}{0.1\,\text{m}}\right) + 0.5}}$$

$$= 1.542\,\text{m/s}$$

Therefore, the initial velocity from the tank, $\boxed{V_2 = 1.542\,\text{m/s}}$.

Step 6 of 9

b)

For any time t, the velocity is given by $V_2 = \sqrt{\frac{2gz}{1 + f\frac{L}{d} + K_L}}$.

Then, the flow rate,

$$\dot{V} = V_2 A_{c,\text{pipe}}$$

$$= \sqrt{\frac{2gz}{1 + f\frac{L}{d} + K_L}} \cdot \frac{\pi}{4}d^2$$

The volume of water in tank is given by

$$dV = A_{\text{tank}}(-dz)$$

$$= -\frac{\pi D^2}{4}dz$$

The negative sign is given because the volume is decreasing when time increases.

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Step 7 of 9

Then, for a differential time interval dt, the amount of flow

Step 7 of 9

Then, for a differential time interval dt, the amount of flow

$$dV = \dot{V}dt$$

$$= \frac{\pi}{4}d^2\sqrt{\frac{2gz}{1+f\frac{L}{d}+K_L}}\cdot dt$$

Substituting the known values,

$$-\frac{\pi D^2}{4}dz = \frac{\pi}{4}d^2\sqrt{\frac{2gz}{1+f\frac{L}{d}+K_L}}\cdot dt$$

$$-D^2dz = d^2\sqrt{\frac{2gz}{1+f\frac{L}{d}+K_L}}\cdot dt$$

$$dt = -\frac{D^2}{d^2}\sqrt{\frac{1+f\frac{L}{d}+K_L}{2gz}}\cdot dz$$

Step 8 of 9

Now, integrating the expression, we get

$$\int_0^t dt = \int_{z_1}^0 -\frac{D^2}{d^2}\sqrt{\frac{1+f\frac{L}{d}+K_L}{2gz}}\,dz$$

$$[t]_0^t = -\frac{D^2}{d^2}\sqrt{\frac{1+f\frac{L}{d}+K_L}{2g}}\left[\frac{z^{1/2}}{1/2}\right]_{z_1}^0$$

$$t = -\frac{D^2}{d^2}\sqrt{\frac{1+f\frac{L}{d}+K_L}{2g}}\left[\frac{0}{1/2}-\frac{z_1^{1/2}}{1/2}\right]$$

$$= \frac{2D^2}{d^2}\sqrt{\frac{\left(1+f\frac{L}{d}+K_L\right)z_1}{2g}}$$

Step 9 of 9

Then, the time required to empty the tank,

$$t = \frac{2(3\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{\left(1+0.015\left(\frac{100\text{ m}}{0.1\text{ m}}\right)+0.5\right)(2\text{ m})}{2\left(9.81\text{ m/s}^2\right)}}$$

$$= 2334.43\text{ s}$$

$$= 38.91\text{ min}$$

Therefore, the time required to empty the tank, $\boxed{t = 38.91\text{ min}}$.

Question: 14.069P

Reconsider Prob. 14-68. In order to drain the tank faster, a pump is installed near the tank exit as in Fig. P14-69. Determine how much pump power input is necessary to establish an average water velocity of 4 m/s when the tank is full at $z = 2$ m. Also, assuming the discharge velocity to remain constant, estimate the time required to drain the tank.

Someone suggests that it makes no difference whether the pump is located at the beginning or at the end of the pipe, and that the performance will be the same in either case, but another person argues that placing the pump near the end of the pipe may cause cavitation. The water temperature is 30°C, so the water vapor pressure is $Pv = 4.246$ kPa = 0.43 m-H2O, and the system is located at sea level. Investigate if there is the possibility of cavitation and if we should be concerned about the location of the pump.

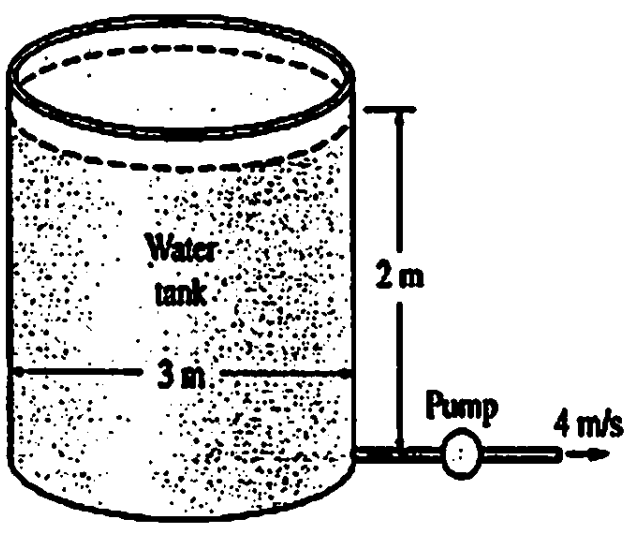


FIGURE P14-69

Step 1 of 8

Given data:

The diameter of the tank, $D = 3\,\text{m}$

The initial height of the tank, $z = h = 2\,\text{m}$

The diameter of the sharp edged orifice, $d = 10\,\text{cm} = 0.1\,\text{m}$

The length of the pipe, $L = 100\,\text{m}$

The friction coefficient of the pipe, $f = 0.015$

The kinetic energy correction factor is neglected. So, $\alpha_1 = \alpha_2 = 1$

The exit velocity, $V_2 = 4\,\text{m/s}$

Step 2 of 8

From the table of properties of water at 30°C,

The density of water at 30°C, $\rho = 996\,\text{kg/m}^3$.

The energy equation is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},a} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L$$

Here, free surface of the tank is taken as the point 1 and exit of the pipe is taken as the point 2.

Let the point 2 be reference level. So, $z_2 = 0$.

Step 3 of 8

The fluid is open to atmosphere at both points 1 and 2. So, $P_1 = P_2 = P_{\text{atm}}$.

At the surface of the tank, the velocity is $V_1 = 0$.

Here, there is no turbine. So, $h_{\text{turbine},e} = 0$.

The head loss,

$$h_L = \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

Initially, the height is $z_1 = h = 2\,\text{m}$.

Step 4 of 8

Then, the energy equation becomes

$$\frac{P_{\text{atm}}}{\rho g} + \frac{0^2}{2g} + z_1 + h_{\text{pump},a} = \frac{P_{\text{atm}}}{\rho g} + \frac{V_2^2}{2g} + 0 + 0 + \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

$$z_1 + h_{\text{pump},a} = \frac{V_2^2}{2g} + \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

$$h_{\text{pump},a} = \frac{V_2^2}{2g} + \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g} - z_1$$

$$= \left(1 + f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g} - z_1$$

$$= \left(1 + 0.015\left(\frac{100}{0.1}\right) + 0.5\right)\frac{4^2}{2\times 9.81} - 2$$

$$= 11.456\,\text{m}$$

Step 5 of 8

The mass flow rate,

$$\dot{m} = \rho A_{c,\text{pipe}} V_2$$

$$= \rho\left(\frac{\pi}{4}d^2\right)V_2$$

$$= \left(996\,\text{kg/m}^3\right)\frac{\pi}{4}(0.1\,\text{m})^2(4\,\text{m/s})$$

$$= 31.29\,\text{kg/s}$$

Now, the pump power required,

$$\dot{W}_{\text{pump},a} = \dot{m} g h_{\text{pump},a}$$

$$= (31.29\,\text{kg/s})\left(9.81\,\text{m/s}^2\right)(11.456\,\text{m})$$

$$= 3516.48\,\text{W}$$

$$= 3.516\,\text{kW}$$

Therefore, the pumping power required, $\boxed{\dot{W}_{\text{pump},a} = 3.516\,\text{kW}}$.

Step 6 of 8

Then, the flow rate is given by,

$$\dot{V} = V_2 A_{c,\text{pipe}}$$

$$= V_2\left(\frac{\pi}{4}d^2\right)$$

$$= (4\,\text{m/s})\left(\frac{\pi}{4}(0.1\,\text{m})^2\right)$$

$$= 0.0314\,\text{m}^3/\text{s}$$

Step 7 of 8

The volume of water in tank is given by

$$V = A_{\text{tank}} h$$

$$= \frac{\pi D^2}{4}h$$

$$= \frac{\pi(3\,\text{m})^2}{4}(2\,\text{m})$$

$$= 14.137\,\text{m}^3$$

Here, it is given that the discharge velocity remains constant. So, the discharge will also be constant.

Step 8 of 8

The discharge time,

$$t = \frac{V}{\dot{V}}$$

$$= \frac{14.137\,\text{m}^3}{0.0314\,\text{m}^3/\text{s}}$$

$$= 450.223\,\text{s}$$

$$= 7.504\,\text{min}$$

The required pump head of water is 11.456 m. This value exceeds the value of the atmospheric pressure at sea level (10.3 m). If the pump exit is at 1 atm, then the absolute pressure at pump inlet must be negative $(-11.4\,\text{kPa})$, which is impossible. Therefore, the system cannot work if the pump is installed near the pipe exit. Thus, the pump must be installed close to the pipe entrance.

Question: 14.070P

Water to a residential area is transported at a rate of 3 m3/s via 90-cm-internal-diameter concrete pipes with a surface roughness of 3 mm and a total length of 1500 m. In order to reduce pumping power requirements, it is proposed to line the interior surfaces of the concrete pipe with 2-cm- thick petroleum-based lining that has a surface roughness thickness of 0.04 mm. There is a concern that the reduction of pipe diameter to 86 cm and the increase in average velocity may offset any gains. Taking ρ = 1000 kg/m3 and $v = 1 \times 10\text{-}6$ m2/s for water, determine the percent increase or decrease in the pumping power requirements due to pipe frictional losses as a result of lining the concrete pipes.

Step 1 of 5

The flow rate of the water is $\dot{V} = 3\,\text{m}^3/\text{s}$

The diameter of the pipe, $D = 0.9\ \text{m}$

The surface roughness of the pipe is $\varepsilon_1 = 3\ \text{mm}$

The length of the pipe, $L = 1500\ \text{m}$

Thickness of the coating is t = 2 cm

Surface roughness of the coating is $\varepsilon_2 = 0.04\ \text{mm}$

The reduced pipe diameter, $D_r = 0.86\ \text{m}$

The density of water is $\rho = 1000\,\text{kg}/\text{m}^3$

The kinematic viscosity is $\nu = 1\times10^{-6}\,\text{m}^2/\text{s}$

Step 2 of 5

The cross sectional area of the pipe before coating is

$$A_c = \frac{\pi}{4}D^2$$
$$= \frac{\pi}{4}(0.9)^2$$
$$= 0.636\ \text{m}^2$$

The average velocity,

$$V = \frac{\dot{V}}{A_c}$$
$$= \frac{3}{0.636}$$
$$= 4.717\ \text{m/s}$$

The Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{(4.717)(0.9)}{1\times10^{-6}}$$
$$= 4245300$$

Since $\text{Re} > 4000$, the flow is turbulent.

Step 3 of 5

For turbulent flow, the Colebrook equation for friction factor (f) is

$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{\varepsilon_1/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f_1}}\right)$$
$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{0.003/0.9}{3.7} + \frac{2.51}{(4245300)\sqrt{f_1}}\right)$$
$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(9.01\times10^{-4} + \frac{5.91\times10^{-7}}{\sqrt{f_1}}\right)$$

Solving the equation for f_1 :

$$f_1 = 0.0051$$

The head loss in the pipe is,

$$h_L = f_1\frac{L}{D}\frac{V^2}{2g}$$
$$= (0.0051)\frac{1500}{0.9}\frac{4.717^2}{2\times9.81}$$
$$= 9.64\ \text{m}$$

Step 4 of 5

The cross sectional area of the pipe after coating is

$$A'_c = \frac{\pi}{4}D_r^2$$
$$= \frac{\pi}{4}(0.86)^2$$
$$= 0.581\ \text{m}^2$$

The average velocity,

$$V_r = \frac{\dot{V}}{A'_c}$$
$$= \frac{3}{0.581}$$
$$= 5.16\ \text{m/s}$$

The Reynolds number,

$$\text{Re} = \frac{V_r D_r}{\nu}$$
$$= \frac{(5.16)(0.86)}{1\times10^{-6}}$$
$$= 4437600$$

Since $\text{Re} > 4000$, the flow is turbulent.

For turbulent flow, the Colebrook equation for friction factor (f) is

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon_2/D_r}{3.7} + \frac{2.51}{\text{Re}\sqrt{f_2}}\right)$$
$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{4\times10^{-5}/0.86}{3.7} + \frac{2.51}{(4437600)\sqrt{f_2}}\right)$$
$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(1.257\times10^{-5} + \frac{5.656\times10^{-7}}{\sqrt{f_2}}\right)$$

Solving the equation for f_2 :

$$f_2 = 0.0022$$

The head loss in the pipe is,

$$h'_L = f_2\frac{L}{D_r}\frac{V_r^2}{2g}$$
$$= (0.0022)\frac{1500}{0.86}\frac{5.16^2}{2\times9.81}$$
$$= 5.207\ \text{m}$$

Step 5 of 5

By comparing the loss in pipe for both the cases, the coating will reduce the head loss and hence decreases the pumping power requirements. The percentage power reduction is

$$\%r = \frac{h'_L}{h_L}\times100$$
$$= \frac{5.207}{9.64}\times100$$
$$= 54.01\%$$

Hence the percentage decrease in the pumping power requirements is **54.01%**

Question: 14.071P

Water is transported by gravity through a 12-cm- diameter 800-m-long plastic pipe with an elevation gradient of 0.01 (i.e., an elevation drop of 1 m per 100 m of pipe length). Taking ρ = 1000 kg/m3 and v = 1 X 10-6 m2/s for water, determine the flow rate of water through the pipe. If the pipe were horizontal, what would the power requirements be to maintain the same flow rate?

Step 1 of 6

Calculate the Reynolds number by using the following equation:

$$\mathrm{Re} = \frac{Vd}{\upsilon}$$

Here, υ is the kinematic viscosity.

Substitute $0.12\,\mathrm{m}$ for d, and $1\times10^{-6}\,\mathrm{m^2/s}$ for υ.

$$\mathrm{Re} = \frac{V\times0.12}{1\times10^{-6}}$$
$$= 120000V$$

Step 2 of 6

Calculate the head loss.

$$h_L = \text{gradient}\times\text{length}$$
$$= \frac{1}{100}\times800$$
$$= 8\,\mathrm{m}$$

Write another equation of head loss.

$$h_L = \frac{fLV^2}{2gd} \quad \ldots\ldots (2)$$

Here, g is the acceleration due to gravity, L is the length of the pipe, and f is the friction factor.

Substitute $800\,\mathrm{m}$ for L, $9.81\,\mathrm{m/s^2}$ for g, $0.12\,\mathrm{m}$ for d, and $8\,\mathrm{m}$ for h_L.

$$8 = \frac{f\times800\times V^2}{2\times9.81\times0.12}$$
$$0.0235 = fV^2$$
$$V = \frac{0.153}{\sqrt{f}}$$
$$\frac{1}{\sqrt{f}} = 6.53V$$

Step 3 of 6

For smooth pipe, roughness $\varepsilon/d = 0$.

Step 4 of 6

Write the equation of friction factor for turbulent flow.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/d}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

Substitute $120000V$ for Re, 0 for ε/d, and $6.53V$ for $\frac{1}{\sqrt{f}}$.

$$6.53V = -2.0\log\left(\frac{0}{3.7} + \frac{2.51\times6.53V}{120000V}\right)$$
$$V = \frac{(-2)\times(-3.86)}{6.53}$$
$$= 1.165\,\mathrm{m/s}$$

Step 5 of 6

Calculate the volume flow rate of water.

$$Q = AV$$
$$= \frac{\pi}{4}d^2V$$

Here, V is the average velocity of the water, A is the cross sectional area of pipe, and d is the diameter of the pipe.

Substitute $0.12\,\mathrm{m}$ for d.

$$Q = \frac{\pi}{4}0.12^2V$$
$$= 0.0113V$$

Substitute $1.165\,\mathrm{m/s}$ for V.

$$Q = 0.0113\times1.165$$
$$= 0.01317\,\mathrm{m^3/s}$$

Therefore, the flow rate of water is $\boxed{0.01317\,\mathrm{m^3/s}}$.

Step 6 of 6

Calculate the required power by using the following equation.

$$W_{pump} = Q\rho g h_L$$

Substitute $0.01317\,\mathrm{m^3/s}$ for Q, $1000\,\mathrm{kg/m^3}$ for ρ, $9.81\,\mathrm{m/s^2}$ for g and $8\,\mathrm{m}$ for h_L.

$$W_{pump} = 0.01317\times1000\times9.81\times8$$
$$= 1.033\,\mathrm{kW}$$

Therefore, the power required by the pump is $\boxed{1.033\,\mathrm{kW}}$.

Question: 14.072P pg 1

Water at 15°C is drained from a large reservoir using two horizontal plastic pipes connected in series. The first pipe is 20 m long and has a 10-cm diameter, while the second pipe is 35 m long and has a 4-cm diameter. The water level in the reservoir is 18 m above the centerline of the pipe. The pipe entrance is sharp-edged, and the contraction between the two pipes is sudden. Neglecting the effect of the kinetic energy correction factor, determine the discharge rate of water from the reservoir.

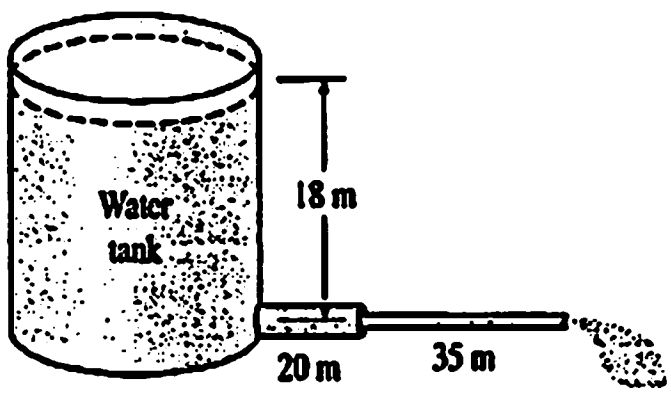


FIGURE P14–72

Step 1 of 7

Choose point 1 to be the surface of the reservoir and point 2 to be the exit of the pipe.

The energy equation between these two points:

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{pump}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{turbine}+h_L$$

Here, pressure at point 1 and 2 is P_1 and P_2, the velocity at points 1 and 2 are V_1 and V_2, acceleration due to gravity is g

Notice that the pressures at both points are atmospheric, that the fluid velocity at point 1is zero, and that the height at point 2 is zero. There is no pump or turbine.

Substitute, 0 for P_1 and P_2, 0 for V_1, 0 for h_{pump} and $h_{turbine}$

$$z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

Substitute, 18 m for z_1, $9.81\ \text{m/s}^2$ for g

$$18\ \text{m}=\frac{V_2^2}{2(9.81\ \text{m/s}^2)}+h_L$$

Step 2 of 7

Next consider the flow velocities in each pipe. Use conservation of mass to compare flow in the two portions of the pipe.

$$V_1A_1=V_2A_2$$
$$V_1=\frac{V_2A_2}{A_1}$$

Here,

$$\frac{A_2}{A_1}=\frac{d_1^2}{d_2^2}$$
$$=\frac{4^2}{10^2}$$
$$=\frac{16}{100}$$
$$=0.16$$

Therefore,

$$V_1=\frac{V_2A_2}{A_1}$$
$$=0.16V_2$$

Step 3 of 7

Consider the head loss relationship h_L by using the following equation:

$$h_L=\left(f_1\frac{L_1}{D_1}+\sum K_{L,entrance}\right)\frac{V_1^2}{2g}+\left(f_2\frac{L_2}{D_2}+\sum K_{L,contraction}\right)\frac{V_2^2}{2g}$$

Here, loss coefficient is at entrance is $\sum K_{L,entrance}$, loss coefficient is at entrance is $\sum K_{L,contraction}$,

Substitute, $20\ \text{m}$ for L_1, $0.10\ \text{m}$ for D_1, 0.5 for $\sum K_{L,entrance}$, 0.46for $\sum K_{L,contraction}$, $20\ \text{m}$ for L_2, $0.04\ \text{m}$ for D_2

$$h_L=\left(f_1\frac{L_1}{D_1}+\sum K_{L,entrance}\right)\frac{V_1^2}{2g}+\left(f_2\frac{L_2}{D_2}+\sum K_{L,contraction}\right)\frac{V_2^2}{2g}$$
$$=\left(f_1\frac{20\ \text{m}}{0.10\ \text{m}}+0.5\right)\frac{V_1^2}{2(9.81\ \text{m/s}^2)}+\left(f_2\frac{20\ \text{m}}{0.04\ \text{m}}+0.46\right)\frac{V_2^2}{2(9.81\ \text{m/s}^2)}$$

Step 4 of 7

Express the average velocity (V) at point 2 by using the following equation:

$$V_2=\frac{\dot{V}}{A_2}$$
$$=\frac{\dot{V}}{\pi D_2^2/4}$$

Here, the volume flow rate ($\dot{V}$), the cross-sectional area (A).

$$V_2=\frac{\dot{V}}{\pi(0.04\ \text{m})^2/4}$$
$$=\frac{\dot{V}}{\pi(0.04\ \text{m})^2/4}$$

Step 5 of 7

$\pi(0.04\text{ m})^2/4$

Step 5 of 7

Calculate the Reynolds numbers (Re) by using the following equation:

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu}$$

Here, the density of water is ρ, the viscosity of water is μ

The viscosity of water, $1.138\times10^{-3}\text{ kg/m}\cdot\text{s}$

Substitute, $1.138\times10^{-3}\text{ kg/m}\cdot\text{s}$ for μ, 0.10 m for D_1, $1.138\times10^{-3}\text{ kg/m}\cdot\text{s}$ for μ

$$\text{Re}_1 = \frac{(999.1\text{ kg/m}^3)V_1(0.10\text{ m})}{1.138\times10^{-3}\text{ kg/m}\cdot\text{s}}$$

Consider the second section of pipe.

$$\text{Re}_2 = \frac{\rho V_2 D_2}{\mu}$$
$$= \frac{(999.1\text{ kg/m}^3)V_2(0.04\text{ m})}{1.138\times10^{-3}\text{ kg/m}\cdot\text{s}}$$

Step 6 of 7

The friction factor (f). Recall that the roughness (ε) of the pipe is zero because the pipe is smooth.

Consider the first section of pipe:

The friction factor is:

$$\frac{1}{\sqrt{f_1}} = -2.0\log\left(\frac{\varepsilon/D_1}{3.7} + \frac{2.51}{\text{Re}_1\sqrt{f_1}}\right)$$
$$= -2.0\log\left(\frac{2.51}{\text{Re}_1\sqrt{f}}\right)$$

Here, the roughness factor is ε

Now consider the second section of pipe.

$$\frac{1}{\sqrt{f_2}} = -2.0\log\left(\frac{\varepsilon/D_2}{3.7} + \frac{2.51}{\text{Re}_2\sqrt{f_2}}\right)$$
$$= -2.0\log\left(\frac{2.51}{\text{Re}_2\sqrt{f_2}}\right)$$

Step 7 of 7

We now have a system of 8 equations and 8 unknowns. Use an equation solver such as EES to solve this system to find the volume flow rate. By using the EES software The discharge rate of water from the reservoir is $0.00595\text{ m}^3/\text{s}$.

Question:

A farmer is to pump water at 70°F from a river to a water storage tank nearby using a 125-ft-long, 5-in-diameter plastic pipe with three flanged 90° smooth bends. The water velocity near the river surface is 6 ft/s, and the pipe inlet is placed in the river normal to the flow direction of water to take advantage of the dynamic pressure. The elevation difference between the river and the free surface of the tank is 12 ft. For a flow rate of 1.5 ft3/s and an overall pump efficiency of 70 percent, determine the required electric power input to the pump.

Step 1 of 9

Apply the energy equation at inlet and exit.

$$\alpha_1 \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \alpha_2 \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \quad \text{..... (1)}$$

Here, P_1 is the inlet pressure, V_1 is the inlet velocity, z_1 is the height of the inlet from the datum, $h_{pump,u}$ is the head loss in the pump, P_2 is the exit pressure, V_2 is the exit velocity, z_2 is the height of the exit from the datum, $h_{turbine,e}$ is the heat loss at turbine exit, h_L is the head loss in the pipe, ρ is the density, and g is the acceleration due to gravity.

Let us consider the free surface river is point 1 and the position of tank is at point 2.

The water at both ends is open to atmosphere.

$$P_1 = P_{atm}$$
$$P_2 = P_{atm}$$
$$P_1 = P_2$$

The velocity at point 2 is zero.

$$V_2 = 0$$

Since, point 1 as reference level,

$$z_1 = 0$$
$$z_2 = 12\,\text{ft}$$

Step 2 of 9

Substitute P_{atm} for P_1, P_{atm} for P_2, 0 for V_2, 0 for $h_{turbine,e}$, 0 for z_1, and 12 ft for z_2 in equation (1).

$$\alpha_1 \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + h_{pump,u} = \alpha_2 \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L$$

$$\alpha_1 \frac{P_{atm}}{\rho g} + \frac{V_1^2}{2g} + (0) + h_{pump,u} = \alpha_2 \frac{P_{atm}}{\rho g} + \frac{0}{2g} + z_2 + 0 + h_L \quad \text{..... (2)}$$

$$\frac{V_1^2}{2g} + h_{pump,u} = z_2 + h_L$$

Step 3 of 9

Calculate the average velocity.

$$V = \dot{v}$$

Here, $\dot{v}$ is the volumetric flow rate, A is the cross sectional area.

Substitute $1.5\ \text{ft}^3/\text{s}$ for $\dot{v}$, $\frac{\pi}{4}D^2$ for A, and 5 in. for D.

$$V = \frac{1.5}{\left(\frac{\pi}{4}D^2\right)}$$
$$= \frac{1.5}{\frac{\pi}{4} \times \left(5\ \text{in} \times \frac{1\ \text{ft}}{12\ \text{in.}}\right)^2}$$
$$= 11\ \text{ft/s}$$

Here, D is the diameter.

Step 4 of 9

Obtain the density, dynamic viscosity, from the table A-15E, "Properties of Saturated Water", at temperature $70°\text{F}$.

Density, $\rho = 62.3\ \text{lbm/ft}^3$

Dynamic viscosity, $\mu = 6.556 \times 10^{-4}\ \text{lbm/ft} \cdot \text{s}$

Calculate the Reynolds number.

$$\text{Re} = \frac{\rho V D}{\mu}$$

Here, μ is the dynamic viscosity.

$$\text{Re} = \frac{\rho V D}{\mu}$$
$$= \frac{62.3 \times 11 \times 0.417}{6.556 \times 10^{-4}}$$
$$= 435890.93$$

Since, the value of Reynolds number is larger than 4000, the flow is turbulent.

Step 5 of 9

Obtain the roughness of the plastic pipe from the table 14.2, "Equivalent Roughness values for new commercial pipes", as zero.
The roughness of plastic pipe, $\varepsilon = 0$

Write an expression for the friction factor.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left[\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right]$$

Substitute 0 for ε, 5 in. for D.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left[\frac{0}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right]$$

Substitute 0 for ε , 5 in. for D .

$$\frac{1}{\sqrt{f}} = -2.0\log\left[\frac{0}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right]$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.51}{435890\sqrt{f}}\right)$$

$$f = 0.003$$

As the pipe is smooth for $f = 0.003$, from the table 14-21, "The friction factor is minimum for a smooth pipe, and increases with roughness", obtain the friction factor as 0.01119

$$f = 0.0119$$

Step 6 of 9

Calculate the total loss coefficient.

$$\sum K_L = K_{L,enter} + 3K_{L,elbow} + K_{L,exit} \quad \text{..... (3)}$$

Here, $K_{L,enter}$ is the entrance loss, $K_{L,elbow}$ is the loss coefficient at the smooth bends, and $K_{L,exit}$ is the exit loss.

As the pipe is smooth, the entrance losses are zero.

$$K_{L,enter} = 0$$

Obtain the loss factor at bends, from the table 14-4, "Loss coefficients of various pipe components for turbulent flow", for $90°$ flanged smooth bends.

The loss coefficient for $90°$ flanged smooth bends, $K_{L,elbow} = 0.3$

The loss coefficient at exit, $K_{L,exit} = 1.0$

Step 7 of 9

Substitute 0 for $K_{L,enter}$, 0.3 for $K_{L,elbow}$, and 1 for $K_{L,exit}$ in equation (3).

$$\begin{aligned}\sum K_L &= K_{L,enter} + 3K_{L,elbow} + K_{L,exit}\\ &= 0 + (3\times 0.3) + 1\\ &= 1.9\end{aligned}$$

Write an expression for the head loss.

$$h_L = \left(f\frac{L}{d} + K_L\right)\frac{V_2^2}{2g}$$

Substitute 0.0119 for f , 125 ft for L , 5 in for d , 1.9 for K_L , 11 ft/s for V_2 , and 32.2 ft/s² for g .

$$h_L = \left[0.0119\times\left(\frac{125}{5\text{ in.}\times\frac{1\text{ ft}}{12\text{ in}}}\right) + 1.9\right]\left[\frac{11^2}{2\times 32.2\text{ ft/s}^2}\right]$$

$$= 10.28\text{ ft}$$

Step 8 of 9

Substitute 6 ft/s for V_1 , 12 ft for z_2 , 10.28 ft for h_L , and 32.2 ft/s² for g in equation (2).

$$\frac{V_1^2}{2g} + h_{pump,u} = z_2 + h_L$$

$$\frac{6^2}{2\times 32.2} + h_{pump,u} = 12 + 10.28$$

$$h_{pump,u} = 21.72\text{ ft}$$

Step 9 of 9

Calculate the electric power required to the pump.

$$\dot{W}_{pump} = \frac{\dot{W}_{pump,u}}{\eta_{pump}}$$

Substitute $\dot{v}gh_{pump,u}$ for $\dot{W}_{pump,u}$, 1.5 ft³/s for $\dot{v}$, 32.2 ft/s² for g , 21.72 ft for $h_{pump,u}$, and 70% for η_{pump} .

$$\dot{W}_{pump} = \frac{\dot{v}gh_{pump,u}}{\eta_{pump}}$$

$$= \frac{\left(\begin{array}{l}\left(1.5\text{ ft}^3/\text{s}\right)\left(62.3\text{ lbm/ft}^3\right)\left(32.2\text{ ft/s}^2\right)\left(21.72\text{ ft}\right)\\ \times\left(\frac{1\text{ kW}}{737\text{ lbf}\cdot\text{ft/s}}\right)\times\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\end{array}\right)}{0.7}$$

$$= \frac{2.8}{0.7}$$

$$= 4\text{ kW}$$

Therefore, the required electric power input to the pump, $\boxed{4\text{ kW}}$.

Question: 14.074P

Reconsider Prob. 14-73E. Using EES (or other) software, investigate the effect of the pipe diameter on the required electric power input to the pump. Let the pipe diameter vary from 1 to 10 in, in increments of 1 in. Tabulate and plot the results, and draw conclusions.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
L=125[ft]                                   "Length of pipe"
"D_in is to be varied"                      "Diameter of pipe"
D_ft=D_in*convert(in,ft)
V_1=6[ft/s]                                 "Inlet velocity of water"
V_2=0
T=70[F]                                     "Temperature of water"
z_1=0[ft]
z_2=12[ft]
eta_pump=0.70                               "Pump efficiency"
V_dot=1.5[ft/s^3]                           "Flow rate"
n_b=3                                       "Number of 90 degrees flange bends"
K_L_b=0.3                                   "Loss Coefficient for 90 degrees flange bends"
K_L_s=1.0                                   "Sharp end"
epsilon=0.0[ft]                             "Roughness value for plastic pipe"
P=1[atm]*convert(atm,Psia)

"Properties of water at temperature T"
rho=Density(Water,T=T,P=1[atm]*convert(atm,Psia))
mu=Viscosity(Water,T=T,P=1[atm]*convert(atm,Psia))*convert(sec,hr)

P/(rho*g#)+V_1^2/(2*g#)+z_1+h_pump = P/(rho*g#)+V_2^2/(2*g#)+z_2+h_L
V=V_dot/(A_c)                               "Average Velocity in the pipe"
A_c=pi/4*D_ft^2                             "Cross-section of the pipe"

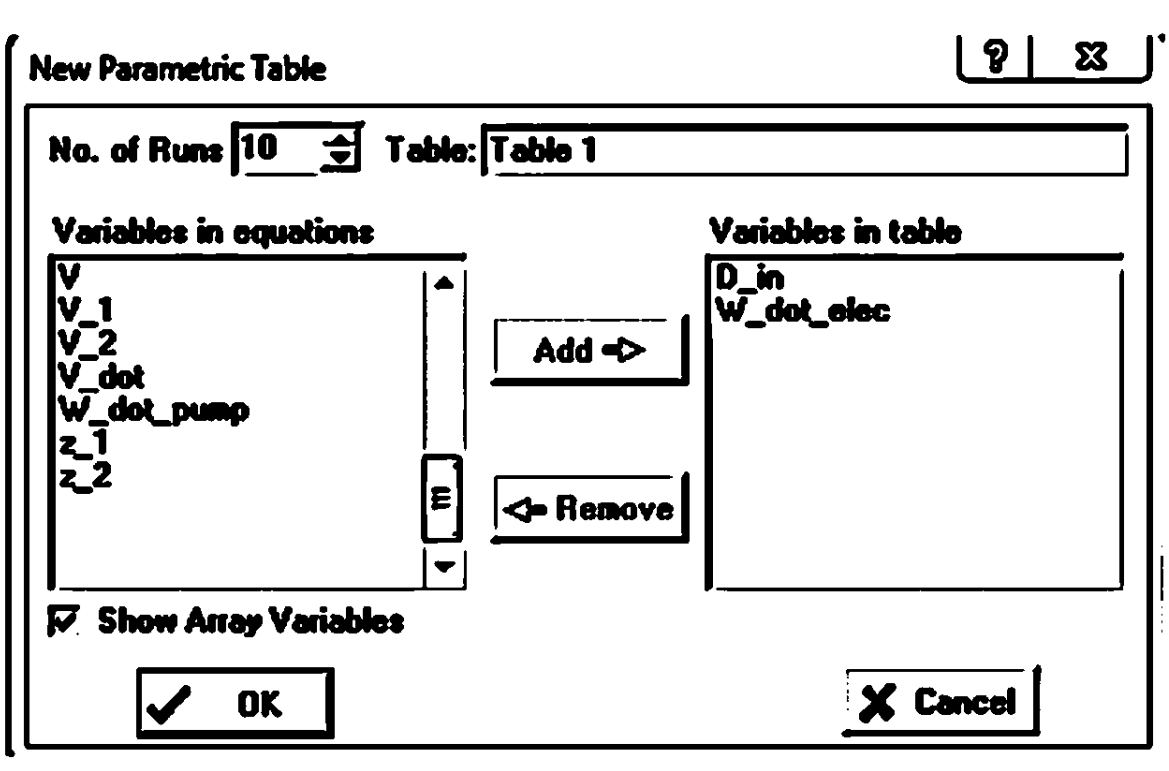
Re=rho*V*D_ft/mu                            "Reynolds number"

1/sqrt(f)=-2.0*log10((epsilon/D_ft)/3.7+2.51/(Re*sqrt(f)))

K_L=n_b*K_L_b+K_L_s
h_L=(f*L/D_ft+K_L)*V^2/2/g#
W_dot_pump=V_dot*rho*g#*h_pump
W_dot_elec*eta_pump=W_dot_pump*convert(lb_m-ft^2/s^3,kW)
```

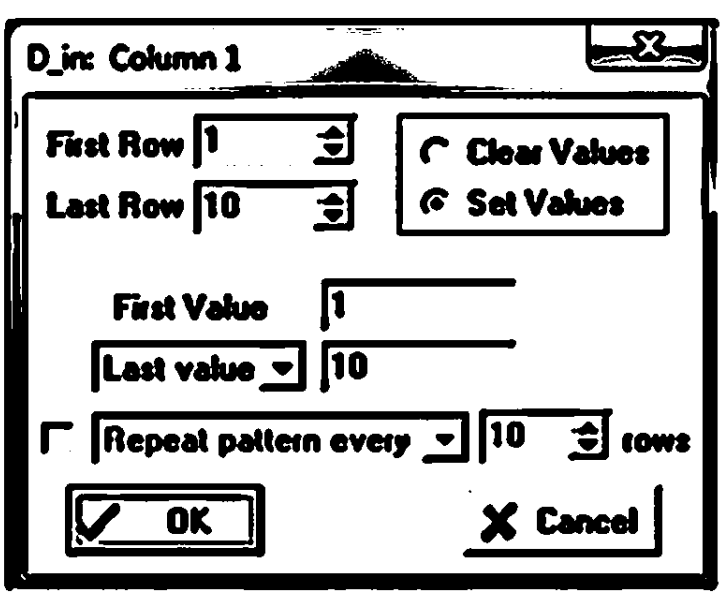
Step 2 of 5

Now to solve the equation for a range of diameter of the pipe, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 10.
Select variables in equations as "D_in" and "W_dot_elec" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

The diameter of the pipe is varied from 1 in. to 10 in.
To enter the values in the "D_in" column, click on the drop tab in "D_in" column. A new windows pops up, fill in the details as shown below and then press 'OK'



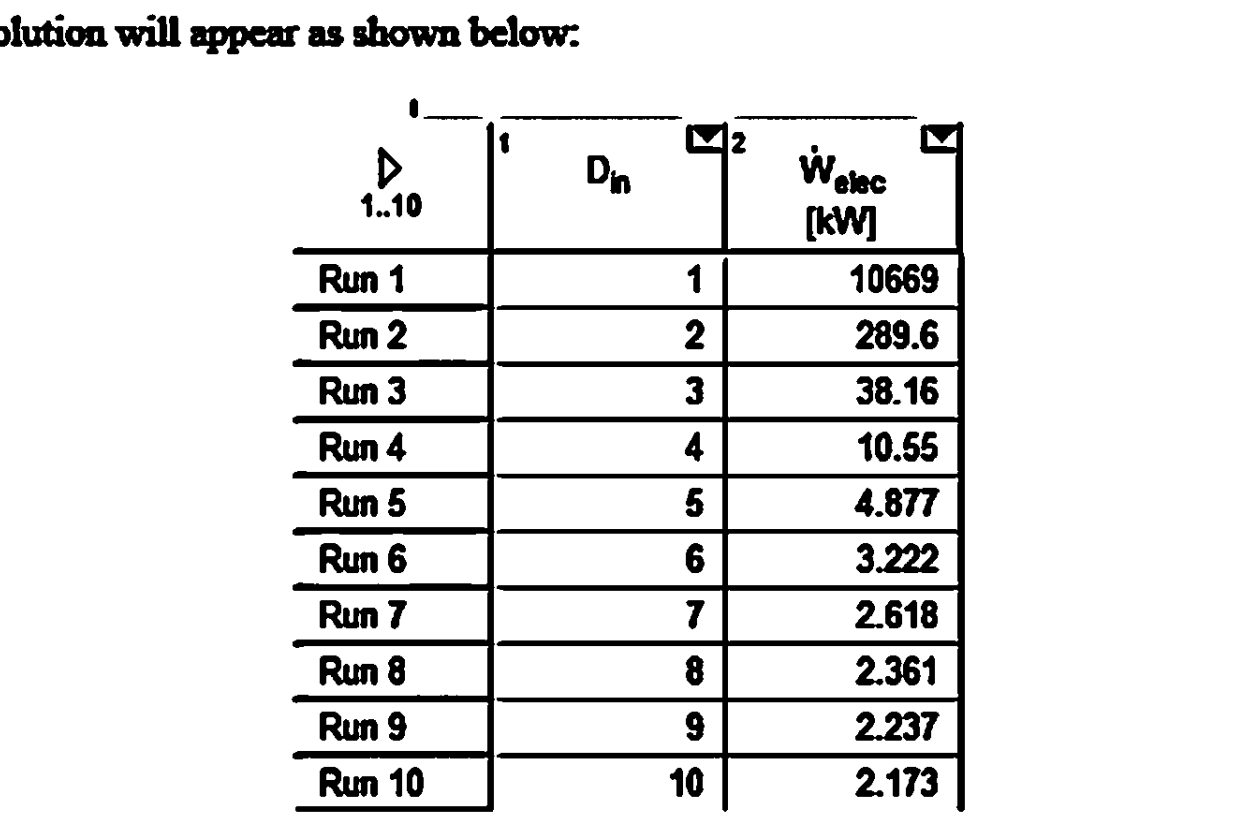
Step 4 of 5

Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

1..10	D_{in}	$\dot{W}_{elec}$ [kW]
Run 1	1	10669
Run 2	2	289.6
Run 3	3	38.16
Run 4	4	10.55
Run 5	5	4.877
Run 6	6	3.222
Run 7	7	2.618
Run 8	8	2.361
Run 9	9	2.237
Run 10	10	2.173

Step 5 of 5

To plot the input power as function of diameter of pipe, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the input power as function of average velocity of water flow is shown:



From the above plot we can interpret that as the diameter of the pipe increases, the electric power required to the pump decreases.

Question: 14.075P

A water tank filled with solar-heated water at 40°C is to be used for showers in a field using gravity-driven flow. The system includes 25 m of 1.5-cm-diameter galvanized iron piping with four miter bends (90°) without vanes and a wide-open globe valve. If water is to flow at a rate of 0.8 L/s through the shower head, determine how high the water level in the tank must be from the exit level of the shower. Disregard the losses at the entrance and at the shower head, and neglect the effect of the kinetic energy correction factor.

Step 1 of 8

Given data:

Temperature of the water, $T = 40°C$

Demeter of the galvanised pipe, $D = 1.5\,\text{cm} = 0.015\,\text{m}$

Length of the galvanised pipe, $L = 25\,\text{m}$

Rate of flow, $\dot{V} = 0.8\,\text{L/s} = 0.0008\,\text{m}^3/\text{s}$

Step 2 of 8

Let set discharge end of the pipe is reference line. hence $z_2 = 0$

By applying the energy balance equation from the water surface of the tank to discharge end of the pipe is given as

$$\frac{P_1}{\rho g} + \alpha \frac{V_1^2}{2g} + z_1 + h_{pump} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L$$

Since atmospheric pressure acting on top surface of the tank and discharge end of the pipe, therefore, $P_1 = P_2 = 0$

Velocity head is negligible at free surface of the tank.

Then, simplifying the above expression, we get

$$\frac{P_1}{\rho g} + \alpha \frac{V_1^2}{2g} + z_1 + h_{pump} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L$$

$$0 + 0 + z_1 + 0 = 0 + \alpha_2 \frac{V_2^2}{2g} + 0 + h_L$$

$$z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

Step 3 of 8

From the continuity equation, we have

$$\dot{V} = AV$$

$$V = \frac{\dot{V}}{A}$$

$$= \frac{\dot{V}}{\left(\frac{\pi}{4} D^2\right)}$$

$$= \frac{0.0008}{\frac{\pi}{4}(0.015)^2}$$

$$= 4.527\,\text{m/s}$$

Step 4 of 8

From the properties of the saturated water temperature table

At $T = 40°C$

Density $\rho = 992.1\,\text{kg/m}^3$

Kinematic viscosity $\mu = 0.653 \times 10^{-3}\,\text{kg/m} \cdot s$

Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{992.1 \times 4.52 \times 0.015}{0.653 \times 10^{-3}}$$

$$= 103008.238$$

Reynolds Number Is greater than The 4000,hence it is turbulent flow.

Step 5 of 8

From the table of roughness values for various pipes

For galvanised pipes,

$\varepsilon = 0.00015\text{m}$

Then roughness ratio is

$$\frac{\varepsilon}{D} = \frac{0.00015}{0.015}$$

$$= 0.01$$

Step 6 of 8

Now, friction factor of the pipe can be calculated by using the Moody chart,

At $\text{Re} = 103008.238$ and $\frac{\varepsilon}{D} = 0.01$

Friction factor is $f = 0.038$

From the table of various loss coefficient for

Now calculating the minor loss of the pipes at various section are

For entrance loss $K_{L\,entrance} = 0$

For elbow, $K_{elbow} = 1.1$

For flanged section, $K_{flanged} = 10$

Now, minor losses are

$$\sum K_L = K_{L\,entrance} + 4K_{elbow} + K_{flanged} + K_{exit}$$

$$= 0 + (4 \times 1.1) + 10 + 0$$

$$= 14.4\,\text{m}$$

Step 7 of 8

Major frictional loss are

$$h_f = \frac{fLV^2}{2gD}$$

$$= \frac{0.038 \times 25 \times 4.52^2}{2 \times 9.81 \times 0.015}$$

$$= 65.949$$

Step 8 of 8

Total losses are

$$h_L = h_f + \sum K_L$$

$$= 65.949 + 14.4$$

$$= 80.349\,\text{m}$$

Thus, elevation of the shower head is

$$z_1 = \alpha_2 \frac{V_2^2}{2g} + h_L$$

$$= \frac{1 \times 4.52^2}{2 \times 9.81} + 80.349$$

$$= 81.390\text{m}$$

Therefore, elevation of the water source is $\boxed{z_1 = 81.390\,\text{m}}$.

Question: 14.076P

Two water reservoirs *A* and *B* are connected to each other through a 40-m-long, 2-cm-diameter cast iron pipe with a sharp-edged entrance. The pipe also involves a swing check valve and a fully open gate valve. The water level in both reservoirs is the same, but reservoir *A* is pressurized by compressed air while reservoir *B* is open to the atmosphere at 88 kPa. If the initial flow rate through the pipe is 1.2 L/s, determine the absolute air pressure on top of reservoir *A*. Take the water temperature to be 10°C.

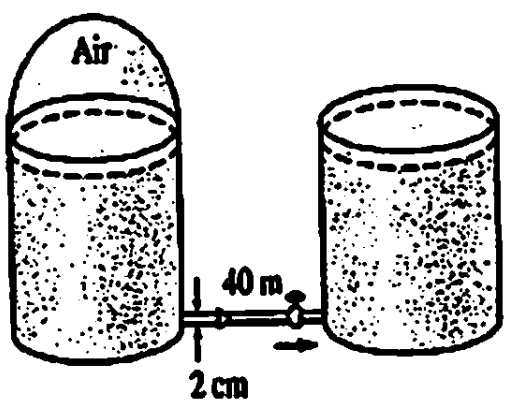


FIGURE P14–76

Step 1 of 5

Given data:

The length of the pipe, $L = 40\ \text{m}$

The diameter of the pipe, $D = 2\ \text{cm}$

$$= 2 \times 10^{-2}\ \text{m}$$

The pipe is made of cast iron.

The entrance of the pipe is sharp edged.

The pipe involves a swing check valve and fully open gate valve.

The atmospheric pressure, $P_{atm} = 88\ \text{kPa}$

The initial flow rate, $\dot{V} = 1.2\ \text{L/s}$

$$= 1.2 \times 10^{-3}\ \text{m}^3/\text{s}$$

The temperature of water is 10°C

Step 2 of 5

Assume that the flow is fully developed, incompressible and steady.

From the table of properties of saturated water, corresponding to 10°C,

The density of water, $\rho = 999.7\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 1.307 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$

The roughness of cast iron pipe, $\varepsilon = 0.26\ \text{mm} = 0.26 \times 10^{-3}\ \text{m}$

The energy equation for a control volume is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \qquad \text{..... (1)}$$

Where,

Point 1 is taken as the surface of the pressurized reservoir and point 2 is taken as the surface of the open reservoir.

Step 3 of 5

Here, both the point 2 is opened to the atmosphere. So, $P_2 = P_{atm}$

The velocity at the free surfaces, $V_1 = V_2 = 0$

Let point 2 be the reference level. Then, $z_2 = 0$ and $z_1 = H$

There is no pump and turbine. So, $h_{pump,s} = h_{turbine,e} = 0$

The average velocity of flow in the pipe,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{1.2 \times 10^{-3}\ \text{m}^3/\text{s}}{\frac{\pi}{4}\left(2 \times 10^{-2}\ \text{m}\right)^2}$$

$$= 3.82\ \text{m/s}$$

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{\left(999.7\ \text{kg/m}^3\right)\left(3.82\ \text{m/s}\right)\left(2 \times 10^{-2}\ \text{m}\right)}{1.307 \times 10^{-3}\ \text{kg/m} \cdot \text{s}}$$

$$= 58.437 \times 10^3$$

Here, $\text{Re} > 4000$, therefore the flow in the pipe is turbulent.

Step 4 of 5

For turbulent flow, the Colebrook equation is $\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$,

where, f is the friction factor.

Substitute the known values.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0.26 \times 10^{-3}\ \text{m}/2 \times 10^{-2}\ \text{m}}{3.7} + \frac{2.51}{\left(58.437 \times 10^3\right)\sqrt{f}}\right)$$

$$= -2.0 \log\left(3.5135 \times 10^{-3} + \frac{4.2952 \times 10^{-5}}{\sqrt{f}}\right)$$

Solve the equation for f.

$f = 0.0424$

The loss coefficient for sharp edged entrance, $K_{L,1} = 0.5$

The loss coefficient for swing check valve, $K_{L,2} = 2$.

The loss coefficient for fully open gate valve, $K_{L,3} = 0.2$

The loss coefficient for exit, $K_{L,4} = 1$

The total loss coefficient,

$$\sum K_L = K_{L,1} + K_{L,2} + K_{L,3} + K_{L,4}$$

$$= 0.5 + 2 + 0.2 + 1$$

$$= 3.7$$

Step 5 of 5

The head loss,

$$h_L = \left(f \frac{L}{D} + \sum K_L\right) \frac{V^2}{2g}$$

$$= \left(0.0424 \frac{40\ \text{m}}{2 \times 10^{-2}\ \text{m}} + 3.7\right) \frac{\left(3.82\ \text{m/s}\right)^2}{2\left(9.81\ \text{m/s}^2\right)}$$

$$= 65.82\ \text{m}$$

Thus, equation (1) becomes,

$$\frac{P_1}{(999.7)(9.81)} + \alpha_1 \frac{(0)^2}{2g} + 0 + 0 = \frac{88 \times 10^3}{(999.7)(9.81)} + \alpha_2 \frac{(0)^2}{2g} + 0 + 0 + 65.82$$

$$\frac{P_1}{(999.7)(9.81)} = \frac{88 \times 10^3}{(999.7)(9.81)} + 65.82$$

$$P_1 = \left(88 \times 10^3\right) + (999.7)(9.81)(65.82)$$

$$P_1 = 733.5 \times 10^3\ \text{N/m}^2$$

$$\boxed{P_1 = 733.5\ \text{kPa}}$$

Therefore, the absolute pressure on the top of reservoir *A*, $P_1 = 733.5\ \text{kPa}$

Question: 14.077P pg 1

A vented tanker is to be filled with fuel oil with ρ = 920 kg/m3 and μ = 0.045 kg/m-s from an underground reservoir using a 25-m-long, 4-cm-diameter plastic hose with a slightly rounded entrance and two 90° smooth bends. The elevation difference between the oil level in the reservoir and the top of the tanker where the hose is discharged is 5 m. The capacity of the tanker is 18 m3 and the filling time is 30 min. Taking the kinetic energy correction factor at the hose discharge to be 1.05 and assuming an overall pump efficiency of 82 percent, determine the required power input to the pump.

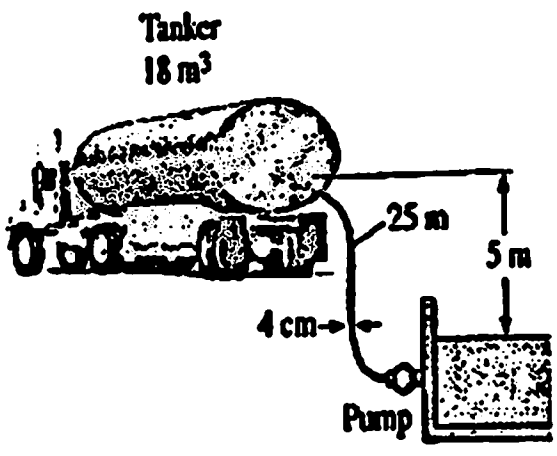


FIGURE P14–77

Step 1 of 7

Calculate the flow rate of oil by using the following equation:

$$Q = \frac{V_{tank}}{t}$$

Here, V_{tank} is the capacity of the tanker, and t is the filling time.

Substitute $18\,\text{m}^3$ for V_{tank} , $30\,\text{min}$ for t.

$$Q = \frac{18}{30\times 60}$$
$$= 0.01\,\text{m}^3/\text{s}$$

Step 2 of 7

Calculate the average velocity of the pipe by using the following equation:

$$V_{pipe} = \frac{Q}{A_p}$$
$$= \frac{Q}{\frac{\pi}{4}D^2}$$

Here, A_p is the cross sectional area of the pipe, and D is the diameter of the pipe.

Substitute $0.01\,\text{m}^3/\text{s}$ for Q , and $0.04\,\text{m}$ for d.

$$V_{pipe} = \frac{0.01}{\frac{\pi}{4}\times 0.04^2}$$
$$= 7.95\,\text{m/s}$$

Step 3 of 7

Calculate the Reynolds number by using the following equation:

$$\text{Re} = \frac{\rho V_{pipe} D}{\mu}$$

Here, ρ is the density of the oil, μ is the dynamic viscosity.

Substitute $920\,\text{kg/m}^3$ for ρ , $7.95\,\text{m/s}$ for V_{pipe} , $0.04\,\text{m}$ for D, and $0.045\,\text{kg/m}$ for μ .

$$\text{Re} = \frac{920\times 7.95\times 0.04}{0.045}$$
$$= \frac{292.56}{0.045}$$
$$= 6501.33$$

Step 4 of 7

Relative roughness of the plastic pipe is $\varepsilon/d = 0$.

From the Moody's chart for friction factor for fully developed flow in a circular pipes, obtain friction factor at $\text{Re} = 6501.33$ and $\varepsilon/d = 0$.

$$f = 0.033$$

Apply Bernoulli's equation at tank inlet and at the reservoir.

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2 - h_p + h_L$$

Here, the pressure at points (2) and (1) are P_2 and P_1 assumed as zero, substitute, 0 for P_1 and P_2 , the average velocity of pipe (1) is $\bar{V}_1$ assumed as zero and substitute, 0 for $\bar{V}_1$ and the average velocity at exit (2) is $\bar{V}_2$, the acceleration due to gravity is g, the datum height at (2) is z_2 , The datum height at (1) is z_1 assumed as zero, substitute, z_1 for 0, the head loss due to pipe is h_p , the total head loss is h_L .

$$0+0+0 = 0+\frac{V^2}{2g}+z_2-h_p+h_L$$

$$h_p = \frac{V^2}{2g}+z_2+h_L \quad \text{..... (1)}$$

Step 5 of 7

Write the equation for the total head loss.

$$h_L = \left(f\frac{L}{D}+\sum K_L\right)\frac{V^2}{2g} \quad \text{..... (2)}$$

Here, L is the length of the pipe, $\sum K_L$ is sum of the loss coefficient, g is the acceleration due to gravity.

Calculate the sum of the loss coefficient by using the following equation:

$$h_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \quad \text{..... (2)}$$ 14.077P pg 2

Here, L is the length of the pipe, $\sum K_L$ is sum of the loss coefficient, g is the acceleration due to gravity.

Calculate the sum of the loss coefficient by using the following equation:

$$\sum K_L = K_{entrance} + 2K_{bend} \quad \text{..... (3)}$$

Here, $K_{entrance}$ is the loss coefficient at entrance, and K_{bend} is the loss coefficient at bend.

From the loss coefficient of various pipe components table, obtain:

$$K_{entrance} = 0.12$$

$$K_{bend} = 0.3$$

Substitute 0.12 for $K_{entrance}$, and 0.3 for K_{bend} in (3).

$$\begin{aligned}\sum K_L &= 0.12 + (2 \times 0.3) \\ &= 0.72\end{aligned}$$

Step 6 of 7

Substitute 0.033 for f, 0.72 for $\sum K_L$, $7.95\,\text{m/s}$ for V_{pipe}, $0.04\,\text{m}$ for D, $9.81\,\text{m/s}^2$ for g, and $25\,\text{m}$ for L in equation (2).

$$\begin{aligned}h_L &= \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \\ &= \left(\left(0.033 \times \frac{25}{0.04}\right) + 0.72\right)\frac{7.95^2}{2 \times 9.81} \\ &= 68.75\,\text{m}\end{aligned}$$

Substitute $68.75\,\text{m}$ for h_L, $7.95\,\text{m/s}$ for V_{pipe}, $9.81\,\text{m/s}^2$ for g, and $5\,\text{m}$ for z_2 in equation (1).

$$\begin{aligned}h_p &= \frac{V^2}{2g} + z_2 + h_L \\ &= \frac{7.95^2}{2 \times 9.81} + 5 + 68.75 \\ &= 76.97\,\text{m}\end{aligned}$$

Step 7 of 7

Calculate the power required by the pump by using the following equation:

$$W_p = \frac{\rho g Q h_p}{\eta}$$

Here, η is pump efficiency.

Substitute $920\,\text{kg/m}^3$ for ρ, $9.81\,\text{m/s}^2$ for g, $0.01\,\text{m}^3/\text{s}$ for Q, $76.97\,\text{m}$ for h_p, and 0.82 for η.

$$\begin{aligned}W_p &= \frac{920 \times 9.81 \times 0.01 \times 76.97}{0.82} \\ &= \frac{6946.81}{0.82} \\ &= 8.471\,\text{kW}\end{aligned}$$

Therefore, the power required by the pump is $\boxed{8.471\,\text{kW}}$.

Question: 14.078P

Two pipes of identical length and material are connected in parallel. The diameter of pipe *A* is twice the diameter of pipe *B*. Assuming the friction factor to be the same in both cases and disregarding minor losses, determine the ratio of the flow rates in the two pipes.

Step 1 of 2

Write the equation for head loss for a steady, incompressible and fully developed flow.

$$h_L = f\frac{L}{D}\cdot\frac{V^2}{2g}$$

Here, f is the friction factor, L is the diameter, D is the diameter, V is the velocity, and g is the acceleration due to gravity.

Since, for a piping system connected in parallel, the head loss in the parallel pipes will be same.

$$(h_L)_A = (h_L)_B$$

$$f_A\frac{L_A}{D_A}\cdot\frac{V_A^2}{2g} = f_B\frac{L_B}{D_B}\cdot\frac{V_B^2}{2g} \quad \text{..... (1)}$$

Write the equation for velocity of flow through pipe *A*.

$$V_A = \frac{\dot{V}_A}{(A_c)_A} = \frac{\dot{V}_A}{\frac{\pi}{4}D_A^2}$$

Here, $\dot{V}_A$ is the flow rate in pipe A.

Substitute $2D_B$ for D_A.

$$V_A = \frac{\dot{V}_A}{\frac{\pi}{4}(2D_B)^2} = \frac{\dot{V}_A}{\pi D_B^2}$$

Step 2 of 2

Calculate the velocity of flow through pipe *B*.

$$V_B = \frac{\dot{V}_B}{(A_c)_B} = \frac{\dot{V}_B}{\frac{\pi}{4}D_B^2} = \frac{4\dot{V}_B}{\pi D_B^2}$$

Substitute the values in the equation (1), we get

$$f_B\frac{L_B}{2D_B}\cdot\left(\frac{\dot{V}_A}{\pi D_B^2}\right)^2\frac{1}{2g} = f_B\frac{L_B}{D_B}\cdot\left(\frac{4\dot{V}_B}{\pi D_B^2}\right)^2\frac{1}{2g}$$

$$\frac{\dot{V}_A^2}{2} = 16\dot{V}_B^2$$

$$\frac{\dot{V}_A^2}{\dot{V}_B^2} = 32$$

$$\frac{\dot{V}_A}{\dot{V}_B} = 5.656$$

Therefore, the ratio of flow rates in the two pipes is $\boxed{5.656}$.

Question: 14.079P

A certain part of cast iron piping of a water distribution system involves a parallel section. Both parallel pipes have a diameter of 30 cm, and the flow is fully turbulent. One of the branches (pipe *A*) is 1000 m long while the other branch (pipe *B*) is 3000 m long. If the flow rate through pipe *A* is 0.4 m3/s, determine the flow rate through pipe *B*. Disregard minor losses and assume the water temperature to be 15°C. Show that the flow is fully rough, and thus the friction factor is independent of Reynolds number.

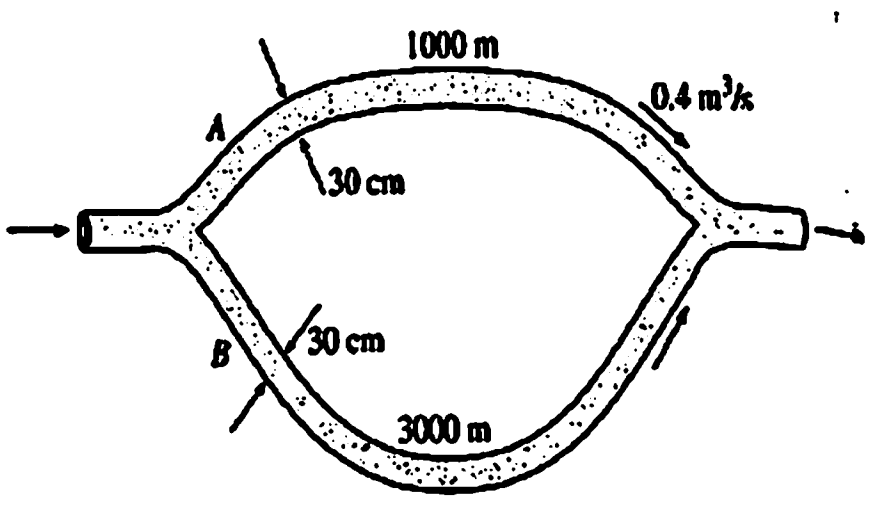


FIGURE P14–79

Step 1 of 5

Given data:

Length of the pipe *A*, $L_A = 1000$ m

Length of the pipe *B*, $L_B = 3000$ m

The diameter of the pipes, $D_A = D_B = 0.3$ m

The flow rate through the pipe *A*, $\dot{V}_A = 0.4\ \text{m}^3/s$

From the properties table of saturated water, at 15°C,

The density of water is $\rho = 999.1\ \text{kg/m}^3$

The dynamic viscosity is $\mu = 1.138 \times 10^{-3}\ \text{kg/m-s}$

Step 2 of 5

The cross sectional area of pipes *A* and *B*,

$$(A_c)_A = (A_c)_B$$
$$= \frac{\pi}{4} D_A^2$$
$$= \frac{\pi}{4}(0.3\,\text{m})^2$$
$$= 0.0707\,\text{m}^2$$

The average velocity of flow through the pipe *A*,

$$V_A = \frac{\dot{V}_A}{(A_c)_A}$$
$$= \frac{0.4\ \text{m}^3/s}{0.0707\ \text{m}^2}$$
$$= 5.658\ \text{m/s}$$

Step 3 of 5

The head loss is given by $h_L = f \frac{L}{D} \cdot \frac{V^2}{2g}$,

where, f is the friction factor,
L is the length of the pipe,
V is the average flow velocity,
D is the diameter of the pipe and
g is the gravitational acceleration.

Considering that the friction factors are same for both the pipes, the head loss will be same in the parallel piping system. Hence

$$(h_L)_A = (h_L)_B$$
$$f_A \frac{L_A}{D_A} \cdot \frac{V_A^2}{2g} = f_B \frac{L_B}{D_B} \cdot \frac{V_B^2}{2g}$$

Step 4 of 5

By substituting the values, we get

$$f_B \frac{1000}{30} \cdot \frac{(5.658)^2}{2g} = f_B \frac{3000}{30} \cdot \frac{V_B^2}{2g}$$
$$3V_B^2 = 5.658^2$$
$$V_B = 3.267\ \text{m/s}$$

The flow rate through the pipe *B*,

$$\dot{V}_B = V_B (A_c)_B$$
$$= (3.267) \times (0.0707)$$
$$= 0.231\ \text{m}^3/s$$

Therefore, the flow rate through the pipe *B*, $\boxed{\dot{V}_B = 0.231\ \text{m}^3/s}$

Step 5 of 5

The Reynolds number in pipe *A*,

$$\text{Re}_A = \frac{\rho V_A D_A}{\mu}$$
$$= \frac{(999.1\ \text{kg/m}^3)(5.658\ \text{m/s})(0.3\ \text{m})}{1.138 \times 10^{-3}\ \text{kg/m} \cdot s}$$
$$= 1.49 \times 10^6$$

Here, $\text{Re} > 4000$, therefore the flow in pipe *A* is turbulent.

The Reynolds number in pipe *B*,

$$\text{Re}_B = \frac{\rho V_B D_B}{\mu}$$
$$= \frac{(999.1\ \text{kg/m}^3)(3.267\ \text{m/s})(0.3\ \text{m})}{1.138 \times 10^{-3}\ \text{kg/m} \cdot s}$$
$$= 0.860 \times 10^6$$

Here, $\text{Re} > 4000$, therefore the flow in pipe *B* is turbulent.

Therefore, the flow in both pipes is fully turbulent, and thus the assumption that the friction factor is the same for both pipes is valid.

Question: 14.080P

Repeat Prob. 14-79 assuming pipe A has a closed gate valve ($KL = 2.1$) while pipe B has a fully open globe valve ($KL = 10$), and the other minor losses are negligible.

Step 1 of 6

Given data:

Length of the pipe A, $L_A = 1000$ m

Length of the pipe B, $L_B = 3000$ m

The diameter of the pipes, $D_A = D_B = 0.3$ m

The flow rate through the pipe A, $\dot{V}_A = 0.4\ m^3/s$

Loss coefficient of the half way closed gate valve in pipe A, $(K_L)_A = 2.1$

Loss coefficient of the fully opened globe valve in pipe B, $(K_L)_B = 10$

The roughness value of cast iron, $\varepsilon = 0.26\ mm = 2.6 \times 10^{-4}$ m

From the properties table of saturated water, at 15°C,

The density of water is $\rho = 999.1\ kg/m^3$

The dynamic viscosity is $\mu = 1.138 \times 10^{-3}$ kg/m-s

Step 2 of 6

The cross sectional area of pipes A and B,

$$(A_c)_A = (A_c)_B$$
$$= \frac{\pi}{4} D_A^2$$
$$= \frac{\pi}{4}(0.3)^2$$
$$= 0.0707\ m^2$$

The average velocity of flow through the pipe A,

$$V_A = \frac{\dot{V}_A}{(A_c)_A}$$
$$= \frac{0.4}{0.0707}$$
$$= 5.658\ m/s$$

The Reynolds number in pipe A,

$$\mathrm{Re}_A = \frac{\rho V_A D_A}{\mu}$$
$$= \frac{(999.1)(5.658)(0.3)}{1.138 \times 10^{-3}}$$
$$= 1.49 \times 10^6$$

Here, $\mathrm{Re} > 4000$, therefore the flow in pipe A is turbulent.

Step 3 of 6

For turbulent flow, the Colebrook equation is

$$\frac{1}{\sqrt{f_A}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\mathrm{Re}_A \sqrt{f_A}}\right),$$

where, f is the friction factor

ε is the roughness factor

D is the diameter of the pipe

Re is the Reynolds number of flow

Substituting the values in Colebrook equation, we get

$$\frac{1}{\sqrt{f_A}} = -2.0 \log\left(\frac{2.6 \times 10^{-4}/0.3}{3.7} + \frac{2.51}{(1.49 \times 10^6)\sqrt{f_A}}\right)$$
$$= -2.0 \log\left(2.342 \times 10^{-4} + \frac{1.685 \times 10^{-6}}{\sqrt{f_A}}\right)$$

By solving the equation for f, we get

$$f_A = 0.0192$$

Then, $f_B = f_A = 0.0192$

Step 4 of 6

The head loss is given by

$$h_L = \left(f \frac{L}{D} + K_L\right) \frac{V^2}{2g},$$

where, L is the length of the pipe,

V is the average flow velocity,

D is the diameter of the pipe and

g is the gravitational acceleration.

The head loss in pipe A,

$$(h_L)_A = \left(f_A \frac{L_A}{D_A} + (K_L)_A\right) \frac{V_A^2}{2g}$$
$$= \left((0.0192)\frac{1000}{0.3} + 2.1\right) \frac{(5.658)^2}{2(9.81)}$$
$$= 107.852\ m$$

Step 5 of 6

For a piping system, connected in parallel, the head loss in the parallel pipes is same.

$$(h_L)_A = (h_L)_B$$
$$= \left(f_B \frac{L_B}{D_B} + (K_L)_B\right) \frac{V_B^2}{2g}$$

By substituting the values, we get

$$107.852 = \left((0.0192)\frac{3000}{0.3} + 10\right) \frac{V_B^2}{2(9.81)}$$
$$107.852 = 10.296 V_B^2$$
$$V_B^2 = 10.475$$
$$V_B = 3.237\ m/s$$

Step 6 of 6

The flow rate through the pipe B,

$$\dot{V}_B = V_B (A_c)_B$$
$$= (3.237)(0.0707).$$
$$= 0.229\ m^3/s$$

Therefore, the flow rate through the pipe B, $\boxed{\dot{V}_B = 0.229\ m^3/s}$.

Question: 14.081P

Gasoline (ρ = 680 kg/m3 and v = 4.29 × 10-7m2/s) is transported at a rate of 400 L/s for a distance of 2km. The surface roughness of the piping is 0.03 mm. If the head loss due to pipe friction is not to exceed 8 m, determine the minimum diameter of the pipe.

Step 1 of 4

Given data:

Length of the pipe, $L = 2 \times 10^3 \text{ m}$

The density of gasoline, $\rho = 680 \text{kg/m}^3$

The kinematic viscosity, $\nu = 4.29 \times 10^{-7} \text{ m}^2/\text{s}$

The flow rate through the pipe, $\dot{V} = 400 \text{L/s} = 0.4 \text{m}^3/\text{s}$

The surface roughness of the piping, $\varepsilon = 3 \times 10^{-5} \text{ m}$

The head loss due to pipe friction, $h_L = 8 \text{ m}$

Step 2 of 4

The cross sectional area of pipe,

$$A_c = \frac{\pi}{4} D^2$$

The average velocity of flow through the pipe,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{4 \times 0.4}{\pi D^2}$$

$$= \frac{0.5093}{D^2}$$

The Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$

$$= \left(\frac{0.5093}{D^2}\right) \frac{D}{4.29 \times 10^{-7}}$$

$$= \frac{1.187 \times 10^6}{D}$$

Step 3 of 4

The head loss is given by the relation,

$$h_L = f \frac{L}{D} \cdot \frac{V^2}{2g}$$

where, f is the friction factor,
L is the length of the pipe,
V is the average flow velocity,
D is the diameter of the pipe and
g is the gravitational acceleration.

Substituting the known values, we get

$$8 = f \frac{(2 \times 10^3)}{D} \cdot \left(\frac{0.5093}{D^2}\right)^2 \frac{1}{2(9.81)}$$

$$8 = 26.441 \frac{f}{D^5}$$

$$f = 0.3026 \, D^5$$

Step 4 of 4

For turbulent flow, the Colebrook equation is

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Substituting the known values, we get

$$\frac{1}{\sqrt{0.3026 D^5}} = -2.0 \log\left(\frac{3 \times 10^{-5}/D}{3.7} + \frac{2.51}{\left(\frac{1.187 \times 10^6}{D}\right)\sqrt{0.3026 D^5}}\right)$$

$$= -2.0 \log\left(\frac{8.108 \times 10^{-6}}{D} + \frac{(2.115 \times 10^{-6})D}{\sqrt{0.3026 D^5}}\right)$$

Solving the equation for D, we get

$D = 0.5232 \text{ m}$

Therefore, the minimum diameter of the pipe, $\boxed{D = 0.5232 \text{m}}$

Question: 14.082P

In large buildings, hot water in a water tank is circulated through a loop so that the user doesn't have to wait for all the water in long piping to drain before hot water starts coming out. A certain recirculating loop involves 40-m-long, 1.2-cm-diameter cast iron pipes with six 90° threaded smooth bends and two fully open gate valves. If the average flow velocity through the loop is 2 m/s, determine the required power input for the recirculating pump. Take the average water temperature to be 60°C and the efficiency of the pump to be 70 percent.

Step 1 of 7

Given data:

Length of the pipe, $L = 40\,\text{m}$

The diameter of the pipe $D = 1.2\,\text{cm} = 0.012\,\text{m}$

The average flow velocity of water $V = 2\,\text{m/s}$

The average water temperature $T = 60°\text{C}$

The efficiency of a pump $\eta_{pump-motor} = 70\% = 0.7$

Step 2 of 7

From the properties of saturated water tables at 60°C temperature the given values.

The density of water, $\rho = 983.3\,\text{kg/m}^3$

The Dynamic viscosity of water, $\mu = 0.467 \times 10^{-3}\,\text{kg/m}\cdot\text{s}$

From the loss coefficient k_L of various pipe components for turbulent flow tables the given values.

For the threaded 90° smooth bent

The loss coefficient is $k_L = 0.9$

For the full open gate valve

The loss coefficient is $k_L = 0.2$

From the Equivalent roughness values for new commercial pipes table the given values

The surface roughness of the cast iron piping, $\varepsilon = 0.26\,\text{m} = 0.00026\,\text{mm}$

Step 3 of 7

The cross sectional area of pipe,

$$A_c = \frac{\pi}{4}D^2$$

The volume flow rate and the pumping power requirements are ,

$$\dot{V} = AV_c$$

$$= \frac{\pi D^2 \times 2}{4}$$

$$= \frac{\pi \times 2 \times (0.012)^2}{4}$$

$$= 0.0002261\,\text{m}^3/\text{s}$$

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{983.3 \times 2 \times 0.012}{0.467 \times 10^{-3}}$$

$$= 50533.61$$

Step 4 of 7

Which is grater than turbulent flow =4000

The relative roughness of the pipe

$$\frac{\varepsilon}{D} = \frac{0.00026}{0.12}$$

$$= 0.0217$$

For turbulent flow, the Colebrook equation is

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Substituting the known values, we get

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0.0217}{3.7} + \frac{2.51}{50533.61\sqrt{f}}\right)$$

Solving the equation for f, we get

The frictional factor $f = 0.05075$

Step 5 of 7

The total head loss is given by the relation,

$$h_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g}$$

Where, f is the friction factor,

L is the length of the pipe,

V is the average flow velocity,

D is the diameter of the pipe and

g is the gravitational acceleration.

Substituting the known values, we get

$$h_L = \left(0.05075 \times \frac{40}{0.012} + (6 \times 0.9 + 2 \times 0.2)\right)\frac{2^2}{2 \times 9.81}$$

$$= (169.17 + 5.8)0.20$$

$$= 35\text{m}$$

Step 6 of 7

The pressure drop

$$\Delta P = \Delta P_L$$

$$= \rho g h_L$$

$$= 983.3 \times 9.81 \times 35$$

$$= 337616.055\,\text{N/m}^2$$

$$= 337.616\,\text{kN/m}^2$$

Step 7 of 7

The required pumping power

$$\dot{W}_{elect} = \frac{\dot{W}_{pump}}{\eta_{pump-motor}}$$

$$= \frac{\dot{V}\Delta P}{\eta_{pump-motor}}$$

$$= \frac{0.0002261 \times 337.616}{0.7}$$

$$= 0.111\,\text{kW}$$

The required pumping power $\boxed{\dot{W}_{elect} = 0.111\,\text{kW}}$

Question: 14.083P

Reconsider Prob. 14-82. Using EES (or other) software, investigate the effect of the average flow velocity on the power input to the recirculating pump. Let the velocity vary from 0 to 3 m/s in increments of 0.3 m/s. Tabulate and plot the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

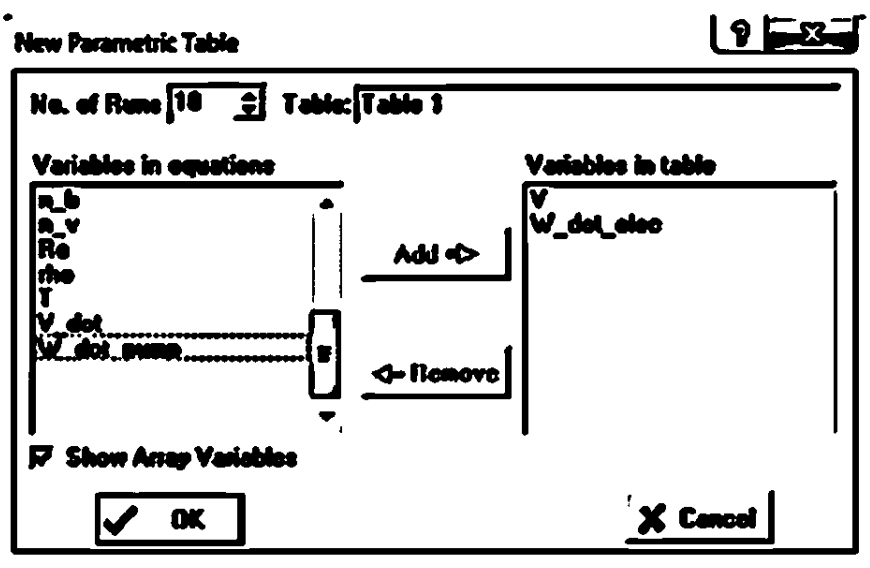
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
L=40[m]                                  "Length of pipe"
D=1.2[cm]*convert(cm,m)                  "Diameter of pipe"
"V is varied from 0 to 3[m/s]"           "Velocity of water"
T=60[C]                                  "Temperature of water"
eta_pump=0.70                            "Pump efficiency"
epsilon=0.26[mm]*convert(mm,m)           "Roughness"
n_b=6                                    "Number of bends"
n_v=2                                    "Number of valves"

"Propertied of water at temperature T"
rho=Density(Water,T=T,P=1[atm]*convert(atm,kPa))
mu=Viscosity(Water,T=T,P=1[atm]*convert(atm,kPa))
k_L_b=0.9                                "Loss Coefficient for 90 Degrees smoothe bent"
k_L_v=0.2                                "Loss Coefficient for gate valve"
A_c=pi*D^2/4
V_dot=A_c*V                              "Flow rate"
Re=rho*V*D/mu                            "Reynolds Number"
1/sqrt(f)=-2.0*log10((epsilon/D)/3.7+2.51/(Re*sqrt(f)))
h_L=(f*L/D+K_L_b*n_b+K_L_v*n_v)*V^2/2/g#
DELTAP=rho*g#*h_L                        "Pressure drop"
W_dot_pump=V_dot*DELTAP
W_dot_elec*eta_pump=W_dot_pump
```

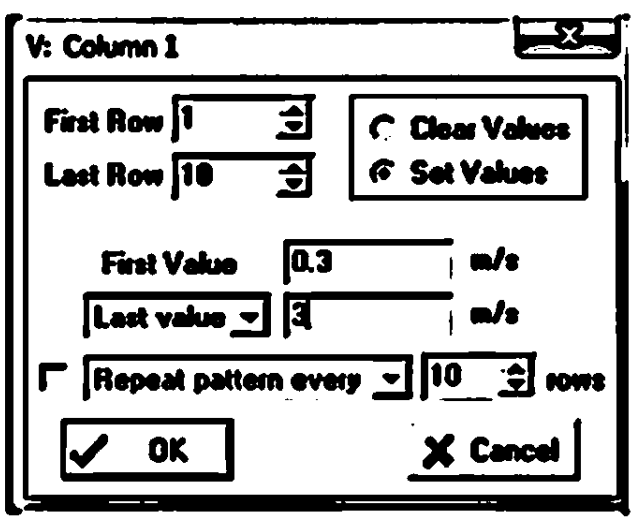
Step 2 of 5

Now to solve the equation for a range of average velocity of flow velociy, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 10.
Select variables in equations as "V" and "W_dot_elec" press 'Add' and then press 'OK'.
The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

The average velocity of flow of water is varied from 0 to 3 m/s
To enter the values in the "V" column, click on the drop tab in "V" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

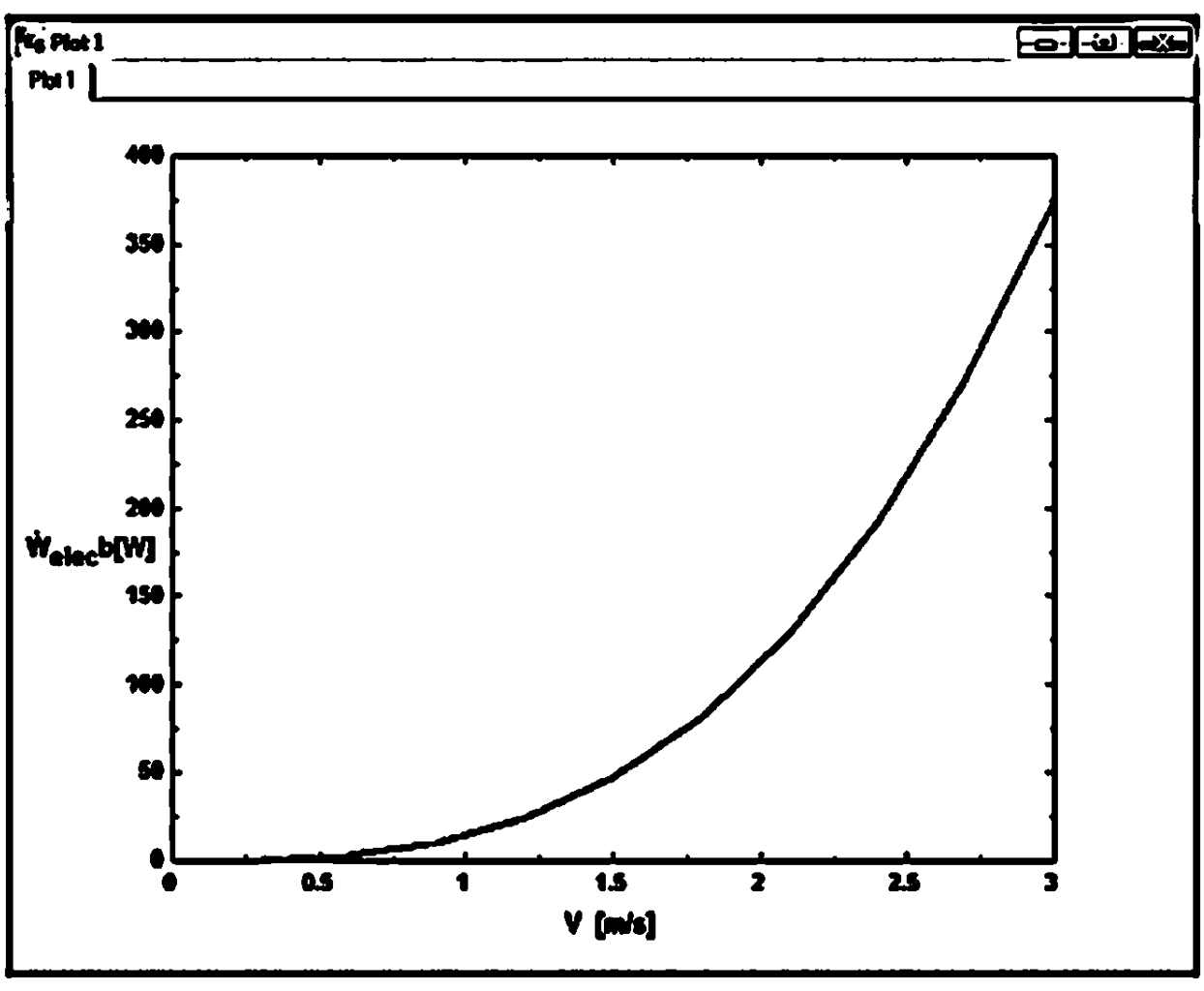
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

$\triangleright_{1..10}$	1 V [m/s]	2 $\dot{W}_{elec}$
Run 1	0.3	0.4032
Run 2	0.6	3.103
Run 3	0.9	10.32
Run 4	1.2	24.3
Run 5	1.5	47.24
Run 6	1.8	81.39
Run 7	2.1	129
Run 8	2.4	192.2
Run 9	2.7	273.3
Run 10	3	374.5

Step 5 of 5

To plot the input power as function of average velocity of water flow, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the input power as function of average velocity of water flow is shown:



From the above plot we can interpret that as the average velocity of flow of water increases the power input to the motor also increases.

Question: 14.084P

Repeat Prob. 14-82 for plastic (smooth) pipes.

Step 1 of 11

The length of cast iron pipe, $L = 40\,\text{m}$

The diameter of cast iron pipe, $D = 1.2\,\text{cm} = 0.012\,\text{m}$

The average flow velocity through the loop, $V = 2\,\text{m/s}$

The average water temperature, $T = 60°\text{C}$

The efficiency of the pump, $\eta_{\text{pump}} = 0.70$

Take density of water, $\rho = 983.3\,\text{kg/m}^3$

Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$

Step 2 of 11

From the properties of saturated water table, at temperature $T = 60°$, we have

Density, $\rho = 983.3\,\text{kg/m}^3$

Dynamic viscosity, $\mu = 0.467\times10^{-3}\,\text{kg/m}\cdot\text{s}$

From the table of Equivalent roughness values for new commercial pipes, we have

The roughness value for plastic, $\varepsilon = 0$

From the table of Valves, we have

The loss of coefficient,

For threaded 90° smooth bend, $K_L = 0.9$

For fully open gate valve, $K_L = 0.2$

The plastic pipe has six 90° threaded smooth bends and two fully open gate valves.

Step 3 of 11

The water is circulating continuously, so this system is considered as control volume.

Let take the 1 and 2 points at the same point on the pipe, then the properties (pressure, velocity and elevation) will be identical at those two points.

Therefore,

Pressure, $P_1 = P_2 = P$

Velocity, $V_1 = V_2 = V$

Elevation, $z_1 = z_2 = z$

Step 4 of 11

By using the energy equation, we have

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L$$

Where,

Turbine work is not there, so we can take $h_{\text{turbine,e}} = 0$

Step 5 of 11

Then,

$$\frac{P}{\rho g}+\alpha_1\frac{V^2}{2g}+z+h_{\text{pump,u}}=\frac{P}{\rho g}+\alpha_2\frac{V^2}{2g}+z+0+h_L$$

$$h_L = h_{\text{pump,u}}$$

$$= h_{L,\text{major}}+h_{L,\text{minor}}$$

$$=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g} \qquad \ldots\ldots (1)$$

Step 6 of 11

The flow rate of water,

$$\dot{V} = VA_c$$

$$= V\left(\frac{\pi D^2}{4}\right)$$

$$=2\left(\frac{\pi(0.012)^2}{4}\right)$$

$$=2.261\times10^{-4}\,\text{m}^3/\text{s}$$

Step 7 of 11

The Reynolds number for the flow,

$$\text{Re}=\frac{\rho VD}{\mu}$$

$$=\frac{983.3\times2\times0.012}{0.467\times10^{-3}}$$

$$=50533.618$$

Re > 4000, therefore, the flow is turbulent.

Step 8 of 11

The friction factor corresponding to the relative roughness and Reynolds number can be determined by using Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{e/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$=-2.0\log\left(\frac{0/D}{3.7}+\frac{2.51}{(50533.618)\sqrt{f}}\right)$$

$$f = 0.01908$$

Step 9 of 11

By substituting f value in equation (1), we get

The total head loss,

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$

$$=\left((0.01908)\frac{40}{0.012}+6\times0.9+2\times0.2\right)\frac{2^2}{2\times9.81}$$

$$=14.148\,\text{m}$$

Step 10 of 11

The pressure drop in the pipe,

$$\Delta P = \rho g h_L$$

$$=983.3\times9.81\times14.148$$

$$=136474.055\,\text{Pa}$$

$$=136.474\,\text{kPa}$$

Step 11 of 11

The required pumping power input for the recirculating pump,

$$\dot{W}_{\text{input}}=\frac{\dot{W}_{\text{pump}}}{\eta_{\text{pump}}}$$

$$=\frac{\dot{V}\Delta P}{\eta_{\text{pump}}}$$

$$=\frac{2.261\times10^{-4}\times136.474}{0.7}$$

$$=0.0441\,\text{kW}$$

Therefore, the required pumping power input for the recirculating pump is $\boxed{\dot{W}_{\text{input}} = 0.0441\,\text{kW}}$.

Question: 14.085P pg 1

Water at 20°C is to be pumped from a reservoir (zA = 2 m) to another reservoir at a higher elevation (zB = 9 m) through two 25-m-long plastic pipes connected in parallel. The diameters of the two pipes are 3 cm and 5 cm. Water is to be pumped by a 68 percent efficient motor-pump unit that draws 7 kW of electric power during operation. The minor losses and the head loss in the pipes that connect the parallel pipes to the two reservoirs are considered to be negligible. Determine the total flow rate between the reservoirs and the flow rates through each of the parallel pipes.

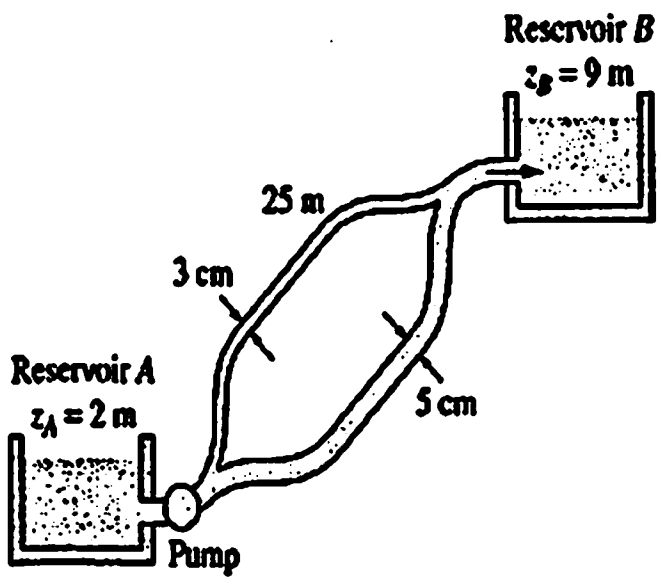


FIGURE P14–85

Step 1 of 6

From properties of saturated water tables, at temperature of $20°C$:

Density, $\rho = 998\ \text{kg/m}^3$

Dynamic viscosity, $\mu = 1.002\times10^{-3}\ \text{kg/m}\cdot\text{s}$

The electric power:

$$\dot{W} = \frac{\dot{V}\rho g h_{pump}}{\eta}$$

Here, the volume flow rate is $\dot{V}$, the density of water is, the acceleration due to gravitational is g, efficiency of the pump is *?*

Substitute, $7000\ \text{W}$ for $\dot{W}$, $998\ \text{kg/m}^3$ for *?*, $9.81\ \text{m/s}^2$ for g, 0.68 for *?*,

$$7000\ \text{W} = \frac{\dot{V}(998\ \text{kg/m}^3)(9.81\ \text{m/s}^2)h_{pump}}{0.68} \quad \text{..... (1)}$$

The energy balance equation between reservoirs is :

$$\frac{P_A}{\rho g} + \alpha_A \frac{V_A^2}{2g} + z_A + h_{pump,u} = \frac{P_B}{\rho g} + \alpha_B \frac{V_B^2}{2g} + z_B + h_L$$

Here, pressure at point 1 and 2 is P_1 and P_2, the velocity at points 1 and 2 are V_1 and V_2, acceleration due to gravity is g

Points *A* and *B* are chosen at the free surfaces of the reservoir. Since the pressures of both the liquids in the reservoir is at atmospheric pressure. Therefore, $P_A = P_B$ and the fluid velocities at both points are nearly zero, $V_A = V_B = 0$. Also z_A and z_B are 2m, 9m respectively.

$$h_{pump,u} = (z_B - z_A) + h_L \qquad h_{pump,u} = 7 + h_L \quad \text{..... (2)}$$
$$= (9-2) + h_L$$

Step 2 of 6

Since, the pipes are connected in parallel; therefore, the head loss is equal to the head loss in both the pipes.

$$h_L = h_{L,1} \quad \text{..... (3)}$$

$$h_L = h_{L,2} \quad \text{..... (4)}$$

Substitute, $0.03\ \text{m}$ for D_1 , and $0.05\ \text{m}$ for D_2 to calculate the cross-sectional area of the first pipe,

Calculate Cross-sectional area of the first pipe,

$$A_{c,1} = \frac{\pi}{4}\times D_1^2$$
$$= \frac{\pi}{4}\times 0.03^2$$
$$= 7.0685\times10^{-4}\ \text{m}^2$$

Calculate Cross-sectional area of the second pipe,

$$A_{c,2} = \frac{\pi}{4}\times D_2^2$$
$$= \frac{\pi}{4}\times 0.05^2$$
$$= 0.001\ \text{m}^2$$

Step 3 of 6

Calculate the average velocity (V) by dividing the volume flow rate ($\dot{V}$) by the cross-sectional area (A):

For First consider point 1:

$$V_1 = \frac{\dot{V}_1}{A_1}$$

$$= \frac{\dot{V}_1}{\pi(0.03\ \text{m})^2/4} \quad \text{..... (5)}$$

For point 2:

$$V_2 = \frac{\dot{V}_2}{A_2}$$

$$= \frac{\dot{V}_2}{\pi(0.03\ \text{m})^2/4} \quad \text{..... (6)}$$

$$= \frac{\dot{V}_2}{\pi (0.03\ \text{m})^2 / 4} \quad \text{..... (6)}$$

Step 4 of 6

Calculate the Reynolds numbers (Re) for first pipe by using the following equation:

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu}$$

Here, the density of water is ?, the viscosity of water is μ

The viscosity of water, $1.002 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$

Substitute, $998\ \text{kg/m}^3$ for ρ, $0.03\ \text{m}$ for D_1, and $1.002 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$ for μ

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu}$$

$$= \frac{(998\ \text{kg/m}^3) V_1 (0.03\ \text{m})}{1.002 \times 10^{-3}\ \text{kg/m} \cdot \text{s}} \quad \text{..... (7)}$$

Consider the second section of pipe.

$$\text{Re}_2 = \frac{\rho \times V_2 \times D_2}{\mu}$$

$$= \frac{998 \times V_2 \times 0.05}{1.002 \times 10^{-3}} \quad \text{..... (8)}$$

From equivalent roughness values for new commercial pipes tables, for plastic pipes, the roughness value, $\varepsilon = 0$

Frictional factor for first pipe is:

$$\frac{1}{\sqrt{f_1}} = -2.0 \log\left(\frac{\varepsilon / D_1}{3.7} + \frac{2.51}{\text{Re}_1 \sqrt{f_1}}\right)$$

$$= -2.0 \log\left(\frac{0/0.03}{3.7} + \frac{2.51}{\text{Re}_1 \sqrt{f_1}}\right)$$

$$\frac{1}{\sqrt{f_1}} = -2.0 \log\left(\frac{2.51}{\text{Re}_1 \sqrt{f_1}}\right) \quad \text{..... (9)}$$

Frictional factor for second pipe is:

$$\frac{1}{\sqrt{f_2}} = -2.0 \log\left(\frac{\varepsilon / D_2}{3.7} + \frac{2.51}{\text{Re}_2 \sqrt{f_2}}\right)$$

$$= -2.0 \log\left(\frac{0/0.05}{3.7} + \frac{2.51}{\text{Re}_2 \sqrt{f_2}}\right)$$

$$\frac{1}{\sqrt{f_2}} = -2.0 \log\left(\frac{2.51}{\text{Re}_2 \sqrt{f_2}}\right) \quad \text{..... (10)}$$

Step 5 of 6

Head loss in the first pipe is:

$$h_{L,1} = f_1 \frac{L}{D_1} \times \frac{V_1^2}{2g}$$

$$= f_1 \times \frac{25}{0.03} \times \frac{V_1^2}{2 \times 9.81}$$

$$h_{L,1} = f_1 \times V_1^2 \times \frac{25}{0.5886} \quad \text{..... (11)}$$

Head loss in the second pipe is:

$$h_{L,2} = f_2 \frac{L}{D_2} \times \frac{V_2^2}{2g}$$

$$= f_2 \times \frac{25}{0.05} \times \frac{V_2^2}{2 \times 9.81}$$

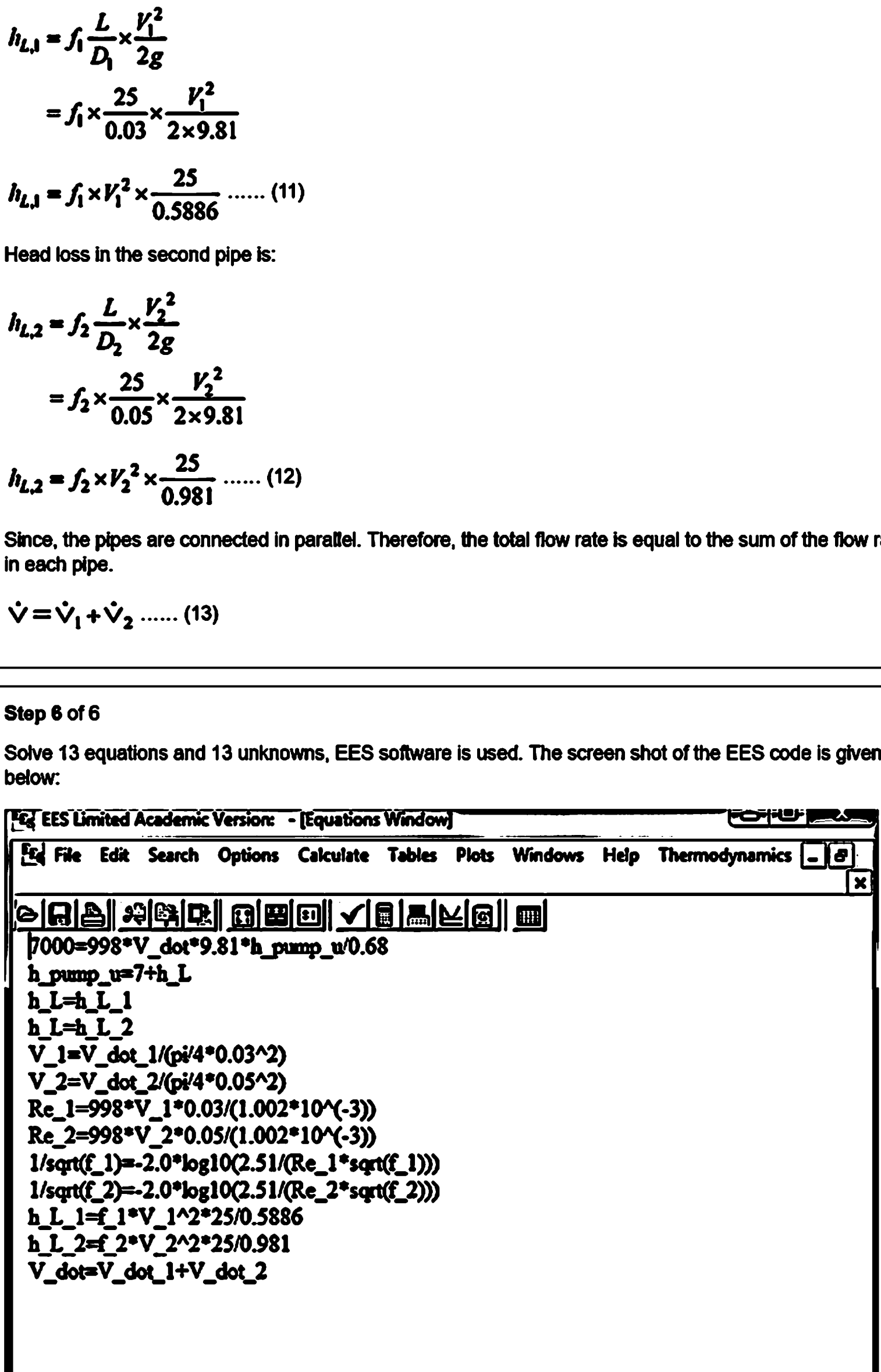
$$h_{L,2} = f_2 \times V_2^2 \times \frac{25}{0.981} \quad \text{..... (12)}$$

Since, the pipes are connected in parallel. Therefore, the total flow rate is equal to the sum of the flow rates in each pipe.

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \quad \text{..... (13)}$$

Step 6 of 6

Solve 13 equations and 13 unknowns, EES software is used. The screen shot of the EES code is given below:



```
h_L_2=f_2*V_2^2*25/0.981
V_dot=V_dot_1+V_dot_2
```


The solution of the EES code is shown below:

EES Limited Academic Version: - [Solution]

File Edit Search Options Calculate Tables Plots Windows Help Thermodynamics

Main

Unit Settings: [kJ]/[C]/[kPa]/[kg]/[degrees]

$f_1 = 0.01638$ [m] $f_2 = 0.01391$ $h_L = 19.53$ [m]

$h_{L,1} = 19.53$ [m] $h_{L,2} = 19.53$ [m] $h_{pump,u} = 26.53$ [m]

$Re_1 = 158345$ $Re_2 = 369722$ $V_1 = 5.299$ [m/s]

$V_2 = 7.424$ [m/s] $\dot{V} = 0.01832$ [m^3/s] $\dot{V}_1 = 0.003746$ [m^3/s]

$\dot{V}_2 = 0.01458$ [m^3/s]

9 potential unit problems were detected. Check Units

Calculation time = .0 sec

Therefore, the total flow rate between the reservoirs, $\dot{V} = 0.01832$ m^3/s

The flow rate of the first pipe, $\dot{V}_1 = 0.003746$ m^3/s

The flow rate of the second pipe, $\dot{V}_2 = 0.01458$ m^3/s

Question: 14.086P pg 1

Shell-and-tube heat exchangers with hundreds of tubes housed in a shell are commonly used in practice for heat transfer between two fluids. Such a heat exchanger used in an active solar hot-water system transfers heat from a water-antifreeze solution flowing through the shell and the solar collector to fresh water flowing through the tubes at an average temperature of 60°C at a rate of 15 L/s. The heat exchanger contains 80 brass tubes 1 cm in inner diameter and 1.5 m in length. Disregarding inlet, exit, and header losses, determine the pressure drop across a single tube and the pumping power required by the tube-side fluid of the heat exchanger.

After operating a long time, 1-mm-thick scale builds up on the inner surfaces with an equivalent roughness of 0.4 mm. For the same pumping power input, determine the percent reduction in the flow rate of water through the tubes.

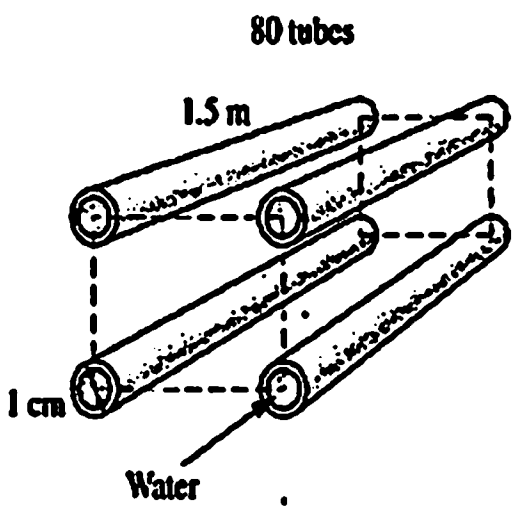


FIGURE P14–86

Step 1 of 14

The number of brass tubes, $N_{tubes} = 80$

The length of brass tubes, $L = 1.5\text{m}$

The inner diameter of brass tubes, $D = 1\text{cm} = 0.01\text{m}$

The flow rate of water, $\dot{V}_{clean} = 15\text{L/s} = 0.015\text{m/s}$

The average water temperature, $T = 60°\text{C}$

Take density of water, $\rho = 983.3\text{kg/m}^3$

Acceleration due to gravity, $g = 9.81\text{m/s}^2$

Step 2 of 14

From the properties of saturated water table, at temperature $T = 60°$, we have

Density, $\rho = 983.3\text{kg/m}^3$

Dynamic viscosity, $\mu = 0.467\times10^{-3}\,\text{kg/m}\cdot\text{s}$

From the table of Equivalent roughness values for new commercial pipes, we have

The roughness value for brass tubing, $\varepsilon = 0.0015\text{mm} = 1.5\times10^{-6}\,\text{m}$

Step 3 of 14

The average velocity of water,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{\dot{V}}{N_{tubes}\left(\frac{\pi D^2}{4}\right)}$$

$$= \frac{0.015}{80\left(\frac{\pi\times0.01^2}{4}\right)}$$

$$= 2.387\,\text{m/s}$$

Step 4 of 14

The Reynolds number for the flow,

$$\text{Re} = \frac{\rho VD}{\mu}$$

$$= \frac{983.3\times2.387\times0.01}{0.467\times10^{-3}}$$

$$= 50260$$

$\text{Re} > 4000$, therefore, the flow is turbulent.

Step 5 of 14

The relative roughness of the pipe,

$$\varepsilon/D = \frac{1.5\times10^{-6}}{0.01}$$

$$= 1.5\times10^{-4}$$

The friction factor corresponding to the relative roughness and Reynolds number can be determined by using Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$= -2.0\log\left(\frac{1.5\times10^{-4}}{3.7} + \frac{2.51}{(50260)\sqrt{f}}\right)$$

$$f = 0.0214$$

Step 6 of 14

The pressure drop in the pipe,

$$\Delta P = f\frac{L}{D}\frac{\rho V^2}{2000}$$

$$= 0.0214\times\frac{1.5}{0.01}\times\frac{983.3\times2.387^2}{2000}$$

$$= 8.99\text{kPa}$$

Therefore, the pressure drop in the pipe is $\boxed{\Delta P = 8.99\text{kPa}}$.

Step 7 of 14

The required pumping power input,

$$\dot{W}_{pump} = \dot{V}\Delta P$$

$$= 0.015\times8.99$$

$$= 0.13485\text{kW}$$

Therefore, the required pumping power input for overcoming the3 frictional losses

The required pumping power input

$$\dot{W}_{pump} = \dot{V}\Delta P$$
$$= 0.015 \times 8.99$$
$$= 0.13485\text{kW}$$

Therefore, the required pumping power input for overcoming the3 frictional losses is $\boxed{\dot{W}_{input} = 0.13485\text{kW}}$.

Step 8 of 14

If the 1mm thick scale builds up on the inner surface of the tubes then the 1cm diameter becomes 0.8cm .

Equivalent roughness value, $\varepsilon = 0.4\text{mm} = 0.0004\text{m}$

Assume that the flow rate and average velocity are unknowns.

Step 9 of 14

The average velocity,

$$V = \frac{\dot{V}}{A_c}$$
$$= \frac{\dot{V}}{N_{tubes}\left(\frac{\pi D^2}{4}\right)}$$
$$= \frac{\dot{V}}{80\left(\frac{\pi(0.008)^2}{4}\right)}$$
$$V = 248.679\dot{V}\ \text{m/s} \qquad \ldots\ldots (1)$$

Step 10 of 14

The Reynolds number,

$$\text{Re} = \frac{\rho VD}{\mu}$$
$$= \frac{983.3 \times V \times 0.008D}{0.467 \times 10^{-3}}$$
$$\text{Re} = 16844.539VD \qquad \ldots\ldots (2)$$

Step 11 of 14

The relative roughness of the pipe,

$$\varepsilon/D = \frac{0.0004}{0.008}$$
$$= 0.05$$

The friction factor corresponding to the relative roughness and Reynolds number can be determined by using Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$
$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0.05}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \qquad \ldots\ldots (3)$$

Step 12 of 14

The pressure drop in the pipe,

$$\Delta P = f\frac{L}{D}\frac{\rho V^2}{2000}$$
$$= f \times \frac{1.5}{0.008} \times \frac{983.3 \times V^2}{2000}$$
$$\Delta P = 92.184V^2\ \text{kPa} \qquad \ldots\ldots (4)$$

The use full power input,

$$\dot{W}_{input} = 0.13485\text{kW} \qquad \ldots\ldots (5)$$

Step 13 of 14

We have five equations and five unknowns, by using trial and error approach and by assuming a velocity value, we get the unknown values as

Friction factor, $f = 0.0723$

Reynolds number, $\text{Re} = 28870$

Average velocity, $V = 1.714\text{m/s}$

Pressure drop, $\Delta P = 19.6\text{kPa}$

Flow rate, $\dot{V}_{dirty} = 0.00689\text{m}^3/\text{s} = 6.89\text{L/s}$

Step 14 of 14

The percentage reduction in the flow rate,

$$\text{Reduction ration} = \frac{\dot{V}_{clean} - \dot{V}_{dirty}}{\dot{V}_{clean}}$$
$$= \frac{15 - 6.89}{15}$$
$$= 0.54$$
$$= 54\%$$

Therefore, 54% percentage of flow rate is reduced due to the formation of scale.

Question: 14.087P

The compressed air requirements of a manufacturing facility are met by a 150-hp compressor that draws in air from the outside through an 12-m-long, 18-cm-diameter duct made of thin galvanized iron sheets. The compressor takes in air at a rate of 0.27 m3/s at the outdoor conditions of 15°C and 95 kPa. Disregarding any minor losses, determine the useful power used by the compressor to overcome the frictional losses in this duct.

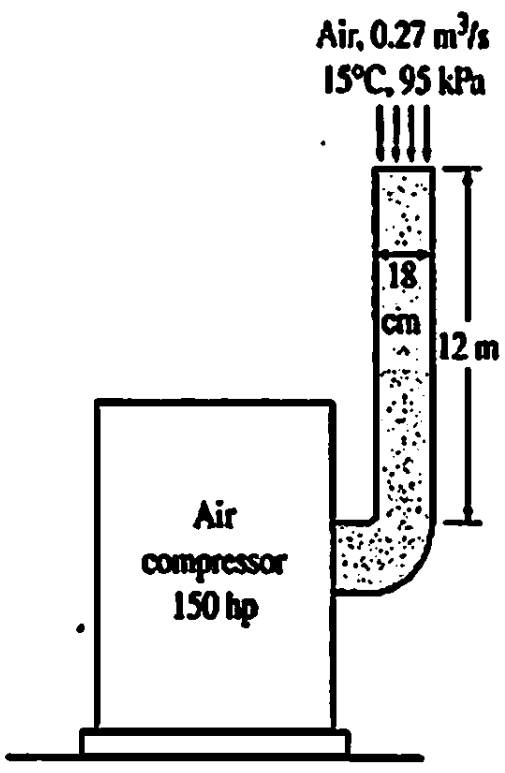


FIGURE P14–87

Step 1 of 5

Given data:

The length of the duct, $L = 12\ \text{m}$

The diameter of the duct, $D = 0.18\ \text{m}$

The rate of air intake, $\dot{V} = 0.27\ \text{m}^3/s$

The outside temperature of the air, $T = 15°\text{C}$

The outside pressure, $P = 95000\ \text{Pa}$

From the properties table of air at 1 atm (101.3 kPa) and 15°C,

The density, $\rho' = 1.225\ \text{kg}/\text{m}^3$.

The dynamic viscosity, $\mu' = 1.802 \times 10^{-5}\ \text{kg}/\text{m} \cdot s$.

Step 2 of 5

Then, the density at 95 kPa,

$$\rho = \frac{P}{P'}\rho'$$

$$= \frac{95}{101.3}(1.225)$$

$$= 1.149\ \text{kg}/\text{m}^3$$

The dynamic viscosity is independent of pressure.

So, the dynamic viscosity at 95 kPa is $\mu = 1.802 \times 10^{-5}\ \text{kg}/\text{m} \cdot s$

For galvanized iron, the roughness is $\varepsilon = 1.5 \times 10^{-4}\ \text{m}$.

The cross-sectional flow area between the glass cover,

$$A_t = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}(0.18)^2$$

$$= 25.45 \times 10^{-3}\ \text{m}^2$$

Step 3 of 5

The average velocity,

$$V = \frac{\dot{V}}{A_t}$$

$$= \frac{0.27}{25.45 \times 10^{-3}}$$

$$= 10.61\ \text{m}/s$$

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(1.149)(10.61)\left(18 \times 10^{-2}\right)}{1.802 \times 10^{-5}}$$

$$= 121773.6$$

Since Re > 4000, the flow is turbulent.

Step 4 of 5

For turbulent flow, the friction factor (f) according to Colebrook equation is

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$= -2.0 \log\left(\frac{1.5 \times 10^{-4}/0.18}{3.7} + \frac{2.51}{121773.6\sqrt{f}}\right)$$

$$= -2.0 \log\left(0.2252 \times 10^{-3} + \frac{20.61 \times 10^{-6}}{\sqrt{f}}\right)$$

Solve the equation for f, we get

$$f = 0.0212$$

Step 5 of 5

The pressure drop,

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2}$$

$$= 0.0212\left[\frac{12}{0.18}\right]\left[\frac{(1.149)(10.61)^2}{2}\right]$$

$$= 91.4\ \text{Pa}$$

The power required,

$$\dot{W}_{\text{pump}} = \dot{V} \times \Delta P$$

$$= (0.27)(91.4)$$

$$= 24.678\ \text{W}$$

Therefore, the power used to overcome the frictional losses is $\boxed{\dot{W}_{\text{pump}} = 24.678\ \text{W}}$

Question: 14.088P

A house built on a riverside is to be cooled in summer by utilizing the cool water of the river. A 15-m-long section of a circular stainless-steel duct of 20-cm diameter passes through the water. Air flows through the underwater section of the duct at 3 m/s at an average temperature of 15°C. For an overall fan efficiency of 62 percent, determine the fan power needed to overcome the flow resistance in this section of the duct.

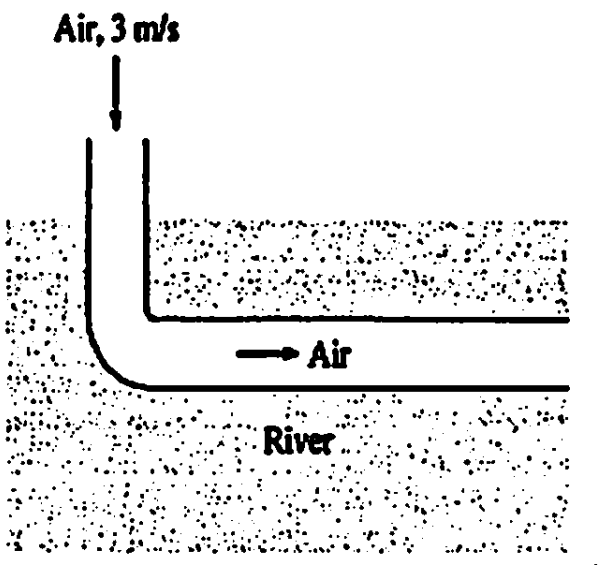


FIGURE P14–88

Step 1 of 6

Given data:

The length of the duct, $L = 15\ \text{m}$

The diameter of the duct, $D = 20\ \text{cm} = 0.2\text{m}$

The velocity of air, $V = 3\ \text{m/s}$

The temperature of the air, $T = 15°\text{C}$

The overall fan efficiency, $\eta = 62\% = 0.62$

The duct is made of stainless steel.

Step 2 of 6

The flow is assumed to be steady, incompressible and fully developed. The minor losses are neglected. The pressure of air is assumed to be 1 atm.

From the properties table of air at 1 atm pressure and 15°C

The density is $\rho = 1.225\ \text{kg/m}^3$

The dynamic viscosity is $\mu = 1.802 \times 10^{-5}\ \text{kg/m} \cdot \text{s}$

For galvanized iron, the roughness is $\varepsilon = 0.002\ \text{mm} = 2 \times 10^{-6}\ \text{m}$

Step 3 of 6

The cross-sectional flow area between the glass covers

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.2\ \text{m})^2$$

$$= 31.416 \times 10^{-3}\ \text{m}^2$$

The volumetric flow rate of air,

$$\dot{V} = VA_c$$

$$= (3\ \text{m/s})(31.416 \times 10^{-3}\ \text{m}^2)$$

$$= 0.0942\ \text{m}^3/\text{s}$$

Step 4 of 6

The Reynolds number,

$$\text{Re} = \frac{\rho VD}{\mu}$$

$$= \frac{(1.225\ \text{kg/m}^3)(3\ \text{m/s})(0.2\text{m})}{1.802 \times 10^{-5}\ \text{kg/m} \cdot \text{s}}$$

$$= 40.788 \times 10^3$$

Since $\text{Re} > 4000$, the flow is turbulent.

Step 5 of 6

For turbulent flow, the Colebrook equation for friction factor (f) is

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2 \times 10^{-6}\ \text{m}/0.2\text{m}}{3.7} + \frac{2.51}{(40.788 \times 10^3)\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(2.703 \times 10^{-6} + \frac{61.54 \times 10^{-6}}{\sqrt{f}}\right)$$

By solving the above equation for f, we get the friction factor as $f = 0.0219$

Step 6 of 6

The pressure drop,

$$\Delta P_L = f \frac{L}{D} \frac{\rho V^2}{2}$$

$$= 0.0219\left[\frac{15\ \text{m}}{0.2\text{m}}\right]\left[\frac{(1.225\ \text{kg/m}^3)(3\ \text{m/s})^2}{2}\right]$$

$$= 9.054\ \text{Pa}$$

The fan power required,

$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump}}}{\eta}$$

$$= \frac{(\dot{V})(\Delta P)}{\eta}$$

$$= \frac{(0.0942\ \text{m}^3/\text{s})(9.054\ \text{Pa})}{0.62}$$

$$= 1.376\ \text{W}$$

Therefore, the fan power required to overcome the flow resistance is $\boxed{\dot{W}_{\text{fan}} = 1.4\ \text{W}}$

Question: 14.089P

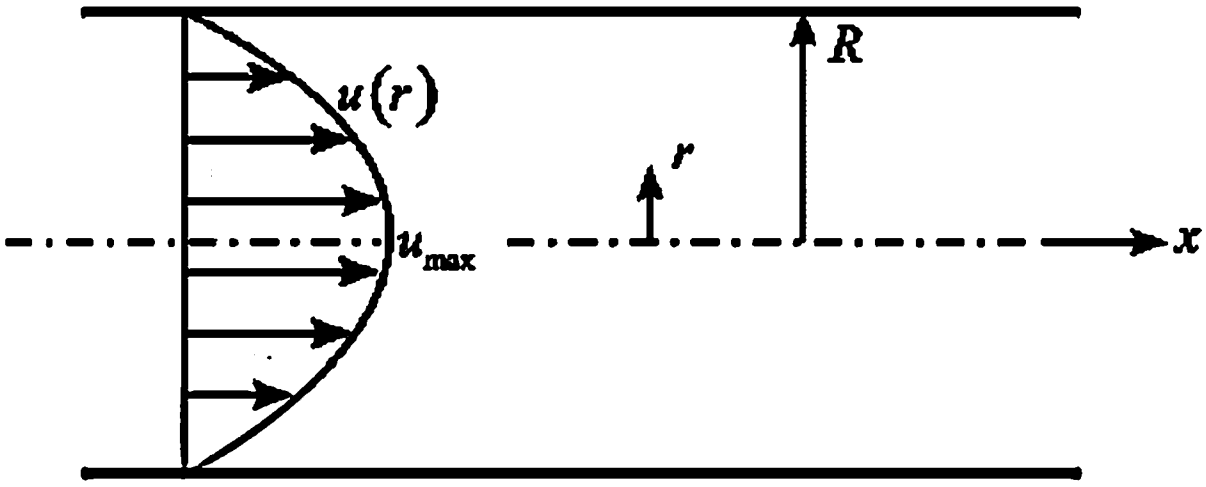
The velocity profile in fully developed laminar flow in a circular pipe, in m/s, is given by $u(r) = 6(1 - 100r^2)$, where r is the radial distance from the centerline of the pipe in m. Determine (*a*) the radius of the pipe, (*b*) the average velocity through the pipe, and (*c*) the maximum velocity in the pipe.

Step 1 of 4

Given data:

The velocity profile of the flow is $u(r) = 6\left(1 - 100r^2\right)$

Velocity profile of the flow is as shown below.



Step 2 of 4

(a)

The velocity profile of the fully developed laminar flow for a circular channel is given by

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right) \qquad \ldots\ldots (1)$$

where, r is the radius of the differential fluid element

R is the radius of the pipe and

u_{max} is the maximum velocity, at the center of the pipe.

By comparing the given velocity profile with equation (1), we get

The maximum velocity as $u_{max} = 6$ m/s

The ratio of squares of the radii,

$$\frac{r^2}{R^2} = 100r^2$$

$$R^2 = \frac{1}{100}$$

$$R = 0.1\,\text{m}$$

Hence the radius of the pipe is $\boxed{R = 0.1\,\text{m}}$

Step 3 of 4

(b)

The average velocity of the flow is given by

$$V_{avg} = \frac{u_{max}}{2}$$

$$= \frac{6\,\text{m/s}}{2}$$

$$= 3\,\text{m/s}$$

Therefore, the average velocity through the pipe is $\boxed{V_{avg} = 3\,\text{m/s}}$

Step 4 of 4

c)

By comparing the given velocity profile with equation (1), we get

The maximum velocity as $\boxed{u_{max} = 6\ \text{m/s}}$

Question: 14.090P

The velocity profile in a fully developed laminar flow of water at 40°F in a 115-ft-long horizontal circular pipe, in ft/s, is given by $u(r) = 0.8(1 - 625r2)$, where r is the radial distance from the centerline of the pipe in ft. Determine (*a*) the volume flow rate of water through the pipe,(*b*) the pressure drop across the pipe, and (*c*) the useful pumping power required to overcome this pressure drop.

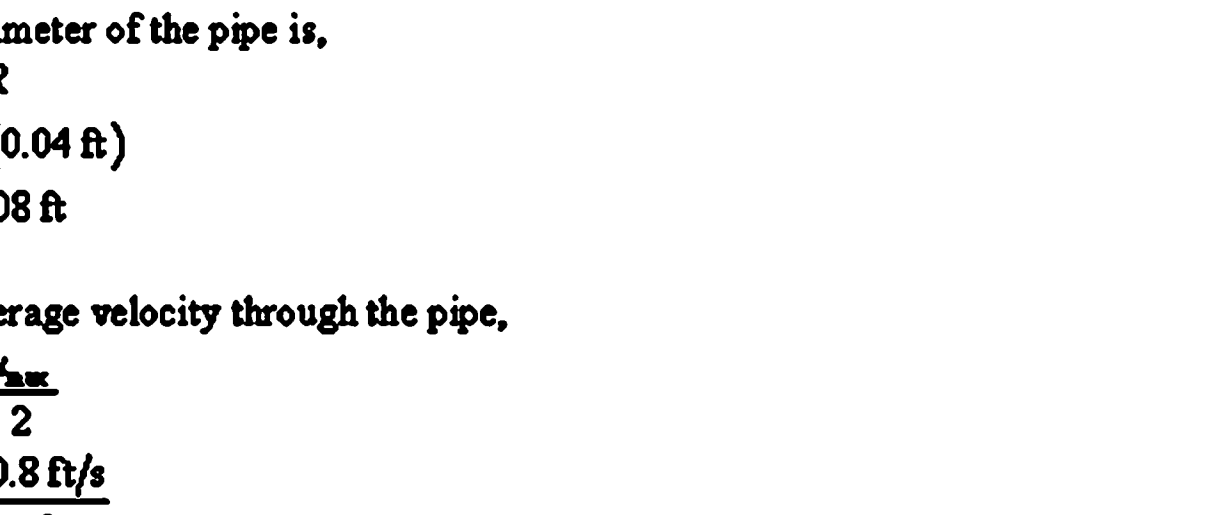
Step 1 of 6

Given data:

The length of the pipe, $L = 155$ ft

The velocity profile, $u(r) = 0.8\left(1 - 625r^2\right)$

Velocity profile of the flow is as shown below.



Step 2 of 6

(a) The velocity profile of the fully developed laminar flow for a circular channel is given by

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right) \qquad \ldots\ldots (1)$$

where, r is the radius of the differential fluid element

R is the radius of the pipe and

u_{max} is the maximum velocity, at the center of the pipe.

By comparing the given velocity profile with equation (1), we get

The maximum velocity as $u_{max} = 0.8$ ft/s

The ratio of squares of the radii,

$$\frac{r^2}{R^2} = 625r^2$$

$$R^2 = \frac{1}{625}$$

$$R = 0.04 \text{ ft}$$

Step 3 of 6

The diameter of the pipe is,

$$D = 2R$$
$$= 2(0.04 \text{ ft})$$
$$= 0.08 \text{ ft}$$

The average velocity through the pipe,

$$V_{avg} = \frac{u_{max}}{2}$$
$$= \frac{0.8 \text{ ft/s}}{2}$$
$$= 0.4 \text{ ft/s}$$

Step 4 of 6

The cross sectional flow area,

$$A_c = \pi R^2$$
$$= \pi(0.04 \text{ ft})^2$$
$$= 5.026 \times 10^{-3} \text{ ft}^2$$

The flow rate,

$$\dot{V} = V_{avg} A_c$$
$$= (0.4 \text{ ft/s})\left(5.026 \times 10^{-3} \text{ ft}^2\right)$$
$$= 2 \times 10^{-3} \text{ ft}^3/\text{s}$$

Therefore, the flow rate, $\boxed{\dot{V} = 2 \times 10^{-3} \text{ ft}^3/\text{s}}$

Step 5 of 6

(b) From the properties table of saturated water, at 40°F

The dynamic viscosity is $\mu = 1.308 \times 10^{-3}$ lbm/ft · s

The pressure drop for laminar flow in a horizontal pipe,

$$\Delta P = \frac{32 \mu L V_{avg}}{D^2}$$
$$= \frac{32\left(1.308 \times 10^{-3} \text{lbm/ft} \cdot \text{s}\right)(155 \text{ ft})(0.4 \text{ ft/s})}{(0.08 \text{ ft})^2}$$
$$= 405.48 \text{ lbf/ft}^2$$

Therefore, the pressure drop for laminar flow in a horizontal pipe is $\boxed{\Delta P = 405.48 \text{ lbf/ft}^2}$

Step 6 of 6

(c) The useful pumping power required,

$$\dot{W}_{pump,u} = (\dot{V})(\Delta P)$$
$$= \left(2 \times 10^{-3} \text{ ft}^3/\text{s}\right)\left(405.48 \text{ lbf/ft}^2\right)$$
$$= 0.811 \text{ lbf} \cdot \text{ft/s}$$

Therefore, the useful pumping power required is $\boxed{\dot{W}_{pump,u} = 0.811 \text{lbf} \cdot \text{ft/s}}$

Question: 14.091P

Repeat Prob. 14-90E assuming the pipe is inclined 12° from the horizontal and the flow is uphill.

Step 1 of 6

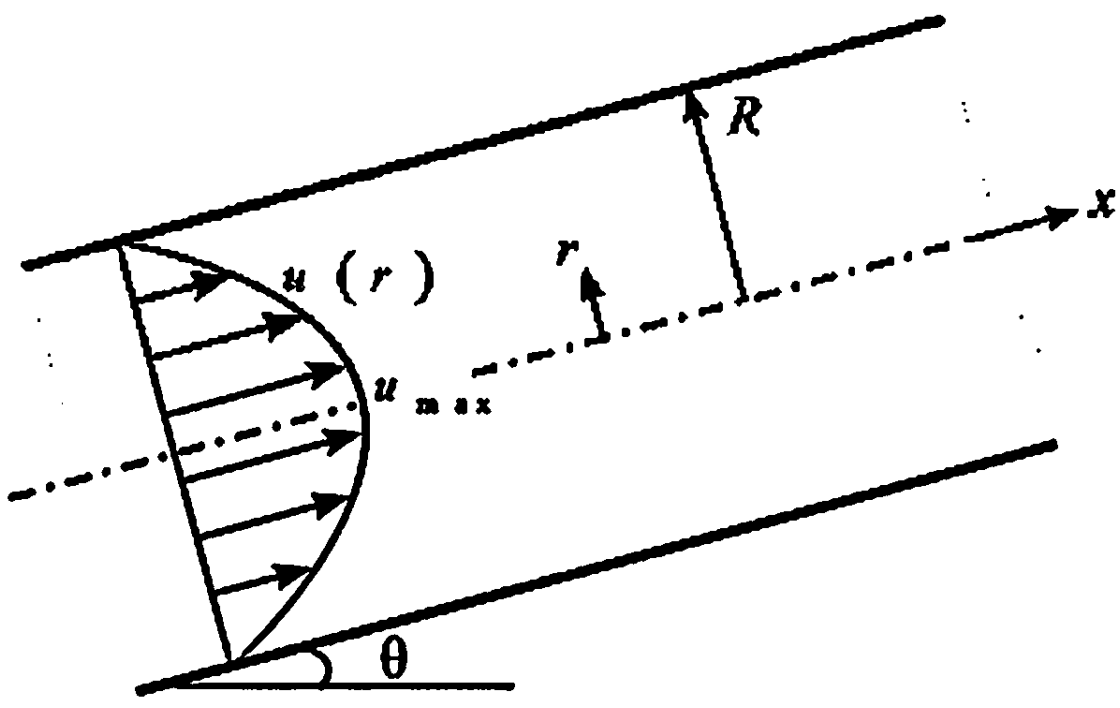
Given data:

The length of the pipe, $L = 155$ ft

The velocity profile, $u(r) = 0.8\left(1 - 625r^2\right)$

The pipe is inclined at an angle of $\theta = 12°$

Velocity profile of the flow is as shown below.



Step 2 of 6

(a)

The velocity profile of the fully developed laminar flow for a circular channel is given by

$$u(r) = u_{max}\left(1 - \frac{r^2}{R^2}\right) \qquad \ldots\ldots (1)$$

where, r is the radius of the differential fluid element

R is the radius of the pipe and

u_{max} is the maximum velocity, at the center of the pipe.

By comparing the given velocity profile with equation (1), we get

The maximum velocity as $u_{max} = 0.8$ ft/s

The ratio of squares of the radii,

$$\frac{r^2}{R^2} = 625r^2$$

$$R^2 = \frac{1}{625}$$

$$R = 0.04 \text{ ft}$$

Step 3 of 6

The diameter of the pipe is,

$$D = 2R$$
$$= 2(0.04 \text{ ft})$$
$$= 0.08 \text{ ft}$$

The average velocity through the pipe,

$$V_{avg} = \frac{u_{max}}{2}$$
$$= \frac{0.8 \text{ ft/s}}{2}$$
$$= 0.4 \text{ ft/s}$$

Step 4 of 6

The cross sectional flow area,

$$A_c = \pi R^2$$
$$= \pi(0.04 \text{ ft})^2$$
$$= 5.026 \times 10^{-3} \text{ ft}^2$$

The flow rate,

$$\dot{V} = V_{avg} A_c$$
$$= (0.4 \text{ ft/s})\left(5.026 \times 10^{-3} \text{ ft}^2\right)$$
$$= 2 \times 10^{-3} \text{ ft}^3/\text{s}$$

Therefore, the flow rate, $\boxed{\dot{V} = 2 \times 10^{-3} \text{ ft}^3/\text{s}}$

Step 5 of 6

(b)

From the properties table of saturated water, at 40°F

The dynamic viscosity is $\mu = 1.308 \times 10^{-3}$ lbm/ft · s

The density of the water is $\rho = 62.42$ lbm/ft^3

The relation between pressure drop and volumetric flow rate is given by

$$\dot{V} = \frac{\left[\Delta P - \rho g L \sin\theta\right]\pi D^4}{128 \mu L}$$

$$\Delta P = \frac{128 \mu L \dot{V}}{\pi D^4} + \rho g L \sin\theta$$
$$= \frac{128\left(1.308 \times 10^{-3}\right)(155)\left(2 \times 10^{-3}\right)}{\pi(0.08)^4} + (62.42)(32.2)(155)\sin 12°$$
$$= 65175.78 \text{ lbf/ft}^2$$

Therefore, the pressure drop for laminar flow in an inclined pipe is $\boxed{\Delta P = 405.48 \text{ lbf/ft}^2}$

Step 6 of 6

(c)

The useful pumping power required,

$$\dot{W}_{pump,u} = (\dot{V})(\Delta P)$$
$$= \left(2 \times 10^{-3} \text{ ft}^3/\text{s}\right)\left(65175.78 \text{ lbf/ft}^2\right)$$
$$= 130.35 \text{ lbf} \cdot \text{ft/s}$$

Therefore, the useful pumping power required is $\boxed{\dot{W}_{pump,u} = 130.35 \text{ lbf} \cdot \text{ft/s}}$

Question: 14.092P

A highly viscous liquid discharges from a large container through a small-diameter tube in laminar flow. Disregarding entrance effects and velocity heads, obtain a relation for the variation of fluid depth in the tank with time.

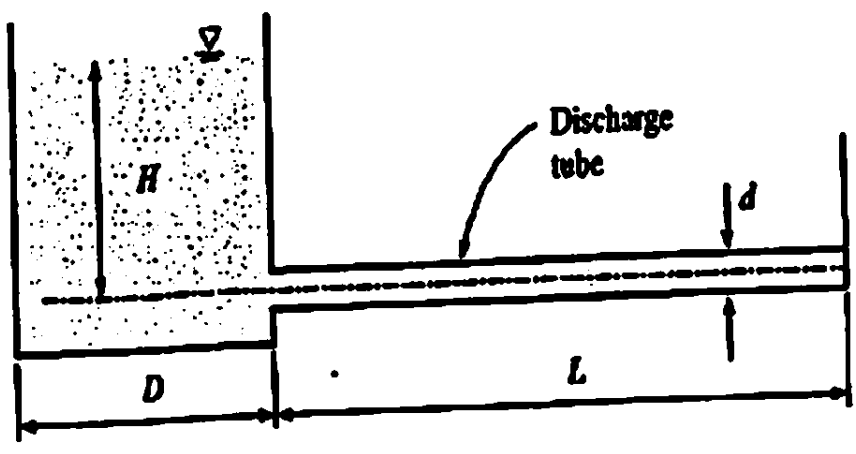


FIGURE P14-92

Step 1 of 5

Consider a tank containing incompressible fluid of height h. The liquid flows through the discharge tube of length L and diameter d. The flow is laminar.

For laminar flow, the friction factor is given by

$$f = \frac{64}{\text{Re}}$$
$$= \frac{64}{(Vd/\nu)}$$

By considering the free surface of the liquid in the tank as point-1 and the exit of the discharge pipe as point-2, from the energy balance equation for a control volume, we have

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L \qquad \text{..... (1)}$$

Step 2 of 5

Here, both the points 1 and 2 are opened to the atmosphere. So the pressure that acts on both these points is atmospheric pressure.

$$P_1 = P_2 = P_{atm}$$

The velocity of the liquid at the free surface is zero as the fluid is stationary at the surface

$$V_1 = 0$$

The velocity head are negligible. Thus,

$$\frac{V_1^2}{2g} = \frac{V_2^2}{2g} \approx 0$$

Let point 2 be the reference level. Then,

$$z_2 = 0$$
$$z_1 = h$$

The average flow velocity,

$$V = \frac{\dot{V}}{A_c}$$
$$= \frac{4\dot{V}}{\pi d^2}$$

Step 3 of 5

By substituting the values in equation (1), we get

$$\frac{P_{atm}}{\rho g} + 0 + h = \frac{P_{atm}}{\rho g} + 0 + 0 + h_L$$
$$h = h_L$$

The expression for head loss is

$$h_L = \frac{f L V^2}{2gd}$$
$$= \frac{64L}{2(Vd/\nu)gd}\left(\frac{4\dot{V}}{\pi d^2}\right)^2$$
$$= \frac{128\nu L\dot{V}}{\pi g d^4}$$

Step 4 of 5

As the flow occurs the height of the liquid in the tank decreases. Hence the volume flow rate can be written as,

$$\dot{V} = -\left(A_{c,\text{tank}}\right)\frac{dh}{dt}$$
$$= -\left(\frac{\pi D^2}{4}\right)\frac{dh}{dt}$$

The negative sign indicates that the rate of change of liquid height is decreasing.

Now the expression for head loss becomes

$$h_L = \frac{128\nu L}{\pi g d^4}\left(-\frac{\pi D^2}{4}\frac{dh}{dt}\right)$$
$$h_L = -\frac{32\nu L D^2}{gd^4}\frac{dh}{dt}$$
$$dt = -\frac{32\nu L D^2}{gd^4}\frac{dh}{h} \qquad \text{..... (2)}$$

Step 5 of 5

By integrating the equation (2), we get

$$\int_0^t dt = -\frac{32\nu L D^2}{gd^4}\int_H^h \frac{dh}{h}$$
$$[t]_0^t = -\frac{32\nu L D^2}{gd^4}\left[\ln(h)\right]_H^h$$
$$(t-0) = -\frac{32\nu L D^2}{gd^4}\left(\ln(h) - \ln(H)\right)$$
$$t = -\frac{32\nu L D^2}{gd^4}\ln\left(\frac{h}{H}\right)$$

Therefore, the relation for the variation of fluid depth in the tank with time is

$$\boxed{t = -\frac{32\nu L D^2}{gd^4}\ln\left(\frac{h}{H}\right)}$$

Question: 14.093P

A student is to determine the kinematic viscosity of an oil using the system shown in Prob. 14-92. The initial fluid height in the tank is $H = 40$ cm, the tube diameter is $d = 6$ mm, the tube length is $L = 0.65$ m, and the tank diameter is $D = 0.63$ m. The student observes that it takes 1750 s for the fluid level in the tank to drop to 36 cm. Find the fluid viscosity.

Step 1 of 6

Given data:

The initial fluid height of the tank, $H = 40 \text{ cm} = 40 \times 10^{-2} \text{ m}$

The tube diameter, $d = 6 \text{ mm} = 6 \times 10^{-3} \text{ m}$

The tube length, $L = 0.65 \text{ m}$

The tank diameter, $D = 0.63 \text{ m}$

The final height of fluid in tank is $h = 36 \text{ cm} = 36 \times 10^{-2} \text{ m}$

The duration of discharge from tank is $t = 1750 \text{ s}$

Step 2 of 6

For laminar flow, the friction factor is given by

$$f = \frac{64}{\text{Re}} = \frac{64}{(Vd/\nu)} = \frac{64\nu}{V\left(6\times10^{-3}\right)}$$

By considering the free surface of the liquid in the tank as point-1 and the exit of the discharge pipe as point-2, from the energy balance equation for a control volume, we have

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_L \qquad \text{..... (1)}$$

Step 3 of 6

Here, both the points 1 and 2 are opened to the atmosphere. So the pressure that acts on both these points is atmospheric pressure.

$P_1 = P_2 = P_{atm}$

The velocity of the liquid at the free surface is zero as the fluid is stationary at the surface

$V_1 = 0$

The velocity head are negligible. Thus,

$$\frac{V_1^2}{2g} = \frac{V_2^2}{2g} \approx 0$$

Let point 2 be the reference level. Then,

$z_2 = 0$

$z_1 = h$

The average flow velocity,

$$V = \frac{\dot{V}}{A_c} = \frac{4\dot{V}}{\pi\left(6\times10^{-3}\right)^2} = \frac{\dot{V}}{\pi\left(9\times10^{-6}\right)}$$

Step 4 of 6

By substituting the values in equation (1), we get

$$\frac{P_{atm}}{\rho g} + 0 + h = \frac{P_{atm}}{\rho g} + 0 + 0 + h_L$$

$h = h_L$

The expression for head loss is

$$h_L = \frac{f L V^2}{2gd} = \frac{64 L V^2}{2(Vd/\nu)gd} = \frac{32\nu L V}{gd^2} = \frac{32\nu(0.65)}{g\left(6\times10^{-3}\right)^2}\left(\frac{\dot{V}}{\pi\left(9\times10^{-6}\right)}\right) = \left(2\times10^{9}\right)\left(\nu\times\dot{V}\right)$$

Step 5 of 6

As the flow occurs the height of the liquid in the tank decreases. Hence the volume flow rate can be written as,

$$\dot{V} = -\left(A_{c,\text{tank}}\right)\frac{dh}{dt} = -\left[\frac{\pi}{4}(0.63)^2\right]\frac{dh}{dt} = -0.312\frac{dh}{dt}$$

The negative sign indicates that the rate of change of liquid height is decreasing.

Now the expression for head loss becomes

$$h_L = \left(2\times10^{9}\right)(\nu)\left(-0.312\frac{dh}{dt}\right)$$

$$h = -624\times10^{6}\times\nu\frac{dh}{dt}$$

$$dt = -624\times10^{6}\times\nu\times\frac{dh}{h} \qquad \text{..... (2)}$$

Step 6 of 6

By integrating the equation (2), we get

$$\int_0^t dt = -624\times10^{6}\times\nu\int_H^h \frac{dh}{h}$$

$$[t]_0^t = -624\times10^{6}\times\nu\left[\ln(h)\right]_H^h$$

$$(t-0) = -624\times10^{6}\times\nu\left(\ln(h) - \ln(H)\right)$$

$$t = -624\times10^{6}\times\nu\ln\left(\frac{h}{H}\right)$$

$$\frac{1}{\nu} = \frac{-624\times10^{6}}{t}\ln\left(\frac{h}{H}\right)$$

$$\frac{1}{\nu} = \frac{624\times10^{6}}{1750}\ln\left(\frac{40}{36}\right)$$

$$\nu = 2.67\times10^{-5} \text{ m}^2/\text{s}$$

Therefore, the kinematic viscosity, $\boxed{\nu = 2.67\times10^{-5} \text{ m}^2/\text{s}}$

Question: 14.094P

Water is to be withdrawn from a 7-m-high water reservoir by drilling a well-rounded 4-cm-diameter hole with negligible loss near the bottom and attaching a horizontal 90° bend of negligible length. Taking the kinetic energy correction factor to be 1.05, determine the flow rate of water through the bend if (*a*) the bend is a flanged smooth bend and (*b*) the bend is a miter bend without vanes.

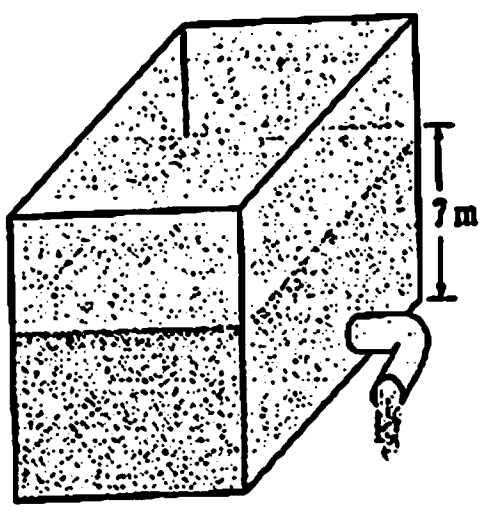


FIGURE P14–94

Step 1 of 5

Given data:

The height of the reservoir, $H = 7\text{ m}$

The diameter of the hole, $D = 4\text{ cm} = 0.04\text{ m}$

The kinetic correction factor, $\alpha_1 = \alpha_2 = 1.05$

Step 2 of 5

Consider the free surface of the reservoir as point 1 and the exit hole as point 2. Now for incompressible and steady flow, the energy balance equation for a control volume is given by

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_L \qquad \text{..... (1)}$$

Here, both the points 1 and 2 are opened to the atmosphere. So

$P_1 = P_2 = P_{atm}$

The velocity of fluid at the free surface is $V_1 = 0$

Let point 2 be the reference level, then

$z_2 = 0$ and

$z_1 = H$

$= 7\text{ m}$

The head loss in the flow is given by

$$h_L = K_L\frac{V_2^2}{2g}$$

Step 3 of 5

By substituting the values in equation (1) we get,

$$\frac{P_{atm}}{\rho g}+(1.05)\frac{(0)^2}{2g}+7=\frac{P_{atm}}{\rho g}+(1.05)\frac{V_2^2}{2g}+0+K_L\frac{V_2^2}{2g}$$

$$7=1.05\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g}$$

$$7=(1.05+K_L)\frac{V_2^2}{2g}$$

$$V_2^2=\frac{14g}{1.05+K_L}$$

$$V_2=\sqrt{\frac{14g}{1.05+K_L}}$$

The volume flow rate is given by

$$\dot{V}=V_2A_c$$

$$\dot{V}=\sqrt{\frac{14g}{1.05+K_L}}\left(\frac{\pi}{4}D^2\right) \qquad \text{..... (2)}$$

Step 4 of 5

(a) The loss coefficient for flanged smooth bend, $K_L = 0.3$

Now by substituting this value in equation (2), we get

$$\dot{V}=\sqrt{\frac{14g}{1.05+K_L}}\left(\frac{\pi}{4}D^2\right)$$

$$=\sqrt{\frac{14\times9.81}{1.05+0.3}}\left(\frac{\pi}{4}0.04^2\right)$$

$$=\sqrt{\frac{137.34}{1.35}}\left(1.257\times10^{-3}\right)$$

$$=0.01268\text{ m}^3/\text{s}$$

$$=12.68\text{ L/s}$$

Therefore, the flow rate of water through the flanged smooth bend is $\boxed{\dot{V}=12.68\text{ L/s}}$

Step 5 of 5

(b) The loss coefficient for miter bends without vanes, $K_L = 1.1$

Now by substituting this value in equation (2), we get

$$\dot{V}=\sqrt{\frac{14g}{1.05+K_L}}\left(\frac{\pi}{4}D^2\right)$$

$$=\sqrt{\frac{14\times9.81}{1.05+1.1}}\left(\frac{\pi}{4}0.04^2\right)$$

$$=\sqrt{\frac{137.34}{2.15}}\left(1.257\times10^{-3}\right)$$

$$=0.01\text{ m}^3/\text{s}$$

$$=10\text{ L/s}$$

Therefore, the flow rate of water through the miter bends without vanes is $\boxed{\dot{V}=10\text{ L/s}}$

Question: 14.095P pg 1

In a geothermal distinct heating system, 10,000 kg/s of hot water must be delivered a distance of 10 km in a horizontal pipe. The minor losses are negligible, and the only significant energy loss arises from pipe friction The friction factor is taken to be 0.015. Specifying a larger-diameter pipe would reduce water velocity, velocity head pipe friction, and thus power consumption. But a larger pipe would also cost more money initially to purchase and install. Otherwise stated, there is an optimum pipe diameter that will minimize the sum of pipe cost and future electric power cost.

Assume the system will run 24 h/day, every day for 30 years. During this time the cost of electricity remains constant at $0.06/kWh. Assume system performance stays constant over the decades (this may not be true, especially if highly mineralized water is passed through the pipeline - scale may form). The pump has an overall efficiency of 80 percent. The cost to purchase, install, and insulate a l0-km pipe depends on the diameter *D* and is given by cost = $106 *D2*, where *D* is in m. Assuming zero inflation and interest rate for simplicity and zero salvage value and zero maintenance cost, determine the optimum pipe diameter.

Step 1 of 8

Apply Bernoulli's equation between two points of the horizontal pipe.

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_L \quad \text{..... (1)}$$

Here, P_1, P_2 is the pressures at points 1 and 2, V_1 , V_2 is the velocities at points 1 and 2, z_1 , z_2 is the datum heads at points 1 and 2, g is the acceleration due to gravity, ρ is the density of the hot water, and h_L is the head loss due to friction and head loss due to pump is h_{pump} , α_1 , α_2 kinetic energy correction factors at point 1 and 2.

In the loop system, the pressure at point 1 and 2 is constant.

$$P_1 = P_2 \quad \text{..... (2)}$$

In the loop system, the velocity at point 1 and 2 is constant.

$$V_1 = V_2 \quad \text{..... (3)}$$

In the loop system, the datum head at point 1 and 2 is constant.

$$z_1 = z_2 \quad \text{..... (4)}$$

Substitute the values of equations (2), (3), and (4) in equation (1).

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump}=\frac{P_1}{\rho g}+\alpha_2\frac{V_1^2}{2g}+z_1+h_L$$
$$\alpha_1+h_{pump}=\alpha_2+h_L$$

Substitute 1 for α_1 , 1 for α_2 .

$$h_{pump}=h_L$$

Step 2 of 8

Calculate the volume flow rate.

$$\dot{V}=\frac{\dot{m}}{\rho}$$

Here, mass flow rate of the water is $\dot{m}$, and ρ is density of the water.

Substitute $10000\ \text{kg/s}$ for $\dot{m}$, and $1000\ \text{kg/m}^3$ for ρ .

$$\dot{V}=\frac{\dot{m}}{\rho}$$
$$=\frac{10000}{1000}$$
$$=10\ \text{m}^3/\text{s}$$

Consider the average velocity expression as follows:

$$V=\frac{\dot{V}}{A}$$

Here, V is the average flow velocity, $\dot{V}$ is the volume flow rate, and A is the area of the pipe.

Substitute $\left(\frac{\pi}{4}D^2\right)$ for A .

$$V=\frac{\dot{V}}{\frac{\pi}{4}D^2}$$
$$=\frac{4\dot{V}}{\pi D^2}$$

Here, D is the diameter of the pipe.

Step 3 of 8

Calculate the head loss in the pipe using the equation as follows:

$$\Delta P=f\frac{L}{D}\frac{\rho V^2}{2}$$

Substitute $\left(\frac{4\dot{V}}{\pi D^2}\right)$ for V .

$$\Delta P=f\frac{L}{D}\times\frac{\rho\left(\frac{4\dot{V}}{\pi D^2}\right)^2}{2}$$
$$=f\frac{8L\rho\dot{V}}{D^5\pi^2}$$

Here, *f* is the friction factor, and *L* is the length of the pipe.

Step 4 of 8

Calculate the pumping power using the equation as follows:

$$\dot{W}_{pump}=\frac{\text{Work done done by the pump}}{\text{Overall efficiency of the pump}}$$
$$=\frac{\dot{V}\Delta P}{}$$

Calculate the pumping power using the equation as follows:

$$\dot{W}_{pump} = \frac{\text{Work done done by the pump}}{\text{Overall efficiency of the pump}}$$
$$= \frac{\dot{V}\Delta P}{\eta_{Overall}}$$

Here, $\dot{V}$ is the volume flow rate, ΔP is the pressure difference, and $\eta_{Overall}$ is the overall efficiency.

Substitute $\left(f\frac{8L\rho\dot{V}}{D^5\pi^2}\right)$ for ΔP.

$$\dot{W}_{pump} = \frac{\dot{V}\left(f\frac{8L\rho\dot{V}^2}{D^5\pi^2}\right)}{\eta_{Overall}} \quad \text{..... (5)}$$
$$= \frac{\left(f\frac{8L\rho\dot{V}^3}{D^5\pi^2}\right)}{\eta_{Overall}}$$

Step 5 of 8

Substitute 0.015 for f, 10 km for L, $1000\ \text{kg/m}^3$ for ρ in equation (5).

$$\dot{W}_{pump} = \frac{\left(f\frac{8L\rho\dot{V}^3}{D^5\pi^2}\right)}{\eta_{Overall}}$$
$$= \frac{\left(0.015\times\frac{8\times10000\times1000\times10^3}{0.8}\right)}{\pi^2}$$
$$= 1.51\times10^8\times\frac{1}{D^5}\ \text{W}$$
$$= \frac{1.51\times10^5}{D^5}\ \text{kW}$$

Calculate the number of hours for one year to run the system.

$$\Delta t = (24\times365)$$
$$= 8760\ \text{h}$$

Step 6 of 8

Calculate the total electrical energy required for one year.

$$E_{Pump} = \dot{W}_{pump}\Delta t$$

Substitute $\left(\frac{1.51\times10^5}{D^5}\ \text{kW}\right)$ for $\dot{W}_{pump}$, $8760\ \text{h}$ for Δt.

$$E_{Pump} = \frac{1.51\times10^5}{D^5}\times8760$$
$$= \frac{13227.6}{D^5}\times10^5\ \text{kW-h}$$

Calculate the total electrical energy cost required for one year.

$$\text{Energy cost} = E_{Pump}\times\text{Unit cost}$$

Substitute $\left(\frac{13227.6}{D^5}\times10^5\right)\text{kW-h}$ for E_{Pump}, and $\$0.06/\text{kWh}$ for Unit cost.

$$\text{Energy cost} = \frac{13227.6\times10^5}{D^5}\times0.06$$
$$= \frac{793.656\times10^5}{D^5}\ \$/\text{yr}$$

Step 7 of 8

Calculate the system cost for 30 years using the equation as follows:

$$\text{System cost} = \frac{\text{Total cost}}{\text{Life time}}$$

Substitute $\$10^6D^2$ for Total cost, 30 years for Life time.

$$\text{System cost} = \frac{\$10^6D^2}{30}$$
$$= 3.3\times10^4\ D^2\ \$/\text{yr}$$

Calculate the total cost of the system using the equation as follows:

$$\text{Total cost} = \text{Energy cost} + \text{System cost}$$

Substitute $\left(\frac{793.656\times10^5}{D^5}\ \$/\text{yr}\right)$ for Energy cost, and $\left(3.3\times10^4\ D^2\ \$/\text{yr}\right)$ for System cost.

$$\text{Total cost} = \frac{793.656\times10^5}{D^5}\ \$/\text{yr} + 3.3\times10^4\ D^2\ \$/\text{yr}$$

Step 8 of 8

Calculate the diameter of the pipe by applying the condition as follows:

$$\frac{\partial(\text{Total cost})}{\partial D} = 0$$

Substitute $\left(\frac{793.656\times10^5}{D^5}\ \$/\text{yr} + 3.3\times10^4\ D^2\ \$/\text{yr}\right)$ for Total cost.

$$\frac{\partial\left(\frac{793.656\times10^5}{D^5}\ \$/\text{yr} + 3.3\times10^4\ D^2\ \$/\text{yr}\right)}{\partial D} = 0$$

$$\left(793.656\times10^5\ \$/\text{yr}\,\partial\left(\frac{1}{D^5}\right) + 3.3\times10^4\ \$/\text{yr}\,\partial\left(D^2\right)\right) = 0$$

$$\frac{\partial\left(\frac{793.656\times10^{5}\ \$/\text{yr}}{D^{5}}+3.3\times10^{4}\ \$/\text{yr}\,D^{2}\right)}{\partial D}=0$$

$$\left(793.656\times10^{5}\ \$/\text{yr}\,\partial\left(\frac{1}{D^{5}}\right)+3.3\times10^{4}\ \$/\text{yr}\,\partial\left(D^{2}\right)\right)=0$$

$$\left(793.656\times10^{5}\ \$/\text{yr}\times(-5)\left(\frac{1}{D^{6}}\right)+3.3\times10^{4}\ \$/\text{yr}\times2D\right)=0$$

$$793.656\times10^{5}\ \$/\text{yr}\times(5)\left(\frac{1}{D^{6}}\right)=3.3\times10^{4}\ \$/\text{yr}\times2D$$

$$D^{7}=\frac{396828000}{66000}$$

$$D=(6012.545)^{1/7}$$

$$=3.46\ \text{m}$$

Therefore, the optimum diameter of the pipe D is $\boxed{3.48\ \text{m}}$.

Question: 14.096P

Two pipes of identical diameter and material are connected in parallel. The length of pipe *A* is three times the length of pipe *B*. Assuming the flow is fully turbulent in both pipes and thus the friction factor is independent of the Reynolds number and disregarding minor losses, determine the ratio of the flow rates in the two pipes.

Step 1 of 4

Given data:

The diameter of pipe *A* is $D_A = D$

The diameter of pipe *B* is $D_B = D$

The length of pipe *A* is $L_A = 3L$

The length of pipe *B* is $L_B = L$

The material is identical for both the pipes.

Step 2 of 4

By considering the flow to be steady and incompressible, the velocity of flow is given by

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{\dot{V}}{\frac{\pi}{4}D^2}$$

$$= \frac{4\dot{V}}{\pi D^2}$$

Step 3 of 4

The head loss for the flow is given by

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

$$= f\frac{L}{D}\frac{\left(4\dot{V}/\pi D^2\right)^2}{2g}$$

$$= 8f\frac{L}{g\pi^2 D^5}\dot{V}^2$$

The head loss for pipe *A* is

$$(h_L)_A = 8f\frac{L_A}{g\pi^2 D_A^5}\dot{V}_A^2$$

$$= 8f\frac{3L}{g\pi^2 D^5}\dot{V}_A^2$$

The head loss for pipe *B* is

$$(h_L)_B = 8f\frac{L_B}{g\pi^2 D_B^5}\dot{V}_B^2$$

$$= 8f\frac{L}{g\pi^2 D^5}\dot{V}_B^2$$

Step 4 of 4

For two pipes connected in parallel, the head loss for each pipe must be same. Thus we get the expression as

$$(h_L)_A = (h_L)_B$$

$$8f\frac{3L}{g\pi^2 D^5}\dot{V}_A^2 = 8f\frac{L}{g\pi^2 D^5}\dot{V}_B^2$$

$$3L\dot{V}_A^2 = L\dot{V}_B^2$$

$$\frac{\dot{V}_A}{\dot{V}_B} = \sqrt{\frac{L}{3L}}$$

$$= 0.577$$

Therefore, the ratio of flow rates in the two pipes is $\boxed{\frac{\dot{V}_A}{\dot{V}_B} = 0.577}$

Question:

A pipeline that transports oil at 40ЭC at a rate of 3 m3/s branches out into two parallel pipes made of commercial steel that reconnect downstream. Pipe *A* is 500 m long and has a diameter of 30 cm while pipe *B* is 800 m long and has a diameter of 45 cm. The minor losses are considered to be negligible. Determine the flow rate through each of the parallel pipes.

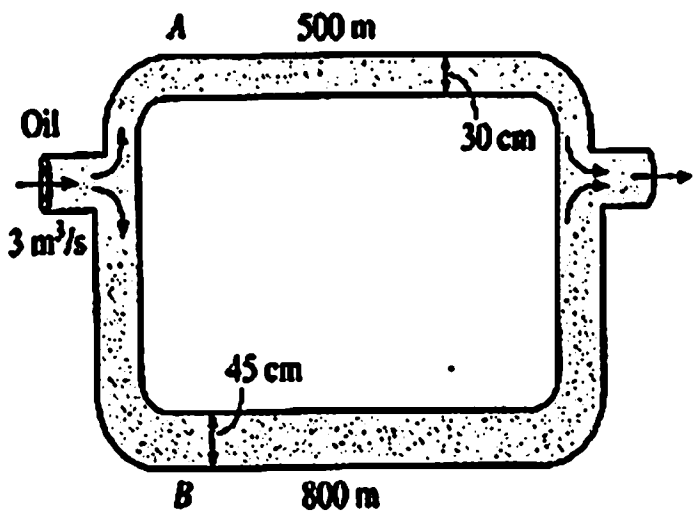


FIGURE P14–97

Step 1 of 8

Initial flow rate of the oil, $\dot{V} = 3\ \text{m}^3/\text{s}$

Length of the pipe A, $L_1 = 500\ \text{m}$

Length of the pipe B, $L_2 = 800\ \text{m}$

Diameter of the pipe A, $D_1 = 30\ \text{cm}$

Diameter of the pipe B, $D_2 = 45\ \text{cm}$

Temperature at which the oil is being transported, 40°C

The roughness of commercial steel pipes is $\varepsilon = 0.000045\ \text{m}$

From the properties table of oil at 40°C we get

Density of oil is $\rho = 876\ \text{kg/m}^3$

Dynamic viscosity of oil is $\mu = 0.2177\ \text{kg/m}\cdot\text{s}$

Step 2 of 8

In this problem let us first assume the flow to be turbulent. Also the head loss in both pipes is equal and the total flow rate must be the sum of the flow rates in the parallel branches.

$$h_{L1} = h_{L2} \qquad \ldots\ldots (1)$$

$$\dot{V} = \dot{V}_1 + \dot{V}_2 \qquad \ldots\ldots (2)$$

Now the average velocity in the pipe is determined by using the continuity equation

$$\dot{V} = AV$$

$$V = \frac{\dot{V}}{A}$$

$$= \frac{\dot{V}_1}{\pi D^2/4}$$

Step 3 of 8

Substituting the known values of pipe (1) in the above relation we get

$$V_1 = \frac{\dot{V}_1}{\pi D_1^2/4}$$

$$= \frac{\dot{V}_1}{\pi(0.3)^2/4}$$

$$= \frac{\dot{V}_1}{0.07068} \qquad \ldots\ldots (3)$$

Similarly for the pipe (2) we get

$$V_2 = \frac{\dot{V}_2}{\pi D_2^2/4}$$

$$= \frac{\dot{V}_2}{\pi(0.450.159)^2/4}$$

$$= \frac{\dot{V}_2}{0.159} \qquad \ldots\ldots (4)$$

Step 4 of 8

Now the Reynolds number for the flow in pipe (1) is

$$\text{Re}_1 = \frac{\rho V_1 D_1}{\mu}$$

$$= \frac{876 \times V_1 \times 0.3}{0.2177}$$

$$= 1207.165 \times V_1 \qquad \ldots\ldots (5)$$

Similarly the Reynolds number for the flow in pipe (2) is

$$\text{Re}_2 = \frac{\rho V_2 D_2}{\mu}$$

$$= \frac{876 \times V_2 \times 0.45}{0.2177}$$

$$= 1810.748 \times V_2 \qquad \ldots\ldots (6)$$

Step 5 of 8

For a fully developed turbulent flow, the friction factor depends upon the Reynolds number and the relative roughness ε/D

Hence the roughness value of the pipe (1) is

$$\Rightarrow \frac{\varepsilon}{D_1}$$

$$\Rightarrow \frac{0.000045}{0.3}$$

$$\Rightarrow \frac{\varepsilon}{D_1}$$

$$\Rightarrow \frac{0.000045}{0.3}$$

$$\Rightarrow 1.5\times10^{-4}$$

Similarly the roughness value of the pipe (2) is

$$\Rightarrow \frac{\varepsilon}{D_2}$$

$$\Rightarrow \frac{0.000045}{0.45}$$

$$\Rightarrow 1\times10^{-4}$$

Step 6 of 8

Now by using the Colebrook equation we shall determine the friction factor in turbulent flow. Hence the friction factor for the pipe (1) is

$$\frac{1}{\sqrt{f_1}} = -2\log\left(\frac{\varepsilon/D_1}{3.7} + \frac{2.5}{\mathrm{Re}_1\sqrt{f_1}}\right)$$

$$\frac{1}{\sqrt{f_1}} = -2\log\left(\frac{1.5\times10^{-4}}{3.7} + \frac{2.5}{(1207.165\times V_1)\sqrt{f_1}}\right) \qquad \ldots\ldots (7)$$

Similarly the friction factor for the pipe (2) is

$$\frac{1}{\sqrt{f_2}} = -2\log\left(\frac{\varepsilon/D_2}{3.7} + \frac{2.5}{\mathrm{Re}_2\sqrt{f_2}}\right)$$

$$\frac{1}{\sqrt{f_2}} = -2\log\left(\frac{1\times10^{-4}}{3.7} + \frac{2.5}{(1810.748\times V_2)\sqrt{f_2}}\right) \qquad \ldots\ldots (8)$$

Step 7 of 8

The head loss through the pipe (1) is

$$h_{f_1} = \frac{f_1 L_1 V_1^2}{2gD_1}$$

$$= \frac{f_1\times500\times V_1^2}{2\times9.81\times0.30}$$

$$= 84.94\times f_1\times V_1^2 \qquad \ldots\ldots (9)$$

Similarly the head loss for the pipe (2) is

$$h_{f_2} = \frac{f_2 L_2 V_2^2}{2gD_2}$$

$$= \frac{f_2\times800\times V_2^2}{2\times9.81\times0.45}$$

$$= 90.61\times f_2\times V_2^2 \qquad \ldots\ldots (10)$$

Step 8 of 8

Now using the equation solver, by solving the equations from (1) to (10) we get the Flow rate in pipe (1) is $\boxed{\dot{V}_1 = 0.92\ \mathrm{m}^3/\mathrm{s}}$ and the flow rate in the pipe (2) is $\boxed{\dot{V}_2 = 2.087\ \mathrm{m}^3/\mathrm{s}}$

Also the Reynolds number for the flow in pipe (1) is $\mathrm{Re}_1 = 15{,}540$ and in pipe (2) is $\mathrm{Re}_2 = 23{,}800$. This is clear that the Reynolds number is greater than 4000, hence our assumption that the flow is turbulent is correct.

Question: 14.098P

Repeat Prob. 14-97 for hot-water flow of a district heating system at 100°C.

Step 1 of 7

Given data:

Water is supplied at 20°C

The flow rate, $\dot{V} = 1.2\ m^3/s$

The length of the pipe, $L = 200\ m$

The diameter of the pipe, $D = 0.35\ m$

The pipe is made of cast iron.

The elevation difference between the free surface of the reservoir and turbine discharge, $H = 115\ m$.

The combined turbine-generator efficiency, $\eta = 80\%$

Step 2 of 7

Assuming that the flow is incompressible and steady:

From the table of properties of saturated water, corresponding to 20°C,

The density of water, $\rho = 998\ kg/m^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ kg/m \cdot s$

The roughness of cast iron pipe, $\varepsilon = 0.26\ mm = 0.26 \times 10^{-3}\ m$

The cross sectional flow area,

$$A_c = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}(0.35\ m)^2$$

$$= 96.211 \times 10^{-3}\ m^2$$

Step 3 of 7

The energy equation for a control volume is given by

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 + h_{pump,u} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \quad \text{..... (1)}$$

where, point 1 is taken as the surface of the reservoir and point 2 is taken as the exit of the hole.

Here, both the points 1 and 2 are opened to the atmosphere. So, $P_1 = P_2$

The velocity at the points 1 and 2 is low. So, the velocities, $V_1 = V_2 = 0$

Let point 2 is the reference level. Then, $z_2 = 0$ and $z_1 = H = 115\ m$.

There is no pump. So, $h_{pump,u} = 0$.

Thus, equation (1) becomes,

$$\frac{P_1}{\rho g} + \alpha_1\frac{(0)^2}{2g} + 115\ m + 0 = \frac{P_1}{\rho g} + \alpha_2\frac{(0)^2}{2g} + 0 + h_{turbine,e} + h_L$$

$$115\ m = h_{turbine,e} + h_L \quad \text{..... (2)}$$

Step 4 of 7

The average velocity of flow in the pipe,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{1.2\ m^3/s}{96.211 \times 10^{-3}\ m^2}$$

$$= 12.47\ m/s$$

The Reynolds number,

$$\mathrm{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(998\ kg/m^3)(12.47\ m/s)(0.35\ m)}{1.002 \times 10^{-3}\ kg/m \cdot s}$$

$$= 4.347 \times 10^6$$

Here, $\mathrm{Re} > 4000$, therefore the flow in the pipe is turbulent.

Step 5 of 7

For turbulent flow, the Colebrook equation is

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

where, f is the friction factor. By substituting the known values.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0.26 \times 10^{-3}\ m/0.35\ m}{3.7} + \frac{2.51}{(4.347 \times 10^6)\sqrt{f}}\right)$$

$$= -2.0\log\left(2.008 \times 10^{-4} + \frac{5.774 \times 10^{-7}}{\sqrt{f}}\right)$$

Solving the equation for f: $f = 0.0184$

Step 6 of 7

The head loss in the flow,

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

$$= 0.0184\left(\frac{200\ m}{0.35\ m}\right)\left(\frac{(12.47\ m/s)^2}{2 \times 9.81\ m/s^2}\right)$$

$$= 83.33\ m$$

Substitute the known values in equation (2).

$$115\ m = h_{turbine,e} + 83.33\ m$$

$$h_{turbine,e} = 31.67\ m$$

Step 7 of 7

The power extracted from the water,

$$\dot{W}_{turbine,e} = \rho \dot{V} g h_{turbine,e}$$

$$= (998\ kg/m^3)(1.2\ m^3/s)(9.81\ m/s^2)(31.67\ m)$$

$$= 372.074 \times 10^3\ W$$

$$= 372.074\ kW$$

The output power of the turbine,

$$\dot{W}_{turbine\text{-}gen} = \eta \dot{W}_{turbine,e}$$

$$= \frac{80}{100}(372.074\ kW)$$

$$= 297.66\ kW$$

Therefore, the electric power output of the plant, $\boxed{\dot{W}_{turbine\text{-}gen} = 297.66\ kW}$.

Question: 14.099P pg 1

A water fountain is to be installed at a remote location by attaching a cast iron pipe directly to a water main through which water is flowing at 70°F and 60 psig. The entrance to the pipe is sharp-edged, and the 70-ft-long piping system involves three 90° miter bends without vanes, a fully open gate valve, and an angle valve with a loss coefficient of 5 when fully open. If the system is to provide water at a rate of 15 gal/min and the elevation difference between the pipe and the fountain is negligible, determine the minimum diameter of the piping system.

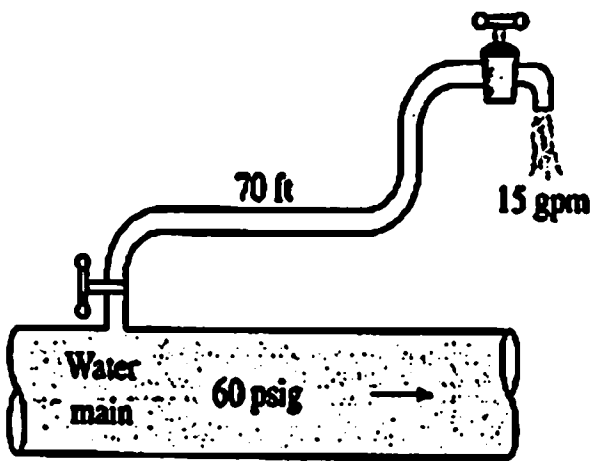


FIGURE P14–99E

Step 1 of 10

Given data:

Temperature of the water $T = 70°F$

The gage pressure, $P_1 = 60 \text{ psi}$

The pipe is having sharp edged entrance, three 90° miter bends, a fully opened gate valve and an angle valve.

The pipe is made of cast iron.

The volume flow rate, $\dot{V} = 15 \text{ gal/min} = 0.0334 \text{ ft}^3/\text{sec}$

The length of the pipe, $L = 70 \text{ ft}$

Step 2 of 10

The flow is assumed to be turbulent, fully developed and incompressible. The elevation deference between the pipe and the fountain is negligible.

From the table of properties of saturated water, corresponding to 70°F,

The density of water, $\rho_f = 62.30 \text{ lbm/ft}^3$

The dynamic viscosity, $\mu_f = 6.556 \times 10^{-4} \text{ lbm/ft-s}$

The roughness of cast iron, $\varepsilon = 0.00085 \text{ ft}$.

Step 3 of 10

Let the point in the main pipe, near the entrance, be 1. At that point the velocity is low. Let the exit be point 2. The pressure at point 2 is equal to the atmospheric pressure.

The energy equation for a control volume between two points is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L \qquad \text{..... (1)}$$

Here, there is no pump or turbine in the pipe system. So, $h_{\text{pump},u} = h_{\text{turbine},e} = 0$.

The effect of kinetic energy correction factor is assumed to be negligible. So,

$\alpha_1 = \alpha_2 = 1$.

Step 4 of 10

Elevation difference between the pipe and the fountain is assumed to be negligible. So,

$z_1 = z_2$

The velocities,

$V_1 \approx 0$

$$V_2 = \frac{\dot{V}}{A_t}$$

$$= \frac{0.0334 \text{ ft}^3/\text{s}}{\frac{\pi}{4} D^2}$$

$$= \frac{0.0425 \text{ ft}^3/\text{s}}{D^2} \qquad \text{..... (2)}$$

Step 5 of 10

The loss coefficient of sharp edged entrance, $K_{L,\text{entrance}} = 0.5$

The loss coefficient of miter bend, $K_{L,\text{bend}} = 1.1$

The loss coefficient of fully opened gate valve, $K_{L,\text{gate}} = 0.2$

The loss coefficient of fully opened angle valve, $K_{L,\text{angle}} = 5$

Total loss coefficient,

$$\sum K_L = K_{L,\text{entrance}} + 3\left(K_{L,\text{bend}}\right) + K_{L,\text{gate}} + K_{L,\text{angle}}$$

$$= 0.5 + 3(1.1) + 0.2 + 5$$

$$= 9$$

Step 6 of 10

The head loss,

$$h_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{V_2^2}{2g}$$

$$= \left(f\frac{70 \text{ ft}}{D} + 9\right)\frac{\left(\frac{0.0425 \text{ ft}^3/\text{s}}{D^2}\right)^2}{2\left(32.2 \text{ ft/s}^2\right)}$$

$$= \left(f\frac{70}{D} + 9\right)\left(\frac{2.825 \times 10^{-5}}{D^4}\right) \qquad \text{.....(3)}$$

Step 7 of 10

Substituting the known values in equation (1):

$$\frac{60\frac{\text{lbm}}{(1/12)^2 \text{ ft}^2}}{(62.30 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)} + 0 + z_1 + 0 = 0 + (1)\frac{\left(\frac{0.0425}{D^2}\right)^2}{2(32.2 \text{ ft/s}^2)} + z_1 + 0$$

Substituting the known values in equation (1):

$$\frac{60\frac{\text{lbm}}{(1/12)^2\,\text{ft}^2}}{(62.30\,\text{lbm/ft}^3)(32.2\,\text{ft/s}^2)}+0+z_1+0=0+(1)\frac{\left(\frac{0.0425}{D^2}\right)^2}{2(32.2\,\text{ft/s}^2)}+z_1+0$$

$$+\left(f\frac{70}{D}+9\right)\left(\frac{2.825\times10^{-5}}{D^4}\right)$$

$$\frac{8640}{2006.06}=\frac{2.80\times10^{-5}}{D^4}+\left(f\frac{70}{D}+9\right)\left(\frac{2.825\times10^{-5}}{D^4}\right)$$

$$4.307=\left(\frac{2.80\times10^{-5}}{D^4}\right)\left(1+f\frac{70}{D}+9\right)$$

$$=\left(\frac{2.80\times10^{-5}}{D^4}\right)\left(10+f\frac{70}{D}\right)$$

Step 8 of 10

On rearranging,

$$\frac{4.306D^4}{2.80\times10^{-5}}-10=f\frac{70}{D}$$

$$f=\left(153.78\times10^3D^4-10\right)\frac{D}{70}$$

Step 9 of 10

The Reynolds number,

$$\text{Re}=\frac{\rho VD}{\mu}$$

$$=\frac{(62.30\,\text{lbm/ft}^3)\left(\frac{0.0425\,\text{ft}^3/\text{s}}{D^2}\right)D}{6.556\times10^{-4}\,\text{lbm/ft}\cdot\text{s}}$$

$$=\frac{4038.66}{D}\qquad\text{..... (5)}$$

Step 10 of 10

For turbulent flow, the Colebrook equation is $\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)$.

where, f is the friction factor.

Substituting the known values:

$$\frac{1}{\sqrt{\left(153.78\times10^3D^4-10\right)\frac{D}{70}}}$$

$$=-2.0\log\left(\frac{0.00085/D}{3.7}+\frac{2.51}{\frac{4038.66}{D}\sqrt{\left(153.78\times10^3D^4-10\right)\frac{D}{70}}}\right)$$

$$=-2.0\log\left(\frac{2.297\times10^{-4}}{D}+\frac{6.215\times10^{-4}D}{\sqrt{\left(153.78\times10^3D^4-10\right)\frac{D}{70}}}\right)$$

$$\frac{1}{\sqrt{\left((153780x^4)-10\right)\times\left(\frac{x}{70}\right)}}=-2\times\ln\left(\frac{(0.0002297)}{x}+\frac{(0.0006215x)}{\sqrt{\left((153780x^4)-10\right)\times\left(\frac{x}{70}\right)}}\right)$$

By solving the above equation,

$D=0.0591\,\text{ft}$

$=0.712\,\text{in.}$

Therefore, the minimum diameter of the piping system is $\boxed{D=0.71\,\text{in.}}$

Question: 14.100P

Repeat Prob. 14-99E for plastic (smooth) pipes.

Step 1 of 8

Given data:

Temperature of the water $T = 20°C$

The gage pressure, $P_1 = 400\ \text{kPa}$

The pipe is having sharp edged entrance, three 90° miter bends, a fully opened gate valve and an angle valve.

The pipe is made of plastic.

The volume flow rate, $\dot{V} = 75\ \text{L/min}$

The length of the pipe, $L = 15\ \text{m}$

Step 2 of 8

From the table of properties of saturated water, corresponding to 20°C,

The density of water, $\rho = 998\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ \text{kg/m} \cdot s$

The roughness of plastic, $\varepsilon = 0$

Let the point in the main pipe, near the entrance, be 1. At this point the velocity is low. Let the exit be point 2. The pressure at point 2 is equal to the atmospheric pressure.

The energy equation for a control volume between two points is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L \qquad \text{..... (1)}$$

Here, there is no pump or turbine in the pipe system. So, $h_{\text{pump},s} = h_{\text{turbine},e} = 0$.

The effect of kinetic energy correction factor is assumed to be negligible. So,

$\alpha_1 = \alpha_2 = 1$.

Step 3 of 8

Elevation difference between the pipe and the fountain is assumed to be negligible. So,

$z_1 = z_2$

The velocities,

$V_1 \approx 0$

$$V_2 = \frac{\dot{V}}{A_t} = \frac{1.25 \times 10^{-3}\ \text{m}^3/\text{s}}{\frac{\pi}{4} D^2} = \frac{1.591 \times 10^{-3}\ \text{m}^3/\text{s}}{D^2} \qquad \text{..... (2)}$$

Step 4 of 8

The loss coefficient of sharp edged entrance, $K_{L,\text{entrance}} = 0.5$

The loss coefficient of miter bend, $K_{L,\text{bend}} = 1.1$

The loss coefficient of fully opened gate valve, $K_{L,\text{gate}} = 0.2$

The loss coefficient of fully opened angle valve, $K_{L,\text{angle}} = 5$

Total loss coefficient,

$$\sum K_L = K_{L,\text{entrance}} + 3\left(K_{L,\text{bend}}\right) + K_{L,\text{gate}} + K_{L,\text{angle}} = 0.5 + 3(1.1) + 0.2 + 5 = 9$$

Step 5 of 8

The head loss in the flow,

$$h_L = \left(f\frac{L}{D} + \sum K_L\right)\frac{V_2^2}{2g} = \left(f\frac{15\ \text{m}}{D} + 9\right)\frac{\left(\frac{1.591 \times 10^{-3}\ \text{m}^3/\text{s}}{D^2}\right)^2}{2\left(9.81\ \text{m/s}^2\right)} = \left(f\frac{15}{D} + 9\right)\left(\frac{1.29 \times 10^{-7}}{D^4}\right) \qquad \text{..... (3)}$$

Step 6 of 8

Substitute the known values in equation (1).

$$\frac{400 \times 10^3\ \text{Pa}}{\left(998\ \text{kg/m}^3\right)\left(9.81\ \text{m/s}^2\right)} + 0 + z_1 + 0 = \left\{ 0 + (1)\frac{\left(\frac{1.591 \times 10^{-3}\ \text{m}^3/\text{s}}{D^2}\right)^2}{2\left(9.81\ \text{m/s}^2\right)} + z_1 + 0 + \left(f\frac{15}{D} + 9\right)\left(\frac{1.29 \times 10^{-7}}{D^4}\right) \right\}$$

$$40.856 = \left(\frac{1.29 \times 10^{-7}}{D^4}\right) + \left(f\frac{15}{D} + 9\right)\left(\frac{1.29 \times 10^{-7}}{D^4}\right) = \left(\frac{1.29 \times 10^{-7}}{D^4}\right)\left(f\frac{15}{D} + 9 + 1\right) = \left(\frac{1.29 \times 10^{-7}}{D^4}\right)\left(f\frac{15}{D} + 10\right)$$

Step 7 of 8

On rearranging,

$$\frac{40.856 D^4}{1.29 \times 10^{-7}} - 10 = f\frac{15}{D}$$

$$f = \left(316.71 \times 10^6 D^4 - 10\right)\frac{D}{15}$$

The Reynolds number,

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{\left(998\ \text{kg/m}^3\right)\left(\frac{1.591 \times 10^{-3}\ \text{m}^3/\text{s}}{D^2}\right)D}{1.002 \times 10^{-3}\ \text{kg/m} \cdot s} = \frac{1584.65}{D} \qquad \text{..... (5)}$$

Step 8 of 8

For turbulent flow, the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

where, f is the friction factor.

Substituting the known values:

$$\frac{1}{\sqrt{\left(316.71 \times 10^6 D^4 - 10\right)\frac{D}{15}}} = -2.0 \log\left(\frac{0/D}{3.7} + \frac{2.51}{\frac{1584.65}{D}\sqrt{\left(316.71 \times 10^6 D^4 - 10\right)\frac{D}{15}}}\right)$$

$D = 0.0169\ \text{m}$

Therefore, the minimum diameter of the piping system, $\boxed{D = 0.0169\ \text{m}}$

Question: 14.101P

In a hydroelectric power plant, water at 20°C is supplied to the turbine at a rate of 1.2 m3/s through a 200-m- long, 0.35-m-diameter cast iron pipe. The elevation difference between the free surface of the reservoir and the turbine discharge is 115 m, and the combined turbine-generator efficiency is 80 percent. Disregarding the minor losses because of the large length-to-diameter ratio, determine the electric power output of this plant.

Step 1 of 8

Given data:

Temperature of the supplied water $T = 20°C$

The flow rate of the water, $\dot{V} = 1.2\ m^3/s$

The length of the pipe, $L = 200\ m$

The diameter of the pipe, $D = 0.35\ m$

The elevation difference between the free surface of the reservoir and turbine discharge, $H = 115\ m$

The combined turbine-generator efficiency, $\eta = 80\%$

The pipe is made of cast iron.

Step 2 of 8

From the table of properties of saturated water, corresponding to 20°C,

The density of water, $\rho = 998\ kg/m^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ kg/m \cdot s$

The roughness of cast iron pipe, $\varepsilon = 0.26\ mm = 0.26 \times 10^{-3}\ m$.

Step 3 of 8

The cross section of the flow area,

$$A_c = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}(0.35\ m)^2$$

$$= 96.211 \times 10^{-3}\ m^2$$

The energy equation for a control volume is given by

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 + h_{pump,e} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \qquad \text{..... (1)}$$

Where, point 1 is taken as the surface of the reservoir
point 2 is taken as the exit of the hole.

Step 4 of 8

Here, both the points 1 and 2 are opened to the atmosphere. So, $P_1 = P_2$

The velocity at the points 1 and 2 is low. So, the velocities, $V_1 = V_2 = 0$

Let point 2 is the reference level. Then, $z_2 = 0$ and $z_1 = H = 115\ m$

There is no pump. So, $h_{pump,e} = 0$

Thus, equation (1) becomes,

$$\frac{P_1}{\rho g} + \alpha_1\frac{(0)^2}{2g} + 115\ m + 0 = \frac{P_1}{\rho g} + \alpha_2\frac{(0)^2}{2g} + 0 + h_{turbine,e} + h_L$$

$$h_{turbine,e} + h_L = 115\ m \qquad \text{..... (2)}$$

Step 5 of 8

The average velocity of flow in the pipe,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{1.2\ m^3/s}{96.211 \times 10^{-3}\ m^2}$$

$$= 12.47\ m/s$$

The Reynolds number,

$$\mathrm{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(998\ kg/m^3)(12.47\ m/s)(0.35\ m)}{1.002 \times 10^{-3}\ kg/m \cdot s}$$

$$= 4.347 \times 10^6$$

Here, $\mathrm{Re} > 4000$, therefore the flow in the pipe is turbulent.

Step 6 of 8

For turbulent flow, the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

Where, f is the friction factor. By substituting the known values

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0.26 \times 10^{-3}\ m/0.35\ m}{3.7} + \frac{2.51}{(4.347 \times 10^6)\sqrt{f}}\right)$$

$$= -2.0\log\left(2.008 \times 10^{-4} + \frac{5.774 \times 10^{-7}}{\sqrt{f}}\right)$$

Solving the equation for f: $f = 0.0184$

Step 7 of 8

The head loss,

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

$$= 0.0184\left(\frac{200\ m}{0.35\ m}\right)\left(\frac{(12.47\ m/s)^2}{2 \times 9.81\ m/s^2}\right)$$

$$= 83.33\ m$$

Substitute the known values in equation (2)

$$115\ m = h_{turbine,e} + 83.33\ m$$

$$h_{turbine,e} = 31.67\ m$$

Step 8 of 8

The power extracted from the water,

$$\dot{W}_{turbine,e} = \rho\dot{V}gh_{turbine,e}$$

$$= (998\ kg/m^3)(1.2\ m^3/s)(9.81\ m/s^2)(31.67\ m)$$

$$= 372.074 \times 10^3\ W$$

$$= 372.074\ kW$$

The output power of the turbine,

$$\dot{W}_{turbine-gen} = \eta\dot{W}_{turbine,e}$$

$$= \frac{80}{100}(372.074\ kW)$$

$$= 297.66\ kW$$

Therefore, the electric power output of the plant, $\boxed{\dot{W}_{turbine-gen} = 297.66\ kW}$

Question: 14.102P pg 1

In Prob. 14-101, the pipe diameter is tripled in order to reduce the pipe losses. Determine the percent increase in the net power output as a result of this modification.

Step 1 of 14

Given data:

Temperature of the water supplied $T = 20°C$

The flow rate of the water, $\dot{V} = 1.2\ m^3/s$

The length of the pipe, $L = 200\ m$

The diameter of the pipe, $D = 0.35\ m$

The elevation difference between the free surface of the reservoir and turbine discharge, $H = 115\ m$

The combined turbine-generator efficiency, $\eta = 80\%$

The pipe is made of cast iron.

Step 2 of 14

From the table of properties of saturated water, corresponding to $20°C$,

The density of water, $\rho = 998\ kg/m^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ kg/m \cdot s$

The roughness of cast iron pipe, $\varepsilon = 0.26\ mm$

Step 3 of 14

The cross section area of the flow,

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.35\ m)^2$$

$$= 96.211 \times 10^{-3}\ m^2$$

Step 4 of 14

The energy equation for a control volume is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump,a} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine,e} + h_L \qquad \text{..... (1)}$$

Where, point 1 is taken as the surface of the reservoir

point 2 is taken as the exit of the hole.

Here, both the points 1 and 2 are opened to the atmosphere. So, $P_1 = P_2$

The velocity at the points 1 and 2 is low. So, the velocities, $V_1 = V_2 = 0$

Let point 2 is the reference level. Then, $z_2 = 0$ and $z_1 = H = 115\ m$.

There is no pump. So, $h_{pump,a} = 0$

Step 5 of 14

Thus, equation (1) becomes,

$$\frac{P_1}{\rho g} + \alpha_1 \frac{(0)^2}{2g} + 115\ m + 0 = \frac{P_1}{\rho g} + \alpha_2 \frac{(0)^2}{2g} + 0 + h_{turbine,e} + h_L$$

$$115\ m = h_{turbine,e} + h_L \qquad \text{..... (2)}$$

The average velocity of flow in the pipe,

$$V = \frac{\dot{V}}{A_c}$$

$$= \frac{1.2\ m^3/s}{96.211 \times 10^{-3}\ m^2}$$

$$= 12.47\ m/s$$

Step 6 of 14

The Reynolds number,

$$\mathrm{Re} = \frac{\rho V D}{\mu}$$

$$= \frac{(998\ kg/m^3)(12.47\ m/s)(0.35\ m)}{1.002 \times 10^{-3}\ kg/m \cdot s}$$

$$= 4.347 \times 10^6$$

Here, $\mathrm{Re} > 4000$, therefore the flow in the pipe is turbulent.

Step 7 of 14

For turbulent flow, the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{e/D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

where, f is the friction factor. By substitute the known values.

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0.26 \times 10^{-3}\ m / 0.35\ m}{3.7} + \frac{2.51}{(4.347 \times 10^6)\sqrt{f}}\right)$$

$$= -2.0 \log\left(2.008 \times 10^{-4} + \frac{5.774 \times 10^{-7}}{\sqrt{f}}\right)$$

By solving the above equation we get, $f = 0.0184$

Step 8 of 14

The head loss,

$$h_L = f \frac{L}{D} \frac{V^2}{2g}$$

$$= 0.0184\left(\frac{200\ m}{0.35\ m}\right)\left(\frac{(12.47\ m/s)^2}{2 \times 9.81\ m/s^2}\right)$$

$$= 83.33\ m$$

Substitute the known values in equation (2).

$$115\ m = h_{turbine,e} + 83.33\ m$$

$$h_{turbine,e} = 31.67\ m$$

Step 9 of 14

Substitute the known values in equation (2).

$115\text{ m} = h_{\text{turbine},e} + 83.33\text{ m}$ 14.102P pg 2

$h_{\text{turbine},e} = 31.67\text{ m}$

Step 9 of 14

Now, considering the system with diameter $3D$ The cross sectional flow area,

$$A_c = \frac{\pi}{4}(3D)^2$$
$$= \frac{9\pi}{4}(0.35\text{ m})^2$$
$$= 865.899 \times 10^{-3}\text{ m}^2$$

The energy equation for a control volume is given by

$$\frac{P_1'}{\rho g} + \alpha_1 \frac{V_1'^2}{2g} + z_1' + h_{\text{pump},s}' = \frac{P_2'}{\rho g} + \alpha_2 \frac{V_2'^2}{2g} + z_2' + h_{\text{turbine},e}' + h_L' \quad \text{..... (3)}$$

where, point 1 is taken as the surface of the reservoir and point 2 is the exit of the hole.

Here, both the points 1 and 2 are opened to the atmosphere. So, $P_1' = P_2'$.

The velocity at the points 1 and 2 is low. So, the velocities, $V_1' = V_2' = 0$.

Let point 2 be the reference level. Then, $z_2' = 0$ and $z_1' = H = 115\text{ m}$.
There is no pump. So, $h_{\text{pump},s}' = 0$.

Step 10 of 14

Thus, equation (3) becomes,

$$\frac{P_1'}{\rho g} + \alpha_1 \frac{(0)^2}{2g} + 115\text{ m} + 0 = \frac{P_1'}{\rho g} + \alpha_2 \frac{(0)^2}{2g} + 0 + h_{\text{turbine},e}' + h_L'$$
$$115\text{ m} = h_{\text{turbine},e}' + h_L' \quad \text{..... (4)}$$

The average velocity of flow in the pipe,

$$V' = \frac{\dot{V}}{A_c'}$$
$$= \frac{1.2\text{ m}^3/\text{s}}{865.899 \times 10^{-3}\text{ m}^2}$$
$$= 1.386\text{ m/s}$$

Step 11 of 14

The Reynolds number,

$$\text{Re}' = \frac{\rho V' D'}{\mu}$$
$$= \frac{(998\text{ kg/m}^3)(1.386\text{ m/s})(3 \times 0.35\text{ m})}{1.002 \times 10^{-3}\text{ kg/m} \cdot \text{s}}$$
$$= 1.449 \times 10^6$$

Here, $\text{Re}' > 4000$, therefore the flow in the pipe is turbulent.

Step 12 of 14

For turbulent flow, the Colebrook equation is

$$\frac{1}{\sqrt{f'}} = -2.0 \log\left(\frac{\varepsilon/D'}{3.7} + \frac{2.51}{\text{Re}'\sqrt{f'}}\right)$$

Substitute the known values.

$$\frac{1}{\sqrt{f'}} = -2.0 \log\left(\frac{0.26 \times 10^{-3}\text{ m}/(3 \times 0.35\text{ m})}{3.7} + \frac{2.51}{(1.449 \times 10^6)\sqrt{f'}}\right)$$
$$= -2.0 \log\left(66.924 \times 10^{-6} + \frac{1.732 \times 10^{-6}}{\sqrt{f'}}\right)$$

By solving the above equation, we get $f' = 0.0149$

Step 13 of 14

The head loss,

$$h_L' = f' \frac{L}{D'} \frac{V'^2}{2g}$$
$$= 0.0149\left(\frac{200\text{ m}}{3 \times 0.35\text{ m}}\right)\left(\frac{(1.386\text{ m/s})^2}{2 \times 9.81\text{ m/s}^2}\right)$$
$$= 0.28\text{ m}$$

Substitute the known values in equation (2).

$115\text{ m} = h_{\text{turbine},e}' + 0.28\text{ m}$

$h_{\text{turbine},e}' = 114.72\text{ m}$

Step 14 of 14

The increase in actual power output,

$$\%\text{ increse} = \frac{\eta \dot{W}_{\text{turbine},e}' - \eta \dot{W}_{\text{turbine},e}}{\eta \dot{W}_{\text{turbine},e}}$$
$$= \frac{\dot{W}_{\text{turbine},e}' - \dot{W}_{\text{turbine},e}}{\dot{W}_{\text{turbine},e}}$$
$$= \frac{\rho \dot{V} g h_{\text{turbine},e}' - \rho \dot{V} g h_{\text{turbine},e}}{\rho \dot{V} g h_{\text{turbine},e}}$$
$$= \frac{h_{\text{turbine},e}' - h_{\text{turbine},e}}{h_{\text{turbine},e}}$$
$$= \frac{114.72\text{ m} - 31.67\text{ m}}{31.67\text{ m}}$$
$$= 2.622$$
$$= 262.2\%$$

Therefore, the percentage increase in output is $\boxed{262.2\%}$.

= 262.2%

14.102P pg 3

Therefore, the percentage increase in output is **262.2%**.

Question: 14.103P pg 1

The drinking water needs of an office are met by large water bottles. One end of a 0.35-in-diameter, 6-ft-long plastic hose is inserted into the bottle placed on a high stand, while the other end with an on/off valve is maintained 3 ft below the bottom of the bottle. If the water level in the bottle is 1 ft when it is full, determine how long it would take to fill an 8-oz glass (= 0.00835 ft3) (*a*) when the bottle is first opened and (*b*) when the bottle is almost empty. Take the total minor loss coefficient, including the on/off valve, to be 2.8 when it is fully open. Assume the water temperature to be the same as the room temperature of 70°F.

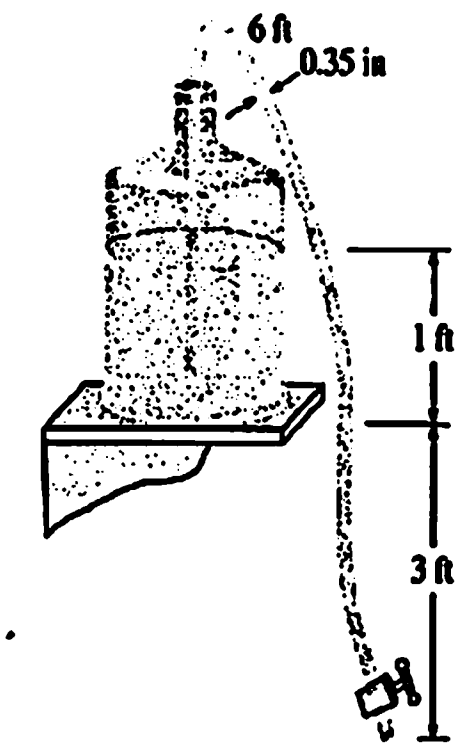


FIGURE P14–103E

Step 1 of 12

Diameter of the plastic hose, $D = 0.35\text{in.} = 0.029\text{ft}$

Total length of the plastic hose, $L = 6\text{ ft}$

Minor loss coefficient, $\sum K_l = 2.8$

Volume to be filled, $Vol = 0.00835\text{ ft}^3$

Acceleration due to gravity, $g = 32.2\text{ ft/s}^2$

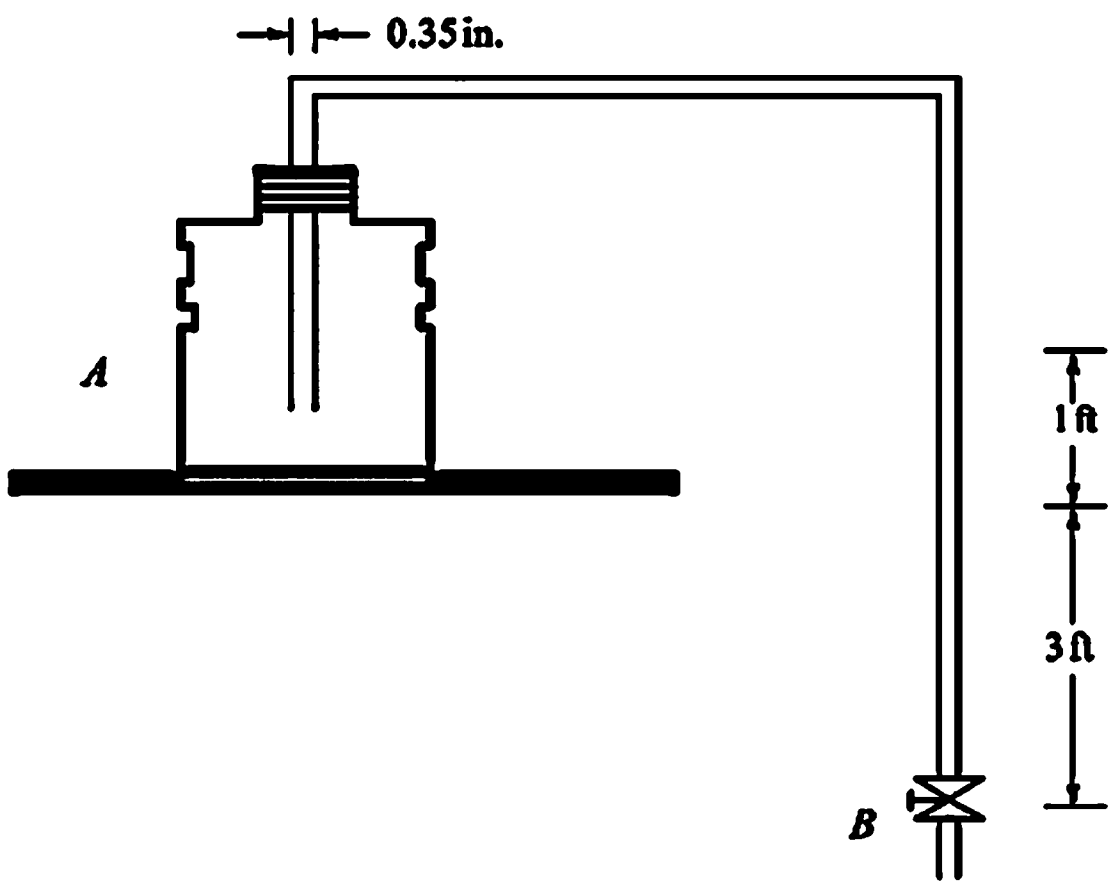
Properties of water at 70°C

Density of water, $\rho = 62.3\text{lbm/ft}^3$

Dynamic viscosity of water, $\nu = 1.052\times10^{-5}\text{ ft}^2/s$

Step 2 of 12

Figure showing dinking water arrangement,



Step 3 of 12

The energy balance equation between free surface of water *A* and valve *B* is given by,

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{turbine}+h_{loss} \qquad \ldots\ldots (1)$$

The pressure acting on the free surface of the water and at the end of the pipe is atmospheric pressure only, $P_1 = P_2$

The velocity of the water at free surface is, $V_1 = 0$

We know that there is no involvement of pumps and turbines therefore,

$h_{pump} = 0,\ h_{turbine} = 0$

We know that total head losses is given by,

$$h_{loss}=\left(f\frac{L}{D}+\sum K_l\right)\frac{V_2^2}{2g}$$

$$=\left(f\frac{6}{0.029}+2.8\right)\frac{V_2^2}{2\times32.2}$$

$$=(3.21f+0.0434)V_2^2$$

Step 4 of 12

The equation (1) reduces to,

$$\frac{P_1}{\rho g}+\alpha_1\frac{0}{2g}+z_1+0=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+0+0+h_{loss}$$

$$z_1=\alpha_2\frac{V_2^2}{2g}+h_{loss}$$

$$=1\times\frac{V_2^2}{2\times32.2}+\left[(3.21f+0.0434)V_2^2\right]$$

$$z_1=(3.21f+0.0584)V_2^2 \qquad \ldots\ldots (2)$$

Step 5 of 12

Discharge by the hose is fully open is given by,

$$\dot{V}=V_2\times A$$

$$=V_2\times\frac{\pi}{4}D^2$$

$$=V_2\times\frac{\pi}{4}\times(0.029)^2$$

$$\dot{V}=6.6\times10^{-4}(V_2) \qquad \ldots\ldots (3)$$

Reynolds number is given by,

$$= V_2 \times \frac{\pi}{4} \times (0.029)^2$$

$$\dot{V} = 6.6 \times 10^{-4} (V_2) \qquad \ldots\ldots (3)$$

Reynolds number is given by,

$$\mathrm{Re} = \frac{V_2 D}{\nu}$$

$$= \frac{V_2 \times 0.029}{1.052 \times 10^{-5}}$$

$$\mathrm{Re} = 2756.65(V_2) \qquad \ldots\ldots (4)$$

Time taken for filing the glass,

$$t = \frac{Vol}{\dot{V}}$$

$$= \frac{0.00835}{6.6 \times 10^{-4} (V_2)}$$

$$t = \frac{12.64}{V_2} \qquad \ldots\ldots (5)$$

Step 6 of 12

Frictional factor is given by,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$= -2.0 \log\left(0 + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \qquad \text{(Since roughness factor, } \varepsilon = 0\text{)}$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \qquad \ldots\ldots (6)$$

Step 7 of 12

a) When the bottle is fist opened,

The height of the frees surface of water,

z_1 = Height from glass to bottle + Height of water free surface

$= 3 + 1$

$= 4\,\mathrm{ft}$

Equation (2) simplifies as,

$$4 = (3.21 f + 0.0584) V_2^2$$

$$f = \frac{1}{3.21}\left(\frac{4}{V_2^2} - 0.0584\right) \qquad \ldots\ldots (7)$$

$$= \frac{1.246}{V_2^2} - 0.0181$$

Step 8 of 12

Substituting in equations (4) and (7) in (6) the equation modifies as follows,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{\frac{1.246}{V_2^2} - 0.0181}} = -2.0 \log\left(\frac{2.51}{2756.65 \times V_2 \times \sqrt{\frac{1.246}{V_2^2} - 0.0181}}\right)$$

$$V_2 = 5.19\,\mathrm{ft/s}$$

Reynolds number form equation (4),

$$\mathrm{Re} = 2756.65(V_2)$$

$$= 2756.65(5.19)$$

$$= 14307.01$$

Since the obtained Reynolds number is grater than 4000 flow is in turbulent nature.

Step 9 of 12

Time taken to fill the glass can be found form equation (5) as,

$$t = \frac{12.64}{V_2}$$

$$= \frac{12.64}{5.19}$$

$$= 2.44\,s$$

Therefore the time taken to fill the glass is $\boxed{2.44 \text{seconds}}$ when the bottle is full.

Step 10 of 12

b) When the bottle is almost empty,

The height of the frees surface of water,

z_1 = Height from glass to bottle + Height of water free surface

$= 3 + 0$

$= 3\,\mathrm{ft}$

Equation (2) simplifies as,

$$3 = (3.21 f + 0.0584) V_2^2$$

$$f = \frac{1}{3.21}\left(\frac{3}{V_2^2} - 0.0584\right) \qquad \ldots\ldots (8)$$

$$= \frac{0.9346}{V_2^2} - 0.0181$$

Step 11 of 12

Substituting in equations (4) and (7) in (6) the equation modifies as follows,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{\frac{0.9346}{V_2^2} - 0.0181}} = -2.0 \log\left(\frac{2.51}{2756.65 \times V_2 \times \sqrt{\frac{0.9346}{V_2^2} - 0.0181}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{\frac{0.9346}{V_2^2} - 0.0181}} = -2.0\log\left(\frac{2.51}{2756.65 \times V_2 \times \sqrt{\frac{0.9346}{V_2^2} - 0.0181}}\right)$$

$V_2 = 4.441 \text{ ft/s}$

Reynolds number form equation (4),

$$\begin{aligned}\mathrm{Re} &= 2756.65(V_2)\\ &= 2756.65(4.441)\\ &= 12242.28\end{aligned}$$

Since the obtained Reynolds number is grater than 4000 flow is in turbulent nature.

Step 12 of 12

Time taken to fill the glass can be found form equation (5) as,

$$\begin{aligned}t &= \frac{12.64}{V_2}\\ &= \frac{12.64}{4.441}\\ &= 2.84 s\end{aligned}$$

Therefore the time taken to fill the glass is $\boxed{t = 2.84 \text{ seconds}}$ when the bottle is almost empty.

Question:

Reconsider Prob. 14-103E. Using EES (or other) software, investigate the effect of the hose diameter on the time required to fill a glass when the bottle is full. Let the diameter vary from 0.2 to 2 in, in increments of 0.2 in. Tabulate and plot the results.

Step 1 of 9

Diameter of the plastic hose, D

Total length of the plastic hose, $L = 6$ ft

Minor loss coefficient, $\sum K_l = 2.8$

Volume to be filled, $Vol = 0.00835$ ft^3

Acceleration due to gravity, $g = 32.2$ ft/s^2

Properties of water at 70°F

Density of water, $\rho = 62.31$ lbm/ft^3

Dynamic viscosity of water, $v = 1.052 \times 10^{-5}$ ft^2/s

Step 2 of 9

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

```
"Given data"
"D=0.35 [in]*convert(in,ft)"
L=6 [ft]
K_l=2.8
Vol=0.00835 [ft^3]
e=0                    "Equivalent roughness"
T_1=70 [F]
P_1=14.7 [psi]
rho_1=Density(Water,T=T_1,P=P_1)
mu_1=Viscosity(Water,T=T_1,P=P_1)
```

Step 3 of 9

```
"Energy balance equation between free surface "
(P_1/(rho_1*g#))+(alpha_1*(V_1^2)/(2*g#))+z_1+h_pump=(P_2/(rho_1*g#))+(alpha_2
*(V_2^2)/(2*g#))+z_2+h_turbine+h_loss

P_2=14.7 [psi]         " pressure acting on the free surface of the water"
V_1=0 [ft/s]           "velocity of the water at free surface"
h_pump=0               "No involvement of pumps and turbines"
h_turbine=0
alpha_1=1
alpha_2=1
z_1=4                  "Height of free surface of water"
z_2=0                  "Datum level"
```

Step 4 of 9

```
"Head loss is given by"
h_loss=((f*L/D)+K_L)*(V_2^2/(2*g#))

"Colebrook equation  frictional factor"
1/sqrt(f)=-2*log10(((e/D)/3.7)+(2.51/(Re*sqrt(f))))

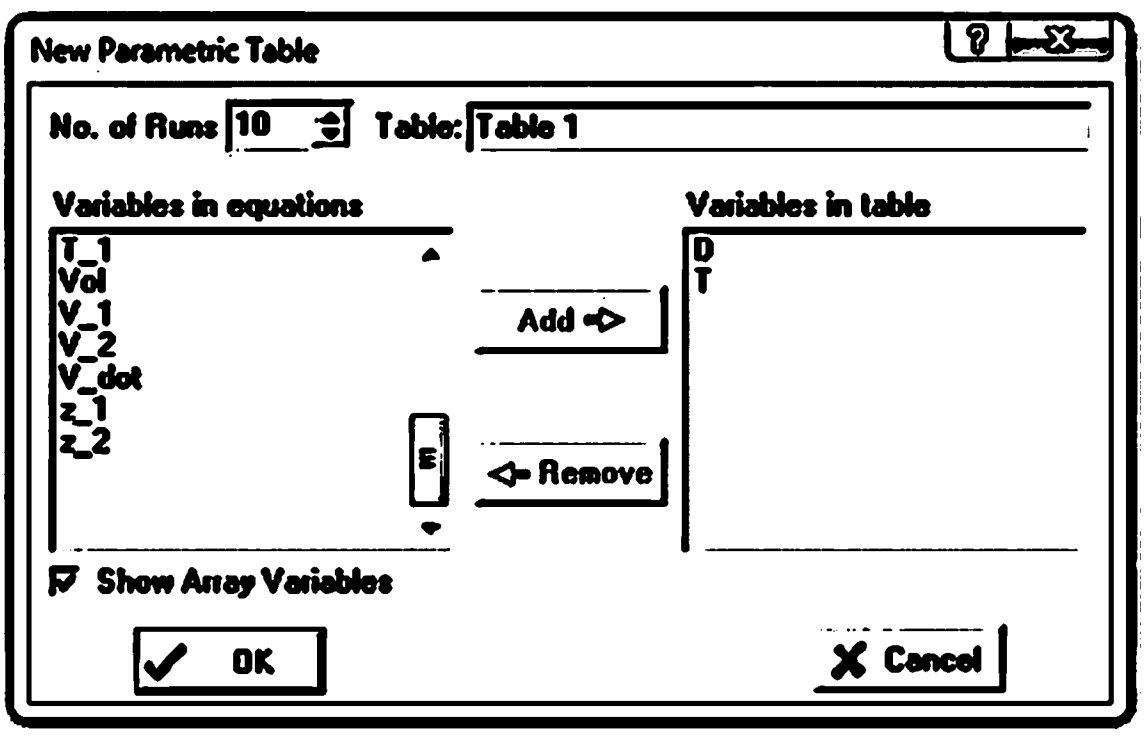
"Discharge"
V_dot=V_2*A
A=pi*D^2/4

"Reynolds number"
Re=rho_1*V_2*D*(3600/mu_1)

"Time taken for filling the glass"
T=Vol/V_dot
```

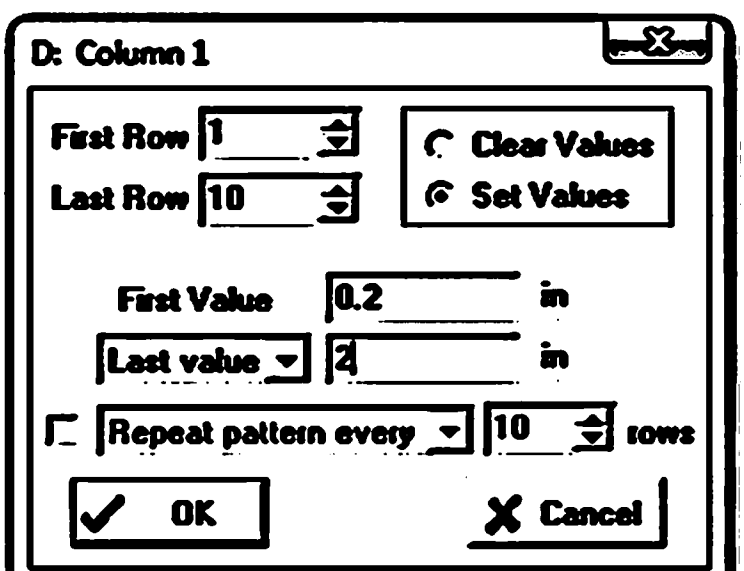
Step 5 of 9

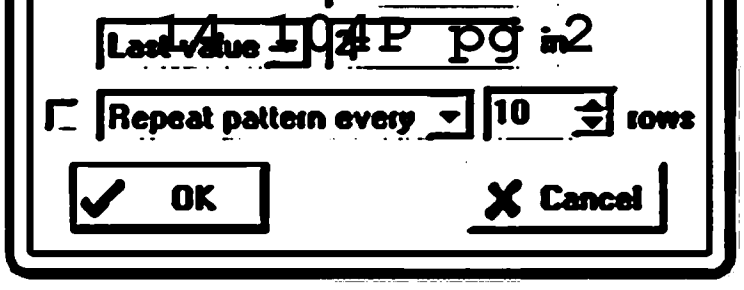
Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 6 of 9

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,





Step 7 of 9

Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

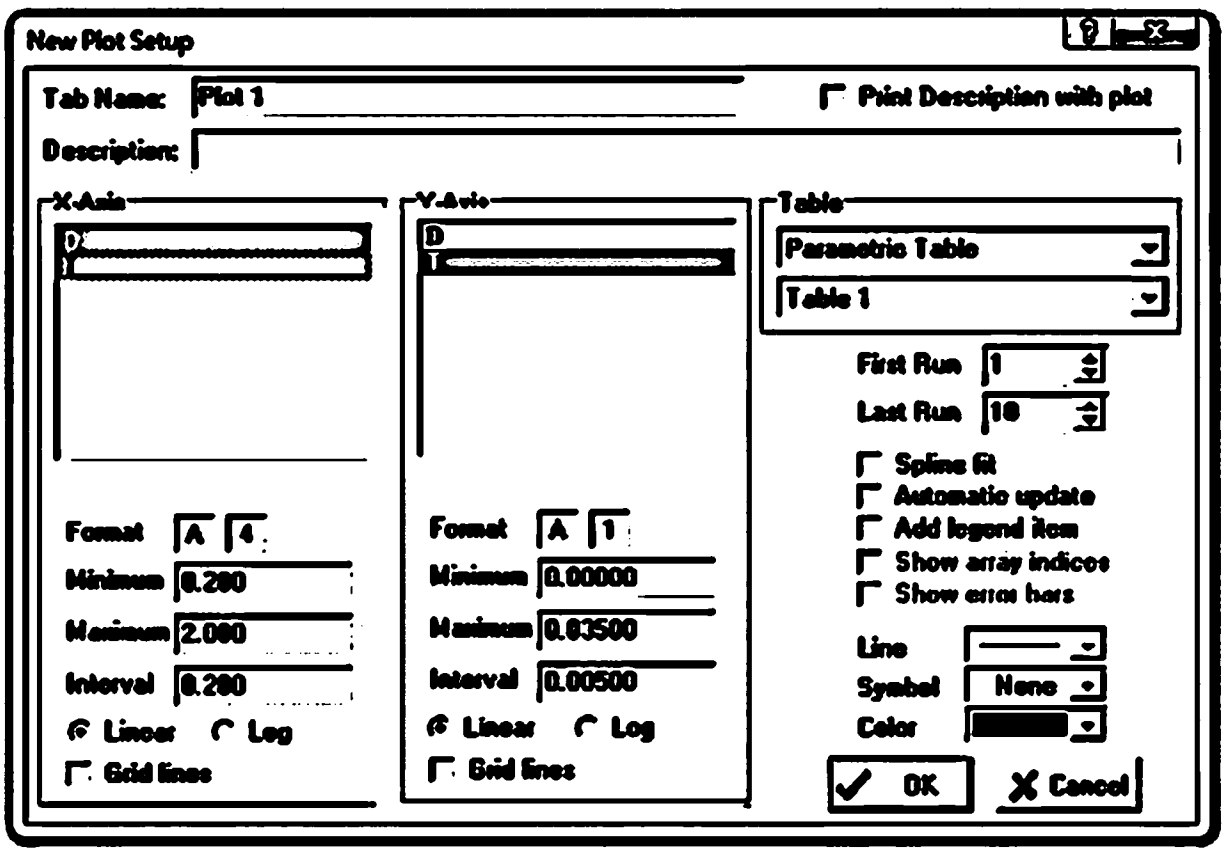
Parametric Table

Table 1

1..10	D [in]	T [s]
Run 1	0.2	0.03435
Run 2	0.4	0.0083
Run 3	0.6	0.003651
Run 4	0.8	0.002043
Run 5	1	0.001304
Run 6	1.2	0.000904
Run 7	1.4	0.0006633
Run 8	1.6	0.0005074
Run 9	1.8	0.0004006
Run 10	2	0.0003243

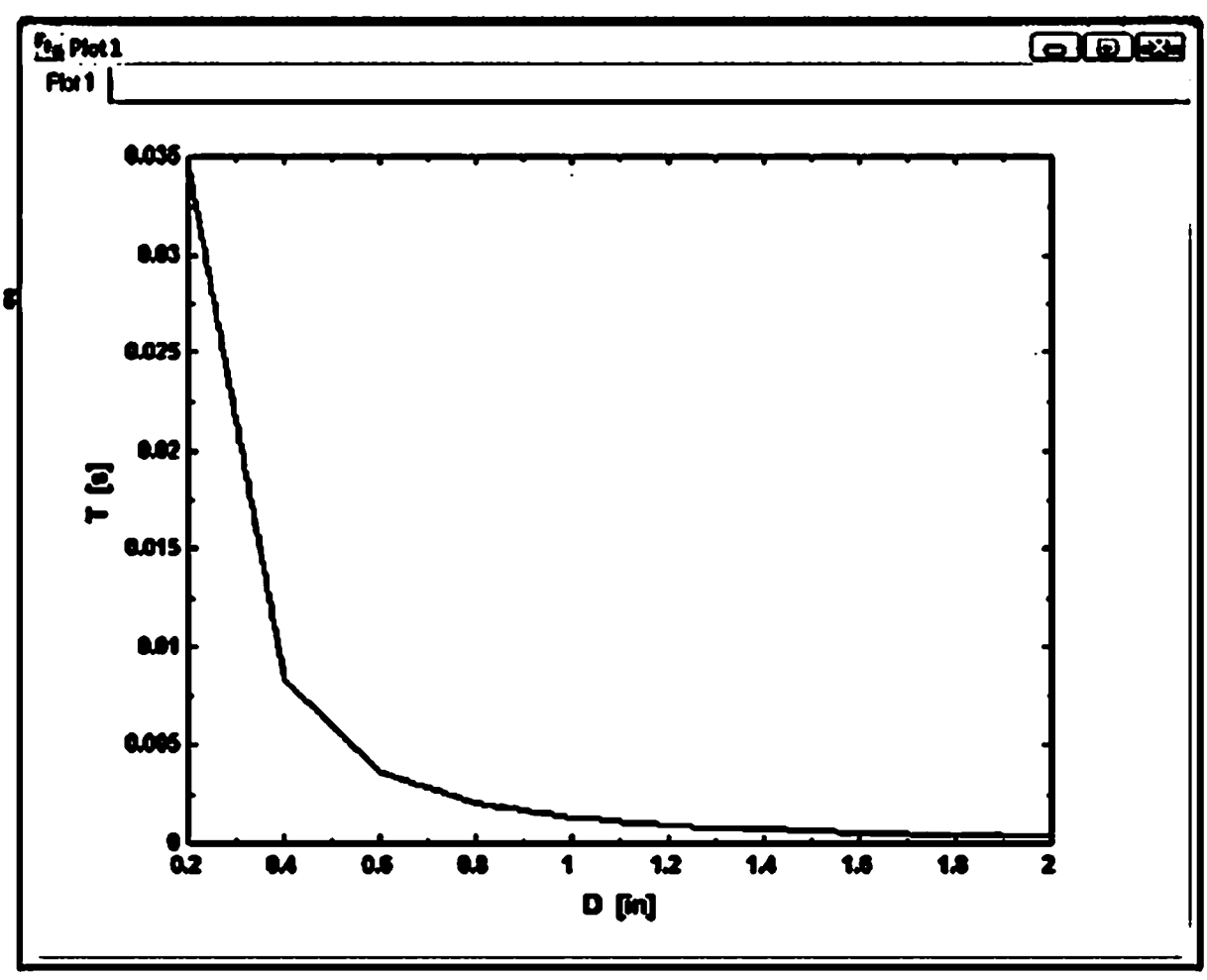
Step 8 of 9

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 9 of 9

By clicking on OK the graph will be generated as shown,



Question: 14.105P pg 1

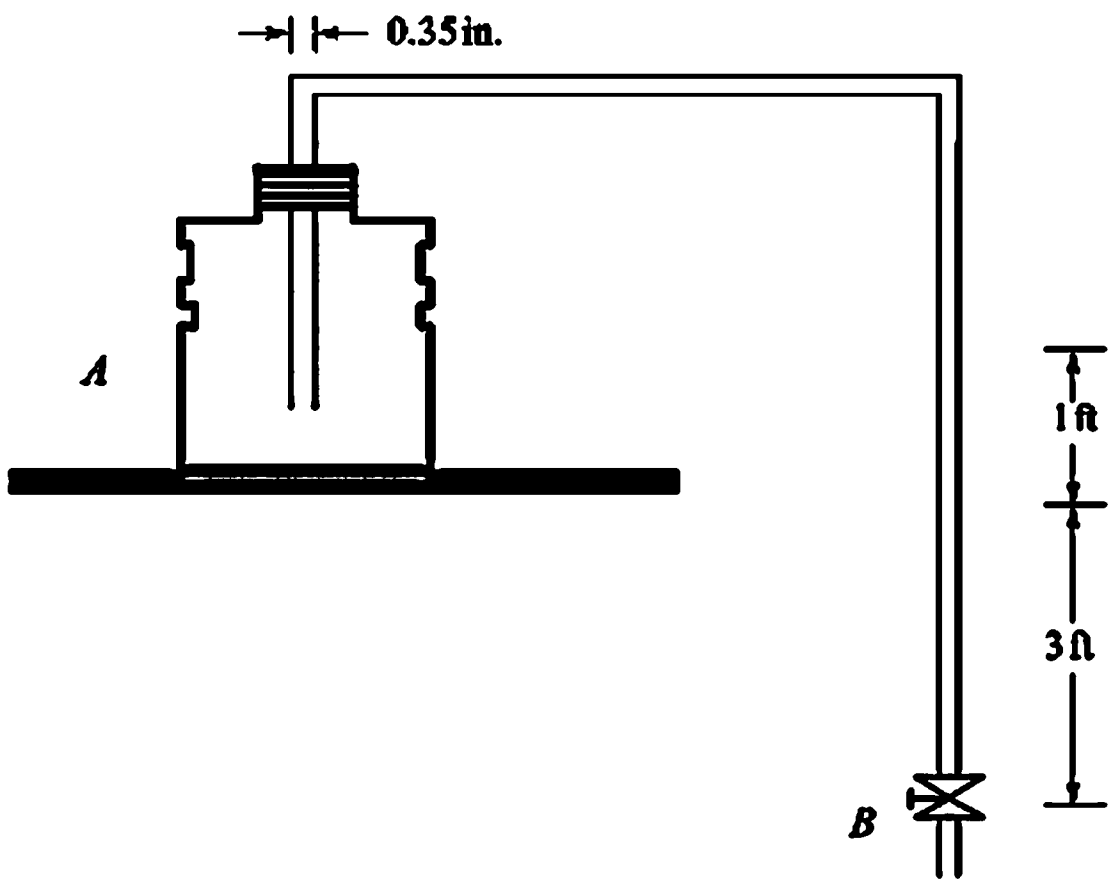
Reconsider Prob. 14-103E. The office worker who set up the siphoning system purchased a 12-ft-long reel of the plastic tube and wanted to use the whole thing to avoid cutting it in pieces, thinking that it is the elevation difference that makes siphoning work, and the length of the tube is not important. So he used the entire 12-ft-long tube. Assuming the turns or constrictions in the tube are not significant (being very optimistic) and the same elevation is maintained, determine the time it takes to fill a glass of water for both cases (bottle nearly full and bottle nearly empty).

Step 1 of 12

Loading solution. Please wait ...

Step 2 of 12

Figure showing dinking water arrangement,



Step 3 of 12

The energy balance equation between free surface of water A and valve B is given by,

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{turbine}+h_{loss} \qquad \text{..... (1)}$$

The pressure acting on the free surface of the water and at the end of the pipe is atmospheric pressure only, $P_1=P_2$

The velocity of the water at free surface is, $V_1=0$

We know that there is no involvement of pumps and turbines therefore,

$h_{pump}=0$, $h_{turbine}=0$

We know that total head losses is given by,

$$h_{loss}=\left(f\frac{L}{D}+\sum K_1\right)\frac{V_2^2}{2g}$$

$$=\left(f\frac{12}{0.029}+2.8\right)\frac{V_2^2}{2\times 32.2}$$

$$=(6.42f+0.0434)V_2^2$$

Step 4 of 12

The equation (1) reduces to,

$$\frac{P_1}{\rho g}+\alpha_1\frac{0}{2g}+z_1+0=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+0+0+h_{loss}$$

$$z_1=\alpha_2\frac{V_2^2}{2g}+h_{loss}$$

$$=1\times\frac{V_2^2}{2\times 32.2}+\left[(6.42f+0.0434)V_2^2\right]$$

$$z_1=(6.42f+0.0584)V_2^2 \qquad \text{..... (2)}$$

Step 5 of 12

Discharge by the hose is fully open is given by,

$$\dot{V}=V_2\times A$$

$$=V_2\times\frac{\pi}{4}D^2$$

$$=V_2\times\frac{\pi}{4}\times(0.029)^2$$

$$\dot{V}=6.6\times10^{-4}(V_2) \qquad \text{..... (3)}$$

Reynolds number is given by,

$$\text{Re}=\frac{V_2D}{\nu}$$

$$=\frac{V_2\times 0.029}{1.052\times10^{-5}}$$

$$\text{Re}=2756.65(V_2) \qquad \text{..... (4)}$$

Time taken for filing the glass,

$$t=\frac{Vol}{\dot{V}}$$

$$=\frac{0.00835}{6.6\times10^{-4}(V_2)}$$

$$t=\frac{12.64}{V_2} \qquad \text{..... (5)}$$

Step 6 of 12

Frictional factor is given by,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$=-2.0\log\left(0+\frac{2.51}{\text{Re}\sqrt{f}}\right) \qquad \text{(Since roughness factor, } \varepsilon=0\text{)}$$

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{2.51}{\text{Re}\sqrt{f}}\right) \qquad \text{..... (6)}$$

$$= -2.0\log\left(0+\frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \quad \text{(Since roughness factor, } \varepsilon = 0\text{)}$$

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \qquad \ldots\ldots (6)$$

Step 7 of 12

a) When the bottle is fist opened,

The height of the frees surface of water,

z_1 = Height from glass to bottle + Height of water free surface

$= 3+1$

$= 4\,\mathrm{ft}$

Equation (2) simplifies as,

$$4 = (6.42f + 0.0584)V_2^2$$

$$f = \frac{1}{6.42}\left(\frac{4}{V_2^2} - 0.0584\right) \qquad \ldots\ldots (7)$$

$$= \left[\frac{0.623}{V_2^2} - \left(9.09\times10^{-3}\right)\right]$$

Step 8 of 12

Substituting in equations (4) and (7) in (6) the equation modifies as follows,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{\frac{0.623}{V_2^2} - \left(9.09\times10^{-3}\right)}} = -2.0\log\left(\frac{2.51}{2756.65\times V_2\times\sqrt{\frac{0.623}{V_2^2} - \left(9.09\times10^{-3}\right)}}\right)$$

$$V_2 = 3.986\,\mathrm{ft/s}$$

Reynolds number form equation (4),

$$\mathrm{Re} = 2756.65(V_2)$$

$$= 2756.65(3.986)$$

$$= 10988$$

Since the obtained Reynolds number is grater than 4000 flow is in turbulent nature.

Step 9 of 12

Time taken to fill the glass can be found form equation (5) as,

$$t = \frac{12.64}{V_2}$$

$$= \frac{12.64}{3.986}$$

$$= 3.171\,\mathrm{s}$$

Therefore the time taken to fill the glass is $\boxed{3.171\,\text{seconds}}$ when the bottle is full.

Step 10 of 12

b) When the bottle is almost empty,

The height of the frees surface of water,

z_1 = Height from glass to bottle + Height of water free surface

$= 3+0$

$= 3\,\mathrm{ft}$

Equation (2) simplifies as,

$$3 = (6.42f + 0.0584)V_2^2$$

$$f = \frac{1}{6.42}\left(\frac{3}{V_2^2} - 0.0584\right) \qquad \ldots\ldots (8)$$

$$= \left[\frac{0.467}{V_2^2} - \left(9.09\times10^{-3}\right)\right]$$

Step 11 of 12

Substituting in equations (4) and (7) in (6) the equation modifies as follows,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{\frac{0.467}{V_2^2} - \left(9.09\times10^{-3}\right)}} = -2.0\log\left(\frac{2.51}{2756.65\times V_2\times\sqrt{\frac{0.467}{V_2^2} - \left(9.09\times10^{-3}\right)}}\right)$$

$$V_2 = 3.395\,\mathrm{ft/s}$$

Reynolds number form equation (4),

$$\mathrm{Re} = 2756.65(V_2)$$

$$= 2756.65(3.395)$$

$$= 9358.82$$

Since the obtained Reynolds number is grater than 4000 flow is in turbulent nature.

Step 12 of 12

Time taken to fill the glass can be found form equation (5) as,

$$t = \frac{12.64}{V_2}$$

$$= \frac{12.64}{3.395}$$

$$= 3.72\,\mathrm{s}$$

Therefore the time taken to fill the glass is $\boxed{t = 3.72\,\text{seconds}}$ when the bottle is almost empty.

$= 3.72s$

Therefore the time taken to fill the glass is $\boxed{t = 3.72 \text{ seconds}}$ when the bottle is almost empty.

Question: 14.106P pg 1

A circular water pipe has an abrupt expansion from diameter $D1$ = 15 cm to $D2$ = 20 cm. The pressure and the average water velocity in the smaller pipe are $P1$ = 120 kPa and 10 m/s, respectively, and the flow is turbulent. By applying the continuity, momentum, and energy equations and disregarding the effects of the kinetic energy and momentum- flux correction factors, show that the loss coefficient for sudden expansion is $KL = (1 - D_1^2/D_2^2)2$, and calculate KL and $P2$for the given case.

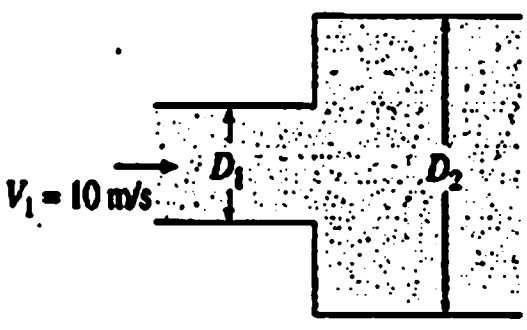


FIGURE P14–106

Step 1 of 9

Given data:

The diameter of the pipe 1, $D_1 = 15$ cm

The diameter of the pipe 2, $D_2 = 20$ cm

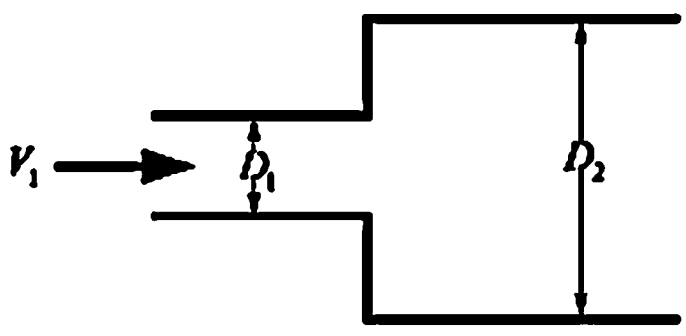
The pressure in pipe 1, $P_1 = 120$ kPa

The average velocity in pipe 1, $V_1 = 10$ m/s

The flow is turbulent.

Step 2 of 9

Given diagram:



Step 3 of 9

From the continuity equation,

$$\rho V_1 A_1 = \rho V_2 A_2$$

$$V_2 = V_1 \frac{A_1}{A_2}$$

The momentum equation for the control volume is given by

$$\sum \vec{F} = \sum_{\text{out}} \beta \dot{m} V - \sum_{\text{in}} \beta \dot{m} V$$

$$P_1 A_1 - P_2 A_2 = \dot{m}(V_2 - V_1)$$

$$P_1 A_1 - P_2 A_2 = (\rho A_2 V_2)(V_2 - V_1)$$

$$P_1 A_1 - P_2 A_2 = \left(\rho A_2 V_1 \frac{A_1}{A_2}\right)\left(V_1 \frac{A_1}{A_2} - V_1\right)$$

$$P_1 A_2 - P_2 A_2 = \rho A_2 V_1^2 \frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right)$$

On simplifying and rearranging,

$$\frac{P_1 - P_2}{\rho} = V_1^2 \frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right) \quad \text{..... (1)}$$

Step 4 of 9

The energy equation is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},s} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L$$

Here, both are placed at same elevation. So, $z_1 = z_2$.

Here, there is no turbine and pump. So, $h_{\text{pump},s} = h_{\text{turbine},e} = 0$.

The head loss,

$$h_L = K_L \frac{V_1^2}{2g}$$

Step 5 of 9

Then, the energy equation becomes

$$\frac{P_1}{\rho g} + (1)\frac{V_1^2}{2g} + z_1 + (0) = \frac{P_2}{\rho g} + (1)\frac{V_2^2}{2g} + z_1 + (0) + K_L \frac{V_1^2}{2g}$$

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + K_L \frac{V_1^2}{2g}$$

$$\frac{P_1 - P_2}{\rho} + \frac{V_1^2 - V_2^2}{2} = K_L \frac{V_1^2}{2}$$

Substituting the equation (1) in above expression

$$V_1^2 \frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right) + \frac{V_1^2 - \left(V_1 \frac{A_1}{A_2}\right)^2}{2} = K_L \frac{V_1^2}{2}$$

$$2V_1^2 \frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right) + V_1^2 - \left(V_1 \frac{A_1}{A_2}\right)^2 = K_L V_1^2$$

Step 6 of 9

Simplifying and rearranging:

$$K_L = 2\frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right) + 1 - \left(\frac{A_1}{A_2}\right)^2$$

$$= 2\left(\frac{A_1}{A_2}\right)^2 - 2\frac{A_1}{A_2} + 1 - \left(\frac{A_1}{A_2}\right)^2$$

$$= \left(\frac{A_1}{A_2}\right)^2 - 2\frac{A_1}{A_2} + 1$$

$$= 2\left(\frac{A_1}{A_2}\right) - 2\frac{A_1}{A_2} + 1 - \left(\frac{A_1}{A_2}\right)$$ 4.106P pg 2

$$= \left(\frac{A_1}{A_2}\right)^2 - 2\frac{A_1}{A_2} + 1$$

$$= \left(\frac{\frac{\pi}{4}D_1^2}{\frac{\pi}{4}D_2^2}\right)^2 - 2\frac{\frac{\pi}{4}D_1^2}{\frac{\pi}{4}D_2^2} + 1$$

$$= \left(\frac{D_1^2}{D_2^2}\right)^2 - 2\frac{D_1^2}{D_2^2} + 1$$

$$= \left(1 - \frac{D_1^2}{D_2^2}\right)^2$$

Thus the expression for loss coefficient for sudden expansion is $\boxed{K_L = \left(1 - \frac{D_1^2}{D_2^2}\right)^2}$

Step 7 of 9

Substituting the known values:

$$K_L = \left(1 - \frac{0.15^2}{0.2^2}\right)^2$$

$$= 0.1914$$

Therefore, the coefficient for sudden expansion is $\boxed{K_L = 0.1914}$

Step 8 of 9

Now, the velocity at exit

$$V_2 = V_1\frac{A_1}{A_2}$$

$$= V_1\frac{\frac{\pi}{4}D_1^2}{\frac{\pi}{4}D_2^2}$$

$$= V_1\frac{D_1^2}{D_2^2}$$

$$= 10\times\frac{0.15^2}{0.2^2}$$

$$= 5.625 \text{ m/s}$$

Step 9 of 9

Substituting the known values in equation (1)

$$\frac{P_1 - P_2}{\rho} = V_1^2\frac{A_1}{A_2}\left(\frac{A_1}{A_2} - 1\right)$$

$$= V_1^2\frac{D_1^2}{D_2^2}\left(\frac{D_1^2}{D_2^2} - 1\right)$$

$$\frac{\left(120\times10^3\right) - P_2}{1000} = (10)^2\times\frac{0.15^2}{0.2^2}\left(\frac{0.15^2}{0.2^2} - 1\right)$$

$$P_2 = 144.61\times10^3 \text{ Pa}$$

Therefore, the pressure, $\boxed{P_2 = 144.61\times10^3 \text{ Pa}}$

Question:

The water at 20°C in a 10-m-diameter. 2-m-high aboveground swimming pool is to be emptied by unplugging a 5-cm-diameter, 25-m-long horizontal plastic pipe attached to the bottom of the pool. Determine the initial rate of discharge of water through the pipe and the time (hours) it would take to empty the swimming pool completely assuming the entrance to the pipe is well-rounded with negligible loss. Take the friction factor of the pipe to be 0.022. Using the initial discharge velocity, check if this is a reasonable value for the friction factor.

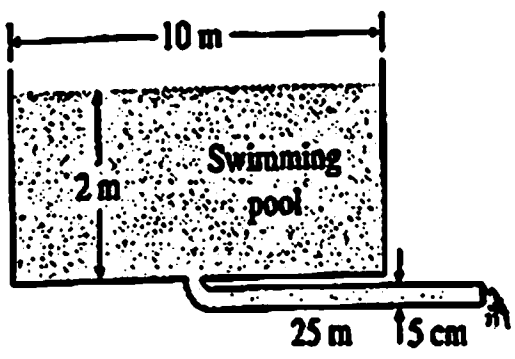


FIGURE P14–107

Step 1 of 12

Given data:

Temperature of the water $T = 20°C$

The diameter of the swimming pool, $D_1 = 10$ m

The height of the swimming pool, $H = 2$ m

The diameter of the pipe, $D = 5$cm

The length of the pipe, $L = 25$ m

The friction factor, $f = 0.022$

The pipe is made of plastic.

Step 2 of 12

From the table of properties of saturated water, corresponding to 20°C,

The density of water, $\rho = 998\ \text{kg/m}^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$

The roughness of plastic, $\varepsilon = 0$

Step 3 of 12

Let point 1 is at the entrance of the pipe

point 2 is at the exit of the pipe.

The velocity at point 1 is low and the velocity at point 2 is the discharge velocity.

Let the point 1 be the reference point.

The energy equation for a control volume between two points is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},e} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L \qquad \text{..... (1)}$$

Here, there is no pump or turbine in the pipe system. So, $h_{\text{pump},e} = h_{\text{turbine},e} = 0$.

The effect of kinetic energy correction factor is assumed to be negligible.

So, $\alpha_1 = \alpha_2 = 1$

The pressures, $P_1 = P_2 = P_{\text{atm}}$

Step 4 of 12

The elevations, when time $t = 0$

$z_1 = 2$ m

$z_2 = 0$

The velocities at time $t = 0$,

$$V_1 \approx 0$$

$$V_{2,t=0} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\frac{\pi}{4}D^2} = \frac{\dot{V}}{\frac{\pi}{4}(0.05\ \text{m})^2} = 509.3\dot{V} \qquad \text{..... (2)}$$

Step 5 of 12

The head loss,

$$h_L = f\frac{L}{D}\frac{V_{2,t=0}^2}{2g} = 0.022\left(\frac{25\ \text{m}}{0.05\ \text{m}}\right)\frac{(509.3\dot{V})^2}{2(9.81\ \text{m/s}^2)} = 145.426 \times 10^3\ \dot{V}^2$$

Step 6 of 12

Substitute the known values in equation (1) for time $t = 0$.

$$\frac{P_{\text{atm}}}{\rho g} + 0 + z_1 + 0 = \frac{P_{\text{atm}}}{\rho g} + (1)\frac{V_{2,t=0}^2}{2g} + 0 + 0 + h_L$$

$$z_1 = \frac{V_{2,t=0}^2}{2g} + h_L$$

$$2\ \text{m} = \frac{(509.3\dot{V})^2}{2(9.81\ \text{m/s}^2)} + 145.426 \times 10^3\ \dot{V}^2$$

$$= 13.22 \times 10^3\ \dot{V}^2 + 145.426 \times 10^3\ \dot{V}^2$$

$$\dot{V} = \sqrt{\frac{2}{158.646 \times 10^3}}$$

$$= 3.55 \times 10^{-3}\ \text{m}^3/\text{s}$$

$$= 3.55\ \text{L/s}$$

Therefore, the initial rate of discharge, $\boxed{\dot{V} = 3.55\ \text{L/s}}$

$= \sqrt{158.646 \times 10^{-6}}$

$= 3.55 \times 10^{-3}\ \text{m}^3/\text{s}$

$= 3.55\ \text{L/s}$

Therefore, the initial rate of discharge, $\boxed{\dot{V} = 3.55\ \text{L/s}}$

Step 7 of 12

The elevation at any given time,

$$z = \frac{V_2^2}{2g} + h_L$$

$$= \frac{V_2^2}{2g} + f\frac{L}{D}\frac{V_2^2}{2g}$$

$$= \frac{V_2^2}{2g}\left(1 + f\frac{L}{D}\right)$$

$$V_2 = \sqrt{\frac{2gz}{1 + f\frac{L}{D}}}$$

The rate of flow at any given time,

$$\dot{V} = V_2 A_t$$

$$= \sqrt{\frac{2gz}{1 + f\frac{L}{D}}}\left(\frac{\pi}{4}D^2\right)$$

$$= \frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D}}}$$

Step 8 of 12

Then, the amount of water that flows through the pipe during a differential time interval dt,

$$dV = \dot{V}dt$$

$$= \frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D}}}\cdot dt \qquad \text{..... (3)}$$

The decrease in the volume of water in the pool,

$$dV = A_{t,\text{tank}}(-dz)$$

$$= \frac{\pi}{4}D_1^2(-dz) \qquad \text{..... (4)}$$

The height $(-dz)$ is negative because the volume is decreasing.

Step 9 of 12

Now, from equations (3) and (4),

$$\frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D}}}\cdot dt = \frac{\pi}{4}D_1^2(-dz)$$

$$dt = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left(z^{-\frac{1}{2}}\right)\cdot dz$$

Step 10 of 12

On integrating,

$$\int_0^t dt = \int_{z_1}^0 -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left(z^{-\frac{1}{2}}\right)\cdot dz$$

$$[t]_0^t = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left[\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right]_{z_1}^0$$

$$t - 0 = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left[\frac{0^{\frac{1}{2}}}{\frac{1}{2}} - \frac{z_1^{\frac{1}{2}}}{\frac{1}{2}}\right]$$

$$= -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left(-2z_1^{\frac{1}{2}}\right)$$

$$t = 2\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D}}{2g}}\left(z_1^{\frac{1}{2}}\right)$$

Step 11 of 12

Now substituting the known values:

$$t = 2\frac{(10\ \text{m})^2}{(0.05\ \text{m})^2}\sqrt{\frac{1 + 0.022\frac{25\ \text{m}}{0.05\ \text{m}}}{2(9.81\ \text{m/s}^2)}}\left((2\ \text{m})^{1/2}\right)$$

$$= 88480.20\ \text{s}$$

$$= 24.57\ \text{h}$$

Therefore, the time for emptying the tank is $\boxed{t = 24.57\ \text{h}}$

Step 12 of 12

The Reynolds number,

$$\text{Re} = \frac{\rho V_{2,t=0} D}{\mu}$$

$$= \frac{(998\ \text{kg/m}^3)(509.3 \times 3.55 \times 10^{-3}\ \text{m/s})(0.05\ \text{m})}{1.002 \times 10^{-3}\ \text{kg/m}\cdot\text{s}}$$

$$= 90.04 \times 10^3$$

For turbulent flow, the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

$$= \frac{(\quad)(\quad)(\quad)}{1.002 \times 10^{-3}\ \text{kg/m}\cdot\text{s}} \quad \text{14 107P pg 3}$$

$$= 90.04 \times 10^3$$

For turbulent flow, the Colebrook equation,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

where, f is the friction factor. Now by substituting the known values:

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{0/0.05\ \text{m}}{3.7} + \frac{2.51}{90.04 \times 10^3 \sqrt{f}}\right)$$

$$= -2.0 \log\left(0 + \frac{2.787 \times 10^{-5}}{\sqrt{f}}\right)$$

$$= -2.0 \log\left(\frac{2.787 \times 10^{-5}}{\sqrt{f}}\right)$$

By solving the above equation, the friction factor, $f = 0.0184$

So, the given value of friction factor is not an accurate one.

Question:

Reconsider Prob. 14-107. Using EES (or other) software, investigate the effect of the discharge pipe diameter on the time required to empty the pool completely. Let the diameter vary from 1 to 10 cm, in increments of 1 cm. Tabulate and plot the results.

Step 1 of 6

Given data:

Temperature of the water $T = 20°C$

The diameter of the swimming pool, $D_1 = 10$ m

The height of the swimming pool, $H = 2$ m

The diameter of the pipe, $D = 5$cm

The length of the pipe, $L = 25$ m

The friction factor, $f = 0.022$

Step 2 of 6

From the table of properties of saturated water, corresponding to 20°C,

The density of water, $\rho = 998\ kg/m^3$

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ kg/m \cdot s$

Acceleration due to gravity is considered as, $g = 9.81 m/s$

The roughness of plastic, $\varepsilon = 0$

Step 3 of 6

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

```
"Given data"
D_1=10 [m]
H=2 [m]
"D=0.01 [m]"
L=25 [m]
f=0.022
rho=998 [kg/m^3]
mu=0.001002 [kg/m-s]
g=9.81 [m/s]

"Energy equation for a control volume between two points"

(P_1/(rho*g))+(alpha_1*(V_1^2)/2*g)+z_1+h_pump=(P_2/(rho*g))+(alpha_2*(V_2^2)/2*g)+z_2+h_turbine+h_l

P_1=P_2                    "Since both are open to atmosphere"

h_pump=0                   "There is no pump or turbine in the pipe system."
h_turbine=0

alpha_1=1                  "The effect of kinetic energy correction factor is assumed to be negligible"
alpha_2=1

V_1=0                      "Water is considered to be stationary in side the pool"

z_1=H                      "Free surface of pool"
z_2=0                      "This level is considered as datum"

h_l=f*(L*V_2^2)/(D*2*g)    "Head loss"

"Discharge of the pipe"

V_2=Vdot/A_c

A_c=pi*D^2/4

V_2=(2*g*H)/((1+f*L/D))^0.5

"Time required to empty the pool completely"

Time=(D_1/D)^2*(2*H/(1+f*L/D)/g)^0.5*convert(s,h)
```

Step 4 of 6

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below ,

Given data

$D_1 = 10$ [m]

$H = 2$ [m]

$D = 0.01$ [m]

$L = 25$ [m]

$f = 0.022$

$\rho = 998$ [kg/m^3]

$\mu = 0.001002$ [kg/m-s]

$g = 9.81$ [m/s]

Energy equation for a control volume between two points

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{pump} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{turbine} + h_l$$

$P_1 = P_2$ Given both are open to atmosphere

$h_{pump} = 0$ There is no pump or turbine in the pipe system

$h_{turbine} = 0$

$\alpha_1 = 1$ The effect of kinetic energy correction factor is assumed to be negligible

$\alpha_2 = 1$

$V_1 = 0$ Water is considered to be stationary in side the pool

$z_1 = H$ Free surface of pool

$z_2 = 0$ This level is considered as datum

$h_l = f \cdot \frac{L \cdot V_2^2}{D \cdot 2 \cdot g}$ Head loss

Discharge of the pipe

$$V_2 = \frac{Vdot}{A_c}$$

$$A_c = \pi \cdot \frac{D^2}{4}$$

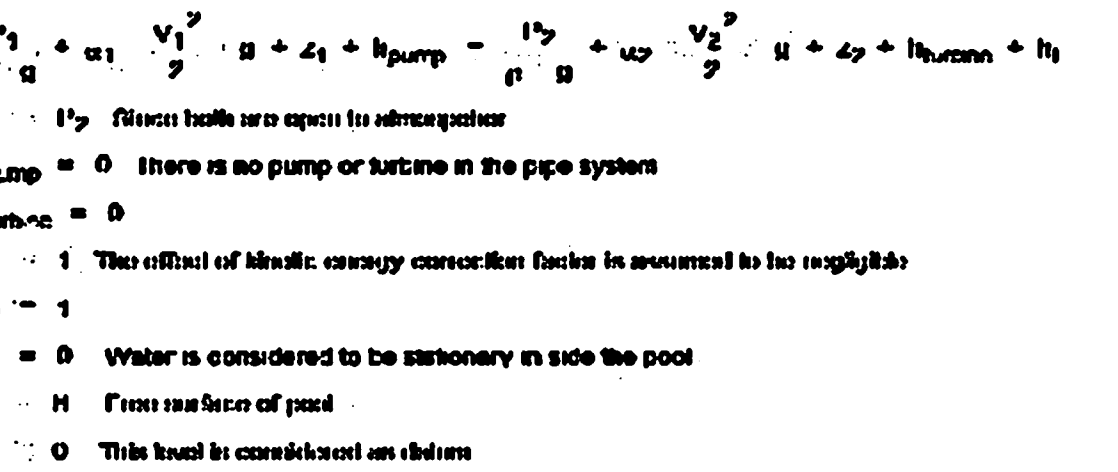
$$V_2 = \left[2 \cdot g \cdot \left[\frac{H}{1 + f \cdot \frac{L}{D}}\right]\right]^{0.5}$$

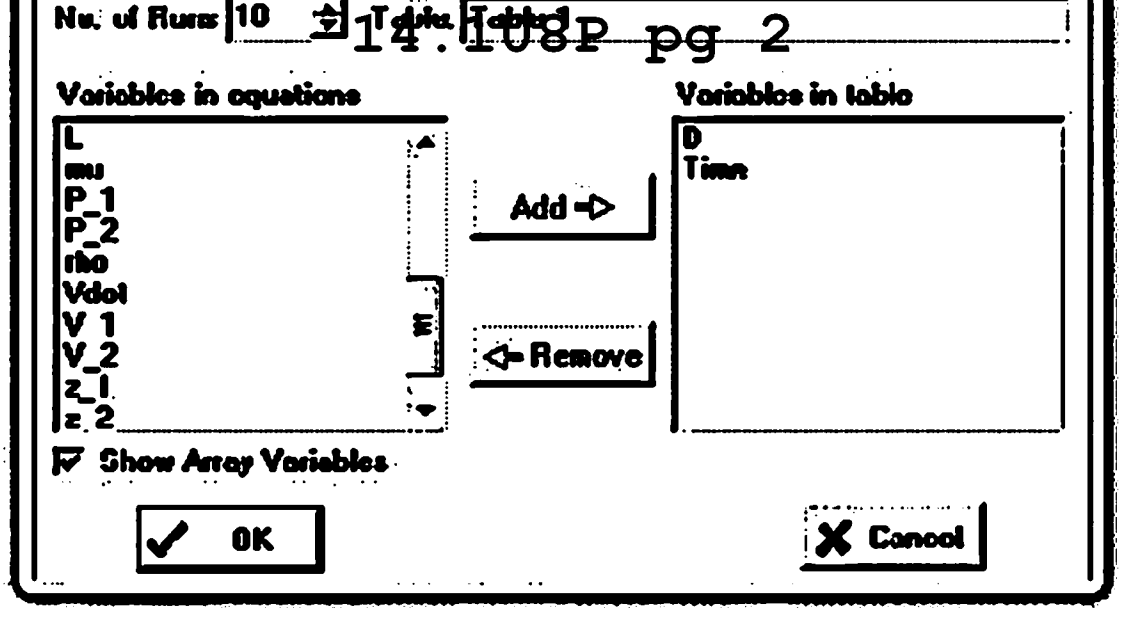
Time required to empty the pool completely

$$time = \left[\frac{D_1}{D}\right]^2 \cdot \left[2 \cdot H \cdot \left[\frac{1 + f \cdot \frac{L}{D}}{g}\right]\right]^{0.5} \cdot \left[0.000277778 \cdot \frac{hr}{s}\right]$$

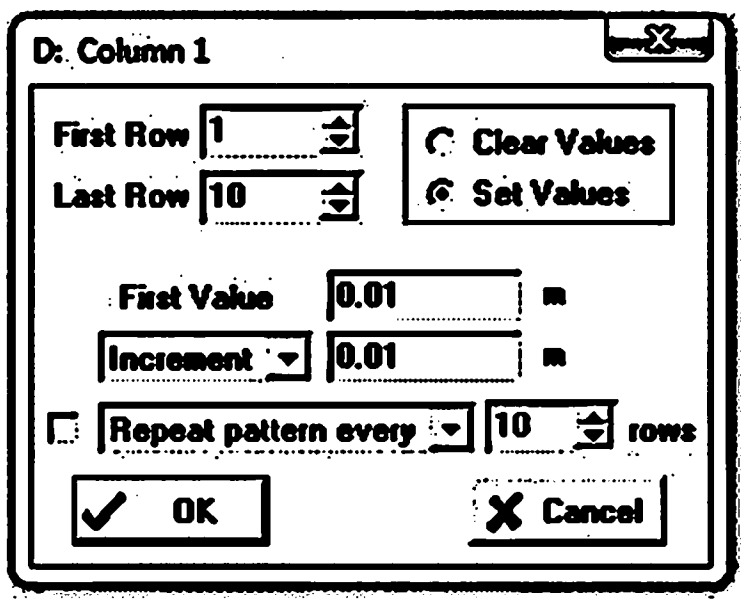
Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,

Step 5 of 6





To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



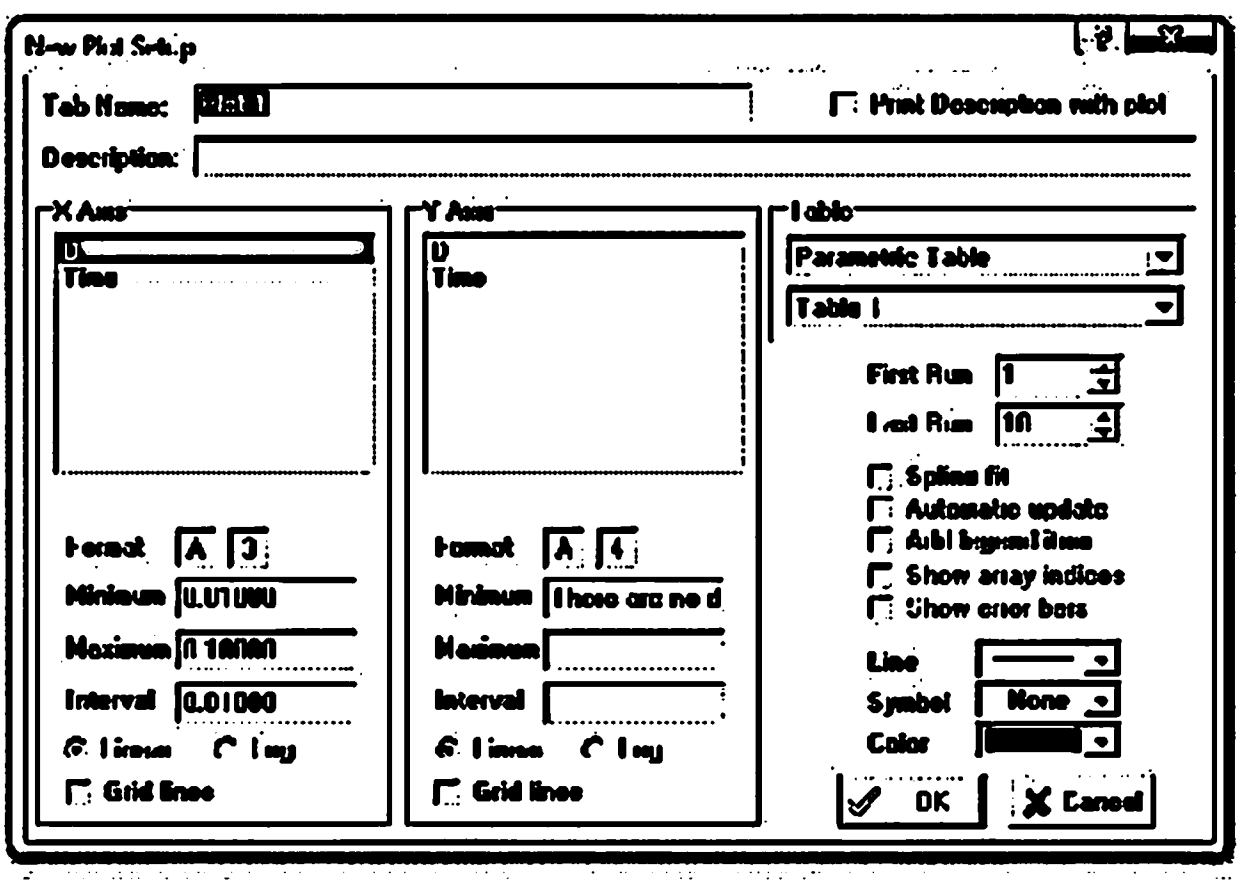
Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

Table 1

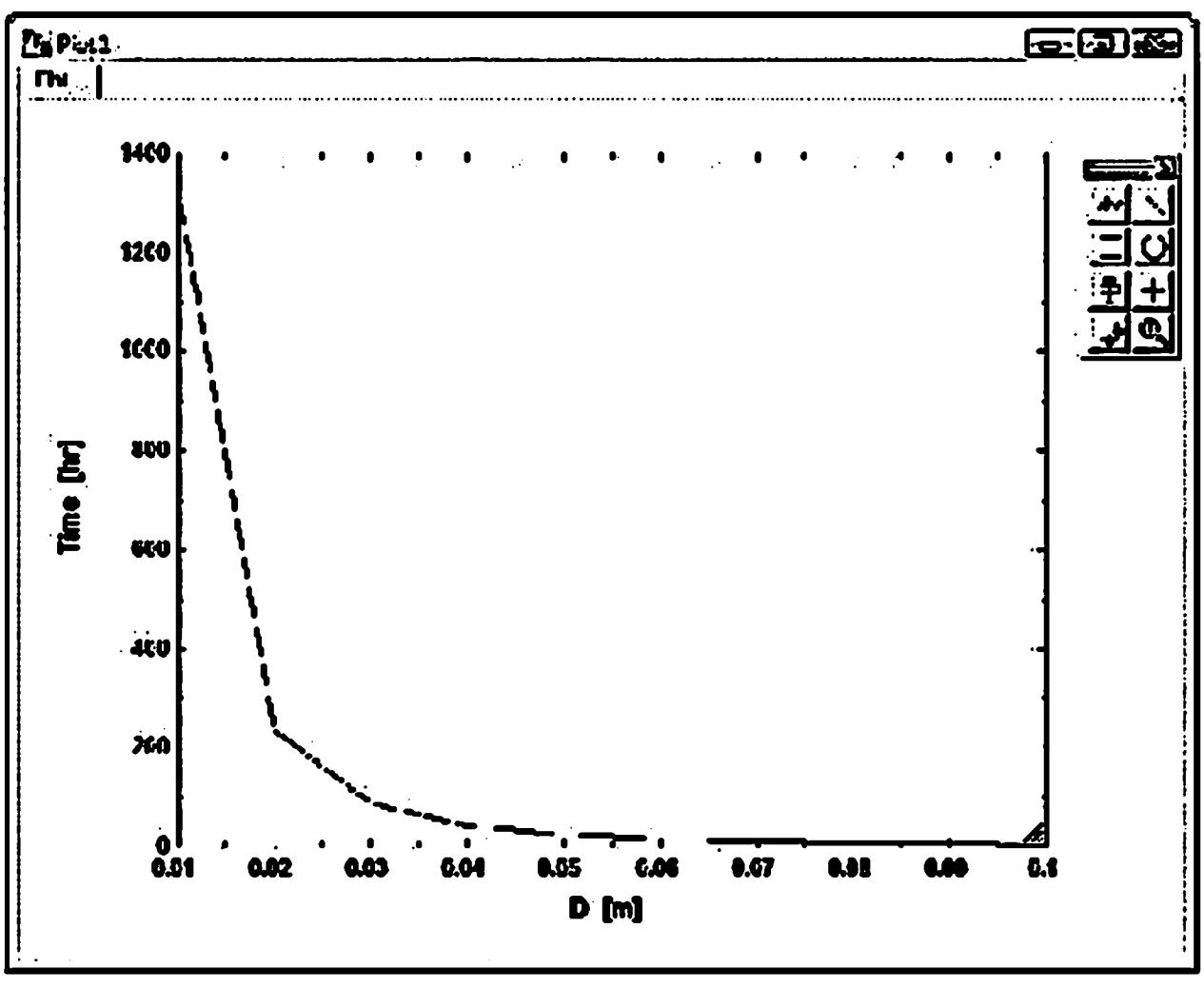
1..10	D [m]	Time [hr]
Run 1	0.01	1327
Run 2	0.02	236.7
Run 3	0.03	86.66
Run 4	0.04	42.58
Run 5	0.05	24.58
Run 6	0.06	15.71
Run 7	0.07	10.77
Run 8	0.08	7.777
Run 9	0.09	5.84
Run 10	0.1	4.522

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 6 of 6

By clicking on OK the graph will be generated as shown,



14.1081 pg 3

Question: 14.109P pg 1

Repeat Prob. 14-107 for a sharp-edged entrance to the pipe with $KL = 0.5$. Is this "minor loss" truly "minor" or not?

Step 1 of 12

Given data:

Temperature of the water, $T = 20°C$

The diameter of the swimming pool, $D_1 = 10$ m

The height of the swimming pool, $H = 2$ m

The diameter of the pipe, $D = 5$ cm $= 0.05$ m

The length of the pipe, $L = 25$ m

The pipe is having sharp edged entrance

The loss coefficient of sharp edged entrance, $K_{L,\text{entrance}} = 0.5$

The pipe is made of plastic.

The friction factor, $f = 0.022$

Step 2 of 12

The flow is assumed to be turbulent, fully developed and incompressible. The elevation deference between the pipe and the fountain is negligible.

From the table of properties of saturated water, corresponding to $20°C$,

The density of water, $\rho = 998\ \text{kg/m}^3$.

The dynamic viscosity, $\mu = 1.002 \times 10^{-3}\ \text{kg/m·s}$

The roughness of plastic, $\varepsilon = 0$

Step 3 of 12

Let point 1 be at the entrance of the pipe and point 2 be at the exit of the pipe. The velocity at point 1 is low and the velocity at point 2 is the discharge velocity.
Let the point 1 be the reference point.

The energy equation for a control volume between two points is given by

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump},u} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine},e} + h_L \qquad \text{..... (1)}$$

Here, there is no pump or turbine in the pipe system. So, $h_{\text{pump},u} = h_{\text{turbine},e} = 0$.

The pressures, $P_1 = P_2 = P_{\text{atm}}$

Step 4 of 12

The elevations, when time $t = 0$

$z_1 = 2$ m

$z_2 = 0$

The velocities at time $t = 0$,

$$V_1 \approx 0$$

$$V_{2,t=0} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\frac{\pi}{4}D^2} = \frac{\dot{V}}{\frac{\pi}{4}(0.05\ \text{m})^2} = 509.3\dot{V} \qquad \text{..... (2)}$$

Step 5 of 12

The head loss,

$$h_L = \left(f\frac{L}{D} + K_L\right)\frac{V_{2,t=0}^2}{2g} = \left[0.022\left(\frac{25\ \text{m}}{0.05\ \text{m}}\right) + 0.5\right]\frac{(509.3\dot{V})^2}{2(9.81\ \text{m/s}^2)} = 152.036 \times 10^3\,\dot{V}^2$$

Step 6 of 12

Substituting the known values in equation (1).for time $t = 0$:

$$\frac{P_{\text{atm}}}{\rho g} + 0 + z_1 + 0 = \frac{P_{\text{atm}}}{\rho g} + (1)\frac{V_{2,t=0}^2}{2g} + 0 + 0 + h_L$$

$$z_1 = \frac{V_{2,t=0}^2}{2g} + h_L$$

$$2\ \text{m} = \frac{(509.3\dot{V})^2}{2(9.81\ \text{m/s}^2)} + 152.036 \times 10^3\,\dot{V}^2 = 13.22 \times 10^3\,\dot{V}^2 + 152.036 \times 10^3\,\dot{V}^2$$

$$\dot{V} = \sqrt{\frac{2}{162.256 \times 10^3}} = 3.51 \times 10^{-3}\ \text{m}^3/\text{s} = 3.51\ \text{L/s}$$

Therefore, the initial rate of discharge is $\boxed{\dot{V} = 3.51\ \text{L/s}}$

Step 7 of 12

The elevation at any given time,

$$z = \frac{V_2^2}{2g} + h_L = \frac{V_2^2}{2g} + \left(f\frac{L}{D} + K_L\right)\frac{V_2^2}{2g} = \frac{V_2^2}{2g}\left(1 + f\frac{L}{D} + K_L\right)$$

$$= \sqrt{\frac{2}{\left(1 + f\frac{L}{D} + K_L\right)}}$$

$$= \frac{V_2^2}{2g} + \left(f\frac{L}{D} + K_L\right)\frac{V_2^2}{2g}$$

$$= \frac{V_2^2}{2g}\left(1 + f\frac{L}{D} + K_L\right)$$

$$V_2 = \sqrt{\frac{2gz}{1 + f\frac{L}{D} + K_L}}$$

Step 8 of 12

The rate of flow at any given time,

$$\dot{V} = V_2 A_c$$

$$= \sqrt{\frac{2gz}{1 + f\frac{L}{D} + K_L}}\left(\frac{\pi}{4}D^2\right)$$

$$= \frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D} + K_L}}$$

Step 9 of 12

Then, the amount of water that flows through the pipe during a differential time interval dt,

$$dV = \dot{V}dt$$

$$= \frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D} + K_L}}\cdot dt \qquad \text{..... (3)}$$

The decrease in the volume of water in the pool,

$$dV = A_{c,\text{tank}}(-dz)$$

$$= \frac{\pi}{4}D_1^2(-dz) \qquad \text{..... (4)}$$

The height $(-dz)$ is negative because the volume is decreasing.

Step 10 of 12

Now, from the equations (3) and (4),

$$\frac{\pi}{4}D^2\sqrt{\frac{2gz}{1 + f\frac{L}{D} + K_L}}\cdot dt = \frac{\pi}{4}D_1^2(-dz)$$

$$dt = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left(z^{-\frac{1}{2}}\right)\cdot dz$$

On integrating,

$$\int_0^t dt = \int_{z_1}^0 -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left(z^{-\frac{1}{2}}\right)\cdot dz$$

$$[t]_0^t = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left[\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right]_{z_1}^0$$

$$t - 0 = -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left[\frac{0^{\frac{1}{2}}}{\frac{1}{2}} - \frac{z_1^{\frac{1}{2}}}{\frac{1}{2}}\right]$$

$$= -\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left(-2z_1^{\frac{1}{2}}\right)$$

$$t = 2\frac{D_1^2}{D^2}\sqrt{\frac{1 + f\frac{L}{D} + K_L}{2g}}\left(z_1^{\frac{1}{2}}\right)$$

Step 11 of 12

Now substitute the known values.

$$t = 2\frac{(10\text{ m})^2}{(0.05\text{ m})^2}\sqrt{\frac{1 + 0.022\frac{25\text{ m}}{0.05\text{ m}} + 0.5}{2(9.81\text{ m/s}^2)}}\left((2\text{ m})^{1/2}\right)$$

$$= 90304.7\text{ s}$$

$$= 25.08\text{ hr}$$

Therefore, the time for emptying the tank is $\boxed{t = 25.08\text{ hr}}$

The Reynolds number,

$$\text{Re} = \frac{\rho V_{2,t-0}D}{\mu}$$

$$= \frac{(998\text{ kg/m}^3)(509.3 \times 3.51 \times 10^{-3}\text{ m/s})(0.05\text{ m})}{1.002 \times 10^{-3}\text{ kg/m}\cdot\text{s}}$$

$$= 89.02 \times 10^3$$

For turbulent flow, the Colebrook equation is $\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$, where, f is the friction factor.

Step 12 of 12

Substitute the known values.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{0/0.05\text{ m}}{3.7} + \frac{2.51}{89.02 \times 10^3\sqrt{f}}\right)$$

$$= -2.0\log\left(0 + \frac{2.82 \times 10^{-5}}{\sqrt{f}}\right)$$

$$= -2.0\log\left(\frac{2.82 \times 10^{-5}}{\sqrt{f}}\right)$$

$$= -2.0 \log\left(0 + \frac{2.82 \times 10^{-5}}{\sqrt{f}}\right)$$

14.109P pg 3

$$= -2.0 \log\left(\frac{2.82 \times 10^{-5}}{\sqrt{f}}\right)$$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{2.82 \times 10^{-5}}{\sqrt{f}}\right)$$

By solving,

$f = 0.0041763$

So, the given value of friction factor is not an accurate one.

CHAPTER 15
SOLUTIONS

Question: 15.001P

Which car is more likely to be more fuel-efficient: one with sharp corners or one that is contoured to resemble an ellipse? Why?

Step 1 of 1

The car that is contoured to resemble an ellipse has more fuel efficient than the car with sharp corners, because the car that is contoured to resemble an ellipse has smaller drag coefficient and thus has smaller air resistance.

The sharp corners promote a fixed flow separation location, whereas roundness corners can lead to unsteadiness in the flow. Therefore, rounding is usually, but not always the best solution.

Question: 15.002P

Consider laminar flow over a flat plate. How does the local friction coefficient change with position?

Step 1 of 1

In a laminar flow the local friction coefficient varies with position. The local friction coefficient variations are based on the changes in the velocity boundary layer along the surface of the plate in the flow direction.

The local friction coefficient at a location x can be determined by the formula

$$C_{f,x} = \frac{0.664}{\mathrm{Re}_x^{0.5}}$$

Where, $C_{f,x}$ is the local friction coefficient at a location x.

x is the distance from the leading edge of the plate.

Re_x is the Reynolds number at the location x.

Here, Reynolds number,

$$\mathrm{Re}_x = \frac{Vx}{\nu}$$

Where, V is the upstream velocity.

x is the distance from the leading edge of the plate.

ν is the Kinematic viscosity.

Question: 15.003P

Explain when an external flow is two-dimensional, three-dimensional, and axisymmetric. What type of flow is the flow of air over a car?

Step 1 of 3

The flow over a body is classified as two-dimensional, three-dimensional, and axisymmetric based on the shape of the body through which the flow occurs.

The flow over a body is said to be two-dimensional when the body is very long and having a constant cross section and the flow is normal to the body.

Example:
Wind blowing over a long pipe perpendicular to its axis. The velocity component in axial direction is zero in this case therefore, the velocity is two-dimensional.

Step 2 of 3

When flow occurs over a body which possesses symmetry about an axis in the flow direction the flow can be considered as axisymmetric.
Example: A bullet piercing through air.

Step 3 of 3

Flow over a body that cannot be considered as two-dimensional flow or axisymmetric flow is called three-dimensional flow.
Example: Flow of air over a vehicle.

Flow of air over a car is a three-dimensional flow.

Question: 15.004P

What is the difference between the upstream velocity and the free-stream velocity? For what types of flow are these two velocities equal to each other?

Step 1 of 1

The velocity at which the fluid approaches a body is called upstream velocity. The velocity of the fluid relative to the body at which it approaches the body is called free-stream velocity. If the flow is uniform and the body is small relative to the rate of flow then upstream velocity and free stream velocity will be equal.

Question: 15.005P

What is the difference between streamlined and bluff bodies? Is a tennis ball a streamlined or bluff body?

Step 1 of 1

All the bodies subjected to fluid flow can be classified as streamlined or bluff bodies.
A body is said to be streamlined if the alignment of the body is altered to reduce the obstruction to the fluid flow. It is very easy to force a streamlined body through a fluid. Now a day's most of the vehicles ships and airplanes are streamlined to reduce the resistance caused by the fluid.

If the alignment of the body is not altered to reduce the obstruction to the fluid flow such a body is called a bluff body.
A tennis ball is a bluff body.

Question: 15.006P

Name some applications in which a large drag is desired.

Step 1 of 1

Larger drag means larger the friction, in some cases larger drag produces beneficial effect and we try to maximize it. In case of the brakes of automobiles large friction will be an advantage, larger the friction smaller will be the stopping distance. It is due to the drag that we are able to parachute, larger drag will decrease the rate of descent of the parachute, so for parachutes we try to maximize the drag.

Question: 15.007P

What is drag? What causes it? Why do we usually try to minimize it?

Step 1 of 3

While the moving of fluid over surface of a body it will cause two types of forces,

(1) Pressure force which is normal to the surface of the body and

(2) Shear force which is parallel to the surface of the body.

Consider only the resultant of these two forces. The resultant will be having two components one is perpendicular component and the other one is the parallel component.

Step 2 of 3

The component of the resultant force which acts in the direction of the flow is called the drag force or simply drag.

The component which acts perpendicular to the flow direction is called lift force.

In general a force which is exerted by the flowing body on the surface of the body in the flow direction is called drag.

Step 3 of 3

Drag will produce frictional effect on the surfaces of the body this will create undesirable effects. Reduction in drag will reduce the fuel consumption in automobiles, ships, submarines and aircrafts. It will also increase the safety and durability of structures subjected to high winds; it will also reduce the noise and vibration.

Question: 15.008P

What is lift? What causes it? Does wall shear contribute to the lift?

Step 1 of 1

The force which causes the body to move in the direction normal to the direction of fluid flow is called lift force and the motion in that direction is called lift. When a fluid moves over a body it exerts pressure force normal to the body and shear force parallel to the body surface. Both these forces have components in the direction of flow and normal to the direction of flow. The resultant components of both pressure force and shear forces in the direction normal to the fluid flow will push the body normal to the fluid flow. Therefore the sum of resultant components of pressure force and shear force will contribute to the lift. **Therefore, wall shear will contribute to the lift.**

Question: 15.009P

During flow over a given body, the drag force, the upstream velocity, and the fluid density are measured. Explain how you would determine the drag coefficient. What area would you use in the calculations?

Step 1 of 3

During flow over a given body if the drag force, upstream velocity, and the fluid density are measured the drag coefficient can be measured using the formula given below.

$$C_D = \frac{F_D}{\frac{1}{2}\rho V^2 A}$$

Where, F_D is the drag force.

ρ is the density of fluid.

V is the velocity.

A is the frontal area.

Step 2 of 3

Here the term $\frac{1}{2}\rho V^2$ is the dynamic pressure.

Now let us have a look at how the above equation can be deducted.

The drag force F_D is directly proportional to the product of dynamic pressure and frontal area.

$$F_D \propto \frac{1}{2}\rho V^2 A$$

In order to equalize RHS and LHS multiply RHS with a constant, this constant is called drag coefficient C_D

Therefore drag force $F_D = C_D \frac{1}{2}\rho V^2 A$.

Step 3 of 3

Here the area used is the frontal area of the body, which is the area projected on a plane normal to the direction of flow. It is the area seen by a person looking at the body from the direction of the approaching fluid. But for thin bodies the area A is the planform area. Planform area is the area seen by a person looking at the body from above in a direction normal to the body. Drag force F_D is the force exerted by the flowing fluid on a body in the flow direction.

Question: 15.010P

Define the frontal area of a body subjected to external flow. When is it appropriate to use the frontal area in drag and lift calculations?

Step 1 of 1

When we look at a body from the direction of fluid flow the area we see is called the frontal area. The frontal area can be used for the calculation of drag and lift if the bodies are blunt bodies like vehicles, airplanes, cylinders etc.

Question: 15.011P

Define the planform area of a body subjected to external flow. When is it appropriate to use the planform area in drag and lift calculations?

Step 1 of 1

When we look at a body from above in a direction normal to the flow, the area of the body we see is called the planform area. In drag and lift calculation of thin or slender bodies, planform area is appropriate to consider. Thus it is used for flat bodies, aero-foils and for other bodies where the frontal area is less.

Question: 15.012P

What is terminal velocity? How is it determined?

Step 1 of 1

When a body falls freely the main forces that act on the body are the drag force (the force a flowing fluid exerts on a body in the flow direction), the buoyancy force (the force which opposes the body's weight) and the weight of the body (acts downwards).

When the body is dropped from a height it first accelerates under the influence of its weight, as the body falls the motion of the body is resisted by the drag force acting opposite to the direction of the motion of the body. This drag force goes on increasing as the velocity of the body increases this continues until all the forces balance each other and the net force acting on the body is zero.

When the net force acting on the body becomes zero the velocity becomes constant and this constant velocity is the maximum velocity that the body can achieve and this velocity is called the terminal velocity.

Terminal velocity can be determined by setting the weight of the body equal to the sum of drag force and the buoyancy force.

Question: 15.013P

What is the difference between skin friction drag and pressure drag? Which is usually more significant for slender bodies such as airfoils?

Step 1 of 2

The force exerted by a fluid on a body in the flow direction is called drag. Drag force is the combined effect of wall shear and pressure forces. The part of drag that is caused by the wall shear stress is called skin friction drag because it is caused by the frictional effect. The part of drag that is caused by the pressure is called pressure drag.

Step 2 of 2

Skin friction drag or friction drag, $F_{D\text{friction}} = \frac{1}{2} C_{D\text{friction}} \rho A V^2$,

where, $C_{D\text{friction}}$ is the friction drag coefficient,

ρ is the density,

A is the frontal area and

V is the velocity.

The pressure drag, $F_{D\text{pressure}} = \frac{1}{2} C_{D\text{pressure}} \rho A V^2$,

where, $C_{D\text{pressure}}$ is the pressure drag coefficient,

ρ is the density,

A is the frontal area and

V is the velocity.

Question: 15.014P

What is the effect of surface roughness on the friction drag coefficient in laminar and turbulent flows?

Step 1 of 1

For laminar flow, the surface roughness doesn't influence the friction drag coefficient provided the roughness in not very large. But if the roughness is large, the friction drag coefficient also increases.

For turbulent flow, the increase in surface roughness increases the value of friction drag coefficient.

Question: 15.015P

In general, how does the drag coefficient vary with the Reynolds number at (*a*) low and moderate Reynolds numbers and (*b*) at high Reynolds numbers (Re > 104)?

Step 1 of 3

Drag coefficient depends on Reynolds number typically for Reynolds number below 10^4. The drag coefficient exhibit different behaviors in the low, moderate and high regions of Reynolds number.

Step 2 of 3

a)
At low and moderate Reynolds number the inertia effect is negligible therefore the drag coefficient is inversely proportional to the Reynolds number.

Step 3 of 3

b)
At higher Reynolds number the drag coefficient remains constant for almost all geometry, because at higher Reynolds number the flow becomes fully turbulent.

Question: 15.016P

What is the effect of streamlining on (*a*) friction drag and (*b*) pressure drag? Does the total drag acting on a body necessarily decrease as a result of streamlining? Explain.

Step 1 of 2

(a)

Streamlining has opposite effects on friction drag and pressure drag. It decreases pressure drag by delaying boundary layer separation and thereby reducing the pressure difference between the front and back of the body.

Step 2 of 2

(b)

Streamlining increases the friction drag by increasing the surface area. On streamlining the total drag may increase or decrease depending on the Reynolds number. Total drag will decrease if flow has high Reynolds number but will increase if the flow is with very low Reynolds number.

Question: 15.017P

What is flow separation? What causes it? What is the effect of flow separation on the drag coefficient?

Step 1 of 1

When a fluid flows over the surface of a body it will remain attached to the body surface. But with increase in flow velocity, the fluid tends to detach itself from the surface of the body. This tendency of fluid is known as Flow separation. The cause for flow separation is Velocity of the fluid. The location of separation point depends on several factors like Reynolds number, roughness of the surface, level of fluctuation in the free stream etc. Flow separation increases the drag coefficient to very high level.

Question: 15.018P

What is drafting? How does it affect the drag coefficient of the drafted body?

Step 1 of 1

Drag is the force which a flowing fluid exerts on a body in the flow direction. As the drag acts in opposite direction of motion, the body that moves in the fluid has low pressure region in its rear end. Hence by making the selected body to move behind another body, we can reduce the drag force acting on the selected body. This method of reducing drag is known as drafting. As drafting involves in making the body to move in low pressure region, the coefficient of drag of a drafted body is reduced to large extent.

Question: 15.019P

A car is moving at a constant velocity of 80 km/h. Determine the upstream velocity to be used in fluid flow analysis if (*a*) the air is calm, (*b*) wind is blowing against the direction of motion of the car at 30 km/h, and (*c*) wind is blowing in the same direction of motion of the car at 50 km/h.

Step 1 of 4

Given data:

The velocity of car, $V_{car} = 80\ \text{km/h}$

Step 2 of 4

a) Let us determine the upstream velocity when air is under calm condition.

Since, the air is calm the relative velocity will be equal to the velocity of car itself because here the velocity of air is zero.

Thus, the upstream velocity, $V = V_{car} = 80\ \text{km/h}$

Therefore, the upstream velocity, $\boxed{V = 80\ \text{km/h}}$

Step 3 of 4

b) Here we have to determine the upstream velocity when wind is blowing against the direction of motion of the car at 30 km/h .

Since wind is blowing against the direction of motion of the car, the relative velocity will be equal to the sum of velocities of car and wind.

Thus, the upstream velocity,

$$V = V_{car} + V_{wind}$$
$$= 80 + 30$$
$$= 110\ \text{km/h}$$

Therefore, upstream velocity, $\boxed{V = 110\ \text{km/h}}$

Step 4 of 4

c) Here we have to determine the upstream velocity when wind is blowing in the same direction of motion of the car at 50 Km/h .

Since wind is blowing in the same direction of motion of the car the relative velocity will be equal to the difference of velocities of car and wind.

Thus, the upstream velocity,

$$V = V_{car} - V_{wind}$$
$$= 80 - 50$$
$$= 30\ \text{km/h}$$

Therefore, the upstream velocity, $\boxed{V = 30\ \text{km/h}}$

Question: 15.020P

The resultant of the pressure and wall shear forces acting on a body is measured to be 580 N, making 35° with the direction of flow. Determine the drag and the lift forces acting on the body.

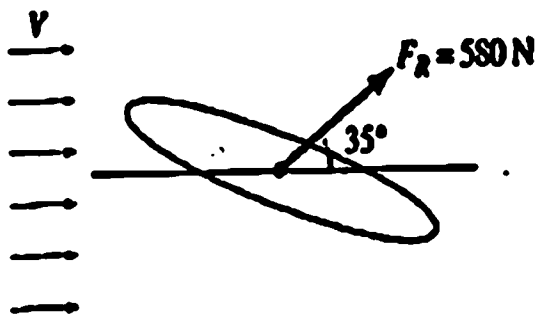


FIGURE P15–20

Step 1 of 3

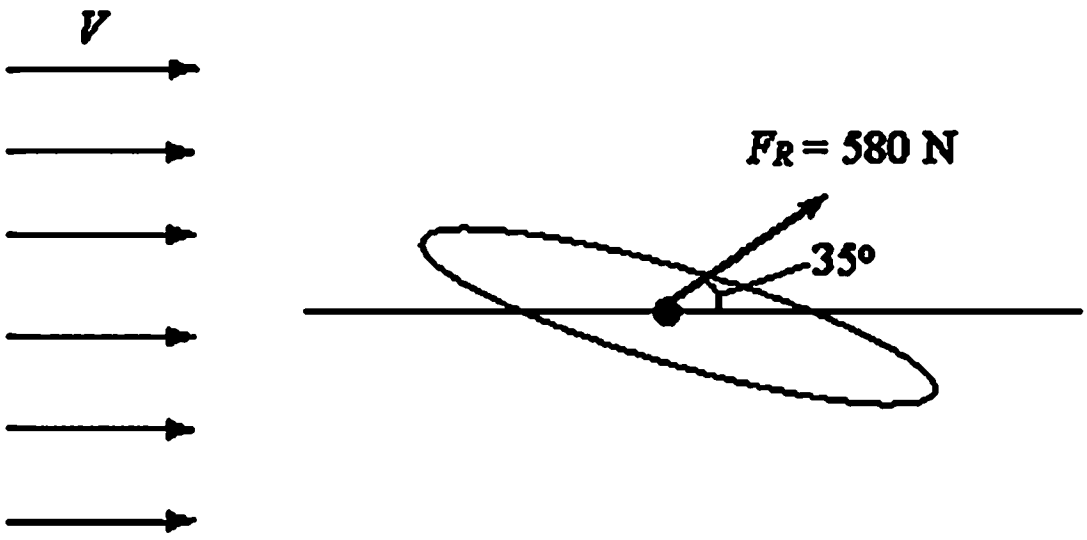
Given data:

The resultant of pressure and wall shear forces acting on the body, $F_R = 580\,\text{N}$

Angle made by the resultant with the horizontal, $\theta = 35°$

Step 2 of 3

Force acting on the body is shown in the diagram drawn below.



Step 3 of 3

From the above figure, Component of F_R which is in the direction of flow is the drag force and normal to the flow is the lift force.

Here component of F_R which is in the direction of flow is $F_R \cos 35°$

And the component of F_R normal to the direction of flow is $F_R \sin 35°$.

Therefore,

Drag force,

$$F_D = F_R \cos 35°$$
$$= 580 \cos 35°$$
$$= 475.11\ \text{N}$$

Lift force,

$$F_L = F_R \sin 35°$$
$$= 580 \sin 35°$$
$$= 332.67\ \text{N}$$

Therefore,

Drag and lift forces are, $\boxed{F_D = 475.11\ \text{N}}$ and $\boxed{F_L = 332.67\ \text{N}}$.

Question: 15.021P

During a high Reynolds number experiment, the total drag force acting on a spherical body of diameter D =12 cm subjected to airflow at 1 atm and 5°C is measured to be 5.2 N. The pressure drag acting on the body is calculated by integrating the pressure distribution (measured by the use of pressure sensors throughout the surface) to be 4.9 N. Determine the friction drag coefficient of the sphere.

Step 1 of 4

Given data:

The diameter of the spherical body, $D = 12$ cm

The total drag force acting on the body, $F_D = 5.2$ N

Pressure drag, $F_{D(P)} = 4.9$ N

Step 2 of 4

We have, total drag force,

$$F_D = F_{D(f)} + F_{D(P)}$$

$$F_{D(f)} = F_D - F_{D(P)}$$

$$= 5.2 - 4.9$$

$$= 0.3 \text{ N}$$

From definition of total drag force, we have

$$F_D = \frac{1}{2} C_D \rho V^2 A \qquad \ldots\ldots (1)$$

From definition of friction drag force, we have

$$F_{D(f)} = \frac{1}{2} C_{D(f)} \rho V^2 A \qquad \ldots\ldots (2)$$

Step 3 of 4

By dividing equation (1) with (2), we get

$$\frac{F_D}{F_{D(f)}} = \frac{\frac{1}{2} C_D \rho V^2 A}{\frac{1}{2} C_{D(f)} \rho V^2 A}$$

$$\frac{F_D}{F_{D(f)}} = \frac{C_D}{C_{D(f)}}$$

$$C_{D(f)} = C_D \frac{F_{D(f)}}{F_D} \qquad \ldots\ldots (3)$$

Step 4 of 4

Now, assume that the flow is turbulent. For turbulent flow, $C_D = 0.2$

By substituting the values in equation (3), we get

$$C_{D(f)} = 0.2 \times \frac{0.3}{5.2}$$

$$= 0.01154$$

Therefore, the frictional drag coefficient of the sphere is $\boxed{C_{D(f)} = 0.01154}$

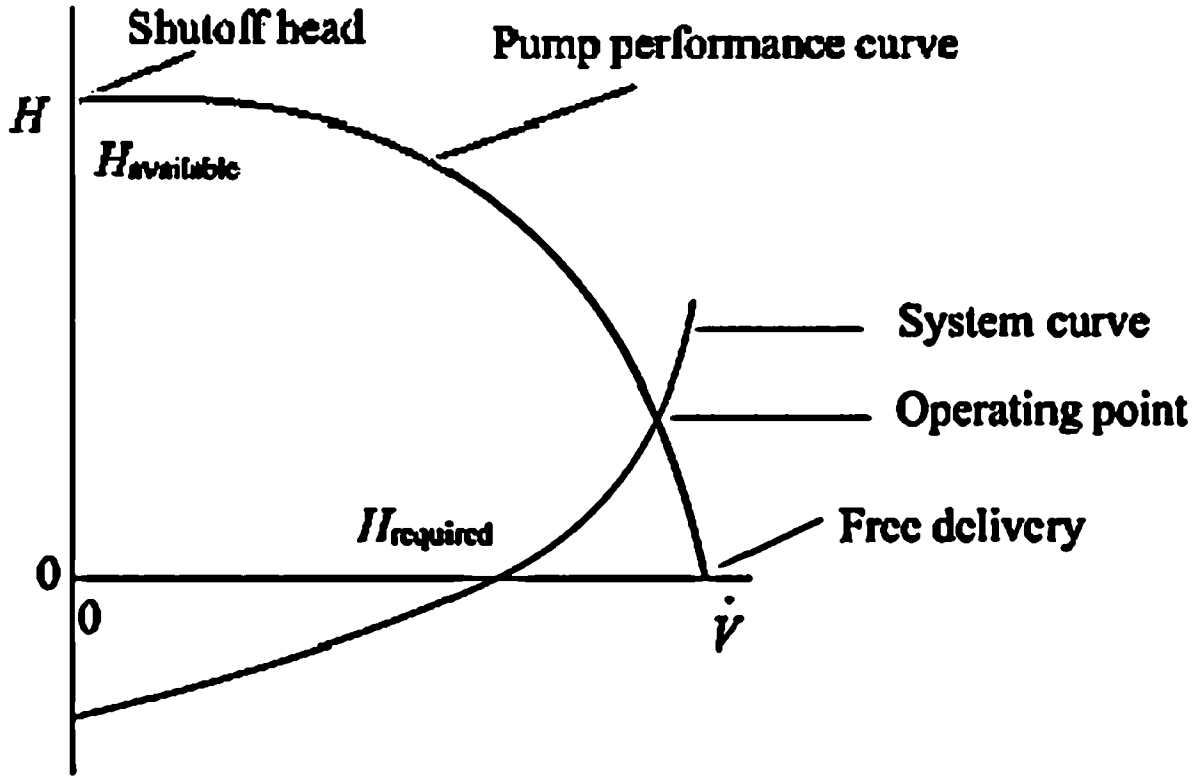
Question: 15.022P

The drag coefficient of a car at the design conditions of 1 atm, 25°C, and 90 km/h is to be determined experimentally in a large wind tunnel in a full-scale test. The height and width of the car are 1.40 m and 1.65 m, respectively. If the horizontal force acting on the car is measured to be 280 N, determine the total drag coefficient of this car.

Step 1 of 1

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In the given zero flow rate system the required head is negative as the elevation of the outlet is lower than that of the free surface of the inlet tank. The qualitative plot is generated.



Question: 15.023P

To reduce the drag coefficient and thus to improve the fuel efficiency, the frontal area of a car is to be reduced. Determine the amount of fuel and money saved per year as a result of reducing the frontal area from 18 to 15 ft2. Assume the car is driven 12,000 mi a year at an average speed of 55 mi/h. Take the density and price of gasoline to be 50 lbm/ft3 and $3.10/gal, respectively; the density of air to be 0.075 lbm/ft3, the heating value of gasoline to be 20,000 Btu/lbm; and the overall efficiency of the engine to be 30 percent.

Step 1 of 6

Given data:

Distance covered by the car in an year, $D = 12{,}000 \text{ miles} = 63.36\times10^6 \text{ ft}$

Average speed of the car $V = 55 \text{ mi/h} = 80.62 \text{ ft/s}$

Densities of air, $\rho_{air} = 0.075 \text{ lbm/ft}^3$

Densities of gasoline, $\rho_{gas} = 50 \text{ lbm/ft}^3$

Cost of gasoline, $C = \$3.10/\text{gal}$

The heating value of gasoline, $H = 20{,}000 \text{ Btu/lbm}$

Initial frontal area, $A_1 = 18 \text{ ft}^2$

Reduced frontal area, $A_2 = 15 \text{ ft}^2$

Overall efficiency of the engine, $\eta = 30\%$

Step 2 of 6

From table 15-2 for a passenger car drag coefficient is considered as $C_D = 0.3$

The drag force acting on a body is given by,

$$F_D = C_D \frac{1}{2} \rho_{air} A_1 V^2$$
$$= 0.3\times\frac{1}{2}\times0.075\times18\times80.63^2$$
$$= 1316.49 \text{ lbf}$$

Step 3 of 6

The amount of work done to overcome this drag force,

$$W_1 = F_D \times D$$
$$= 1316.49\times63.36\times10^6$$
$$= 83.41\times10^9 \text{ lbf}\cdot\text{ft/year}$$

Energy expended in doing this work,

$$E = \frac{W_1}{\eta}$$
$$= \frac{83.41\times10^9}{0.3}$$
$$= 278.03\times10^9 \text{ lbf-ft/year}$$
$$= 375.29\times10^6 \text{ Btu/year}$$

Step 4 of 6

The amount of fuel used is given by,

$$F_{amount} = \frac{m}{\rho_{gas}}$$
$$= \frac{E/H}{\rho_{gas}}$$
$$= \frac{375.29\times10^6/20000}{50}$$
$$= 357.29 \text{ ft}^3/\text{year}$$

Then the amount costs expended on fuel,

$$C_{total} = C \times F_{amount}$$
$$= 375.29\times3.10\times\frac{7.84 \text{ ft}^3}{1 \text{ gal}}$$
$$= 8683.61 \$/\text{year}$$

Step 5 of 6

The percentage of reduced area is,

$$A_p = \frac{A_1 - A_2}{A_1}$$
$$= \frac{18-15}{18}$$
$$= 16.67\%$$

The amount of fuel saved due to reduction in area,

$$F_{sav} = F_{amount} \times A_p$$
$$= 375.29\times0.1667$$
$$= 62.56 \text{ ft}^3/\text{year}$$

Step 6 of 6

Amount of money saved due to reduced area ,

$$C_{saved} = F_{sav} \times C$$
$$= 62.56\times3.1\times\frac{7.84 \text{ ft}^3}{1 \text{ gal}}$$
$$= 1520.48 \$/\text{year}$$

Therefore due to reduction of frontal area the amount of fuel saved is $\boxed{62.56 \text{ ft}^3/\text{year}}$ and the amount of money saved is $\boxed{1520.48 \$/\text{year}}$.

Question:

Reconsider Prob. 15-23E. Using EES (or other) software, investigate the effect of frontal area on the annual fuel consumption of the car. Let the frontal area vary from 10 to 30 ft2 in increments of 2 ft2. Tabulate and plot the results.

Step 1 of 5

Distance covered by the car in an year, $D = 12{,}000$ miles $= 63.36 \times 10^6$ ft

Average speed of the car $V = 55$ mi/h $= 80.62$ ft/s

Densities of air, $\rho_{air} = 0.075$ lbm/ft^3

Densities of gasoline, $\rho_{gas} = 50$ lbm/ft^3

Cost of gasoline, $C = \$3.10$/gal

The heating value of gasoline, $H = 20{,}000$ Btu/lbm

Frontal area of car, A

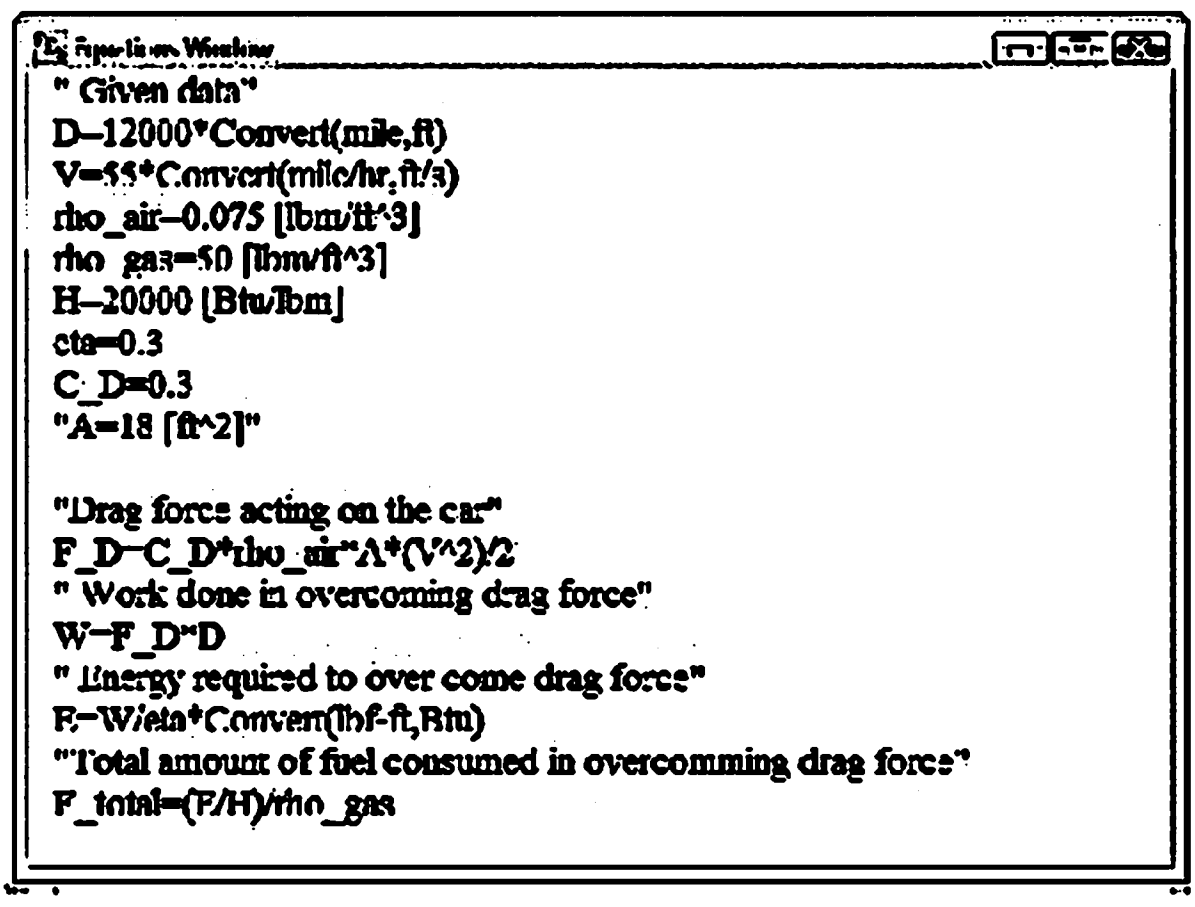
Overall efficiency of the engine, $\eta = 30\%$

Step 2 of 5

From table 15-2 for a passenger car drag coefficient is considered as $C_D = 0.3$

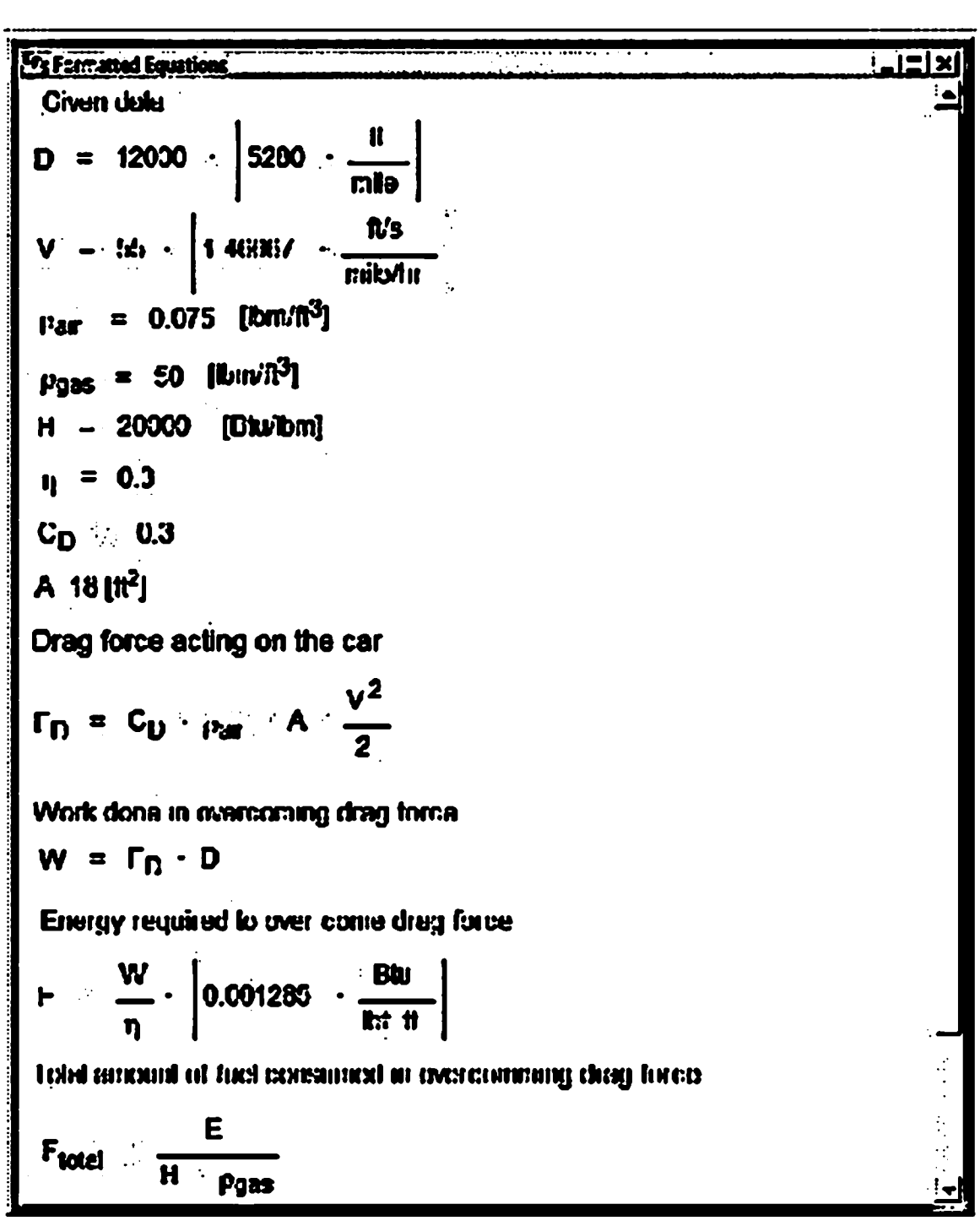
This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,



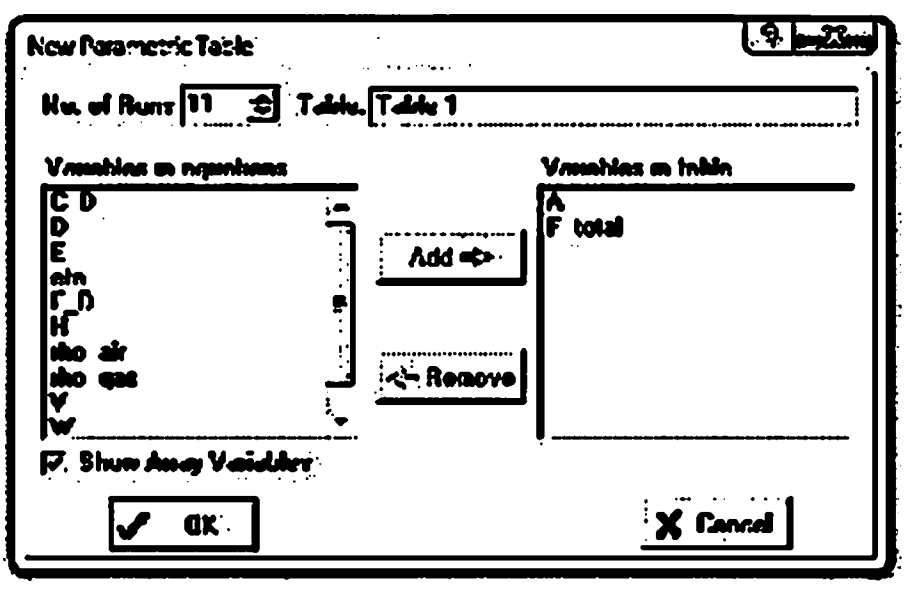
Step 3 of 5

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below ,



Step 4 of 5

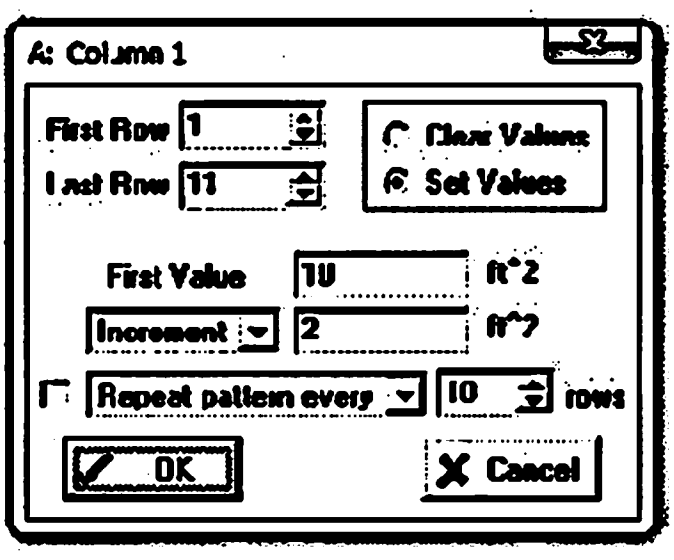
Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 5 of 5

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



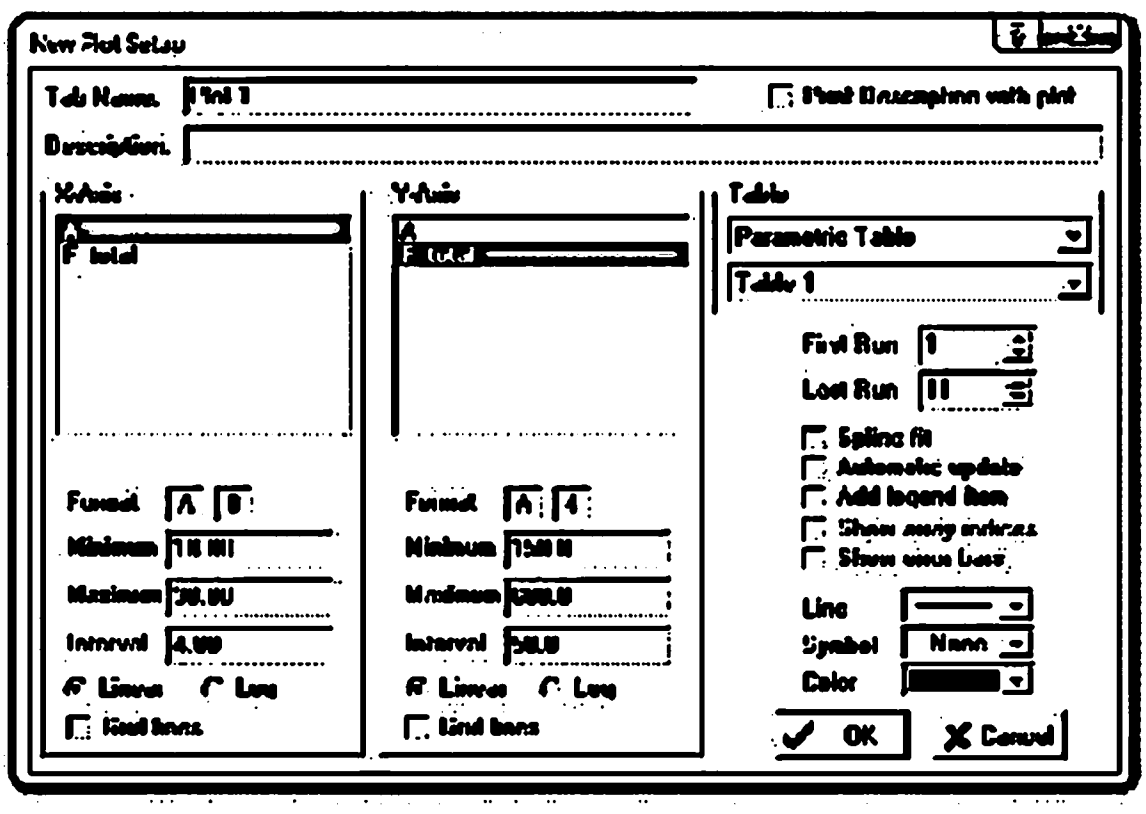
Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric table

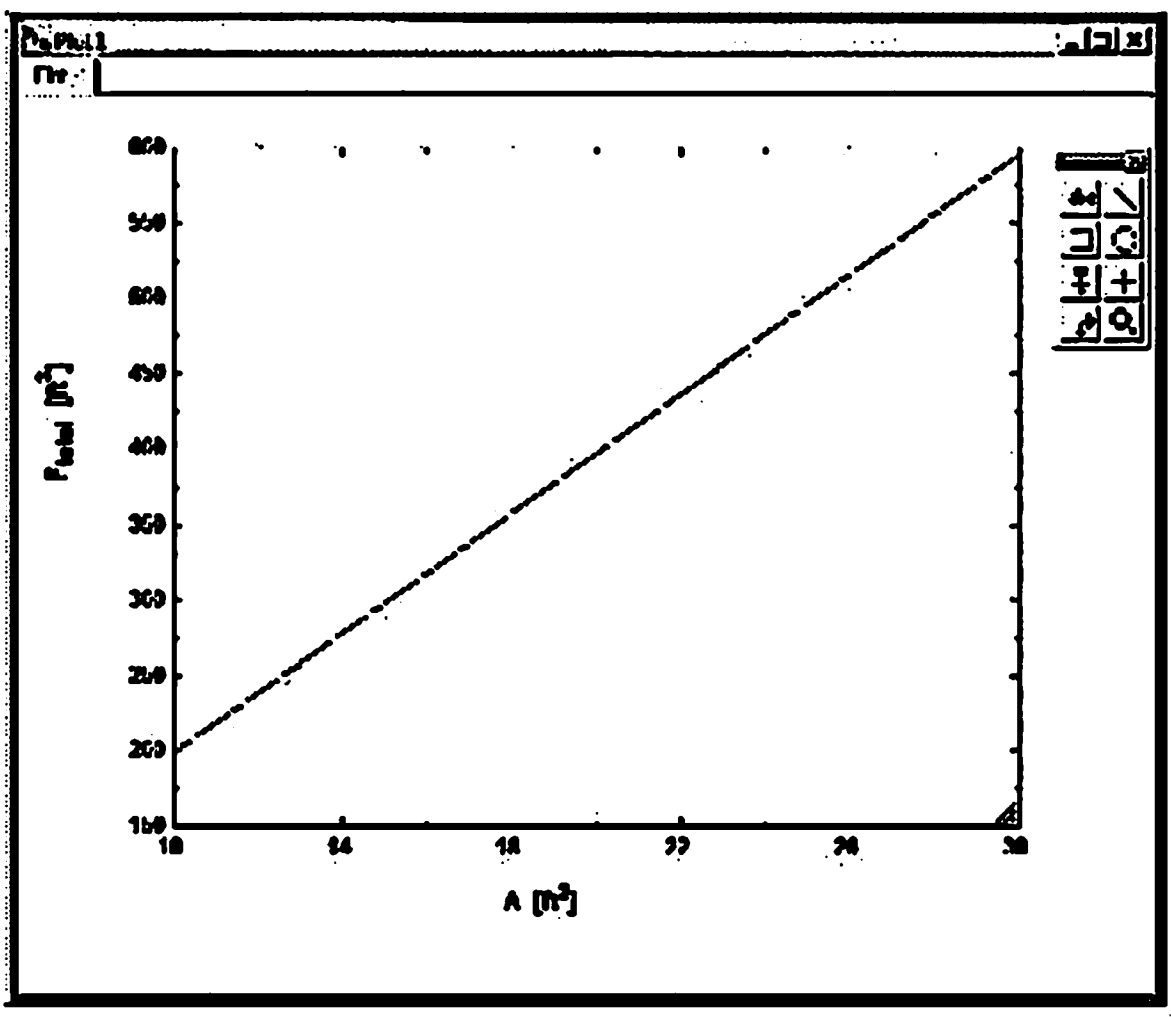
Table 1

	A [ft2]	F total [ft3]
Run 1	10	198.7
Run 2	12	238.4
Run 3	14	278.1
Run 4	16	317.9
Run 5	18	357.6
Run 6	20	397.3
Run 7	22	437.1
Run 8	24	476.8
Run 9	26	516.6
Run 10	28	556.3
Run 11	30	596

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



By clicking on OK the graph will be generated as shown,



Question: 15.025P

A circular sign has a diameter of 50 cm and is subjected to normal winds up to 150 km/h at 10°C and 100 kPa. Determine the drag force acting on the sign. Also determine the bending moment at the bottom of its pole whose height from the ground to the bottom of the sign is 1.5 m. Disregard the drag on the pole.

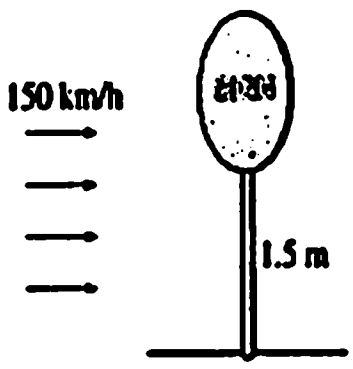


FIGURE P15–25

Step 1 of 3

Given data:

Diameter of the circular sine board, $d = 50\,\text{cm} = 0.5\,\text{m}$

Velocity of the wind, $V = 150\,\text{km/h} = 41.6\,\text{m/s}$

Length of the pole, $l = 1.5\,\text{m}$

Pressure of air, $P = 100\,\text{kPa}$

Temperature of the air, $T = 10°\text{C} = 283\,\text{K}$

Coefficient of drag, $C_D = 1.1$

Universal gas constant, $R = 0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K}$

Step 2 of 3

Density of air at given pressure and temperature,

$$\rho = \frac{P}{RT}$$
$$= \frac{100}{0.287 \times 283}$$
$$= 1.23\,\text{kg/m}^3$$

Frontal area is given by subjected to wind flow,

$$A = \frac{\pi}{4} d^2$$
$$= \frac{\pi}{4} \times 0.5^2$$
$$= 0.1963\,\text{m}^2$$

Step 3 of 3

The drag force on the circular plate is given by,

$$F_D = C_D\left(\frac{1}{2}\rho V^2 A\right)$$
$$= 1.1\left(\frac{1}{2} \times 1.23 \times 41.66^2 \times 0.1963\right)$$
$$= 1.1 \times 209.524$$
$$= 230.47\,\text{N}$$

The bending moment on the bottom surface of the pole,

$$M = F_D \times l$$
$$= 230.47 \times 1.5$$
$$= 345.705\ \text{N}\cdot\text{m}$$

The drag force acting on the circular sine bored at given conditions obtained as $\boxed{F_D = 230.47\,\text{N}}$ and the moment at the bottom of the sine board pole is obtained as $\boxed{M = 345.705\ \text{N}\cdot\text{m}}$

Question: 15.026P

Wind loading is a primary consideration in design of the supporting mechanisms of billboards, as evidenced by many billboards being knocked down during high winds. Determine the wind force acting on an 8-ft-high, 20-ft-wide billboard due to 75-mi/h winds in the nori™ direction when the atmospheric conditions are 14.3 psia and 40°F.

Step 1 of 3

Given data:

Height of the billboard, $D = 8\,\text{ft}$

With of the billboard, $L = 20\,\text{ft}$

Velocity of the wind, $V = 75\,\text{mi/h} = 110\,\text{ft/s}$

Pressure of air, $P = 14.3\,\text{psi}$

Temperature of the air, $T = 40°\text{F} = 500\,\text{R}$

Universal gas constant, $R = 0.3704\ \text{psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R}$

Step 2 of 3

Coefficient of drag for rectangular plate is given by,

$$\begin{aligned} C_D &= 1.10 + 0.02\left[(L/D) + (D/L)\right] \\ &= 1.10 + 0.02\left[\left(\frac{20}{8}\right) + \left(\frac{8}{20}\right)\right] \\ &= 1.10 + 0.02\left[2.5 + 0.4\right] \\ &= 1.158 \end{aligned}$$

Density of air at given pressure and temperature,

$$\begin{aligned} \rho &= \frac{P}{RT} \\ &= \frac{14.3}{0.3704 \times 500} \\ &= \frac{14.3}{185.2} \\ &= 0.0772\ \text{lbm/ft}^3 \end{aligned}$$

Step 3 of 3

Frontal area is given by subjected to wind flow,

$$\begin{aligned} A &= L \times D \\ &= 20 \times 8 \\ &= 160\,\text{ft}^2 \end{aligned}$$

The drag force on the billboard is given by,

$$\begin{aligned} F_D &= C_D\left(\frac{1}{2}\rho V^2 A\right) \\ &= 1.158\left(\frac{1}{2} \times 0.0772 \times 110^2 \times 160\right) \\ &= 86536.87\,\text{lb} \\ &= \frac{86536.87}{32.2} \end{aligned}$$

$$= 2687.48\,\text{lbf}$$

Therefore the drag force acting on the billboard at given conditions obtained as $\boxed{F_D = 2687.48\,\text{lbf}}$

Question: 15.027P

Advertisement signs are commonly carried by taxi cabs for additional income, but they also increase the fuel cost Consider a sign that consists of a 0.30-m-high, 0.9-m-wide, and 0.9-m-long rectangular block mounted on top of a taxi cab such that the sign has a frontal area of 0.3 m by 0.9 m from all four sides. Determine the increase in the annual fuel cost of this taxicab due to this sign. Assume the taxicab is driven 60,000 km a year at an average speed of 50 km/h and the overall efficiency of the engine is 28 percent. Take the density, unit price, and heating value of gasoline to be 0.72 kg/L, \$1.10/L, and 42,000 kJ/kg, respectively, and the density of air to be 1.25 kg/m3.



FIGURE P15–27

Step 1 of 5

Given data:

Height of the billboard $D = 0.3\text{m}$

With of the billboard $W = 0.9\text{m}$

Length of the billboard $L = 0.9\text{m}$

Frontal area is given by, $A = 0.3\text{m} \times 0.9\text{m} = 0.27\,\text{m}^2$

Distance covered by the cab in a year, $d = 60,000\text{km/year}$

Average speed of the cab, $V_{cab} = 50\text{km/h} = 13.88\text{m/s}$

Velocity of the wind $V = 150\text{km/h} = 41.6\text{ m/s}$

Over all efficiency of the engine $\eta = 28\%$

Density of gasoline, $\rho_{gas} = 0.72\text{kg/L}$

Unit price of gasoline, $P_g = \$1.10/\text{L}$

Heating value of gasoline, $h = 42,000\text{kJ/kg}$

Density of air, $\rho = 1.25\text{kg/m}^3$

Step 2 of 5

Coefficient of drag for the sine board is as $C_D = 2.2$ from the table 15-1.

The drag force on the circular plate is given by,

$$F_D = C_D\left(\frac{1}{2}\rho V^2 A\right)$$
$$= 2.2\left(\frac{1}{2}\times 1.25\times 13.88^2\times 0.27\right)$$
$$= 71.523\text{N}$$

Step 3 of 5

Amount of drag force acting on the cab due to sine board for entire year can be obtained as,

$$W_{drag} = F_D \times d$$
$$= 71.523\times 60000$$
$$= 4.29\times 10^6\text{ kJ/year}$$

Energy expended on overcoming drag force due to the sine board for an entire year,

$$E_w = \frac{W_{drag}}{\eta}$$
$$= \frac{4.29\times 10^6}{0.28}$$
$$= 15.32\times 10^6\text{ kJ/year}$$

Step 4 of 5

The total amount of fuel consumed for entire year,

$$A_{fuel} = \frac{m_{gas}}{\rho_{gas}}$$
$$= \frac{E_w / h}{\rho_{gas}}$$
$$= \frac{15.32\times 10^6 / 42000}{0.72}$$
$$= 506.61\text{L/year}$$

Step 5 of 5

Annual fuel cost is given by,

$$P_{year} = P_g \times A_{fuel}$$
$$= 1.10\times 506.61$$
$$= 557.27\,\$/\text{year}$$

Increase in annual fuel cost due to the action of drag force on sign is $\boxed{557.27\,\$/\text{year}}$.

Question: 15.028P

At highway speeds, about half of the power generated by the car's engine is used to overcome aerodynamic drag, and thus the fuel consumption is nearly proportional to the drag force on a level road. Determine the percentage increase in fuel consumption of a car per unit time when a person who normally drives at 55 mi/h now starts driving at 75 mi/h.

Step 1 of 3

Given data:

Normal driving speed, $V_1 = 55\,\text{mi/h}$

New speed, $V_2 = 75\,\text{mi/h}$

Step 2 of 3

Equation for drag force,

$$F_D = \frac{1}{2} C_D \rho A V^2$$

Where F_D is the force acting on body,

A is the frontal area,

ρ is the density of air,

V is the velocity of wind,

C_D is the coefficient of drag

As all the parameters of the drag force here are constant except velocity, we get

$$F_D \propto V^2$$

$$\frac{(F_D)_2}{(F_D)_1} = \left(\frac{V_2}{V_1}\right)^2 \qquad \ldots\ldots (1)$$

Step 3 of 3

Fuel consumption is directly proportional to the drag force. Hence percentage increase in fuel consumption is equal to the percentage increase in drag force.

$$\% \Delta f = \% \Delta F_D$$

$$= \frac{(F_D)_2 - (F_D)_1}{(F_D)_1} \times 100$$

$$= \left[\frac{(F_D)_2}{(F_D)_1} - 1\right] \times 100 \qquad \ldots\ldots (2)$$

From equations (1) and (2), we get

$$\% \Delta f = \left[\frac{V_2^2}{V_1^2} - 1\right] \times 100$$

$$= \left[\frac{(75)^2}{(55)^2} - 1\right] \times 100$$

$$= 85.95\,\%$$

Hence percentage increase in fuel consumption is $\boxed{\% \Delta f = 85.95\,\%}$

Question: 15.029P

Bill gets a job delivering pizzas. The pizza company makes him mount a sign on the roof of his car. The frontal area of the sign is A = 0.612 ft2, and he estimates the drag coefficient to beCD = 0.94 at nearly all air speeds. Estimate how much additional money it costs Bill per year in fuel to drive with the sign on his roof compared to without the sign. Use the following additional information: He drives about 10,000 miles per year at an average speed of 45 mph. The overall car efficiency is 0.332, ρfuel = 50.2 lbm/ft3, and the heating value of the fuel is 1.53 × 107 ft.lbf/lbm. The fuel costs $3.50 per gallon. Use standard air properties. Be careful with unit conversions.

Step 1 of 4

Sign frontal area, $A = 0.612\,\text{ft}^2$

Sign board drag coefficient, $C_D = 0.94$

Distance covered by the car in an year, $D = 10{,}000\ \text{miles} = 52.8\times10^6\,\text{ft}$

Average speed of the car $V = 45\,\text{mi/h} = 66\,\text{ft/s}$

Overall efficiency of the engine, $\eta = 33.2\%$

Densities of gasoline, $\rho_{gas} = 50.2\ \text{lbm/ft}^3$

Cost of gasoline, $C = \$3.50/\text{gal}$

The heating value of gasoline, $H = 1.53\times10^7\ \text{ft}\cdot\text{lbf/lbm}$

Consider densities of air, $\rho_{air} = 0.075\ \text{lbm/ft}^3$

Step 2 of 4

The drag force acting on a sign board is given by,

$$F_D = C_D\frac{1}{2}\rho_{air}AV^2$$
$$= 0.94\times\frac{1}{2}\times0.075\times0.612\times66^2$$
$$= 93.97\,\text{lbf}$$

The amount of work done to overcome this drag force for an entire year,

$$W = F_D\times D$$
$$= 93.97\times52.8\times10^6$$
$$= 4961.72\times10^6\ \text{lbf}\cdot\text{ft}$$

Step 3 of 4

Energy expended in doing this work for the whole year is,

$$E = \frac{W}{\eta}$$
$$= \frac{4961.72\times10^6}{0.332}$$
$$= 14.94\times10^3\ \text{lbf}\cdot\text{ft}$$

The amount of fuel consumed in the entire given year is,

$$F_{amount} = \frac{m}{\rho_{gas}}$$
$$= \frac{E/H}{\rho_{gas}}$$
$$= \frac{14.944\times10^9/1.53\times10^7}{50.2}$$
$$= 194.57\,\text{ft}^3$$

Step 4 of 4

Then the amount costs expended on fuel,

$$C_{total} = C\times F_{amount}$$
$$= 3.5\times0.2948\times\frac{1\,\text{gal}}{7.84\,\text{ft}^3}$$
$$= 86.86\$$

Therefore the annual amount of expenditure that costs Bill due to the sine board is $\boxed{86.86\$}$

Question: 15.030P

During major windstorms, high vehicles such as RVs d semis may be thrown off the road and boxcars off their tracks, especially when they are empty and in open areas. Consider a 5000-kg semi that is 9 m long, 2.5 m high, and 2 m wide. The distance between the bottom of the truck and the road is 0.75 m. Now the truck is exposed to winds from its side surface. Determine the wind velocity that will tip the truck over to its side. Take the air density to be 1.1 kg/m3 and assume the weight to be uniformly distributed.

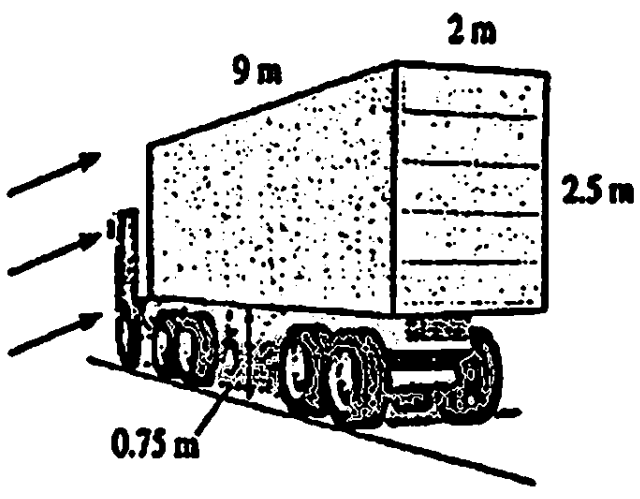


FIGURE P15–30

Step 1 of 6

Given data:

Mass of the vehicle, $m = 5000\ \text{kg}$

Length of the vehicle, $L = 9\ \text{m}$

Height of the vehicle, $h = 2.5\ \text{m}$

Width of the vehicle, $w = 2\ \text{m}$

Distance between the bottom of the truck and the road, $l = 0.75\ \text{m}$

Density of air, $\rho = 1.1\ \text{kg/m}^3$

Coefficient of drag, $C_D = 2.2$

Let the frontal area be *A* and the velocity of wind be *V*

Step 2 of 6

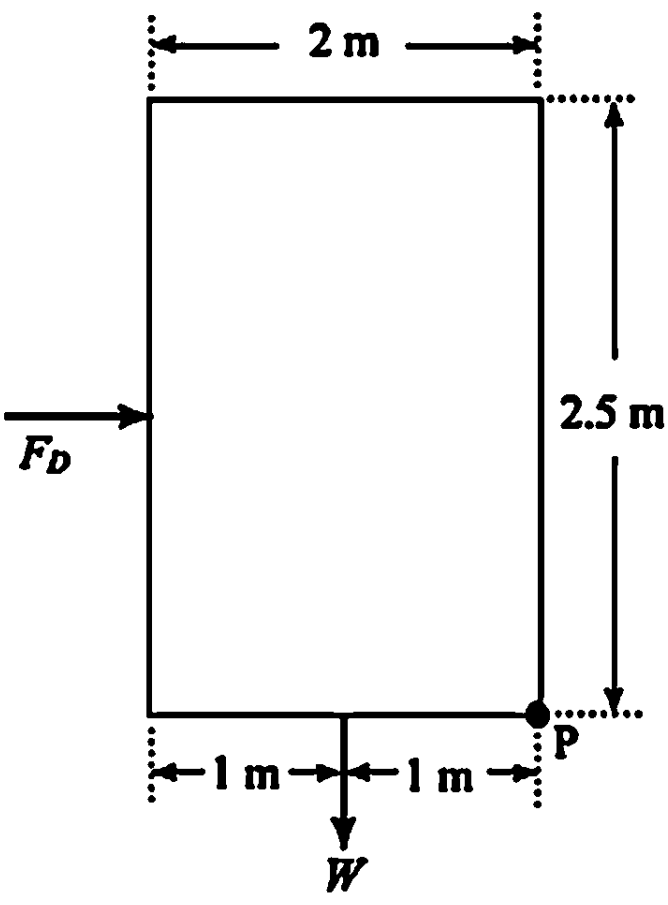
When the truck is tipped off the wheels on wind loaded side will be off the ground first. Therefore, all the reactions will act on the wheels of other side. To find out the drag force that caused the truck to tip over. Consider an axis passing through the wheels on the non-wind loaded side of the truck and setting the moment about that point equal to zero will give the required drag force.

Weight of the truck,

$$\begin{aligned} W &= mg \\ &= 5000 \times 9.81 \\ &= 49050\ \text{N} \end{aligned}$$

Step 3 of 6

Figure showing the forces acting on truck,



Step 4 of 6

As the Drag force (F_D) on truck acts at middle, the moment about point P would be,

$$\begin{aligned} \Sigma M_P &= 0 \\ F_D\left(\frac{2.5}{2}\right) - W\left(\frac{2}{2}\right) &= 0 \\ (F_D \times 1.25) &= (49050 \times 1) \\ F_D &= \frac{49050}{1.25} \\ &= 39240\ \text{N} \end{aligned}$$

Step 5 of 6

The drag force acting on the truck,

$$F_D = \frac{1}{2} C_D \rho A V^2 \quad \text{..... (1)}$$

The frontal area is given by,

$$\begin{aligned} A &= L \times h \\ &= 9 \times 2.5 \\ &= 22.5\ \text{m}^2 \end{aligned}$$

Step 6 of 6

Now, by substituting the values in equation (1), we get

$$\begin{aligned} F_D &= \frac{1}{2} C_D \rho A V^2 \\ 39240 &= \frac{1}{2} \times 2.2 \times 1.1 \times 22.5 \times V^2 \\ V^2 &= 1441.32 \\ V &= 37.96\ \text{m/s} \end{aligned}$$

Therefore the wind velocity that will tip the truck over to its side is, $\boxed{V = 37.96\ \text{m/s}}$

Question: 15.031P

An 80-kg bicyclist is riding her 15-kg bicycle downhill on a road with a slope of 12° without pedaling or braking. The bicyclist has a frontal area of 0.45 m2 and a drag coefficient of 1.1 in the upright position, and a frontal area of 0.4 m2 and a drag coefficient of 0.9 in the racing position. Disregarding the rolling resistance and friction at the bearings, determine the terminal velocity of the bicyclist for both positions. Take the air density to be 1.25 kg/m3.

Step 1 of 6

Given data:

Mass of bicyclist is $m_1 = 80\,\text{kg}$.

Mass of bicycle is $m_2 = 15\,\text{kg}$.

Slope of road is $\theta = 12°$.

In upright position,

Frontal area is $A_1 = 0.45\ \text{m}^2$.

Drag coefficient is $C_{D1} = 1.1$.

In racing position,

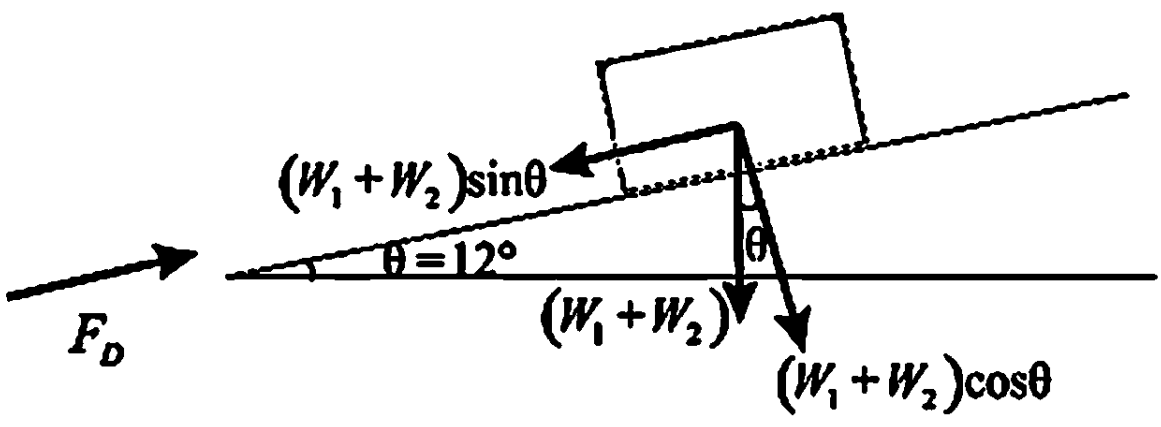
Frontal area is $A_2 = 0.4\ \text{m}^2$.

Drag coefficient is $C_{D_2} = 0.9$.

Density of air is $\rho = 1.25\,\text{kg/m}^3$.

Step 2 of 6

Diagram for the given conditions:



Step 3 of 6

Terminal velocity is attained when the sum of the weight of the bicycle and bicyclist is equal to the drag force acting opposite to it.

$$F_D = (W_1+W_2)\sin\theta \qquad \ldots\ldots (1)$$

Here W_1 is the weight of the bicyclist and W_2 is the weight of the bicycle.

$$W_1 = m_1 g$$
$$= 80 \times 9.81\ \text{kg}\cdot\text{m/s}^2$$
$$= 784.8\ \text{N}$$

Now,

$$W_2 = m_2 g$$
$$= 15 \times 9.81\ \text{kg}\cdot\text{m/s}^2$$
$$= 147.15\ \text{N}$$

Step 4 of 6

Substituting the values of W_1, W_2 and 6 in equation (1).

$$F_D = (W_1+W_2)\sin 12°$$
$$= (784.8+147.15)\sin 12°$$
$$= 193.76\ \text{N}$$

Then the drag force is $F_D = \frac{1}{2} C_D A \rho V^2$ ..... (2)

Where, V is the velocity.

A is the frontal area.

ρ is the density of air.

C_D is the drag coefficient.

Step 5 of 6

Consider upright position.

Here,

Frontal area, $A_1 = 0.45\ \text{m}^2$

Drag coefficient, $C_{D1} = 1.1$

Drag force acting for upright position,

$$F_D = \frac{1}{2} C_{D1} A_1 \rho V_1^2$$

Substituting the values of F_D, A_1, C_{D1} and ρ in the above equation.

$$193.76 = \frac{1}{2} \times 1.1 \times 0.45 \times 1.25 \times V_1^2$$
$$V_1^2 = 626.29$$
$$V_1 = 25.03\ \text{m/s}$$
$$= 25.03\left(\frac{18}{5}\right)$$
$$= 90.108\ \text{km/h}$$

Step 6 of 6

Now, consider racing position.

Here

Frontal area, $A_2 = 0.4\ \text{m}^2$

Drag coefficient, $C_{D_2} = 0.9$

Drag force acting for racing position,

$$F_D = \frac{1}{2} C_{D2} A_2 \rho V_2^2$$

Substituting the values of F_D, A_2, C_{D2} and ρ in the above equation.

$$193.76 = \frac{1}{2} \times 0.9 \times 0.4 \times 1.25 \times V_2^2$$
$$V_2^2 = 861.15$$
$$V_2 = 29.35\ \text{m/s}$$
$$= 29.35\left(\frac{18}{5}\right)$$
$$= 105.6\ \text{km/h}$$

Therefore, the terminal velocity of the bicyclist in the upright position is $\boxed{V_1 = 90.108\ \text{km/h}}$.

And the terminal velocity of the bicyclist in the racing position is $\boxed{V_2 = 105.6\ \text{km/h}}$.

Question: 15.032P

A wind turbine with two or four hollow hemispherical cups connected to a pivot is commonly used to measure wind speed. Consider a wind turbine with four 8-cm-diameter cups with a center-to-center distance of 40 cm, as shown in

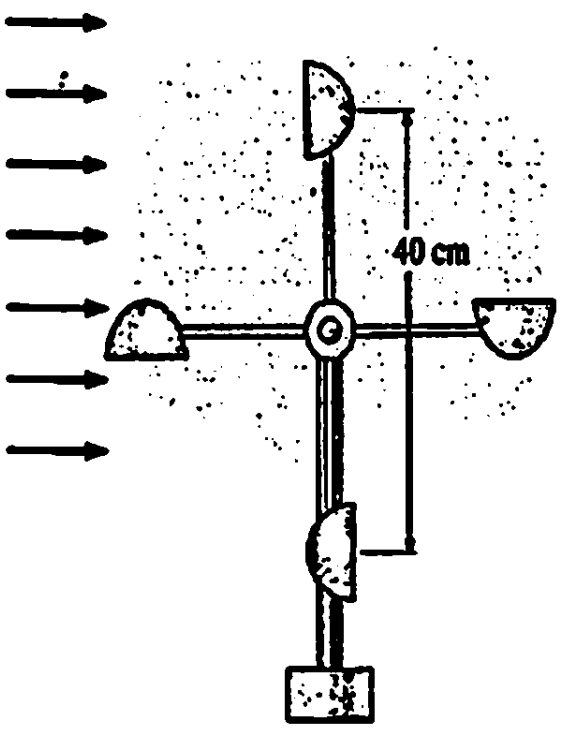


FIGURE P15–32

Fig. PI5-32. The pivot is stuck as a result of some malfunction, and the cups stop rotating. For a wind speed of 15 m/s and air density of 1.25 kg/m3, determine the maximum torque this turbine applies on the pivot.

Step 1 of 7

Given data:

Diameter of the cup, $D = 8$ cm

Center to center distance, $L = 40$ cm

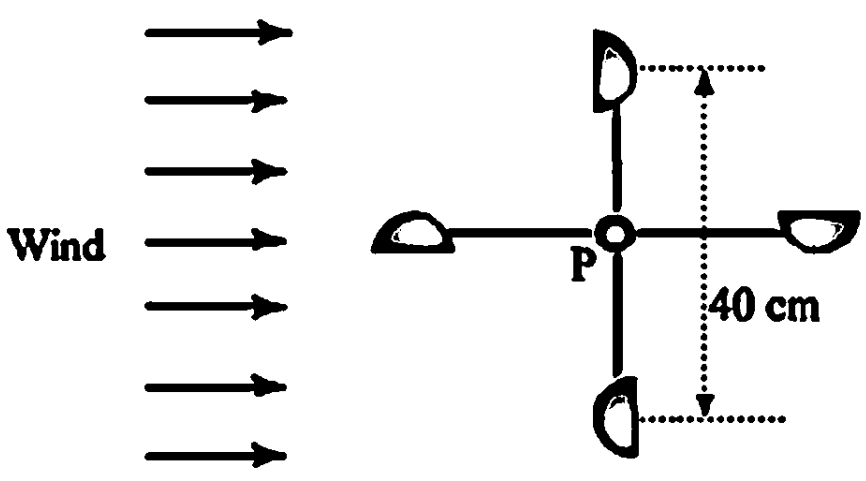
Wind speed, $V = 15$ m/s

Density of air, $\rho = 1.25\ \text{kg/m}^3$

Let the frontal area of the hemispherical cup be A and the velocity of wind be V

Step 2 of 7

Maximum torque occurs when two cups are normal to the wind because the length of the moment arm is maximum in this case.



Step 3 of 7

Drag force acting on the convex side,

$$F_{D_1} = \frac{1}{2} C_{D_1} \rho A V^2 \qquad \ldots\ldots (1)$$

Frontal area,

$$A = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.08)^2$$

$$= 0.005\ \text{m}^2$$

Here, the drag coefficient for the hemisphere is $C_{D_1} = 0.4$ from table of drag coefficient for various three-dimensional bodies based on frontal area.

Step 4 of 7

Now, substitute the values of C_{D1}, ρ, A and V in equation (1) we have,

$$F_{D_1} = \frac{1}{2} \times 0.4 \times 1.25 \times 0.005 \times (15)^2$$

$$= 0.28\ \text{N}$$

Now, drag force acting on the concave side is given by,

$$F_{D_2} = \frac{1}{2} C_{D_2} \rho A V^2 \qquad \ldots\ldots (2)$$

Frontal area,

$$A = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.08)^2$$

$$= 0.005\ \text{m}^2$$

Step 5 of 7

Here, the drag coefficient for the hemisphere is $C_{D2} = 1.2$ from table of drag coefficient for various three-dimensional bodies based on frontal area

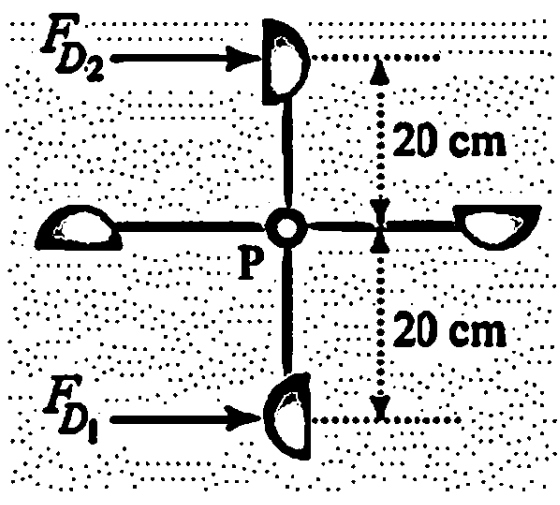
Now, substitute the values of C_{D2}, ρ, A and V in equation (2)

$$F_{D_2} = \frac{1}{2} \times 1.2 \times 1.25 \times 0.005 \times (15)^2$$

$$= 0.84\ \text{N}$$

Step 6 of 7

Free body diagram to show the forces acting on the hemispherical cups.



Step 7 of 7

To find the maximum torque acting, take moment about the pivot when two cups are normal to the direction of wind.

$+\circlearrowleft \sum M_P =$ Maximum torque

$$\text{Maximum torque} = F_{D_2}\left(\frac{L}{2}\right) - F_{D_1}\left(\frac{L}{2}\right)$$

$$= 0.84(0.02) - 0.28(0.02)$$

$$= 0.0112\ \text{N}\cdot\text{m}$$

Therefore the maximum torque the turbine applies on the pivot is $\boxed{T_{\max} = 0.0112\ \text{N}\cdot\text{m}}$

Question: 15.033P pg 1

Reconsider Prob. 15-32. Using EES (or other) Mm software, investigate the effect of wind speed on the torque applied on the pivot. Let the wind speed vary from 0 to 50 m/s in increments of 5 m/s. Tabulate and plot the results.

Step 1 of 2

4749-15-33P AID: 1083 | 14/05/2012

Given data:

Diameter of the cup, $D = 8$ cm

Center to center distance, $L = 40$ cm

Wind speed, $V = 15$ m/s

Density of air, $\rho = 1.25$ kg/m^3

Frontal area of the hemispherical cup, A

Velocity of wind, V

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

Equations Window

```
" Given data "
C_D1=0.40                    " Coefficient of drag for hemispherical cup on convex side "
C_D2=1.2                     " Coefficient of drag for hemispherical cup on concave side "
rho=1.25 [kg/m3]
D=0.08 [m]
L=0.4 [m]
A=pi*D^2/4
" V=15 [m/s] "
" Drag force acting on convex face of the cup "
F_D1=C_D1*A*(rho*V^2)/2
" Drag force acting on concave face of the cup"
F_D2=C_D2*A*(rho*V^2)/2
" Resutant force aling on the wind turbine"
F_Dnet=F_D2-F_D1
" Net torque acting on the  turbine"
Torque=(F_D2-F_D1)*L/2
```

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below ,

Formatted Equations

Given data

$C_{D1} = 0.4$ Coefficient of drag for hemispherical cup on convex side

$C_{D2} = 1.2$ Coefficient of drag for hemispherical cup on concave side

$\rho = 1.25$ [kg/m3]

$D = 0.08$ [m]

$L = 0.4$ [m]

$$A = \pi \cdot \frac{D^2}{4}$$

V=15 [m/s]

Drag force acting on convex face of the cup

$$F_{D1} = C_{D1} \cdot A \cdot \frac{\rho \cdot V^2}{2}$$

Drag force acting on concave face of the cup

$$F_{D2} = C_{D2} \cdot A \cdot \frac{\rho \cdot V^2}{2}$$

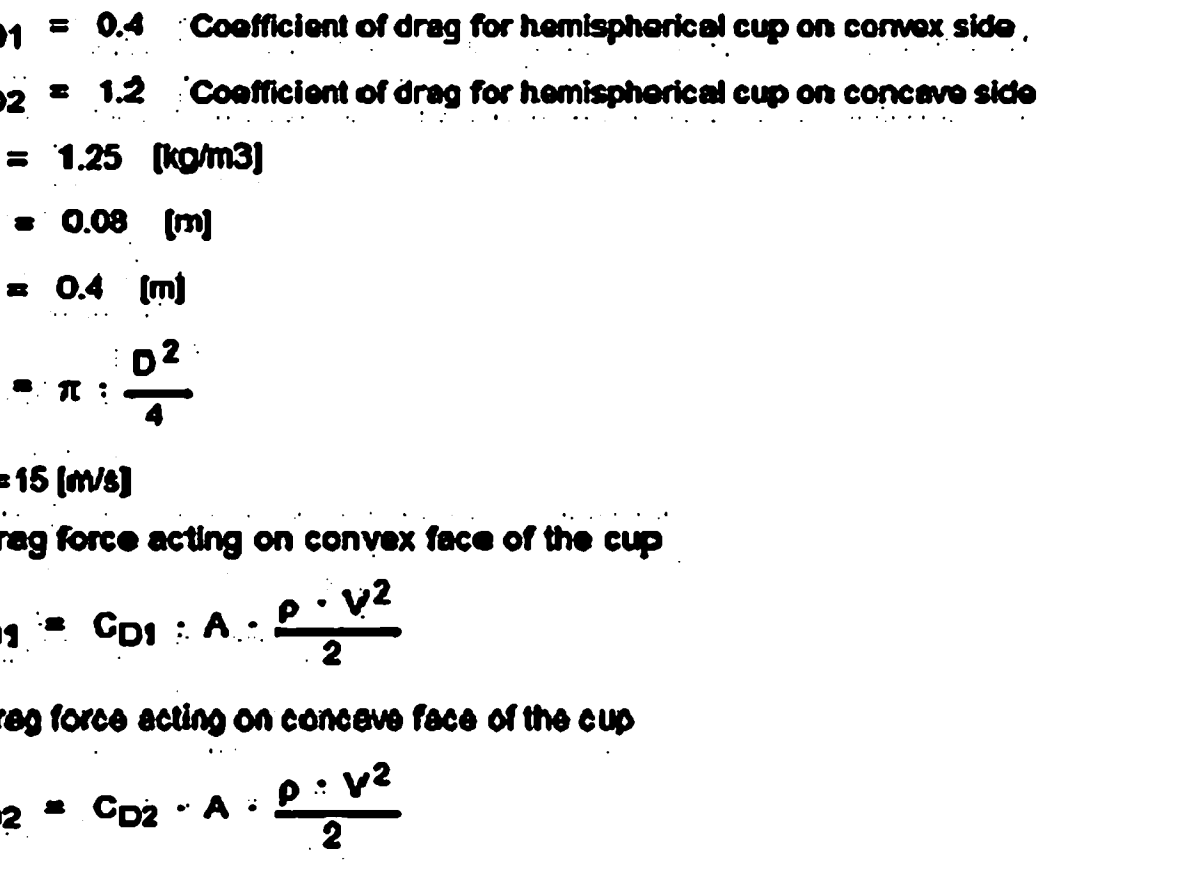
Resutant force aling on the wind turbine

$$F_{Dnet} = F_{D2} - F_{D1}$$

Net torque acting on the turbine

$$Torque = (F_{D2} - F_{D1}) \cdot \frac{L}{2}$$

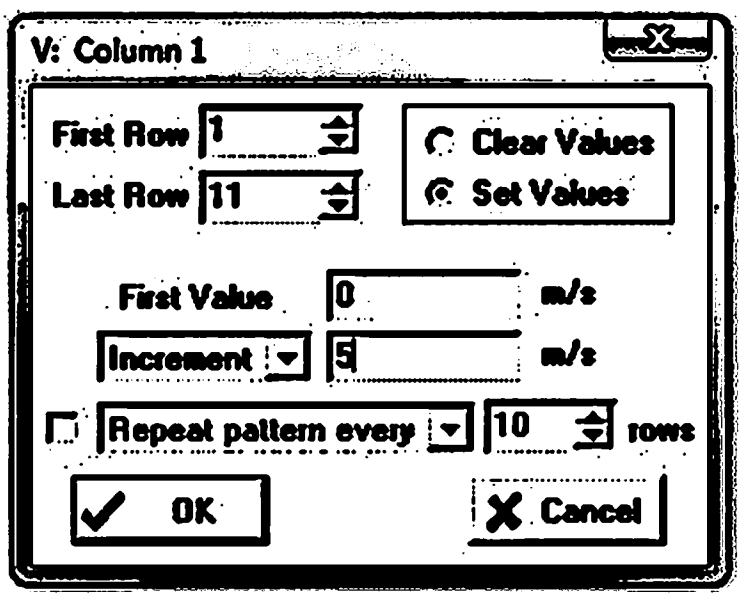
Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 2 of 2

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



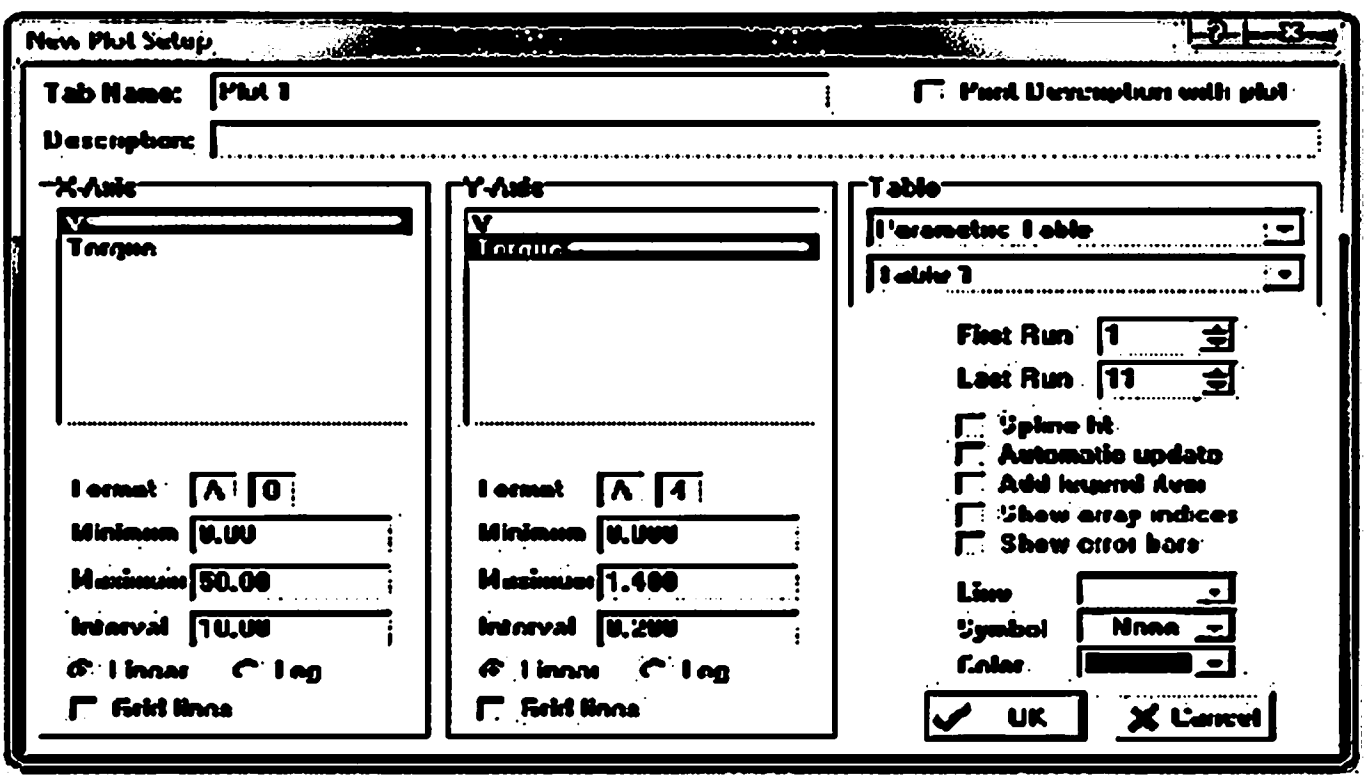
Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

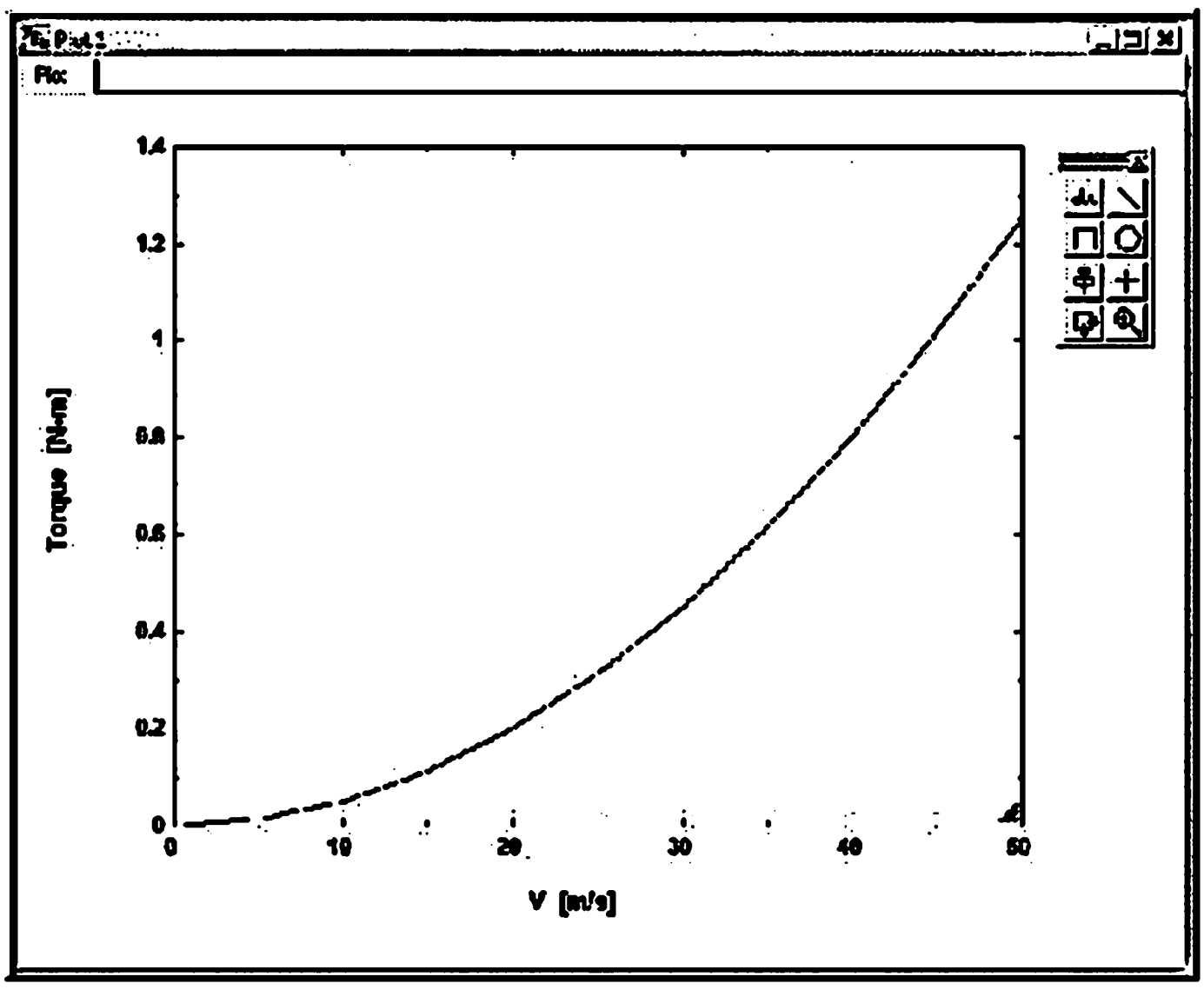
Table 1

1..11	V [m/s]	Torque [N-m]
Run 1	0	0
Run 2	5	0.01257
Run 3	10	0.05027
Run 4	15	0.1131
Run 5	20	0.2011
Run 6	25	0.3142
Run 7	30	0.4524
Run 8	35	0.6158
Run 9	40	0.8042
Run 10	45	1.018
Run 11	50	1.257

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



By clicking on OK the graph will be generated as shown,



Question: 15.034P

A 5-ft-diameter spherical tank completely submerged in freshwater is being towed by a ship at 12 ft/s. Assuming turbulent flow, determine the required towing power.

Step 1 of 4

Given data:

Diameter of the submerged spherical tank, $D = 5\text{ft}$

Velocity of the ship towing it, $V = 12\text{ft/s}$

Density of water is consider as, $\rho_{water} = 62.4\text{lb/ft}^3$

Step 2 of 4

From table 15-2 for a spherical body under turbulent conditions drag force coefficient is considered as $C_D = 0.2$

Frontal area of the spherical tank,

$$A = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}5^2$$

$$= 19.63\text{ft}^2$$

Step 3 of 4

The drag force acting on a body is given by,

$$F_D = C_D \frac{1}{2}\rho_{air} A V^2$$

$$= 0.2 \times \frac{1}{2} \times 62.4 \times 19.63 \times 12^2$$

$$= 17638.73\text{lbf}$$

$$= 17.63 \times 10^3\text{ lbf}$$

Step 4 of 4

The power required in towing the spherical tank,

$$P = F_d \times V$$

$$= 17.63 \times 10^3 \times 12$$

$$= 211 \times 10^3\text{ Btu/s}$$

The power required by the ship for towing a spherical tank is obtained as $\boxed{211 \times 10^3\text{ Btu/s}}$

Question: 15.035P

During steady motion of a vehicle on a level road, the power delivered to the wheels is used to overcome aerodynamic drag and rolling resistance (the product of the rolling resistance coefficient and the weight of the vehicle), assuming the friction at the bearings of the wheels is negligible. Consider a car that has a total mass of 950 kg, a drag coefficient of 0.32, a frontal area of 1.8 m2, and a rolling resistance coefficient of 0.04. The maximum power the engine can deliver to the wheels is 80 kW. Determine (*a*) the speed at which the rolling resistance is equal to the aerodynamic drag force and (*b*) the maximum speed of this car. Take the air density to be 1.20 kg/m3.

Step 1 of 6

Given data:

Mass of the car is $m = 950$ kg

Drag coefficient is $C_D = 0.32$

The rolling resistance coefficient is $C_R = 0.04$

The frontal area is $A = 1.8\ \text{m}^2$

Maximum power engine can deliver to the wheel is $P = 80$ kW

Density of air is $\rho = 1.20\ \text{kg/m}^3$

Step 2 of 6

Rolling resistance of the car,

In the question it is given that rolling resistance is the product of the rolling resistance coefficient and the weight of the vehicle.

$$R_R = C_R W \qquad \text{..... (1)}$$

Where, C_R is the rolling resistance coefficient

W is the weight of the body

Here,

$$W = mg$$
$$= 950 \times 9.81$$
$$= 9319.5\ \text{N}$$

Now, substituting the value of C_R and W in equation (1)

$$R_R = 0.04(9319.5)$$
$$= 372.78\ \text{N}$$

Step 3 of 6

Now, aerodynamic drag force acting on the car,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \text{..... (2)}$$

Where,

C_D is the drag coefficient,

ρ is the density,

A is the frontal area and

V is the velocity.

Substituting the values of C_D, ρ and A in equation (2)

$$F_D = \frac{1}{2} \times 0.32 \times 1.20 \times 1.8 \times V^2$$
$$= 0.3456 \times V^2$$

Step 4 of 6

To find the velocity at which rolling resistance becomes equal to the aerodynamic drag force, equate the values of rolling resistance and aerodynamic drag force.

$$R_R = F_D$$

Therefore,

$$372.78 = 0.3456 \times V^2$$
$$V^2 = \frac{372.78}{0.3456}$$
$$= 1078.65$$
$$V = 32.8\ \text{m/s}$$
$$V = 32.8\left(\frac{18}{5}\right)$$
$$= 118.08\ \text{km/h}$$

Therefore, the speed at which the rolling resistance is equal to the aerodynamic drag force is $\boxed{V = 118.08\ \text{km/h}}$

Step 5 of 6

Power required overcoming aerodynamic drag and rolling resistance, $P = (R_R + F_D)V$.

$$P = \left(372.78 + 0.3456V^2\right)V$$
$$P = 372.78V + 0.3456V^3 \qquad \text{..... (3)}$$

Maximum power is delivered at maximum speed.

So, in order to find the maximum speed, substitute the value of maximum power in equation (3).

$$80000 = 372.78V + 0.3456V^3$$
$$0.3456V^3 + 0V^2 + 372.78V - 80000 = 0$$

Step 6 of 6

On solving the above equation, we get two values for V which are, 55.5 and -27.78.

Since velocity cannot be negative we cannot take -27.78, so we take the value 55.5 as the velocity.

$$V = 55.56\left(\frac{18}{5}\right)$$
$$= 200\ \text{km/h}$$

Therefore, the maximum speed of the car is $\boxed{V = 200\ \text{km/h}}$

Question: 15.036P pg 1

Reconsider Prob. 15-35. Using EES (or other) software, investigate the effect of car speed on the required power to overcome (*a*) rolling resistance, (*b*) the aerodynamic drag, and (*c*) their combined effect. Let the car speed vary from 0 to 150 km/h in increments of 15 km/h. Tabulate and plot the results.

Step 1 of 5

Given data:

Total mass of the car, $m = 950\,\text{kg}$

Coefficient of drag of the car, $C_d = 0.32$

Frontal area of the car, $A = 1.8\,\text{m}^2$

Rolling resistance coefficient, $C_{roll} = 0.04$

The maximum power the of engine, $P_{max} = 80\,\text{kW}$

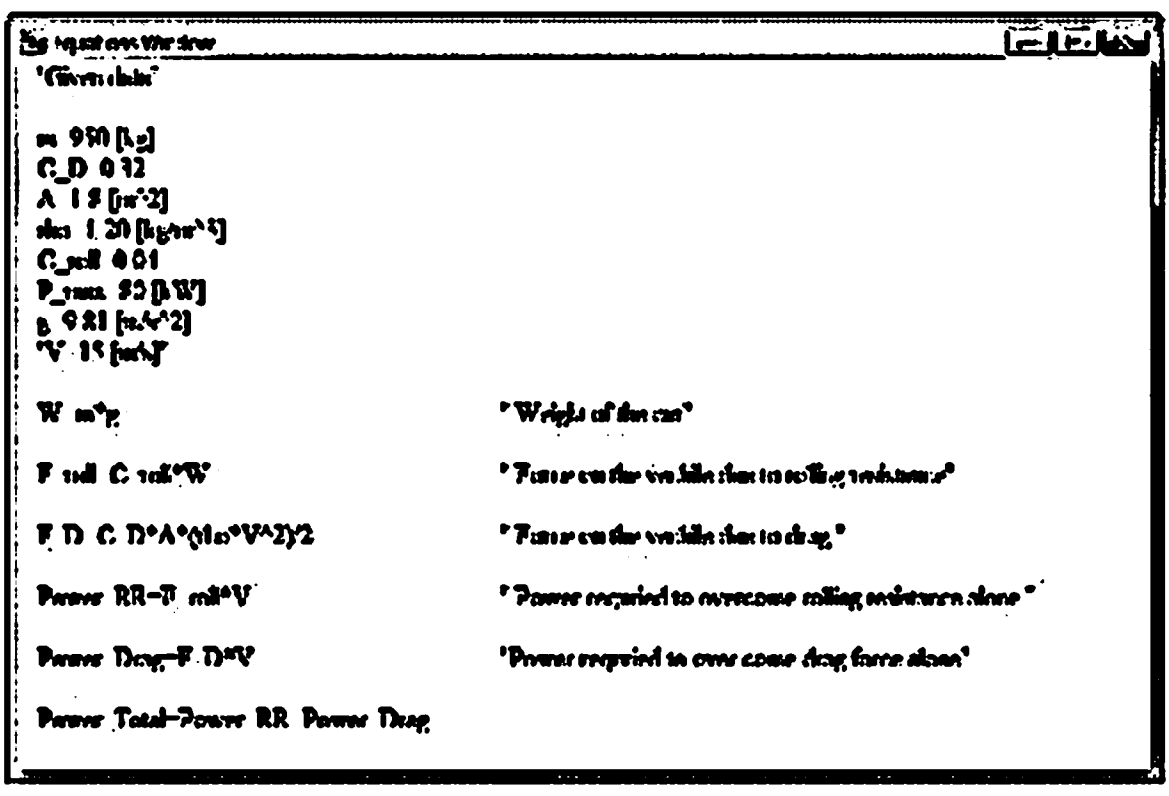
Density of air, $\rho = 1.20\,\text{kg/m}^3$

Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$

Step 2 of 5

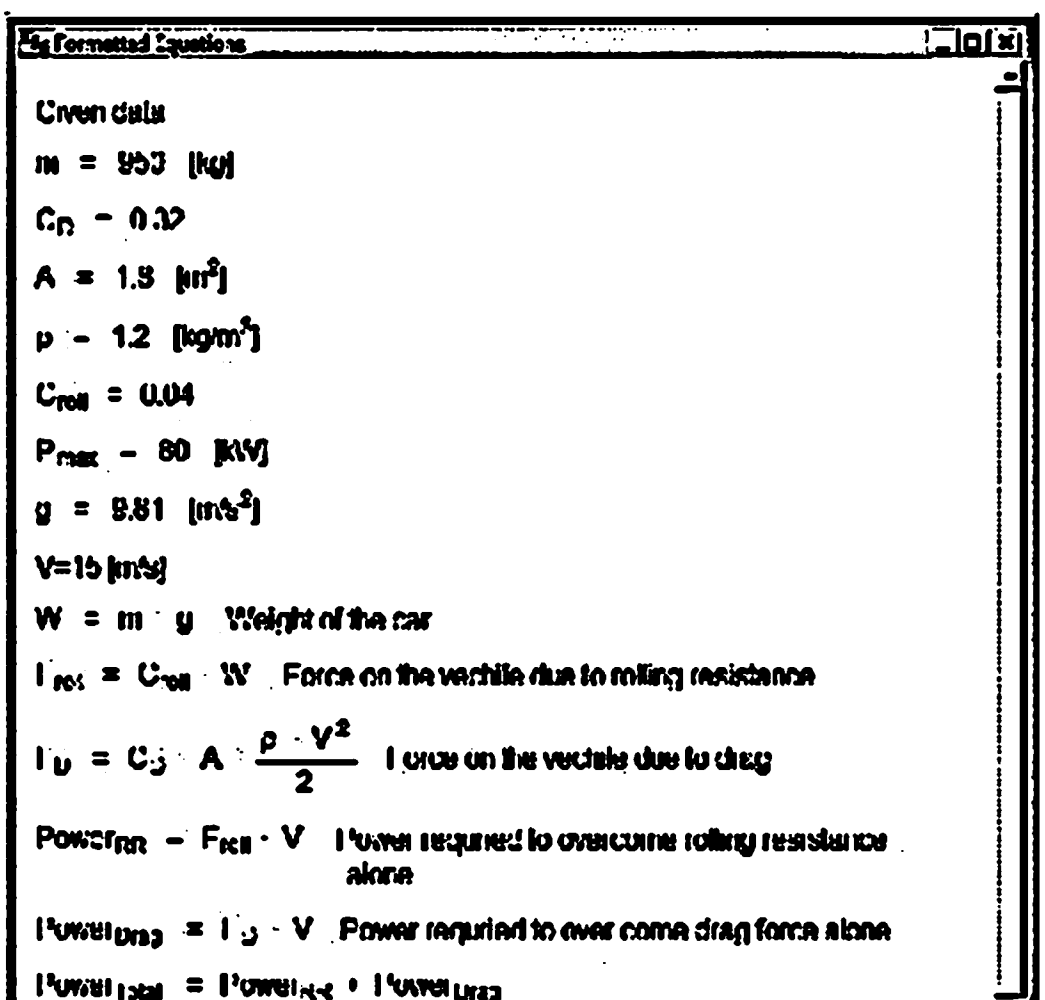
This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,



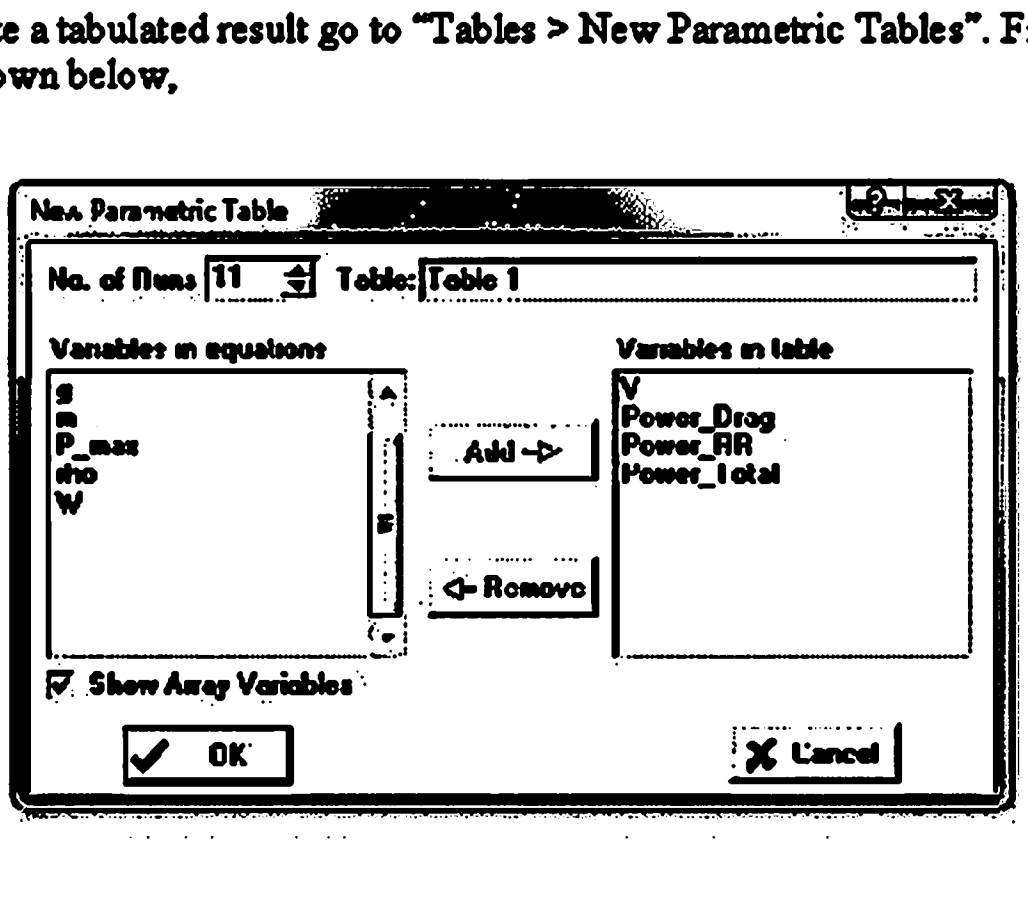
Step 3 of 5

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below ,



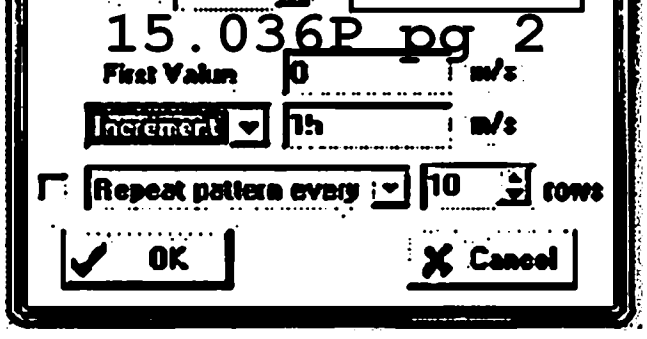
Step 4 of 5

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,

Step 5 of 5



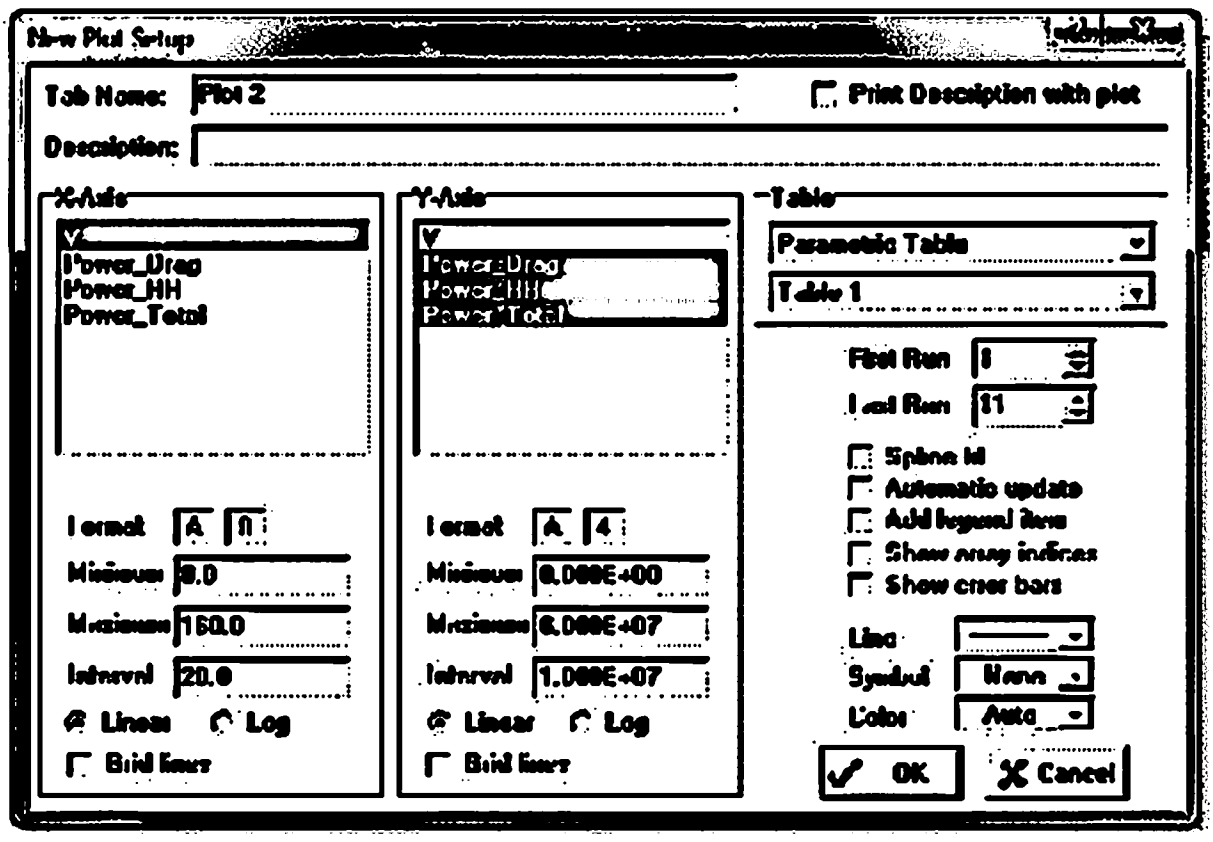
Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

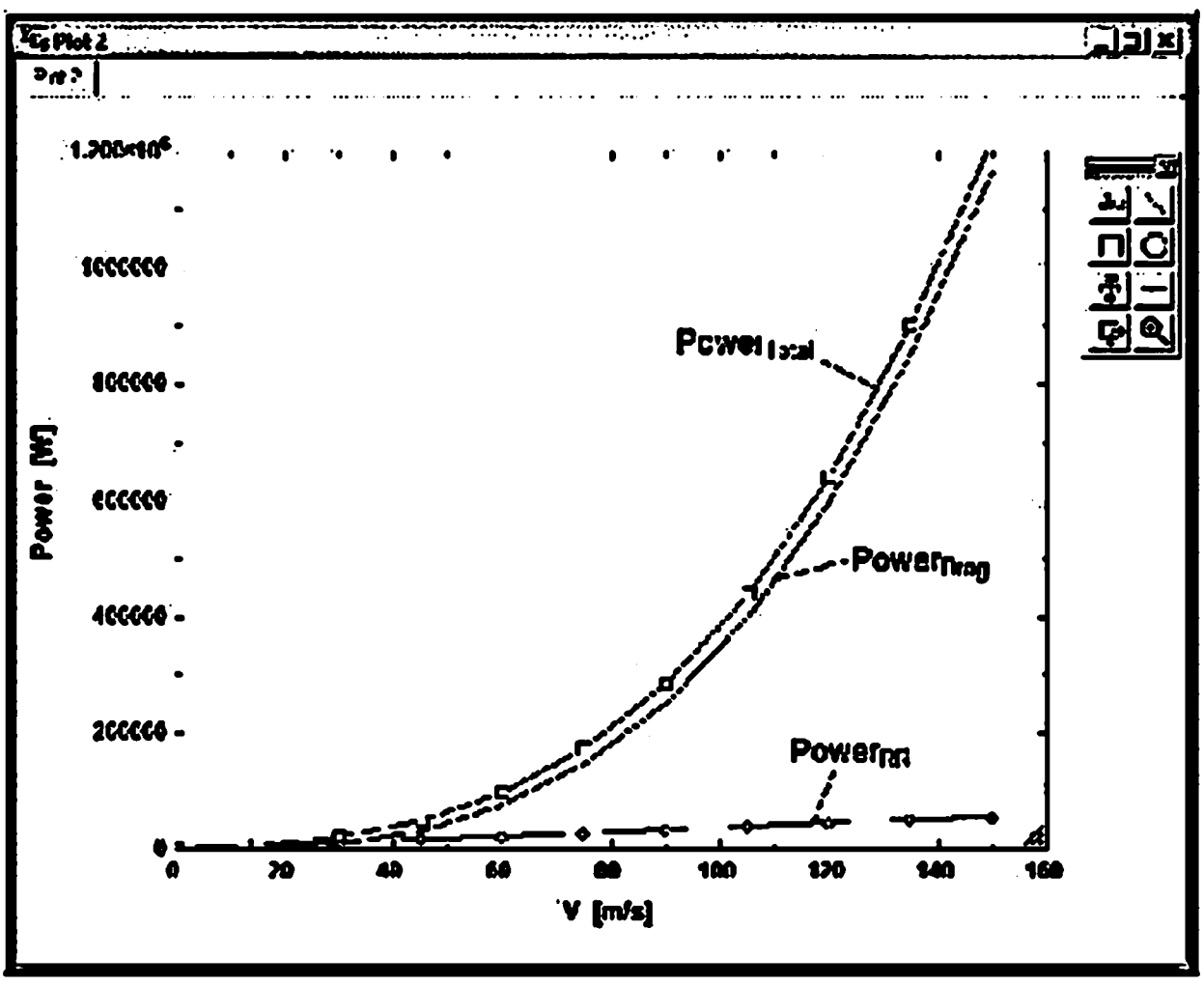
Table 1

	V [m/s]	$Power_{Drag}$ [W]	$Power_{RR}$ [W]	$Power_{Total}$
Run 1	0	0	0	0
Run 2	15	1166	5592	6758
Run 3	30	9331	11183	20515
Run 4	45	31493	16775	48268
Run 5	60	74650	22367	97016
Run 6	75	145800	27959	173759
Run 7	90	251942	33550	285493
Run 8	105	400075	39142	439217
Run 9	120	597197	44734	641930
Run 10	135	850306	50325	900631
Run 11	150	1.165E+06	55917	1.222E+06

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



By clicking on OK the graph will be generated as shown,



Question: 15.037P

A submarine can be treated as an ellipsoid with a diameter of 5 m and a length of 25 m. Determine the power required for this submarine to cruise horizontally and steadily at 65 km/h in seawater whose density is 1025 kg/m3. Also determine the power required to tow this submarine in air whose density is 1.30 kg/m3. Assume the flow is turbulent in both cases.

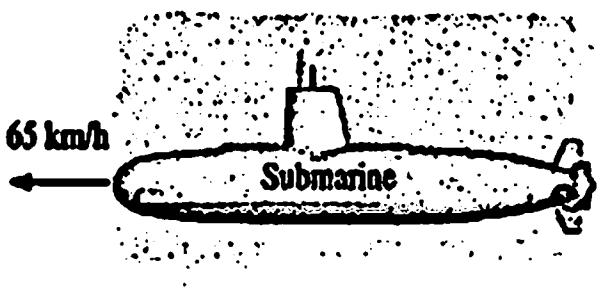


FIGURE P15–37

Step 1 of 5

Given data:

The diameter of the ellipsoid, $D = 5\ \text{m}$

The length of the submarine, $L = 25\ \text{m}$

The velocity of the submarine, $V = 65\ \text{km/h} = 18.05\ \text{m/s}$

The density of sea water, $\rho_{water} = 1025\ \text{kg/m}^3$

The density of air, $\rho_{air} = 1.30\ \text{kg/m}^3$

Drag coefficient of an ellipsoid with $\frac{L}{D} = \frac{25}{5} = 5$ is $C_D = 0.1$

Step 2 of 5

Equation for drag force,

$$F_D = \frac{1}{2} C_D \rho_{water} A V^2$$

Where F_D is the force acting on body

A is the frontal area,

ρ_{water} is the density of water,

V is the velocity of submarine,

C_D is the coefficient of drag

Frontal area of submarine,

$$A = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(5)^2$$

$$= 19.625\ \text{m}^2$$

Step 3 of 5

Substituting the values, we get

$$F_D = \frac{1}{2} \times 0.1 \times 1025 \times 19.625 \times (18.05)^2$$

$$= 327686.05\ \text{N}$$

Now, the power required to overcome this drag,

$P = F_D \times V$,

Where, F_D is the drag force and

V is the velocity.

Substituting the values,

$$P = 327686.05(18.05)$$

$$= 5914733.20\ \text{Watts}$$

$$= 5914.73\ \text{kW}$$

Therefore, the power required by the submarine to cruise horizontally and steadily at 65 km/h in sea water is $\boxed{P = 5914.73\ \text{kW}}$

Step 4 of 5

Drag force acting on the submarine when it is cruising through air,

$$F_D = \frac{1}{2} C_D A \rho_{air} V^2$$

Where C_D is the drag coefficient,

A is the frontal area of the ellipsoid,

V is the velocity of the submarine and

ρ_{air} is the density of air.

Substituting the values,

$$F_D = \frac{1}{2} \times 0.1 \times 19.625 \times 1.30 \times (18.05)^2$$

$$= 415.60\ \text{N}$$

Step 5 of 5

Now, the power required to overcome this drag,

$P = F_D \times V$

Where, F_D is the drag force and

V is the velocity.

Substituting the values,

$$P = 415.60(18.05)$$

$$= 7501.58\ \text{Watts}$$

$$= 7.5\ \text{kW}$$

Therefore, the power required by the submarine to cruise horizontally and steadily at 65 km/h through air is $\boxed{P = 7.5\ \text{kW}}$

Question: 15.038P

An 0.90-m-diameter, 1.1-m-high garbage can is found in the morning tipped over due to high winds during the night. Assuming the average density of the garbage inside to be 150 kg/m3 and taking the air density to be 1.25 kg/m3, estimate the wind velocity during the night wh.en the can was tipped over. Take the drag coefficient of the can to be 0.7.

Step 1 of 7

Given data:

The diameter of the garbage is $D = 0.90$ m

The height of the garbage is $H = 1.1$ m

The density of garbage is $\rho_g = 150\ \text{kg/m}^3$

The density of air is $\rho_{air} = 1.25\ \text{kg/m}^3$

Drag coefficient is $C_D = 0.7$

Step 2 of 7

Volume of the garbage can,

$$V = \frac{\pi}{4}D^2H$$

$$= \frac{\pi}{4}(0.90)^2(1.1)$$

$$= 0.7\ \text{m}^3$$

Now, weight of the garbage can,

$$W = mg \qquad \ldots\ldots (1)$$

But we know that $\text{Density}(\rho) = \dfrac{\text{Mass}(m)}{\text{Volume}(V)}$

Therefore, $m = V\rho_g$

Substituting the value of mass in equation (1)

$$W = V\rho_g g$$

$$= 0.7 \times 150 \times 9.81$$

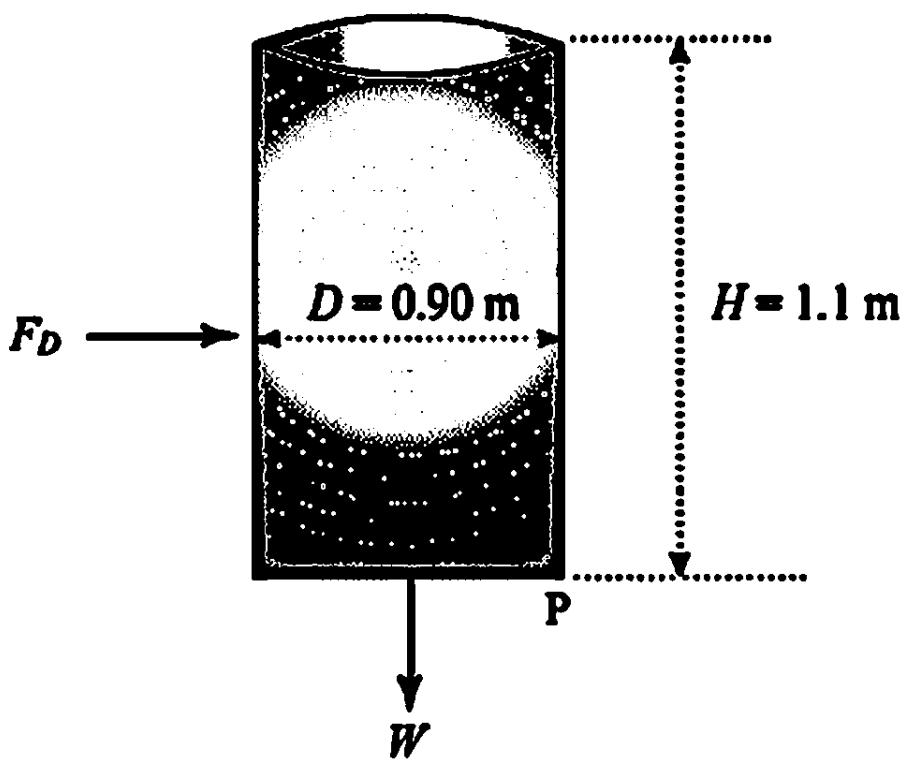
$$= 1030.05\ \text{N}$$

Step 3 of 7

When the garbage can is first tipped the wind loaded side will be off the ground first. Therefore, all the reactions will act on the other edge.
Now, we have to find out the drag force that caused the garbage to tip over.
For that consider an axis passing through the contact point and setting the moment about that point equal to zero will give the required drag force.

Step 4 of 7

Draw the garbage can to show the forces acting on the garbage can.



Step 5 of 7

Now, take moment about the contact point P

$$+\circlearrowleft \sum M_p = 0$$

$$F_D\left(\frac{H}{2}\right) - W\left(\frac{D}{2}\right) = 0$$

$$F_D\left(\frac{1.1}{2}\right) - 1030.05\left(\frac{0.9}{2}\right) = 0$$

$$F_D(0.55) = 463.52$$

$$F_D = \frac{463.52}{0.55}$$

$$= 842.76\ \text{N}$$

Step 6 of 7

Now the wind velocity that will cause this drag force can be determined by the equation given below.

$$F_D = \frac{1}{2}C_D\rho_{air}AV^2 \qquad \ldots\ldots (2)$$

Here, C_D is the drag coefficient.

ρ_{air} density of air

A is the frontal area

V is the velocity of air

Here frontal area,

$$A = DH$$

$$= 0.9 \times 1.1$$

$$= 0.99\ \text{m}^2$$

Step 7 of 7

Substituting the values of F_D C_D, ρ_{air}, and A in equation (2)

$$842.76 = \frac{1}{2} \times 0.7 \times 1.25 \times 0.99 \times V^2$$

$$V^2 = 1945.77$$

$$V = 44.11\ \text{m/s}$$

$$= 44.11\left(\frac{18}{5}\right)$$

$$= 158.8$$

$$= 159\ \text{km/h}$$

Therefore, the wind velocity during the night when the can was tipped over was,

$$\boxed{V = 159\ \text{km/h}}$$

Question: 15.039P pg 1

To reduce the drag coefficient and thus to improve the fuel efficiency of cars, the design of side rearview mirrors has changed drastically in recent decades from a simple circular plate to a streamlined shape. Determine the amount of fuel and money saved per year as a result of replacing a 13- cm-diameter flat mirror by one with a hemispherical back, as shown in the figure. Assume the car is driven 24,000 km a year at an average speed of 95 km/h. Take the density and price of gasoline to be 0.75 kg/L and $0.90/L, respectively; the heating value of gasoline to be 44,000 kJ/kg; and the overall efficiency of the engine to be 30 percent.

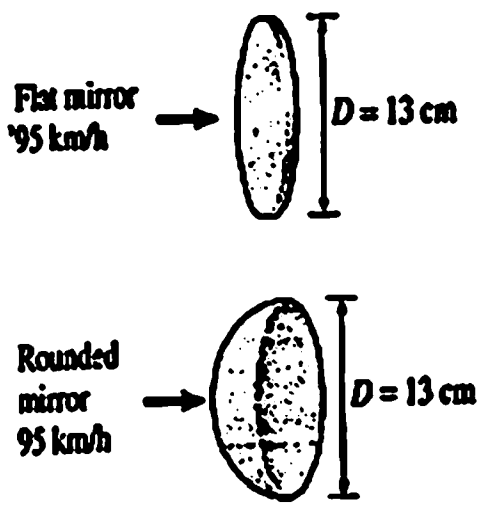


FIGURE P15–39

Step 1 of 8

Given data:

The diameter of the flat mirror, $D = 0.13\text{m}$

The diameter of the rounded mirror, $D = 0.13\ \text{m}$

Fuel costs, $0.90 per liter

Overall efficiency of the car is $\eta = 30\%$

The density of the fuel, $\rho_{fuel} = 0.75\ \text{kg/L}$

The average velocity of the car, $V = 95\ \text{km/h}$

Heating value of the fuel, $HV = 44{,}000\ \text{kJ/kg}$

The density of air can be taken as, $\rho_{air} = 1.25\ \text{kg/m}^3$

Total distance covered by the car in one year, $d = 24{,}000\ \text{km/yr}$

Step 2 of 8

The drag force acting on the flat mirror can be calculated by using the formula,

$$F_D = \frac{1}{2} C_D A \rho V^2$$

Where, C_D is the drag coefficient ($C_D = 1.1$)

ρ is the density of air

A is the frontal area

V is the velocity of wind

Now, the frontal area, is given by

$$A = \frac{\pi D^2}{4}$$

$$= \frac{\pi(0.13)^2}{4}$$

$$= 0.0133\ \text{m}^2$$

The velocity, is given by

$$V = 95\left(\frac{5}{18}\right)$$

$$= 26.4\ \text{m/s}$$

Step 3 of 8

By substituting the values in drag force, we get

$$F_D = \frac{1}{2} \times 1.1 \times 0.0133 \times 1.25 \times (26.4)^2$$

$$= 6.37\ \text{N}$$

Now, the work required to overcome this drag is

$$W = F_D \times d$$

$$= (6.37\ \text{N}) \times (24000\ \text{km/yr})$$

$$= 152880\ \text{kJ/yr}$$

Now, required energy input for a travelling 24000 km is,

$$E_{in} = \frac{W}{\eta}$$

$$= \frac{152880}{0.3}$$

$$= 509600\ \text{kJ/yr}$$

Step 4 of 8

The amount of fuel required to develop this energy,

$$f_1 = \frac{E_{in}}{HV(\rho_{fuel})}$$

$$= \frac{509600}{44000(0.75)}$$

$$= 15.44\ \text{L/yr}$$

The cost of the fuel required is

C_1 = Amount of fuel × Unit cost of fuel

$$= 15.44 \times 0.90$$

$$= \$13.9$$

Step 5 of 8

The drag force acting on the rounded mirror can be calculated by using the formula,

$$F_D = \frac{1}{2} C_D A \rho V^2$$

Where, C_D is the drag coefficient ($C_D = 0.4$)

ρ is the density of air

A is the frontal area

V is the velocity of wind

Now, the frontal area, is given by

$$A = \frac{\pi D^2}{4}$$

$$= \frac{\pi(0.13)^2}{4}$$

V is the velocity of wind

Now, the frontal area, is given by

$$A = \frac{\pi D^2}{4}$$
$$= \frac{\pi(0.13)^2}{4}$$
$$= 0.0133\ \text{m}^2$$

Step 6 of 8

By substituting the values in drag force, we get

$$F_D = \frac{1}{2} \times 0.4 \times 0.0133 \times 1.25 \times (26.4)^2$$
$$= 2.32\ \text{N}$$

Now, the work required to overcome this drag is

$$W = F_D \times d$$
$$= (2.32\ \text{N}) \times (24000\ \text{km/yr})$$
$$= 55680\ \text{kJ/yr}$$

Now, required energy input for a travelling 24000 km is,

$$E_{in} = \frac{W}{\eta}$$
$$= \frac{55680}{0.3}$$
$$= 185600\ \text{kJ/yr}$$

Step 7 of 8

The amount of fuel required to develop this energy,

$$f_2 = \frac{E_{in}}{HV(\rho_{fuel})}$$
$$= \frac{185600}{44000(0.75)}$$
$$= 5.62\ \text{L/yr}$$

The cost of the fuel required is

C_1 = Amount of fuel × Unit cost of fuel
$$= 5.62 \times 0.90$$
$$= \$5.06$$

Step 8 of 8

The amount of fuel saved,

$$f = f_1 - f_2$$
$$= 15.44 - 5.62$$
$$= 9.82\ \text{L}$$

Hence the amount of fuel saved in a year is $\boxed{f = 9.82\ \text{L}}$

The amount of money saved,

$$C = C_1 - C_2$$
$$= 13.9 - 5.06$$
$$= \$\,8.84$$

Hence the amount of money saved in a year is $\boxed{C = \$\,8.84}$

Question: 15.040P

A 5-m-diameter hot air balloon that has a total mass of 230 kg is standing still in air on a windless day. The balloon is suddenly subjected to 40 km/h winds. Determine the initial acceleration of the balloon in the horizontal direction.

Step 1 of 4

Given data:-

Diameter of the balloon, $D = 5$ m

Mass of the balloon, $m = 230$ kg

Velocity of the wind, $V = 40$ km/h

Step 2 of 4

Now, the drag force acting on the balloon can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag coefficient

ρ is the density of air.

A is the frontal area of the balloon.

V is the velocity of the wind.

Frontal area of the parachute,

$$A = \frac{\pi}{4} D^2$$

Here, $D = 5$ m

Substituting the value in the above equation

$$A = \frac{\pi}{4}(5)^2$$

$$= 19.625 \text{ m}^2$$

Drag coefficient for the balloon is, $C_D = 0.1$

Step 3 of 4

From the table properties of air,

Density of air at 1atm pressure and 25°C is, $\rho = 1.184$ kg/m^3

Velocity of air,

$$V = 40 \text{ km/h}$$

$$= 40\left(\frac{5}{18}\right) \text{m/s}$$

$$= 11.11 \text{ m/s}$$

Now, substitute all the known values in equation (1)

$$F_D = \frac{1}{2} C_D \rho A V^2$$

$$= \frac{1}{2} \times 0.1 \times 1.184 \times 19.625 \times (11.11)^2$$

$$= 143.4 \text{ N}$$

Step 4 of 4

Now, we know that force is the product of mass and acceleration,

Therefore,

Therefore, Force,

$F = \text{mass} \times \text{acceleration}$ ..... (2)

Here mass is the mass of balloon, $m = 230$ kg

Force is the drag force acting on the balloon,

Now, substitute the known values in equation (2)

$$F = \text{mass} \times \text{acceleration}$$

$$143.4 = 230 \times a$$

$$a = \frac{143.4}{230}$$

$$= 0.62 \text{ m/s}^2$$

Therefore acceleration, $\boxed{a = 0.62 \text{ m/s}^2}$

Question: 15.041P

A 6-mm-diameter plastic sphere whose density is 1150 kg/m3 is dropped into water at 20°C. Determine the terminal velocity of the sphere in water.

Step 1 of 4

Given data:

The diameter of the plastic sphere, $D = 6$ mm

The density of the sphere, $\rho_{sphere} = 1150\ kg/m^3$

The temperature of water, $T = 20°C$

The density of water at 20°C, $\rho_{water} = 998\ kg/m^3$

The dynamic viscosity of water at 20°C, $\mu = 1.002 \times 10^{-3}$ kg/m-s

Drag coefficient of a sphere in laminar flow, $C_D = 0.5$

Step 2 of 4

The frontal area of the sphere,

$$A = \frac{\pi D^2}{4}$$

$$= \frac{\pi(0.006)^2}{4}$$

$$= 2.83 \times 10^{-5}\ m^2$$

The drag force acting on the sphere is

$$F_D = \frac{1}{2} C_D A \rho_{water} V^2$$

Where C_D is the drag coefficient

A is the frontal area of the car

V is the velocity of the car and

ρ_{water} is the density of water.

$$F_D = \frac{1}{2} C_D A \rho_{water} V^2$$

$$= \frac{1}{2} \times 0.5 \times 2.83\left(10^{-5}\right) \times 998 \times V^2$$

$$= 7.06 \times 10^{-3} \times V^2$$

Step 3 of 4

Now, weight of the body,

$$W = \rho_{sphere} g \forall$$

$$= \rho_{sphere} g \left(\frac{\pi D^3}{6}\right)$$

$$= 1150 \times 9.81 \times \frac{\pi(0.006)^3}{6}$$

$$= 0.00127\ N$$

Now, buoyant force,

$$F_B = \rho_{water} g \forall$$

$$= \rho_{water} g \left(\frac{\pi D^3}{6}\right)$$

$$= 998 \times 9.81 \times \frac{\pi(0.006)^3}{6}$$

$$= 0.0011\ N$$

Step 4 of 4

For a freely falling body terminal velocity is reached when the sum of the drag force (F_D) and buoyant force (F_B) acting on the body becomes equal to the weight of the body (W). Therefore,

$$W = F_D + F_B$$

$$0.00127 = \left(7.06 \times 10^{-3} \times V^2\right) + 0.0011$$

$$1.7 \times 10^{-4} = 7.06 \times 10^{-3} \times V^2$$

$$V = \sqrt{\frac{1.7 \times 10^{-4}}{7.06\left(10^{-3}\right)}}$$

$$= 0.155\ m/s$$

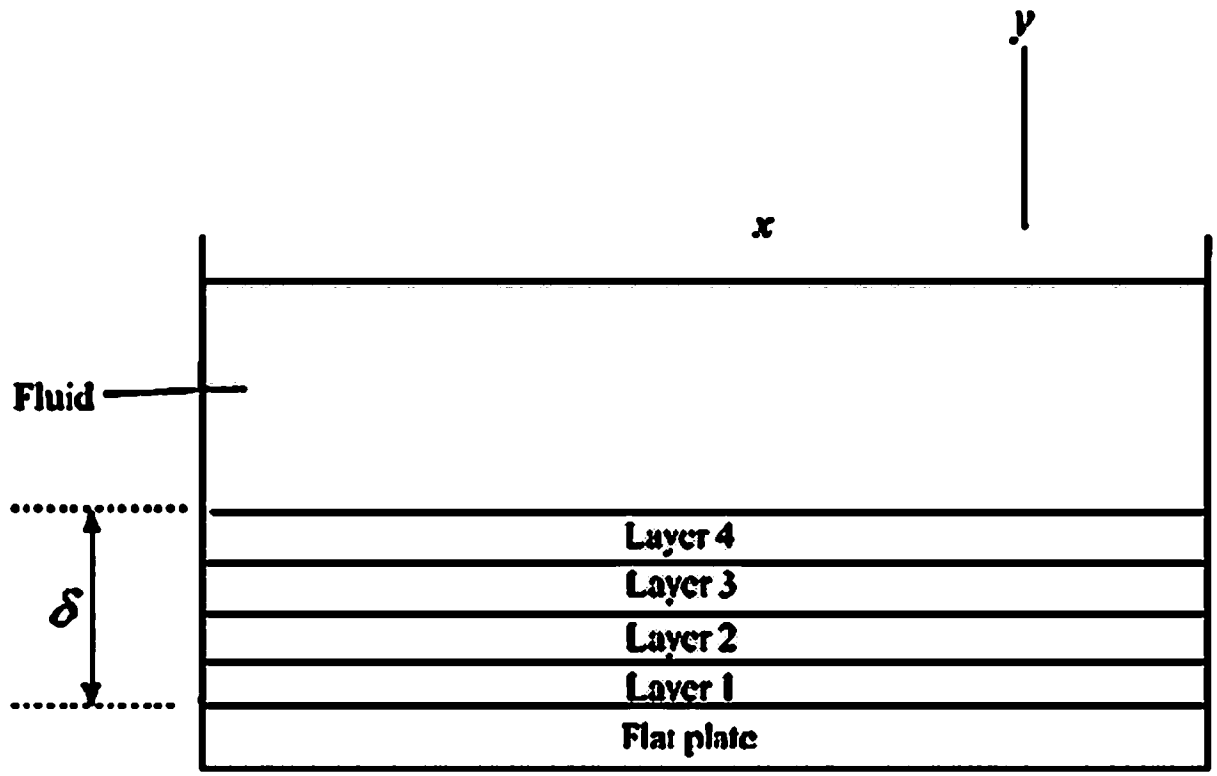
Therefore, the terminal velocity of the sphere in water is $\boxed{V = 0.155 m/s}$

Question: 15.042P

What does the friction coefficient represent in flow over a flat plate? How is it related to the drag force acting on the plate?

Step 1 of 2

Diagram for the boundary layer:



Step 2 of 2

Consider a fluid flow over a flat plate. The x coordinate is measured along the plate in the direction of flow and y coordinate is measured normal to the plate or perpendicular to the direction of flow.

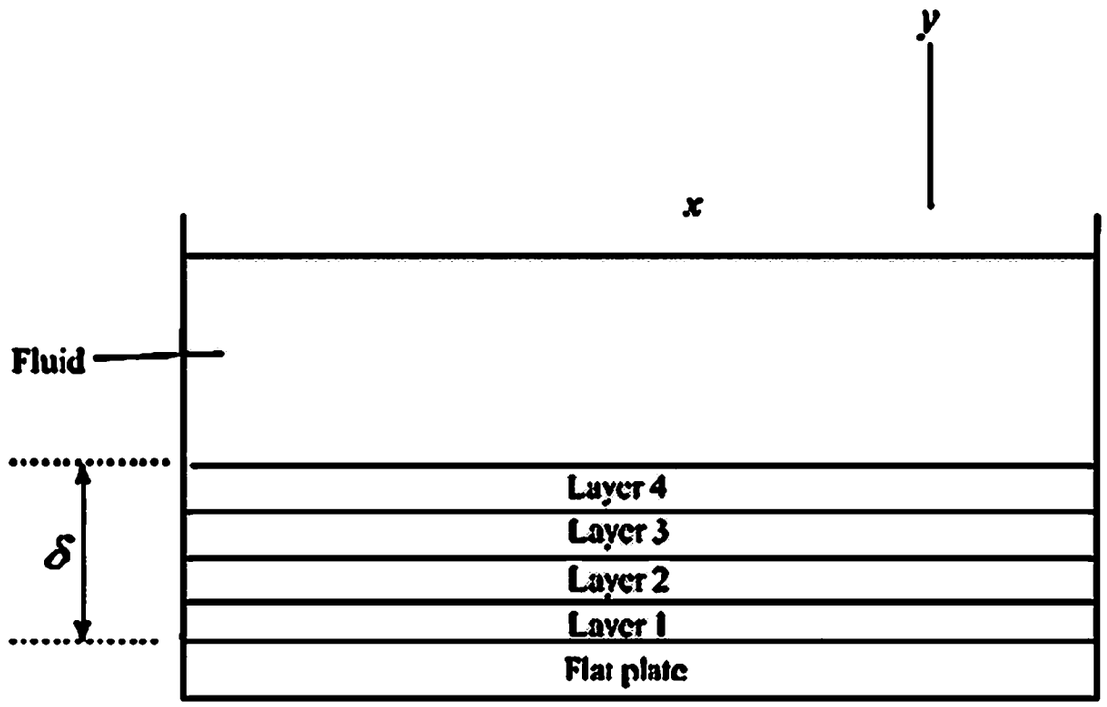
The fluid approaches the plate in the x direction with a uniform velocity of V. We are assuming that the fluid consist of adjacent layers one over the other, in the figure it is layer 1, layer 2, layer 3 etc. Each layer will offer a resistance to the flow of adjacent layer; this is due to the frictional resistance between the two layers. This frictional resistance of fluid is called viscosity.

Question: 15.043P

What fluid property is responsible for the development of the velocity boundary layer? What is the effect of the velocity on the thickness of the boundary layer?

Step 1 of 3

Diagram for the boundary layer:



Step 2 of 3

Consider a fluid flow over a flat plate. The x coordinate is measured along the plate in the direction of flow and y coordinate is measured normal to the plate or perpendicular to the direction of flow.

The fluid approaches the plate in the x direction with a uniform velocity of V. We are assuming that the fluid consist of adjacent layers one over the other, in the figure it is layer 1, layer 2, layer 3 etc. Each layer will offer a resistance to the flow of adjacent layer; this is due to the frictional resistance between the two layers. This frictional resistance of fluid is called viscosity.

Step 3 of 3

The velocity of the particles in the first layer (layer 1) is zero because of the no slip condition. This motionless layer will slow down the particles of the neighboring fluid layer (layer 2) due to the frictional resistance between the two layers. Now, the layer 2 will offer frictional resistance to layer 3 and so on. This will continue up to a distance of δ from the plate surface beyond that distance the free stream velocity virtually remain unchanged, in other word beyond that distance the fluid will flow at a velocity of V (typically $0.99V$). As a result the x component (velocity) varies from zero to V (typically $0.99V$).

The region above the plate up to distance δ is called velocity boundary layer. From the above explanation we can understand that viscosity is the fluid property responsible for the development of boundary layer. Velocity brings boundary layer closer to the plate surface, so higher the velocity lower will be the thickness of the boundary layer and vise versa.

Question: 15.044P

How is the average friction coefficient determined in flow over a flat plate?

Step 1 of 2

One of he ways of determining the average friction coefficient is by dividing the drag force by dynamic pressure. For this we should first determine the drag force acting on the body experimentally and the area of the surface through which the fluid is flowing, the density of the fluid and the velocity of the fluid flow also should be known.

Coefficient of friction, $C_f = \dfrac{F_D}{\frac{1}{2}\rho A V^2}$

Where, $\frac{1}{2}\rho A V^2$ is the dynamic pressure.

Here, F_D is the drag force,

ρ is the density,

A is the area of the plate through which fluid is flowing and

V is the velocity of flow.

Step 2 of 2

Now, we can determine the average friction coefficient by another method, for this we have to know the local friction coefficient $\left(C_{f,x}\right)$ which is the local value of friction coefficient at a particular region in the plate. The local friction coefficient varies along the surface of the flat plate due to the change in velocity boundary layer in the flow direction so the values of local friction coefficient will be different in different regions of the plate.

To determine the average friction coefficient for the entire plate we should integrate the local friction coefficient for the entire length L and divide it by the length L.

$$C_f = \frac{1}{L}\int_0^L C_{f,x}\,dx$$

For laminar flow, $C_{f,x} = \dfrac{0.664}{\mathrm{Re}_x^{0.5}}$

Where, Re_x is the Reynolds number at a distance x from the leading edge of the plate.

$\mathrm{Re}_x = \dfrac{Vx}{\nu}$. Also, $\mathrm{Re}_x \le 5 \times 10^5$

V is the velocity of fluid, x is the length of the plate and ν is the kinematic viscosity of the fluid.

Question: 15.045P

Light oil at 75°F flows over a 22-ft-long flat plate with a free-stream velocity of 6 ft/s. Determine the total drag force per unit width of the plate.

Step 1 of 3

The total drag force acting per unit width can be calculated using the formula given below.

Drag force, $F_D = \frac{1}{2} C_f \rho A V^2$ ..... (1)

Where,

The density of air be ρ

The velocity of flow be V

The coefficient frictional be C_f

The area of the plate through which air is flowing be A

Step 2 of 3

Since the value of Reynolds number is unknown we have to calculate the Reynolds number.

To calculate the Reynolds number we can use the formula given below.

$$\mathrm{Re}_L = \frac{VL}{\nu} \text{ ..... (2)}$$

Here,

The velocity of flow be V

The kinematic viscosity be ν

The length of the plate be L

Substituting the values of V, L and ν in equation (2), we get

$$\mathrm{Re}_L = \frac{6(22)}{7.751 \times 10^{-3}}$$
$$= 17030.06$$

We can calculate coefficient of friction using the formula given below,

$$C_f = \frac{1.33}{\mathrm{Re}_L^{0.5}}$$
$$= \frac{1.33}{(17030.06)^{0.5}}$$
$$= 0.01019$$

Step 3 of 3

Now, we have to calculate the area of the surface of the flat plate through which oil is flowing A

Area of the surface through which oil is flowing,

$$A = L \times w$$

Here, L is the length of the plate and w is the width of the plate.

$$A = 22 \times 1$$
$$= 22 \text{ ft}^2$$

Substituting the values V, A, ρ and c_f in the equation (1), we get

$$F_D = \frac{1}{2} C_f \rho A V^2$$
$$= \frac{1}{2} \times 0.01019 \times 55.3 \times 22 \times 6^2$$
$$= 223.14 \text{ lbf}$$

Therefore, the total drag force acting per unit width of the plate is $\boxed{F_D = 223.14 \text{ lbf}}$

Question: 15.046P pg 1

The local atmospheric pressure in Denver, Colorado (elevation 1610 m) is 83.4 kPa. Air at this pressure and at 25°C flows with a velocity of 6 m/s over a 2.5-m × 8-m flat plate. Determine the drag force acting on the top surface of the plate if the air flows parallel to the (*a*) 8-m-long side and (*b*) the 2.5-m-long side.

Step 1 of 8

Given data:

Pressure of the air is $P = 83.4\ \text{kPa}$.

Temperature of air is $T = 25°\text{C}$.

Velocity of flow is $V = 6\ \text{m/s}$.

Dimensions of the flat plate is $2.5\ \text{m} \times 8\ \text{m}$.

Area of the flat plate is $A = 20\ \text{m}^2$.

Step 2 of 8

a) Air is flowing parallel to 8m long side.

Dynamic viscosity of air at $25°\text{C}$ is $\mu = 1.849 \times 10^{-5}\ \text{kg/m·s}$

Density of air at temperature, $T = 25°\text{C}$ and pressure, $P = 83.4\ \text{kPa}$

$$\rho = \frac{P}{RT} \quad \ldots\ldots (1)$$

Here, P is the pressure, T is the temperature and R is the gas constant.

$$T = 273 + 25$$
$$= 298\ \text{K}$$

$$R = 287.058\ \text{J/kg K}$$
$$= 0.287\ \text{kJ/kg K}$$

Substituting the values of P, R and T in equation (1).

$$\rho = \frac{83.4}{0.287 \times 298}$$
$$= 0.975\ \text{kg/m}^3$$

Step 3 of 8

Reynolds number is $\text{Re}_L = \frac{\rho V L}{\mu}$ ..... (2)

Where, ρ is the density of air.

V is the velocity of flow.

L is the length of the plate.

μ is the kinematic viscosity.

Here, length $L = 8\ \text{m}$

Substitute these values in equation (2).

$$\text{Re}_L = \frac{\rho V L}{\mu}$$
$$= \frac{0.975 \times 6 \times 8}{1.849 \times 10^{-5}}$$
$$= 2.531 \times 10^6$$

Step 4 of 8

We know that the critical value of Reynolds number is $\text{Re}_{cr} = 5 \times 10^5$. Since the obtained value of Reynolds number is greater then the critical value the flow will be a combined laminar and turbulent flow.

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$C_f = \frac{0.074}{\text{Re}_L^{1/5}} - \frac{1742}{\text{Re}_L}$$
$$= \frac{0.074}{\left(2.531 \times 10^6\right)^{1/5}} - \frac{1742}{2.531 \times 10^6}$$
$$= 0.0039 - 0.000688$$
$$= 0.0032$$

Step 5 of 8

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2 \quad \ldots\ldots (3)$$

Where, ρ is the density of air.

A is the area of the flat plate through which air is flowing.

V is the velocity of flow.

Now, substitute the value of ρ, A, V, and C_f in equation (3).

$$F_D = \frac{1}{2} \times 0.0032 \times 0.975 \times 20 \times (6)^2$$
$$= 1.12\ \text{N}$$

Therefore, the drag force acting on the flat plate when air flows parallel to 8 m long side, $\boxed{F_D = 1.12\ \text{N}}$.

Step 6 of 8

b) Air is flowing parallel to 2.5m long side.

Reynolds number, $\text{Re}_L = \frac{\rho V L}{\mu}$

Step 6 of 8

b) Air is flowing parallel to 2.5m long side.

Reynolds number, $\mathrm{Re}_L = \dfrac{\rho V L}{\mu}$

Where, ρ is the density of air.

V is the velocity of flow.

L is the length of the plate.

μ is the kinematic viscosity.

Here, length $L = 2.5\ \mathrm{m}$

Substituting these values in the above equation.

$$\begin{aligned}\mathrm{Re}_L &= \frac{\rho V L}{\mu}\\ &= \frac{0.975 \times 6 \times 2.5}{1.849 \times 10^{-5}}\\ &= 7.9 \times 10^5\end{aligned}$$

We know that the critical value of Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$.

Since the obtained value of Reynolds number is greater then the critical value the flow will be a combined laminar and turbulent flow.

Step 7 of 8

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$\begin{aligned}C_f &= \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L}\\ &= \frac{0.074}{\left(7.9 \times 10^5\right)^{1/5}} - \frac{1742}{7.9 \times 10^5}\\ &= 0.00489 - 0.0022\\ &= 0.00269\end{aligned}$$

Step 8 of 8

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2$$

Were, ρ is the density of air.

A is the area of the flat plate through which air is flowing.

V is the velocity of flow.

Substituting the value of ρ, A, V, and C_f in the above equation.

$$\begin{aligned}F_D &= \frac{1}{2} \times 0.00269 \times 0.975 \times 20 \times (6)^2\\ &= 0.944\ \mathrm{N}\end{aligned}$$

Therefore, the drag force acting on the flat plate when air flows parallel to 2.5 m long side,

$\boxed{F_D = 0.944\ \mathrm{N}}$.

Question: 15.047P pg 1

During a winter day, wind at 55 km/h, 5°C, and .1 atmis blowing parallel to a 4-m-high and 10-m-long wall of a house. Approximating the wall surfaces as smooth, determine the friction drag acting on the wall. What would your answer be if the wind velocity has doubled? How realistic is it to treat the flow over side wall surfaces as flow over a flat plat?

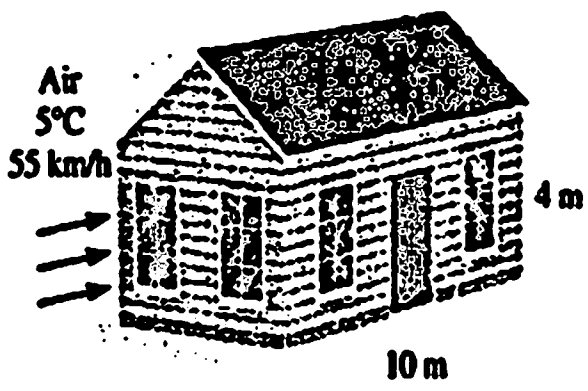


FIGURE P15–47

Step 1 of 7

Given data:

Velocity of wind is $V = 55\ \text{km/h}$.

Temperature of the air is $T = 5°\text{C}$.

Pressure of the air is $P = 1\ \text{atm}$.

Height of the wall is $h = 4\ \text{m}$.

Length of the wall is $L = 10\ \text{m}$.

Step 2 of 7

Density of air at $5°\text{C}$ and 1 atm pressure, $\rho = 1.269\ \text{kg/m}^3$

Kinematic viscosity of air at $5°\text{C}$ and 1 atm pressure, $\nu = 1.382 \times 10^{-5}\ \text{m}^2/\text{s}$

Now, the Reynolds number at the end of the wall that means at a length of 10 m of the wall,

$$\text{Re}_L = \frac{VL}{\nu} \quad \ldots\ldots (1)$$

Where, V is the velocity of flow.

L is the length of the wall.

ν is the kinematic viscosity of fluid.

Step 3 of 7

Converting velocity

$$V = 55\left(\frac{5}{18}\right)\text{m/s}$$
$$= 15.3\,\text{m/s}$$

Substituting the values of V, L and ν in equation (1).

$$\text{Re}_L = \frac{VL}{\nu}$$
$$= \frac{15.3 \times 10}{1.382 \times 10^{-5}}$$
$$= 1.1 \times 10^7$$

Step 4 of 7

We know that the critical value of Reynolds number is $\text{Re}_{cr} = 5 \times 10^5$. Since the obtained value of Reynolds number is greater then the critical value the flow will be a combined laminar and turbulent flow.

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$C_f = \frac{0.074}{\text{Re}_L^{1/5}} - \frac{1742}{\text{Re}_L}$$
$$= \frac{0.074}{\left(1.1 \times 10^7\right)^{1/5}} - \frac{1742}{1.1 \times 10^7}$$
$$= 0.0029 - 0.00016$$
$$= 0.00274$$

Step 5 of 7

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2 \quad \ldots\ldots (2)$$

Were, ρ is the density of air.

A is the area of the wall through which air is flowing.

V is the velocity of flow.

Here, area of the wall through which air is flowing,

$$A = L \times h$$
$$= 10 \times 4$$
$$= 40\ \text{m}^2$$

Then, substituting the value of ρ, A, V, and C_f in equation (2).

$$F_D = \frac{1}{2} \times 0.00274 \times 1.269 \times 40 \times (15.3)^2$$
$$= 16.3\ \text{N}$$

Step 6 of 7

Now, the velocity is doubled so the new velocity is, $V = 110\ \text{km/h}$

Step 6 of 7

Now, the velocity is doubled so the new velocity is, $V = 110\ \text{km/h}$

$$V = 110\left(\frac{5}{18}\right)\text{m/s}$$
$$= 30.55\,\text{m/s}$$

The new Reynolds number is,

$$\text{Re}_L = \frac{VL}{\nu}$$
$$= \frac{30.55 \times 10}{1.382 \times 10^{-5}}$$
$$= 2.21 \times 10^7$$

We know that the critical value of Reynolds number is $\text{Re}_{cr} = 5 \times 10^5$. Since the obtained value of Reynolds number is greater then the critical value the flow will be a combined laminar and turbulent flow.

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$C_f = \frac{0.074}{\text{Re}_L^{1/5}} - \frac{1742}{\text{Re}_L}$$
$$= \frac{0.074}{\left(2.21 \times 10^7\right)^{1/5}} - \frac{1742}{2.21 \times 10^7}$$
$$= 0.0025 - 0.0000788$$
$$= 0.0024$$

Step 7 of 7

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2$$

Were, ρ is the density of air.

A is the area of the wall through which air is flowing.

V is the velocity of flow.

Then, substituting the value of ρ, A, V, and C_f in the above equation.

$$F_D = \frac{1}{2} \times 0.0024 \times 1.269 \times 40 \times (30.55)^2$$
$$= 57\,\text{N}$$

Therefore, the drag force acting on the wall when velocity is $55\,\text{km/h}$ is $\boxed{F_D = 16.3\ \text{N}}$, and the drag force acting when the velocity is $110\,\text{km/h}$ is $\boxed{F_D = 57\ \text{N}}$.

Question: 15.048P

Air at 70°F flows over a 10-ft-long flat plate ɯlɯ at 25 ft/s. Determine the local friction coefficient at intervals of 1 ft and plot the results against the distance from the leading edge.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

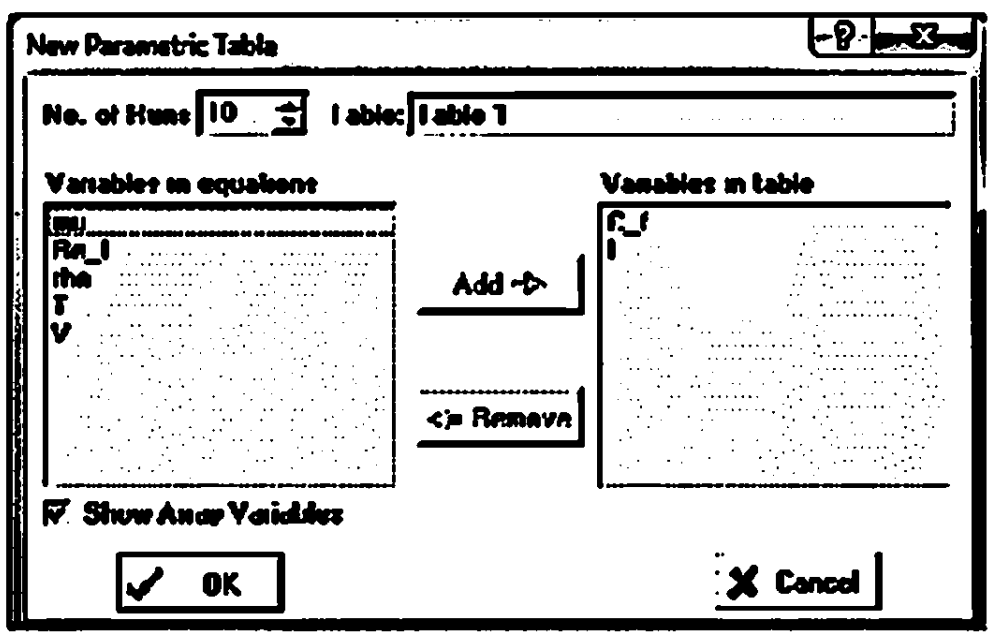
```
Function IFTEST(X, Y)
if(X<Y) Then
A:=0.664/X^(1/2)
Else
A:=0.059/(X)^(1/5)
EndIf
IFTEST:=A
End

T=70[F]                                   "Temperature of Air"
"L is varied from 0 to 10[ft]"            "Lenght of the plate"
V=25[ft/s]                                "Velocity of air"

"Properties of air"
rho=Density(Air,T=T,P=1[atm]*convert(atm,psia))
mu=Viscosity(Air,T=T)*convert(lb_m/ft hr, lb_m/ft s)
Re_L=rho*V*L/mu                           "Reynolds number"
C_f=IFTEST(Re_L,500000)                   "Friction Coefficient"
```

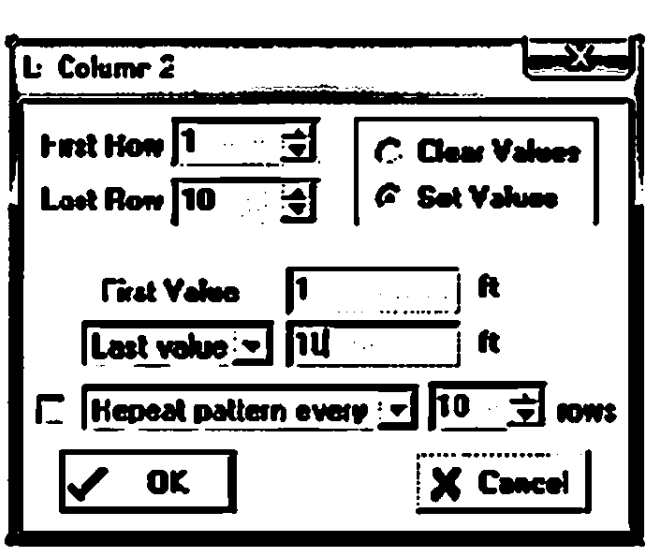
Step 2 of 5

Now to solve the equation for a range of length of the air, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 10.
Select variables in equations as "C_f", and "L" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

The length of the plate is varied from 0 to 10 ft
To enter the values in the "L" column, click on the drop tab in "L" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

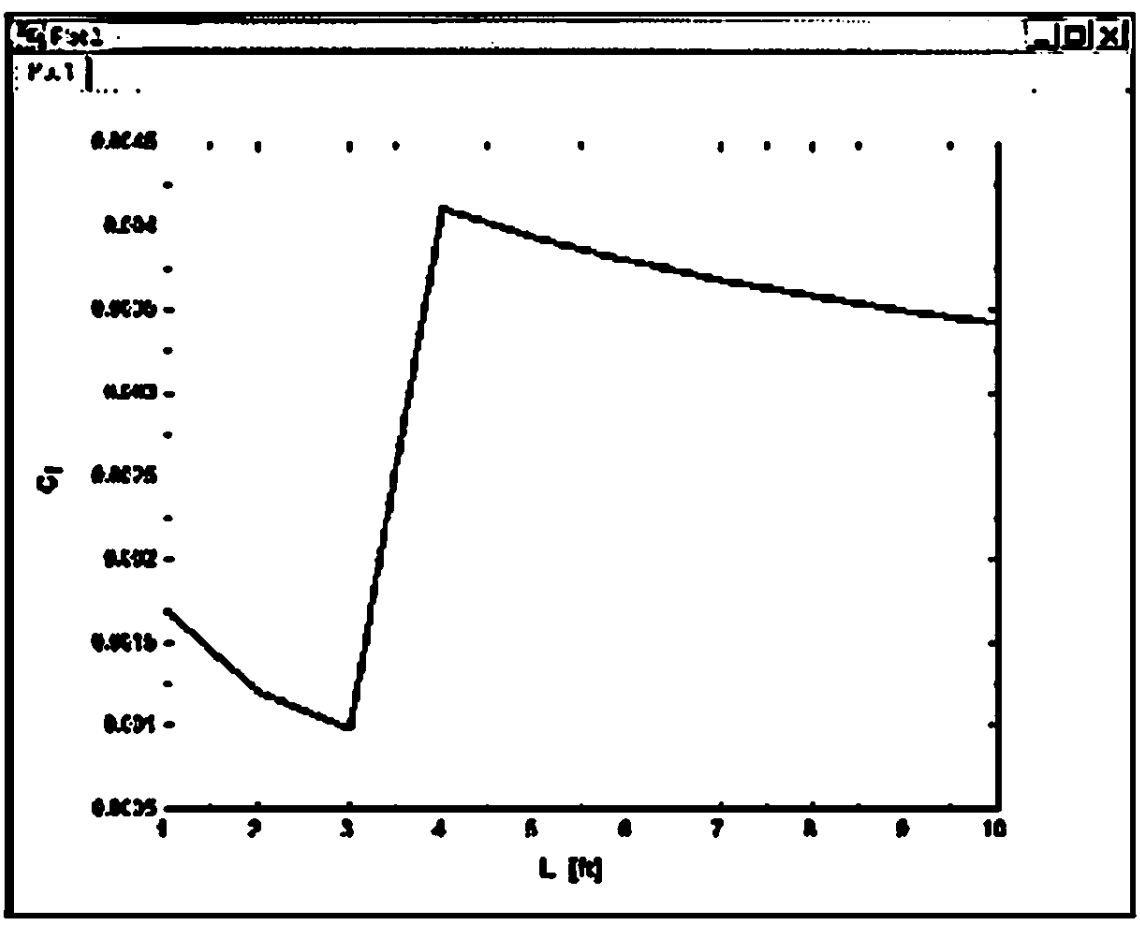
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..10	C_f	L [ft]
Run 1	0.001702	1
Run 2	0.001203	2
Run 3	0.0009826	3
Run 4	0.004111	4
Run 5	0.003932	5
Run 6	0.003791	6
Run 7	0.003676	7
Run 8	0.003579	8
Run 9	0.003496	9
Run 10	0.003423	10

Step 5 of 5

To plot the coefficient of friction as a function of length of the plate, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the coefficient of friction as a function of length of the plate is shown:



Question: 15.049P

Consider laminar flow of a fluid over a flat plate. Now the free-stream velocity of the fluid is doubled. Determine the change in the drag force on the plate. Assume the flow to remain laminar.

Step 1 of 3

For a laminar fluid drag force is given by,

$$F_{D_1} = \frac{1}{2} C_f \rho A V^2 \text{ ..... (1)}$$

Where, C_f is the frictional coefficient.

ρ is the density of air.

A is the area of the plate through which air is flowing.

V is the velocity of flow.

Step 2 of 3

Then, the frictional coefficient is $C_f = \frac{1.328}{\text{Re}^{0.5}}$.

Where, Re is the Reynolds number.

Substitute the value of C_f in equation (1).

$$F_{D_1} = \frac{1}{2}\left(\frac{1.328}{\text{Re}^{0.5}}\right)\rho A V^2 \text{ ..... (2)}$$

Now, the relation for the Reynolds number is, $\text{Re} = \frac{VL}{\nu}$

Substituting the value of Re in equation (2)

$$F_{D_1} = \frac{1}{2}\left(\frac{1.328}{\left(\frac{VL}{\nu}\right)^{0.5}}\right)\rho A V^2$$

$$F_{D_1} = \frac{0.664 \rho A \nu^{0.5} V^{3/2}}{L^{0.5}} \text{ ..... (3)}$$

Step 3 of 3

When the free stream velocity is doubled,

$$V = 2V$$

Then substituting values in equation (3) drag force becomes,

$$\begin{aligned} F_{D_2} &= \frac{0.664 \rho A \nu^{0.5} (2V)^{3/2}}{L^{0.5}} \\ &= 2^{3/2}\left(\frac{0.664 \rho A \nu^{0.5} V^{3/2}}{L^{0.5}}\right) \\ &= 2.83\left(\frac{0.664 \rho A \nu^{0.5} V^{3/2}}{L^{0.5}}\right) \\ &= 2.83 F_{D_1} \end{aligned}$$

$$\boxed{F_{D_2} = 2.83 F_{D_1}}.$$

Therefore, when the free stream velocity is doubled the drag force increases by 2.83 times.

Question: 15.050P

Consider a refrigeration truck traveling at 70 mi/h at a location where the air is at 1 atm and 80°F. The refrigerated compartment of the truck can be considered to be a 9-ft- wide, 8-ft-high, and 20-ft-long rectangular box. Assuming the airflow over the entire outer surface to be turbulent and attached (no flow separation), determine the drag force acting on the top and side surfaces and the power required to overcome this drag.

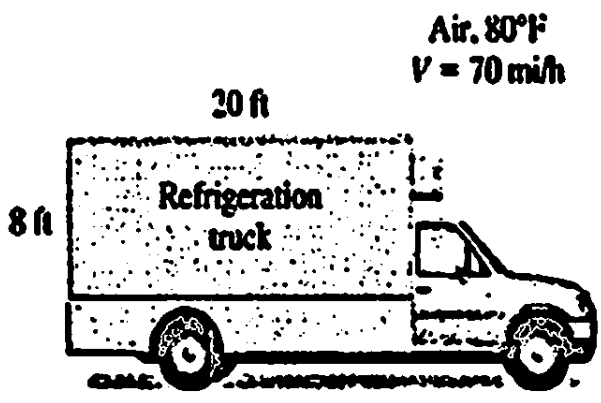


FIGURE P15–50E

Step 1 of 5

Given data:

Velocity of the truck,

$$V = 70 \text{ mi/h}$$
$$= 70 \times 1.46 \text{ ft/s}$$
$$= 102.2 \text{ ft/s}$$

Width of the refrigerated compartment is $w = 9$ ft

Length of the refrigerated compartment is $L = 20$ ft

Height of the refrigerated compartment is $h = 8$ ft

Pressure of the air, $P = 1$ atm

Temperature of the air, $T = 80°\text{F}$

Step 2 of 5

From the table properties of air,

At pressure, $P = 1$ atm

At temperature, $T = 80°\text{F}$

The density of air, $\rho = 0.0735 \text{ lbm/ft}^3$

Kinematic viscosity of air, $\nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s}$

Now, the Reynolds number at a length of 6 m of the top surface,

$$\text{Re}_L = \frac{VL}{\nu} \quad \ldots\ldots (1)$$

Where,

The velocity of flow be V

The kinematic viscosity of air be ν

The length of the refrigerated compartment be L

Step 3 of 5

Substituting the values of V, L and ν in equation (1), we get

$$\text{Re}_L = \frac{VL}{\nu}$$
$$= \frac{102.2 \times 20}{1.697 \times 10^{-4}}$$
$$= 1.204 \times 10^7$$

The air flow over the entire outer surface is assumed to be turbulent.

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$C_f = \frac{0.074}{\text{Re}_L^{1/5}}$$
$$= \frac{0.074}{\left(1.204 \times 10^7\right)^{1/5}}$$
$$= 2.83 \times 10^{-3}$$

Step 4 of 5

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2 \quad \ldots\ldots (2)$$

Where,

The density of air be ρ

The area of the top and side surface of the truck be A

The velocity of flow be V

Area of the top and side surface of the truck is,

$$A = \text{Area of the top surface} + 2(\text{Area of the side surface})$$
$$= (L \times w) + 2(h \times L)$$
$$= (20 \times 9) + 2(8 \times 20)$$
$$= 500 \text{ ft}^2$$

Step 5 of 5

Substituting the value of ρ, A, V, and C_f in equation (2), we get

$$F_D = \frac{1}{2} \times 2.83 \times 10^{-3} \times 0.0735 \times 500 \times (102.2)^2$$
$$= 543.14 \text{ lb}$$

Therefore, the drag force acting on the top and side surfaces is $\boxed{F_D = 543.14 \text{ lb}}$

The power required to overcome this drag is,

$$P = F_D \times V$$
$$= 543.14 \times 102.2$$
$$= 55508.9 \text{ lbf-ft/s}$$
$$= \frac{55508.9}{550} \text{ hp}$$
$$= 100.9 \text{ hp}$$

Therefore, the power required to overcome this drag is $\boxed{P = 100.9 \text{ hp}}$

Question: 15.051P

Reconsider Prob. 15-50E. Using EES (or other) software, investigate the effect of truck speed on the total drag force acting on the top and side surfaces, and the power required to overcome it. Let the truck speed vary from 0 to 100 mi/h in increments of 10 mi/h. Tabulate and plot the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

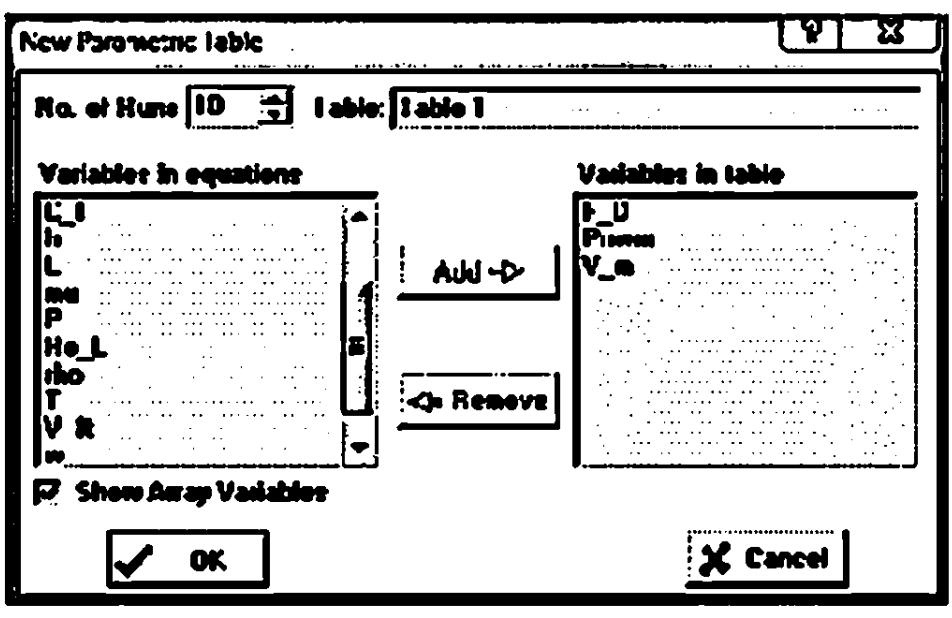
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"V_m is to be varied from 0 to 10 mi/hr"        "Velocity of truck"
V_ft=V_m*convert(mile/hr,ft/s)
w=9[ft]                                          "Width of refrigeration truck"
L=20[ft]                                         "Length of refrigeration truck"
h=8[ft]                                          "Height of refrigeration tuck"
P=1[atm]*convert(atm,psia)                       "Pressure of air"
T=80[F]                                          "Temperature of air"

rho=Density(Air,T=T,P=P)                         "Properties of air"
mu=Viscosity(Air,T=T)*convert(lb_m/ft hr, lb_m/ft s)
Re_L=rho*V_ft*L/mu                               "Reynolds number"
C_f=0.074/(Re_L)^(1/5)                           "Friction Coefficient"
A_t=L*W                                          "Area of the top surface"
A_s=2*L*h                                        "Area of the sides"
A=A_t+A_s                                        "Total Area"
F_D=1/2*C_f*rho*V_ft^2*A/g#                      "Drag Force"
Power=F_D*V_ft*convert(lb_f ft/s,kW)             "Power required"
```

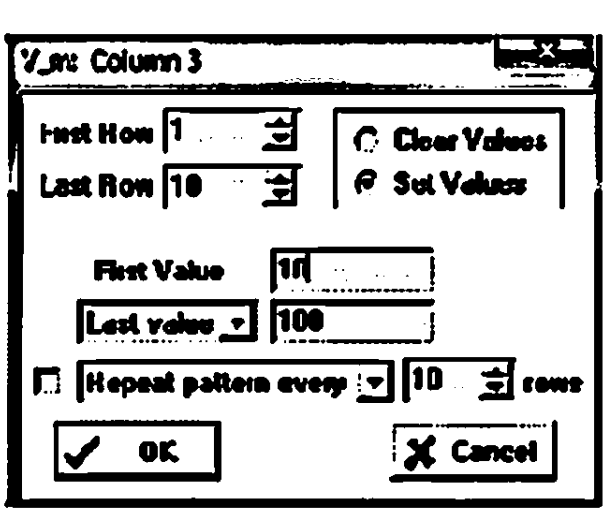
Step 2 of 5

Now to solve the equation for a range of velocity of truck, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 10.
Select variables in equations as "F_D", "Power" and "V_m" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

The velocity of the truck is varied from 0 to 100 mi/hr
To enter the values in the "V_m" column, click on the drop tab in "V_m" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

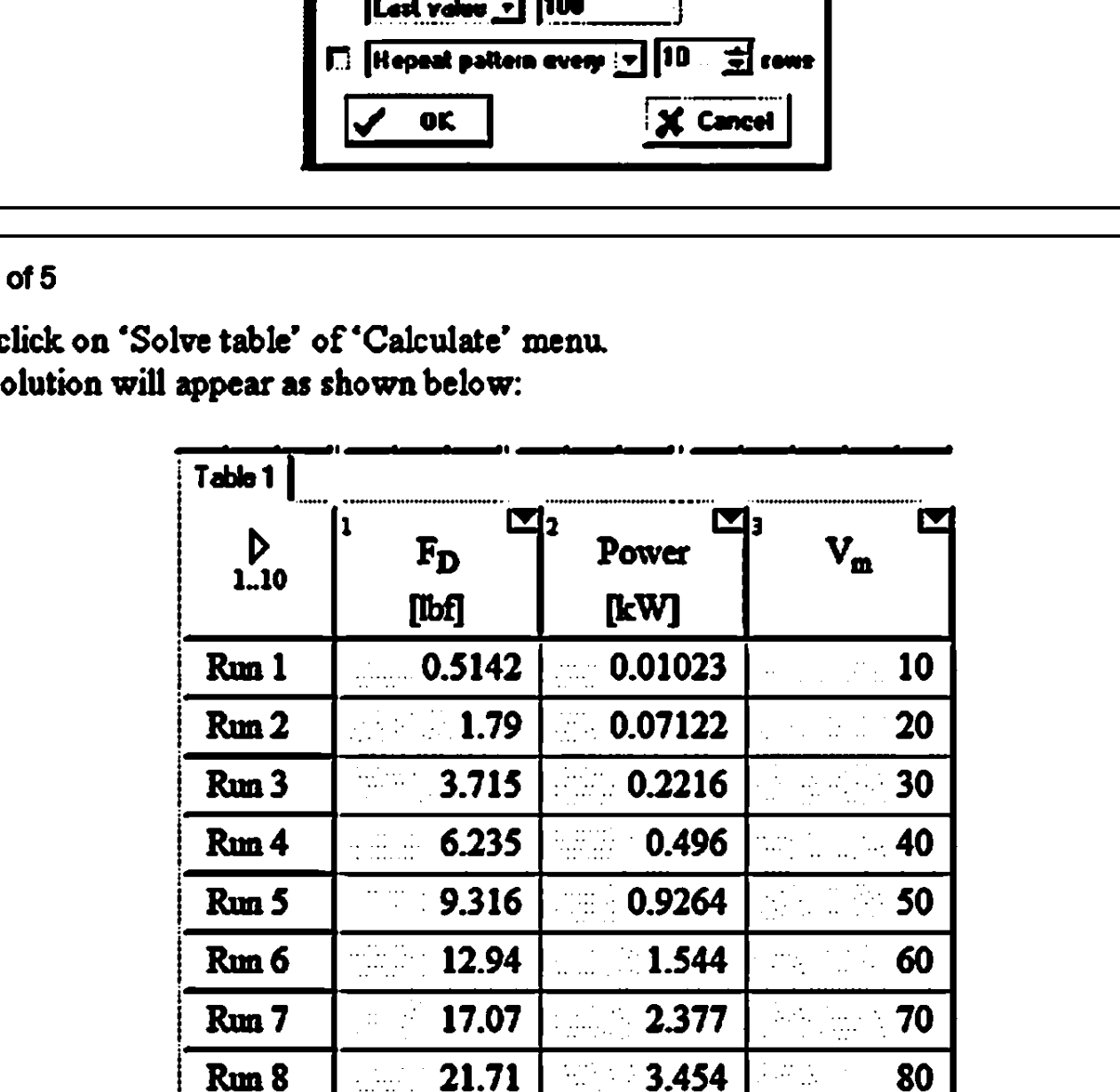
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..10	F_D [lbf]	Power [kW]	V_m
Run 1	0.5142	0.01023	10
Run 2	1.79	0.07122	20
Run 3	3.715	0.2216	30
Run 4	6.235	0.496	40
Run 5	9.316	0.9264	50
Run 6	12.94	1.544	60
Run 7	17.07	2.377	70
Run 8	21.71	3.454	80
Run 9	26.84	4.804	90
Run 10	32.44	6.452	100

Step 5 of 5

To plot the drag force and power required as a function of truck speed, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the drag force and power required as a function of truck speed is shown:



From the above plot we can interpret that as the velocity of the truck increases, the drag force and power required to overcome this drag also increases.

Question: 15.052P

Air at 25°C and 1 atm is flowing over a long flat plate with a velocity of 8 m/s. Determine the distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location.

Step 1 of 4

Given data:
Velocity of flow is $V = 8\text{m/s}$.
Atmospheric conditions,
Temperature of air is $T = 25°\text{C}$.
Atmospheric pressure is $P = 1\text{ atm}$.

Step 2 of 4

From the table properties of air at 1 atm pressure.
Density of air at 25°C and 1atm pressure is $\rho = 1.184\text{kg/m}^3$.
Kinematic viscosity of air at 25°C and 1atm pressure is $\nu = 1.562\times10^{-5}\text{ m}^2/\text{s}$.

When a fluid flows through a flat plate the transition of flow from laminar to turbulent happens when the Reynolds number becomes equal to the critical Reynolds number.
The value of critical Reynolds number is, $\text{Re}_{cr} = 5\times10^5$
The critical Reynolds number can be found out using the formula,

$$\text{Re}_{cr} = \frac{Vx_{cr}}{\nu} \qquad \ldots\ldots (1)$$

Where, V is the velocity of flow.
ν is the kinematic viscosity.
x_{cr} is the distance from the leading edge of the plate where the flow becomes turbulent.

Step 3 of 4

Substituting the value of Re_{cr}, V, and ν in equation (1).

$$\text{Re}_{cr} = \frac{Vx_{cr}}{\nu}$$

$$5\times10^5 = \frac{8(x_{cr})}{1.562\times10^{-5}}$$

$$x_{cr} = \frac{(5\times10^5)(1.562\times10^{-5})}{8}$$

$$= 0.97625\text{ m}$$

Therefore the flow becomes turbulent at a distance of $\boxed{x = 0.97625\text{ m}}$ from the leading edge of the plate.

Step 4 of 4

Now, the thickness of the boundary layer at that location can be found out using the equation given below.

$$\delta = \frac{4.91x}{\text{Re}_x^{0.5}}$$

$$= \frac{4.91x_{cr}}{\text{Re}_{cr}^{0.5}}$$

Substituting the values of x_{cr}, and Re_{cr} in the above equation.

$$\delta = \frac{4.91x_{cr}}{\text{Re}_{cr}^{0.5}}$$

$$= \frac{4.91(0.97625)}{(5\times10^5)^{0.5}}$$

$$= 0.00678\text{ m}$$

Therefore, the thickness of the boundary layer is $\boxed{\delta = 0.00678\text{ m}}$.

Question: 15.053P

Repeat Prob. 15-52 for water.

Step 1 of 4

Given data:

Velocity of flow is $V = 8\ \text{m/s}$.

Atmospheric conditions,

Temperature of air is $T = 25°\text{C}$.

Atmospheric pressure is $P = 1\ \text{atm}$.

Step 2 of 4

From the table properties of air at 1 atm pressure.

Density of water at 25°C and 1atm pressure is $\rho = 997\ \text{kg/m}^3$.

Dynamic viscosity of water at 25°C and 1atm pressure is $\mu = 0.891 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$.

When a fluid flows through a flat plate the transition of flow from laminar to turbulent happens when the Reynolds number becomes equal to the critical Reynolds number.

The value of critical Reynolds number is, $\text{Re}_{cr} = 5 \times 10^5$

The critical Reynolds number can be found out using the formula,

$$\text{Re}_{cr} = \frac{\rho V x_{cr}}{\mu} \qquad \ldots\ldots (1)$$

Where, V is the velocity of flow

μ is the dynamic viscosity

x_{cr} is the distance from the leading edge of the plate where the flow becomes turbulent.

Step 3 of 4

Substituting the value of Re_{cr}, V, and μ in equation (1).

$$\text{Re}_{cr} = \frac{\rho V x_{cr}}{\mu}$$

$$5 \times 10^5 = \frac{997 \times 8 \times x_{cr}}{0.891 \times 10^{-3}}$$

$$x_{cr} = \frac{\left(5 \times 10^5\right)\left(0.891 \times 10^{-3}\right)}{997 \times 8}$$

$$= 0.056\ \text{m}$$

Therefore the flow becomes turbulent at a distance of $\boxed{x = 0.056\ \text{m}}$ from the leading edge of the plate.

Step 4 of 4

Now, the thickness of the boundary layer at that location can be found out using the equation given below.

$$\delta = \frac{4.91x}{\text{Re}_x^{0.5}}$$

$$= \frac{4.91x_{cr}}{\text{Re}_{cr}^{0.5}}$$

Substituting the values of x_{cr}, and Re_{cr} in the above equation.

$$\delta = \frac{4.91x_{cr}}{\text{Re}_{cr}^{0.5}}$$

$$= \frac{4.91(0.0559)}{\left(5 \times 10^5\right)^{0.5}}$$

$$= 0.00039\ \text{m}$$

Therefore, the thickness of the boundary layer is $\boxed{\delta = 0.00039\ \text{m}}$.

Question: 15.054P

The top surface of the passenger car of a train moving at a velocity of 70 km/h is 3.2 m wide and 8 m long. If the outdoor air is at 1 atm and 25°C, determine the drag force acting on the top surface of the car.

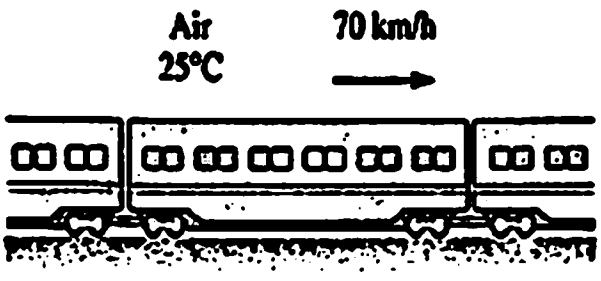


FIGURE P15–54

Step 1 of 4

Given data:

Velocity of passenger car is $V = 70$ km/h.

Width of the passenger car is $w = 3.2$ m.

Length of the passenger car is $L = 8$ m.

Atmospheric condition,

Pressure is $P = 1$ atm.

Temperature is $T = 25°C$.

Step 2 of 4

From the properties of air tables at 1 atm pressure.

The density of air at 1atm and 25°C is $\rho = 1.184$ kg/m^3.

Kinematic viscosity of air at 1atm and 25°C is $\nu = 1.562 \times 10^{-5}$ m^2/s.

Now, the Reynolds number at the end of the plate that means at a length of 10 m of the plate,

$$\mathrm{Re}_L = \frac{VL}{\nu} \qquad \ldots\ldots (1)$$

Where, V is the velocity of flow.

L is the length of the plate.

ν is the kinematic viscosity of fluid.

Converting units of velocity

$$V = 70\left(\frac{5}{18}\right) \text{m/s}$$

$$= 19.44 \text{ m/s}$$

Step 3 of 4

Substituting the values of V, L and ν in equation (1).

$$\mathrm{Re}_L = \frac{VL}{\nu}$$

$$= \frac{19.44 \times 8}{1.562 \times 10^{-5}}$$

$$= 9.956 \times 10^6$$

We know that the critical value of Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$ since the obtained value of Reynolds number is greater then the critical value the flow will be a combined laminar and turbulent flow.

Therefore, we can now calculate the frictional coefficient by using the formula given below.

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L}$$

$$= \frac{0.074}{\left(9.956 \times 10^6\right)^{1/5}} - \frac{1742}{9.956 \times 10^6}$$

$$= 0.0029 - 0.00017$$

$$= 0.00273$$

Step 4 of 4

Since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2 \qquad \ldots\ldots (2)$$

Were, ρ is the density of air.

A is the area of the top surface of the passenger car.

V is the velocity of flow.

The area of the top surface of the passenger car, through which air is flowing,

$$A = L \times w$$

$$= 8 \times 3.2$$

$$= 25.6 \text{ m}^2$$

Now, substituting the value of ρ, A, V, and C_f in equation (2).

$$F_D = \frac{1}{2} \times 0.00273 \times 1.184 \times 25.6 \times (19.44)^2$$

$$= 15.63 \text{ N}$$

Therefore, the drag force acting on the top surface of the car is $\boxed{F_D = 15.63 \text{ N}}$.

Question: 15.055P

The weight of a thin flat plate 50 cm × 50 cm in size is balanced by a counterweight that has a mass of 2 kg, as shown in Fig. P15-55. Now a fan is turned on, and air at 1 atm and 25°C flows downward over both surfaces of the plate (front and back in the sketch) with a free-stream velocity of 10 m/s. Determine the mass of the counterweight that needs to be added in order to balance the plate in this case.

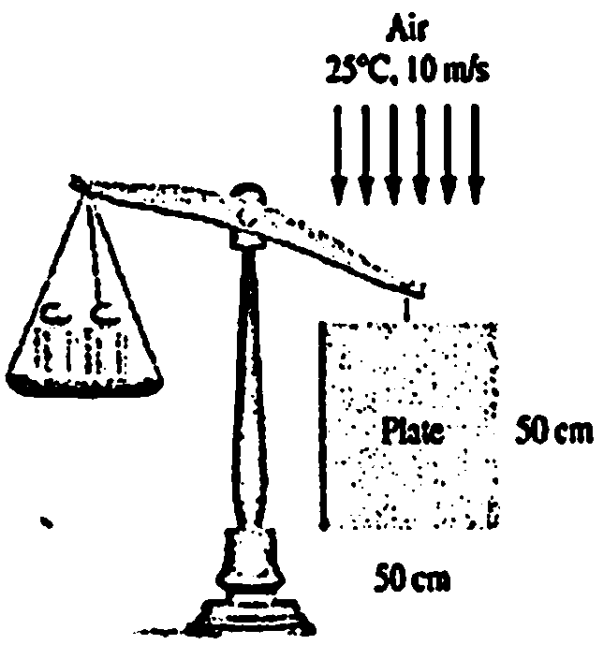


FIGURE P15–55

Step 1 of 6

Given data:

Free stream velocity of air is $V = 10\ m/s$.

Breadth of the thin flat plate is $b = 50\ cm$.

Length of the thin flat plate is $L = 50\ cm$.

Mass of the counter weight is $m = 2\ kgm$.

Atmospheric condition,

Pressure is $P = 1\ atm$.

Temperature is $T = 25°C$.

Step 2 of 6

From the table properties of air at 1 atm pressure.

The density of air at 1atm and 25°C is $\rho = 1.184\ kg/m^3$.

Kinematic viscosity of air at 1atm and 25°C is $\nu = 1.562 \times 10^{-5}\ m^2/s$.

Now, the Reynolds number for a length L is,

$$Re_L = \frac{VL}{\nu} \qquad \ldots\ldots (1)$$

Where, V is the velocity of air.

L is the length of the thin flat plate.

ν is the kinematic viscosity of fluid.

Here, length,

$L = 50\ cm$

$= 0.5\ m$

Step 3 of 6

Substituting the values of V, L and ν in equation (1).

$$Re_L = \frac{VL}{\nu}$$

$$= \frac{10 \times 0.5}{1.562 \times 10^{-5}}$$

$$= 3.2 \times 10^5$$

We know the critical Reynolds number is 5×10^5, since the obtained Reynolds number is less than the critical Reynolds number the flow is laminar.

Step 4 of 6

The friction coefficient can be found out using the formula given below.

$$C_f = \frac{1.328}{Re_L^{0.5}} \qquad \ldots\ldots (2)$$

Where Re_L is the Reynolds number.

Substituting the value of Re_L in equation (2)

$$C_f = \frac{1.328}{\left(3.2 \times 10^5\right)^{0.5}}$$

$$= 0.00235$$

Step 5 of 6

Here since the pressure drag is zero the drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_f \rho A V^2 \qquad \ldots\ldots (3)$$

Were, ρ is the density of air.

A is the area of the flat plate through which air is flowing.

V is the velocity of flow.

Here, area of the flat plate,

$A = L \times b$

We have, $b = 50\ cm = 0.5\ m$

$A = 0.5 \times 0.5$

$= 0.25\ m^2$

Here since air is flowing through both sides of the plate area should be taken as,

$A = 2 \times 0.25$

$= 0.5\ m^2$

Step 6 of 6

Now, substituting the value of ρ, A, V, and C_f in equation (3).

$$F_D = \frac{1}{2} \times 0.00235 \times 1.184 \times 0.5 \times (10)^2$$

$$= 0.06956\ N$$

Now, we have to find out the mass of the body whose weight is 0.06956 N using the formula given below.

$$m = \frac{F_D}{g}$$

Where F_D is the drag force and g is the gravitational acceleration.

Where, $g = 9.81\ m/s^2$

$$m = \frac{0.06956}{9.81}$$

$$= 0.0071\ kg$$

Therefore, a counterweight of mass $\boxed{m = 0.0071\ kg}$ is needed to balance the plate.

Question: 15.056P

In flow over cylinders, why does the drag coefficient suddenly drop when the boundary layer becomes turbulent? Isn't turbulence supposed to increase the drag coefficient instead of decreasing it?

Step 1 of 1

In flow over cylinder the drag coefficient suddenly drops when the flow in the boundary layer suddenly becomes turbulent. This is because when the flow becomes turbulent the separation point will be moved to the rear of the body, which will reduce the size of the wake and there by reduces the magnitude of pressure drag. But for stream lined bodies this turbulence will increase the drag coefficient because of friction drag.

Question: 15.057P

In flow over bluff bodies such as a cylinder, how does the pressure drag differ from the friction drag?

Step 1 of 1

Friction drag is caused due to the shear stress on the surface of the body. Pressure drag is caused due to the pressure difference between the back and front of the body. When fluid flow over a blunt body a wake is formed in the rear part of the body this wake is responsible for the pressure difference between the front and back of the body.

Question: 15.058P

Why is flow separation in flow over cylinders delayed when the boundary layer is turbulent?

Step 1 of 1

For turbulent flow the flow separation occurs at $\theta \cong 140°$. The delay in separation in turbulent flow is due to the rapid fluctuations of fluid in the transverse direction which will enable the turbulent boundary layer to travel a long distance before the separation occurs, due to this a narrow wake is formed as a result the pressure drag will be smaller.

Question: 15.059P

A 1.2-in-outer-diameter pipe is to span across a river at a 140-ft-wide section while being completely immersed in water. The average flow velocity of the water is 10 ft/s. and its temperature is 70°F. Determine the drag force exerted on the pipe by the river.

Step 1 of 3

Given data:

Outer diameter of the pipe $D = 1.2\text{in.}$

Width of the section $w = 140\text{ft}$

Average flow velocity of water $V = 10\text{ft/s}$

Temperature of water $T = 70°\text{F}$

Step 2 of 3

From properties of water table at $T = 70°\text{F}$ properties of water are

$\rho = 1.94\text{slugs/ft}^3$

$\mu = 2.04\times10^{-5}\text{lb}\cdot\text{s/ft}^2$

The Reynolds number can be found out using the formula given below.

$$\text{Re} = \frac{\rho VD}{\mu}$$

Where ρ is the density of water.

V is the velocity of flow.

D is the diameter of the pipe.

μ is the dynamic viscosity.

Substituting the values in the above formula, we get

$$\text{Re} = \frac{\rho VD}{\mu}$$

$$= \frac{1.94\times10\times\frac{1.2}{12}}{2.04\times10^{-5}}$$

$$= 9.509\times10^{4}$$

Using Reynolds number value the drag coefficient can be obtained from graph "average drag coefficient for cross flow over a smooth circular cylinder and sphere"

$C_D = 1.1$

Step 3 of 3

Now, the drag force acting on the pipe can be found out using the formula given below.

$$F_D = \frac{1}{2}C_D\rho AV^2.$$

Where C_D is the drag coefficient.

ρ is the density of water.

A is the frontal area of the pipe.

V is the velocity of flow.

The frontal area of the pipe is,

$$A = w\times D$$

$$= 140\times\frac{1.2}{12}$$

$$= 14\text{ft}^2$$

Substituting the values in the above formula, we get

$$F_D = \frac{1}{2}\times1.1\times1.94\times14\times(10)^2$$

$$= 1493.81\text{bf}$$

Therefore the drag force acting on the cylinder $\boxed{F_D = 1493.81\text{bf}}$

Question: 15.060P

A long 8-cm-diameter steam pipe passes through some area open to the wind. Determine the drag force acting on the pipe per unit of its length when the air is at 1 atm and 5°C and the wind is blowing across the pipe at a speed of 50 km/h.

Step 1 of 2

Obtain the properties of air at pressure of 1 atm and temperature of $5°C$ from the table A-9 "properties of the air at 1 atm" of the text book.

Density of air, $\rho = 1.269\ \text{kg/m}^3$.

Kinematic viscosity of air, $\nu = 1.382 \times 10^{-5}\ \text{m}^2/\text{s}$.

Convert the velocity of wind from km/h to m/s.

$$\begin{aligned} V &= 50\ \text{km/h} \\ &= 50\left(\frac{5}{18}\right)\text{m/s} \\ &= 13.88\ \text{m/s} \end{aligned}$$

Calculate the Reynolds number by using the formula.

$$\text{Re} = \frac{VD}{\nu}.$$

Here, V is the velocity of wind, D is the diameter of the pipe, and ν is the kinematic viscosity of air.

Substitute $13.88\ \text{m/s}$ for V, 0.08 m for D, and $1.382 \times 10^{-5}\ \text{m}^2/\text{s}$ for ν .

$$\begin{aligned} \text{Re} &= \frac{13.88(0.08)}{1.382 \times 10^{-5}} \\ &= 8.03 \times 10^4 \end{aligned}$$

Obtain the C_D value at Reynolds number of 8.03×10^4 from the plot of "average drag coefficient for cross flow over a smooth circular cylinder and a smooth sphere" as, $C_D = 1$

Step 2 of 2

Calculate the frontal area for flow past a cylinder.

$$A = L \times D$$

Here, A is the frontal area for flow past a cylinder, L is the length of pipe.

Calculate the drag force per unit length acting on the pipe by using the formula.

$$\begin{aligned} F_D &= \frac{\left(\frac{1}{2}C_D\rho AV^2\right)}{L} \\ &= \frac{\left(\frac{1}{2}C_D\rho(DL)V^2\right)}{L}. \\ &= \frac{1}{2}C_D\rho DV^2 \end{aligned}$$

Here, C_D is the drag coefficient, A is the frontal area, and V is the velocity of flow.

Substitute 1 for C_D , $1.269\ \text{kg/m}^3$ for ρ , 0.08 m for D, and $13.88\ \text{m/s}$ for V.

$$\begin{aligned} F_D &= \frac{1}{2} \times 1 \times 1.269 \times 0.08 \times (13.88)^2 \\ &= 9.779\ \text{N/m} \end{aligned}$$

Therefore, the drag force acting per unit length is, $\boxed{9.779\ \text{N/m}}$.

Question:

Consider 0.8-cm-diameter hail that is falling freely in atmospheric air at 1 atm and 5°C. Determine the terminal velocity of the hail. Take the density of hail to be 910 kg/m3.

Step 1 of 10

Given data:

Diameter of the hail is $D = 0.8\ \text{cm}$.

Density of the hail is $\rho_{hail} = 910\ \text{kg/m}^3$.

Pressure of the air is $P = 1\ \text{atm}$.

Temperature of the air is $T = 5°\text{C}$.

Step 2 of 10

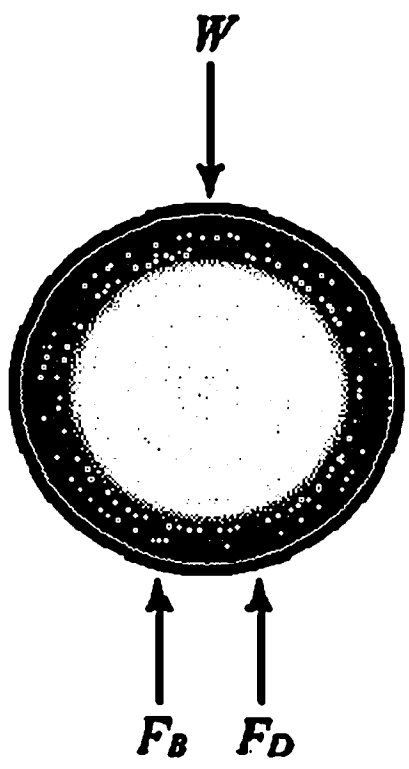
Weight of a freely falling body is equal to the sum of drag force and buoyancy force acting on the body. The drag force and buoyancy force will act opposite to the direction of motion of the body and weight of a body always acts downwards. Since the body is falling downwards the drag force and buoyancy force will act upwards.

When the body is dropped from a height it first accelerates under the influence of its weight, as the body falls the motion of the body is resisted by the drag force acting opposite to the direction of the motion of the body. This drag force goes on increasing as the velocity of the body increases, this continues until all the forces balance each other and the net force acting on the body is zero.

When the net force acting on the body becomes zero the velocity becomes constant and this constant velocity is the maximum velocity that the body can achieve and this velocity is called the terminal velocity.

Step 3 of 10

The free body diagram of the hail showing different forces acting on it.



Therefore, terminal velocity is achieved when, $F_D + F_B = W$.

Where, F_D is the drag force.

F_B is the buoyancy force.

W is the weight of the body.

Step 4 of 10

Now, we know that,

$F_D + F_B = W$

Now the drag force, $F_D = W - F_B$.

Here,

$$F_D = \frac{1}{2} C_D \rho_{air} A V^2$$

Where, C_D is the drag force.

ρ_{air} is the density of air.

A is the frontal area of the hail.

V is the velocity of the hail.

Step 5 of 10

Now, weight of the hail,,

$W = mg$ ..... (1)

We know that, $\text{Density} = \frac{\text{mass}}{\text{volume}}$

$$\rho = \frac{m}{V}$$

$$m = \rho V$$

Therefore, equation (1) becomes,

$W = \rho_{hail} V g$

Where, ρ_{hail} is the density of hail.

V is the volume of the hail.

g is the acceleration due to gravity.

Step 6 of 10

From the table properties of air at 1 atm pressure.

Density of air at 1atm and 5°C is $\rho_{air} = 1.269\ \text{kg/m}^3$.

Kinematic viscosity of air at 1atm and 5°C is $\nu = 1.382 \times 10^{-5}\ \text{m}^2/\text{s}$.

We have, $F_D = W - F_B$

Here,

$$F_D = \frac{1}{2} C_D \rho_{air} A V^2$$

$W = \rho_{hail} V g$

$F_B \cong 0$

Step 7 of 10

Substituting these expressions in the above equation.

$$\frac{1}{2} C_D \rho_{air} A V^2 = \rho_{hail} V g - 0$$

$$\frac{1}{2} C_D \rho_{air} A V^2 = \rho_{hail} V g$$ ..... (2)

Substituting these expressions in the above equation.

$$\frac{1}{2}C_D\rho_{air}AV^2 = \rho_{hail}Vg - 0$$

$$\frac{1}{2}C_D\rho_{air}AV^2 = \rho_{hail}Vg \qquad \ldots\ldots (2)$$

Here the frontal area is $A = \frac{\pi}{4}D^2$.

Volume of the hail is $V = \frac{\pi D^3}{6}$

Substituting these values in equation (2).

$$\frac{1}{2}C_D\rho_{air}\left(\frac{\pi}{4}D^2\right)V^2 = \rho_{hail}\left(\frac{\pi D^3}{6}\right)g$$

$$\frac{1}{4}C_D\rho_{air}V^2 = \frac{1}{3}\rho_{hail}gD$$

$$V^2 = \frac{4\rho_{hail}gD}{3C_D\rho_{air}}$$

$$V = \sqrt{\frac{4\rho_{hail}gD}{3C_D\rho_{air}}}$$

$$= \sqrt{\frac{4\times 910\times 9.81\times 0.008}{3\times C_D\times 1.269}}$$

$$= \frac{8.662}{\sqrt{C_D}}$$

$$V = \frac{8.662}{\sqrt{C_D}} \qquad \ldots\ldots (3)$$

Step 8 of 10

Then, calculating the Reynolds number.

$$\mathrm{Re} = \frac{VD}{\nu}.$$

Where, V is the velocity if the hail.
D is the diameter of the hail.
ν is the kinematic viscosity of air.

$$\mathrm{Re} = \frac{V\times 0.008}{1.382\times 10^{-5}}$$

$$= 578.87V$$

$$\mathrm{Re} = 578.87V \qquad \ldots\ldots (4)$$

Step 9 of 10

Now, assume a value for the velocity and substituting the assumed value in equation (4) and calculate the value of Reynolds number based on the obtained Reynolds number find out the value of C_D using the graph 'Average drag coefficient for cross flow over a smooth circular cylinder and a smooth sphere', and substitute the value of C_D in equation (3) and check whether the assumed value matches the calculated value, if not assume another value for V and repeat the process until the assumed value becomes equal to the calculated value and that velocity will be the terminal velocity of the hail.

Step 10 of 10

Here, we assume the velocity as $V = 13.7$ m/s or 49 km/h.

Substituting the value in equation (4)

$$\mathrm{Re} = 578.87\times 13.7$$

$$= 7930.5$$

Taking C_D value from the average drag coefficient for cross flow over a smooth circular cylinder and a smooth sphere.

The corresponding value of C_D is $C_D = 0.4$

Substituting the value of C_D in equation (3)

$$V = \frac{8.662}{\sqrt{0.4}}$$

$$= 13.7 \text{ m/s}$$

Since the assumed value of C_D matches the calculated value of C_D it is the terminal velocity.

Therefore, the terminal velocity of the hail is $\boxed{V = 13.7 \text{ m/s}}$.

Question: 15.062P

A 0.1-mm-diameter dust particle whose density is 2.1 g/cm3 is observed to be suspended in the air at 1 atm and 25°C at a fixed point. Estimate the updraft velocity of air motion at that location. Assume Stokes law to be applicable. Is this a valid assumption?

Step 1 of 4

Given data:

Diameter of the dust particle, $D = 0.1\,\text{mm} = 0.0001\,\text{m}$

Density of the dust particle, $\rho_d = 2.1\,\text{g/cm}^3 = 2.1\times10^3\ \text{kg/m}^3$

Pressure of air, $P = 1\,\text{atm}$

Temperature of the air, $T = 25°\text{C}$

Density of air at 1 atm pressure and 25°C, $\rho_a = 1.184\ \text{kg/m}^3$

Viscosity of air at 1 atm pressure and 25°C, $\mu = 1.849\times10^{-5}\ \text{kg/m-s}$

Step 2 of 4

The dust particle is suspended in the air so the updraft velocity of air should be equal to the terminal velocity of dust particle. Disregarding the initial transient period during which the dust particle accelerate to their terminal velocity, velocity at which the dust particle fall is equal to the terminal velocity of the particle. For a body terminal velocity is reached when the weight of the body becomes equal to the sum of drag force and buoyancy force acting on the body.

$$W = F_D + F_B \qquad \ldots\ldots (1)$$

Where W is the weight of the body $(W = \rho_d gV)$

F_B is the buoyancy force on body $(F_B = \rho_a gV)$

F_D is the drag force as per Stokes law $(F_D = 3\pi\mu VD)$

V is the velocity of the particle.

V is the Volume of the dust particle

Step 3 of 4

Volume of the dust particle (particle has a spherical shape),

$$V = \frac{\pi D^3}{6}$$

$$= \frac{\pi(0.0001)^3}{6}$$

$$= 5.23\times10^{-13}\ \text{m}^3$$

Weight of the dust particle,

$$W = \rho_d gV$$

$$= \left(2.1\times10^3\right)\times9.81\times\left(5.23\times10^{-13}\right)$$

$$= 1.077\times10^{-8}\ \text{N}$$

Step 4 of 4

Buoyancy force acting on dust particle,

$$F_B = \rho_a gV$$

$$= (1.184)\times9.81\times\left(5.23\times10^{-13}\right)$$

$$= 6.074\times10^{-12}\ \text{N}$$

The drag force on dust particle according to equation (1) is

$$F_D = W - F_B$$

$$3\pi\mu VD = \left(1.077\times10^{-8}\right) - \left(6.074\times10^{-12}\right)$$

$$V = \frac{1.0764\times10^{-8}}{3\pi\mu D}$$

$$= \frac{1.0764\times10^{-8}}{3\pi\left(1.849\times10^{-5}\right)0.0001}$$

$$= 0.6177\ \text{m/s}$$

Therefore, the velocity of the dust particle is, $\boxed{V = 0.6177\ \text{m/s}}$

Question: 15.063P

Dust particles of diameter 0.06 mm and density 1.6 g/cm3 are unsettled during high winds and rise to a height of 350 m by the time things calm down. Estimate how long it takes for the dust particles to fall back to the ground in still air at 1 atm and 15°C, and their velocity. Disregard the initial transient period during which the dust particles accelerate to their terminal velocity, and assume Stokes law to be applicable.

Step 1 of 5

Given data:

Diameter of the dust particle, $D = 0.06\ \text{mm} = 6\times10^{-5}\ \text{m}$

Density of the dust particle, $\rho_d = 1.6\ \text{g/cm}^3 = 1.6\times10^{3}\ \text{kg/m}^3$

Height through which the dust particle rises, $h = 350\ \text{m}$

Pressure of air, $P = 1\ \text{atm}$

Temperature of the air, $T = 15°\text{C}$

Density of air at 1 atm pressure and 15°C, $\rho_a = 1.225\ \text{kg/m}^3$

Viscosity of air at 1 atm pressure and 15°C, $\mu = 1.802\times10^{-5}\ \text{kg/m}\cdot\text{s}$

Step 2 of 5

Since we can disregard the initial transient period during which the dust particle accelerate to their terminal velocity, the velocity at which the dust particle fall will be equal to the terminal velocity of the particle. For a body terminal velocity is reached when the weight of the body becomes equal to the sum of drag force and buoyancy force acting on the body.

$$W = F_D + F_B \qquad \ldots\ldots (1)$$

Where W is the weight of the body $(W = \rho_d gV)$

F_B is the buoyancy force on body $(F_B = \rho_a gV)$

F_D is the drag force as per Stokes law $(F_D = 3\pi\mu VD)$

V is the velocity of the particle.

V is the Volume of the dust particle

Step 3 of 5

Volume of the dust particle (particle has a spherical shape),

$$V = \frac{\pi D^3}{6}$$

$$= \frac{\pi\left(6\times10^{-5}\right)^3}{6}$$

$$= 1.131\times10^{-13}\ \text{m}^3$$

Weight of the dust particle,

$$W = \rho_d gV$$

$$= \left(1.6\times10^{3}\right)\times9.81\times\left(1.131\times10^{-13}\right)$$

$$= 1.775\times10^{-9}\ \text{N}$$

Step 4 of 5

Buoyancy force acting on dust particle,

$$F_B = \rho_a gV$$

$$= (1.225)\times9.81\times\left(1.131\times10^{-13}\right)$$

$$= 1.36\times10^{-12}\ \text{N}$$

The drag force on dust particle according to equation (1) is

$$F_D = W - F_B$$

$$3\pi\mu VD = \left(1.775\times10^{-9}\right) - \left(1.36\times10^{-12}\right)$$

$$V = \frac{1.77\times10^{-9}}{3\pi\mu D}$$

$$= \frac{1.77\times10^{-9}}{3\pi\left(1.802\times10^{-5}\right)\left(6\times10^{-5}\right)}$$

$$= 0.1737\ \text{m/s}$$

Step 5 of 5

The time taken by the dust particle to fall back to the ground,

$$t = \frac{h}{V}$$

$$= \frac{350}{0.1737}$$

$$= 2014.97\ s$$

$$= 33.583\ \text{min}$$

Therefore time taken by the dust particle to fall back to the ground is $\boxed{t = 33.583\ \text{min}}$

Question: 15.064P pg 1

A 2-m-long, 0.2-m-diameter cylindrical pine log (density = 513 kg/m3) is suspended by a crane in the horizontal position. The log is subjected to normal winds of 40 km/h at 5°C and 88 kPa. Disregarding the weight of the cable and its drag, determine the angle θ the cable will make with the horizontal and the tension on the cable.

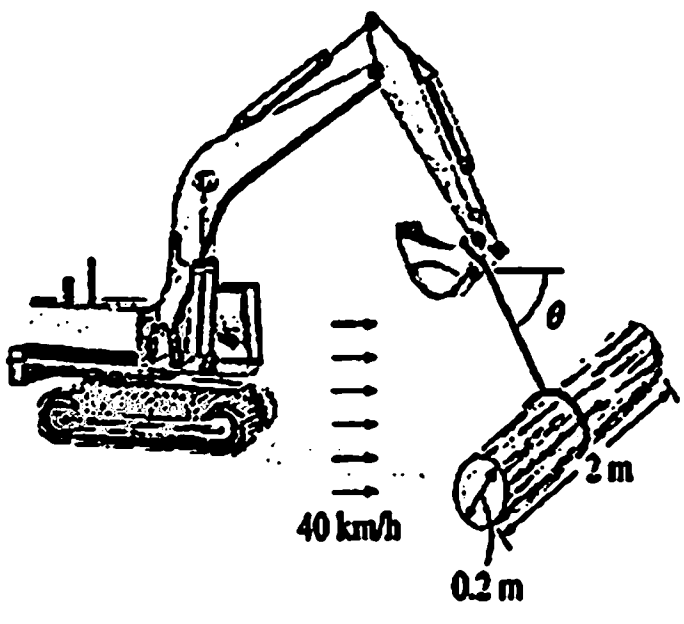


FIGURE P15–64

Step 1 of 11

Calculate the volume of the log by using the equation,

$$V_l = \frac{\pi}{4} D^2 L$$

Here, V_l is the volume of the log, D is the diameter of the cylinder and L is the length of the cylinder.

Substitute $0.2\,\text{m}$ for D and $2\,\text{m}$ for L.

$$V_l = \frac{\pi}{4} \times 0.2^2 \times 2$$
$$= 0.0628\,\text{m}^3$$

Step 2 of 11

Calculate the weight of the pine log using the equation,

$$W = \rho_l g V_l$$

Here, W is the weight of the log, ρ_l is the density of the log, g is the acceleration due to gravity and V_l is the volume of the log.

Substitute $513\,\text{kg/m}^3$ for ρ_l, $9.81\,\text{m/s}^2$ for g and $0.0628\,\text{m}^3$ for V_l.

$$W = 513 \times 9.81 \times 0.0628$$
$$= 316\,\text{N}$$

Step 3 of 11

Calculate the density of air using the relation,

$$\rho = \frac{P}{RT}$$

Here, ρ is the density of air, P is the pressure, R is the universal gas constant and T is the absolute temperature.

Substitute $88\,\text{kPa}$ for P, $0.287\,\text{kJ/kg}\cdot\text{K}$ for R and $278\,\text{K}$ for T.

$$\rho = \frac{88}{0.287 \times 278}$$
$$= 1.103\,\text{kg/m}^3$$

Step 4 of 11

Obtain the dynamic viscosity of air at $5°\text{C}$ from the properties of air.

$$\mu = 1.754 \times 10^{-5}\ \text{kg/m}\cdot\text{s}$$

Here, μ is the dynamic viscosity of air.

Calculate the kinematic viscosity of air by using the equation,

$$\nu = \frac{\mu}{\rho}$$

Here, ν is the kinematic viscosity of air.

Substitute $1.754 \times 10^{-5}\ \text{kg/m}\cdot\text{s}$ for μ, $1.103\,\text{kg/m}^3$ for ρ.

$$\nu = \frac{1.754 \times 10^{-5}}{1.103}$$
$$= 1.59 \times 10^{-5}\,\text{m}^2/\text{s}$$

Step 5 of 11

Convert the wind speed from km/h to m/s by using the following conversion factor,

$$1\,\text{km/hr} = \frac{5}{18}\,\text{m/s}$$

$$V = 40\,\text{km/h}$$
$$= 40 \times \frac{5}{18}$$
$$= 11.11\,\text{m/s}$$

Here, V is the wind speed.

Calculate the Reynold's number for the wind flow over cylindrical pine log by using the equation,

$$\text{Re} = \frac{VD}{\nu}$$

Here, Re is the Reynold's number for the wind flow, D is the diameter of the cylindrical pine log.

$$\mathrm{Re}=\frac{VD}{\nu}$$

Here, Re is the Reynold's number for the wind flow, D is the diameter of the cylindrical pine log.

Substitute $11.11\,\mathrm{m/s}$ for V , $0.2\,\mathrm{m}$ for D and $1.59\times10^{-5}\,\mathrm{m^2/s}$ for ν .

$$\begin{aligned}\mathrm{Re}&=\frac{11.11\times0.2}{1.59\times10^{-5}}\\&=1.397\times10^{5}\end{aligned}$$

Step 6 of 11

Obtain the drag coefficient corresponding to the above Reynold's number by using average drag coefficient for cross-flow over a smooth circular cylinder and a smooth sphere charts.

$$C_D=1.2$$

Here, C_D is the drag coefficient corresponding to the above Reynold's number.

Calculate the frontal area for the flow past a cylinder by using the relation,

$$A=LD$$

Here, A is the frontal area for the flow past a cylinder, L is the length of the cylinder and D is the diameter of the cylinder.

Substitute $2\,\mathrm{m}$ for L and $0.2\,\mathrm{m}$ for D .

$$\begin{aligned}A&=2\times0.2\\&=0.4\,\mathrm{m^2}\end{aligned}$$

Step 7 of 11

Calculate the drag force on the log by using the equation,

$$F_D=C_DA\frac{\rho V^2}{2}$$

Here, F_D is the drag force on the log.

Substitute 1.2 for C_D , $0.4\,\mathrm{m^2}$ for A , $1.103\,\mathrm{kg/m^3}$ for ρ and $11.11\,\mathrm{m/s}$ for V .

$$\begin{aligned}F_D&=1.2\times0.4\times\frac{1.103\times11.11^2}{2}\\&=32.67\,\mathrm{N}\end{aligned}$$

Step 8 of 11

Calculate the resultant force acting on the log by using the equation,

$$F_R=\sqrt{F_D^2+W^2}$$

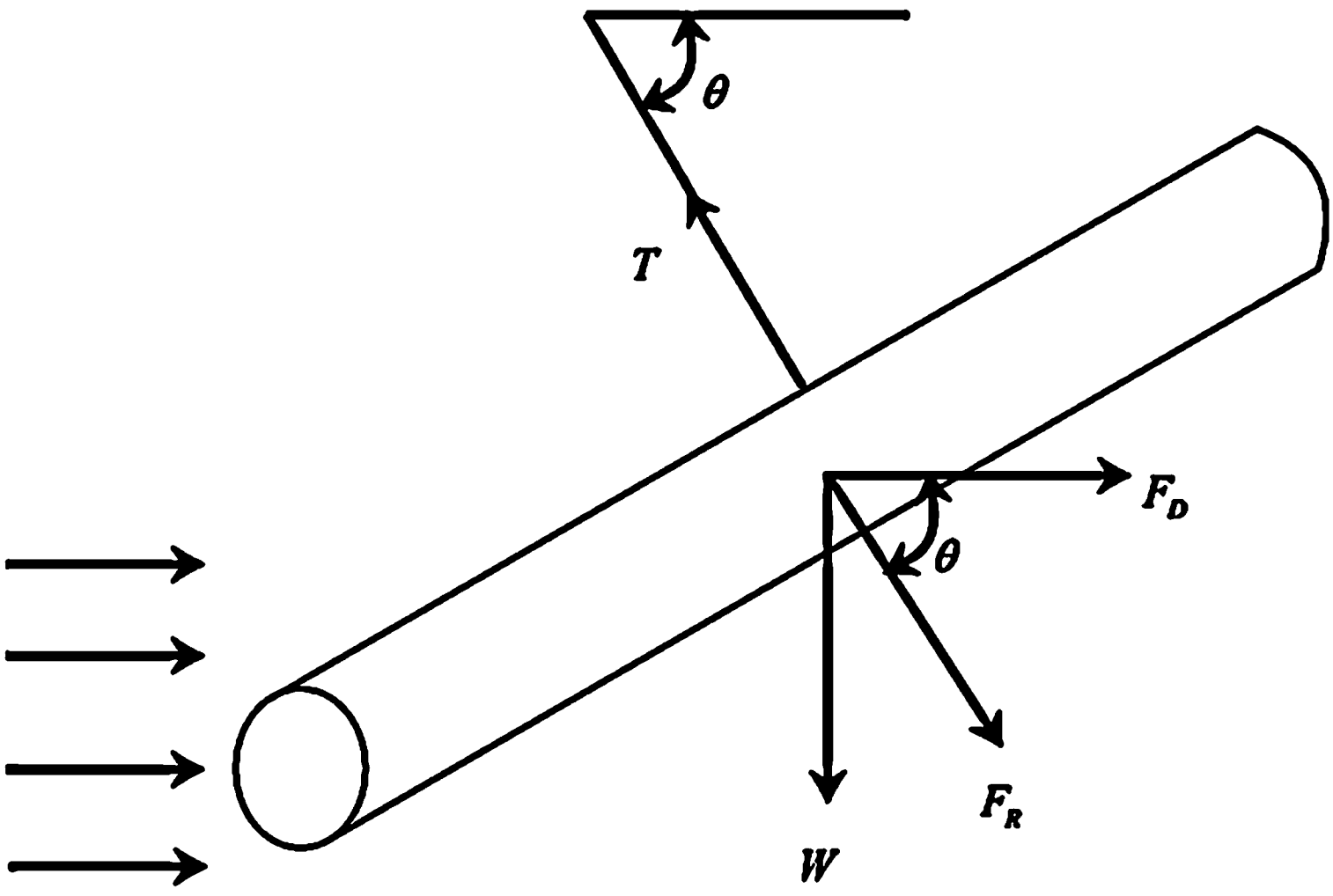
Here, F_R is the resultant force acting on the log.

Substitute $32.67\,\mathrm{N}$ for F_D and $316\,\mathrm{N}$ for W .

$$\begin{aligned}F_R&=\sqrt{F_D^2+W^2}\\&=\sqrt{32.67^2+316^2}\\&=317.68\,\mathrm{N}\end{aligned}$$

Step 9 of 11

Draw the free body diagram of the pine log.



Step 10 of 11

From the free body diagram of the pine log, calculate the angle θ the cable will make with the horizontal by using the equation,

$$\theta=\tan^{-1}\left(\frac{W}{F_D}\right)$$

Here, θ is the angle the cable will make with the horizontal.

Substitute $316\,\mathrm{N}$ for W and $32.67\,\mathrm{N}$ for F_D

$$\begin{aligned}\theta&=\tan^{-1}\left(\frac{316}{32.67}\right)\\&=84^\circ\end{aligned}$$

Therefore, the angle the cable will make with the horizontal is $\boxed{\theta=84^\circ}$.

$= 84°$

Therefore, the angle the cable will make with the horizontal is $\boxed{\theta = 84°}$.

Step 11 of 11

From the free body diagram of the pine log, use force balance to calculate the tension in the cable.

$$T = F_R$$

Here, T is the tension in the cable.

Substitute $317.68\,\text{N}$ for F_R.

$$T = 317.68\,\text{N}$$

Therefore, the tension in the cable is $\boxed{T = 317.68\,\text{N}}$.

Question: 15.065P

A person extends his uncovered arms into the windy air outside at 1 atm and 60°F and 20 mi/h in order to feel nature closely. Treating the arm as a 2-ft-long and 3-in- diameter cylinder, determine the combined drag force on both arms.

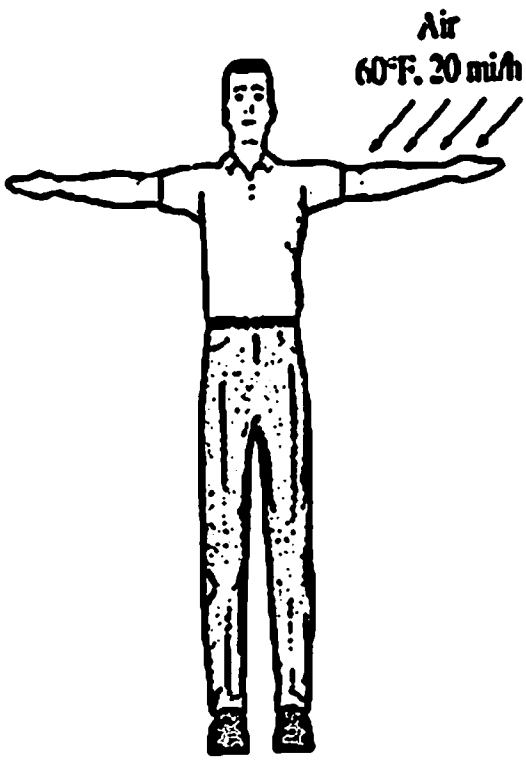


FIGURE P15–65E

Step 1 of 4

Given data:

Pressure of outside air, $P = 1\text{atm}$

Temperature of outside air, $T = 60°\text{F}$

Velocity of air, $V = 20\,\text{mi/h}$

Length of arm, $L = 2\text{ft}$

Diameter of cylinder, $D = 3\text{in.}$

Step 2 of 4

From the table of properties ideal gas of air

At $T = 60°\text{F}$

Density of air is $\rho = 0.07633\text{lbm/ft}^3$

Kinematic viscosity is

$$\nu = 0.5718\text{ft}^2/\text{h}$$

$$= \left(0.5718\text{ft}^2/\text{h}\right)\left(1\text{h}/3600\,\text{s}\right)$$

$$= 1.588\times10^{-4}\,\text{ft}^2/\text{s}$$

Reynolds number is given by

$$\text{Re} = \frac{VD}{\nu}$$

FIGURE P15–65E

Step 1 of 4

Given data:

Pressure of outside air, $P = 1\text{atm}$

Temperature of outside air, $T = 60°\text{F}$

Velocity of air, $V = 20\,\text{mi/h}$

Length of arm, $L = 2\text{ft}$

Diameter of cylinder, $D = 3\text{in.}$

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From the table of properties ideal gas of air

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$$= 1.588\times10^{-4}\,\text{ft}^2/\text{s}$$

Reynolds number is given by

$$\text{Re} = \frac{VD}{\nu}$$

$$= \left[1\times1\times\frac{0.07633\left[20\,\text{mi/h}\times\frac{1.46\,\text{ft/s}}{1\,\text{mi/h}}\right]}{2}\right]\left(\frac{1\text{lbf}}{32.2\text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 1.020\text{lbf}$$

Therefore, drag force acting on both the arm is $\boxed{F = 1.020\text{lbf}}$.

$$= \left[1\times1\times\frac{0.07633\left[20\,\text{mi/h}\times\frac{1.46\,\text{ft/s}}{1\,\text{mi/h}}\right]}{2}\right]\left(\frac{1\text{lbf}}{32.2\text{lbm}\cdot\text{ft/s}^2}\right)$$

$$= 1.020\text{lbf}$$

Therefore, drag force acting on both the arm is $\boxed{F = 1.020\text{lbf}}$.

Question: 15.066P

A 6-mm-diameter electrical transmission line is exposed to windy air. Determine the drag force exerted on a 160-m-long section of the wire during a windy day when the air is at 1 atm and 15°C and the wind is blowing across the transmission line at 65 km/h.

Step 1 of 5

Given data:

Diameter of the electrical transmission line $D = 6\text{mm}$

Length of the wire $L = 160\text{m}$

Velocity of wind $V = 65\text{km/h}$

Pressure of the air $P = 1\text{atm}$

Temperature of the air $T = 15°\text{C}$

Step 2 of 5

The drag force can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag force

ρ is the density of air

A is the frontal area of the transmission line

V is the velocity of the wind

Step 3 of 5

From the properties of air table at 1 atm pressure

The density of air at 1atm and 15°C $\rho = 1.225\,\text{kg/m}^3$

Kinematic viscosity of air at 1atm and 15°C $\nu = 1.470 \times 10^{-5}\,\text{m}^2/\text{s}$

Then, calculating the Reynolds number using the formula given below

$$\text{Re} = \frac{VD}{\nu} \qquad \ldots\ldots (2)$$

Where, V is the velocity of the hail.

D is the diameter of the hail.

ν is the kinematic viscosity of air.

Step 4 of 5

Here, diameter of the wire,

$D = 6\text{ mm}$

$= 0.006\text{ m}$

Now, velocity of wind,

$$V = \frac{65}{2}\,\text{km/h}$$

Where, C_D is the drag force

ρ is the density of air

A is the frontal area of the transmission line

V is the velocity of the wind

Step 3 of 5

From the properties of air table at 1 atm pressure

The density of air at 1atm and 15°C $\rho = 1.225\,\text{kg/m}^3$

Kinematic viscosity of air at 1atm and 15°C $\nu = 1.470 \times 10^{-5}\,\text{m}^2/\text{s}$

Then, calculating the Reynolds number using the formula given below

$$\text{Re} = \frac{VD}{\nu} \qquad \ldots\ldots (2)$$

Where, V is the velocity of the hail.

D is the diameter of the hail.

ν is the kinematic viscosity of air.

Step 4 of 5

Here, diameter of the wire,

$D = 6\text{ mm}$

$= 0.006\text{ m}$

Now, velocity of wind,

Question: 15.067P

Air is flowing past a nonsymmetrical airfoil at zero angle of attack. Is the (*a*) lift and (*b*) drag acting on the airfoil zero or nonzero?

Step 1 of 2

a) When air is flowing past a symmetrical airfoil at zero angle of attack ($\alpha = 0$) the lift will be zero. Because of symmetry, and the stagnation points are at the leading and trailing edges.

Step 2 of 2

b) When air is flowing past a symmetrical airfoil at zero angle of attack ($\alpha = 0$) the drag will be nonzero.

Question: 15.068P

Why is the contribution of viscous effects to lift usually negligible for airfoils?

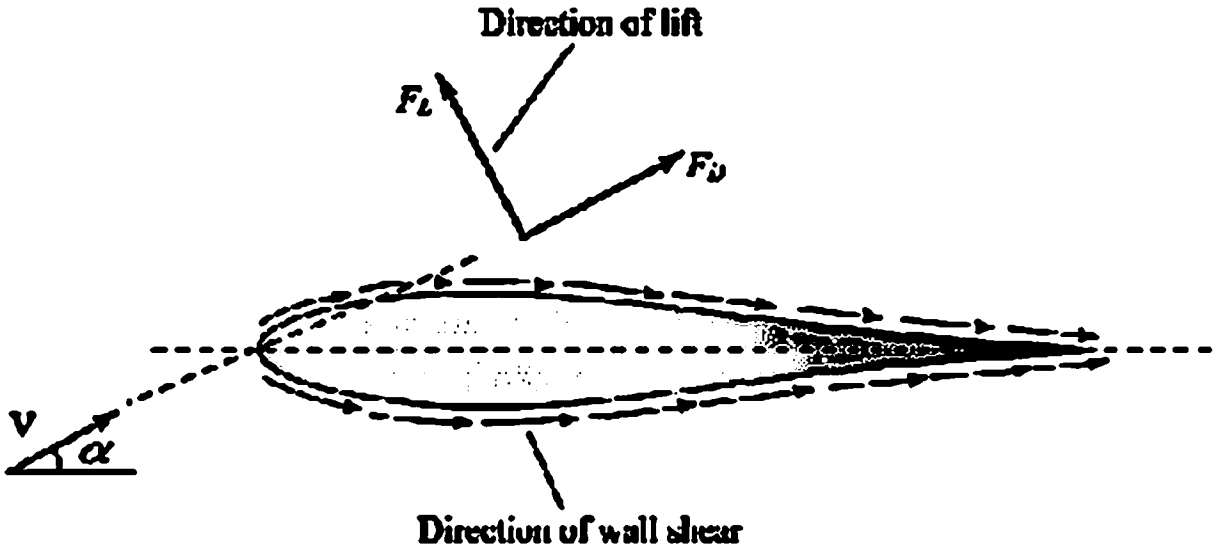
Step 1 of 2

Contribution of viscous effects to lift is negligible because the airplane bodies are streamlined and therefore the direction of wall shear will be parallel to the surface of the body and almost normal to the direction of lift.

The lift slope is depends on the viscous effect, while the presence of viscous effects tend to decrease the lift slope therefore it seems reasonable to ignore the very thin boundary layer on the airfoil and calculate the pressure distribution around the airfoil from the relatively simple potential flow theory for which net viscous forces are zero for flow past an airfoil.

Step 2 of 2

The figure below represents an air foil, the direction of different forces are shown.



Question: 15.069P

Air is flowing past a symmetrical airfoil at an angle of attack of 5°. Is the (*a*) lift and (*b*) drag acting on the airfoil zero or nonzero?

Step 1 of 2

a) If air is flowing past a symmetrical airfoil at an angle of attack of 5° ($\alpha = 5°$) lift acting on the airfoil will be nonzero. Therefore, planes with symmetrical wing sections must fly with their wings at higher angles of attack in order to produce the same lift.

Step 2 of 2

b) If air is flowing past a symmetrical airfoil at an angle of attack of 5° ($\alpha = 5°$) drag acting on the airfoil will be nonzero.

Question: 15.070P

What is stall? What causes an airfoil to stall? Why are commercial aircraft not allowed to fly at conditions near stall?

Step 1 of 1

The lift coefficient (C_l) of an aircraft will increase with increase in angle of attack, but this increase is only up to a particular angle of attack, which is approximately 16° ($\alpha = 16°$) beyond that the lift coefficient will decrease drastically. This decrease in lift with increase in angle of attack beyond 16° is called stall. The stall occurs when airflow separates from the upper wing surface this happens when the plane under too higher angle of attack.

Decrease in lift of an aircraft is a serious problem with the safety of the aircraft if an aircraft losses lift it will fall down, so commercial aircrafts are not allowed to fly at a condition near stall. Stall also increases drag. It is caused due to flow separation, and due to flow separation a wide wake is formed on the top surface of the airfoil and causes stall.

Question: 15.071P

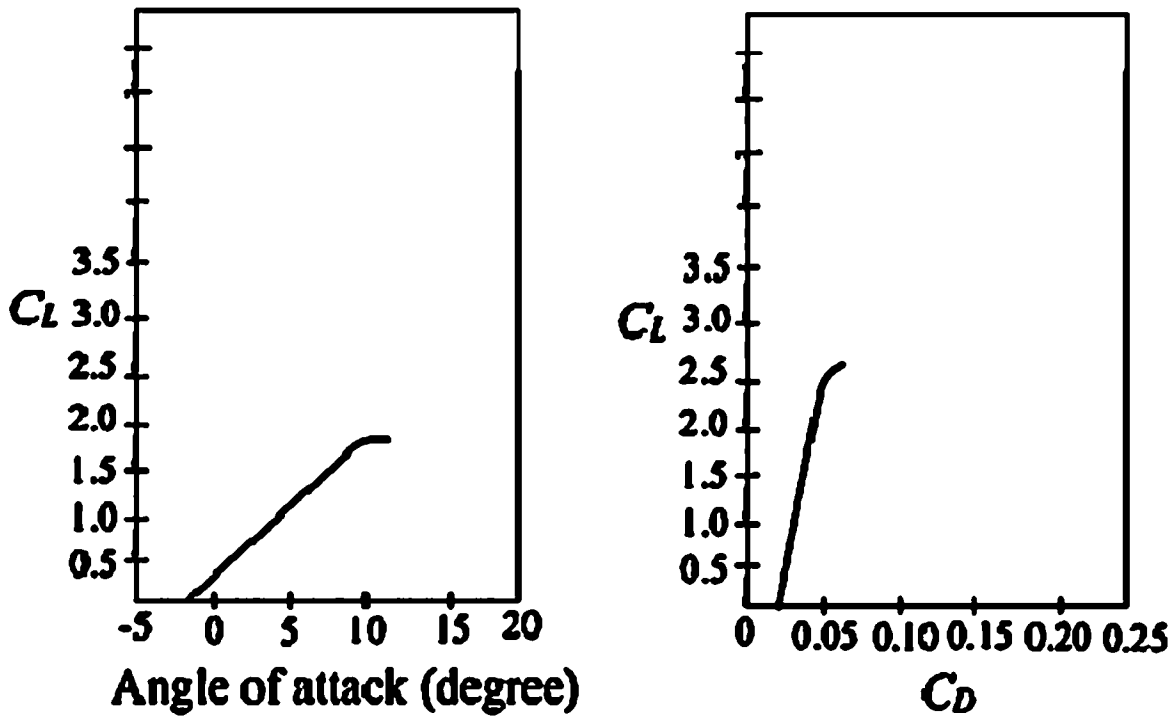
Both the lift and the drag of an airfoil increase with an increase in the angle of attack. In general, which increases at a higher rate, the lift or the drag?

Step 1 of 2

Both lift and drag will increase with an increase in angle of attack but in general lift will increase at a higher rate than drag. The lift coefficient (C_L) of an aircraft will increase with increase in angle of attack, but this increase is only up to a particular angle of attack, which is approximately 16° ($\alpha = 16°$) beyond that the lift coefficient will decrease drastically.

Step 2 of 2

The graph below represents the variations of C_L and C_D at different angle of attacks (α).



Question: 15.072P

Why are flaps used at the leading and trailing edges of the wings of large aircraft, during takeoff and landing? Can an aircraft take off or land without them?

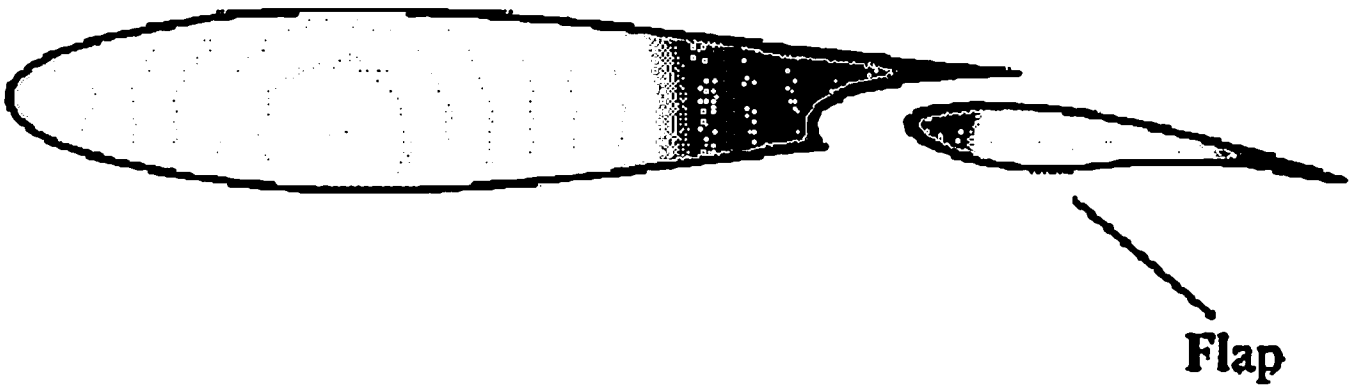
Step 1 of 2

Flaps are used to alter the shape of the wing so that the aircraft could land and takeoff safely at low speed. These moving flaps will increase the lift and drag of the aircraft, an increase in drag will decrease the speed of the aircraft. A movable leading edge and trailing edge flap is commonly used in modern aircrafts. During cruising altitude these flaps will be retracted and come into normal shape to minimize the drag and increase the fuel efficiency.

An aircraft can land and takeoff without the flaps but only at high speed, which is not desirable.

Step 2 of 2

The figure shown below represents the wing of an aircraft with a flap attached to it.



Question: 15.073P

How do flaps affect the lift and the drag of wings?

Step 1 of 1

Flaps are used to alter the shape of the wing so that the aircraft could land and takeoff safely at low speed. These moving flaps will increase both lift and drag of the aircraft, an increase in drag will decrease the speed of the aircraft.

Question: 15.074P

What is the effect of wing tip vortices (the air circulation from the lower part of the wings to the upper part) on the drag and the lift?

Step 1 of 1

Wing tip vortices will increase the drag and decrease the lift. Since the air circulate from the lower part of the wing to the upper part of the wing it will create downdraft, this downdraft is very dangerous for small aircrafts.

Question: 15.075P

What is induced drag on wings? Can induced drag be minimized by using long and narrow wings or short and wide wings?

Step 1 of 1

When the tip vortices of the wing come into contact with the free stream some forces will be developed on the wing tip, these forces will act in all directions including the flow direction. The component of the force in the flow direction will add to the drag and this drag is called induced drag. Induced drag can be minimized by using long and narrow wings.

Question: 15.076P

Explain why endplates or winglets are added to some airplane wings.

Step 1 of 1

Endplates and winglets are attached to the tips of the wings perpendicular to the top surface to reduce the strength of the tip vortices and the induced drag. The endplates will block the leakage around the wing tips and this will result in the reduction in strength of the tip vortices and the induced drag.

Question: 15.077P

Consider an aircraft that takes off at 190 km/h when it is fully loaded. If the weight of the aircraft is increased by 20 percent as a result of overloading, determine the speed at which the overloaded aircraft will take off.

Step 1 of 6

Given data:

Mass of the aircraft when fully loaded is m_1.

Take off speed is $V_1 = 190$ km/h.

Let V_2 be the takeoff speed of the airplane when the weight of the airplane is increased.

Step 2 of 6

An aircraft can takeoff when lift is equal to the total weight of the airplane.

$F_L = W$.

Lift force,

$$F_L = \frac{1}{2} C_L \rho A V^2.$$

Where, C_L is the lift coefficient.

ρ is the density of air.

A is the wing area.

V is the take off velocity.

Therefore, the equation becomes

$$W = \frac{1}{2} C_L \rho A V^2$$

$$V^2 = \frac{2W}{C_L \rho A}$$

$$V = \sqrt{\frac{2W}{C_L \rho A}} \qquad \ldots\ldots (1)$$

Step 3 of 6

In equation (1) A C_L and ρ are constant. Therefore the takeoff velocity is directly proportional to the weight of the aircraft.

Now, the takeoff velocity of aircraft when the aircraft is fully loaded,

$$V_1 = \sqrt{\frac{2W_1}{C_L \rho A}} \qquad \ldots\ldots (2)$$

Where, W_1 is the weight of the aircraft when fully loaded.

Now, the takeoff velocity of aircraft when the weight of the aircraft is increased by 20%.

$$V_2 = \sqrt{\frac{2W_2}{C_L \rho A}} \qquad \ldots\ldots (3)$$

Where, W_2 is the weight of the aircraft when the weight of the aircraft is increased by 20%.

Step 4 of 6

Divide equation (3) by (2).

$$\frac{V_2}{V_1} = \frac{\sqrt{\frac{2W_2}{C_L \rho A}}}{\sqrt{\frac{2W_1}{C_L \rho A}}}$$

$$= \frac{\sqrt{W_2}}{\sqrt{W_1}}$$

But we know that weight is $W = mg$.

Where,

m is the mass.

g is the acceleration due to gravity.

Step 5 of 6

Therefore,

$$\frac{V_2}{V_1} = \frac{\sqrt{m_2 g}}{\sqrt{m_1 g}}$$

$$= \frac{\sqrt{m_2}}{\sqrt{m_1}}$$

$$V_2 = V_1 \frac{\sqrt{m_2}}{\sqrt{m_1}} \qquad \ldots\ldots (4)$$

Where, m_2 is the mass of airplane when the weight of the airplane is increased by 20%.

$$m_2 = m_1 + \left(\frac{20}{100}\right) m_1$$

$$= \left(\frac{120}{100}\right) m_1$$

$$= 1.2 m_1$$

Step 6 of 6

Substituting the values of m_1 and m_2 in equation (4)

$$V_2 = V_1 \left(\frac{\sqrt{1.2 m_1}}{\sqrt{m_1}}\right)$$

We know that $V_1 = 190$ km/h.

$$V_2 = 190 \left(\frac{\sqrt{1.2 m_1}}{\sqrt{m_1}}\right)$$

$$= 190 \left(\sqrt{1.2}\right)$$

$$= 208.1 \text{ km/h}$$

Therefore the takeoff speed of the airplane when the weight of the airplane is increased by 20% $\boxed{V_2 = 208 \text{ km/h}}$.

Question: 15.078P pg 1

Consider an airplane whose takeoff speed is 220 km/h and that takes 15 s to take off at sea level. For an airport at an elevation of 1600 m (such as Denver), determine (*a*) the take off speed, (*b*) the takeoff time, and (*c*) the additional runway length required for this airplane. Assume constant acceleration for both cases.

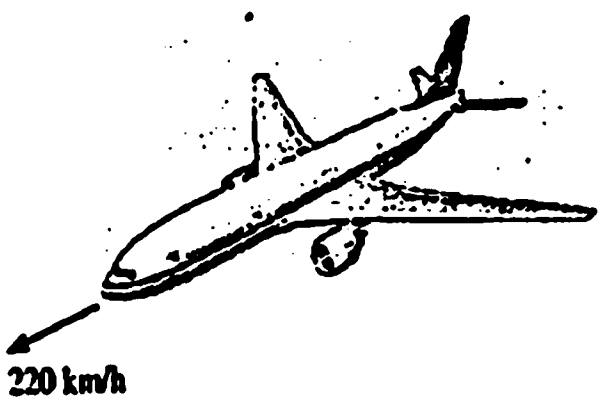


FIGURE P15–78

Step 1 of 10

Given data:

Take off speed at sea level, $V_1 = 220$ km/h

Time taken to takeoff at sea level, $t_1 = 15$ sec

Step 2 of 10

a) An aircraft can takeoff when lift is equal to the total weight of the airplane.

Means that, $F_L = W$

Lift force, $F_L = \frac{1}{2} C_L \rho A V^2$

Where, C_L is the lift coefficient

ρ is the density of air.

A is the wing area.

V is the take off velocity.

Therefore,

$$W = \frac{1}{2} C_L \rho A V^2$$

$$V^2 = \frac{2W}{C_L \rho A}$$

$$V = \sqrt{\frac{2W}{C_L \rho A}} \qquad \ldots\ldots (1)$$

Step 3 of 10

In equation (1) W, C_L and A are constants.

If the altitude is changes the density of air will also change.

Therefore, takeoff velocity is inversely proportional to the density of air.

Density of air at sea level, $\rho_1 = 1.225$ kg/m^3

Density of air at an altitude of 1600 m, $\rho_2 = 1.048$ kg/m^3.

Now, takeoff velocity of airplane at sea level,

$$V_1 = \sqrt{\frac{2W}{C_L \rho_1 A}} \qquad \ldots\ldots (2)$$

Takeoff velocity of airplane at an altitude of 1600 m,

$$V_2 = \sqrt{\frac{2W}{C_L \rho_2 A}} \qquad \ldots\ldots (3)$$

Step 4 of 10

Divide equation (3) by (2)

$$\frac{V_2}{V_1} = \frac{\sqrt{\frac{2W}{C_L \rho_2 A}}}{\sqrt{\frac{2W}{C_L \rho_1 A}}}$$

$$= \frac{\sqrt{\rho_1}}{\sqrt{\rho_2}}$$

$$V_2 = V_1 \sqrt{\frac{\rho_1}{\rho_2}}$$

We know that,

$V_1 = 220$ km/h.

$$V_2 = 220 \sqrt{\frac{1.225}{1.048}}$$

$$= 237.85$$

$$= 238 \text{ km/h}$$

Therefore takeoff velocity of aircraft at an altitude of 1600 m, $\boxed{V_2 = 238 \text{ km/h}}$

Step 5 of 10

b)

Acceleration of the aircraft at the sea level,

$$a_1 = \frac{\Delta V_1}{\Delta t_1} \qquad \ldots\ldots (4)$$

Where, a_1 is the acceleration at sea level.

ΔV_1 is the change in velocity during takeoff at sea level.

Δt_1 time for takeoff at sea level.

$$a_1 = \frac{220 - 0}{15}$$

$$= 14.67$$

$$= 14.67\left(\frac{5}{18}\right) \text{m/s}^2$$

$$= 4.075 \text{ m/s}^2$$

Step 6 of 10

Acceleration of the aircraft at an altitude of 1600 m

$$a_2 = \frac{\Delta V_2}{\Delta t_2} \qquad \ldots\ldots (5)$$

Where, a_1 is the acceleration at an altitude of 1600 m.

Acceleration of the aircraft at an altitude of 1600 m

$$a_2 = \frac{\Delta V_2}{\Delta t_2} \qquad \ldots\ldots (5)$$

Where, a_1 is the acceleration at an altitude of 1600 m.

ΔV_1 Is the change in velocity during takeoff at an altitude of 1600 m.

Δt_1 Time for takeoff at an altitude of 1600 m.

$$a_2 = \frac{238 - 0}{\Delta t_2}$$

Since the acceleration is constant in both the cases, $a_1 = a_2$.

Step 7 of 10

Therefore,

$$4.075 = \frac{238\left(\frac{5}{18}\right) - 0}{\Delta t_2}$$

$$\Delta t_2 = \frac{66.11}{4.075}$$

$$= 16.2 \text{ sec}$$

Therefore the time required to takeoff at an altitude of 1600 m, $\boxed{\Delta t_2 = 16.2 \text{ sec}}$

Step 8 of 10

c) Now we have to find out the additional runway length required for the airplane.
Runway length required for the airplane to takeoff at sea level,

$$S_1 = V_0 t_1 + \frac{1}{2} a_1 t_1^2 \qquad \ldots\ldots (6)$$

Where, S_1 is the length of the runway at sea level

V_0 is the initial velocity of the airplane at sea level.

t_1 is the time required to takeoff at sea level.

a_1 is the acceleration of the aircraft at sea level.

Here initial velocity of the airplane is zero, $V_0 = 0$.

$t_1 = 15$ sec

$a_1 = 4.075 \text{ m/s}^2$

Substitute all these values in equation (6)

$$S_1 = 0 + \frac{1}{2} \times 4.075 \times (15)^2$$

$$= 458.44 \text{ m}$$

Step 9 of 10

Runway length required for the airplane to takeoff at an altitude of 1600 m,

$$S_2 = V_0 t_2 + \frac{1}{2} a_2 t_2^2 \qquad \ldots\ldots (6)$$

Where, S_2 is the length of the runway at an altitude of 1600 m.

V_0 is the initial velocity of the airplane at an altitude of 1600 m.

t_2 is the time required to takeoff at an altitude of 1600 m.

a_2 is the acceleration of the aircraft at an altitude of 1600 m.

Here initial velocity of the airplane is zero, $V_0 = 0$.

$t_2 = 16.2$ sec

$a_2 = 4.075 \text{ m/s}^2$.

Substitute all these values in equation (6)

$$S_2 = 0 + \frac{1}{2} \times 4.075 \times (16.2)^2$$

$$= 534.72 \text{ m}$$

Step 10 of 10

Now, the additional runway length required for this airplane to takeoff at an altitude of 1600m is,

$$S_{additional} = S_2 - S_1$$

$$= 534.72 - 458.44$$

$$= 76.28 \text{ m}$$

Therefore, the additional runway length required for this airplane to takeoff at an altitude of 1600 m, $\boxed{S_{additional} = 76.28 \text{ m}}$

Question: 15.079P

An airplane is consuming fuel at a rate of 7 gal/min when cruising at a constant altitude of 10,000 ft at constant speed. Assuming the drag coefficient and the engine efficiency to remain the same, determine the rate of fuel consumption at an altitude of 30,000 ft at the same speed.

Step 1 of 5

Given data:

Consumption of fuel by the airplane at an altitude of 3000 m, $f_1 = 7\,\text{gal/min}$

Cruising Altitude $h_1 = 10000$ ft

Step 2 of 5

When an aircraft cruises at a constant altitude the acceleration will be zero as a result the net force acting on the aircraft is zero.

Therefore the thrust developed by the engine is used to overcome the drag only as a result the thrust produced will be equal to the drag force acting, (Thrust $= F_D$)

Therefore power required to overcome drag,

$P = \text{Thrust} \times V$

$= F_D \times V$

Where, V is the velocity of airplane.

F_D is the drag force.

Here, $F_D = \frac{1}{2} C_D \rho A V^2$

Substitute this value in the above equation,

Power, $P = \frac{1}{2} C_D \rho A V^2 \times V$ ..... (1)

Where, C_D is the drag coefficient.

ρ is the density of air.

A is the wing area.

V is the velocity of aircraft.

Step 3 of 5

Now, power is also the product of rate of fuel energy supplied and the propulsive efficiency.

$P = \eta f_{c1} \text{HV}$ ..... (2)

(Rate of fuel energy supplied $= f_c \text{HV}$)

Where, η is the efficiency.

f_{c1} is the rate of fuel consumption at an altitude of

HV is the heating value.

Equate equations (1) and (2).

$$\frac{1}{2} C_D \rho A V^2 \times V = \eta f_{c1} \text{HV}$$

$$f_{c1} = \frac{\frac{1}{2} C_D \rho_1 A V^2 \times V}{\eta \text{HV}} \quad \text{..... (3)}$$

ρ_1 is the density of air at $h_1 = 10000$ ft

Step 4 of 5

Rate of fuel consumption at an altitude of $h_2 = 30000$ ft

$$f_{c2} = \frac{\frac{1}{2} C_D \rho_2 A V^2 \times V}{\eta \text{HV}} \quad \text{..... (4)}$$

Where, η is the efficiency

f_{c2} is the rate of fuel consumption at an altitude of $h_2 = 30000$ ft

HV is the heating value.

ρ_2 is the density of air at $h_2 = 30000$ ft

Step 5 of 5

From equation (3) and (4) we can understand that fuel consumption is directly proportional to density of air.

Divide equation (3) by (4).

$$\frac{f_{c1}}{f_{c2}} = \frac{\frac{\frac{1}{2} C_D \rho_1 A V^2 \times V}{\eta \text{HV}}}{\frac{\frac{1}{2} C_D \rho_2 A V^2 \times V}{\eta \text{HV}}}$$

$$\frac{f_{c1}}{f_{c2}} = \frac{\rho_1}{\rho_2} \quad \text{..... (5)}$$

Now, from the table 'Properties of the atmosphere at high altitude'

Density of air at $h_1 = 10000$ ft, $\rho_1 = 0.05648\ \text{lbm/ft}^3$

Density of air at $h_2 = 30000$ ft, $\rho_2 = 0.02866\ \text{lbm/ft}^3$

Substitute all the values in equation (5)

$$\frac{7}{f_{c2}} = \frac{0.909}{0.467}$$

$$f_{c2} = \frac{0.05648 \times 7}{0.02866}$$

$= 13.79\ \text{gal/min}$

Therefore, the rate of fuel consumption at an altitude of 30000 ft is $\boxed{f_{c2} = 13.79\ \text{gal/min}}$.

Question: 15.080P

A jumbo jet airplane has a mass of about 400,000 kg when fully loaded with over 400 passengers and takes off at a speed of 250 km/h. Determine the takeoff speed when the airplane has 100 empty seats. Assume each passenger with luggage is 140 kg and the wing and flap settings are maintained the same.

Step 1 of 6

Given data:-

Mass of the aircraft when fully loaded, $m_1 = 400000$ kg.

Take off speed, $V = 250$ km/h.

Mass of each passenger with luggage, $m = 140$ kg.

Number of passengers when the aircraft is fully loaded is 400.

Step 2 of 6

An aircraft can takeoff when lift is equal to the total weight of the airplane.

$F_L = W$.

Lift force, $F_L = \frac{1}{2} C_L \rho A V^2$.

Where, C_L is the lift coefficient.

ρ is the density of air.

A is the wing area.

V is the take off velocity.

Therefore,

$$W = \frac{1}{2} C_L \rho A V^2$$

$$V^2 = \frac{2W}{C_L \rho A}$$

$$V = \sqrt{\frac{2W}{C_L \rho A}} \qquad \ldots\ldots (1)$$

Step 3 of 6

It is given that there is no change in wing and flap settings so area A remains constant in equation (1) also C_L and ρ is also constant. Therefore the takeoff velocity is directly proportional to the weight of the aircraft.

Now, the takeoff velocity of aircraft when the aircraft is fully loaded,

$$V_1 = \sqrt{\frac{2W_1}{C_L \rho A}} \qquad \ldots\ldots (2)$$

Where, W_1 is the weight of the aircraft when fully loaded.

Now, the takeoff velocity of aircraft when the aircraft has 100 empty seats.

$$V_2 = \sqrt{\frac{2W_2}{C_L \rho A}} \qquad \ldots\ldots (3)$$

Where, W_2 is the weight of the aircraft when the aircraft has 100 empty seats.

Step 4 of 6

Divide equation (3) by (2).

$$\frac{V_2}{V_1} = \frac{\sqrt{\frac{2W_2}{C_L \rho A}}}{\sqrt{\frac{2W_1}{C_L \rho A}}}$$

$$= \frac{\sqrt{W_2}}{\sqrt{W_1}}$$

But we know that weight, $W = mg$.

Where, m is the mass.

g is the gravitational acceleration.

Step 5 of 6

Therefore,

$$\frac{V_2}{V_1} = \frac{\sqrt{m_2 g}}{\sqrt{m_1 g}}$$

$$= \frac{\sqrt{m_2}}{\sqrt{m_1}}$$

$$V_2 = V_1 \frac{\sqrt{m_2}}{\sqrt{m_1}} \qquad \ldots\ldots (4)$$

Where, m_2 is the mass of airplane when 100 seats are empty.

$$m_2 = m_1 - m(100)$$

$$= 400000 - 140(100)$$

$$= 400000 - 14000$$

$$= 386000 \text{ kg}$$

Step 6 of 6

Substitute the values of m_1 and m_2 in equation (4)

$$V_2 = V_1 \left(\frac{\sqrt{386000}}{\sqrt{400000}} \right)$$

We know that

$V_1 = 250$ km/h.

$$V_2 = 250 \left(\frac{\sqrt{386000}}{\sqrt{400000}} \right)$$

$$= 245.59 \text{ km/h}$$

$$= 246 \text{ km/h}$$

Therefore the takeoff speed of the airplane when it has 100 empty seats is

$\boxed{V_2 = 246 \text{ km/h}}$.

Question: 15.081P

Reconsider Prob. 15-80. Using EES (or other) software, investigate the effect of passenger count on the takeoff speed of the aircraft. Let the number of passengers vary from 0 to 500 in increments of 50. Tabulate and plot the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

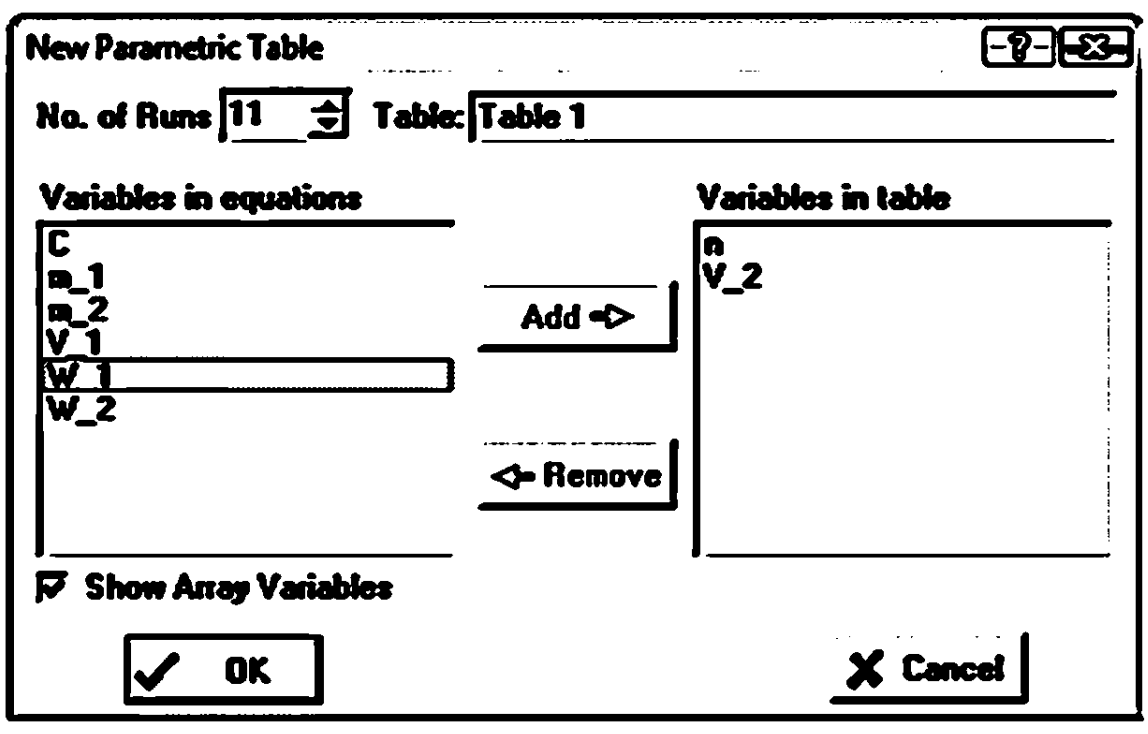
1. **Open EES software**
2. **Create a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in dailog box**

```
m_1=400000[kg]            "Mass of the jumbo jet airplane"
V_1=250[km/hr]            "Takeoff speed"
m_2=140[kg]               "Mass of each passenger"
"n is varied from 0 to 500"  "Number of seats empty"

W_1=m_1*g#                "Weight of the airplane loaded with 500 passengers"
W_2=W_1-m_2*g#*n          "Weight of the airplane when n seats are empty"
"Let C_L*rho*A be 'C' "
W_1=0.5*C*V_1^2           "Lift Force"
W_2=0.5*C*V_2^2
```

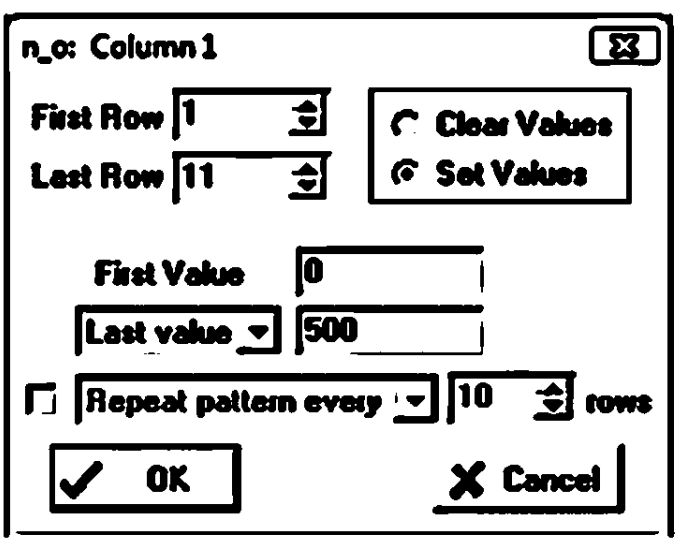
Step 2 of 5

Now to solve the equation for a range of hot surface temperature values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "n" and "V_2" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

To enter the values in the "n" column click on the drop tab in "n" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

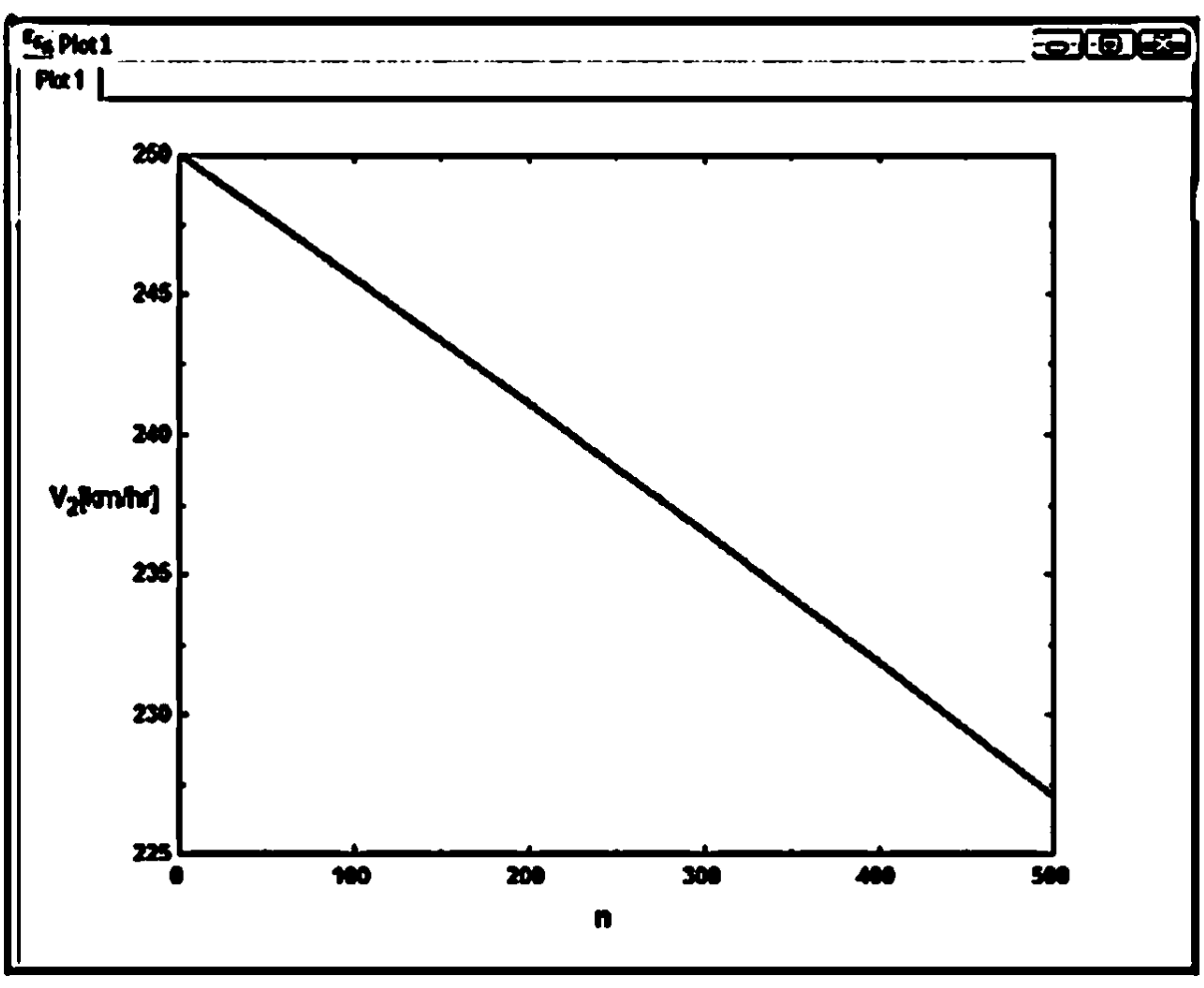
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	n	V_2
Run 1	0	250
Run 2	50	247.8
Run 3	100	245.6
Run 4	150	243.3
Run 5	200	241.1
Run 6	250	238.8
Run 7	300	236.5
Run 8	350	234.2
Run 9	400	231.8
Run 10	450	229.5
Run 11	500	227.1

Step 5 of 5

Inorder to plot the rate of natural convection heat transfer as a function of hot surface temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as "n" and Y-axis as "V_2" and press 'OK'
The Graph will be plotted as shown below:



From the above graph we can interpret that as the number of empty seats increases, the velocity of the plane decreases.

Question: 15.082P

A small aircraft has a wing area of 28 m2, a lift coefficient of 0.45 at takeoff settings, and a total mass of 2500 kg. Determine (*a*) the takeoff speed of this aircraft at sea level at standard atmospheric conditions, (*b*) the wing loading, and (*c*) the required power to maintain a constant cruising speed of 300 km/h for a cruising drag coefficient of 0.035.

Step 1 of 8

Mass of the aircraft, $m = 2500$ kg

Wing area, $A = 28\ \text{m}^2$

Lift coefficient, $C_L = 0.45$

Drag coefficient, $C_D = 0.035$

Step 2 of 8

a) An aircraft can takeoff when lift is equal to the total weight of the airplane.

Means that, $F_L = W$

Lift force, $F_L = \frac{1}{2} C_L \rho A V^2$

Where, C_L is the lift coefficient

ρ is the density of air.

A is the wing area.

V is the take off velocity.

Therefore,

$$W = \frac{1}{2} C_L \rho A V^2$$

$$V^2 = \frac{2W}{C_L \rho A}$$

$$V = \sqrt{\frac{2W}{C_L \rho A}} \qquad \ldots\ldots (1)$$

Step 3 of 8

Density of air at sea level, $\rho = 1.225\ \text{kg/m}^3$

Now, substitute all the known values in equation (1)

$$V = \sqrt{\frac{2mg}{C_L \rho A}}$$

$$= \sqrt{\frac{2 \times 2500 \times 9.81}{0.45 \times 1.225 \times 28}}$$

$$= \sqrt{3177.84}$$

$$= 56.37\ \text{m/s}$$

Therefore, the takeoff speed of this aircraft at sea level and at standard atmospheric condition, $\boxed{V = 56.37\ \text{m/s}}$

Step 4 of 8

b) Now, we have to find out the wing loading.

Wing loading is the lift force per unit plan form area.

Loading on the wing, $F_{\text{Wing loding}} = \frac{F_L}{A}$ ..... (2)

where, F_L is the lift force.

A is the plan form area.

Planform area is the area seen by a person when looking at the body from above the body in a direction normal to the body.

We know that $F_L = W$.

Since $W = mg$,

$$F_L = mg \qquad \ldots\ldots (3)$$

Where, m is the mass of the airplane.

g is the acceleration due to gravity.

Step 5 of 8

Substitute the value of m and g in equation (3)

$$F_L = 2500(9.81)$$

$$= 24525\ \text{N}$$

Substitute the value of F_L in equation (2)

$$F_{\text{Wing loding}} = \frac{24525}{28}$$

$$= 875.89\ \text{N/m}^2$$

Therefore the wing loading is, $\boxed{F_{\text{Wing loding}} = 875.89\ \text{N/m}^2}$.

Step 6 of 8

c)When an aircraft cruises at a constant altitude the acceleration will be zero as a result the net force acting on the aircraft is zero therefore the thrust developed by the engine is used to overcome the drag only, as a result the thrust produced will be equal to the drag force acting, (Thrust $= F_D$).

We know that the drag force acting,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (4)$$

Where C_D is the drag coefficient.

ρ is the density of air.

A is the wing area.

V is the velocity of airplane.

Step 7 of 8

Here,

$C_D = 0.035$.

$\rho = 1.225\ \text{kg/m}^3$.

$A = 28\ \text{m}^2$.

$V = 300$ km/h

$$= 300\left(\frac{5}{18}\right) \text{m/s}$$

$$= 83.33\ \text{m/s}$$

Substitute all these values in equation (4)

$$F_D = \frac{1}{2} C_D \rho A V^2$$

$$= \frac{1}{2} \times 0.035 \times 1.225 \times 28 \times (83.33)^2$$

$$= 4168.07\ \text{N}$$

Step 8 of 8

The power required to maintain a constant cruising speed of 300 km/h for a cruising drag coefficient of 0.035.

$P =$ Thrust $\times V$

$= F_D \times V$

Where, V is the velocity of airplane.

F_D is the drag force.

$P = 4168.07 \times 83.33$

$= 347325.27$ Watt

$= 347.3$ kW

Therefore the power required to maintain a constant cruising speed of 300 km/h for a cruising drag coefficient of 0.035 is, $\boxed{P = 347.3\ \text{kW}}$

Question: 15.083P

A small airplane has a total mass of 1800 kg and a wing area of 42 m2. Determine the lift and drag coefficients of this airplane while cruising at an altitude of 4000 m at a constant speed of 280 km/h and generating 190 kW of power.

Step 1 of 7

Given data:

Total mass of the airplane, $m = 1800$ kg

Wing area, $A = 42$ m^2

Cruising altitude, 4000 m

Speed of the aircraft, $V = 280$ km/h

Power generated by the aircraft engine, $P = 190$ kW

Step 2 of 7

An aircraft can fly when lift is equal to the total weight of the airplane.

Means that $F_L = W$

Lift force, $F_L = \frac{1}{2} C_L \rho A V^2$

Where, C_L is the lift coefficient

ρ is the density of air

A is the wing area

V is the take off velocity.

Therefore,

$$W = \frac{1}{2} C_L \rho A V^2$$

$$V^2 = \frac{2W}{C_L \rho A}$$

$$V = \sqrt{\frac{2W}{C_L \rho A}} \quad \text{..... (1)}$$

Step 3 of 7

Density of air at $h = 4000$ m, $\rho = 0.8194$ kg/m^3

Speed of the aircraft,

$V = 280$ km/h

$= 280\left(\frac{5}{18}\right)$ m/s

$= 77.8$ m/s

Weight,

$W = mg$

$= 1800 \times 9.81$

$= 17658$ N

Wing area, $A = 42$ m^2

Step 4 of 7

Substitute all these values in equation (1)

$$V = \sqrt{\frac{2W}{C_L \rho A}}$$

$$77.8 = \sqrt{\frac{2 \times 17658}{C_L \times 0.8194 \times 42}}$$

Squaring on both sides,

$$(77.8)^2 = \left(\sqrt{\frac{2 \times 17658}{C_L \times 0.8194 \times 42}}\right)^2$$

$$6052.84 = \frac{2 \times 17658}{C_L \times 0.8194 \times 42}$$

$$C_L = \frac{2 \times 17658}{6052.84 \times 0.8194 \times 42}$$

$$C_L = \frac{2 \times 17658}{6052.84 \times 0.8194 \times 42}$$

$= 0.17$

Therefore the lift coefficient, $\boxed{C_L = 0.17}$

Step 5 of 7

When an aircraft cruises at a constant altitude the acceleration will be zero as a result the net force acting on the aircraft is zero therefore the thrust developed by the engine is used to overcome the drag only, as a result the thrust produced will be equal to the drag force acting. (Thrust $= F_D$).

Power, $P = F_D \times V$ ..... (2)

Where, F_D is the drag force.

V is the velocity of the airplane.

We know that the drag force acting,

$$F_D = \frac{1}{2} C_D \rho A V^2 \quad \text{..... (3)}$$

Where C_D is the drag coefficient.

ρ is the density of air.

A is the wing area.

V is the velocity of airplane.

Step 6 of 7

Substituting the values of ρ, A and V in equation (3),

$$F_D = \frac{1}{2} C_D \left\{0.8194 \times 42 \times (77.8)^2\right\}$$

$= 104153.64\, C_D$

Power,

$P = 190$ kW

$= 190 \times 10^3$ Watt

Step 7 of 7

Now, substitute the value of P, F_D and V in equation (2),

$190 \times 10^3 = 104153.64\, C_D \times 77.8$

$$C_D = \frac{190 \times 10^3}{104153.64 \times 77.8}$$

$= 0.02345$

Therefore the drag coefficient, $\boxed{C_D = 0.02345}$

Question: 15.084P

The NACA 64(1)-412 airfoil has a lift-to-drag ratio of 50 at 0° angle of attack, as shown in Fig. 15-43.,At what angle of attack does this ratio increase to 80?

Step 1 of 1

From the graph, 'The variation of the lift-to-drag ratio with angle of attack for a two-dimensional airfoil' for an angle of attack of 3 the lift to drag ratio becomes 80.

$\boxed{\frac{C_L}{C_D} = 80}$ When $\alpha = 3°$.

Question: 15.085P

Consider a light plane that has a total weight of 15,000 N and a wing area of 46 m2 and whose wings resemble ϕe NAC A 23012 airfoil with no flaps. Using data from pig. 15-45, determine the takeoff speed at an angle of attack of 5° at sea level. Also determine the stall speed.

Step 1 of 6

Given data:
Weight of the plane is $W = 15000\ \text{N}$.
Wing area is $A = 46\ \text{m}^2$.
Angle of attack is $\alpha = 5°$.

Step 2 of 6

Taking density of air at sea level is $\rho = 1.225\ \text{kg/m}^3$.
From the graph, 'Effect of flaps on the lift and drag coefficients of an airfoil'

For an angle of attack of $\alpha = 5°$,
Lift coefficient, $C_L = 0.58$
Drag coefficient, $C_D = 0.02$

Step 3 of 6

There was a problem loading this step. We have been notified and we will fix it shortly.

Step 4 of 6

Substituting all values in equation (1)

$$V_{takeoff} = \sqrt{\frac{2 \times 15000}{0.58 \times 1.225 \times 46}}$$
$$= 30.29\ \text{m/s}$$
$$= 30.29\left(\frac{18}{5}\right)\ \text{km/h}$$
$$= 109.04\ \text{km/h}$$

Therefore the takeoff velocity is $\boxed{V_{takeoff} = 109.04\ \text{km/h}}$.

Step 5 of 6

Now, the stall speed can be determined by using the formula given below,

$$V_{stall} = \sqrt{\frac{2W}{C_{L\max}\rho A}} \qquad \ldots\ldots (2)$$

Where, $C_{L\max}$ is the maximum lift coefficient.
ρ is the density of air.
A is the wing area.
W is the weight of the airplane.

Step 6 of 6

Here, $C_{L\max} = 1.52$
Substituting all values in equation (2)

$$V_{stall} = \sqrt{\frac{2W}{C_{L\max}\rho A}}$$
$$= \sqrt{\frac{2 \times 15000}{1.52 \times 1.225 \times 46}}$$
$$= 18.72\ \text{m/s}$$
$$= 18.72\left(\frac{18}{5}\right)\ \text{km/h}$$
$$= 67.4\ \text{km/h}$$

Therefore the stall speed is $\boxed{V_{stall} = 67.4\ \text{km/h}}$.

Question: 15.086P

An airplane has a mass of 50,000 kg, a wing W area of 300 m2, a maximum lift coefficient of 3.2, and a cruising drag coefficient of 0.03 at an altitude of 12,000 m. Determine (*a*) the takeoff speed at sea level, assuming it is 20 percent over the stall speed, and (*b*) the thrust that the engines must deliver for a cruising speed of 700 km/h.

Step 1 of 3

Mass of the airplane, $m = 50000\text{kg}$

Area of the wing, $a = 300\ \text{m}^2$

Maximum lift coefficient, $C_{L,\max} = 3.2$

Drag coefficient, $C_D = 0.03$

Altitude, $h = 12000\text{m}$

Step 2 of 3

From the table of properties of the atmosphere at high altitude,

Density of standard air at sea level, $\rho_1 = 1.225\ \text{kg/m}^3$

Density of standard air at an altitude of 12,000 m, $\rho_2 = 0.312\text{kg/m}^3$

(*a*) An aircraft will takeoff when lift equals the total weight.

Therefore,

$$W = F_L$$

$$= \frac{1}{2} C_L \rho V^2 A$$

So,

$$V = \sqrt{\frac{2W}{\rho C_L A}}$$

$$= \sqrt{\frac{2mg}{\rho C_L A}}$$

The stall velocity is determined by using the maximum lift coefficient.

Hence,

$$V_{\min} = \sqrt{\frac{2mg}{\rho C_{L,\max} A}}$$

$$= \sqrt{\frac{2(50000 \times 9.81)}{1.225 \times 3.2 \times 300}}$$

$$= 28.88\,\text{m/s}$$

Then the "safe" minimum velocity to avoid the stall region,

$$V_{\min,\text{safe}} = 1.2 V_{\min}$$

$$= 1.2 \times (28.9)$$

$$= 34.7\ \text{m/s}$$

$$= 124.77\text{km/hr}$$

Therefore, the take off speed at sea level over the stall speed, $\boxed{V_{\min,\text{safe}} = 124.77\text{km/hr}}$

Step 3 of 3

(*b*) When the aircraft cruises steadily at a constant altitude, the net force acting on the aircraft is zero, and thus the thrust provided by the engines must be equal to the drag force.

Drag force,

$$F_D = C_D A \frac{\rho_2 V^2}{2}$$

$$= 0.03 \times 300\ \text{m}^2 \left(\frac{0.312\ \text{kg/m}^3 \times \left(\frac{700}{3.6}\text{m/s} \right)^2}{2} \right) \left(\frac{1\,\text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2} \right)$$

$$= 53.08\ \text{kN}$$

So,

$$\text{Power} = \text{Thrust} \times \text{Velocity}$$

$$= F_D \times V$$

$$= (53.08\ \text{kN}) \left(\frac{700}{3.6}\ \text{m/s} \right) \left(\frac{1\,\text{kW}}{1\ \text{kN}\cdot\text{m/s}} \right)$$

$$= 10{,}321.11\,\text{kW}$$

Therefore, the thrust force deliver for the crushing speed is $\boxed{\text{Power} = 10{,}321.11\,\text{kW}}$

Question: 15.087P

The passenger compartment of a minivan traveling at 60 mi/h in ambient air at 1 atm and 80°F is modeled as a 3.2-ft-high, 6-ft-wide, and 11-ft-long rectangular box. The airflow over the exterior surfaces is assumed to be turbulent because of the intense vibrations involved. Determine the drag force acting on the top and the two side surfaces of the van and the power required to overcome it.

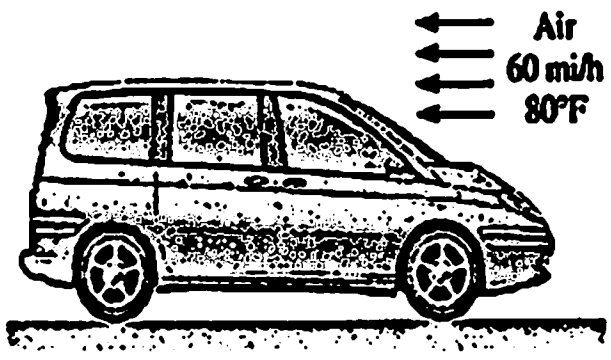


FIGURE P15–87E

Step 1 of 4

Velocity f the minivan, $V_\infty = 60\,\text{mi/h} = 86.82\,\text{ft/s}$

Ambient air pressure, $P = 1\,\text{atm}$

Ambient air temperature, $T = 80°\text{F}$

Height of the minivan, $h = 3.2\,\text{ft}$

With of the minivan, $w = 6\,\text{ft}$

Length of the minivan, $l = 11\,\text{ft}$

The air flow is assumed to be turbulent.

Step 2 of 4

At the given conditions of air the coefficient of kinematic viscosity, $\nu = 1.697\times10^{-4}\,\text{ft}^2/\text{s}$

The density of air at this ambient air conditions , $\rho = 0.074\,\text{lbm/ft}^3$

Reynold's number is given by,

$$\begin{aligned}\text{Re} &= \frac{V_\infty l}{\nu}\\ &= \frac{86.82\times11}{1.697\times10^{-4}}\\ &= 56.277\times10^5\end{aligned}$$

Since the air flow is assumed to be turbulent the coefficient of drag is calculated as,

$$\begin{aligned}C_f &= \frac{0.074}{\text{Re}^{1/5}}\\ &= \frac{0.074}{\left(56.277\times10^5\right)^{1/5}}\\ &= 3.305\times10^{-3}\end{aligned}$$

Step 3 of 4

The total are encountering drag force,

Total area= Area of top + area of sides + area of sides

$$\begin{aligned}A_{Total} &= A_{top} + 2A_{sides}\\ &= (6\times11)+2(3.2\times11)\\ &= 136.4\,\text{ft}^2\end{aligned}$$

Total drag force acting on the minivan die to this surfaces,

$$\begin{aligned}F_D &= C_f \times \frac{1}{2}\rho A_{Total} V_\infty^2\\ &= 3.305\times10^{-3}\times\frac{1}{2}\times0.074\times136.4\times86.82^2\\ &= 125.68\,\text{lbm}\\ &= 3.9\,\text{lbf}\end{aligned}$$

Step 4 of 4

The power required to over come the drag force is given by,

$$\begin{aligned}P &= F_D V_\infty\\ &= 3.9\times86.82\\ &= 338.89\,\text{lbf}\cdot\text{ft/s}\\ &= 0.459\,\text{kW}\end{aligned}$$

The over all drag force due the top and both the side surfaces is found to be $\boxed{3.9\,\text{lbf}}$ and power required by the minivan to over the drag force is obtained as $\boxed{0.459\,\text{kW}}$

Question: 15.088P

A 1.2-m-external-diameter spherical tank is located outdoors at 1 atm and 25°C and is subjected to winds at 48 km/h. Determine the drag force exerted on it by the wind.

Step 1 of 5

Given data:
External diameter of the spherical tank is $D = 1.2$ m.
Velocity of the wind is $V = 48$ km/h.

Step 2 of 5

(From the table of properties of air)
Density of air at 1 atm pressure and 25°C is $\rho = 1.184\ \text{kg/m}^3$.
Kinematic viscosity of air at 1 atm and 25°C is $\nu = 1.562 \times 10^{-5}\ \text{m}^2/\text{s}$.
Here, velocity,

$$V = 48\ \text{km/h}$$
$$= 48\left(\frac{5}{18}\right)\text{m/s}$$
$$= 13.33\ \text{m/s}$$

Step 3 of 5

Now, we have to find out the Reynolds number using the formula given below,

$$\text{Re} = \frac{VD}{\nu}$$

Where, V is the velocity of the air.
D is the diameter of the spherical tank.
ν is the kinematic viscosity.

Substituting all these values in the above equation,

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{13.33 \times 1.2}{1.562 \times 10^{-5}}$$
$$= 1.02 \times 10^{6}$$

From the graph, 'Effect of surface roughness on the drag coefficient of a sphere'
The drag coefficient (C_D) corresponding to the obtained Reynolds number is,

$$C_D = 0.14$$

Step 4 of 5

Now, the drag force acting on the sphere,

$$F_D = \frac{1}{2}C_D\rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag coefficient.
ρ is the density of air.
A is the area of the spherical tank.
V is the velocity of wind.

Here, area,

$$A = \frac{\pi}{4}D^2$$
$$= \frac{\pi}{4}(1.2)^2$$
$$= 1.13\ \text{m}^2$$

Step 5 of 5

Substituting all the values in equation (1).

$$F_D = \frac{1}{2}C_D\rho A V^2$$
$$= \frac{1}{2} \times 0.14 \times 1.184 \times 1.13 \times (13.33)^2$$
$$= 16.6\ \text{N}$$

Therefore the drag force acting on the sphere is $\boxed{F_D = 16.6\ \text{N}}$.

Question: 15.089P pg 1

A 2-m-high, 4-m-wide rectangular advertisement panel is attached to a 4-m-wide, 0.15-m-high rectangular concrete block (density = 2300 kg/m3) by two 5-cm-diameter, 4-m-high (exposed part) poles, as shown in Fig. PI5-89. If the sign is to withstand 150 km/h winds from any direction, determine (*a*) the maximum drag force on the panel, (*b*) the drag force acting on the poles, and (*c*) the minimum length *L* of the concrete block for the panel to resist the winds. Take the density of air to be 1.30 kg/m3.

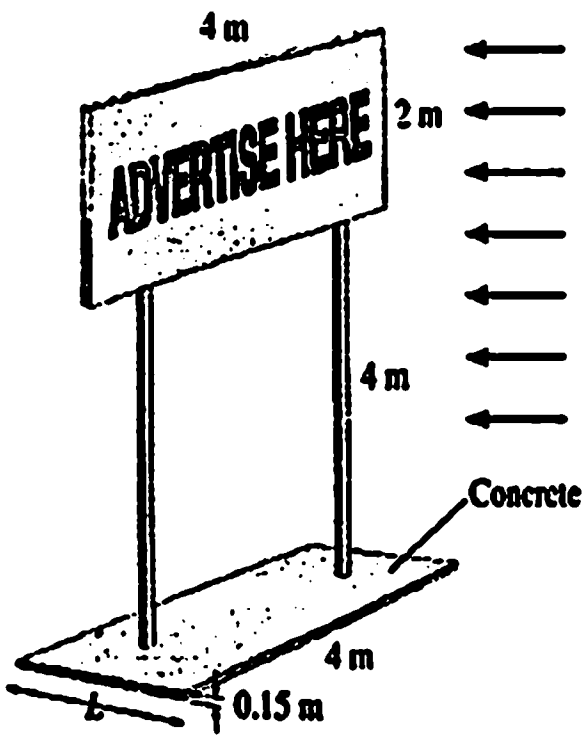


FIGURE P15–89

Step 1 of 9

Given data:

Height of rectangular advertisement panel is $h_1 = 2\ \text{m}$.

Width of rectangular advertisement panel is $w_1 = 4\ \text{m}$.

Height of rectangular concrete block is $h_2 = 0.15\ \text{m}$.

Width of rectangular concrete block is $w_2 = 4\ \text{m}$.

Diameter of the poles is $D = 5\ \text{cm}$.

Height of the poles is $h_3 = 4\ \text{m}$.

Velocity of the wind is $V = 150\ \text{km/h}$.

Density of the concrete block is $\rho_c = 2300\ \text{kg/m}^3$.

Density of the air is $\rho_{air} = 1.30\ \text{kg/m}^3$.

Assume the flow is turbulent.

Step 2 of 9

a) For calculating the maximum drag force on the panel.

Drag force, $F_D = \frac{1}{2} C_D \rho_{air} A V^2$ ..... (1)

Where,

C_D is the drag coefficient.

ρ_{air} is the density of air.

A is the frontal area of the panel.

V is the velocity of the wind.

Step 3 of 9

Here velocity,

$$V = 150\ \text{km/h}$$

$$= 150\left(\frac{5}{18}\right)\ \text{m/s}$$

$$= 41.7\ \text{m/s}$$

Frontal area of the panel,

$$A = h_1 \times w_1$$

$$= 2 \times 4$$

$$= 8\ \text{m}^2$$

In turbulent flow drag coefficient for a thin rectangular plate is, $C_D = 2$

Step 4 of 9

Substituting all these values in equation (1).

$$F_D = \frac{1}{2} C_D \rho_{air} A V^2$$

$$= \frac{1}{2} \times 2 \times 1.30 \times 8 \times (41.7)^2$$

$$= 18084.5\ \text{N}$$

Therefore the maximum drag force on the panel is $\boxed{F_{D\,\text{Panel}} = 18084.5\ \text{N}}$.

Step 5 of 9

b) Now, we have to find out the drag force acting on the pole.

It can be found out using equation (1)

In turbulent flow drag coefficient for a circular rod is, $C_D = 0.3$

Now, the frontal area of the circular rod,

$$A = D \times h_3$$

$$= 0.05 \times 4$$

$$= 0.2\ \text{m}^2$$

Substituting all the values in equation (1)

$$F_D = \frac{1}{2} C_D \rho_{air} A V^2$$

$$= \frac{1}{2} \times 0.3 \times 1.30 \times 0.2 \times (41.7)^2$$

$$= 67.8\ \text{N}$$

Therefore the drag force acting on each pole is $\boxed{F_{D\,\text{pole}} = 67.8\ \text{N}}$.

Step 6 of 9

c) Now, we have to find out the minimum length L of the concrete block for the panel to resist the winds.

The resultant drag force acting on the panel passes through the centre of the panel, the drag force acting on the pole passes through the centre of the pole, the weight of the concrete block acts through the centre of the concrete block.

Assume that the concrete block is tipped due to the wind force, when the concrete block is first tipped the wind loaded side of the panel will be lifted off the ground as

to resist the winds.

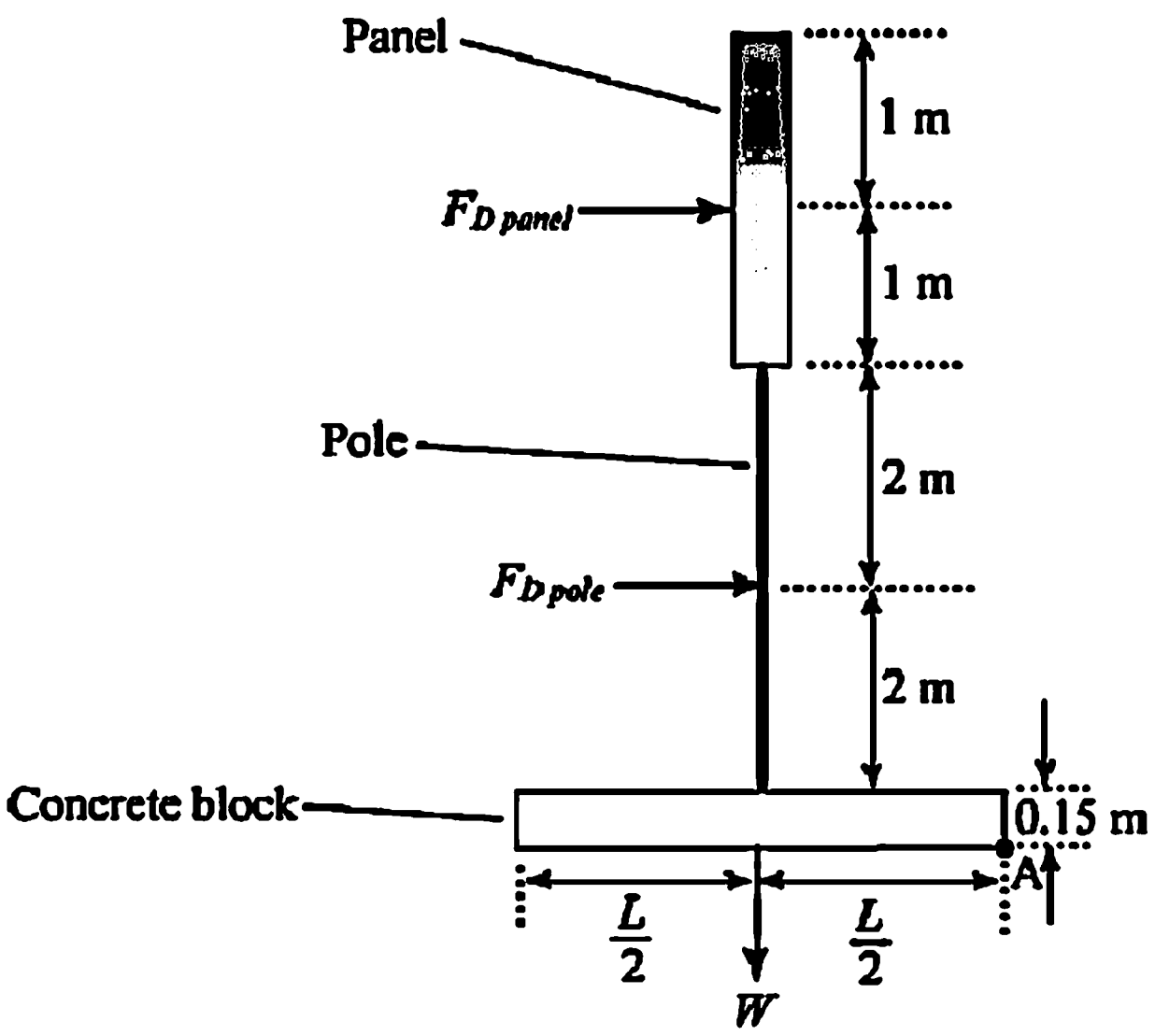
15.089P pg 2

The resultant drag force acting on the panel passes through the centre of the panel, the drag force acting on the pole passes through the centre of the pole, the weight of the concrete block acts through the centre of the concrete block.

Assume that the concrete block is tipped due to the wind force, when the concrete block is first tipped the wind loaded side of the panel will be lifted off the ground as a result all the reaction force from the ground will act on the other side of the block. Take moment about this side and equate it to zero.

Step 7 of 9

Draw the figure of the side view of the advertisement panel.



Step 8 of 9

Taking moment about point A.

$$+\circlearrowleft \sum M_A = 0$$

$$F_{D\,Panel}(5.15) + F_{D\,Pole}(2.15) - W\left(\frac{L}{2}\right) = 0 \qquad \ldots\ldots (2)$$

Now, weight of the concrete block,

$W = mg$

Since,

mass = Density × volume

$= \rho \times V$

$W = \rho_c V g$

Here, volume of the concrete block, $V = L \times h_2 \times w_2$

$V = L \times 0.15 \times 4$

$= 0.6L$

Now, mass,

$m = 2300 \times 0.6L$

$= 1380L$

Now, weight,

$W = 1380L \times 9.81$

$= 13537.8L$

Step 9 of 9

Substituting all the values in equation (2).

$$F_{D\,Panel}(5.15) + F_{D\,Pole}(2.15) - W\left(\frac{L}{2}\right) = 0$$

$$18084.5(5.15) + 67.8(2)(2.15) - 13537.8L\left(\frac{L}{2}\right) = 0$$

$$13537.8L\left(\frac{L}{2}\right) = 93426.715$$

$L^2 = 13.8$

$L = 3.7$ m

Therefore the minimum length of the concrete block for the panel to resist the wind is $\boxed{L = 3.7 \text{ m}}$.

Question: 15.090P

A plastic boat whose bottom surface can be approximated as a 1.5-m-wide, 2-m-long flat surface is to move through water at 15°C at speeds up to 45 km/h. Determine the friction drag exerted on the boat by the water and the power needed to overcome it.

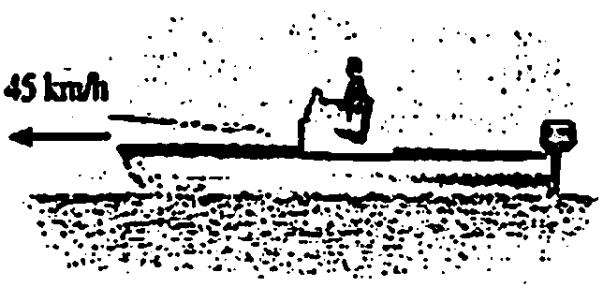


FIGURE P15–90

Step 1 of 5

Given data:-

Width of the bottom surface of the boat, $w = 1.5\ \text{m}$.

Length of the bottom surface of the boat, $L = 2\ \text{m}$.

Temperature of water, $T = 15°\text{C}$.

Speed of the boat, $V = 45\ \text{km/h}$.

Step 2 of 5

Frictional drag exerted on the boat by the water can be found out using the formula given below

$$F_D = \frac{1}{2} C_f \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_f is the friction coefficient.

ρ is the density of water.

A is the frontal area of the bottom surface of the boat.

V is the velocity of the boat.

Here the frictional coefficient C_f is unknown, to find the frictional coefficient use the formula given below,

(The flow is assumed to be turbulent)

$$C_f = \frac{0.074}{\text{Re}_L^{0.2}} \qquad \ldots\ldots (2)$$

Where, Re_L is the Reynolds number.

Here Re_L is unknown which can be found out using the formula given below,

$$\text{Re}_L = \frac{\rho V L}{\mu} \qquad \ldots\ldots (3)$$

Where, ρ is the density of water.

V is the velocity of the boat.

L is the length of the bottom surface of the boat.

μ is the dynamic viscosity of water.

Step 3 of 5

From the table 'properties of saturated water'

Density of water at 15°C, $\rho = 999.1\ \text{kg/m}^3$.

Dynamic viscosity of water at 15°C, $\mu = 1.138 \times 10^{-3}\ \text{kg/m} \cdot \text{s}$

Now, velocity of the boat,

$$V = 45\ \text{km/h}$$

$$= 45\left(\frac{5}{18}\right)\ \text{m/s}$$

$$= 12.5\ \text{m/s}$$

Substitute all the values in equation (3)

$$\text{Re}_L = \frac{\rho V L}{\mu}$$

$$= \frac{999.1 \times 12.5 \times 2}{1.138 \times 10^{-3}}$$

$$= 2.2 \times 10^7$$

Step 4 of 5

Now, substitute the value of Re_L in equation (2)

$$C_f = \frac{0.074}{\text{Re}_L^{0.2}}$$

$$= \frac{0.074}{\left(2.2 \times 10^7\right)^{0.2}}$$

$$= 0.0025$$

Now, frontal area of the bottom surface of the boat,

$$A = w \times L$$

$$= 1.5 \times 2$$

$$= 3\ \text{m}^2$$

Now, substitute the values of C_f, ρ, A, and V in equation (1)

$$F_D = \frac{1}{2} C_f \rho A V^2$$

$$= \frac{1}{2} \times 0.0025 \times 999.1 \times 3 \times (12.5)^2$$

$$= 585.41\ \text{N}$$

Therefore the frictional drag exerted on the boat by the water is, $\boxed{F_D = 585.41\ \text{N}}$.

Step 5 of 5

Now, the power required to overcome this frictional drag has to be determined.

Power required is the product of frictional drag force and velocity of the boat.

Power, $P = F_D \times V \qquad \ldots\ldots (4)$

Where, F_D frictional drag and V is the velocity of boat.

Substitute these values in equation (4)

$$P = F_D \times V$$

$$= 585.41 \times 12.5$$

$$= 7317.625\ \text{Watt}$$

Therefore the power required to overcome the frictional drag is, $\boxed{P = 7.3\ \text{kW}}$.

Question: 15.091P

Reconsider Prob. 15-90. Using EES (or other) NU software, investigate the effect of boat speed on the drag force acting on the bottom surface of the boat, and the power needed to overcome it. Let the boat speed vary from 0 to 100 km/h in increments of 10 km/h. Tabulate and plot the results.

Step 1 of 5

Wide of boat, $W = 1.5\text{m}$

Length of boat, $L = 2\text{m}$

Step 2 of 5

Drag force acting on the boat is

$$F_D = c_D \frac{\rho A V^2}{2}$$

$$= \frac{1000 \times 2 \times 1.5 V^2}{2} \times \frac{0.074}{\text{Re}}$$

$$= \frac{1000 \times 2 \times 1.5 V^2}{2} \times \frac{0.074}{\left(\frac{V \times L}{\nu}\right)}$$

$$= \frac{1000 \times 2 \times 1.5 V^2}{2} \times \frac{0.074}{\left(\frac{V \times 2}{0.001138}\right)}$$

$$= 1500 V^2 \times \frac{5}{18} \text{ m/s}$$

$$= 0.00583 V$$

Step 3 of 5

Power required is

$$P = F_D \times V$$

$$= 0.00583 V \times V$$

$$= 0.0583 V^2$$

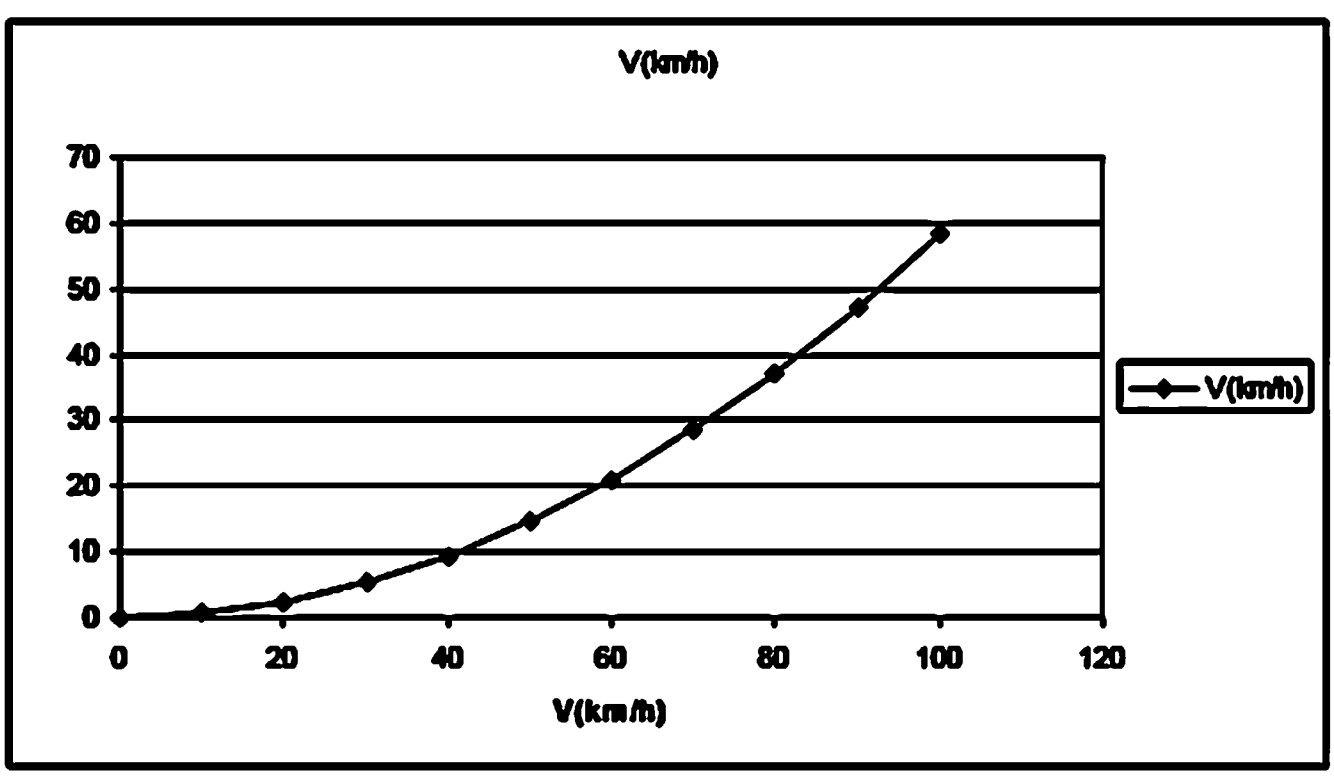
Step 4 of 5

Calculation of velocity and power:

V(km/h)	F(N)	P(kW)
0	0	0
10	0.0583	0.583
20	0.1166	2.332
30	0.1749	5.247
40	0.2332	9.328
50	0.2915	14.575
60	0.3498	20.988
70	0.4081	28.567
80	0.4664	37.312
90	0.5247	47.223
100	0.583	58.3

Step 5 of 5

Power vs. velocity:



Question: 15.092P

The cylindrical chimney of a factory has an external diameter of 1.6 m and is 35 m high. Determine the bending moment at the base of the chimney when winds at 110 km/h are blowing across it. Take the atmospheric conditions to be 20°C and 1 atm.

Step 1 of 8

Given data:-

External diameter of the chimney, $D = 1.6$ m.

Height of the chimney, $h = 35$ m.

Velocity of the wind, $V = 110$ km/h.

Atmospheric conditions,

Temperature of the air, $T = 20°C$.

Pressure of the air, $P = 1$ atm.

Step 2 of 8

(From the table of properties of air)

Density of air at 1 atm pressure and 20°C, $\rho = 1.204$ kg/m^3.

Kinematic viscosity of air at 1 atm and 20°C is $\nu = 1.516 \times 10^{-5}$ m^2/s.

When wind blows across the chimney a drag force will act on the chimney, this drag force will act at the centre of the chimney.

Step 3 of 8

Drag force acting on the chimney can be found out using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \quad \ldots\ldots (1)$$

Where, C_D is the drag coefficient.

ρ is the density of air.

A is the frontal area of the chimney.

V is the velocity of wind.

In the above equation drag coefficient C_D is unknown so in order to find out the drag coefficient we have to find out the Reynolds number.

Step 4 of 8

Reynolds number can be found out using the formula given below,

$$\text{Re} = \frac{VD}{\nu} \quad \ldots\ldots (2)$$

Where, V is the velocity of the air.

D is the diameter of the spherical tank.

ν is the kinematic viscosity.

Here, velocity,

$$V = 110 \text{ km/h}$$

$$= 110\left(\frac{5}{18}\right) \text{m/s}$$

$$= 30.55 \text{ m/s}$$

Step 5 of 8

Substitute all the values in equation (2)

$$\text{Re} = \frac{30.55(1.6)}{1.516 \times 10^{-5}}$$

$$= 3.22 \times 10^6$$

From the graph, 'Effect of surface roughness on the drag coefficient of a sphere'

The drag coefficient (C_D) corresponding to the obtained Reynolds number is,

$$C_D = 0.15$$

Step 6 of 8

Now, frontal area of the chimney,

$$A = h \times D$$

$$= 35 \times 1.6$$

$$= 56 \text{ m}^2$$

Now, substitute all the values in equation (1)

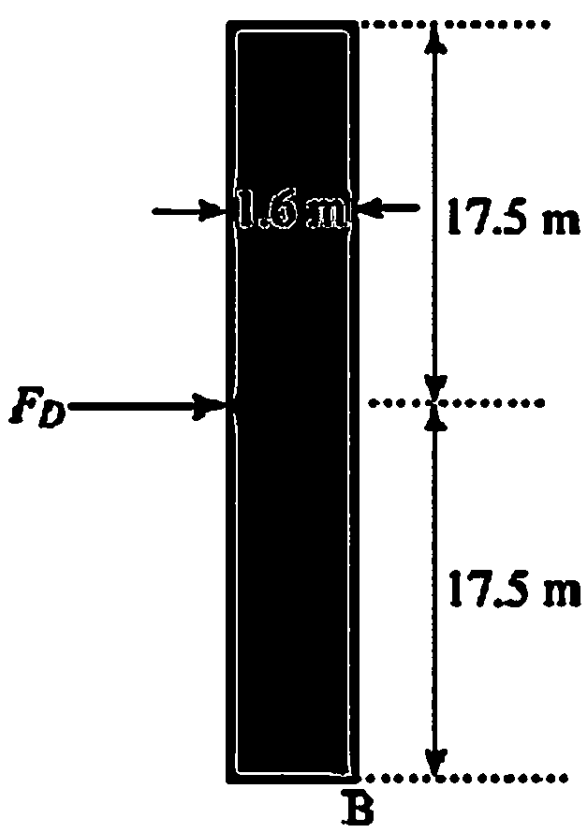
$$F_D = \frac{1}{2} C_D \rho A V^2$$

$$= \frac{1}{2} \times 0.15 \times 1.204 \times 56 \times (30.55)^2$$

$$= 4719.5 \text{ N}$$

Step 7 of 8

Now, draw the free body diagram of the chimney to show the forces acting on it.



Step 8 of 8

Now, to find out the bending moment about the base of the chimney, take moment about B.

$$+\circlearrowleft \sum M_B$$

$$M_B = F_D(17.5)$$

$$= 4719.5(17.5)$$

$$= 82591.25 \text{ N} \cdot \text{m}$$

Therefore the bending moment at the base of the chimney, $\boxed{M_B = 82591.25 \text{ N} \cdot \text{m}}$.

Question: 15.093P

Consider a blimp that can be approximated as a 3-m diameter, 8-m long ellipsoid and is connected to the ground. On a windless day, the rope tension due to the net buoyancy effect is measured to be 120 N. Determine the rope tension when there are 50 km/h winds blowing along the blimp (parallel to the blimp axis).

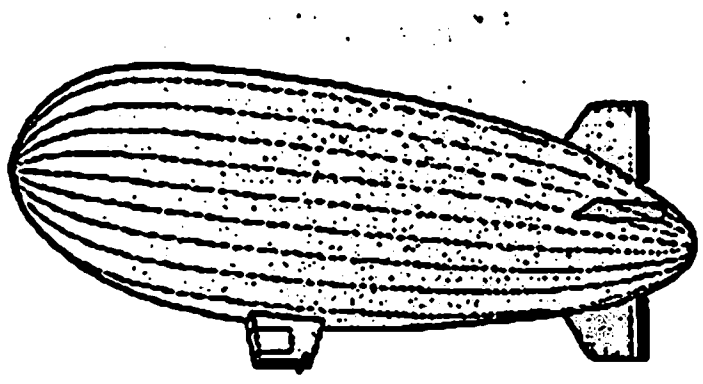


FIGURE P15–93

Step 1 of 6

Diameter of the blimp, $D = 3\,\text{m}$

Length of the blimp, $L = 8\,\text{m}$

Rope tension, $T = 120\,\text{N}$

Velocity of the wind, $V = 50\,\text{km/h}$

Step 2 of 6

The drag force acting on the blimp can be calculated using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag coefficient.

ρ is the density of air.

A is the frontal area of the blimp.

V is the velocity of the wind.

Step 3 of 6

Frontal area of the parachute,

$$A = \frac{\pi}{4} D^2$$

Here, $D = 3\,\text{m}$.

Substitute the value in the above equation.

$$A = \frac{\pi}{4}(3)^2$$

$$= 7.065\,\text{m}^2$$

Drag coefficient for the balloon is, $C_D = 0.1$

Step 4 of 6

From the table properties of air,

Density of air at 1atm pressure and 25°C is, $\rho = 1.184\,\text{kg/m}^3$

Velocity of air,

$$V = 50\,\text{km/h}$$

$$= 50\left(\frac{5}{18}\right)\text{m/s}$$

$$= 13.88\,\text{m/s}$$

Step 5 of 6

Now, substitute all the known values in equation (1)

$$F_D = \frac{1}{2} C_D \rho A V^2$$

$$= \frac{1}{2} \times 0.1 \times 1.184 \times 7.065 \times (13.88)^2$$

$$= 80.577\,\text{N}$$

Step 6 of 6

Now, the rope tension is the sum of rope tension due to buoyancy effect and the force due to wind blowing.

Therefore, rope tension is,

$$T = 120 + F_D$$

$$= 120 + 80.577$$

$$= 200.577\,\text{N}$$

Therefore, the rope tension is, $\boxed{T = 200.577\,\text{N}}$

Question: 15.094P

A commercial airplane has a total mass of 150,000 lbm and a wing planform area of 1800 ft2. The plane has a cruising speed of 550 mi/h and a cruising altitude of 38,000 ft where the air density is 0.0208 lbm/ft3. The plane has double-slotted flaps for use during takeoff and landing, but it cruises with all flaps retracted. Assuming the lift and drag characteristics of the wings can be approximated by NAC A 23012, determine (*a*) the minimum safe speed for takeoff and landing with and without extending the flaps, (*b*) the angle of attack to cruise steadily at the cruising altitude, and (*c*) the power that needs to be supplied to provide enough thrust to overcome drag. Take the air density on the ground to be 0.075 lbm/ft3.

Step 1 of 5

Mass of the air plane is $m = 150000\,\text{lbm}$

Speed of the air plane, $V = 550\,\text{mi/h} = 806.7\,\text{ft/s}$

Area of the wines, $A = 1800\,\text{ft}^2$

Air density, $\rho = 0.0208\,\text{lbm/ft}^3$

Step 2 of 5

From the figure of variation of lift coefficient with angle of attack for a symmetrical section"

$C = 3.48$

$C_2 = 1.52$

Weight of air plane is

$$W = mg$$
$$= 150000 \times 32.2$$
$$= 150000\,\text{lbf}$$

Mean velocity corresponding stalling is

$$V_1 = \sqrt{\frac{2W}{\rho C A}}$$
$$= \sqrt{\frac{2 \times 150000}{0.075 \times 3.48 \times 1800}}$$
$$= 217 \frac{\text{ft}}{\text{s}}$$
$$V_2 = \sqrt{\frac{2W}{\rho C_2 A}}$$
$$= \sqrt{\frac{2 \times 150000}{0.075 \times 3.48 \times 1800}}$$
$$= 143 \frac{\text{ft}}{\text{s}}$$

Step 3 of 5

Without flaps,

$$V = 1.2(V_1)$$
$$= 1.2(217)$$
$$= 178\,\text{mph}$$
$$V_{flaps} = 1.2V_2$$
$$= 1.2(143)$$
$$V_{flaps} = 172 \frac{\text{ft}}{\text{s}}$$

Therefore, velocity of the air plane with flaps is $\boxed{V_{flaps} = 172 \frac{\text{ft}}{\text{s}}}$

Step 4 of 5

b)

Coefficient of lift is given as

$$C_L = \frac{F_L}{\frac{1}{2}\rho V^2 A}$$
$$= \frac{150000}{\frac{1}{2} \times 0.0208 \times 806.7}$$
$$= 0.400$$

From the figure of drag coefficient corresponds angle of attack is $\boxed{\alpha = 3.4^\circ}$

Step 5 of 5

c)

Power needed to run the plane is

$$P = F \times V$$
$$= \left(C_D A \frac{\rho V^2}{2} \right) V$$
$$= \left(0.0153 \times 1800 \times \frac{0.0208 \times 806.7}{2} \right) 806.7$$
$$= 6199.90\,\text{W}$$

Therefore, power needed to run the air plane is $\boxed{P = 6199.90\,\text{W}}$

Question: 15.095P

A 9-cm-diameter smooth sports ball has a velocity of 36 km/h during a typical hit. Determine the percent increase in ithe drag coefficient if the ball is given a spin of 3500 rpm in'air at 1 atm and 25°C.

Step 1 of 6

Given data:-
Diameter of the ball, $D = 9$ cm.
Velocity of the ball, $V = 36$ km/h.
Number of revolutions, $N = 3500$ rpm.
Atmospheric conditions,
Temperature of the air, $T = 25\,^{\circ}C$.
Pressure of the air, $P = 1$ atm.

Step 2 of 6

Since we have to find out the percentage increase in drag coefficient we have to determine the drag coefficient of the ball with spinning and with out spinning.
Since here the drag force acting is not given it is not possible to calculate the drag coefficient directly so in order to calculate the drag coefficient we have to first calculate the Reynolds number and nondimensional rate of rotation after calculating that we can find the drag coefficient using the graphs 'The effect of surface roughness on the drag coefficient of a sphere' for a ball without spinning and 'The variation of lift and drag coefficients of a smooth sphere with the nondimensional rate of rotation for $\mathrm{Re} = 6\times10^4$' for a ball with spinning.
First of all let us calculate the drag coefficient of the ball with out spinning.
For that first of all calculate the Reynolds number using the formula given below.

$$\mathrm{Re} = \frac{VD}{\nu} \qquad \ldots\ldots (1)$$

Where, V is the velocity of the ball.
D is the diameter of the ball.
ν is the kinematic viscosity of air.
From the table 'Properties of air at 1atm pressure',
Kinematic viscosity of air, $\nu = 1.562\times10^{-5}\,\mathrm{m^2/s}$.

Step 3 of 6

Now, the velocity of ball,

$$V = 36\ \mathrm{km/h}$$
$$= 36\left(\frac{5}{18}\right)\mathrm{m/s}$$
$$= 10\ \mathrm{m/s}$$

Diameter of the ball,

$$D = 9\ \mathrm{cm}$$
$$= 0.09\ \mathrm{m}$$

Step 4 of 6

Now, substitute all the known values in equation (1)

$$\mathrm{Re} = \frac{VD}{\nu}$$
$$= \frac{10\times0.09}{1.562\times10^{-5}}$$
$$= 5.76\times10^4$$

Now, using the graph 'The effect of surface roughness on the drag coefficient of a sphere' the drag coefficient of the ball corresponding to the obtained Reynolds number is, $\boxed{C_D = 0.49}$, this is for a ball with out spinning.

Step 5 of 6

Now, to calculate the drag coefficient of the ball with spin, calculate the nondimensional rate of rotation.

$$\text{nondimensional rate of rotation} = \frac{\varpi D}{2V} \qquad \ldots\ldots (2)$$

Where, ϖ is the angular velocity.
D is the diameter of the ball.
V is the linear velocity of the ball.
Here the angular velocity ϖ is unknown so we have to calculate it using the formula given below.

$$\varpi = \frac{2\pi N}{60}$$

Where, N is the number of revolutions.

$$\varpi = \frac{2\times3.14\times3500}{60}$$
$$= 366.33$$
$$= 367\ \mathrm{rad/sec}$$

Now, substitute all the values in equation (2)

$$\text{nondimensional rate of rotation} = \frac{367\times0.09}{2\times10}$$
$$= 1.65\ \mathrm{rad}$$

Step 6 of 6

The Reynolds number that we have obtained is $\mathrm{Re} = 5.76\times10^4$ which is close to $\mathrm{Re} = 6\times10^4$ we can use the graph 'The variation of lift and drag coefficients of a smooth sphere with the nondimensional rate of rotation for $\mathrm{Re} = 6\times10^4$'.
Now, using the graph 'The variation of lift and drag coefficients of a smooth sphere with the nondimensional rate of rotation for $\mathrm{Re} = 6\times10^4$' the drag coefficient corresponding to the obtained nondimensional rate of rotation is, $\boxed{C_D = 0.59}$.
Therefore,
Drag coefficient (without spin), $C_{D\,no\,spin} = 0.49$
Drag coefficient (with spin), $C_{D\,spin} = 0.59$
Now, the percent increase in drag coefficient,

$$C_{D\,increase} = \frac{C_{D\,spin} - C_{D\,no\,spin}}{C_{D\,no\,spin}}$$
$$= \frac{0.59 - 0.49}{0.49}$$
$$= 0.20$$

Now, percent increase in C_D

$$C_{D\,increase} = 0.20\times100$$
$$= 20\%$$

Therefore, the percent increase is $\boxed{C_{D\,increase} = 20\%}$.

Question: 15.096P

A paratrooper and his 8-m-diameter parachute weigh 950 N. Taking the average air density to be 1.2 kg/m3, determine the terminal velocity of the paratrooper.

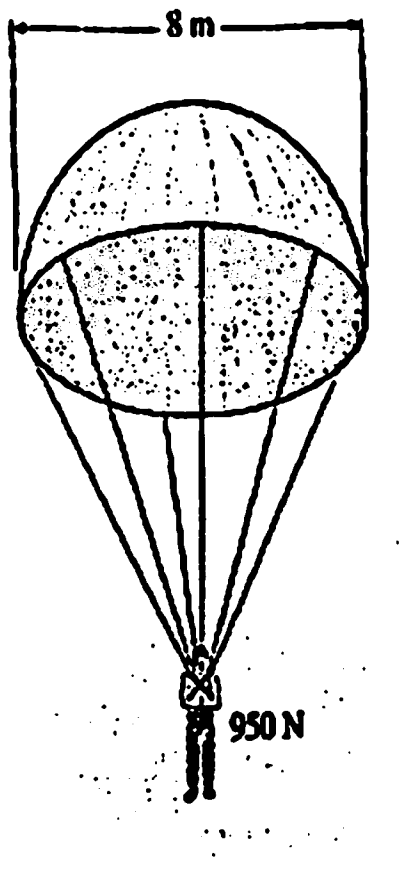


FIGURE P15–96

Step 1 of 4

Given data:-

Diameter of the parachute, $D = 8\ \text{m}$.

Weight of the paratrooper and the parachute, $W = 950\ \text{N}$.

Density of air, $\rho = 1.2\ \text{kg/m}^3$

Step 2 of 4

When a body falls freely the acceleration of the body will be equal to $9.8\ \text{m/s}^2$, the weight of the body will acts downwards and the drag force and buoyancy force will acts upwards. Terminal velocity of the body will be reached when the drag force acting will be equal to the difference of weight and buoyancy force acting on it.

$F_D = W - F_B$

Where, F_D is the drag force.

W is the weight of the body.

F_B is the buoyancy force.

Here, $F_B \cong 0$ (F_B is approximately equal to zero)

Therefore, $F_D = W$ ..... (1)

Now, drag force,

$$F_D = \frac{1}{2} C_D \rho A V^2 \quad \text{..... (2)}$$

Where, C_D is the drag coefficient.

ρ is the density of air.

A is the frontal area of the parachute.

V is the terminal velocity.

Step 3 of 4

Frontal area of the parachute,

$$A = \frac{\pi}{4} D^2$$

Here, $D = 8\ \text{m}$.

Substitute the value in the above equation.

$$A = \frac{\pi}{4}(8)^2$$

$$= 50.24\ \text{m}^2$$

Drag coefficient for a parachute is, $C_D = 1.3$

Step 4 of 4

Substitute all the known values in equation (2)

$$F_D = \frac{1}{2} \times 1.3 \times 1.2 \times 50.24 \times V^2$$

$$= 39.2\, V^2$$

Substitute the value of F_D in equation (1)

$$39.2\, V^2 = 950$$

$$V^2 = \frac{950}{39.2}$$

$$V^2 = 24.23$$

$$V = 4.9\ \text{m/s}$$

Therefore the terminal velocity of the paratrooper, $\boxed{V = 4.9\ \text{m/s}}$.

Question: 15.097P

It is proposed to meet the water needs of a recreational vehicle (RV) by installing a 2-m-long, 0.5-m-diameter cylindrical tank on top of the vehicle. Determine the additional power requirement of the RV at a speed of 95 km/h when the tank is installed such that its circular surfaces face (*a*) the front and back (as sketched) and (*b*) the sides of the RV. Assume atmospheric conditions are 87 kPa and 20°C.

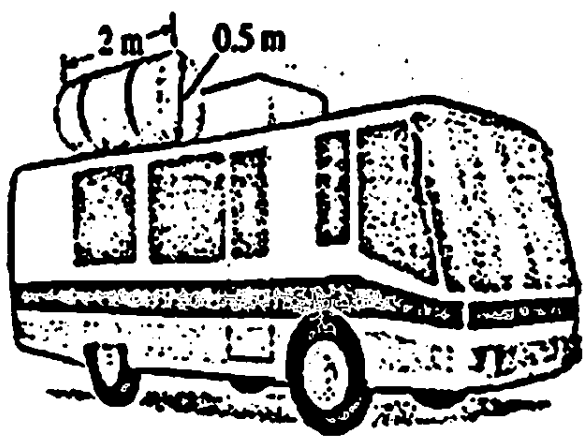


FIGURE P15–97

Step 1 of 5

Given data:

Length of the cylindrical tank, $L = 2$ m

Diameter of the cylindrical tank, $D = 0.5$ m

Speed of the vehicle, $V = 95$ km/h

Atmospheric conditions,

Pressure of the air, $P = 87$ kPa

Temperature of the air, $T = 20°C$

Step 2 of 5

a) Circular surfaces face the front and back of the vehicle.

Drag force acting on the tank is found out using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag coefficient

ρ is the density of air.

A is the frontal area of the tank.

V is the velocity of the vehicle.

Velocity of the vehicle,

$$V = 95 \text{ km/h}$$

$$= 95\left(\frac{5}{18}\right) \text{m/s}$$

$$= 26.4 \text{ m/s}$$

Since for horizontal cylinder with ratio $L/D = 4$, drag coefficient, $C_D = 0.9$

Step 3 of 5

Here, frontal area,

$$A = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.5)^2$$

$$= 0.19625 \text{ m}^2$$

From the table 'properties of air'

Density of air, $\rho = 1.027 \text{ kg/m}^3$

Substitute all the values in equation (1)

$$F_D = \frac{1}{2} \times 0.9 \times 1.027 \times 0.19625 \times (26.4)^2$$

$$= 63.21 \text{ N}$$

Power required for overcoming drag force,

$$P = F_D \times V$$

$$= 63.21 \times 26.4$$

$$= 1668.74 \text{ Watt}$$

Therefore the additional power required is, $\boxed{P = 1.67 \text{ kW}}$

Step 4 of 5

b) Circular surfaces face the sides of the vehicle.

Drag force acting on the tank is found out using the formula given below,

$$F_D = \frac{1}{2} C_D \rho A V^2 \qquad \ldots\ldots (1)$$

Where, C_D is the drag coefficient.

ρ is the density of air.

A is the frontal area of the tank.

V is the velocity of the vehicle.

Velocity of the vehicle,

$$V = 95 \text{ km/h}$$

$$= 95\left(\frac{5}{18}\right) \text{m/s}$$

$$= 26.4 \text{ m/s}$$

Here, drag coefficient, $C_D = 0.8$

Here, frontal area,

$$A = L \times D$$

$$= 2 \times 0.5$$

$$= 1 \text{ m}^2$$

Step 5 of 5

From the table 'properties of air'

Density of air, $\rho = 1.027 \text{ kg/m}^3$

Substitute all the values in equation (1)

$$F_D = \frac{1}{2} \times 0.8 \times 1.027 \times 1 \times (26.4)^2$$

$$= 286.3 \text{ N}$$

Power required for overcoming this drag,

$$P = F_D \times V$$

$$= 286.3 \times 26.4$$

$$= 7558.6 \text{ Watt}$$

Therefore the additional power required is, $\boxed{P = 7.55 \text{ kW}}$

Question: 15.098P

A 17,000-kg tractor-trailer rig has a frontal area of 92 m2, a drag coefficient of 0.96, a rolling resistance coefficient of 0.05 (multiplying the weight of a vehicle by the rolling resistance coefficient gives the rolling resistance), a bearing friction resistance of 350 N, and a maximum speed of 110 km/h on a level road during steady cruising in calm weather with an air density of 1.25 kg/m3. Now a fairing is installed to the front of the rig to suppress separation and to streamline the flow to the top surface, and the drag coefficient is reduced to 0.76. Determine the maximum speed of the rig with the fairing.

Step 1 of 7

Loading solution. Please wait ...

Step 2 of 7

In order to calculate the maximum speed of the rig with the fairing, let us calculate the maximum power of the vehicle

Power developed can be found out using the equation given below.

P = Force × Velocity

Here,

Force is the sum of all the resistive forces acting on the tractor-trailer rig which are bearing friction, drag force and rolling resistance.

Velocity is the maximum velocity of the tractor-trailer rig, maximum power occurs at maximum velocity.

Therefore, we have maximum power,

$$P = \left(F_{D1} + F_{\text{Bearing}} + F_{RR}\right) V_1 \qquad \text{..... (1)}$$

Where, F_{D1} is the drag force acting on the tractor-trailer rig with out fairing.

F_{Bearing} is the bearing resistance

F_{RR} is the rolling resistance

V_1 is the maximum velocity of the tractor-trailer rig

Step 3 of 7

Here, F_{D1} and F_{RR} are unknown so we have to calculate it as follows,

Now, the drag force acting on the tractor-trailer rig with out fairing,

$$F_{D1} = \frac{1}{2} C_{D_1} \rho A V_1^2 \qquad \text{..... (2)}$$

Where, C_{D1} drag coefficient.

ρ is the density of air.

A is the frontal area.

V_1 is the maximum speed

Here, maximum speed,

$$V_1 = 110 \text{ km/h}$$

$$= 110\left(\frac{5}{18}\right)$$

$$= 30.55 \text{ m/s}$$

Substitute all the known values in equation (2), we get

$$F_{D1} = \frac{1}{2} C_{D_1} \rho A V_1^2$$

$$= \frac{1}{2} \times 0.96 \times 1.25 \times 9.2 \times (30.55)^2$$

$$= 5151.83 \text{ N}$$

Step 4 of 7

It has given that rolling resistance is the product of weight of the vehicle and rolling resistance coefficient.

Therefore, rolling resistance $F_{RR} = W \times C_{RR}$ ..... (3)

Weight of the vehicle, $W = mg$

Where, m is the mass.

g is the acceleration due to gravity

Therefore, $W = 17000 \times 9.81$

Substitute all the values in equation (3), we get

$$F_{RR} = 17000 \times 9.81 \times 0.05$$

$$= 8339 \text{ N}$$

Now, substitute all the calculated values in equation (1).

$$P = \left(F_{D1} + F_{\text{Bearing}} + F_{RR}\right) V_1$$

$$= (5151.83 + 350 + 8339)\, 30.55$$

$$= 422837.36 \text{ Watt}$$

$$= \frac{422837.36}{1000} \text{ kW}$$

$$= 423 \text{ kW}$$

Therefore, the maximum power the vehicle can developed is, $P = 423$ kW

Step 5 of 7

Now, a fairing is installed and the drag coefficient reduces to 0.76.

Therefore, $C_{D2} = 0.76$

Let us calculate the maximum speed of the rig in this condition.

It can be calculated using the equation given below.

$$P = \left(F_{D2} + F_{\text{Bearing}} + F_{RR}\right) V_2 \qquad \text{..... (4)}$$

Where, F_{D2} is the drag force acting on the tractor-trailer rig with fairing.

F_{Bearing} is the bearing resistance

F_{RR} is the rolling resistance

V_2 is the maximum velocity of the tractor-trailer rig

Step 6 of 7

$$F_{D2} = \frac{1}{2} C_{D_2} \rho A V_2^2$$

Where, C_{D2} drag coefficient

ρ is the density of air

A is the frontal area

V_2 is the maximum speed

Step 7 of 7

Substitute all the values in the above equation, we get,

$$F_{D2} = \frac{1}{2} C_{D_2} \rho A V_2^2$$

$$= \frac{1}{2} \times 0.76 \times 1.25 \times 9.2 \times V_2^2$$

$$= 4.37\, V_2^2$$

Substitute all the values in equation (4)

$$P = \left(F_{D2} + F_{\text{Bearing}} + F_{RR}\right) V_2$$

$$423000 = \left(4.37\, V_2^2 + 350 + 8339\right) V_2$$

$$423000 = \left(4.37\, V_2^2 + 8689\right) V_2$$

$$423000 = 4.37\, V_2^3 + 8689\, V_2$$

$$4.37\, V_2^3 + 8689\, V_2 - 423000 = 0$$

Solving the above equation we get V_2 as 32.08 m/s

$$V_2 = \frac{32.08 \times 3600}{10^3}$$

$$= 115.5 \text{ km/h}$$

Therefore, the maximum speed of the rig with the fairing is, $\boxed{V_2 = 115.5 \text{ km/h}}$

Question: 15.099P

Stokes law can be used to determine the viscosity of a fluid by dropping a spherical object in it and measuring the terminal velocity of the object in that fluid. This can be done by plotting the distance traveled against time and observing when the curve becomes linear. During such an experiment a 3-inm-diameter glass ball (ρ = 2500 kg/m3) is dropped into a fluid whose density is 875 kg/m3, and the terminal velocity is measured to be 0.16 m/s. Disregarding the wall effects, determine the viscosity of the fluid.

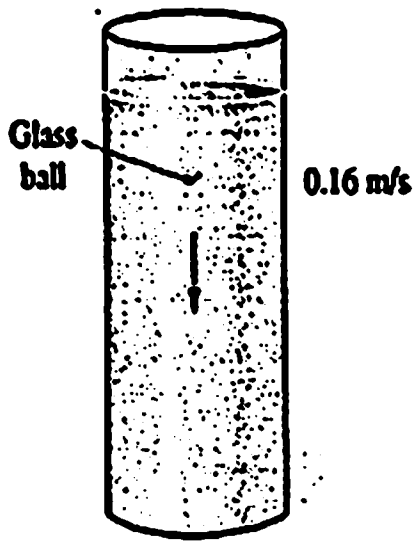


FIGURE P15–99

Step 1 of 4

Given data:

Diameter of the glass ball, $D = 3$ mm

Density of the glass ball, $\rho_s = 2500\ \text{kg/m}^3$

Density of the fluid, $\rho_f = 875\ \text{kg/m}^3$

Terminal velocity of the ball, $V = 0.16$ m/s

Step 2 of 4

The terminal velocity of freely falling body is reached when the drag force becomes equal to the difference of weight of the body and buoyancy force acting on it.

$F_D = W - F_B$ ..... (1)

Where, F_D drag force.

W is the weight of the body.

F_B is the buoyancy force.

According to Stokes law,

$F_D = 3\pi\mu VD$

Where, μ is the dynamic viscosity of fluid.

V is the terminal velocity.

D is the diameter of the glass ball.

Step 3 of 4

Now, weight of the body,

$W = \rho_s gV$

Where, ρ_s is the density of the glass ball.

g is the acceleration due to gravity.

V is the volume of the glass.

The buoyancy force applied by the fluid,

$F_B = \rho_f gV$

Where, ρ_f is the density of the fluid.

g is the acceleration due to gravity.

V is the volume of the glass.

Step 4 of 4

By substituting all the values in equation (1), we get,

$3\pi\mu VD = \rho_s gV - \rho_f gV$

$$3\pi\mu VD = g\frac{\pi D^3}{6}(\rho_s - \rho_f)$$

$$\mu = \frac{gD^2}{18}(\rho_s - \rho_f)$$

$$= \frac{9.81 \times .003^2}{18}(2500 - 875)$$

$$= 0.05\ \text{kg/m} \cdot \text{s}$$

Therefore, the viscosity of the fluid is, $\boxed{\mu = 0.05\ \text{kg/m} \cdot \text{s}}$

Question: 15.100P

During an experiment, three aluminum balls (ρs =2600 kg/m3) having diameters 2, 4, and 10 mm, respectively, are dropped into a tank filled with glycerin at 22°C (ρf = 1274 kg/m3 and μ = 1 kg/m-s). The terminal settling velocities of the balls are measured to be 3.2, 12.8, and 60.4 mm/s, respectively. Compare these values with the velocities predicted by Stokes law for drag force $FD = 3\pi\mu DV$, which is valid for very low Reynolds numbers (Re $\ll$ 1). Determine the error involved for each case and assess the accuracy of Stokes law.

Step 1 of 9

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Step 2 of 9

The terminal velocity of freely falling body is reached when the drag force becomes equal to the difference of weight of the body and buoyancy force acting on it.

$$F_D = W - F_B \qquad \ldots\ldots (1)$$

Where, F_D drag force.

W is the weight of the ball.

F_B is the buoyancy force

According to Stokes law,

$$F_D = 3\pi\mu VD$$

Where, μ is the dynamic viscosity of fluid.

V is the terminal velocity.

D is the diameter of the ball.

Step 3 of 9

Now, weight of the body,

$$W = \rho_s gV$$

Where, ρ_s is the density of the ball.

g is the acceleration due to gravity.

V is the volume of the ball.

Now, the buoyancy force applied by the fluid,

$$F_B = \rho_f gV$$

Where, ρ_f is the density of the fluid.

g is the acceleration due to gravity.

V is the volume of the ball, $V = \frac{\pi}{6}D^3$

Step 4 of 9

Now, substitute all the values in equation (1), we get,

$$3\pi\mu VD = \rho_s gV - \rho_f gV$$

$$V = \frac{gV(\rho_s - \rho_f)}{3\pi\mu D}$$

$$= \frac{g\pi D^3(\rho_s - \rho_f)}{18\pi\mu D}$$

$$= \frac{9.81(0.002)^2(2600-1274)}{18\times 1}$$

$$= 0.0029 \text{ m/s}$$

Terminal setting velocity of the first ball,

$$V_1 = 3.2 \text{ mm/s}$$

$$= 0.0032 \text{ m/s}$$

Therefore the velocity of the first ball according to Stokes law is, $V = 0.0029$ m/s and experimental velocity is, $V_1 = 0.0032$ m/s

Step 5 of 9

Therefore the error is,

$$\text{Error} = \frac{V_1 - V}{V_1}\times 100$$

$$= \frac{0.0032 - 0.0029}{0.0032}\times 100$$

$$= 9.37\ \%$$

Therefore, the error involved in first ball is, $\boxed{\text{Error} = 9.37\ \%}$

Step 6 of 9

The velocity of second ball,

$$V = \frac{gV(\rho_s - \rho_f)}{3\pi\mu D}$$

$$= \frac{g\pi D^3(\rho_s - \rho_f)}{18\pi\mu D}$$

$$= \frac{9.81(0.004)^2(2600-1274)}{18\times 1}$$

$$= 0.012 \text{ m/s}$$

Terminal setting velocity of the second ball,

$$V_2 = 12.8 \text{ mm/s}$$

$$= 0.0128 \text{ m/s}$$

Therefore, the velocity of second ball according to Stokes law is, $V = 0.012$ m and experimental velocity is, $V_2 = 0.0128$ m/s

Step 7 of 9

Now, the error involved for the second ball,

$$\text{Error} = \frac{V_2 - V}{V_2}\times 100$$

$$= \frac{0.0128 - 0.012}{0.0128}\times 100$$

$$= 6.25\ \%$$

Therefore, the error involved in second ball is, $\boxed{\text{Error} = 6.25\ \%}$

Step 8 of 9

The velocity of third ball,

$$V = \frac{gV(\rho_s - \rho_f)}{3\pi\mu D}$$

$$= \frac{g\pi D^3(\rho_s - \rho_f)}{18\pi\mu D}$$

$$= \frac{9.81(0.01)^2(2600-1274)}{18\times 1}$$

$$= 0.072 \text{ m/s}$$

Terminal setting velocity of the second ball,

$$V_3 = 60.4 \text{ mm/s}$$

$$= 0.0604 \text{ m/s}$$

Therefore the velocity according to Stokes law is, $V = 0.072$ m and experimental velocity is, $V_3 = 0.0604$ m/s.

Step 9 of 9

Now, the error involved for the third ball,

$$\text{Error} = \frac{V_3 - V}{V_3}$$

$$= \frac{0.0604 - 0.072}{0.0604}$$

$$= -0.192$$

$$= -19.2\ \%$$

Therefore, the error involved in the third ball is, $\boxed{\text{Error} = -19.2\ \%}$

Question: 15.101P pg 1

Repeat Prob. 15-100 by considering the more general form of Stokes law expressed as$F_D = 3\pi\mu DV + (9\pi/16)\rho V^2D^2$ whereρ is the fluid density.

Step 1 of 11

Given data:

Density of aluminum balls, $\rho_s = 2600\ \text{kg/m}^3$

Diameter of the first aluminum ball, $D_1 = 2\ \text{mm}$

Diameter of the second aluminum ball, $D_2 = 4\ \text{mm}$

Diameter of the third aluminum ball, $D_3 = 10\ \text{mm}$

Temperature of glycerin, $T = 22°\text{C}$

Density of glycerin, $\rho_f = 1274\ \text{kg/m}^3$

Dynamic viscosity of glycerin, $\mu = 1\ \text{kg/m}\cdot\text{s}$

Terminal setting velocity of the first ball, $V_1 = 3.2\ \text{mm/s}$

Terminal setting velocity of the second ball, $V_2 = 12.8\ \text{mm/s}$

Terminal setting velocity of the third ball, $V_3 = 60.4\ \text{mm/s}$

Step 2 of 11

The terminal velocity of freely falling body is reached when the drag force becomes equal to the difference of weight of the body and buoyancy force acting on it.

$$F_D = W - F_B \qquad \ldots\ldots (1)$$

Where, F_D drag force.

W is the weight of the ball.

F_B is the buoyancy force

According to Stokes law,

$$F_D = 3\pi\mu VD + \left(\frac{9\pi}{16}\right)\rho V^2D^2$$

Where, μ is the dynamic viscosity of fluid.

V is the terminal velocity.

D is the diameter of the ball.

ρ is the density of fluid.

Step 3 of 11

Now, weight of the body,

$$W = \rho_s gV$$

Where, ρ_s is the density of the ball

g is the acceleration due to gravity.

V is the volume of the ball.

Now, the buoyancy force applied by the fluid,

$$F_B = \rho_f gV$$

Where, ρ_f is the density of the fluid.

g is the acceleration due to gravity.

V is the volume of the ball.

Step 4 of 11

Now, by substituting all the values in equation (1), we get

$$3\pi\mu VD + \left(\frac{9\pi}{16}\right)\rho V^2D^2 = \rho_s gV - \rho_f gV$$

$$3\pi\mu VD + \left(\frac{9\pi}{16}\right)\rho V^2D^2 = gV\left(\rho_s - \rho_f\right)$$

Now, to find the velocity of ball solve the above quadratic for V.

$$V = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \qquad \ldots\ldots (2)$$

Where,

$$a = \left(\frac{9\pi}{16}\right)\rho D^2$$

$$b = 3\pi\mu D$$

$$c = -gV\left(\rho_s - \rho_f\right)$$

Step 5 of 11

Diameter of the first ball,

$$D_1 = 2\ \text{mm}$$
$$= 0.002\ \text{m}$$

Terminal setting velocity of the first ball,

$$V_1 = 3.2\ \text{mm/s}$$
$$= 0.0032\ \text{m/s}$$

For first ball,

$$a = \left(\frac{9\pi}{16}\right)\rho D^2$$
$$= \left(\frac{9\pi}{16}\right) \times 1274 \times (0.002)^2$$
$$= 0.009$$
$$b = 3\pi\mu D$$
$$= 3 \times 3.14 \times 1 \times 0.002$$
$$= 0.01884$$
$$c = -gV\left(\rho_s - \rho_f\right)$$
$$= -g\frac{\pi D^3}{6}\left(\rho_s - \rho_f\right)$$
$$= -9.81 \times \frac{3.14 \times (0.002)^3}{6}(2600 - 1274)$$
$$= -0.0000545$$

Step 6 of 11

Substitute all known values in equation (2), we get,

$$V = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$
$$= \frac{-0.01884 + \sqrt{0.01884^2 - 4(0.009 \times -0.0000545)}}{2 \times 0.009}$$
$$= 0.0029\ \text{m/s}$$

$$V = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

$$= \frac{-0.01884 + \sqrt{0.01884^2 - 4(0.009 \times -0.0000545)}}{2 \times 0.009}$$

$$= 0.0029 \text{ m/s}$$

Therefore, the velocity of first ball according to Stokes law is, $V = 0.0029$ m/s and experimental velocity is, $V_1 = 0.0032$ m/s

Therefore the error is,

$$\text{Error} = \frac{V_1 - V}{V_1} \times 100$$

$$= \frac{0.0032 - 0.0029}{0.0032} \times 100$$

$$= 9.375\ \%$$

Therefore, here the error involved is, $\boxed{\text{Error} = 9.375\ \%}$

Step 7 of 11

Diameter of the second ball,

$D_2 = 4$ mm

$= 0.004$ m

Terminal setting velocity of the second ball,

$V_2 = 12.8$ mm/s

$= 0.0128$ m/s

For second ball

$$a = \left(\frac{9\pi}{16}\right)\rho D^2$$

$$= \left(\frac{9\pi}{16}\right) \times 1274 \times (0.004)^2$$

$$= 0.036$$

$$b = 3\pi\mu D$$

$$= 3 \times 3.14 \times 1 \times 0.004$$

$$= 0.03768$$

$$c = -gV(\rho_s - \rho_f)$$

$$= -g\frac{\pi D^3}{6}(\rho_s - \rho_f)$$

$$= -9.81 \times \frac{3.14 \times (0.004)^3}{6}(2600 - 1274)$$

$$= -0.00044$$

Step 8 of 11

Substitute all known values in equation (2), we get,

$$V = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

$$= \frac{-0.03768 + \sqrt{0.03768^2 - 4(0.036 \times -0.00044)}}{2 \times 0.036}$$

$$= 0.0115 \text{ m/s}$$

Therefore the velocity of second ball according to Stokes law is, $V = 0.0115$ m/s and experimental velocity is, $V_2 = 0.0128$ m/s

Now, the error involved for the second ball,

$$\text{Error} = \frac{V_2 - V}{V_2} \times 100$$

$$= \frac{0.0128 - 0.0115}{0.0128} \times 100$$

$$= 10.16\ \%$$

Therefore, here the error involved is, $\boxed{\text{Error} = 10.16\ \%}$

Step 9 of 11

Diameter of the third ball,

$D_3 = 10$ mm

$= 0.01$ m

Terminal setting velocity of the second ball,

$V_3 = 60.4$ mm/s

$= 0.0604$ m/s

For the third ball,

$$a = \left(\frac{9\pi}{16}\right)\rho D^2$$

$$= \left(\frac{9\pi}{16}\right) \times 1274 \times (0.01)^2$$

$$= 0.225$$

Step 10 of 11

$$b = 3\pi\mu D$$

$$= 3 \times 3.14 \times 1 \times 0.01$$

$$= 0.0942$$

$$c = -gV(\rho_s - \rho_f)$$

$$= -g\frac{\pi D^3}{6}(\rho_s - \rho_f)$$

$$= -9.81 \times \frac{3.14 \times (0.01)^3}{6}(2600 - 1274)$$

$$= -0.0068$$

Step 11 of 11

Substitute all known values in equation (2)

$$V = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

$$= \frac{-0.0942 + \sqrt{0.0942^2 - 4(0.225 \times -0.0068)}}{2 \times 0.225}$$

$$= 0.063 \text{ m/s}$$

$V = \dfrac{}{2a}$ 15.101P pg 3

$$= \frac{-0.0942 + \sqrt{0.0942^2 - 4(0.225 \times -0.0068)}}{2 \times 0.225}$$

$$= 0.063 \text{ m/s}$$

Therefore the velocity of third ball according to Stokes law is, $V = 0.063$ m/s and experimental velocity is, $V_3 = 0.0604$ m/s

Now, the error involved for the third ball,

$$\text{Error} = \frac{V_3 - V}{V_3} \times 100$$

$$= \frac{0.0604 - 0.063}{0.0604} \times 100$$

$$= -4.3\%$$

Therefore, here the error involved is, $\boxed{\text{Error} = -4.3\%}$

Question:

Engine oil at 40°C is flowing over a long flat plate with a velocity of 6 m/s. Determine the distance xcr from the leading edge of the plate where the flow becomes turbulent, and calculate and plot the thickness of the boundary layer over a length of 2xCr.

Step 1 of 6

From the table A-19 "Properties of liquid" in the text book, take the value of kinematic viscosity for engine oil at temperature, $T = 40°C$.

$$\nu = \left(2.485\times10^{-4}\right)\ \text{m}^2/\text{s}$$

Calculate the distance x_{cr} from the leading edge of the plate where the flow becomes turbulent.

$$\text{Re}_{cr} = \frac{Vx_{cr}}{\nu}$$

Here, Re_{cr} is the critical Reynolds number and V is the velocity.

Substitute 5×10^5 for Re_{cr}, $\left(2.485\times10^{-4}\right)\ \text{m}^2/\text{s}$ for ν, and $6\ \text{m/s}$ for V .

$$5\times10^5 = \frac{6\times x_{cr}}{\left(2.485\times10^{-4}\right)}$$

$$\left(6\times x_{cr}\right) = 124.25$$

$$x_{cr} = 20.7\ \text{m}$$

Therefore, the distance x_{cr} from the leading edge of the plate where the flow becomes turbulent is $\boxed{20.7\ \text{m}}$.

Step 2 of 6

Use laminar flow relation to calculate the boundary layer thickness for first half of the flow, $x = 0$ to $20.7\ \text{m}$:

Assume $x = 4\ \text{m}$:

Calculate the Reynolds number.

$$\text{Re}_x = \frac{Vx}{\nu}$$

Substitute $4\ \text{m}$ for x, $\left(2.485\times10^{-4}\right)\ \text{m}^2/\text{s}$ for ν, and $6\ \text{m/s}$ for V .

$$\text{Re}_x = \frac{6\times4}{\left(2.485\times10^{-4}\right)}$$

$$= 96579.4768$$

Calculate the thickness of the boundary layer for laminar flow.

$$\left(\delta_x\right)_{\text{laminar}} = \frac{4.91x}{\text{Re}_x^{1/2}}$$

Substitute $4\ \text{m}$ for x, and 96579.4768 for Re_x.

$$\left(\delta_x\right)_{\text{laminar}} = \frac{4.91\times4}{\left(96579.4768\right)^{1/2}}$$

$$= 0.0632\ \text{m}$$

Step 3 of 6

Use turbulent flow relation to calculate the boundary layer thickness for second half of the flow.

$x = 20.7\ \text{m}$ to $2x_{cr}$

Substitute $20.7\ \text{m}$ to x_{cr}.

$$x = 20.7\ \text{m to}\ \left(2\times20.7\right)$$

$$= 20.7\ \text{m to}\ 41.4\ \text{m}$$

Assume $x = 24\ \text{m}$:

Calculate the Reynolds number.

$$\text{Re}_x = \frac{Vx}{\nu}$$

Substitute $24\ \text{m}$ for x, $\left(2.485\times10^{-4}\right)\ \text{m}^2/\text{s}$ for ν, and $6\ \text{m/s}$ for V .

$$\text{Re}_x = \frac{6\times24}{\left(2.485\times10^{-4}\right)}$$

$$= 579476.86116$$

Step 4 of 6

Calculate the thickness of the boundary layer for turbulent flow.

$$\left(\delta_x\right)_{\text{turbulent}} = \frac{0.38x}{\text{Re}_x^{1/5}}$$

Substitute $24\ \text{m}$ for x, and 579476.86116 for Re_x.

$$\left(\delta_x\right)_{\text{turbulent}} = \frac{0.38\times24}{\left(579476.86116\right)^{1/5}}$$

$$= \frac{9.12}{14.21043}$$

$$= 0.64178\ \text{m}$$

Step 5 of 6

Tabulate the values of thickness of the boundary layer for laminar and turbulent flows over a length of $41.4\ \text{m}$.

$x\ (\text{m})$	Re_x	$\left(\delta_x\right)_{\text{laminar}}$	$\left(\delta_x\right)_{\text{turbulent}}$
0	0	0	

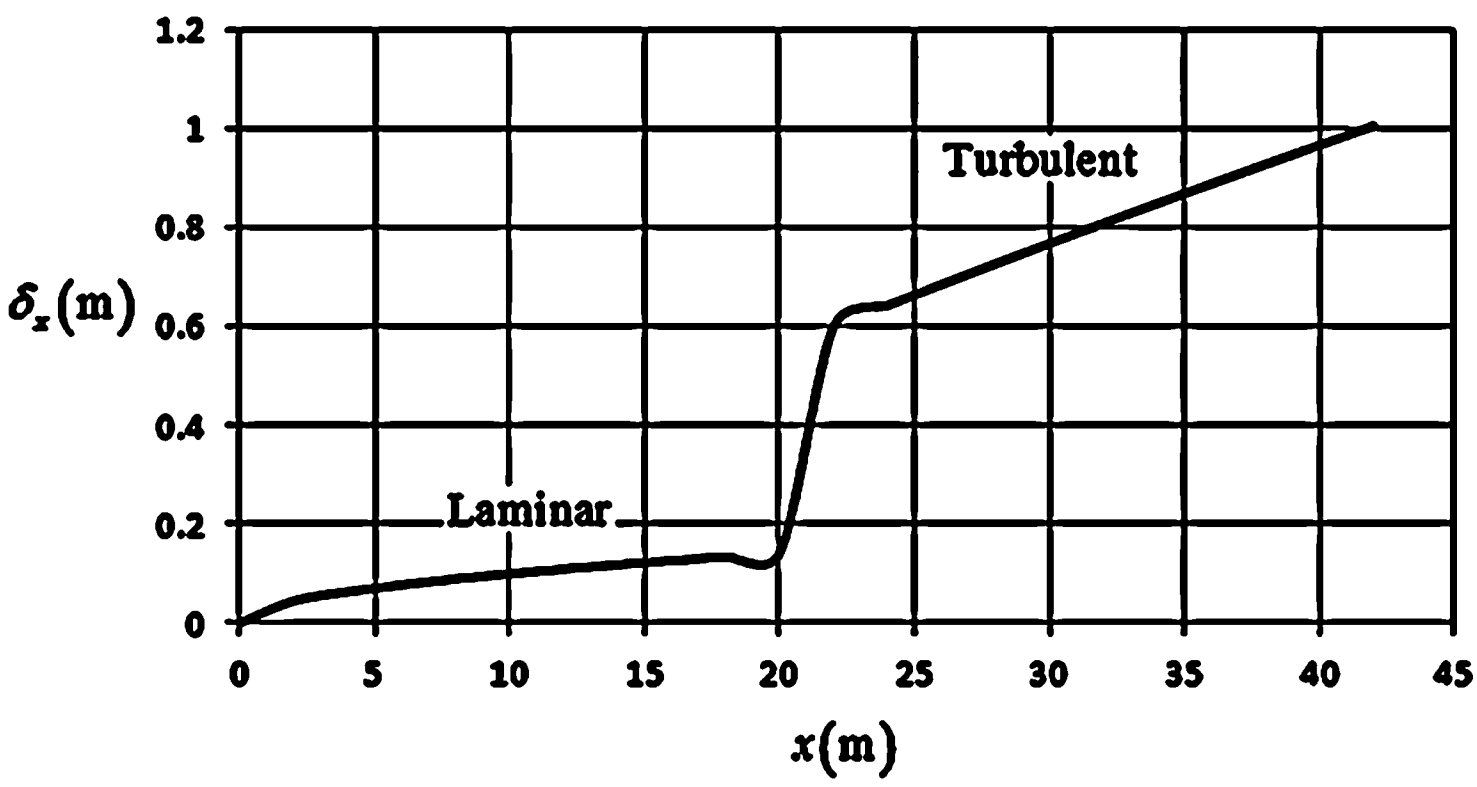
Tabulate the values of thickness of the boundary layer for laminar and turbulent flows over a length of 41.4 m .

15.102P pg 2

x (m)	Re_x	$(\delta_x)_{\text{laminar}}$	$(\delta_x)_{\text{turbulent}}$
0	0	0	
2	48289.73843	0.044687	
4	96579.47686	0.063197	
6	144869.2153	0.077401	
8	193158.9537	0.089375	
10	241448.6922	0.099924	
12	289738.4306	0.109461	
14	338028.169	0.118231	
16	386317.9074	0.126395	
18	434607.6459	0.134062	
20	482897.3843	0.141314	
22	531187.1227		0.598628
24	579476.8612		0.641782
26	627766.5996		0.684222
28	676056.338		0.726014
30	724346.0765		0.767212
32	772635.8149		0.807865
34	820925.5533		0.848012
36	869215.2918		0.887689
38	917505.0302		0.926927
40	965794.7686		0.965754
42	1014084.507		1.004195

Step 6 of 6

Plot the thickness of the boundary layer over a length of $2x_{cr}\,(41.4\text{ m})$.



Question: 15.103P

Janie likes to drive with a tennis ball on her car antenna. The ball diameter is D = 2.62 in and its equivalent roughness factor is ε/D - 1.5 × 10–3. Her friends tell her she is wasting gas because of the additional drag on the ball. Estimate how much money (in dollars) she wastes per year by driving around with this tennis ball on her antenna. Use the following additional information: She drives mostly on the highway, about 12,000 miles per year at an average speed of 55 mph. The overall car efficiency is 0.308, ρfuel, = 50.2 lbm/ft3, and the heating value of the fuel is 1.47 × 107ft.lbf/lbm. The fuel costs $4.00 per gallon. Use standard air properties. Be careful with unit conversions. Should Janie remove the tennis ball?

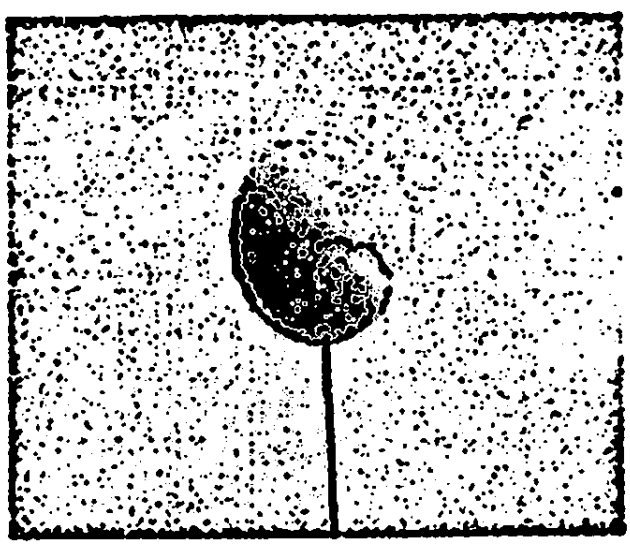


FIGURE P15–103E

Step 1 of 6

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Step 2 of 6

Now, the drag force acting on the ball can be calculated by using the formula given as

$$F_D = \frac{1}{2} C_D A \rho_{air} V^2 \qquad \text{..... (1)}$$

Where C_D is the drag coefficient

A is the frontal area of the ball.

V is the velocity of the car.

ρ_{air} is the density of air.

Now, velocity of car,

$$V = 55 \text{ mph}$$

$$= \frac{55 \times 5280}{3600} \text{ ft/s}$$

$$= 80.666 \text{ ft/s}$$

Now, frontal area of the ball,

$$A = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4} (0.2183)^2$$

$$= 0.03742 \text{ ft}^2$$

Step 3 of 6

The value C_D is unknown, but roughness factor is given in the question, so by calculating the Reynolds number we can find out the drag coefficient from the graph,

Reynolds number,

$$\text{Re} = \frac{VD}{\nu} \qquad \text{..... (2)}$$

Where, V is the velocity of car.

D is the diameter of the ball.

ν is the kinematic viscosity of air.

At standard air condition kinematic viscosity of air,

$$\nu = 1.535 \times 10^{-4} \text{ ft}^2/\text{s}$$

Step 4 of 6

Now, substitute all the known values in equation (2)

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{80.666 \times 0.218}{1.535 \times 10^{-4}}$$

$$= 1.145 \times 10^5$$

Based on the obtained Reynolds number, from the graph "The effect of surface roughness

$$= \frac{\pi}{4} (0.2183)^2$$

$$= 0.03742 \text{ ft}^2$$

Step 3 of 6

The value C_D is unknown, but roughness factor is given in the question, so by calculating the Reynolds number we can find out the drag coefficient from the graph,

Reynolds number,

$$\text{Re} = \frac{VD}{\nu} \qquad \text{..... (2)}$$

Where, V is the velocity of car.

D is the diameter of the ball.

ν is the kinematic viscosity of air.

At standard air condition kinematic viscosity of air,

$$\nu = 1.535 \times 10^{-4} \text{ ft}^2/\text{s}$$

Step 4 of 6

Now, substitute all the known values in equation (2)

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{80.666 \times 0.218}{1.535 \times 10^{-4}}$$

$$= 1.145 \times 10^5$$

Based on the obtained Reynolds number, from the graph "The effect of surface roughness

The amount of fuel required to develop this much energy for one year,

$$f = \frac{E_{in}}{\text{HV}(\rho_{fuel})}$$

$$= \frac{984662.33}{1.47 \times 10^7 (6.711)}$$

$$= 0.01 \text{ gallons/year}$$

Cost of fuel = Amount of fuel × Unit cost of fuel.

$$= 0.01 \times 4.00$$

$$= \$0.04$$

Therefore, the amount wasted per year to overcome the drag produced by tennis ball is $\boxed{\$0.04}$.

CHAPTER 16
SOLUTIONS

Question: 16.001P

Which is a better heat conductor, diamond or silver?

Step 1 of 1

Diamond is a better heat conductor than silver. Because of thermal conductivity of the Diamond is $K = 2000 \frac{\mathrm{W}}{\mathrm{m} \cdot \mathrm{K}}$ and silver is $K_s = 430 \frac{\mathrm{W}}{\mathrm{m} \cdot \mathrm{K}}$.

Then we know that heat transfer rate can calculate by

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L}$$

From the above Value we know that rate of heat transfer directly proportional to thermal conductivity. so that it is better conductivity than the silver.

Question: 16.002P

Define thermal conductivity and explain its significance in heat transfer.

Step 1 of 2

Thermal conductivity: It is a property of a material by which the rate of heat transfer is measure through a unit thickness of the material per unit area and per unit temperature difference. It is a measure of the ability of material to conduct heat.

Step 2 of 2

By the help of thermal conductivity of a material we can measure how fast heat will be conducted in that material. A high value of thermal conductivity signifies that material is highly conductor of heat and low value how that material is less conductor of heat that mean insulator.

Question: 16.003P

What are the mechanisms of heat transfer? How are they distinguished from each other?

Step 1 of 3

There are three mechanisms of heat transfer which are conduction, convection and radiation.

Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. So in this method heat can be transfer from solid to solid. Conduction is expressed by Fourier's law of conduction as

$$Q = kA\frac{dt}{dx}$$

Where

$\frac{dt}{dx}$ is the temperature gradient,

K is the thermal conductivity, and

A is the area which is normal to the direction of heat transfer.

Step 2 of 3

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. So when heat is transfer between solid to fluid or vice versa Convection is expressed by Newton's law of cooling as

$$Q_{conv} = hA(T_0 - T_\infty)$$

Where

h is the convection heat transfer coefficient,

A Is the surface area through which convection heat transfer takes place,

T_0 Is the surface temperature and

T_∞ Is the temperature of the fluid sufficiently far from the surface.

Step 3 of 3

Radiation is energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules. So in radiation no medium is required. Radiation is expressed by Stefan-Boltzman law as

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{surr}^4)$$

Where

ε Is the emissivity of surface,

A_s Is the surface area,

T_s Is the surface temperature

T_{surr} is average surrounding surface temperature and

$\sigma = 5.67 \times 10^{-8}\frac{W}{m^2 \cdot K^4}$ Is the Stefan-Boltzman constant.

Question: 16.004P

What is the physical mechanism of heat conduction in a solid, a liquid, and a gas?

Step 1 of 1

Physical mechanism of heat conduction in solid is due to the combination of the vibrations of the molecules in a lattice and the energy transport by free electrons. In gases and liquids, it is due to the collisions of the molecules during their random motion.

Question: 16.005P

Consider heat transfer through a windowless wall of a house on a winter day. Discuss the parameters that affect the rate of heat conduction through the wall.

Step 1 of 3

Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. So in this method heat can be transfer from solid to solid. Conduction is expressed by Fourier's law of conduction as

$$Q = kA\frac{dt}{dx}$$

Where

$\frac{dt}{dx}$ is the temperature gradient,

K is the thermal conductivity, and

A is the area which is normal to the direction of heat transfer.

From the above expression we know that rate of heat transfer directly proportional to thermal conductivity and inversely proportional to the wall thickness

Step 2 of 3

Thermal conductivity: It is a property of a material by which the rate of heat transfer is measure through a unit thickness of the material per unit area and per unit temperature difference. It is a measure of the ability of material to conduct heat'

A high value of thermal conductivity signifies that material is highly conductor of heat and low value how that material is less conductor of heat that mean insulator.

Step 3 of 3

The parameters that affect the rate of heat conduction through a windowless wall are

- The geometry and
- Surface area of wall,
- Its thickness,
- The material of the wall,
- And the temperature difference across the wall.

Question: 16.006P

Write down the expressions for the physical laws that govern each mode of heat transfer, and identify the variables involved in each relation.

Step 1 of 3

There are three mechanisms of heat transfer which are conduction, convection and radiation.

Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. So in this method heat can be transfer from solid to solid. Conduction is expressed by Fourier's law of conduction as

$$Q = kA\frac{dt}{dx}$$

Where

$\frac{dt}{dx}$ is the temperature gradient,

K is the thermal conductivity, and

A is the area which is normal to the direction of heat transfer.

Step 2 of 3

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. So when heat is transfer between solid to fluid or vice versa Convection is expressed by Newton's law of cooling as

$$Q_{conv} = hA(T_0 - T_\infty)$$

Where

h is the convection heat transfer coefficient,

A Is the surface area through which convection heat transfer takes place,

T_0 Is the surface temperature and

T_∞ Is the temperature of the fluid sufficiently far from the surface.

Step 3 of 3

Radiation is energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules. So in radiation no medium is required. Radiation is expressed by Stefan-Boltzman law as

$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{surr}^4)$

Where

ε Is the Emissivity of surface,

A_s Is the surface area,

T_s Is the surface temperature

T_{surr} is average surrounding surface temperature and

$\sigma = 5.67\times10^{-8}\frac{W}{m^2\cdot K^4}$ Is the Stefan-Boltzman constant.

Question: 16.007P

How does heat conduction differ from convection?

Step 1 of 2

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. So when heat is transfer between solid to fluid or vice versa Convection is expressed by Newton's law of cooling as

$$Q_{conv} = hA(T_0 - T_\infty)$$

Where

h is the convection heat transfer coefficient,

A Is the surface area through which convection heat transfer takes place,

T_0 Is the surface temperature and

T_∞ Is the temperature of the fluid sufficiently far from the surface.

Step 2 of 2

In the case of solids we can only conduction. Because of there is no fluid motion in the solids so amount of heat transfer is only due to conduction only .

Question: 16.008P

Does any of the energy of the sun reach the earth by conduction or convection?

Step 1 of 1

Conduction and convection requires the presence of material medium and heat transfer through a vacuum is by radiation only. Since space is almost completely empty so the heat energy of the sun reaches the earth by radiation only. Therefore there is no energy of the sun reach the earth by conduction and convection.

Question: 16.009P

How does forced convection differ from natural convection?

Step 1 of 1

If the fluid is forced to flow over the surface by external means such as fan, pump or blower then the convection is called Forced convection.
And if the fluid motion is caused by buoyancy forces that are induced by density differences due to variation of temperature in the fluid then the convection is called Natural convection.

Question: 16.010P

Define emissivity and absorptivity. What is Kirch-hoff's law of radiation?

Step 1 of 3

Emissivity:
It is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Emissivity is denoted by ε.
and its Value of Emissivity is $0 \leq \varepsilon \leq 1$
So Emissivity is a property of a material which shows how closely a surface approximates a black body whose emissivity is 1.

Step 2 of 3

Absorptivity:
It is the fraction of radiation incident on a surface that is absorbed by the surface. It is denoted by α.
And its value is $0 \leq \alpha \leq 1$.
A black body can absorb entire radiation incident on it. So for black body the value of α is 1.

Step 3 of 3

Kirchoffs Law of radiation: -
The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength.

Question: 16.011P

What is a blackbody? How do real bodies differ from blackbodies?

Step 1 of 1

A blackbody is an idealized body which emits the maximum amount of radiation at a given temperature and which absorbs the entire radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

Question: 16.012P

Judging from its unit W/m·K, can we define thermal conductivity of a material as the rate of heat transfer through the material per unit thickness per unit temperature difference? Explain.

Step 1 of 1

No, such a definition will imply that doubling the thickness would double the heat transfer rate. The equivalent but "more correct" unit of thermal conductivity is $\frac{W \cdot m}{m^2 \cdot {}^\circ C}$

That indicates product of heat transfer rate and thickness per unit surface area per unit temperature difference.

Question:

16.013P

Consider heat loss through the two walls of a house on a winter night. The walls are identical, except that one of them has a tightly fit glass window. Through which wall will the house lose more heat? Explain.

Step 1 of 2

Then we know that heat transfer rate can calculate by

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L}.$$

From the above expression we know that rate of heat transfer directly proportional to thermal conductivity and inversely proportional to the wall thickness.

Step 2 of 2

In this house, heat loss through the wall with glass window will be larger since the glass is much thinner than a wall, and its thermal conductivity is higher than the average conductivity of a wall.

Question: 16.014P

Consider two houses that are identical, except that the walls are built using bricks in one house, and wood in the other. If the walls of the brick house are twice as thick, which house do you think will be more energy efficient?

Step 1 of 2

Thermal conductivity (K) of brick $K_b = 0.72 \frac{W}{m \cdot K}$

And for wood it is $K_W = 0.17 \frac{W}{m \cdot K}$

So if the brick wall thickness is twice than the wood wall thickness.
Then we know that heat transfer rate can calculate by

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L} .$$

From the above expression we know that rate of heat transfer directly proportional to thermal conductivity and inversely proportional to the wall thickness.

Step 2 of 2

So if the brick wall thickness is twice than the wood wall thickness.
For brick wall value of thermal conductivity

$$k = \frac{0.72 \frac{W}{m \cdot K}}{2}$$

$$k = 0.36 \frac{W}{m \cdot K}$$

And for wood $0.17 \frac{W}{m^2 \cdot K}$ rest parameter are same.

The house with the lower rate of heat transfer through the walls will be the more efficient. so from clearly shows that wood wall is more efficient then the brick wall because of it has lower thermal conductivity than the brick.

Question: 16.015P

Consider two walls of a house that are identical except that one is made of 10-cm-thick wood, while the other is made of 25-cm-thick brick. Through which wall will the house lose more heat in winter?

Step 1 of 4

Given:

Thickness of the wood $L_{thick} = 10\,\text{cm}$
$= 0.1\,\text{m}$

Thickness of the Brick $L_{Brick} = 25\,\text{cm}$
$= 0.25\,\text{m}$

Step 2 of 4

From the Data book we get the

Thermal conductivity of the Wood is $k = 0.16\dfrac{\text{W}}{\text{m}\cdot\text{K}}$

Thermal conductivity of the Brick is $k_b = 0.72\dfrac{\text{W}}{\text{m}\cdot\text{K}}$

Step 3 of 4

The rate of heat transfer through both walls can be expressed as

$$\dot{Q}_{\text{wood}} = k_{\text{wood}} A\frac{T_1 - T_2}{L_{\text{wood}}}$$
$$= (0.16\ \text{W/m.°C})A\frac{T_1 - T_2}{0.1\,\text{m}}$$
$$= 1.6A(T_1 - T_2)$$

Step 4 of 4

$$\dot{Q}_{\text{brick}} = k_{\text{brick}} A\frac{T_1 - T_2}{L_{\text{brick}}}$$
$$= (0.72\ \text{W/m}\cdot\text{°C})A\frac{(T_1 - T_2)}{0.25\,\text{m}}$$
$$= 2.88A(T_1 - T_2)$$

Therefore, heat transfer through the brick wall will be larger despite its higher thickness.

Question: 16.016P

How do the thermal conductivity of gases and liquids vary with temperature?

Step 1 of 1

For gases, the molecules are in continues random motion. as the temperature increases, velocities of molecules become higher than in a lower temperature region. the molecules move from a high temperature region to a region of low temperature and gives up their energy through collisions to lower energy molecules. thus the thermal conductivity of gas increases with an increases in temperature and is proportional to the square root of the absolute temperature

The thermal conductivity of most liquids, however, decreases with increasing temperature, with water being a notable exception.

Question: 16.017P

Why is the thermal conductivity of superinsulation orders of magnitude lower than the thermal conductivity of ordinary insulation?

Step 1 of 1

Super insulations are obtained by using layers of highly reflective sheets separated by glass fibers in an evacuated space. Radiation heat transfer between two surfaces is inversely proportional to the number of sheets used and thus heat loss by radiation will be very low by using these highly reflective sheets.

At the same time, evacuating the space between the layers forms a vacuum under 0.000001 atm pressure which minimizes conduction or convection through the air space between the layers. Thermal conductivity of super-insulation is very low.

Question: 16.018P

Why do we characterize the heat conduction ability of insulators in terms of their apparent thermal conductivity instead of the ordinary thermal conductivity?

Step 1 of 1

Generally insulations are obtained by mixing fibers, powders, or flakes of insulating materials with air. Heat transfer through such insulations is by three modes of heat transfer namely, conduction, convection or radiation.
Such systems are characterized by apparent thermal conductivity instead of the ordinary thermal conductivity in order to incorporate these convection and radiation effects.

Question: 16.019P

Consider an alloy of two metals whose thermal conductivities are *k*1 and *k*2. Will the thermal conductivity of the alloy be less than *k*1 greater than *k*2, or between *k*1 and *k*2?

Step 1 of 1

Suppose an alloy made of two metals of thermal conductivities k_1 and k_2, then thermal conductivity of alloy k is in between k_1 and k_2. But this turns out not to be the case. The thermal conductivity of an alloy of two metals is usually much lower than that of either metal.

For example,

At 300 K

Thermal conductivity of Copper, $k_{Cu} = 401\,W/m \cdot °C$

Thermal conductivity of Aluminum, $k_{Al} = 237\,W/m \cdot °C$

Thermal conductivity of commercial bronze $(90\%Cu + 10Al)$, $k_{bronze} = 52\,W/m \cdot °C$

Question: 16.020P

Two surfaces of a 2-cm-thick plate are maintained at 0°C and 80°C, respectively. If it is determined that heat is transferred through the plate at a rate of 500 W/m2, determine its thermal conductivity.

Step 1 of 2

Given

Thickness of plate, $L = 2\,\text{cm}$

Temperatures both surfaces, $T_1 = 0°\text{C}$

$T_2 = 80°\text{C}$

Heat transfer rate through the plate, $\dot{Q} = 500\,\text{W/m}^2$

Step 2 of 2

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Question: 16.021P

Four power transistors, each dissipating 12 W, are mounted on a thin vertical aluminum plate 22 cm × 22 cm in size. The heat generated by the transistors is to be dissipated by both surfaces of the plate to the surrounding air at 25°C, which is blown over the plate by a fan. The entire plate can be assumed to be nearly isothermal, and the exposed surface area of the transistor can be taken to be equal to its base area. If the average convection heat transfer coefficient is 25 W/m2·K, determine the temperature of the aluminum plate. Disregard any radiation effects.

Step 1 of 2

Calculate the area of cross section.

$$A_s = 2(l \times b)$$

Here, l is the length, and b is the breadth.

Substitute $22\ \text{cm}$ for l and $22\ \text{cm}$ for b .

$$\begin{aligned} A_s &= 2\times(22\times22)\text{cm}^2 \times \frac{10^{-4}\ \text{m}^2}{1\ \text{cm}^2} \\ &= 2\times(22\times22\times10^{-4}) \\ &= 0.0968\,\text{m}^2 \end{aligned}$$

Calculate the heat balance equation to the aluminum plate and surrounding air.

$$Q = hA_s\Delta T$$

Here, h is the heat transfer coefficient, A_s is the area of cross section, and ΔT is the temperature difference.

Substitute $25\ \text{W/m}^2\cdot\text{K}$ for h , $(T - T_{air})$ for ΔT , $25°\text{C}$ for T_{air} , and $0.0968\,\text{m}^2$ for A_s .

$$\begin{aligned} Q &= 25(0.0968)(T - T_{air}) \\ &= 25(0.0968)(T - 25) \quad \text{..... (1)} \\ &= 2.42(T - 25) \end{aligned}$$

Step 2 of 2

The heat dissipation from each transistor,

$$Q_T = 12\ \text{W}$$

Then total heat transfer is,

$$\begin{aligned} Q &= 4Q_T \\ &= 4\times12 \\ &= 48\ \text{W} \end{aligned}$$

Substitute $48\ \text{W}$ for Q_T in equation (1).

$$\begin{aligned} Q_T &= 2.42(T - 25) \\ 48 &= 2.42(T - 25) \\ (T - 25) &= 19.835 \\ T &= 44.835°\text{C} \end{aligned}$$

Therefore, the temperature of aluminum plate is approximately $\boxed{45°\text{C}}$.

Question: 16.022P

In a power plant, pipes transporting superheated vapor are very common. Superheated vapor is flowing at a rate of 0.3 kg/s inside a pipe with 5 cm in diameter and 10 m in Length. The pipe is located in a power plant at 20°C, and has a uniform pipe surface temperature of 100°C. If the temperature drop between the inlet and exit of the pipe is 30°C, and the specific heat of the vapor is 2190 J/kg·K, determine the heat transfer coefficient as a result of convection between the pipe surface and the surrounding.

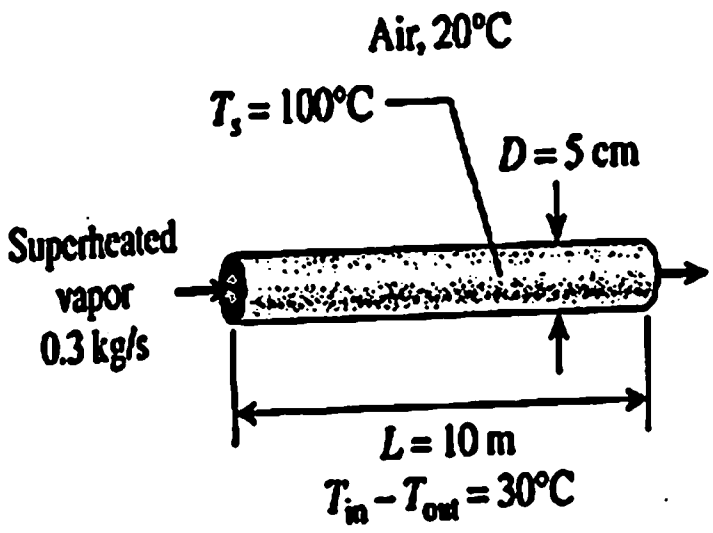


FIGURE P16–22

Step 1 of 3

Given data:

Heat is dissipated through the pipes of a power plant transporting superheated vapor

Superheated vapor mass flow rate, $\dot{m} = 0.3 kg/s$

Inside diameter of the pipe, $D = 5 cm$

Length of the pipe, $L = 10 m$

Temperature drop between inlet and exit of the pipe, $\Delta T_D = 30 °C$

Surface temperature of the pipe, $T_s = 100 °C$

Surrounding temperature of the air, $T_\infty = 20 °C$

Potential energy and kinetic energy changes are negligible

Step 2 of 3

The heat balance equation to the pipes,

$P_p \times \Delta t = \Delta U$

Taking the specific heat of vapor, $S = 2190 J/kg \cdot K$

Assume the convection heat transfer takes place from steam to pipe and then to the surrounding air respectively.

The rate of change in internal energy in the steam can be written as

$$\frac{\Delta U}{\Delta t} = \dot{m} S \Delta T_D$$
$$= 0.3 \times 2190 \times 30$$
$$= 19710 \text{ W}$$

Step 3 of 3

The power loss through the pipe to the ambient air by convection process so that from balance equation of heat, the heat transfer coefficient can be

$P_p \times \Delta t = \Delta U$

$$hA_s \Delta T = \frac{\Delta U}{\Delta t}$$

$$h \times (\pi \times d \times l) \times (100 - 20) = 19710$$

$$h \times (\pi \times 0.05 \times 10) \times (100 - 20) = 19710$$

$$h = 156.847 \text{ W/m}^2 \cdot \text{K}$$

Therefore, the convective heat transfer coefficient between pipe and air is

$$\boxed{h = 157 \text{ W/m}^2 \cdot \text{K}}$$

Question: 16.023P

An electric current of 5 A passing through a resistor has a measured voltage of 6 V across the resistor. The resistor is cylindrical with a diameter of 2.5 cm and length of 15 cm. The resistor has a uniform temperature of 90°C and the room air temperature is 20°C. Assuming that heat transfer by radiation is negligible, determine the heat transfer coefficient by convection.

Step 1 of 3

Given data:

An electric current passing through the resistor, $I = 5\text{A}$

Voltage across the resistor, $V = 6\text{V}$

Diameter of the cylindrical resistor, $D = 2.5\text{cm}$

Length of the cylindrical resistor, $L = 15\text{cm}$

The resistor has uniform temperature, $T = 90^\circ\text{C}$

The room air temperature, $T_{air} = 20^\circ\text{C}$

Step 2 of 3

Let h be the convective heat transfer coefficient,

The heat balance equation to the room and resistor can be

$$Q = hA_s\Delta T$$

For cylindrical resistor, surface area of heat dissipation to the air should be

$$A_s = \left(2\times\frac{\pi D^2}{4}\right)+\pi DL$$

$$= 2\times\frac{\pi\times(0.025)^2}{4}+\pi\times 0.025\times 0.15$$

$$= 0.0128\ \text{m}^2$$

Step 3 of 3

The rate of heat dissipation due to current passing through the resistor from the heat balance equation

$$Q = hA_s\Delta T$$

$$VI = h\times 0.0128\times(90-20)$$

$$h = \frac{5\times 6}{0.0128\times(90-20)}$$

$$= 33.48\ \text{W/m}^2\cdot\text{K}$$

Therefore, the convective heat transfer coefficient between resistor and air is

$$\boxed{h = 33.6\ \text{W/m}^2\cdot\text{K}}$$

Question: 16.024P

The inner and outer surfaces of a 4-m × 7-m brick wall of thickness 30 cm and thermal conductivity 0.69 W/m-K are maintained at temperatures of 26°C and 8°C, respectively. Determine the rate of heat transfer through the wall, in W.

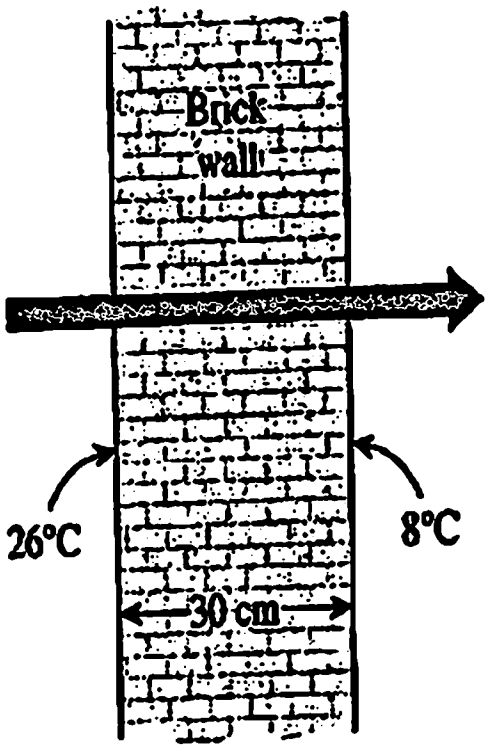


FIGURE P16–24

Step 1 of 2

Given data:

The dimensions of brick wall, $4\,\text{m} \times 7\,\text{m}$

Thickness of the wall, $L = 30\text{cm} = 0.3\text{m}$

Thermal conductivity, $k = 0.69\ \text{W/m} \cdot \text{K}$

Temperature of inner surface of the wall,

$T_1 = 26°\text{C}$

$= 26 + 273$

$= 299\ \text{K}$

Temperature of inner surface of the wall,

$T_2 = 8°\text{C}$

$= 8 + 273$

$= 281\ \text{K}$

Step 2 of 2

Heat transfer area of the wall,

$A = 4 \times 7$

$= 28\,\text{m}^2$

Under steady conditions,

The rate of heat transfer through the wall by conduction is given by

$$\dot{Q} = -kA\frac{T_2 - T_1}{L}$$

$$= -0.69 \times 28 \times \frac{(281 - 299)}{0.3}$$

$$= 1159.2\ \text{W}$$

Therefore, the rate of heat transfer through the wall by conduction is $\boxed{\dot{Q} = 1159.2\ \text{W}}$.

Question: 16.025P

The inner and outer surfaces of a 0.5-cm thick 2-m × 2-m window glass in winter are 10°C and 3°C, respectively If the thermal conductivity of the glass is 0.78 W/m·K, determine the amount of heat loss through the glass over a period of 5 h. What would your answer be if the glass were 1 cm thick?

Step 1 of 4

Given

Inner temperature of window glass, $T_1 = 10°C$

Outer surface temperature of window glass, $T_1 = 3°C$

Thickness of window glass $L = 0.5$ cm

Thermal conductivity of the glass, $k = 0.78\ W/m \cdot C$

Time period, $t = 5h$

Step 2 of 4

Area of window glass, $A = 2 \times 2$

$= 4\ m^2$

The rate of heat transfer through the glass by conduction is given by

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L}$$

$$= 0.78 \times 4 \times \frac{(10-3)}{0.005}$$

$$= \frac{21.84}{0.005}$$

$$= 4368\ W$$

Step 3 of 4

Then the amount of heat transfer over a period of $t = 5h$ becomes

$$Q = \dot{Q}_{cond} t$$

$$= 4368 \times (5 \times 3600)$$

$$= 78624000\ J$$

$$= 78.624\ MJ$$

The amount of heat transfer over a period of $t = 5h$, $\boxed{Q = 78.624\ MJ}$

Step 4 of 4

If the thickness of the glass doubled to $L = 1$ cm

Then the amount of heat transfer will go down by half.

So, amount of heat transfer, $Q_{L=1cm} = \frac{Q_{L=0.5cm}}{2}$

$$= \frac{78.6}{2}$$

$$= 39.3MJ$$

The amount of heat transfer over a period of $t = 5h$, $\boxed{Q = 39.3MJ}$

Question: 16.026P

Reconsider Prob. 16–25. Using EES (or other) software, plot the amount of heat loss through the glass as a function of the window glass thickness in the range of 0.1 cm to 1.0 cm. Discuss the results.

Step 1 of 7

Given

Inner temperature of window glass, $T_1 = 10°C$

Outer surface temperature of window glass, $T_1 = 3°C$

Thickness range of window glass, $0.1\text{cm} \le L \le 1.0\text{ cm}$

Thermal conductivity of the glass, $k = 0.78\ \text{W/m} \cdot \text{C}$

Time period, $t = 5\text{h}$

Step 2 of 7

To solve this question using EES software open EES software

Then FILE->New

A dailogbox will open type the following in dailog box

```
"GIVEN"
A=2*2 "[m^2]"
T_1=10 "[C]"
T_2=3 "[C]"
k=0.78 "[W/m-C]"
time=5*3600 "[s]"

"ANALYSIS"
Q_dot_cond=k*A*(T_1-T_2)/L
Q_cond=Q_dot_cond*time*Convert(J, kJ)
```

Step 3 of 7

Then from Tables -> New parametric table

A dailog box will open

Select no of runs = 10

Select variables in equations as L and Q_{cond} press ADD and then press ok

A table will appear put values of L as 0.001,0.002,0.003....0.01 in run 1to 10

Then from Calculate -> Solve table

Step 4 of 7

Solution will appear

L (m)	Q_{cond} (kJ)
0.001	393120
0.002	196560
0.003	131040
0.004	98280
0.005	78624
0.006	65520
0.007	56160
0.008	49140
0.009	43680
0.01	39312

Step 5 of 7

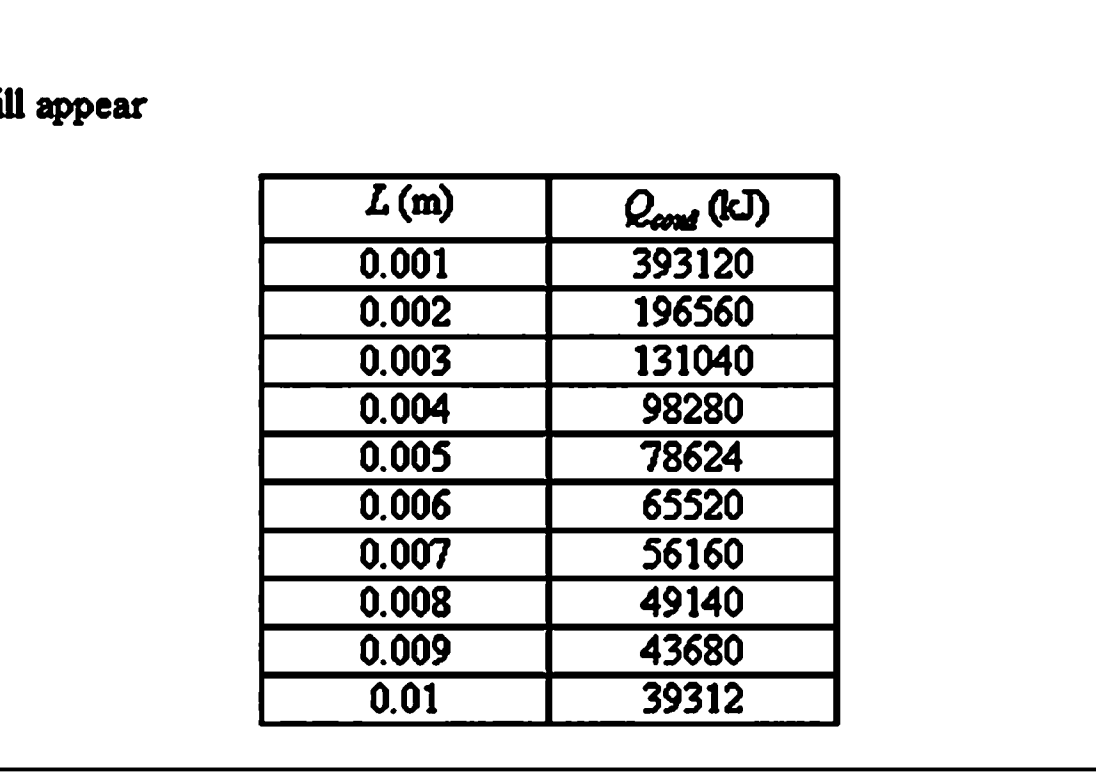
Then from Plot -> New plot window -> X-Y Plot

Select X-axis L and Y-axis Q_{cond}

Press OK

Step 6 of 7

Graph will be plotted.



Step 7 of 7

Result:

Heat conduction through the glass is decreased with increasing thickness of glass.

Question: 16.027P

An aluminum pan whose thermal conductivity 237 W/m·K has a flat bottom with diameter 15 cm and thickness 0.4 cm. Heat is transferred steadily to boiling water in the pan through its bottom at a rate of 1400 W. If the inner surface of the bottom of the pan is at 105°C, determine the temperature of the outer surface of the bottom of the pan.

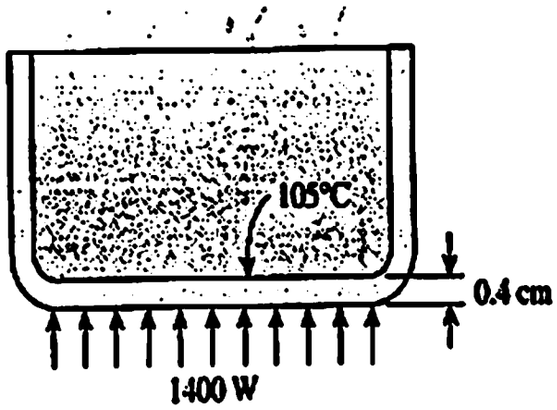


FIGURE P16–27

Step 1 of 2

Given data:

Thermal conductivity of the aluminum, $k = 237\ \text{W/m}\cdot\text{K}$

Diameter of the pan, $d = 15\,\text{cm} = 0.15\,\text{m}$

Thickness, $L = 0.4\,\text{cm} = 4\times10^{-3}\ \text{m}$

Total heat transfer rate through bottom, $Q = 1400\,\text{W}$

Temperature of inner surface of bottom of pan,

$T_1 = 105°\text{C}$

$= 105 + 273$

$= 378\,\text{K}$

Step 2 of 2

Area of the pan,

$$A = \frac{\pi d^2}{4}$$

$$= \frac{\pi \times 0.15^2}{4}$$

$$= 0.0177\ \text{m}^2$$

Under steady state condition,

The rate of heat transfer through the bottom of the pan by conduction is given by

$$\dot{Q} = kA\frac{T_2 - T_1}{L}$$

$$1400 = 237 \times 0.0177 \times \frac{(T_2 - 378)}{0.004}$$

$$T_2 - 378 = 1.334$$

$$T_2 = 379.334\ \text{K}$$

Therefore, the temperature of outer surface of bottom of pan is $\boxed{T_2 = 379.334\ \text{K}}$.

Question: 16.028P

The north wall of an electrically heated home is 20 ft long. 10 ft high, and 1 fl thick, and is made of brick whose thermal conductivity is $k = 0.42$ Btu/h·ft·°F. On certain winter night, the temperatures of the inner and the outer surfaces of the wall are measured to be at about 62°F and 25°F, respectively, for a period of 8 h. Determine (*a*) the rate of heat loss through the wall that night and (*b*) the cost of that heat loss to the home owner if the cost of electricity is $0.07/kWh.

Step 1 of 3

Given

Length of the wall, $L = 20\,\text{ft}$

Height of the wall, $h = 10\,\text{ft}$

Thickness of the wall, $t = 1\text{ft}$

Thermal conductivity of brick, $k = 0.42\,\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

Temperature of inner surface of the wall, $T_1 = 62^\circ\text{F}$

Temperature of outer surface of the wall, $T_2 = 25^\circ\text{F}$

Time period, $\Delta t = 8\text{h}$

Cost of electricity, $0.07/kWh

Step 2 of 3

(a) Surface area of wall, $A = L \times h$

$= 20 \times 10$

$= 200\,\text{ft}^2$

Rate of heat transfer through the wall in the night is given by

$$\dot{Q} = kA\frac{T_1 - T_2}{L}$$

$$= 0.42 \times 200 \times \frac{(62-25)}{1}$$

$$= 3108\,\text{Btu/h}$$

Rate of heat transfer through the wall, $\boxed{\dot{Q} = 3108\,\text{Btu/h}}$

Step 3 of 3

(b) Rate of heat transfer through the wall, $\dot{Q} = 3108\,\text{Btu/h}$

$= 0.911\,\text{kW}$ $(\because 1\text{kW} = 3412\ \text{Btu/h})$

The amount of heat lost during an 8 hour period

$$Q = \dot{Q}\Delta t$$

$$= 0.911 \times 10^3 \times 8$$

$$= 7.288\,\text{kWh}$$

Cost = (Amount of energy)(Unit cost of energy)

$= 7.288 \times 0.07$

$= \$0.51$

The cost of the heat loss through the wall, $\boxed{\$0.51}$

Question: 16.029P

In a certain experiment, cylindrical samples of diameter 4 cm and length 7 cm are used (see Fig. 16–9). The two thermocouples in each sample are placed 3 cm apart. After initial transients, the electric heater is observed to draw 0.6 A at 110 V, and both differential thermometers read a temperature difference of 8°C. Determine the thermal conductivity of the sample.

Step 1 of 4

Given data:

Diameter of the cylinder, $d = 4\,\text{cm} = 0.04\,\text{m}$

Length of the cylinder, $L = 7\,\text{cm} = 0.07\,\text{m}$

Distance between two thermocouples, $a = 3\,\text{cm} = 0.03\,\text{m}$

Temperature difference reading,

$\Delta T = 8^\circ\text{C} = 8\,\text{K}$

Specifications of electric heater

Current, $I = 0.6\,\text{A}$

Voltage, $V = 110\,\text{V}$

Step 2 of 4

Area of the surface,

$$A = \frac{\pi d^2}{4}$$

$$= \frac{\pi \times 0.04^2}{4}$$

$$= 0.00125\,\text{m}^2$$

Step 3 of 4

The electrical power consumed by the heater is given by

$$\dot{W}_e = VI$$

$$= 110 \times 0.6$$

$$= 66\,\text{W}$$

The rate of heat flow through each sample is

$$\dot{Q} = \dot{W}_e / 2$$

$$= 66/2$$

$$= 33\,\text{W}$$

Step 4 of 4

Thermal conductivity of the sample can be calculating as

$$\dot{Q} = kA\frac{\Delta T}{L}$$

$$k = \frac{\dot{Q}L}{A\Delta T}$$

$$= \frac{33 \times 0.03}{0.00125 \times 8}$$

$$= \frac{0.99}{0.01}$$

$$= 99\,\text{W/m} \cdot \text{K}$$

Therefore, the thermal conductivity of the sample is $\boxed{k = 99\,\text{W/m} \cdot \text{K}}$

Question: 16.030P

One way of measuring the thermal conductivity of a material is to sandwich an electric thermofoil heater between two identical rectangular samples of the material and to heavily insulate the four outer edges, as shown in the figure. Thermocouples attached to the inner and outer surfaces of the samples record the temperatures.

During an experiment, two 0.5-cm thick samples 10 cm × 10 cm in size arc used. When steady operation is reached, the heater is observed to draw 25 W of electric power, and the temperature of each sample is observed to drop from 82°C at the inner surface to 74°C at the outer surface. Determine the thermal conductivity of the material at the average temperature.

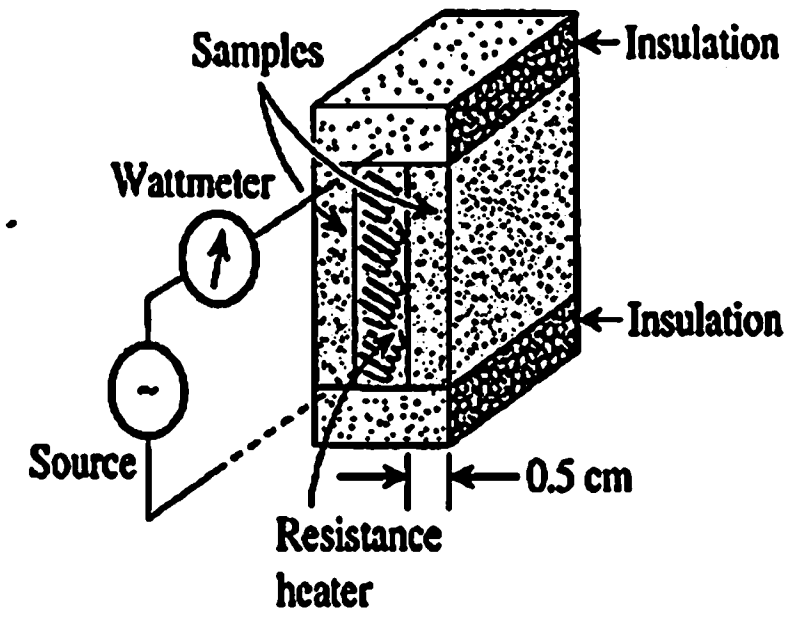


FIGURE P16–30

Step 1 of 2

Calculate the area of each sample as follows:

$$A=\left(10\,\text{cm}\times\frac{1\,\text{m}}{100\,\text{cm}}\right)\times\left(10\,\text{cm}\times\frac{1\,\text{m}}{100\,\text{cm}}\right)$$
$$=0.01\,\text{m}^2$$

Find the rate of heat transfer for each sample using the following equation.

$$\dot{Q}=\frac{W_{\text{elec}}}{2}$$

Substitute $25\,\text{W}$ for W_{elec}.

$$\dot{Q}=\frac{25}{2}$$
$$=12.5\,\text{W}$$

Step 2 of 2

Calculate the thermal conductivity using the following equation.

$$\dot{Q}=kA\frac{(T_2-T_1)}{L}$$
$$k=\frac{\dot{Q}L}{A(T_2-T_1)}$$

Here, k is the thermal conductivity of the material, L is the thickness of the insulation, A is the cross sectional area, T_1 and T_2 are the temperatures at inside and outside respectively.

Substitute $12.5\,\text{W}$ for $\dot{Q}$, $0.005\,\text{m}$ for L, $0.01\,\text{m}^2$ for A, $82°\text{C}$ for T_2 and $74°\text{C}$ for T_1.

$$k=\frac{12.5\times0.005}{0.01(82-74)}$$
$$=\frac{0.0625}{0.08}$$
$$=0.781\,\text{W/m}\cdot°\text{C}$$

Therefore, the thermal conductivity of the material is $\boxed{0.781\,\text{W/m}\cdot°\text{C}}$

Question: 16.031P

Repeat Prob. 16–30 for an electric power consumption of 20W.

Step 1 of 2

Given

Thickness of sample, $L = 0.5$ cm

Area of sample, $A = 10\,\text{cm} \times 10\,\text{cm}$

Temperature of outer surface, $T_2 = 82°\text{C}$

Temperature of inner surface, $T_1 = 74°\text{C}$

Power consumed by heater, $P = 20\,\text{W}$

Step 2 of 2

Area of each sample, $A = 0.1 \times 0.1$

$= 0.01\,\text{m}^2$

Rate of heat transfer for each sample

$\dot{Q} = 20/2$

$= 10\,\text{W}$

And we have

$$\therefore \dot{Q} = kA\frac{(T_2 - T_1)}{L}$$

$$\Rightarrow k = \frac{\dot{Q}L}{A(T_2 - T_1)}$$

$$= \frac{10 \times 0.005}{0.01(82-74)}$$

$$= \frac{0.05}{0.08}$$

$$= 0.625\,\text{W/m} \cdot °\text{C}$$

The thermal conductivity of the material, $\boxed{k = 0.625\,\text{W/m} \cdot °\text{C}}$

Question: 16.032P

A heat flux meter attached to the inner surface of a 3-cm-thick refrigerator door indicates a heat flux of 32 W/m2 through the door. Also, the temperatures of the inner and the outer surfaces of the door are measured to be 7°C and 15°C. respectively. Determine the average thermal conductivity of the refrigerator door.

Step 1 of 2

Given data:

Thickness of refrigerator door, $L = 3\text{cm} = 0.03\text{ m}$

Heat flux through the door, $\dot{q} = 32\ \text{W/m}^2$

Temperature of inner surface,

$T_1 = 7°\text{C}$

$= 7 + 273$

$= 280\ \text{K}$

Temperature of outer surface,

$T_2 = 15°\text{C}$

$= 15 + 273$

$= 288\ \text{K}$

Step 2 of 2

Heat flux through the slab is given by

$$\dot{q} = k\frac{(T_2 - T_1)}{L}$$

$$k = \frac{\dot{q}L}{(T_2 - T_1)}$$

$$= \frac{32 \times 0.03}{(288 - 280)}$$

$$= \frac{0.96}{8}$$

$$= 0.12\ \text{W/m} \cdot \text{K}$$

Therefore, the average thermal conductivity of the door material is $\boxed{k = 0.12\ \text{W/m} \cdot \text{K}}$

Question: 16.033P

Consider a person standing in a room maintained at 20°C at all times. The inner surfaces of the walls, floors, and ceiling of the house are observed to be at an average temperature of 12°C in winter and 23°C in summer. Determine the rates of radiation heat transfer between this person and the surrounding surfaces in both summer and winter if the exposed surface area, emissivity, and the average outer surface temperature of the person are 1.6 m2. 0.95, and 32°C, respectively.

Step 1 of 3

Given

Temperature of inner surface in winter, $T_{winter} = 12°C$

Temperature of inner surface in summer, $T_{summer} = 23°C$

Emissivity of person, $\varepsilon = 0.95$

Surface area, $A = 1.6\,m^2$

Temperature of outer surface, $T_s = 32°C$

Step 2 of 3

In summer:

Rate of radiation heat transfer is given by

$$\dot{Q}_{rad} = \varepsilon\sigma A\left(T_s^4 - T_{summer}^4\right)$$

$$= 0.95\times\left(5.67\times10^{-8}\right)\times1.6\times\left[(32+273)^4 - (23+273)^4\right]$$

$$= 8.62\times10^{-8}\times\left[305^4 - 296^4\right]$$

$$= 84.2\,W$$

Rate of radiation heat transfer in summer, $\boxed{\dot{Q}_{rad} = 84.2\,W}$

Step 3 of 3

In winter:

Rate of radiation heat transfer is given by

$$\dot{Q}_{rad} = \varepsilon\sigma A\left(T_s^4 - T_{winter}^4\right)$$

$$= 0.95\times\left(5.67\times10^{-8}\right)\times1.6\times\left[(32+273)^4 - (12+273)^4\right]$$

$$= 8.62\times10^{-8}\times\left[305^4 - 285^4\right]$$

$$= 177.2\,W$$

Rate of radiation heat transfer in winter, $\boxed{\dot{Q}_{rad} = 177.2\,W}$

Question: 16.034P

Reconsider Prob. 16–33. Using EES (or other) software, plot the rate of radiation heat transfer in winter as a function of the temperature of the inner surface of the room in the range of 8°C to 18°C. Discuss the results.

Step 1 of 7

Given

Temperature range of inner surface in winter, $8°C \le T_{sur,winter} \le 18°C$

$281K \le T_{sur,winter} \le 291K$

Temperature of inner surface in summer, $T_{sur,sum} = 23°C$

Emissivity of person, $\varepsilon = 0.95$

Surface area, $A = 1.6 m^2$

Temperature of outer surface, $T_s = 32°C$

Step 2 of 7

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
T_infinity=20+273 "[K]"
"T_surr_winter=12+273 [K], parameter to be varied"
T_surr_summer=23+273 "[K]"
A=1.6 "[m^2]"
epsilon=0.95
T_s=32+273 "[K]"

"ANALYSIS"
sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzman constant"
"(a)"
Q_dot_rad_summer=epsilon*sigma*A*(T_s^4-T_surr_summer^4)
"(b)"
Q_dot_rad_winter=epsilon*sigma*A*(T_s^4-T_surr_winter^4)
```

Step 3 of 7

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 11
Select variables in equations as $T_{sur,winter}$ and $Q_{rad,winter}$ press ADD and then press ok
A table will appear put values of L as 281,282,283....291 in run 1to 11
Then from Calculate -> Solve table

Step 4 of 7

Solution will appear

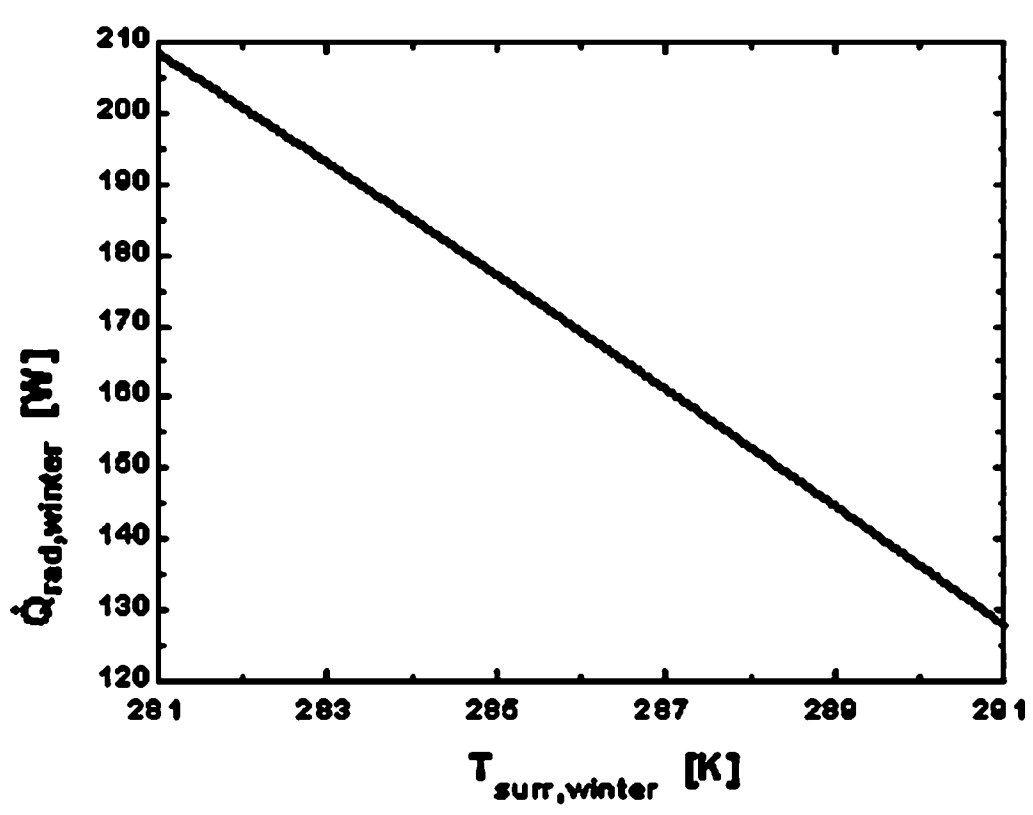
$T_{sur,winter}$ (T)	$Q_{rad,winter}$ (W)
281	208.5
282	200.8
283	193
284	185.1
285	177.2
286	169.2
287	161.1
288	152.9
289	144.6
290	136.2
291	127.8

Step 5 of 7

Then from Plot -> New plot window -> X-Y Plot
Select X-axis $T_{sur,winter}$ and Y-axis $Q_{rad,winter}$
Press OK

Step 6 of 7

Graph will be plotted



Step 7 of 7

Result:
Rate of radiation heat transfer is decreased with increasing temperature of surroundings in winter. It varies linearly.

Question: 16.035P

For heat transfer purposes, a standing man can be modeled as a 30-cm-diameter, 170-cm-long vertical cylinder with both the top and bottom surfaces insulated and with the side surface at an average temperature of 34°C. For a convection heat transfer coefficient of 8 W/m2·K, determine the rate of heat loss from this man by convection in an environment at l8°C.

Step 1 of 2

Given data:

A standing man can be modeled as a vertical cylinder.

Diameter of the cylinder, $d = 30\text{cm} = 0.03\text{m}$

Length of the cylinder, $L = 170\,\text{cm} = 1.7\,\text{m}$

Top and bottom surfaces are insulated

Temperature of side surface,

$T_2 = 34°\text{C}$

$= 34 + 273$

$= 307\,\text{K}$

Temperature of environment,

$T_1 = 18°\text{C}$

$= 18 + 273$

$= 291\text{K}$

Coefficient of convection heat transfer, $h = 8\,\text{W/m}^2\cdot\text{K}$

Step 2 of 2

Surface area of the person,

$A = \pi dL$

$= \pi \times 0.03 \times 1.7$

$= 0.1602\,\text{m}^2$

The rate of heat transfer by convection is given by

$\dot{Q}_{conv} = hA(T_2 - T_1)$

$= 8 \times 1.602 \times (307 - 291)$

$= 205.056\,\text{W}$

Therefore, the rate of heat loss from man by convection, $\boxed{\dot{Q}_{conv} = 205.056\,\text{W}}$

Question: 16.036P

Hot air at 80°C is blown over a 2-m × 4-m flat surface at 30°C. If the average convection heat transfer coefficient is 55 W/m2·K, determine the rate of heat transfer from the air to the plate, in kW.

Step 1 of 2

Given

Temperature of hot air, $T_2 = 80°C$

Dimensions of flat surface, $2\,m \times 4\,m$

Temperature of flat surface, $T_1 = 30°C$

Coefficient of heat transfer, $h = 55\,W/m^2 \cdot °C$

Step 2 of 2

Area of flat surface, $A = 2 \times 4$

$= 8\,m^2$

Rate of heat transfer by convection is given by

$$\dot{Q}_{conv} = hA(T_2 - T_1)$$

$$= 55 \times 8 \times (80 - 30)$$

$$= 22000\,W$$

$$= 22\,kW$$

Rate of heat transfer from air to the plate, $\boxed{\dot{Q}_{conv} = 22\,kW}$

Question: 16.037P

Reconsider Prob. 16–36. Using EES (or other) software, plot the rate of heat transfer as a function of the heat transfer coefficient in the range of 20 W/m2·K to 100 W/m2·K. Discuss the results.

Step 1 of 7

Given

Temperature of hot air, $T_\infty = 80°C$

Dimensions of flat surface, $2\,m \times 4\,m$

Temperature of flat surface, $T_s = 30°C$

Range of coefficient of heat transfer, $20\ W/m^2 \cdot °C \le h \le 55\ W/m^2 \cdot °C$

Step 2 of 7

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
T_infinity=80 "[C]"
A=2*4 "[m^2]"
T_s=30 "[C]"
"h=55 [W/m^2-C], parameter to be varied"

"ANALYSIS"
Q_dot_conv=h*A*(T_infinity-T_s)
```

Step 3 of 7

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 10
Select variables in equations as h and $\dot{Q}_{conv}$ press ADD and then press ok
A table will appear put values of h as 20,30,40....,100 in run 1to 10
Then from Calculate -> Solve table

Step 4 of 7

Solution will appear

$h\ (W/m^2 \cdot °C)$	$\dot{Q}_{conv}$ (W)
20	8000
30	12000

A dailogbox will open type the following in dailog box

```
"GIVEN"
T_infinity=80 "[C]"
A=2*4 "[m^2]"
T_s=30 "[C]"
"h=55 [W/m^2-C], parameter to be varied"

"ANALYSIS"
Q_dot_conv=h*A*(T_infinity-T_s)
```

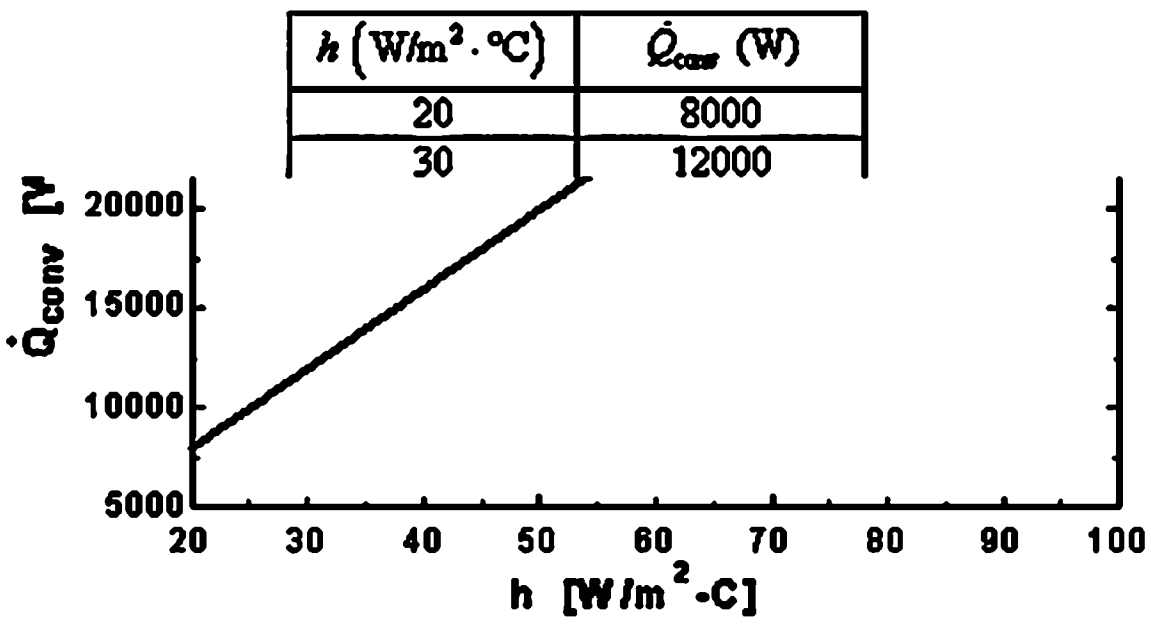
Step 3 of 7

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 10
Select variables in equations as h and $\dot{Q}_{conv}$ press ADD and then press ok
A table will appear put values of h as 20,30,40....,100 in run 1to 10
Then from Calculate -> Solve table

Step 4 of 7

Solution will appear

$h\ (W/m^2 \cdot °C)$	$\dot{Q}_{conv}$ (W)
20	8000
30	12000



Step 7 of 7

From graph, rate of heat transfer increases with coefficient of heat transfer increases linearly.

Question: 16.038P

The outer surface of a spacecraft in space has an emissivity of 0.8 and a solar absorptivity of 0.3. If solar radiation is incident on the spacecraft at a rate of 950 W/m2, determine the surface temperature of the spacecraft when the radiation emitted equals the solar energy absorbed.

Step 1 of 3

Given data:

The outer surface of a spacecraft has emissivity, $\varepsilon = 0.8$

The outer surface of a spacecraft has absorptivity, $\alpha = 0.3$

The rate of solar radiation is incident on the spacecraft, $q_{rad} = 950\ W/m^2$

Step 2 of 3

Assume the radiation emitted is equal to the solar energy absorbed.

Heat absorbed by the spacecraft due to solar radiation,

$$\begin{aligned} Q_{abs} &= \alpha A_s q_{rad} \\ &= 0.3 \times A_s \times 950 \\ &= 258 A_s \end{aligned}$$

Heat transferred from the spacecraft due to solar radiation,

$$\begin{aligned} Q_{rad} &= \varepsilon \sigma A_s T^4 \\ &= 0.8 \times \left(5.67 \times 10^{-8}\right) \times A_s \times T^4 \\ &= 4.536 \times 10^{-8} A_s \times T^4 \end{aligned}$$

Step 3 of 3

From the given condition of equality between the radiation emitted and the solar energy absorbed the temperature,

$$Q_{abs} = Q_{rad}$$

$$285 A_s = 4.536 \times 10^{-8} A_s \times T^4$$

$$\begin{aligned} T^4 &= \frac{285}{4.536 \times 10^{-8}} \\ &= \sqrt[4]{6.283 \times 10^9} \\ &= 281.5\text{K} \end{aligned}$$

Therefore, the surface temperature of spacecraft, $\boxed{T = 282\ \text{K}}$

$= 258 A_s$

Heat transferred from the spacecraft due to solar radiation,

$$\begin{aligned} Q_{rad} &= \varepsilon \sigma A_s T^4 \\ &= 0.8 \times \left(5.67 \times 10^{-8}\right) \times A_s \times T^4 \\ &= 4.536 \times 10^{-8} A_s \times T^4 \end{aligned}$$

Step 3 of 3

From the given condition of equality between the radiation emitted and the solar energy absorbed the temperature,

$$Q_{abs} = Q_{rad}$$

$$285 A_s = 4.536 \times 10^{-8} A_s \times T^4$$

Question: 16.039P

The heat generated in the circuitry on the surface of a silicon chip (k= 130 W/m·K) is conducted to the ceramic substrate to which it is attached. The chip is 6 mm × 6 mm in size and 0.5 mm thick and dissipates 5 W of power. Disregarding any heat transfer through the 0.5-mm high side surfaces, determine the temperature difference between the front and back surfaces of the chip in steady operation.

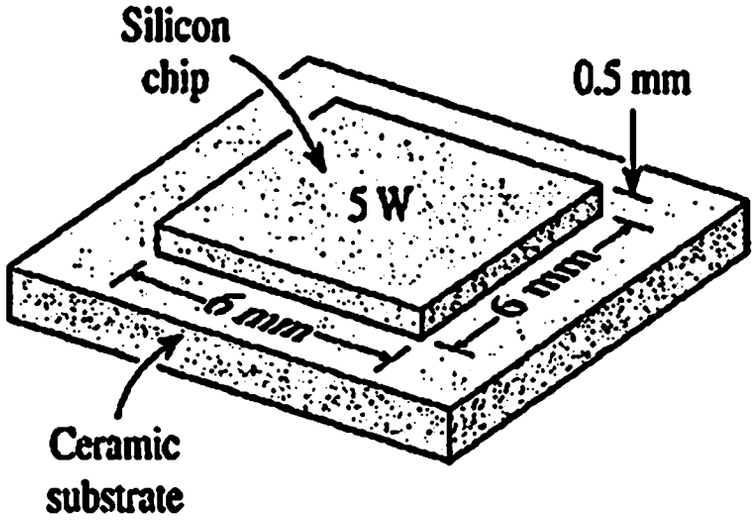


FIGURE P16–39

Step 1 of 2

Loading solution. Please wait ...

Step 2 of 2

Let the temperature difference between the front and back surfaces of the chip be ΔT
Heat transfer area of the chip,

$$A = 0.006 \times 0.006$$

$$= 3.6 \times 10^{-5}\ \text{m}^2$$

Heat transfer by conduction is given by

$$Q = kA\frac{\Delta T}{L}$$

$$\Delta T = \frac{QL}{kA}$$

$$= \frac{5 \times 0.0005}{130 \times 3.6 \times 10^{-5}}$$

$$= \frac{2.5 \times 10^{-3}}{4.68 \times 10^{-3}}$$

$$= 0.534\,\text{K}$$

Therefore, the temperature difference between the front and back surfaces of the chip,

$$\boxed{\Delta T = 0.534\,\text{K}}$$

Question: 16.040P

A 40-cm-long, 800-W electric resistance heating element with diameter 0.5 cm and surface temperature 120°C is immersed in 75 kg of water initially at 20°C. Determine how long it will take for this heater to raise the water temperature to 80°C. Also, determine the convection heat transfer coefficients at the beginning and at the end of the heating process.

Step 1 of 5

Given

Length of heating element, $L = 40\,\text{cm}$

Diameter of heating element, $d = 0.5\text{cm}$

Surface temperature of heating element, $T_s = 120°\text{C}$

Amount of water, $m = 75\text{kg}$

Initial temperature of water, $T_1 = 20°\text{C}$

Final temperature of water, $T_2 = 80°\text{C}$

Generated heat of heating element, $\dot{Q} = 800W$

Step 2 of 5

From table, properties of common liquids, solids and foods

Specific heat of water at room temperature, $C_p = 4.18\text{kJ/kg}\cdot\text{K}$

Here generated heat is equal to the rate of heat gain by water.

Noting that this is the only mechanism of energy transfer, the time it takes to raise the water temperature from 20°C to 80°C is determined to be

$$Q = mC_p(T_2 - T_1)$$

$$\Rightarrow \dot{Q}\Delta t = mC_p(T_2 - T_1)$$

$$\Rightarrow \Delta t = \frac{mC_p(T_2 - T_1)}{\dot{Q}}$$

$$= \frac{75\times 4.18\times 10^3 \times (80-20)}{800}$$

$$= \frac{18810000}{800}$$

$$= 23512s$$

$$= 6.53\text{h}$$

Time taken for heater to raise the water temperature to 80°C, $\boxed{\Delta t = 6.53\text{h}}$

Step 3 of 5

The surface area of the wire,

$$A = \pi dL$$

$$= \pi\times 0.005\times 0.4$$

$$= 0.00628\,\text{m}^2$$

The Newton's law of cooling for convection heat transfer is given by

$$\dot{Q} = hA(T_s - T_\infty)$$

Disregarding any heat transfer by radiation and thus assuming all the heat loss from the wire to occur by convection.

Step 4 of 5

The convection heat transfer coefficient at the beginning process is calculated as

$$h_1 = \frac{\dot{Q}}{A(T_s - T_{\infty 1})}$$

$$= \frac{800}{0.00628\times(120-20)}$$

$$= \frac{800}{0.628}$$

$$= 1273.8\,\text{W/m}^2\cdot°\text{C}$$

The convection heat transfer coefficient at the beginning process, $\boxed{h_1 = 1273.8\,\text{W/m}^2\cdot°\text{C}}$

Step 5 of 5

The convection heat transfer coefficient at the end of the process is determined as

$$h_2 = \frac{\dot{Q}}{A(T_s - T_{\infty 2})}$$

$$= \frac{800}{0.00628\times(120-80)}$$

$$= \frac{800}{0.2512}$$

$$= 3184.7\,\text{W/m}^2\cdot°\text{C}$$

The convection heat transfer coefficient at the end of the process, $\boxed{h_2 = 3184.7\,\text{W/m}^2\cdot°\text{C}}$

Question: 16.041P

A 5-cm-external-diameter, 10-m-long hot-water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/m2·K. Determine the rate of heat loss from the pipe by natural convection.

Step 1 of 2

Given

Diameter of hot water pipe, $d = 5\text{cm}$

Length of hot water pipe, $L = 10\text{m}$

Temperature of pipe, $T_2 = 80°\text{C}$

Temperature of surroundings, $T_2 = 5°\text{C}$

Coefficient convection heat transfer, $h = 25\,\text{W/m}^2 \cdot °\text{C}$

Step 2 of 2

The heat transfer surface area

$$\begin{aligned} A &= \pi dL \\ &= \pi \times 0.05 \times 10 \\ &= 1.571\text{m}^2 \end{aligned}$$

Under steady conditions,

The rate of heat transfer by convection is given by

$$\begin{aligned} \dot{Q}_{conv} &= hA(T_2 - T_1) \\ &= 25 \times 1.571 \times (80 - 5) \\ &= 2945.6\,\text{W} \end{aligned}$$

The rate of heat transfer by convection, $\boxed{\dot{Q}_{conv} = 2945.6\,\text{W}}$

Question: 16.042P

A hollow spherical iron container with outer diameter 20 cm and thickness 0.2 cm is filled with iced water at 0°C.

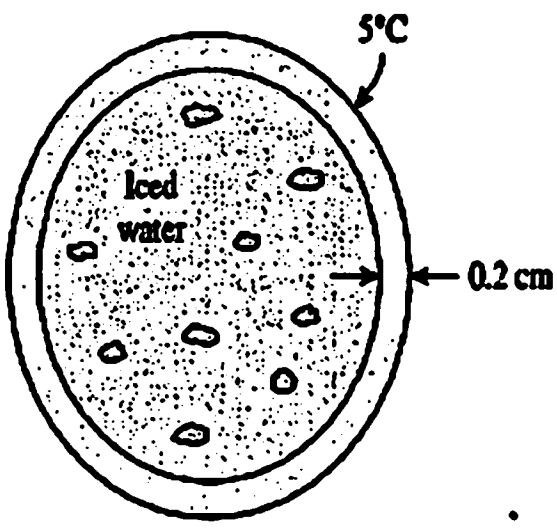


FIGURE P16–42

If the outer surface temperature is 5°C, determine the approximate rate of heat loss from the sphere in kW, and the rate at which ice melts in the container. The heat of fusion of water is 333.7 kJ/kg.

Step 1 of 4

Given data:

Outer diameter of the spherical container, $d_o = 20$ cm

Thickness of the container, $t = 0.2$ cm

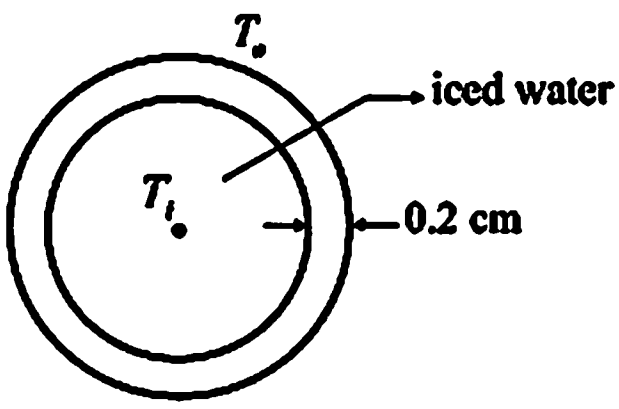
Temperature of iced water, $T_i = 0°C$

Outer surface temperature, $T_o = 5°C$

Heat of fusion of water, $h_{if} = 333.7$ kJ/kg

Step 2 of 4

A schematic of the given iron container is shown below.



Step 3 of 4

The conduction heat transfer through the iron container is given by,

$$Q_{cond} = kA\frac{\Delta T}{dx}$$

where, A is the outer surface area of the container.

k is the thermal conductivity of the container, i.e., $k_{Fe} = 80.2$ W/m·K

dx is the length of the container.

$$Q_{cond} = (80.2 \text{ W/m}\cdot\text{K})\left(4\pi\times(0.1 \text{ m})^2\right)\times\frac{(5-0) \text{ K}}{0.002 \text{ m}}$$

$$= 80.2\times0.125\times2500$$

$$= 25{,}195.57 \text{ W}$$

Therefore, the total heat transferred through the spherical container is $\boxed{Q_{cond} = 25.195 \text{ kW}}$.

Step 4 of 4

Rate at which ice melts is given by,

$$\dot{m} = \frac{Q}{h_{if}}$$

$$= \frac{25.195 \text{ kW}}{333.7 \text{ kJ/s}}$$

$$= 0.075 \text{ kg/s}$$

Hence, the rate at which the ice melts due to the conduction heat transfer is $\boxed{\dot{m} = 0.075 \text{ kg/s}}$.

Question: 16.043P

Reconsider Prob. 16–12. Using EES (or other) software, plot the rate at which ice melts as a function of the container thickness in the range of 0.1 cm to 1.0 cm. Discuss the results.

Step 1 of 5

Given data:

Outer diameter of the spherical container, $d_o = 20$ cm

Thickness of the container, $t = 0.2$ cm

Temperature of iced water, $T_i = 0°C$

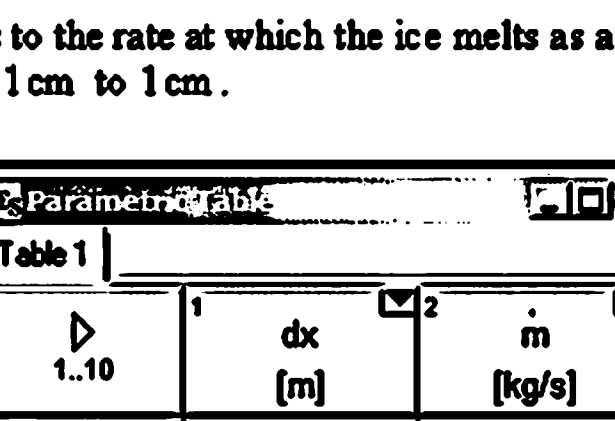
Outer surface temperature, $T_o = 5°C$

Heat of fusion of water, $h_{if} = 333.7$ kJ/kg

The thickness of the shell ranges from 0.1 cm to 1 cm.

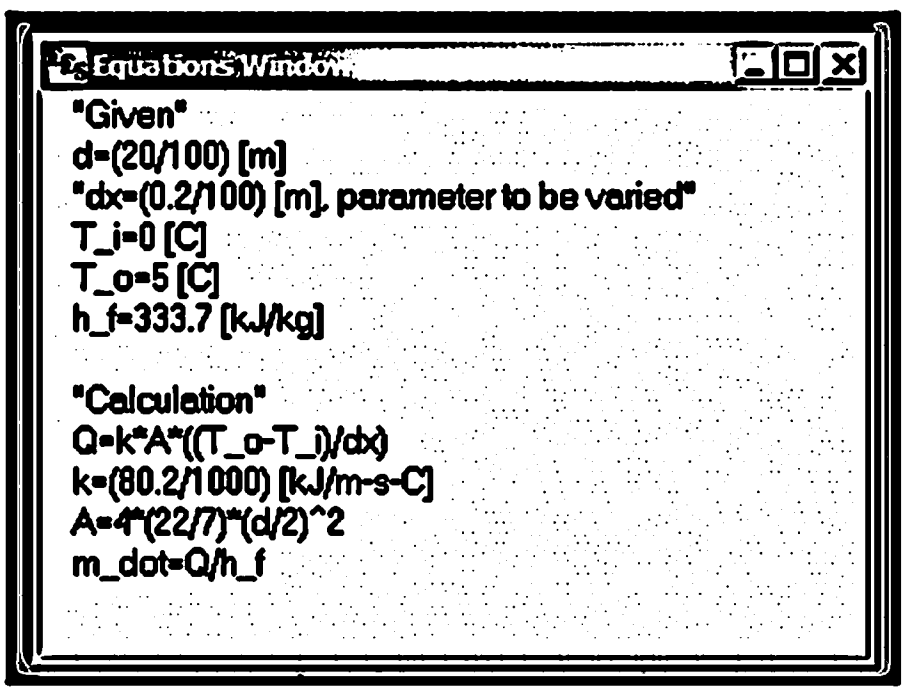
Step 2 of 5

A schematic of the given iron container is shown below.



Step 3 of 5

The following program represents the given parameters followed by the calculation of the rate of ice melt for the thickness of the shell varied.



Step 4 of 5

The following table refers to the rate at which the ice melts as a function of container thickness ranging from 0.1 cm to 1 cm.

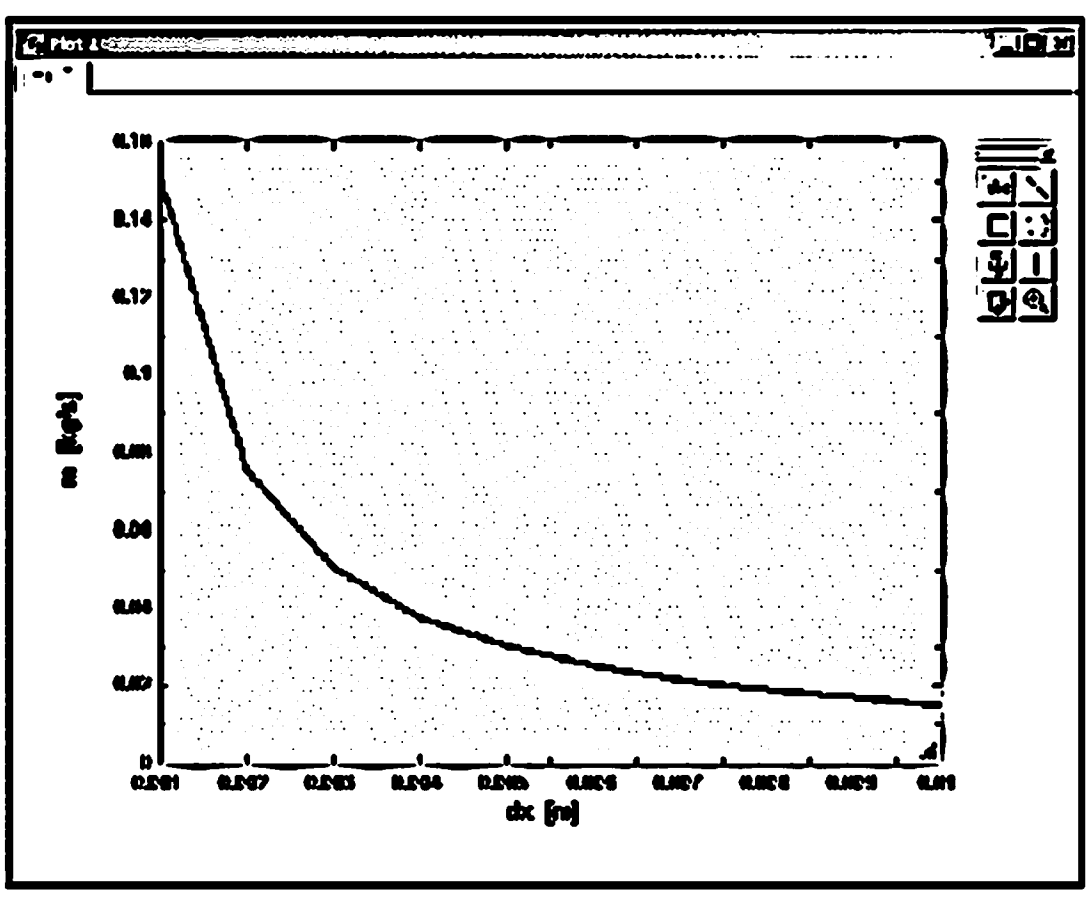
Parametric Table

Table 1

1..10	1 dx [m]	2 $\dot{m}$ [kg/s]
Run 1	0.001	0.1511
Run 2	0.002	0.07553
Run 3	0.003	0.05036
Run 4	0.004	0.03777
Run 5	0.005	0.03021
Run 6	0.006	0.02518
Run 7	0.007	0.02158
Run 8	0.008	0.01888
Run 9	0.009	0.01679
Run 10	0.01	0.01511

Step 5 of 5

Following is the graph showing the variation in the rate of ice melt as the thickness of the shell varies from 0.1 cm to 1 cm.



Question:

16.044P

The inner and outer glasses of a 4-ft × 4-ft double-pane window are at 60°F and 48°F, respectively. If the 0.25-in space between the two glasses is filled with still air, determine the rate of heat transfer through the window.

Step 1 of 3

Given

Dimensions of glass, $4\text{ft}\times 4\text{ft}$

Temperature of inner glass, $T_1 = 60°\text{F}$

Temperature of outer glass, $T_2 = 48°\text{F}$

Space between two glasses, $L = 0.25\text{in}$

Step 2 of 3

Surface area of glass, $A = 4\times 4$

$$= 16\text{ft}^2$$

The average temperature of glass,

$$T_{avg} = \frac{T_1+T_2}{2}$$

$$= \frac{60+48}{2}$$

$$= 54°\text{F}$$

Step 3 of 3

From table, properties of air at 1atm pressure,

The thermal conductivity of air at $54°\text{F}$, $k = 0.01419\text{Btu/h}\cdot\text{ft}\cdot°\text{F}$

The rate of heat loss through window

$$\dot{Q} = kA\frac{(T_1-T_2)}{L}$$

$$= 0.01419\times 16\times\frac{(60-48)}{(0.25/12)}$$

$$= \frac{32.694}{0.25}$$

$$= 130.8\text{Btu/h}$$

The rate of heat loss through window, $\boxed{\dot{Q} = 130.8\text{Btu/h}}$

Question: 16.045P

A transistor with a height of 0.4 cm and a diameter of 0 6 cm is mounted on a circuit board. The transistor is cooled by air flowing over it with an average heat transfer coefficient of 30 W/m2·K.If the air temperature is 55°C and the transistor case temperature is not to exceed 70°C, determine the amount of power this transistor can dissipate safely. Disregard any heat transfer from the transistor base.

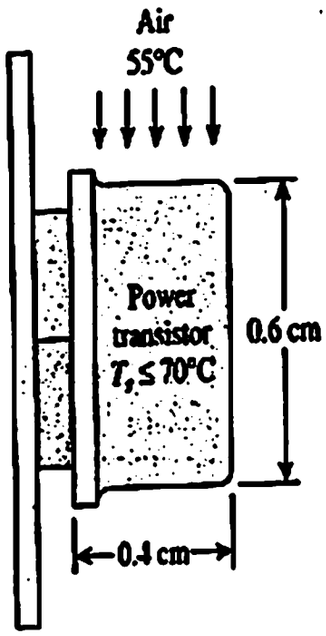


FIGURE P16–45

Step 1 of 3

Given

Height of transistor, $h = 0.4\,\text{cm}$

Diameter of transistor, $d = 0.6\,\text{cm}$

Coefficient of heat transfer, $h = 30\,\text{W/m}\cdot{}^\circ\text{C}$

Temperature of air, $T_1 = 55°\text{C}$

Temperature of transistor case, $T_2 = 70°\text{C}$

Neglect heat transfer from transistor base

Step 2 of 3

Total heat transfer area of the transistor is

$$\begin{aligned} A &= \pi dh + \pi d^2/4 \\ &= \pi \times 0.6 \times 0.4 + \pi \times 0.6^2/4 \\ &= 0.754 + 0.283 \\ &= 1.037\ \text{cm}^2 \\ &= 1.037 \times 10^{-4}\ \text{m}^2 \end{aligned}$$

Step 3 of 3

Amount of power dissipated by transistor is given by rate of heat transfer from the power transistor

$$\begin{aligned} \dot{Q} &= hA(T_2 - T_1) \\ &= 30 \times \left(1.037 \times 10^{-4}\right) \times (70 - 55) \\ &= 0.047\,\text{W} \end{aligned}$$

Amount of power dissipated by transistor, $\boxed{\dot{Q} = 0.047\,\text{W}}$

Question: 16.046P

Reconsider Prob. 16–45. Using EES (or other) software, plot the amount of power the transistor can dissipate safely as a function of the maximum case temperature in the range of 60°C to 90°C. Discuss the results.

Step 1 of 7

Given

Height of transistor, $h = 0.4\,\text{cm}$

Diameter of transistor, $d = 0.6\,\text{cm}$

Coefficient of heat transfer, $h = 30\,\text{W/m}\cdot{}^\circ\text{C}$

Temperature of air, $T_\infty = 55^\circ\text{C}$

Range of temperature of transistor case, $60^\circ\text{C} \le T_2 \le 90^\circ\text{C}$

Neglect heat transfer from transistor base

Step 2 of 7

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
L=0.004 "[m]"
D=0.006 "[m]"
h=30 "[W/m^2-C]"
T_infinity=55 "[C]"
"T_case_max=70 [C], parameter to be varied"

"ANALYSIS"
A=pi*D*L+pi*D^2/4
Q_dot=h*A*(T_case_max-T_infinity)
```

Step 3 of 7

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 13
Select variables in equations as T_{max} and Q press ADD and then press ok
A table will appear put values of T_{max} as 60,62.5,65.....90 upto 0.01 in run 1to 13
Then from Calculate -> Solve table

Step 4 of 7

Solution will appear

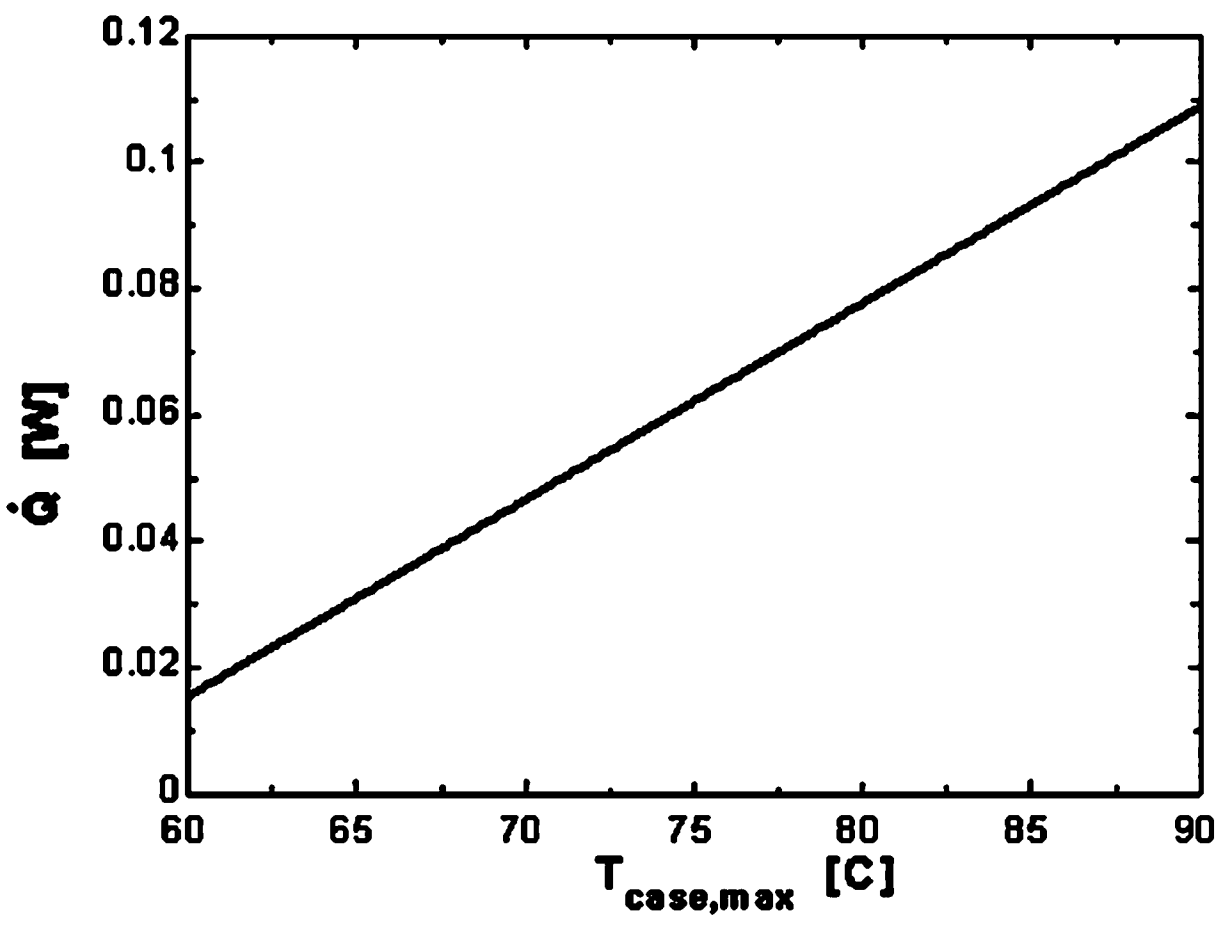
T_{max} (°C)	Q (W)
60	0.01555
62.5	0.02333
65	0.0311
67.5	0.03888
70	0.04665
72.5	0.05443
75	0.0622
77.5	0.06998
80	0.07775
82.5	0.08553
85	0.09331
87.5	0.1011
90	0.1089

Step 5 of 7

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_{max} and Y-axis Q
Press OK

Step 6 of 7

Graph will be plotted.



Step 7 of 7

From graph, amount of power increases with increasing maximum case temperature linearly.

Question: 16.047P

A 300-ft-long section of a steam pipe whose outer diameter is 4 in passes through an open space at 50°F. The average temperature of the outer surface of the pipe is measured be 280°F, and the average heat transfer coefficient on that surface is determined to be 6 Btu/h-ft2-°F. Determine (*a*) the rate of heat loss from the steam pipe and (*b*) the annual cost of this energy loss if steam is generated in a natural as furnace having an efficiency of 86 percent, and the priced natural gas is $ 1.10/therm (1 therm = 100,000 Btu).

Step 1 of 3

Use the conversion factor to convert the outer diameter of the pipe from in. to ft.

$$1\,\text{ft} = 12\,\text{in.}$$

$$\begin{aligned} d &= 4\,\text{in} \\ &= \frac{4\,\text{in.} \times 1\,\text{ft}}{12\,\text{in.}} \\ &= 0.33333\,\text{ft} \end{aligned}$$

Calculate the total area through which heat transfer.

$$A = \pi dL$$

Here, d is the outer diameter of the steam pipe, and L is the length of the steam pipe.

Substitute $0.333\,\text{ft}$ for d, and $300\,\text{ft}$ for L.

$$\begin{aligned} A &= \pi dL \\ &= \pi \times 0.33333 \times 300 \\ &= 314.156\,\text{ft}^2 \end{aligned}$$

Step 2 of 3

Calculate the Rate of heat loss from pipe.

$$\dot{Q}_{pipe} = hA\left(T_s - T_{air}\right)$$

Here, h is the heat transfer coefficient, T_s is surface temperature, and T_{air} is ambient air temperature.

Substitute $6\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for h, $50°\text{F}$ for T_{air}, $280°\text{F}$ for T_s, and $314.156\,\text{ft}^2$ for A.

$$\begin{aligned} \dot{Q}_{pipe} &= 6 \times 314.156 \times (280 - 50) \\ &= 433535.45\,\text{Btu/h} \end{aligned}$$

Therefore, the rate of heat loss from pipe is $\boxed{433535.45\,\text{Btu/h}}$.

Step 3 of 3

(b)

Calculate the amount of heat loss per year.

$$Q = \dot{Q}t$$

Substitute $433535.45\,\text{Btu/h}$ for $\dot{Q}$.

$$\begin{aligned} Q &= 433535.45\,\text{Btu/h} \times \left(\frac{365\,\text{days}}{1\,\text{year}} \times \frac{24\,\text{h}}{1\,\text{days}}\right) \\ &= 3.79 \times 10^9\ \text{Btu/yr} \end{aligned}$$

Calculate the Amount of gas consumed per year in the furnace with efficiency of 86%.

$$\begin{aligned} \text{Annual Energy loss} &= \frac{3.79 \times 10^9}{0.86 \times 100000} \\ &= 44160.122\ \text{therms/yr} \end{aligned}$$

Calculate the total annual cost for energy loss.

$$\text{Energy cost} = \left[(\text{Annual energy loss}) \times (\text{Unit cost of energy})\right]$$

Substitute $44160.122\ \text{therms/yr}$ for for annual energy loss and 1.10 for Unit cost of energy.

$$\begin{aligned} \text{Energy cost} &= 44160.122 \times 1.10 \\ &= 48576.134/\text{yr} \end{aligned}$$

Therefore, the total annual cost for energy loss is $\boxed{48576.134/\text{yr}}$.

Question: 16.048P

The boiling temperature of nitrogen at atmospheric pressure at sea level (1 atm) is - 196°C. Therefore, nitrogen is commonly used in low temperature scientific studies since the temperature of liquid nitrogen in a tank open to the atmosphere remains constant at –196°C until the liquid nitrogen in the tank is depleted. Any heat transfer to the tank results in the evaporation of some liquid nitrogen, which has a heat of vaporaization of 198 kJ/kg and a density of 810 kg/m3- at 1 atm.

Consider a 4-m-diameter spherical tank initially filled with liquid nitrogen at 1 atm and –196°C. The tank is exposed to 20°C ambient air with a heat transfer coefficient of 25 W/m2·K. The temperature of the thin-shelled spherical tank is observed to be almost the same as the temperature of the nitrogen inside. Disregarding any radiation heat exchange, determine the rate of evaporation of the liquid nitrogen in the tank as a result ofthe heat transfer from the ambient air.

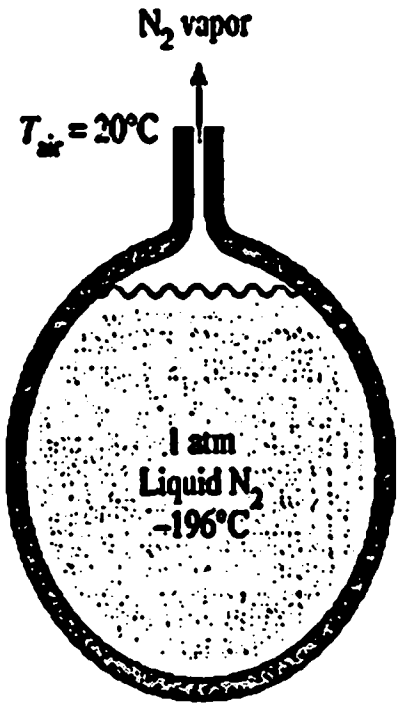


FIGURE P16–48

Step 1 of 2

Calculate the heat transfer area.

$$A = \pi d^2$$

Here, d is the diameter of the spherical tank.

Substitute $4\,\text{m}$ for d .

$$A = \pi \times 4^2$$
$$= 50.27\,\text{m}^2$$

Calculate the rate of heat transfer for nitrogen tank.

$$\dot{Q} = hA(T_{air} - T_s)$$

Here, h is the convective heat transfer coefficient, T_{air} is the ambient temperature, and T_s is the temperature of the liquid nitrogen.

Substitute $25\,\text{W/m}^2$ for h , $50.27\,\text{m}^2$ for A , $20°\text{C}$ for T_{air} , and $-196°\text{C}$ for T_s .

$$\dot{Q} = 25 \times 50.27\left[20 - (-196)\right]$$
$$= 271458\,\text{W}$$

Step 2 of 2

Calculate the rate of evaporation of liquid nitrogen.

$$\dot{Q} = \dot{m} h_{fg}$$

Here, h_{fg} is the heat of vaporization of liquid nitrogen.

Substitute $271458\,\text{W}$ for $\dot{Q}$ and $198 \times 10^3\,\text{J/kg}$ for h_{fg} .

$$271485 = \dot{m}\left(198 \times 10^3\right)$$
$$\dot{m} = 1.371\,\text{kg/s}$$

Therefore, the rate of evaporation of liquid nitrogen is $\boxed{1.371\,\text{kg/s}}$.

Question: 16.049P

Repeat Prob. 16–18 for liquid oxygen, which has a boiling temperature of −183°C. a heat of vaporization of 213 kJ/kg. and a density of 1140 kg/m3 at 1atm pressure.

Step 1 of 3

Given

Boiling temperature of nitrogen at 1atm , $T_s = -183°C$

Diameter of spherical tank, $d = 4\,m$

Temperature of ambient air, $T_{air} = 20°C$

Heat transfer coefficient, $h = 25\,W/m^2 \cdot °C$

Heat of vaporization for nitrogen, $h_{fg} = 213 kJ/kg$

Neglect any radiation heat exchange

Step 2 of 3

Heat transfer area,

$$A = \pi d^2$$
$$= \pi \times 4^2$$
$$= 50.27\,m^2$$

Rate of heat transfer for nitrogen tank is given by

$$\dot{Q} = hA(T_{air} - T_s)$$
$$= 25 \times 50.27\left[20 - (-183)\right]$$
$$= 255120\,W$$

Step 3 of 3

Rate of evaporation of liquid nitrogen is calculated as

$$\dot{Q} = \dot{m} h_{fg}$$
$$\Rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}}$$
$$= \frac{255120}{213 \times 10^3}$$
$$= 1.2 kg/s$$

Rate of evaporation of liquid nitrogen, $\boxed{\dot{m} = 1.2 kg/s}$

Question: 16.050P

Reconsider Prob. 16–48. Using EES (or other) software, plot the rate of evaporation of liquid nitrogen as a function of the ambient air temperature in the range of 0°C to 40°C. Discuss the results.

Step 1 of 4

Given data:

Boiling temperature of nitrogen at 1 atm is $T_s = -196°C$

Diameter of spherical tank, $D = 4$ m

Temperature of ambient air, $T_{air} = 20°C$

Heat transfer coefficient, $h = 25\ W/m^2 \cdot K$

Heat of vaporizations for nitrogen, $h_{fg} = 198$ kJ/kg

Step 2 of 4

To solve this question using EES software, open EES software
Then open a new file from 'FILE' menu by cliking on 'New'
A dailogbox will open type the following in dailog box

```
"GIVEN"
D=4 [m]
T_s=-196 [C]
"T_air=20 [C], parameter to be varied"
h=25 [W/m^2-C]

"PROPERTIES"
h_fg=198 [kJ/kg]

"ANALYSIS"
A=pi*D^2
Q_dot=h*A*(T_air-T_s)
m_dot_evap=(Q_dot*Convert(J/s, kJ/s))/h_fg
```

Step 3 of 4

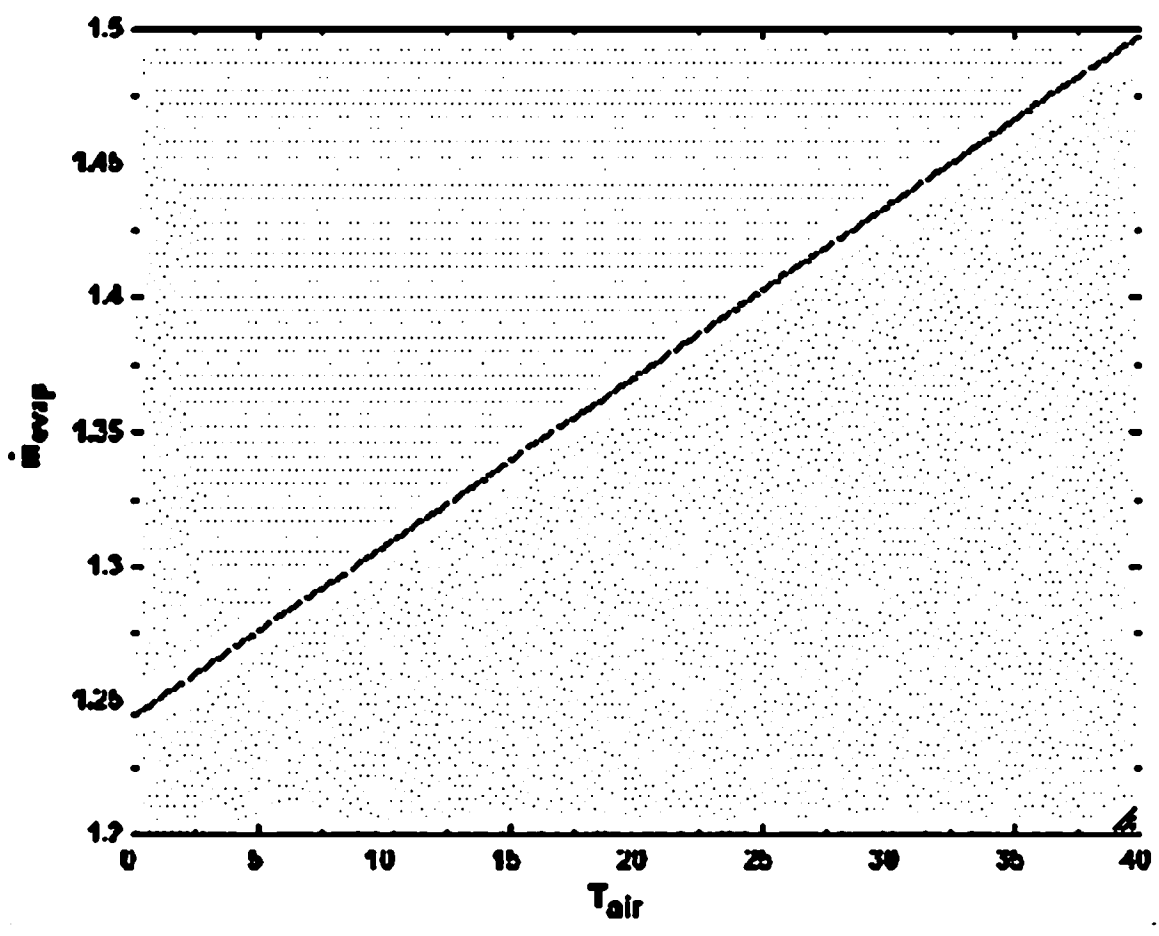
Then from 'Tables' menu, select 'New parametric table'. A dailog box will open
Select no of runs = 21
Select variables in equations as T_{air} and $\dot{m}_{evap}$ press 'ADD' and then press 'OK'
A table will appear, assign the values of T_{air} as 0,2,4.....40 in run 1 to 21 for that table
Then from 'Calculate' menu click 'Solve table'

Solution will be displayed as below:

1 21	1 $\dot{m}_{evap}$	2 T_{air}
Run 1	1.244	0
Run 2	1.257	2
Run 3	1.269	4
Run 4	1.282	6
Run 5	1.295	8
Run 6	1.307	10
Run 7	1.32	12
Run 8	1.333	14
Run 9	1.345	16
Run 10	1.358	18
Run 11	1.371	20
Run 12	1.384	22
Run 13	1.396	24
Run 14	1.409	26
Run 15	1.422	28
Run 16	1.434	30
Run 17	1.447	32
Run 18	1.46	34
Run 19	1.472	36
Run 20	1.485	38
Run 21	1.498	40

Step 4 of 4

Then from 'Plot' menu, click on 'New plot window' and then select 'X-Y Plot'.
Select X-axis as T_{air} and Y-axis as m_{evap}
Press 'OK' the Graph will be plotted as shown.



Question: 16.051P

Consider a person whose exposed surface area is 1.7 m2, emissivity is 0.5, and surface temperature is 32°C. Determine the rate of heat loss from that person by radiation in a large room having walls at a temperature of (*a*) 300 K and (*b*) 280 K.

Step 1 of 3

Given: -

Exposed surface area of person $A_s = 1.7\ m^2$

Emissivity $\varepsilon = 0.5$

Surface temperature $T_s = 32°C$

Step 2 of 3

(a) Net rate of heat transfer is given by

$T_{sur} = 300\ K$

$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{sur}^4)$

Where

Stefan Boltzmann constant $\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4}$

$\dot{Q}_{rad} = (0.5)(5.67 \times 10^{-8}\ (\frac{W}{m^2 \cdot K^4}))(1.7\ m^2)[(32+273)^4 - (300\ K)^4]K^4$

$\dot{Q}_{rad} = 26.7\ W$

$\boxed{Q_{rad} = 26.7\ W}$

Step 3 of 3

(b) Net rate of heat transfer is given by

$T_{sur} = 280\ K$

$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{sur}^4)$

Where

Stefan Boltzmann constant $\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4}$

$\dot{Q}_{rad} = (0.5)(5.67 \times 10^{-8}\ (\frac{W}{m^2 \cdot K^4}))(1.7\ m^2)[(32+273)^4 - (280\ K)^4]K^4$

$\dot{Q}_{rad} = 121\ W$

$\boxed{Q_{rad} = 121\ W}$

Question: 16.052P

A 0.3-cm-thick. 12-cm-high, and 18-cm-long circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. The board is impregnated with copper fillings and has an effective thermal conductivity of 16 W/m·K. All the heat generated in the chips is conducted across the circuit board and is dissipated from the back side of the board to the ambient air. Determine the temperature difference between the two sides of the circuit board.

Step 1 of 4

Given:

Thickness of circuit board $L = 0.3\text{cm}$

$= 0.003\text{ m}$

Height of circuit board $H = 12\text{ cm}$

$= 0.12\text{ m}$

Length of circuit board $W = 18\text{ cm}$

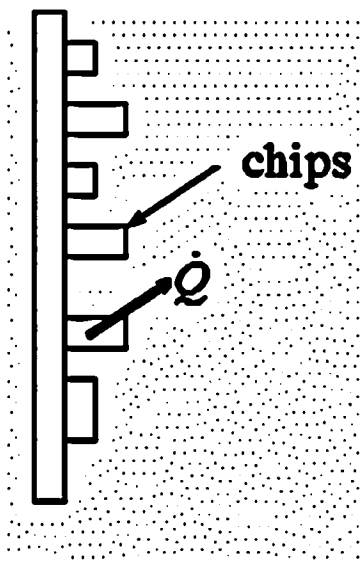
$= 0.18\text{ m}$

No of chips $n = = 80$

Power dissipated by each chip $Q = 0.06\text{ W}$

Effective thermal conductivity of board $k = 16\ \frac{\text{W}}{\text{m}\cdot^\circ\text{C}}$

Step 2 of 4



Step 3 of 4

Total rate of heat dissipated by the chips

$Q_{Total} = n \times Q$

$\dot{Q}_{Total} = 80 \times (0.06\text{ W})$

$= 4.8\text{ W}$

Step 4 of 4

Temperature difference between two sides of the board is

$A = (0.12\text{ m})(0.18\text{ m})$

$= 0.0216\text{ m}^2$

Heat transfer due to conduction is

$$\dot{Q} = kA\frac{\Delta T}{L}$$

$$\Delta T = \frac{\dot{Q}L}{kA}$$

$$= \frac{(4.8\text{ W})(0.003\text{ m})}{(16\text{ W/m}\cdot^\circ\text{C})(0.0216\text{ m}^2)}$$

$=0.042^\circ\text{C}$

$\boxed{\Delta T = 0.042^\circ\text{C}}$

Question: 16.053P

Consider a sealed 20-cm-high electronic box whose base dimensions are 50 cm × 50 cm placed in a vacuum chamber. The emissivity of the outer surface of the box is 0.95. If the electronic components in the box dissipate a total of 120 W of power and the outer surface temperature of the box is not to exceed 55°C, determine the temperature at which the surrounding surfaces must be kept if this box is to be cooled by radiation alone. Assume the heat transfer from the bottom surface of the box to the stand to be negligible.

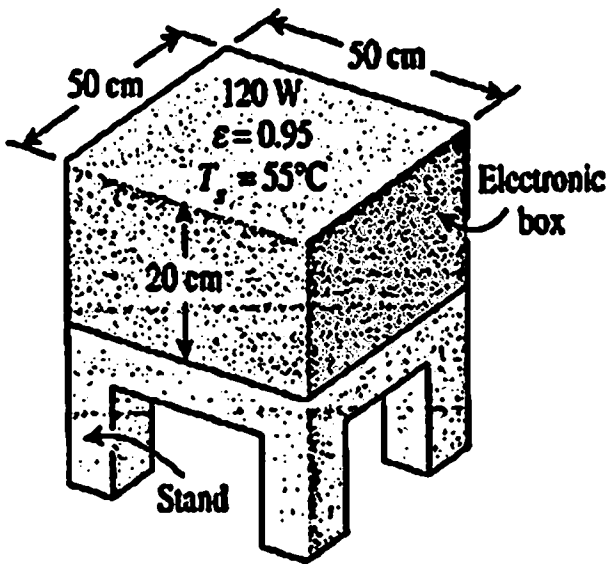


FIGURE P16–53

Step 1 of 3

Given data:

The emissivity, $\varepsilon = 0.95$.

Area of the box, $A = 50 \times 50\ \text{cm}^2$.

Height of the box, $l = 20\ \text{cm}$.

Total amount of heat dissipated by the electronic box, $Q_{rad} = 120\ \text{W}$.

Outer surface temperature of the electronic box, $T_s = 55°\text{C} = 328\ \text{K}$.

Step 2 of 3

The heat transferred by the radiation is given by,

$$Q_{rad} = \varepsilon \sigma A_s \left(T_s^4 - T_\infty^4\right)$$

The surface radiation of the electronic box from which heat is dissipates can be

A_s = Area of Top surface + Area of four side surfaces

Here, the bottom surface area is negligible.

$$A_s = (50 \times 50) + 4 \times (20 \times 50)$$
$$= 2500 + 4000$$
$$= 6500\ \text{cm}^2$$
$$= 0.65\ \text{m}^2$$

Step 3 of 3

Let T_∞ be the temperature of the surrounding at which the surfaces must be kept if box is to cooled by radiation alone.

$$Q_{rad} = \varepsilon \sigma A_s \left(T_s^4 - T_\infty^4\right)$$

$$120 = 0.95 \times \left(5.67 \times 10^{-8}\right) \times 0.65 \times \left(328^4 - T_\infty^4\right)$$

$$\left(328^4 - T_\infty^4\right) = \frac{120}{0.95 \times \left(5.67 \times 10^{-8}\right) \times 0.65}$$

$$T_\infty = \sqrt[4]{328^4 - \left(3.427 \times 10^9\right)}$$
$$= \sqrt[4]{8.146 \times 10^9}$$
$$= 300.43\ \text{K}$$
$$= 27.28\ °\text{C}$$

Hence, the surrounding air temperature should be less than $\boxed{T_\infty = 27.28°\text{C}}$.

Question: 16.054P

Using the conversion factors between W and Btu/h, m and ft, and K and R, express the Stefan-Boltz-mann constant σ= 5.67 × 10−8 W/m2·K4 in the English unit Btu/h-ft2·R4.

Step 1 of 3

Given:

Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8}\ \mathrm{W/m^2.K^4}$

Step 2 of 3

The conversion factors for W, m, and K are taken from conversion tables

$1\ \mathrm{W} = 3.41214\ \mathrm{Btu/h}$

$1\ \mathrm{m} = 3.2808\ \mathrm{ft}$

$1\ \mathrm{K} = 1.8\ \mathrm{R}$

Step 3 of 3

Substituting gives the Stefan-Boltzmann constant in the desired units,

$$\sigma = 5.67 \times 10^{-8}\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K^4}}$$

$$= (5.67)\ \frac{(3.41214\ \mathrm{Btu/h})}{(3.2808\ \mathrm{ft})^2 (1.8\ \mathrm{R})^4}$$

$$= 0.171\ \frac{\mathrm{Btu}}{\mathrm{h \cdot ft^2 \cdot R^4}}$$

$$\boxed{0.171\ \frac{\mathrm{Btu}}{\mathrm{h \cdot ft^2 \cdot R^4}}}$$

Question: 16.055P

An engineer who is working on the heat transfer analysis of a house in English units needs the convection heat transfer coefficient on the outer surface of the house, But the only value he can find from his handbooks is 22 W/m2·K. which is in SI units. The engineer does not have a direct conversion factor between the two unit systems for the convection heat transfer coefficient. Using the conversion factors between W and Btu/h, m and ft, and °C and °F, express the given convection heat transfer coefficient in Btu/h·ft2·°F.

Step 1 of 2

Given data:

Convective heat transfer coefficient on the outer surface of the house, $h = 22\ \text{W/m}^2\cdot\text{K}$

By using some known conversion factors, we have,

$1\text{m} = 3.28084\ \text{ft}$

$1\text{W} = 3.412\ \text{Btu/h}$

$1\ ^\circ\text{C} = \frac{9}{5} + 32^\circ\text{F}$

Step 2 of 2

Now converting the heat transfer coefficient into the English units:

$$h = 22\ \text{W/m}^2\cdot\text{K}$$

$$= 22 \times \frac{3.412}{(3.28084)^2 \times \left(\frac{9}{5} + 32 - 32\right)}$$

$$= 22 \times \frac{3.412}{19.375}$$

$$= 3.874\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Hence, the convective heat transfer coefficient in English units is $\boxed{h = 3.87\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$.

Question: 16.056P

Water at 0°C releases 333.7 kJ/kg of heat as it freezes to ice (ρ = 920 kg/m3) at 0°C. An aircraft flying under icing conditions maintains a heat transfer coefficient of 150 W/m2·K between the air and wing surfaces. What temperature must the wings be maintained at to prevent ice from forming on them during icing conditions at a rate of 1 mm/min or less?

Step 1 of 2

Given: -

Wings initial temperature $T_1 = 0°C$

Latent heat of water $L = 333.7 \text{kJ/kg}$

Density of ice $\rho = 920 \frac{\text{kg}}{\text{m}^3}$

Heat transfer coefficient $h = 150 \frac{\text{W}}{\text{m}^2 \cdot °\text{C}}$

Rate of ice formation on wings $V = 1 \frac{\text{mm}}{\text{min}}$

$$= \frac{0.001}{60} \frac{\text{m}}{\text{sec}}$$

Step 2 of 2

Assume steady state condition we can write

Heat transfer by wing = heat taken by ice

$hA(T_{wing} - T_{ice}) = \rho A V h_{if}$

Cancelling area on both sides we get

$$\left(150 \frac{\text{W}}{\text{m}^2 \cdot °\text{C}}\right)(T_{wing} - 0°\text{C}) = \left(920 \frac{\text{kg}}{\text{m}^3}\right)\left(\frac{0.001\text{m}}{60s}\right)\left(333{,}700 \frac{\text{J}}{\text{kg}}\right)$$

$T_{wing} = 34.11°\text{C}$

$\boxed{T_{wing} = 34.11°\text{C}}$

Question: 16.057P

A 2.1-m-long, 0.2-cm-diameter electrical wire extends across a room that is maintained at 20°C. Heat is generated in the wire as a result of resistance heating, and the surface temperature of the wire is measured to be 180°C in steady operation. Also, the voltage drop and electric current through the wire are measured to be 110 V and 3 A, respectively. Disregarding any heat transfer by radiation, determine the convection heat transfer coefficient for heat transfer between the outer surface of the wire and the air in the room.

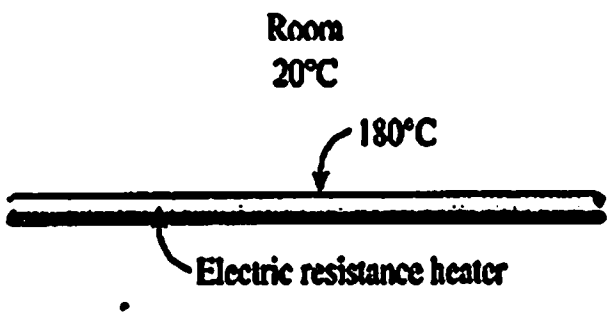


FIGURE P16–57

Step 1 of 4

Given data:

Diameter of the wire, $d = 0.2$ cm

Length of the wire, $l = 2.1$ m

Ambient temperature, $T_A = 20°C$

Surface temperature of the wire, $T_s = 180°C$

Voltage drop across the wire, $V = 110$ V

Current across the wire, $I = 3$ A

Step 2 of 4

Given figure:

Step 3 of 4

For an electrical wire, the surface area is determined using the formula,

$$A_s = \pi dl$$

$$= \pi \times \frac{0.2}{100} \times 2.1$$

$$= \pi \times 0.002 \times 2.1$$

Diameter of the wire, $d = 0.2$ cm

Length of the wire, $l = 2.1$ m

Ambient temperature, $T_A = 20°C$

Surface temperature of the wire, $T_s = 180°C$

Voltage drop across the wire, $V = 110$ V

Current across the wire, $I = 3$ A

Step 2 of 4

Given figure:

Step 3 of 4

For an electrical wire, the surface area is determined using the formula,

Question: 16.058P

Reconsider Prob. 16–57. Using EES (or other) software, plot the convection heat transfer coefficient as a function of the wire surface temperature in the range of 100°C to 300°C. Discuss the results.

Step 1 of 7

Given data:

Diameter of the wire, $d = 0.2\,\text{cm}$

Length of the wire, $l = 2.1\,\text{m}$

Room temperature, $T_A = 20°\text{C}$

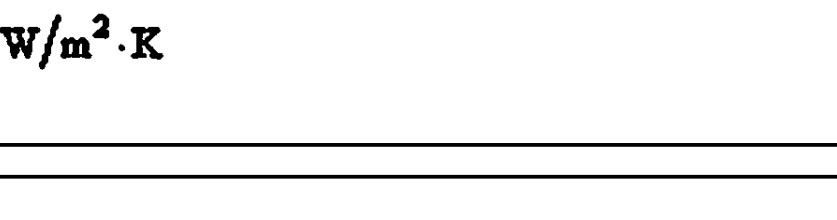
Surface temperature of the wire varies from, $T_s = 100\,°\text{C}$ to $300\,°\text{C}$

Voltage drop in the wire, $V = 110\,\text{V}$

Current across the wire, $I = 3\,\text{A}$

Step 2 of 7

Given figure:



Step 3 of 7

For an electrical wire, the surface area is given by,

$$A_s = \pi dl$$

$$= \pi \times \frac{0.2}{100} \times 2.1$$

$$= \pi \times 0.002 \times 2.1$$

$$= 0.0132\,\text{m}^2$$

Step 4 of 7

Let h be the heat transfer coefficient between the ambient air and the wire.

Rate of heat dissipation due to the current passing through the electrical wire is given by

$$Q = hA_s\Delta T$$

$$I \times V = h \times 0.0132 \times (T_S - 20)$$

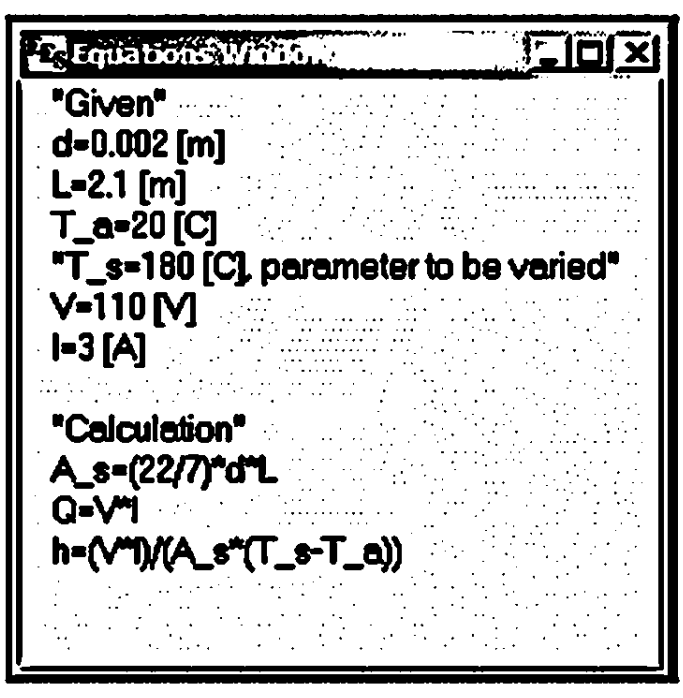
$$110 \times 3 = h \times 0.0132 \times (T_S - 20)$$

$$h = \frac{110 \times 3}{0.0132 \times (T_S - 20)}$$

$$= \frac{25000}{(T_S - 20)}\,\text{W/m}^2 \cdot \text{K}$$

Step 5 of 7

The following program represents the given parameters followed by the calculation of the heat transfer coefficient.

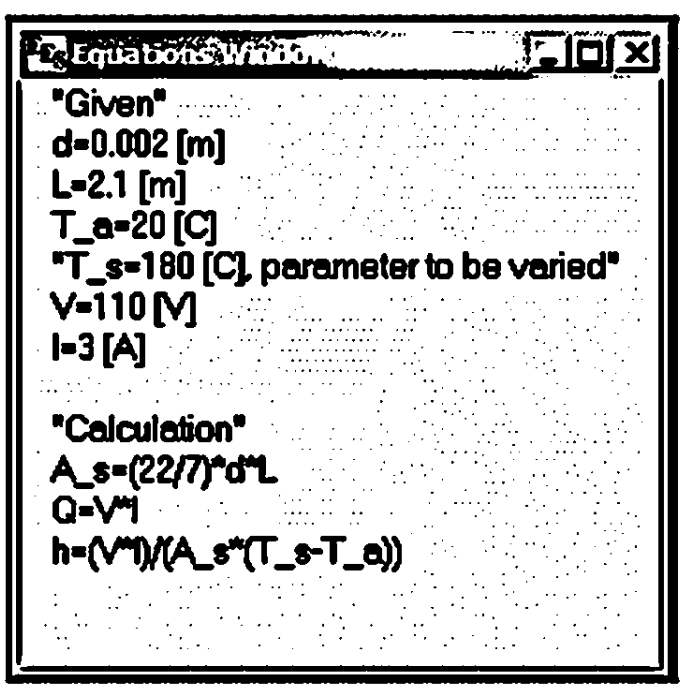


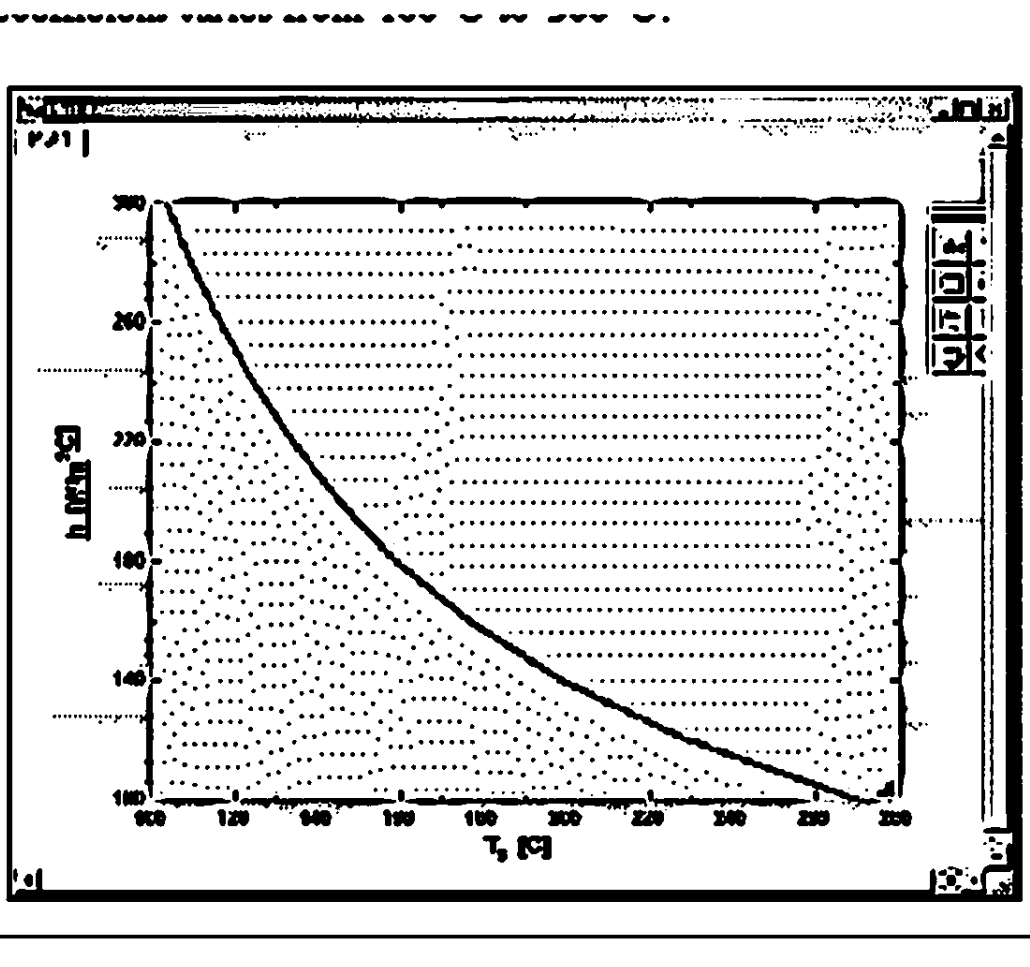
$$h = \frac{110 \times 3}{0.0132 \times (T_S - 20)}$$

$$= \frac{25000}{(T_S - 20)}\,\text{W/m}^2 \cdot \text{K}$$

Step 5 of 7

The following program represents the given parameters followed by the calculation of the heat transfer coefficient.





Question: 16.059P

Can all three modes of heat transfer occur simultaneously (in parallel) in a medium?

Step 1 of 1

All three modes of heat transfer cannot occur simultaneously in a medium.
Because of medium required only for conduction and convention. But not required for radiation. A medium may involve two of them simultaneously.

Question: 16.060P

Can a medium involve (*a*) conduction and convection, (*b*) conduction and radiation, or (*c*) convection and radiation simultaneously? Give examples for the "yes" answers.

Step 1 of 3

(a)
Heat transfer through fluid is either by conduction or convention, depending upon the presence of any bulk fluid motion. So that Conduction and convection not occurs simultaneously.

Step 2 of 3

(b)
The combined mode of Conduction and radiation occurs simultaneously.
Example: A roof heating of a house in winter. Where the heat from hot air the outside roof is transferred to the colder ambient environmental inside the room. Another example hot cup tea to the surrounding air thought the side walls of the cup contains in the tea.

Step 3 of 3

(c)
The combined mode of Convection and radiation occurs simultaneously.
The combined mode of Conduction and radiation occurs simultaneously where the solid surface either the loses the heat or gain the heat from the surrounding medium. This modes are occurs simultaneously in order to determine the total heat transfer.
Example: Heat transferred from the hot furnace wall to the surrounding air
And the outer surface of the cold piece of rock will warm up in a warmer environmental as a result by heat gain by convention (from the air) and radiation from the sun or warmer surrounding surface.

Step 3 of 3

(c)
The combined mode of Convection and radiation occurs simultaneously.
The combined mode of Conduction and radiation occurs simultaneously where the solid surface either the loses the heat or gain the heat from the surrounding medium. This modes are occurs simultaneously in order to determine the total heat transfer.
Example: Heat transferred from the hot furnace wall to the surrounding air
And the outer surface of the cold piece of rock will warm up in a warmer environmental as a result by heat gain by convention (from the air) and radiation from the sun or warmer surrounding surface.

Question: 16.061P

The deep human body temperature of a healthy person remains constant at 37°C while the temperature and the humidity of the environment change with time. Discuss the heat transfer mechanisms between the human body and the environment both in summer and winter, and explain how a person can keep cooler in summer and warmer in winter.

Step 1 of 2

The human body loses heat by convection, radiation, and evaporation in both summer and winter.

In summer season our body temperature is more than that of the surrounding .so in order to maintain the body temperature there is formation of boundary layer on the body due to wet on the surface or evaporation process. In summer, we can keep cool by dressing lightly, staying in cooler environments, turning a fan on, avoiding humid places and direct exposure to the sun.

Step 2 of 2

In winter season our body contains fat which is the high calorific value which generates more heat in body to maintain the temperature constant in winter season by means of melting of the fat. In winter, we can keep warm by dressing heavily, staying in a warmer environment, and avoiding drafts.

Question: 16.062P

We often turn the fan on in summer to help us cool. Explain how a fan makes us feel cooler in the summer. Also explain why some people use ceiling fans also in winter.

Step 1 of 2

The fan increases the air motion around the body and thus the convection heat transfer coefficient, which increases the rate of heat transfer from the body by convection and evaporation

When using the ceiling fans in a summer day, make certain to let the blades rotate ins the counter clockwise in order that to blow the air downward towards to you as well as in your space. The gust of wind from the fan evaporates perspiration on human skin faster which in effect leaves our body felling cooler.

Step 2 of 2

In rooms with high ceilings, ceiling fans are used in winter to force the warm air at the top downward to increase the air temperature at the body level. This is usually done by forcing the air up which hits the ceiling and moves downward in a gently manner to avoid drafts.

Question: 16.063P

Consider a person standing in a room at 18°C. Determine the total rate of heat transfer from this person if the exposed surface area and the skin temperature of the person are 1.7 m2 and 32°C. respectively, and the convection heat transfer coefficient is 5 W/m2·K. Take the emissivity of the skin and the clothes to be 0.9, and assume the temperature of the inner surfaces of the room to be the same as the air temperature.

Step 1 of 3

Given data:

Temperature of the ambience, $T_A = 18°C = 291\ K$

Surface area of the person, $A_s = 1.7\ m^2$

Surface temperature of the person, $T_s = 32°C = 305\ K$

Convective heat transfer coefficient, $h = 5\ W/m^2 \cdot K$

Emissivity between man and air, $\varepsilon = 0.9$.

Step 2 of 3

The total heat transfer from the person is given by,

$$Q = Q_{rad} + Q_{conv}$$
$$= \varepsilon\sigma A_s \left(T_s^4 - T_\infty^4\right) + hA_s\Delta T$$

where, $\sigma = 5.67\times10^{-8}\ W/m^2 \cdot K^4$

Heat transfer from the person to the surroundings due to radiation is given by

$$Q_{rad} = \varepsilon\sigma A_s \left(T_s^4 - T_\infty^4\right)$$
$$= \varepsilon\sigma A_s \left(T_s^4 - T_A^4\right)$$
$$= 0.9\times5.67\times10^{-8}\times1.7\times\left((32+273)^4 - (18+273)^4\right)$$
$$= 0.9\times\left(5.67\times10^{-8}\right)\times1.7\times\left(305^4 - 291^4\right)$$
$$= 128.63\ W$$

Step 3 of 3

Heat transfer from the person to the surroundings due to convection:

$$Q_{conv} = hA_s\Delta T$$
$$= hA_s\left(T_s - T_A\right)$$
$$= 5\times1.7\times(32-18)$$
$$= 119\ W$$

Now, the total heat transfer from the person due to convection and radiation together is given by,

$$Q = Q_{rad} + Q_{conv}$$
$$= 128.63 + 119$$
$$= 247.63\ W$$

Hence, the rate of total heat transfer from the person to the surroundings is $\boxed{Q = 248\ W}$.

Question: 16.064P

Consider steady heat transfer between two large parallel plates at constant temperatures of $T1$ = 290 K and $T2$=150 K that are L=2 cm apart. Assuming the surfaces to be black (emissivity ε = 1). determine the rate of heal transfer between the plates per unit surface area assuming the gap between the plates is (*a*) filled with atmospheric air, (*b*) evacuated, (*c*) filled with fiberglass insulation, and (*d*) filled with superinsulation having an apparent thermal conductivity of 0.00015 W/m·K.

Step 1 of 10

Given: -

Temperature of plate 1 $(T_1) = 290\ K$

Temperature of plate 2 $(T_2) = 150\ K$

Distance between two plates $(L) = 2\ cm$

$= 0.02\ m$

Emissivity $\varepsilon = 1$

Thermal conductivities for super insulation $(k) = 0.00015\ \frac{W}{m.^\circ C}$

Step 2 of 10

Mean film temperature

$$T_{mean} = \frac{T_1 + T_2}{2}$$

$$T_{mean} = \frac{290 + 150}{2}$$

$$T_{mean} = 220\,K$$

$$T_{mean} = -53^\circ C$$

Step 3 of 10

From the Properties of air

At -53°C

By using the interpolation we get

$$k_{air} = 0.01582 + \frac{(0.01979 - 0.01582)(-53 - (-50))}{-50 - (-100)}$$

$$k_{air} = 0.0195519\ \frac{W}{m.^\circ C}$$

From the Properties of glass

For fiberglass insulation at -53°C

$$k = 0.036\ \frac{W}{m.^\circ C}$$

Step 4 of 10

(a) Rates of heat transfer by conduction

$$\dot{Q}_{cond} = kA\frac{T_1 - T_2}{L}$$

$$= (0.0195519\ \frac{W}{m^2.^\circ C})(1\ m^2)\frac{(290 - 150)\ K}{0.02\ m}$$

$$= 136.863\ W$$

Step 5 of 10

Rates of heat transfer by radiation

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_1^4 - T_2^4)$$

$$= 1(5.67 \times 10^{-8}\ W/m^2.K^4)(1m^2)\left[(290\ K)^4 - (150\ K)^4\right]$$

$$= 372\ W$$

Step 6 of 10

Total heat transfer is

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{rad}$$

$$= 136.833 + 372$$

$$= 508.86\ W$$

$\boxed{Q_{Total} = 508.86\ W}$

Step 7 of 10

(b) In case of vacuum there will no heat transfer through conduction and all heat Transfer will be through radiation so

Rates of heat transfer by radiation

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_1^4 - T_2^4)$$

$$= 1(5.67 \times 10^{-8}\ W/m^2.K^4)(1m^2)\left[(290\ K)^4 - (150\ K)^4\right]$$

$$= 372\ W$$

$$\dot{Q}_{total} = \dot{Q}_{rad} = 372\ W$$

$\boxed{\dot{Q}_{total} = 372\ W}$

Step 8 of 10

C) In case of fiberglass entire heat transfer will be through conduction

Rates of heat transfer by conduction

$$\dot{Q}_{total} = \dot{Q}_{cond}$$

$$= kA\frac{(T_1 - T)_2}{L}$$

$$= (0.036\ \frac{W}{m.^\circ C})(1\ m^2)\frac{(290 - 150)\ K}{0.02\ m}$$

$$= 252\ W$$

$\boxed{Q_{Total} = 252\ W}$

Step 9 of 10

e) The rate of heat transfer in case of super-insulation will be

Rates of heat transfer by conduction

$$\dot{Q}_{total} = \dot{Q}_{cond}$$

$$= kA\frac{(T_1 - T_2)}{L}$$

Step 10 of 10

$$= (0.00015\ W/m.^\circ C)(1\ m^2)\frac{(290 - 150)\ K}{0.02\ m}$$

$$= 1.05\ W$$

$\boxed{Q_{Total} = 1.05\ W}$

Question: 16.065P

The inner and outer surfaces of a 25-cm-thiek wall in summer are at 27?C and 44?C, respectively. The outer surface of the wall exchanges heat by radiation with surrounding surfaces at 40?C, and convection with ambient air also at 40?C with a convection heat transfer coefficient of 8 W/m2?K. Solar radiation is incident on the surface at a rate of 150 W/m2. If both the emissivity and the solar absorptivity of the outer surface are 0.8, determine the effective thermal conductivity of the wall.

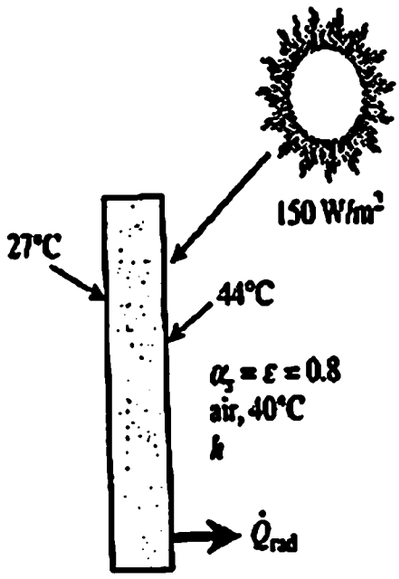


FIGURE P16–65

Step 1 of 1

Apply steady state condition on the surface to calculate the required effective thermal conductivity.

$$\dot{q}_{cond} = \dot{q}_{conv} + \dot{q}_{rad} + \dot{q}_{solar}$$

Here, $\dot{q}_{conv}$ is the convection heat transfer, $\dot{q}_{cond}$ is the conduction heat transfer, $\dot{q}_{solar}$ is the solar heat transfer and $\dot{q}_{rad}$ is the radiation heat transfer.

Substitute $k\left(\frac{T_2 - T_1}{L}\right)$ for $\dot{q}_{cond}$, $h(T_0 - T_2)$ for $\dot{q}_{conv}$, $\varepsilon\sigma\left(T_{surr}^4 - T_2^4\right)$ for $\dot{q}_{rad}$, and $\alpha_s I$ for $\dot{q}_{solar}$.

$$k\left(\frac{T_2 - T_1}{L}\right) = h(T_0 - T_2) + \varepsilon\sigma\left(T_{surr}^4 - T_2^4\right) + \alpha_s I$$

Here, T_1 is the inner temperature of the wall, T_2 is outer temperature of the wall, L is the thickness of the wall, k is the thermal conductivity, h is the heat transfer co efficient, T_0 is the ambient air temperature, ε is the emissivity of the surface, σ is the Stefan Boltzmann constant, T_{surr} is surrounding temperature, α_s is the solar absorptivity of the outer surface, and I is the solar radiation incident on surface.

Substitute 0.25 m for L, $27°C$ for T_1, $44°C$ for T_2, $40°C$ for T_{surr}, $8\,W/m^2 \cdot °C$ for h, $40°C$ for T_0, $150\,W/m^2$ for I, $\left(5.67\times10^{-8}\right)$ for σ, 0.8 for ε, and 0.8 for α_s.

$$k\times\left(\frac{44-27}{0.25}\right) = 8\times(40-44) + (0.8)\left(5.67\times10^{-8}\right)\left[313^4 - 317^4\right] + (0.8)(150)$$

$$k = 0.960\,W/m \cdot °C$$

Therefore, the required effective thermal conductivity of the wall is $\boxed{0.960\,W/m \cdot °C}$.

Question: 16.066P

A 2-in-diameter spherical ball whose surface is maintained at a temperature of 170°F is suspended in the middle of a room at 70°F. If the convection heat transfer coefficient is 15 Btu/h·ft2·°F and the emissivity of the surface is 0.8, determine the total rate of heat transfer from the ball.

Step 1 of 5

Given:

Diameter of the spherical ball $D = 2\text{in}$

A spherical ball surface temperature $T_s = 170°\text{F}$

Room temperature $T_0 = 70°\text{F}$

The Emissivity of the ball surface is given to be $\varepsilon = 0.8$.

Convection heat coefficient $h = 15\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$

Step 2 of 5

The heat transfer surface area is

$$A_s = \pi D^2$$
$$= \pi\left(\frac{2}{12}\text{ ft}\right)^2$$
$$= 0.08727\text{ ft}^2$$

Step 3 of 5

Under steady conditions, the rates of convection and radiation heat transfer are

$$\dot{Q}_{conv} = hA_s\Delta T$$
$$= \left(15\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(0.08727\text{ ft}^2)(170-70)^\circ\text{F}$$
$$= 130.905\frac{\text{Btu}}{\text{h}}$$

Step 4 of 5

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_\infty^4)$$
$$= 0.8(0.08727\text{ ft}^2)\left(0.1714\times10^{-8}\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot\text{R}^4}\right)[(170+460\text{ R})^4 - (70+460\text{ R})^4]$$
$$= 9.4\ \frac{\text{Btu}}{\text{h}}$$

Step 5 of 5

Therefore, Total Heat transfer is

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= 130.9 + 9.4$$
$$= 140.3\frac{\text{Btu}}{\text{h}}$$

$$\boxed{Q_{Total} = 140.3\frac{\text{Btu}}{\text{h}}}$$

Question: 16.067P

An 800-W iron is left on the iron board with its base exposed to the air at 20°C. The convection heat transfer coefficient between the base surface and the surrounding air is 35 W/m2·K. If the base has an emissivity of 0.6 and a surface area of 0.02 m2, determine the temperature of the base of the iron.

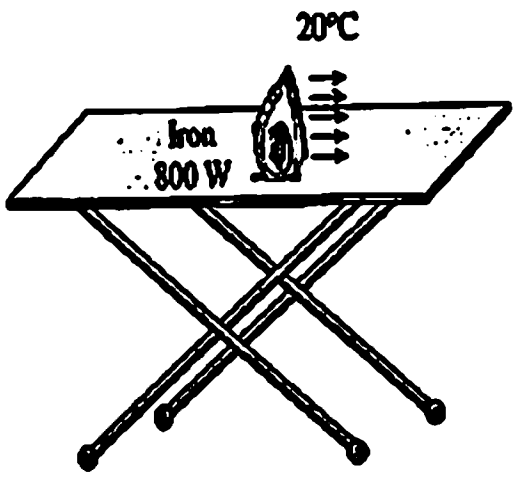


FIGURE P16–67

Step 1 of 3

Given data:

Capacity of the an iron box, $Q = 800\ \text{W}$

The ambient temperature, $T_\infty = 20°\text{C} = 293\ \text{K}$

Convective heat transfer coefficient, $h = 35\ \text{W/m}^2\cdot\text{K}$

The emissivity of the base, $\varepsilon = 0.6$

The surface area of the base, $A_s = 0.02\ \text{m}^2$

Step 2 of 3

Thermal properties of man are assumed to be constant and the heat transfer takes place due to both convective and radiation.

$$Q = Q_{rad} + Q_{conv}$$
$$= \varepsilon\sigma A_s\left(T^4 - T_\infty^4\right) + hA_s\Delta T$$

Heat transfer from the person to the surroundings due to radiation is given by

$$Q_{rad} = \varepsilon\sigma A_s\left(T^4 - T_\infty^4\right)$$
$$= 0.6\times\left(5.67\times10^{-8}\right)\times0.02\times\left(T^4 - 293^4\right)$$

Step 3 of 3

Heat transfer from the iron to the surroundings due to convection:

$$Q_{conv} = hA_s\Delta T$$
$$= 35\times0.02\times(T - 293)$$

Now the total heat transfer can be obtained by,

$$Q = Q_{rad} + Q_{conv}$$
$$800 = \left(0.6\times\left(5.67\times10^{-8}\right)\times0.02\times\left(T^4 - 293^4\right)\right) + \left(35\times0.02\times(T - 293)\right)$$
$$800 = \left(0.06804\times10^{-8}T^4 - 5.0145\right) + (0.7T - 205.1)$$
$$0.06804\times10^{-8}T^4 + 0.7T - 210.11 = 800$$
$$0.06804\times10^{-8}T^4 + 0.7T = 1{,}010.11\ \text{W}$$

By trial and error method, we have,

$T = 601°\text{C}$

Hence, the base temperature of iron is $\boxed{T = 601°\text{C}}$.

Question: 16.068P

A 3-m-internal-diameter spherical tank made of 1-cm-thick stainless steel is used to store iced water at 0°qC.The lank is located outdoors at 25°C. Assuming the entire steel lank to be at 0°C and thus the thermal resistance of the tank to be negligible, determine (*a*) the rate of heat transfer to the iced water in the lank and (*b*) the amount of ice at 0°C that melts during a 24-hour period. The heat of fusion of water at atmospheric pressure is *hif* =333.7 kJ/kg. Theemissivity of the outer surface of the tank is 0.75. and the convection heat transfer coefficient on the outer surface can be taken to be 30 W/m2·K. Assume the average surrounding surface temperature for radiation exchange to be 15°C.

Step 1 of 4

Given

Internal diameter of spherical tank, $d = 3\text{m}$

Thickness, $t = 1\text{cm}$

Average surrounding surface temperature for radiation exchange, $T_{sur} = 15°\text{C}$

Temperature of entire steel tank, $T_s = 0°\text{C}$

Outdoor temperature, $T_\infty = 25°\text{C}$

The heat of fusion of water at atmospheric pressure, $h_{if} = 333.7\,\text{kJ/kg}$.

The emissivity of the outer surface of the tank, $\varepsilon = 0.75$

Heat transfer coefficient, $h = 30\,\text{W/m}^2\cdot°\text{C}$

Step 2 of 4

(a) The outer surface area of the spherical tank,

$$A = \pi d_2^2$$
$$= \pi \times 3.02^2$$
$$= 28.65\,\text{m}^2$$

The rate of heat transfer to the tank by convection is given by

$$\dot{Q}_{conv} = hA(T_\infty - T_s)$$
$$= 30 \times 28.65 \times (25-0)$$
$$= 21487.5\,\text{W}$$
$$= 21.5\text{kW}$$

Step 3 of 4

The rate of heat transfer to the tank by radiation is given by

$$\dot{Q}_{rad} = \varepsilon A\sigma\left(T_{sur}^4 - T_s^4\right)$$
$$= 0.75 \times 28.65 \times \left(5.67\times10^{-8}\right) \times \left(288^4 - 273^4\right)$$
$$= 1614.5\,\text{W}$$
$$= 1.61\text{kW}$$

The total rate of heat transfer,

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= 21.5 + 1.61$$
$$= 23.11\text{kW}$$

Total rate of heat transfer, $\boxed{\dot{Q}_{total} = 23.11\text{kW}}$

Step 4 of 4

(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}t$$
$$= 23.1 \times (24 \times 3600)$$
$$= 1996704\,\text{kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if}$$
$$\Rightarrow m = \frac{Q}{h_{if}}$$
$$= \frac{1996704}{333.7}$$
$$= 5983.5\text{kg}$$

The amount of ice that melts during 24 h period, $\boxed{m = 5983.5\text{kg}}$

Question: 16.069P



The roof of a house consists of a 22-cm-thick concrete slab (k = 2 W/m·K) that is 15 m wide and 20 m long. The emissivity of the outer surface of the 1roof is 0.9, and the convection heat transfer coefficient on that surface is estimated to be 15 W/m2·K. The inner surface of the roof is maintained at 15°C. On a clear winter night, the ambient air is reported to be at 10°C while the night sky temperature for radiation heat transfer is 255 K. Considering both radiation and convection heat transfer, determine the outer surface temperature and the rate of heat transfer through the roof.

If the house is heated by a furnace burning natural gas with an efficiency of 85 percent, and the unit cost of natural gas is $1.20/therm (1 therm = 105.500 kJ of energy content), determine the money lost through the roof that night during a 14-hour period.

Step 1 of 3

Consider the outer surface temperature of the roof be T_{out} .

Since, the total heat conducted is equal to the sum of the heat transferred through convection and the rest by radiation.

$$Q_{cond} = Q_{rad} + Q_{conv}$$

Re-write the equation as follows:

$$kA_s\frac{\Delta T}{L} = \varepsilon\sigma A_s\left(T_s^4 - T_\infty^4\right) + hA_s\Delta T$$

$$k\frac{\Delta T}{L} = \varepsilon\sigma\left(T_s^4 - T_\infty^4\right) + h\Delta T$$

$$k\frac{T_{in} - T_{out}}{L} = \varepsilon\sigma\left(T_{out}^4 - T_{rad}^4\right) + h\left(T_{out} - T_\infty\right)$$

Here, k is the thermal conductivity of the roof, L is the thickness, T_{in} the inner temperature, T_{out} is the outer temperature, ε is the emissivity, σ is the Stephen-Boltzmann Constant, T_{rad} is the sky temperature for radiation heat transfer, h is the heat transfer coefficient, and T_∞ is the ambient temperature.

Substitute $2\ \text{W/m}\cdot\text{K}$ for k, $15°\text{C}$ for T_{in} , 0.22 m for L, 0.9 for ε , $5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ for σ , 255 K for T_{rad} , $15\ \text{W/m}^2\cdot\text{K}$ for h, and $10°\text{C}$ for T_∞ .

$$2\times\frac{15 - T_{out}}{0.22} = \left[0.9\times\left(5.67\times10^{-8}\right)\times\left(\left(T_{out}+273\right)^4 - 255^4\right)\right] + \left[15\times\left(T_{out} - 10\right)\right]$$

$$T_{out} = 7.7°\text{C}$$

Hence, the temperature of outer surface of the roof is $\boxed{T_{out} = 7.7°\text{C}}$.

Step 2 of 3

Calculate the surface area of the roof.

$$A_s = b\times l$$

Here, b is the width, and l is the length.

Substitute 15 m for b and 20 m for l.

$$A_s = 15\times20$$
$$= 300\ \text{m}^2$$

Write the equation for conduction.

$$Q_{cond} = kA_s\frac{\Delta T}{L}$$

Substitute $2\ \text{W/m}\cdot\text{K}$ for k, $300\ \text{m}^2$ for A_s , $(15-7.7)°\text{C}$ for ΔT , and 0.22 m for L.

$$Q_{cond} = 2\times300\times\frac{15 - 7.7}{0.22}$$
$$= 19{,}909\ \text{W}$$

Therefore, the total heat transferred through conduction is $\boxed{Q_{cond} = 19{,}909\ \text{W}}$.

Step 3 of 3

Consider the amount of natural gas required be R and the cost incurred in running the furnace through the night be M.

$$R = \frac{Q_{cond}}{0.85}\times T$$

Duration of time, $T = 14\times3600\ \text{s}$

$$R = \frac{19.909\times\left(14\times3600\right)}{0.85}\ \text{kJ}$$
$$= \frac{19.909\times\left(14\times3600\right)}{0.85\times105{,}500}\ \text{therm}$$
$$= 11.2\ \text{therm}$$

And the required money for this gas,

$$M = 11.2\times\$1.2$$
$$= \$13.44$$

Therefore, the money lost through the roof due to the heat transfer, $\boxed{M = \$13.44}$.

Question: 16.070P

Consider a flat-plate solar collector placed horizontally on the flat roof of a house. The collector is 5 ft wide and 15 ft long, and the average temperature of the exposed surface of the collector is 100°F. The emissivity of the exposed surface of the collector is 0.9. Determine the rate of heat loss from the collector by convection and radiation during a calm day when the ambient air temperature is 70°F and the effective sky temperature for radiation exchange is 50°F. Take the convection heat transfer coefficient on the exposed surface to be 2.5 Btu/h·ft2·°F.

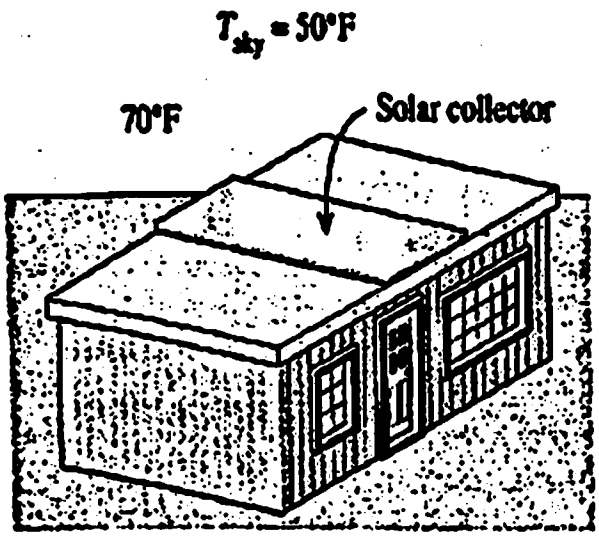


FIGURE P16–70E

Step 1 of 2

Given

Width of the collector, $w = 5\text{ft}$

Length of the collector, $L = 15\text{ft}$

Temperature of exposed surface of collector, $T_w = 100°\text{F}$

The emissivity of the outer surface of the collector, $\varepsilon = 0.9$

Ambient temperature of air, $T_\infty = 70°\text{F}$

Sky temperature for radiation, $T_{sky} = 50°\text{F}$

Convection heat transfer coefficient, $h = 2.5\,\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$

Step 2 of 2

Heat transfer area, $A = 5\times15$

$= 75\text{ ft}^2$

The total rate of heat loss from the collector to the environment by convection and radiation.

Rate of heat loss by convection is given by

$$\dot{Q}_{conv} = hA(T_w - T_\infty)$$

$$= 2.5\times75\times(100-70)$$

$$= 5625\text{ Btu/h}$$

Rate of heat loss by radiation is given by

$$\dot{Q}_{rad} = \varepsilon A\sigma\left(T_w^4 - T_{sky}^4\right)$$

$$= 0.9\times75\times\left(0.1714\times10^{-8}\right)\times\left[(100+460)^4 - (50+460)^4\right]$$

$$= 3551\text{ Btu/h}$$

The total rate of heat loss is given by

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

$$= 5625+3551$$

$$= 9176\,\text{Btu/h}$$

The total rate of heat loss from the collector to the environment, $\boxed{\dot{Q}_{total} = 9176\,\text{Btu/h}}$

Question: 16.071P

A soldering iron has a cylindrical tip of 2.5 mm in diameter and 20 mm in length. With age and usage, the tip has oxidized and has an emissivity of 0.80. Assuming that the average convection heat transfer coefficient over the soldering iron tip is 25 W/m2·K, and the surrounding air temperature is 20°C, determine the power required to maintain the tip at 400°C.

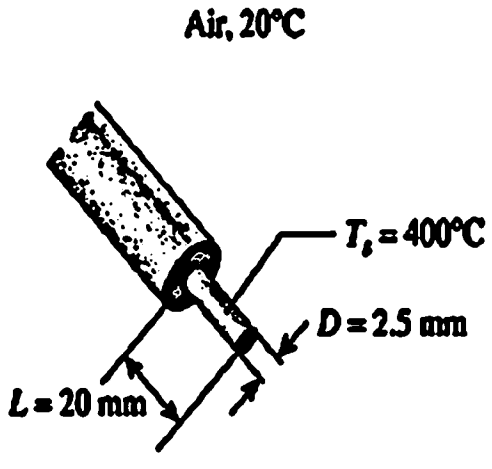


FIGURE P16–71

Step 1 of 4

Given data:

Diameter of the cylindrical tip, $D = 2.5\text{mm}$

Length of the tip, $L = 20\text{mm}$

Emissivity between tip and air, $\varepsilon = 0.8$.

Convective heat transfer coefficient between tip and air, $h = 25\ \text{W/m}^2\cdot\text{K}$

Ambient temperature, $T_\infty = 20\,°\text{C}$

Temperature of the tip, $T = 400\,°\text{C}$

Step 2 of 4

The total heat transfer is due to convection followed by radiation.

$$Q = Q_{rad} + Q_{conv}$$
$$= \varepsilon\sigma A_s\left(T^4 - T_\infty^4\right) + hA_s\Delta T$$

The surface area of the cylindrical tip is given by

$$A_s = \pi DL$$
$$= \pi\times\left(\frac{2.5}{1,000}\right)\times\left(\frac{20}{1,000}\right)$$
$$= 1.57\times10^{-4}\ \text{m}^2$$

Step 3 of 4

Heat transfer due to radiation:

$$Q_{rad} = \varepsilon\sigma A_s\left(T^4 - T_\infty^4\right)$$
$$= 0.8\times\left(5.67\times10^{-8}\right)\times1.57\times10^{-4}\times\left[(400+273)^4 - (20+273)^4\right]$$
$$= 1.41\ \text{W}$$

Heat transfer due to convection:

$$Q_{conv} = hA_s\Delta T$$
$$= 25\times\left(1.57\times10^{-4}\right)\times(400-20)$$
$$= 1.49\ \text{W}$$

Step 4 of 4

Total heat transfer between tip and the air,

$$Q = Q_{rad} + Q_{conv}$$
$$= 1.41+1.49$$
$$= 2.9\ \text{W}$$

Therefore, the power required to maintain a tip temperature of 400 °C is $\boxed{Q = 2.9\ \text{W}}$

Question: 16.072P

It is well known that wind makes the cold air feel much colder as a result of the *wind chill effect*that is due to the increase in the convection heat transfer coefficient with increasing air velocity. The wind chill effect is usually expressed in terms of the *wind chill temperature*(WCT), which is the apparent temperature felt by exposed skin. For outdoor air temperature of 0°C, for example, the wind chill temperature is −5°C at 20 km/h winds and −9°C at 60 km/h winds. That is. a person exposed to 0°C windy air at 20 km/h will feel as cold as a person exposed to −5°C calm air (air motion under 5 km/h).

For heat transfer purposes, a standing man can be modeled as a 30-cm-diameter, 170-cm-long vertical cylinder with both the top and bottom surfaces insulated and with the side surface at an average temperature of 34°C. For a convection heat transfer coefficient of 15 W/m2·K, determine the rate of heat loss from this man by convection in still air at 20°C. What would your answer be if the convection heat transfer coefficient is increased to 30 W/m2·K as a result of winds? What is the wind chill temperature in this case?

Step 1 of 3

Given data:

Diameter of the standing man, $D = 30$ cm

Length of the standing man, $L = 170$ cm

Average temperature of side surface, $T_s = 34°C$

Convective heat transfer coefficient between man and air is $h = 15\ \text{W/m}^2\cdot\text{K}$

Still air temperature, $T_{air} = 20°C$

For wind chill effect, convective heat transfer coefficient between man and air is $h = 30\ \text{W/m}^2\cdot\text{K}$

Step 2 of 3

Rate of heat loss from the man by convection in still air is given by,

$$Q = hA_s\Delta T$$

Surface area of the man is given by

$$\begin{aligned} A_s &= \pi dl \\ &= \pi\times 0.3\times 1.7 \\ &= 1.6\ \text{m}^2 \end{aligned}$$

$$\begin{aligned} Q &= hA_s\Delta T \\ &= 15\times 1.6\times(34-20) \\ &= 336\ \text{W} \end{aligned}$$

Thus, rate of heat loss from the man is $\boxed{Q = 336\ \text{W}}$

Step 3 of 3

For windy air, if Q is the rate of heat dissipation due to generation of heat, we have,

$$\begin{aligned} Q &= hA_s\Delta T \\ &= 30\times 1.6\times(34-20) \\ &= 672\ \text{W} \end{aligned}$$

For an effective balance of heat to cause wind-chill effect,

$$Q = hA_s\Delta T$$

$$672 = 15\times 1.6\times(34-T)$$

$$(34-T) = \frac{672}{15\times 1.6}$$

$$T = 6°C$$

Thus, rate of heat loss due to windy air, $\boxed{Q = 672\ \text{W}}$

Therefore, the wind chill temperature turns out to be $\boxed{T = 6°C}$.

Question: 16.073P

A thin metal plate is insulated on the back and exposed to solar radiation on the front surface. The exposed surface of the plate has an absorptivity of 0.7 for solar radiation. If solar radiation is incident on the plate at a rate of 550 W/m2and the surrounding air temperature is 10°C. determine the surface temperature of the plate when the heat loss by convection equals the solar energy absorbed by the plate. Take the convection heat transfer coefficient to be 25 W/m2·K, and disregard any heat loss by radiation.

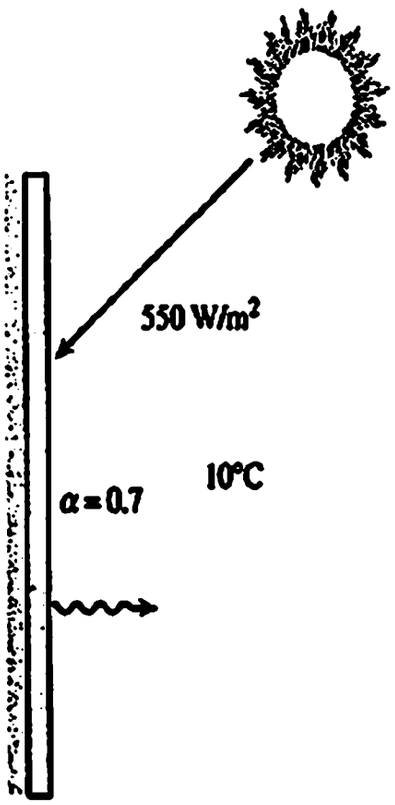


FIGURE P16–73

Step 1 of 2

Given

Surrounding temperature, $T_{surr} = 10°C$

Heat absorb by the surface, $\dot{Q}_{abs} = 550\,W/m^2$

The solar absorptivity of the plate, $\alpha = 0.7$

Heat transfer coefficient, $h = 25\,W/m^2 \cdot °C$

Step 2 of 2

Let heat transfer area is A.

And surface temperature of the plate is $T_{surface}$

When the heat loss from the plate by convection equals the solar radiation absorbed,

The surface temperature of the plate can be determined from

$$\dot{Q}_{solar,abs} = \dot{Q}_{conv}$$

$$\Rightarrow \alpha A \dot{Q}_{solar} = hA\left(T_{surface} - T_{surr}\right)$$

$$\Rightarrow 0.7 \times A \times 550 = 25 \times A \times \left(T_{surface} - 10\right)$$

$$\Rightarrow \left(T_{surface} - 10\right) = 15.4$$

$$\Rightarrow T_{surface} = 25.4°C$$

The surface temperature of the plate, $\boxed{T_{surface} = 25.4°C}$

Question: 16.074P

Consider a 3-m × 3-m × 3-m cubical furnace whose top and side surfaces closely approximate black surfaces at a temperature of 1200 K. The base surface has an emissivity of ε = 0.4, and is maintained at 800 K. Determine the net rate of radiation heat transfer to the base surface from the top and side surfaces.

Step 1 of 3

Given data:

The length of the side of the cube is $3\ \text{m}$

Emissivity between furnace and air, $\varepsilon = 0.4$

Temperature of the black body surfaces, $T_\infty = 1200\ \text{K}$

Base surface temperature, $T = 800\ \text{K}$

Step 2 of 3

The heat emitted from the top and side wall surfaces through radiation is equal to the heat absorbed by the base surface. Hence, the total heat transfer to the base surface,

$$Q = Q_{rad}$$

$$= \varepsilon \sigma A_s \left(T^4 - T_\infty^4\right)$$

where, surface area of heat transferring face,

$$A_s = 3 \times 3$$

$$= 9\ \text{m}^2$$

Step 3 of 3

Heat transfer due to radiation,

$$Q_{rad} = \varepsilon \sigma A_s \left(T^4 - T_\infty^4\right)$$

$$= 0.4 \times \left(5.67 \times 10^{-8}\right) \times 9 \times \left(1200^4 - 800^4\right)$$

$$= 339{,}655\ \text{W}$$

$$= 340\ \text{kW}$$

Therefore, heat transfer due to radiation is $\boxed{Q_{rad} = 340\ \text{kW}}$

Question: 16.075P

Engine valves (cp= 440 J/kg·K and ρ= 7840 kg/m3) are to be heated from 40°C to 800° in 5 min in the heat treatment section of a valve manufacturing facility. The valves have a cylindrical stem with a diameter of 8 mm and a length of 10 cm. The valve head and the stem may be assumed to be of equal surface area, with a total mass of 0.0788 kg. For a single valve, determine (*a*) the amount of heat transfer, (*b*) the average rate of heat transfer, (*c*) the average heat flux, and (*d*) the number of valves that can be heat treated per day if the heating section can hold 25 valves and it is used 10 h per day.

Step 1 of 5

Given

Specific heat of valve material, $c_p = 440\,\text{J/kg}\cdot^\circ\text{C}$

Density of valve material, $\rho = 7840\,\text{kg/m}^3$

Initial temperature, $T_1 = 40^\circ\text{C}$

Final temperature, $T_2 = 800^\circ\text{C}$

Time taken, $\Delta t = 5\,\text{min}$

Diameter of cylinder stem, $d = 8\,\text{mm}$

Length of cylinder stem, $L = 10\,\text{cm}$

Total mass of valve, $m = 0.0788\,\text{kg}$

Step 2 of 5

(a) The amount of heat transferred for a single valve is given by change in its internal energy.

So, $Q = \Delta U$

$$= mc_p(T_2 - T_1)$$
$$= 0.0788\times 440\times(800-40)$$
$$= 26350.7\,\text{J}$$
$$= 26.35\,\text{kJ}$$

Amount of heat transferred for a single valve, $\boxed{Q = 26.35\,\text{kJ}}$

Step 3 of 5

(b) The average rate of heat transfer is calculated as Amount of heat transfer divided by total time taken.

$$\dot{Q}_{ave} = \frac{Q}{\Delta t}$$
$$= \frac{26.35}{5\times 60}$$
$$= 0.0878\,\text{kW}$$

Average rate of heat transfer, $\boxed{\dot{Q}_{ave} = 0.0878\,\text{kW}}$

Step 4 of 5

(c) The average heat flux is determined from

$$\dot{q}_{ave} = \frac{\dot{Q}_{ave}}{A}$$
$$= \frac{\dot{Q}_{ave}}{2\pi dL}$$
$$= \frac{0.0878\times 10^3}{2\pi\times 0.008\times 0.1}$$
$$= \frac{87.8}{5.026\times 10^{-3}}$$
$$= 17467.2\,\text{W/m}^2$$

Average heat flux, $\boxed{\dot{q}_{ave} = 17467.2\,\text{W/m}^2}$

Step 5 of 5

(d) The heating section can hold 25 valves and it is used 10 h per day.

The total number of valves that can be heat treated daily is

Number of valves,

$$n = \frac{(10\times 60)\times 25}{5}$$
$$= 3000\,\text{valves}$$

Number of valves, $\boxed{n = 3000\,\text{valves}}$

Question: 16.076P

Consider a flat-plate solar collector placed on the roof of a house. The temperatures at the inner and outer surfaces of the glass cover are measured to be 33°C and 31°C, respectively. The glass cover has a surface area of 2.5 m2, a thickness of 0.6 cm, and a thermal conductivity of 0.7 W/m·K. Heat is lost from the outer surface of the cover by convection and radiation with a convection heat transfer coefficient of 10 W/m2·K and an ambient temperature of 15°C. Determine the fraction of heat lost from the glass cover by radiation.

Step 1 of 3

Given data:

The surface area of the glass cover, $A_s = 2.5\ \text{m}^2$

Thickness of the cover, $L = 0.6\ \text{cm}$

Thermal conductivity of the cover, $k = 0.7\ \text{W/m}\cdot\text{K}$

Convective heat transfer coefficient between plate and air, $h = 10\ \text{W/m}^2\cdot\text{K}$

Inner surface temperature of the glass cover, $T_{in} = 31°\text{C}$

Outer surface temperature of the glass cover, $T_{out} = 33°\text{C}$

Ambient temperature, $T_\infty = 15°\text{C}$

Step 2 of 3

The total heat conducted across the cover is transferred by convection, followed by radiation.

$$Q_{cond} = Q_{rad} + Q_{conv}$$

$$kA_s \frac{\Delta T}{L} = Q_{rad} + hA_s\Delta T$$

Substituting the known values in the above equation, we have,

$$\begin{aligned} Q_{rad} &= Q_{cond} - Q_{conv} \\ &= \left(0.7\times 2.5\times \frac{33-31}{0.006}\right) - \left(10\times 2.5\times (31-15)\right) \\ &= 583.334 - 400 \\ &= 183.334\ \text{W} \end{aligned}$$

Step 3 of 3

The fraction of heat lost through radiation is the ratio of the heat lost by radiation to the total heat conducted. Let this fraction be x.

$$\begin{aligned} x &= \frac{Q_{rad}}{Q_{cond}} \\ &= \frac{Q_{rad}}{kA_s \frac{\Delta T}{L}} \\ &= \frac{183.334}{0.7\times 2.5\times \frac{33-31}{0.006}} \\ &= 0.314 \end{aligned}$$

The fraction of heat lost by radiation of the total heat loss is $\boxed{x = 0.314}$

Question: 16.077P

The rate of heat loss through a unit surface area of a window per unit temperature difference between the indoors and the outdoors is called the *U*-factor. The value of the *U*-factor ranges from about 1.25 W/m2·K (or 0.22 Btu/h·ft2·°F) for low-ecoated, argon-filled, quadruple-pane windows to 6.25 W/m2·K (or 1.1 Btu/h·ft2·°F) for a singlc: pane window with aluminum frames. Determine the range for the rate of heat loss through a 1.2-m × 1.8-m window of a house that is maintained at 20°C when the outdoor air temperature is –8°C.

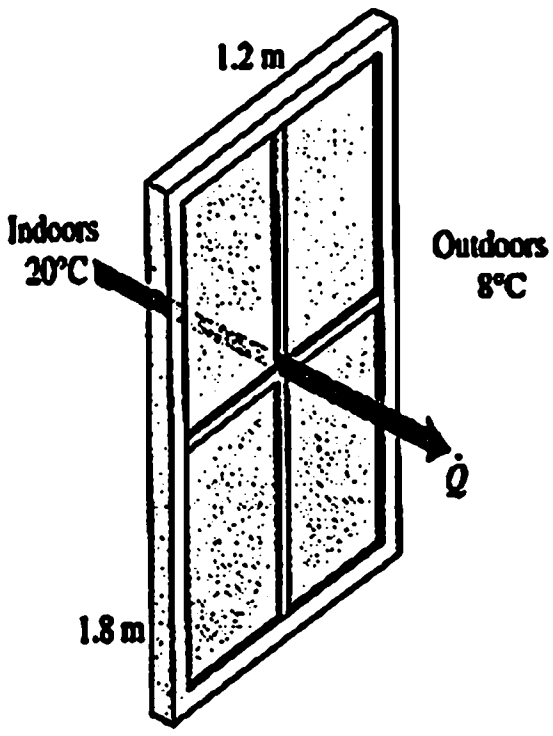


FIGURE P16–77

Step 1 of 4

Given:

U- Factor ranges from $1.25\ \frac{W}{m^2 \cdot °C}$ to $6.25\ \frac{W}{m^2 \cdot °C}$

Area of window $A = 1.2m \times 1.8m$

Inside temperature of house $T_{inside} = 20°C$

Outside temperature $T_{outside} = -8°C$

Step 2 of 4

The rate of heat loss $\dot{Q}_{window}$ through the window is given by

$\dot{Q}_{window} = U_{overall} A_{window} (T_{in} - T_{out})$

$U_{overall}$ Is the U-factor (the overall heat transfer coefficient) of the window,

A_{window} Is the window area.

Step 3 of 4

Substituting,

Maximum heat loss:

$$\dot{Q}_{window,max} = \left(6.25\ \frac{W}{m^2 \cdot °C}\right)(1.2 \times 1.8\ m^2)[20 - (-8)]°C$$

$Q_{window\,max} = 378\ W$

$\boxed{Q_{window\,max} = 378\ W}$

Step 4 of 4

Minimum heat loss:

$$\dot{Q}_{window,min} = \left(1.25\ \frac{W}{m^2 \cdot °C}\right)(1.2 \times 1.8\ m^2)[20 - (-8)]°C$$

$\dot{Q}_{window,min} = 75.6\ W$

$\boxed{\dot{Q}_{window,min} = 75.6\ W}$

Range of rate of heat loss through the window for low e-coated, argon filled, quadruple-pane windows to single-pane with aluminum frames is from 75.6 W to 378 W.

Question: 16.078P

Reconsider Prob. 16–77. Using EES (or other)software, plot the rate of heat loss through the window as a function of the *U*-factor. Discuss the results.

Step 1 of 6

Given:

U- Factor ranges from $1.25\ \frac{W}{m^2 \cdot C}$ to $6.25\ \frac{W}{m^2 \cdot C}$

Area of window $A = 1.2m \times 1.8m$

Inside temperature of house $T_{inside} = 20°C$

Outside temperature $T_{outside} = -8°C$

Step 2 of 6

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box
"GIVEN"
A=1.2*1.8 "[m^2]"
T_1=20 "[C]"
T_2=-8 "[C]"
"U=1.25 [W/m^2-C], parameter to be varied"

"ANALYSIS"
Q_dot_window=U*A*(T_1-T_2)

Step 3 of 6

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 11
Select variables in equations as U and Q_{window} press ADD and then press ok
A table will appear put values of U as 1.25,1.75,2.25....6.25 in run 1to 11
Then from Calculate -> Solve table

Step 4 of 6

Solution will appear

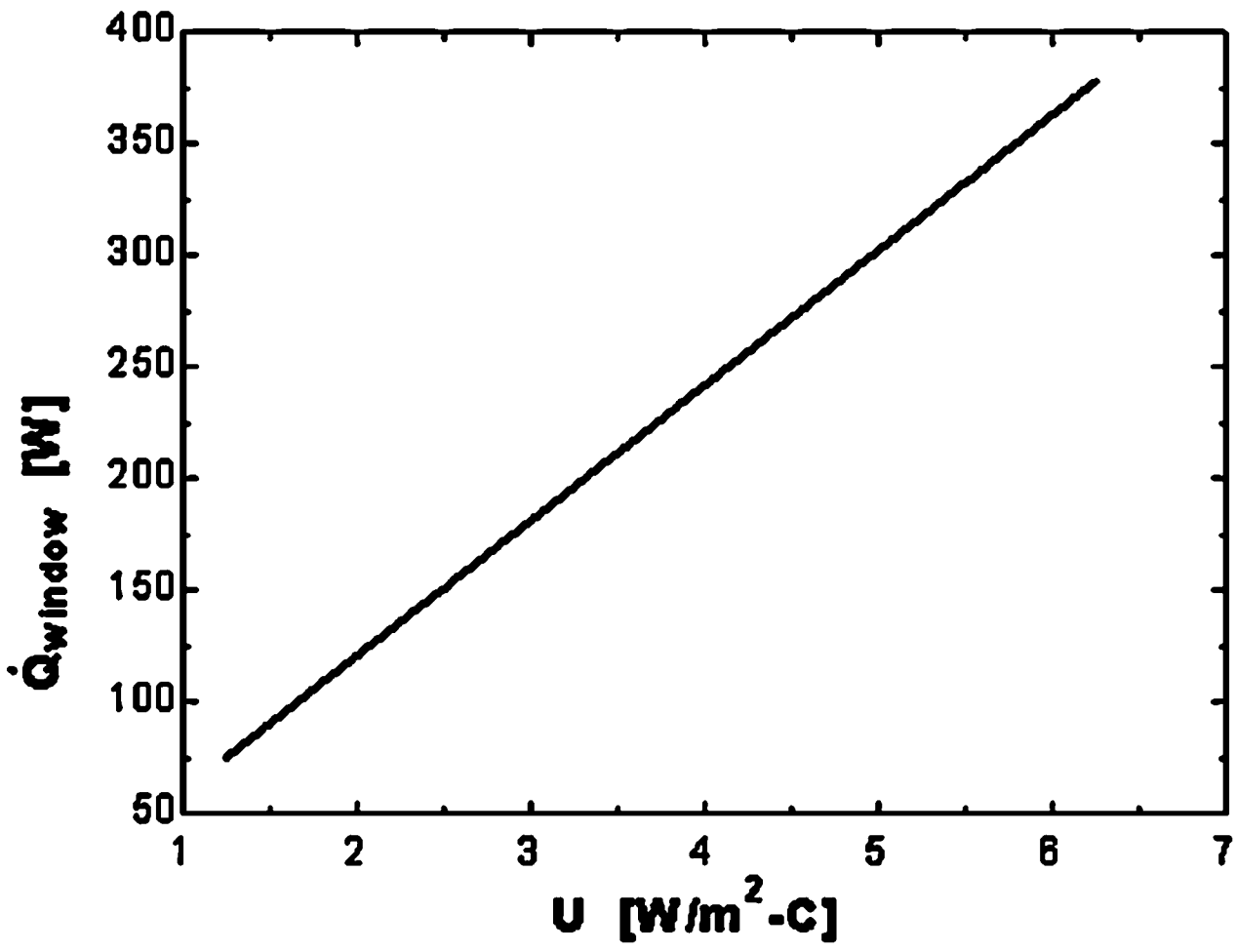
$U\ [W/m^2.C]$	$Q_{window}\ [W]$
1.25	75.6
1.75	105.8
2.25	136.1
2.75	166.3
3.25	196.6
3.75	226.8
4.25	257
4.75	287.3
5.25	317.5
5.75	347.8
6.25	378

Step 5 of 6

Then from Plot -> New plot window -> X-Y Plot
Select X-axis U and Y-axis Q_{window}
Press OK

Step 6 of 6

Graph will be plotted



Question: 16.079P

An electric heater with the total surface area of 0.25 m2 and emissivity 0.75 is in a room where the air has a temperature of 20°C and the walls are at 10°C. When the heater consumes 500 W of electric power, its surface has a steady temperature of 120°C. Determine the temperature of the heater surface when it consumes 700 W, Solve the problem (*a*) assuming negligible radiation and (*b*) taking radiation into consideration. Based on your results, comment on the assumption made in part (*a*).

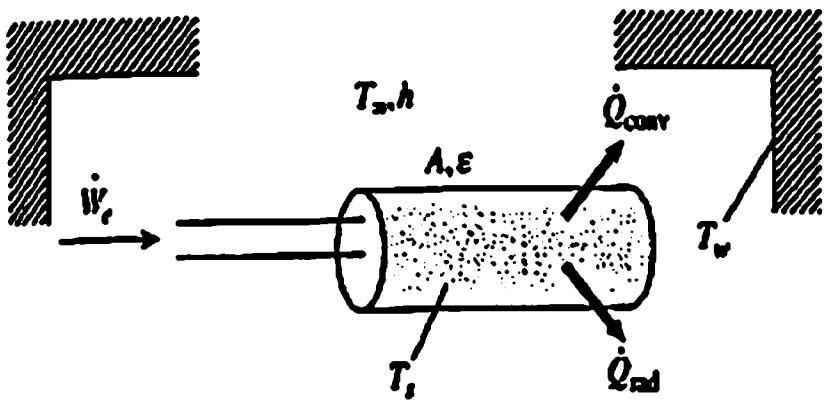


FIGURE P16–79

Step 1 of 4

Given: -

Surface area of heater $(A) = 0.25\text{m}^2$

Emissivity of heater surface $(\varepsilon) = 0.75$

Temperature of surrounding air $T_\infty = 20^\circ\text{C}$

Temperature of walls $T_w = 10^\circ\text{C}$

Power consumed by heater $Q = 500\text{ W}$

Surface temperature of temperature $T_s = 120^\circ\text{C}$

Step 2 of 4

(a) For steady flow condition

Heat developed in heater = heat flow by convection + heat flow by radiation

Neglecting Radiation

So Heat developed in heater = Heat flow by convection

$$(500\text{W}) = h \times (0.25\text{m}^2) \times (120 - 20)^\circ\text{C}$$

$$h = 20 \frac{\text{W}}{\text{m}^2 \cdot {}^\circ\text{C}}$$

Step 3 of 4

Surface temperature of heater when heater consumes 700 W

$$(700\text{W}) = \left(20 \frac{\text{W}}{\text{m}^2 \cdot {}^\circ\text{C}}\right) \times (0.25\text{m}^2) \times (T - 20)^\circ\text{C}$$

$$(T - 20)^\circ\text{C} = 140^\circ\text{C}$$

$$T = 160^\circ\text{C}$$

Step 4 of 4

(b) Taking radiation into consideration

$$Q = hA(\Delta T) + \varepsilon\sigma A(T^4 - T_2^4)$$

$$(500\text{W}) = h \times (0.25\text{m}^2) \times (120 - 20)^\circ\text{C} + 0.75 \times (5.67 \times 10^{-8} \frac{\text{W}}{\text{m}^2 \cdot \text{K}^4}) \times (0.25\text{m}^2) \times \left[(393\text{K})^4 - (283\text{K})^4\right]$$

$$h = 12.58 \frac{\text{W}}{\text{m}^2 \cdot {}^\circ\text{C}}$$

When Q = 700W then surface temperature of heater will be

$$700 = 12.58 \times 0.25 \times (T - 293) + 0.75 \times 5.67 \times 10^{-8} \times 0.25 \times (T^4 - 283^4)$$

$$T_s = 425.9\text{K}$$

$$\boxed{T_s = 152.9^0\text{C}}$$

So comparing the two answers we can say that we cannot affect of radiation in this case as temperature difference in two case is up to 7°C

CHAPTER 17
SOLUTIONS

Question: 17.001P

Consider heat conduction through a wall of thickness L and area A. Under what conditions will the temperature distributions in the wall be a straight line?

Step 1 of 2

The thermal resistance of a medium represents the resistance of that medium against heat transfer. Heat flow through any medium can be compared with the electrical circuit

For electrical circuits, $dV = IR$

Here, dV is voltage difference in the resistance of the circuit, and I is the current flow.

This equation can be rearranged as $I = dV/R$ .....(1)

Step 2 of 2

Write the equation of Fourier's as follows:

$$Q = \frac{dT}{\left(\frac{dx}{KA}\right)} \quad \text{.....(2)}$$

Here, Q is the heat flow through medium, dx is the change of thickness, K is the thermal conductivity, and A is the cross-sectional area.

Compare the equations (1) and (2), resistance (R) of the electrical circuit can be compared with the terms $\frac{dx}{KA}$ and it is called thermal resistance.

Question: 17.002P

Consider heat conduction through a plane wall. Does the energy content of the wall change during steady heat conduction? How about during transient conduction? Explain.

Step 1 of 1

During the steady heat conduction, the rate of heat transfer into the wall is equal to the rate of heat transfer out of it. Also, the temperature at any point in the wall remains constant. Therefore, the energy content of the wall does not change.

During transient conduction the temperature along the wall will change with time, thus the energy content of the wall will change.

Question: 17.003P

Consider steady heat transfer through the wall of a room in winter. The convection heat transfer coefficient at the outer surface of the wall is three times that of the inner surface as a result of the winds. On which surface of the wall do you think the temperature will be closer to the surrounding air temperature? Explain.

Step 1 of 1

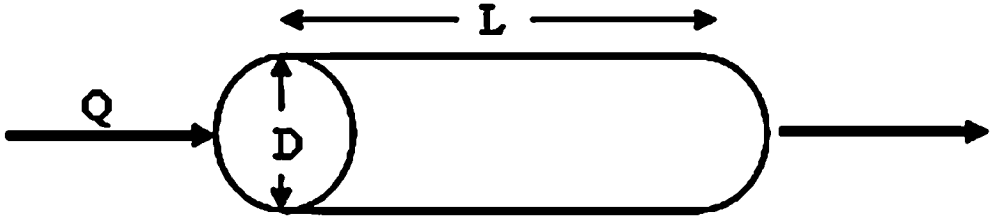
We know that convection heat transfer through the wall is expressed as $\dot{Q} = hA_s(T_s - T_\infty)$. In steady heat transfer, heat transfer rate to the wall and from the wall are equal. Therefore at the outer surface which has convection heat transfer coefficient three times that of the inner surface will experience three times smaller temperature drop compared to the inner surface. Therefore, at the outer surface, the temperature will be closer to the surrounding air temperature.

Question: 17.004P

Consider one-dimensional heat conduction through a cylindrical rod of diameter *D* and length *L*. What is the heat transfer area of the rod if (*a*) the lateral surfaces of the rod are insulated and (*b*) the top and bottom surfaces of the rod are insulated?

Step 1 of 3

Consider a one – dimensional heat conduction through a cylindrical rod of diameter D and length L.



Step 2 of 3

(a)

If the lateral surfaces of the rod is insulated, the heat transfer surface area of the cylindrical rod is the bottom or the top surface area of the rod, $A_s = \pi D^2/4$.

Step 3 of 3

(b)

When the top and the bottom surfaces of the rod are insulated, the heat transfer area of the rod is the lateral surface area of the rod, $A = \pi DL$.

Question: 17.005P

What does the thermal resistance of a medium represent?

Step 1 of 2

The thermal resistance of a medium represents the resistance of that medium against heat transfer. Heat flow through any medium can be compared with the electrical circuit

For electrical circuits, $dV = IR$

Here, dV is voltage difference in the resistance of the circuit, and I is the current flow.

This equation can be rearranged as $I = dV/R$ .....(1)

Step 2 of 2

Write the equation of Fourier's as follows:

$$Q = \frac{dT}{\left(\frac{dx}{KA}\right)} \quad \text{.....(2)}$$

Here, Q is the heat flow through medium, dx is the change of thickness, K is the thermal conductivity, and A is the cross-sectional area.

Compare the equations (1) and (2), resistance (R) of the electrical circuit can be compared with the terms $\frac{dx}{KA}$ and it is called thermal resistance.

Question: 17.006P

How is the combined heat transfer coefficient defined? What convenience does it offer in heat transfer calculations?

Step 1 of 1

The combined heat transfer coefficient represents the combined effects of radiation and convection heat transfers on a surface, and is defined as $h_{combined} = h_{convection} + h_{radiation}$.
It offers the convenience of incorporating the effects of radiation in the convection heat transfer coefficient, and to ignore radiation in heat transfer calculations.

Question: 17.007P

Can we define the convection resistance for a unit surface area as the inverse of the convection heat transfer coefficient?

Step 1 of 1

Yes, from Newton's law of cooling heat flow (Q) = hAdT where A is the surface area and dT is the change of temperature.This equation can compare with electrical analogy equation dV is voltage difference, R is the restence of the circuit and I is the current flow I = dV/R This R after compare with thermal resistence it is the inverse of the convection heat transfer coefficient per unit surface area since it is defined as $R_{conv} = \frac{1}{hA}$

Question: 17.008P

Why are the convection and the radiation resistances at a surface in parallel instead of being in series?

Step 1 of 1

A surface exposed to the surrounding air involves convection and radiation simultaneously, and the total heat transfer at the surface is determined by adding the radiation and convection components. The convection and radiation are parallel to each other. When $T_{surr} \approx T_{\infty}$

Convection can occurs when heat is transferred between two different medium and radiation can occurs with out medium. So both the convection and radiation heat transfers occur simultaneously and they are parallel.

Question: 17.009P

Consider a surface of area A at which the convection and radiation heat transfer coefficients are $h$conv and $h$rad, respectively. Explain how you would determine (*a*) the single equivalent heat transfer coefficient, and (*b*) the equivalent thermal resistance. Assume the medium and the surrounding surfaces are at the same temperature.

Step 1 of 2

(a) For a surface of A at which the convection and radiation heat transfer coefficients are h_{conv} and h_{rad}. When a surface exposed to the surroundings air involves convection and radiation simultaneously, the convection and radiation resistances are parallel to each other, the single equivalent heat transfer coefficient is $h_{eqv} = h_{conv} + h_{rad}$ when the medium and the surrounding surfaces are at the same temperature.

Step 2 of 2

(b) The equivalent thermal resistance will be $R_{eqv} = 1/(h_{eqv}A)$ where A is the surface area.

Question: 17.010P

How does the thermal resistance network associated with a single-layer plane wall differ from the one associated with a five-layer composite wall?

Step 1 of 1

The thermal resistance network associated with a five-layer composite wall involves five single-layer resistances connected in series. If thermal resistance of any single all is R_{th} then for five layer it will be $R_{th1} + R_{th2} + R_{th3} + R_{th4} + R_{th5}$

Question: 17.011P

Consider steady one-dimensional heat transfer through a multilayer medium. If the rate of heat transfer Q is known, explain how you would determine the temperature drop across each layer.

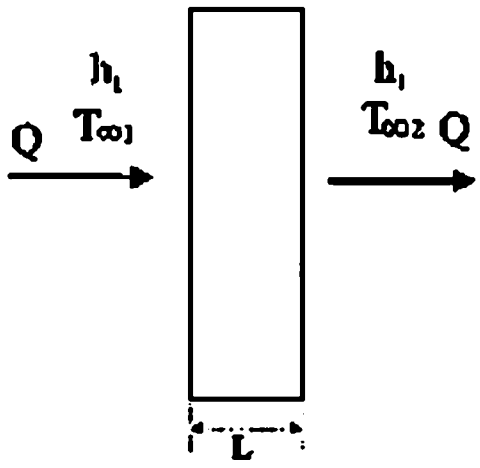
Step 1 of 1

Once the rate of heat transfer $\dot{Q}$ is known, the temperature drop across any layer can be determined by multiplying heat transfer rate by the thermal resistance across that layer, $\Delta T_{layer} = \dot{Q} R_{layer}$

Question: 17.012P

Consider steady one-dimensional heat transfer through a plane wall exposed to convection from both sides to environments at known temperatures $T_{\infty 1}$ and $T_{\infty 2}$ with known heat transfer coefficients h_1, and h_2. Once the rate of heat transfer Q has been evaluated, explain how you would determine the temperature of each surface.

Step 1 of 6



Step 2 of 6

Let the thickness of the body is L
Surface area is A
Convective heat transfer coefficient is h_1 for inside and h_2 for outside
Inside surface temperature, T_1
Inside surrounding temperature, $T_{\infty 1}$
And outside surface temperature is T_2
Outside surrounding temperature, $T_{\infty 2}$
Thermal conductivity of material is K

Step 3 of 6

Then heat transfer from surface to wall due to inside convection is

$$Q = h_1 A(T_{\infty 1} - T_1)$$

$$(T_{\infty 1} - T_1) = \frac{Q}{h_1 A} \quad \text{----------(1)}$$

Step 4 of 6

Heat transfer through wall to wall due to conduction is

$$Q = KA(T_1 - T_2)$$

$$(T_1 - T_2) = \frac{Q}{KA} \quad \text{----------(2)}$$

Step 5 of 6

Heat transfer from wall to surface due to outside convection is

$$Q = h_2 A(T_2 - T_{\infty 2})$$

$$(T_2 - T_{\infty 2}) = \frac{Q}{h_2 A} \quad \text{----------(3)}$$

Step 6 of 6

So after addition equation (1), (2), and (3)

$$T_{\infty 1} - T_{\infty 2} = Q\left(\frac{1}{h_1 A} + \frac{1}{KA} + \frac{1}{h_2 A}\right)$$

$$\boxed{T_1 = T_{\infty 1} - \frac{Q}{h_1 A}}$$

$$\boxed{T_2 = \frac{Q}{h_2 A} + T_{\infty 2}}$$

Question: 17.013P

Someone comments that a microwave oven can be viewed as a conventional oven with zero convection resistance at the surface of the food. Is this an accurate statement?

Step 1 of 1

Yes, it is. Because when zero convection resistance there is no convective heat transfer so it is treated as a conventional oven

Question: 17.014P

Consider a window glass consisting of two 4-mm-thick glass sheets pressed tightly against each other. Compare the heat transfer rate through this window with that of one consisting of a single 8-mm-thick glass sheet under identical conditions.

Step 1 of 1

The window glass which consists of two 4 mm thick glass sheets pressed tightly against each other will probably have thermal contact resistance which serves as an additional thermal resistance to heat transfer through window, and thus the heat transfer rate will be smaller relative to the one which consists of a single 8 mm thick glass sheet.

Question: 17.015P

Consider two cold canned drinks, one wrapped in a blanket and the other placed on a table in the same room. Which drink will warm up faster?

Step 1 of 1

The blanket will introduce additional resistance due to conduction and convection so the heat transfer from the wrapped blanket surface is less. Where as the heat transfer from the drink which is placed on a table is more because in this resistance due to convection only. Therefore, the drink left on a table will warm up faster.

Question: 17.016P

Consider a 3-m-high, 6-m-wide, and 0.25-m-thick brick wall whose thermal conductivity is k =0.8 W/m·K. On a certain day, the temperatures of the inner and the outer surfaces of the wall are measured to be 14°C and 5°C, respectively. Determine the rate of heat loss through the wall on that day.

Step 1 of 4

Given data:

Height of the wall $h = 3\,\text{m}$

Wide of the wall $b = 6\,\text{m}$

Thickness of the wall, $L = 0.25\text{m}$

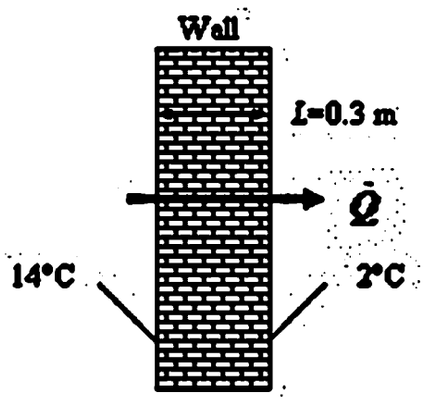
The thermal conductivity $k = 0.8\,\text{W/m}\cdot\text{K}$

Inner surface temperature $T_1 = 14°\text{C}$

Outer surface temperature $T_2 = 5°\text{C}$

Step 2 of 4

Diagram:



Step 3 of 4

If the heat transfer through the wall is steady since the surface temperatures remain constant at the specified values and the heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors

Step 4 of 4

The surface area of the wall and the rate of heat loss through the wall are

$A = bh$

$A = (3\,\text{m}) \times (6\,\text{m})$

$A = 18\,\text{m}^2$

Rate of heat transfer,

$$\dot{Q} = kA\frac{T_1 - T_2}{L}$$

$$= (0.8)(18) \times \frac{(14-5)}{0.25}$$

$$= 518.4\,\text{W}$$

Hence, the rate of heat transfer is $\boxed{\dot{Q} = 518.4\,\text{W}}$

Question: 17.017P

Consider a 1.5-m-high and 2.4-m-wide glass window. whose thickness is 6 mm and thermal conductivity is k = 0.78 W/m·K. Determine the steady rate of heat transfer through this glass window and the temperature of its inner surface for a day during which the room is maintained at 24°C while the temperature of the outdoors is −5°C. Take the convection heat transfer coefficients on the inner and outer surfaces of the window to be $h1$, = 10 W/m2·K and $h2$ = 25 W/m2·K, and disregard any heat transfer by radiation.

Step 1 of 6

Given data:

Height of the glass window, $H = 1.5\,\text{m}$.

Width of the glass window, $W = 2.4\text{m}$.

Thickness of the glass, $L = 0.006\,\text{m}$.

Thermal conductivity of the glass, $k = 0.78\,\text{W/m-K}$.

Temperature of the room, $T_{\infty 1} = 24°\text{C}$.

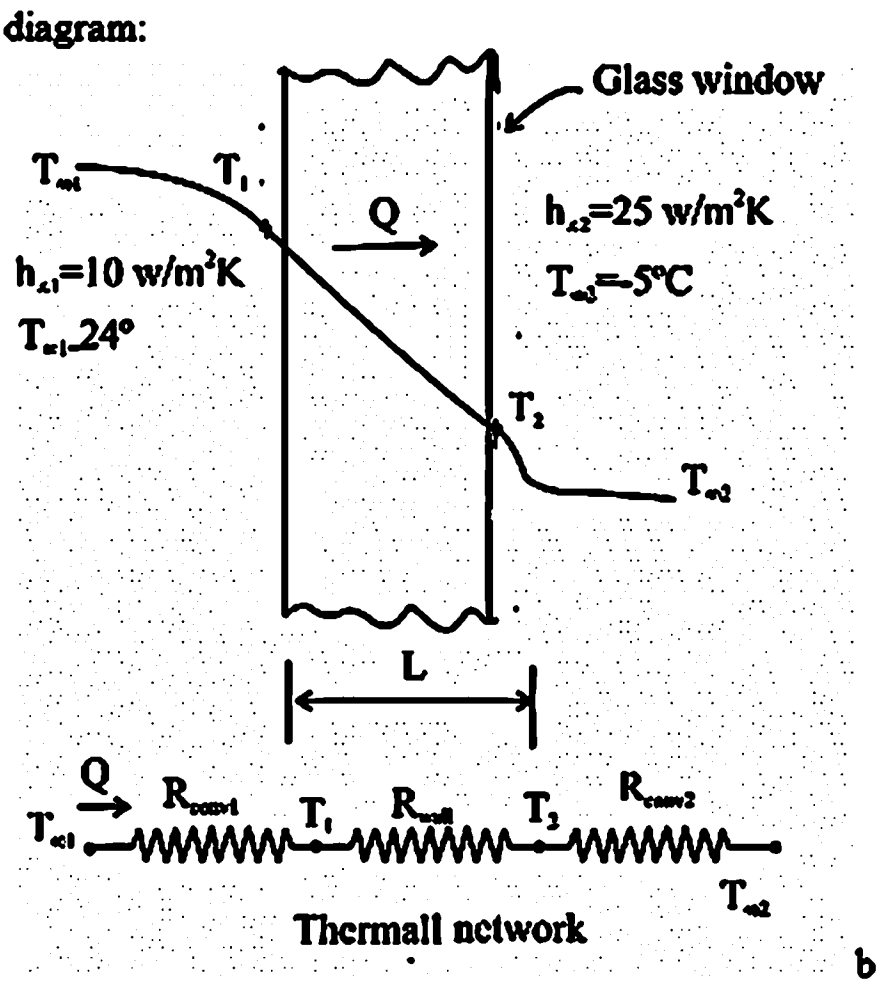
Temperature of the outdoors, $T_{\infty 2} = -5°\text{C}$.

Heat transfer coefficient on the inner surface of the window, $h_1 = 10\,\text{W/m}^2\text{-K}$.

Heat transfer coefficient on the outer surface of the window, $h_2 = 25\,\text{W/m}^2\text{-K}$.

Step 2 of 6

Thermal network diagram:



Step 3 of 6

The heat transfer through the glass window can be determined using the thermal resistance network concept as follows,

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{conv,1} + R_{glass} + R_{conv,2}}$$

The cross sectional area of the glass window can be calculated as

$$A = H \times W$$
$$= 1.5(\text{m}) \times 2.4(\text{m})$$
$$= 3.6\,\text{m}^2$$

Step 4 of 6

The individual resistances are calculated as follows

$$R_{conv,1} = \frac{1}{h_1 A}$$
$$= \frac{1}{10(\text{W/m}^2\text{-K}) \times 3.6(\text{m}^2)}$$
$$= 0.02778\,\text{K/W}$$
$$R_{glass} = \frac{L}{kA}$$
$$= \frac{0.006(\text{m})}{0.78(\text{W/m-K}) \times 3.6(\text{m}^2)}$$
$$= 0.002137\,\text{K/W}$$
$$R_{conv,2} = \frac{1}{h_2 A}$$
$$= \frac{1}{25(\text{W/m}^2\text{-K}) \times 3.6(\text{m}^2)}$$
$$= 0.0111\,\text{K/W}$$

Step 5 of 6

Therefore, the heat transfer through glass window is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{conv,1} + R_{glass} + R_{conv,2}}$$
$$= \frac{297.15(\text{K}) - 268.15(\text{K})}{0.02778(\text{K/W}) + 0.002137(\text{K/W}) + 0.0111(\text{K/W})}$$
$$= 707.024\,\text{W}$$

Therefore, the heat transfer through glass window is $\boxed{\dot{Q} = 707.024\,\text{W}}$.

Step 6 of 6

The inner surface temperature of the glass window can be expressed as

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{conv,1}}$$
$$707.024(\text{W}) = \frac{297.15(\text{K}) - (T_1)(\text{K})}{0.02778(\text{K/W})}$$
$$\Rightarrow T_1 = 277.51\,\text{K}$$
$$\Rightarrow T_1 = 4.36\,°\text{C}$$

$\boxed{T_1 = 4.36\,°\text{C}}$

Question: 17.018P pg 1

Consider a 1.5-m-high and 2.4-m-wide double-pane window consisting of two 3-mm-thick layers of glass (k = 0.78 W/m·K) separated by a 12-mm-wide stagnant air space (k = 0.026 W/m·K). Determine the steady rate of heat transfer through this double-pane window and the temperature of its inner surface for a day during which the room is maintained at 21°C while the temperature of the outdoors is −5°C. Take the convection heat transfer coefficients on the inner and outer surfaces of the window to be $h1$, = 10 W/m2·K and $h2$ = 25 W/m2·K, and disregard any heat transfer by radiation.

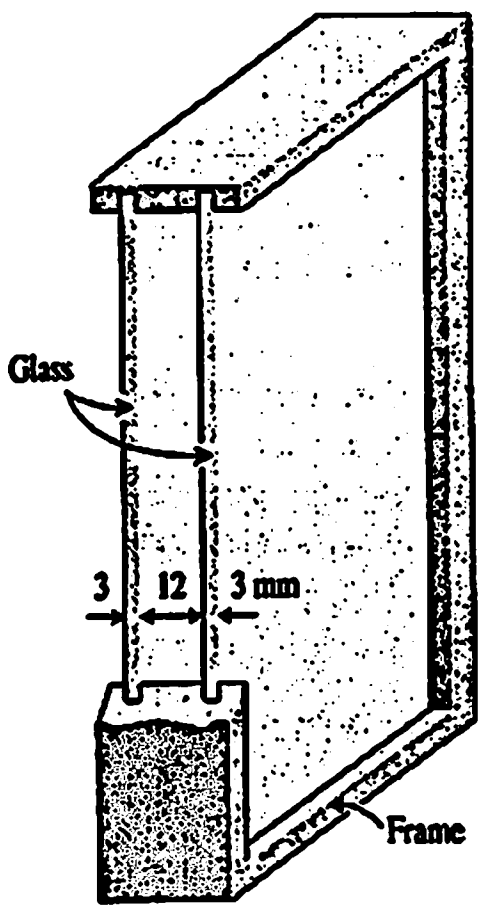


FIGURE P17–18

Step 1 of 8

Given data:

Height of the glass window, $H = 1.5\,\text{m}$.

Width of the glass window, $W = 2.4\,\text{m}$.

Thickness of the glass,

$$L_1 = 3\,\text{mm}$$
$$= \frac{3}{1000}\,\text{m}$$
$$= 0.003\,\text{m}$$

Distance between two glasses,

$$L_2 = 12\,\text{mm}$$
$$= \frac{12}{1000}\,\text{m}$$
$$= 0.012\,\text{m}$$

Thermal conductivity of the glass, $k = 0.78\ \text{W/m}\cdot\text{K}$

Thermal conductivity of the air, $k_2 = 0.026\ \text{W/m}\cdot\text{K}$

Temperature of the room,

$$T_{\infty 1} = 21°\text{C}$$
$$= 21 + 273.15\ \text{K}$$
$$= 294.15\ \text{K}$$

Temperature of the outdoors,

$$T_{\infty 2} = -5°\text{C}$$
$$= -5 + 273.15\ \text{K}$$
$$= 2268.15\ \text{K}$$

Heat transfer coefficient on the inner surface of the window, $h_1 = 10\ \text{W/m}^2\cdot\text{K}$

Heat transfer coefficient on the outer surface of the window, $h_2 = 25\ \text{W/m}^2\cdot\text{K}$

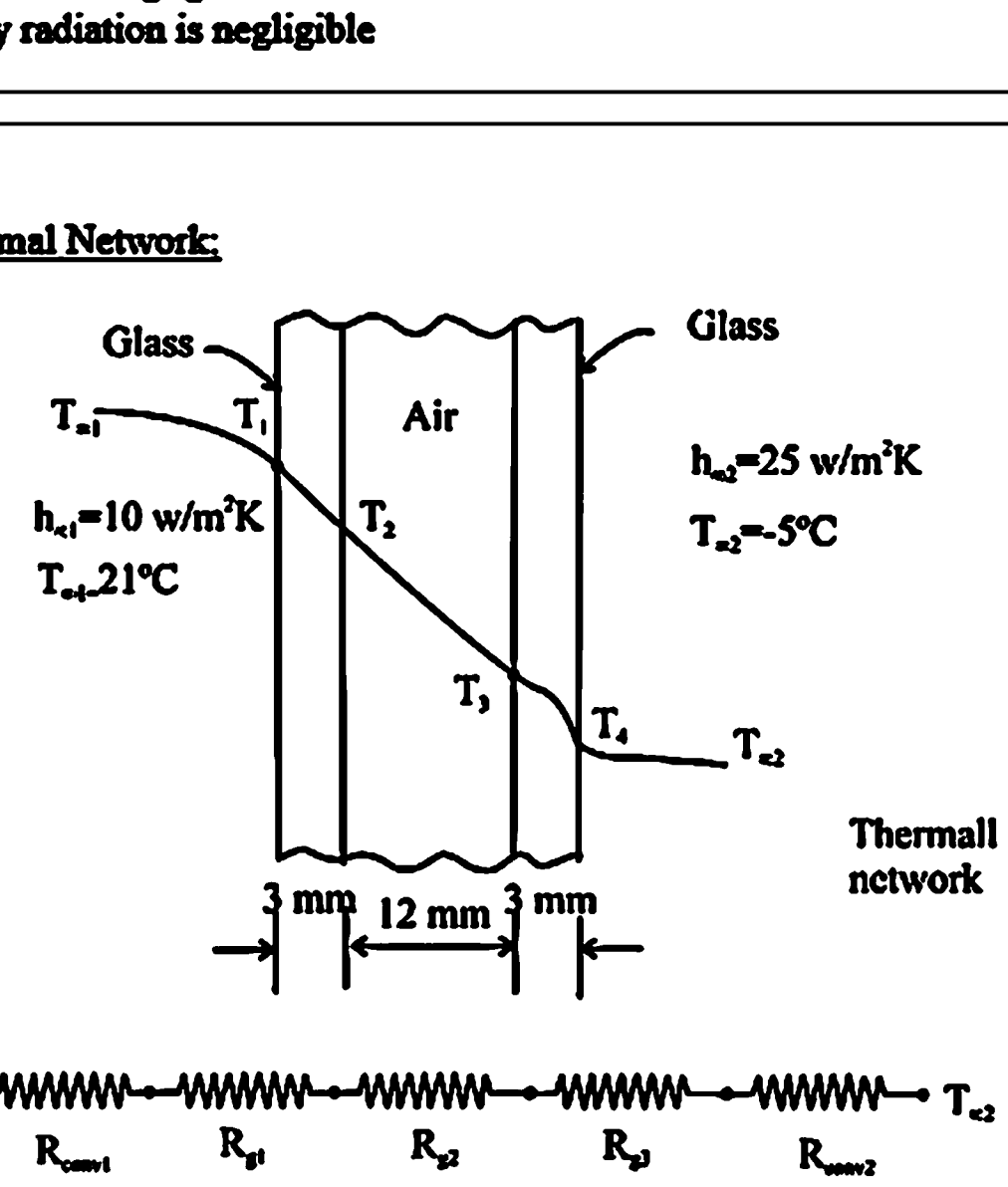
Step 2 of 8

Assumptions:

1. Heat transfer through the glass window is under steady state.
2. Thermal conductivity of the glass window is constant.
3. Heat transfer through the glass window is one dimensional and the gradients in Width and height direction are negligible.
4. Heat transfer by radiation is negligible

Step 3 of 8

Diagram for Thermal Network:



Step 4 of 8

The individual resistances are calculated as follows

$$R_{conv,1} = \frac{1}{h_1 A}$$
$$= \frac{1}{10 \times 3.6}$$
$$= 0.02778\,\text{K/W}$$

$$R_{glass,1} = \frac{L_1}{k_1 A}$$
$$= \frac{0.003}{0.78 \times 3.6}$$
$$= 0.0010684\,\text{K/W}$$

$$R_{glass,1}=\frac{L_1}{k_1A}$$
$$=\frac{0.003}{0.78\times3.6}$$
$$=0.0010684\,\text{K/W}$$

Step 5 of 8

$$R_{air,2}=\frac{L_2}{k_2A}$$
$$=\frac{0.012}{0.026\times3.6}$$
$$=0.1282\,\text{K/W}$$
$$R_{glass,3}=\frac{L_3}{k_3A}$$
$$=\frac{0.003}{0.78\times3.6}$$
$$=0.0010684\,\text{K/W}$$

Step 6 of 8

$$R_{conv,2}=\frac{1}{h_2A}$$
$$=\frac{1}{25\times3.6}$$
$$=0.0111\,\text{K/W}$$

The cross sectional area of the glass window can be calculated as

$$A=H\times W$$
$$=1.5(\text{m})\times2.4(\text{m})$$
$$=3.6\,\text{m}^2$$

Step 7 of 8

The heat transfer through the glass window can be determined using the thermal resistance network concept as follows,

$$\dot{Q}=\frac{T_{\infty1}-T_{\infty2}}{R_{conv,1}+R_{glass,1}+R_{air,2}+R_{glass,3}+R_{conv,2}}$$
$$=\frac{21-(-5)}{0.02778+0.001068+0.1282+0.001068+0.0111}$$
$$=153.65\,\text{W}$$

Hence, heat transfer through glass window is $\boxed{\dot{Q}=153.65\,\text{W}}$

Step 8 of 8

The inner surface temperature of the glass window can be expressed as,

$$\dot{Q}=\frac{T_{\infty1}-T_1}{R_{conv,1}}$$
$$153.65=\frac{294.15-(T_1)}{0.02778}$$
$$T_1=289.8812\,\text{K}$$
$$=16.73^\circ\text{C}$$

Hence, inner surface temperature of the glass window is $\boxed{T_1=16.73^\circ\text{C}}$

Question: 17.019P pg 1

Repeat Prob. 17–18, assuming the space between the two glass layers is evacuated.

Step 1 of 12

Given data:

Height of the glass window, $H = 1.5\,\text{m}$

Width of the glass window, $W = 2.4\,\text{m}$

Thickness of the glass,

$L_1 = 3\,\text{mm}$

$= \frac{3}{1000}\,\text{m}$

$= 0.003\,\text{m}$

Distance between two glasses,

$L_2 = 12\,\text{mm}$

$= \frac{12}{1000}\,\text{m}$

$= 0.012\,\text{m}$

Thermal conductivity of the glass, $k_1 = 0.78\ \text{W/m}\cdot\text{K}$

Temperature of the room,

$T_{\infty 1} = 21°\text{C}$

$= 21 + 273.15\ \text{K}$

$= 294.15\ \text{K}$

Temperature of the outdoors,

$T_{\infty 2} = -5°\text{C}$

$= -5 + 273.15\ \text{K}$

$= 2268.15\ \text{K}$

Heat transfer coefficient on the inner surface of the window, $h_1 = 10\ \text{W/m}^2\cdot\text{K}$

Heat transfer coefficient on the outer surface of the window, $h_2 = 25\ \text{W/m}^2\cdot\text{K}$

Stefan Boltzmann constant, $\sigma = 5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$

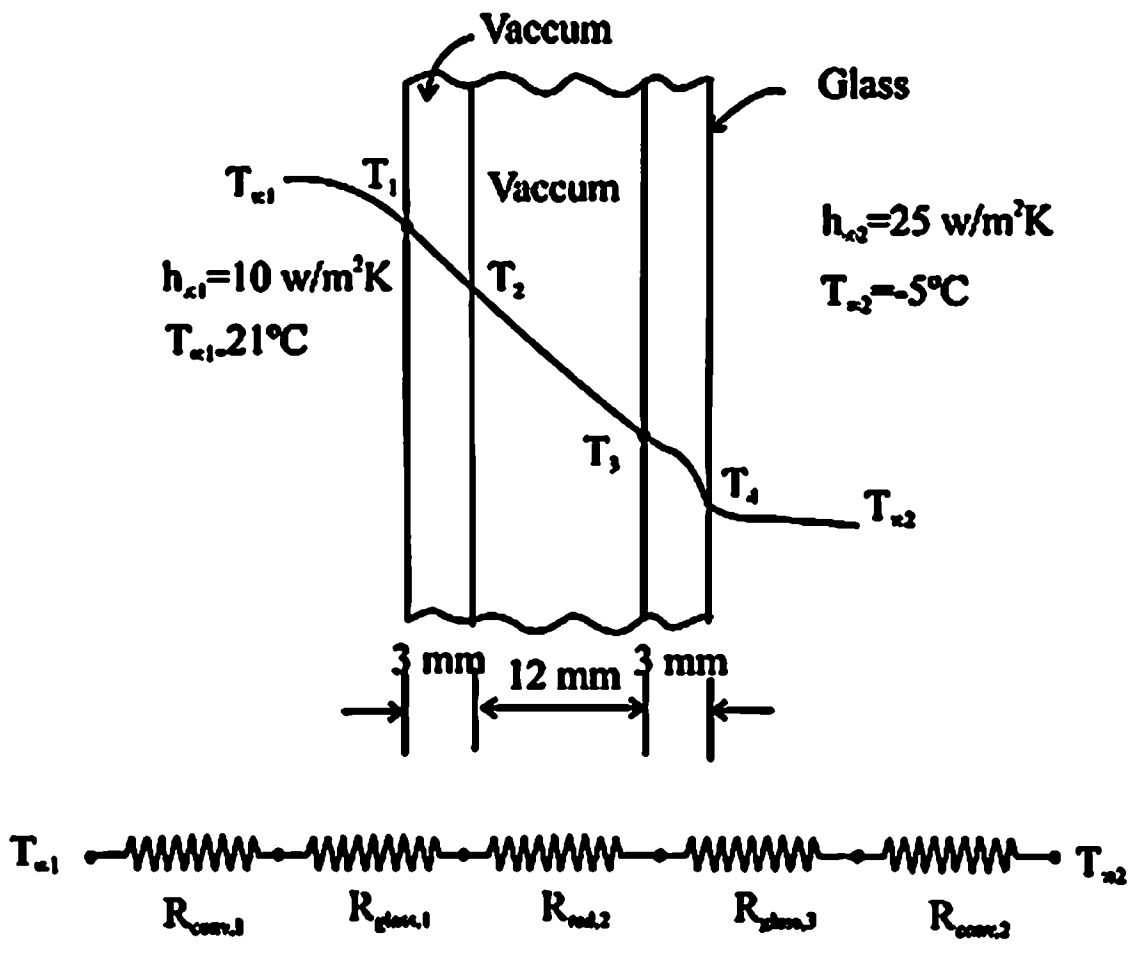
Step 2 of 12

Assumptions:

1. Heat transfer through the glass window is under steady state.
2. Thermal conductivity of the glass window is constant.
3. Heat transfer through the glass window is one dimensional and the gradients in width and height direction are negligible.

Step 3 of 12

Diagram for thermal network:



Step 4 of 12

The space between the two glass plates is evacuated and hence it can be considered that pure vacuum in the evacuated space. The mode of heat transfer between the glasses is radiation. In order to calculate the heat resistance with in the vacuum space, we need to know the inner surface temperature of the glasses. But we don't know the inner surface temperature of the glasses.

Therefore, assume the inner surface temperature of the glasses to be $T_2 = 18°\text{C}$ and $T_3 = 3°\text{C}$, and the emissivity of the glass is to be 1.

Therefore, the heat transfer through glasses can be determined as

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{conv,1} + R_{glass,1} + R_{vac,2} + R_{glass,3} + R_{conv,2}} \qquad \ldots\ldots (1)$$

Step 5 of 12

The cross sectional area of the glass window can be calculated as

$A = H \times W$

$= 1.5(\text{m}) \times 2.4(\text{m})$

$= 3.6\ \text{m}^2$

The individual resistances are calculated as follows,

$R_{conv,1} = \frac{1}{h_1 A}$

$= \frac{1}{10 \times 3.6}$

$= 0.02778\,\text{K/W}$

Step 6 of 12

$R_{glass,1} = \frac{L_1}{k_1 A}$

$= \frac{0.003}{0.78 \times 3.6}$

$= 0.0010684\,\text{K/W}$

$$R_{glass,1}=\frac{L_1}{k_1A}$$
$$=\frac{0.003}{0.78\times3.6}$$
$$=0.0010684\,\text{K/W}$$

$$R_{rad,2}=\frac{1}{\varepsilon\sigma A(T_2^2+T_3^2)(T_2+T_3)}$$
$$=\frac{1}{1\times5.67\times10^{-8}\times3.6\times(291.15^2+276.15^2)\times(291.15+276.15)}$$
$$=0.0536\,\text{K/W}$$

$$R_{glass,3}=\frac{L_3}{k_3A}$$
$$=\frac{0.003}{0.78\times3.6}$$
$$=0.0010684\,\text{K/W}$$

Step 7 of 12

$$R_{conv,2}=\frac{1}{h_2A}$$
$$=\frac{1}{25\times3.6}$$
$$=0.0111\,\text{K/W}$$

Substituting $R_{conv,1}$, $R_{glass,1}$, $R_{rad,2}$, $R_{glass,3}$ and $R_{conv,2}$ in equation (1), we get

$$\dot{Q}=\frac{T_{\infty1}-T_{\infty2}}{R_{conv,1}+R_{glass,1}+R_{rad,2}+R_{glass,3}+R_{conv,2}}$$
$$=\frac{21-(-5)}{0.02778+0.001068+0.0536+0.001068+0.0111}$$
$$=274.8\text{ W}$$

Hence, heat transfer through glass window is $\boxed{\dot{Q}=274.8\text{ W}}$

Step 8 of 12

The inner surface temperature of the glass window can be expressed as

$$\dot{Q}=\frac{T_{\infty1}-T_1}{R_{conv,1}}$$
$$274.8=\frac{294.15-T_1}{0.02778}$$
$$T_1=286.515\text{ K}$$
$$=13.36^\circ\text{C}$$

So, the $T_1=13.6^\circ\text{C}<T_2=18^\circ\text{C}$, it is impossible and in reality $T_1>T_2$. Therefore, the assumption of T_2 and T_3 are wrong. Hence the calculation needs to be repeated by re assuming inner surface temperatures of glasses are 12 and 1°C .

The resistance in the vacuum space can be determined as,

$$R_{rad,2}=\frac{1}{\varepsilon\sigma A(T_2^2+T_3^2)(T_2+T_3)}$$
$$=\frac{1}{1\times5.67\times10^{-8}\times3.6\times(285.15^2+274.15^2)\times(285.15+274.15)}$$
$$=0.056\text{ K/W}$$

Step 9 of 12

Therefore, the heat transfer through glass can be calculated as

$$\dot{Q}=\frac{T_{\infty1}-T_{\infty2}}{R_{conv,1}+R_{glass,1}+R_{rad,2}+R_{glass,3}+R_{conv,2}}$$
$$=\frac{21-(-5)}{0.02778+0.001068+0.056+0.001068+0.0111}$$
$$=268\text{ W}$$

Hence, the heat transfer through glass is $\boxed{\dot{Q}=268\,\text{W}}$

Step 10 of 12

The inner surface temperature of the glass window can be expressed as

$$\dot{Q}=\frac{T_{\infty1}-T_1}{R_{conv,1}}$$
$$268=\frac{294.15-T_1}{0.02778}$$
$$T_1=286.7\text{K}$$
$$=13.55^\circ\text{C}$$

Hence, the inner surface temperature of the glass window is $\boxed{T_1=13.55^\circ\text{C}}$

Step 11 of 12

Similarly the inner surface temperatures of glasses calculated as

$$\dot{Q}=\frac{T_1-T_2}{R_{glass,1}}$$
$$268=\frac{286.7-T_2}{0.001068}$$
$$T_2=286.4\text{K}$$
$$=13.26^\circ\text{C}$$

Hence, the inner surface temperature of the glass window and vacuum is $\boxed{T_2=13.26^\circ\text{C}}$

Step 12 of 12

$$\dot{Q}=\frac{T_2-T_3}{R_{glass,1}}$$
$$268=\frac{286.4-T_3}{0.056}$$
$$T_3=271.4\text{K}$$

$$\dot{Q} = \frac{T_3 - T_2}{R_{glass,1}}$$

$$268 = \frac{286.4 - T_3}{0.056}$$

$$T_3 = 271.4\text{K}$$

$$= -1.76°\text{C}$$

Hence, the outer surface temperature of the glass window and vacuum is $\boxed{T_2 = -1.76°\text{C}}$

Therefore the assumed values are very close to the calculated results.

Question: 17.020P

Reconsider Prob. 17–18. Using EES (or other) software, plot the rate of heat transfer through the window as a function of the width of air space in the range of 2 mm to 20 mm, assuming pure conduction through the air. Discuss the results.

Step 1 of 7

Given data:

Height of the glass window, $H = 1.5\,\text{m}$.

Width of the glass window, $W = 2.4\,\text{m}$.

Thickness of the glass, $L_1 = 0.003\,\text{m}$.

Distance between two glasses, $L_2 = 0.002\,\text{to}\,0.012\,\text{m}$.

Thermal conductivity of the glass, $k_1 = 0.78\,\text{W/m-K}$.

Thermal conductivity of the air, $k_2 = 0.026\,\text{W/m-K}$.

Temperature of the room, $T_{\infty 1} = 21^\circ\text{C}$.

Temperature of the outdoors, $T_{\infty 2} = -5^\circ\text{C}$.

Heat transfer coefficient on the inner surface of the window, $h_1 = 10\,\text{W/m}^2\text{-K}$.

Heat transfer coefficient on the outer surface of the window, $h_2 = 25\,\text{W/m}^2\text{-K}$.

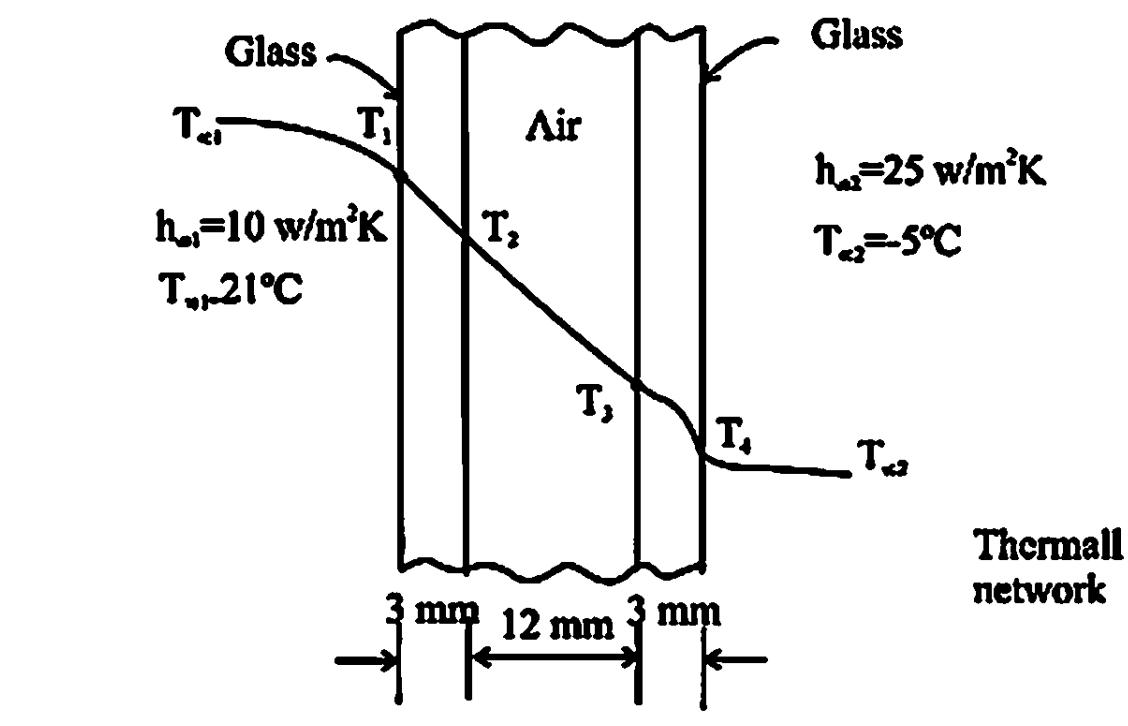
Step 2 of 7

Assumptions:

1. Heat transfer through the glass window is under steady state.
2. Thermal conductivity of the glass window is constant.
3. Heat transfer through the glass window is one dimensional and the gradients in width and height direction are negligible.
4. Heat transfer by radiation is negligible

Step 3 of 7

Diagram for thermal network:



Step 4 of 7

The heat transfer through the glass window can be determined using the thermal resistance network concept as follows,

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{conv,1} + R_{glass,1} + R_{air,2} + R_{glass,3} + R_{conv,2}}$$

The cross sectional area of the glass window can be calculated as

$$A = H \times W$$
$$= 1.5(\text{m}) \times 2.4(\text{m})$$
$$= 3.6\,\text{m}^2$$

Step 5 of 7

The individual resistances are calculated as follows

$$R_{conv,1} = \frac{1}{h_1 A}$$
$$= \frac{1}{10(\text{W/m}^2\text{-K}) \times 3.6(\text{m}^2)}$$
$$= 0.02778\,\text{K/W}$$

$$R_{glass,1} = \frac{L_1}{k_1 A}$$
$$= \frac{0.003(\text{m})}{0.78(\text{W/m-K}) \times 3.6(\text{m}^2)}$$
$$= 0.0010684\,\text{K/W}$$

$$R_{air,2} = \frac{L_2}{k_2 A}$$
$$= \frac{L_2(\text{m})}{0.026(\text{W/m-K}) \times 3.6(\text{m}^2)}$$
$$= 10.684 L_2\,\text{K/W}$$

$$R_{glass,3} = \frac{L_3}{k_3 A}$$
$$= \frac{0.003(\text{m})}{0.78(\text{W/m-K}) \times 3.6(\text{m}^2)}$$
$$= 0.0010684\,\text{K/W}$$

$$R_{conv,2} = \frac{1}{h_2 A}$$
$$= \frac{1}{25(\text{W/m}^2\text{-K}) \times 3.6(\text{m}^2)}$$
$$= 0.0111\,\text{K/W}$$

Step 6 of 7

Therefore, the heat transfer through glass window is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{conv,1} + R_{glass,1} + R_{air,2} + R_{glass,3} + R_{conv,2}}$$
$$= \frac{294.15(\text{K}) - 268.15(\text{K})}{0.02778(\text{K/W}) + 0.001068(\text{K/W}) + 10.684 L_2(\text{K/W}) + 0.001068(\text{K/W}) + 0.0111(\text{K/W})}$$

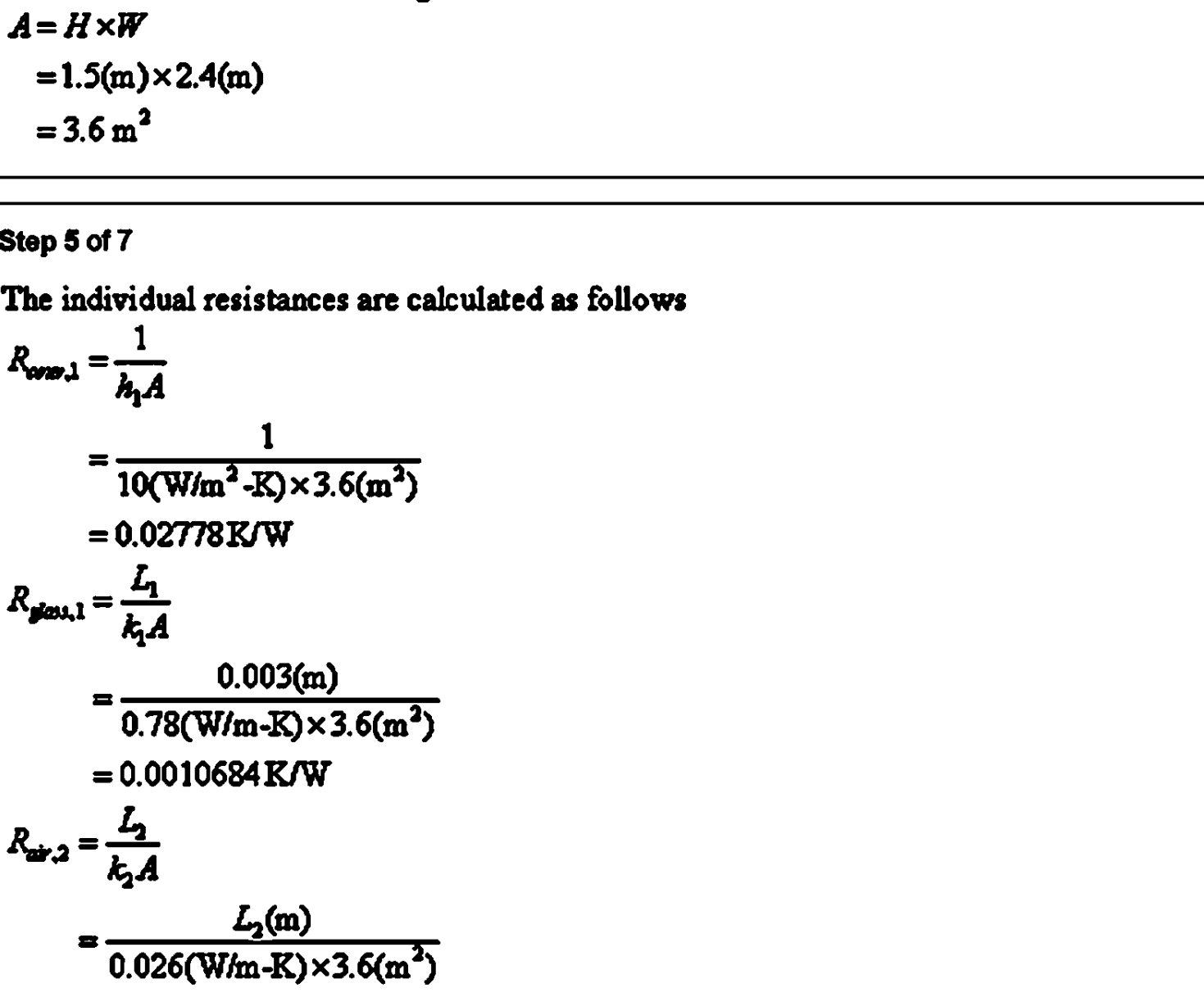
By varying the Length values from 0.002 to 0.02 by using the excel or EES software .

The problem is solved using the Microsoft Excel and the solution is given below

Lair(m)	Q(W)
0.002	416.7735
0.004	310.4403
0.006	247.3364
0.008	205.5531
0.01	175.8468
0.012	153.6425
0.014	136.4171
0.016	122.6647
0.018	111.4311
0.02	102.0825

Step 7 of 7

Grapgh between the heatvtrasfer rate and Width of the space :



Therefore, from the graphical plot we can infer that by increasing the width of the air space the net heat transfer rate through window glasses decreasing parabolically.

Question: 17.021P

Consider an electrically heated brick house (k = 0.40 Btu/h·ft·°F) whose walls are 9 ft high and 1 ft thick. Two of the walls of the house are 50 ft long and the others are 35 ft long. The house is maintained at 70°F at all times while the temperature of the outdoors varies. On a certain day, the temperature of the inner surface of the walls is measured to be at 55°F while the average temperature of the outer surface is observed to remain at 45°F during the day for 10 h and at 35°F at night for 14 h. Determine the amount of heat lost from the house that day. Also determine the cost of that heat loss to the home owner for an electricity price of $0.09/kWh.

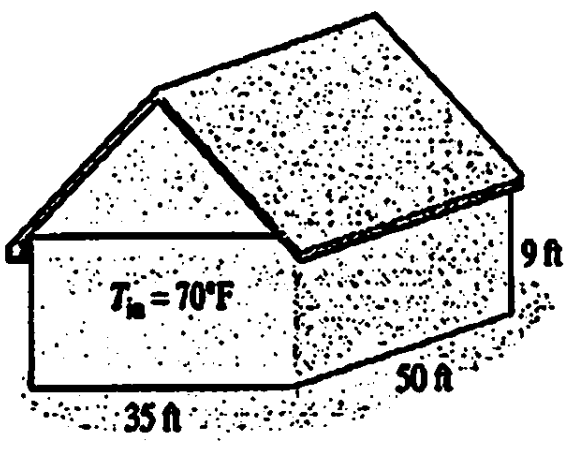
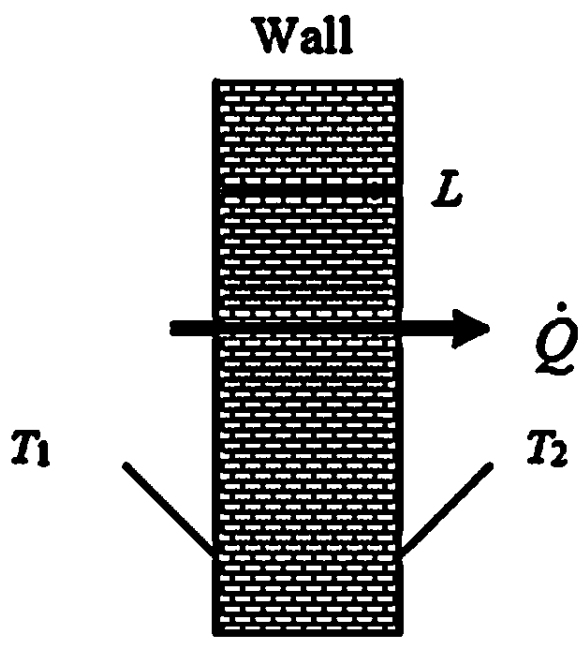


FIGURE P17–21E

Step 1 of 6

Draw a schematic diagram for steady state heat conduction through a wall.



Step 2 of 6

If heat transfer through the walls is steady since the surface temperatures of the walls remain constant at the specified values during the time period considered and heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors considering heat loss through the walls only.

Calculate the total heat transfer area.

$$A = 2(50\times9 + 35\times9)$$
$$= 1530\ \text{ft}^2$$

Step 3 of 6

Calculate the rate of heat loss during the day time using the following equation:

$$\dot{Q}_{day} = kA\frac{T_1 - T_2}{L}$$

Here, k is the thermal conductivity, A is the heat transfer area, T_1 and T_2 are temperatures at inner and outer wall surfaces and L is thickness of the wall.

Substitute 0.40 Btu/h·ft·°F for k, 1530 ft² for A, 55°F for T_1, 45°F for T_2 and 1 ft for L.

$$\dot{Q}_{day} = (0.40\ \text{Btu/h}\cdot\text{ft}\cdot\text{°F})(1530\ \text{ft}^2)\frac{(55-45)\text{°F}}{1\ \text{ft}}$$
$$= 6120\ \text{Btu/h}$$

Step 4 of 6

Calculate the rate of heat loss during night time as follows:

$$\dot{Q}_{night} = kA\frac{T_1 - T_2}{L}$$

Substitute 0.40 Btu/h·ft·°F for k, 1530 ft² for A, 55°F for T_1, 45°F for T_2 and 1 ft for L.

$$\dot{Q}_{night} = (0.40\ \text{Btu/h}\cdot\text{ft}\cdot\text{°F})(1530\ \text{ft}^2)\frac{(55-35)\text{°F}}{1\ \text{ft}}$$
$$= 12240\ \text{Btu/h}$$

Step 5 of 6

Calculate the total amount of heat loss from the house that day.

$$\dot{Q} = \frac{Q}{\Delta t}$$
$$Q = \dot{Q}\Delta t$$
$$= 10\dot{Q}_{day} + 14\dot{Q}_{night}$$

Substitute 6120 Btu/h for $\dot{Q}_{day}$ and 12240 Btu/h for $\dot{Q}_{night}$.

$$Q = (10\ \text{h})(6120\ \text{Btu/h}) + (14\ \text{h})(12240\ \text{Btu/h})$$
$$= 232560\ \text{Btu}$$

Therefore, The total amount of heat loss from the house that day is **232560 Btu**.

Step 6 of 6

Calculate the cost of this heat loss for that day as follows:

$$\text{Cost} = \left(232560\ \text{Btu}\times\frac{0.000293071\ \text{kWh}}{1\ \text{Btu}}\right)\times\left(\frac{\$0.09}{\text{kWh}}\right)$$
$$= \$6.134$$

Therefore, the electricity cost of that heat loss is **$6.134**.

Question: 17.022P

A cylindrical resistor element on a circuit board dissipates 0.15 W of power in an environment at 35°C. The resistor is 1.2 cm long, and has a diameter of 0.3 cm. Assuming heat to be transferred uniformly from all surfaces, determine (*a*) the amount of heat this resistor dissipates during a 24-h period; (*b*) the heat flux on the surface of the resistor, in W/m2; and (*c*) the surface temperature of the resistor for a combined convection and radiation heat transfer coefficient of 9 W/m2·K.

Step 1 of 5

Given data:

Power dissipates $\dot{Q} = 0.15\text{W}$

Environment temperature $T = 35°\text{C}$

Resistor length,

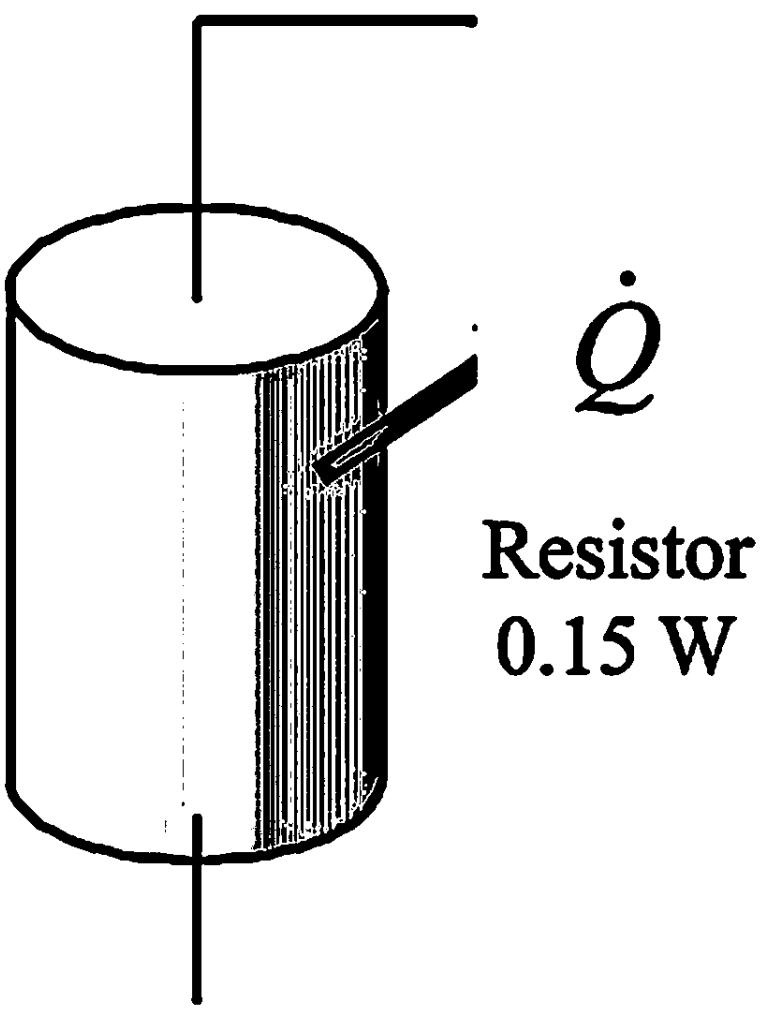
$$\begin{aligned} L &= 1.2\text{ cm} \\ &= \frac{1.2}{100}\text{ m} \\ &= 0.012\text{ m} \end{aligned}$$

Resistor diameter,

$$\begin{aligned} D &= 0.3\text{ cm} \\ &= \frac{0.3}{100}\text{ m} \\ &= 0.003\text{ m} \end{aligned}$$

Step 2 of 5

Diagram:



Step 3 of 5

If steady operating conditions exist and heat is transferred uniformly from all surfaces of the resistor then

(*a*)

Time period, $\Delta t = 24\text{ hr}$

The amount of heat this resistor dissipates during a 24-hour period is

$$\begin{aligned} Q &= \dot{Q}\Delta t \\ &= (0.15 \times 24) \\ &= 3.6\text{ Wh} \end{aligned}$$

Hence, the amount of heat this resistor dissipates during a 24-hour period is $\boxed{Q = 3.6\text{ Wh}}$

(*b*)

The heat flux on the surface of the resistor is

$$\begin{aligned} A_s &= 2\frac{\pi D^2}{4} + \pi DL \\ &= 2\frac{\pi(0.003\text{ m})^2}{4} + \pi(0.003\text{ m})(0.012\text{ m}) \\ &= 0.000127\text{ m}^2 \end{aligned}$$

Step 4 of 5

Heat flux on the surface of the transistor,

$$\begin{aligned} \dot{q} &= \frac{\dot{Q}}{A_s} \\ &= \frac{0.15\text{ W}}{0.000127\text{ m}^2} \\ &= 1179\text{ W/m}^2 \end{aligned}$$

Hence, Heat flux on the surface of the transistor is $\boxed{\dot{q} = 1181\text{ W/m}^2}$

Step 5 of 5

(*c*)

The surface temperature of the resistor can be determined from

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$\begin{aligned} T_s &= T_\infty + \frac{\dot{Q}}{hA_s} \\ &= 35 + \frac{0.15}{(9)(0.000127)} \\ &= 166°\text{C} \end{aligned}$$

Hence, the surface temperature of the resistor is $\boxed{T_s = 166°\text{C}}$

Question: 17.023P

To defog the rear window of an automobile, a very thin transparent heating element is attached to the inner surface of the window. A uniform heat flux of 1300 W/m2 is providedto the heating element for defogging a rear window with thickness of 5 mm. The interior temperature of the automobile is22°C and the convection heat transfer coefficient is 15 W/m2·K The outside ambient temperature is −5°C and the convection heat transfer coefficient is 100 W/m2·K. If the thermal conductivity of the window is 1.2 W/m·K, determine the inner surface temperature of the window.

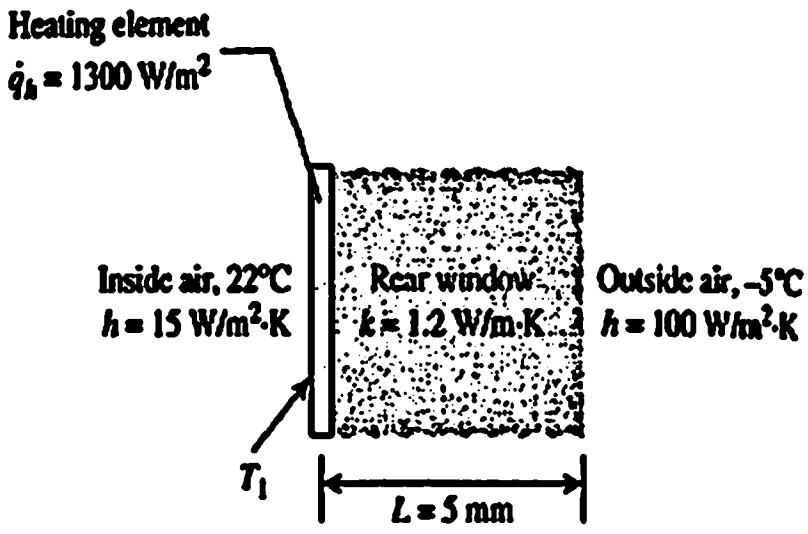


FIGURE P17–23

Step 1 of 3

Apply equilibrium equation to the system under steady state condition. The amount of internal heat generation in the heating element is equal to the amount of heat convection either of the heating element.

$$q'' = q_{conv,i} + q_{conv,o}$$
$$q'' = h_{conv,i}\left(T_1 - T_{\infty,i}\right) + h_{conv,o}\left(T_2 - T_{\infty,o}\right)$$

Here, $h_{conv,i}$ is the heat transfer coefficient of inside air, $h_{conv,o}$ is the heat transfer coefficient of outside air, $T_{\infty,i}$ is the temperature of inside air, $T_{\infty,o}$ is the temperature of outside air, T_1 is inner surface of the window, and T_2 is outer surface of the window.

Substitute $15\ W/m^2.K$ for $h_{conv,i}$, $100\ W/m^2.K$ for $h_{conv,o}$, $22°C$ for $T_{\infty,i}$, $5°C$ for $T_{\infty,o}$, and

$1300\ W/m^2$ for q''

$$1300 = 15\left(T_1 - 22\right) + 100\left(T_2 - (-5)\right)$$
$$1300 = 15\left(T_1 - 22\right) + 100\left(T_2 + 5\right)$$

$$15T_1 + 100T_2 = 1130 \quad \text{..... (1)}$$

Step 2 of 3

The heat transfer due to conduction through the glass is equal to the heat transfer due to convection out of the glass.

$$q_{cond} = q_{conv}$$
$$\frac{k\left(T_1 - T_2\right)}{L} = h_{conv,o}\left(T_2 - T_{\infty}\right)$$

Here, k is thermal conductivity of the rear window and L is thickness of the window.

Substitute $1.2\ W/m.K$ for k, 0.005 m for L, $100\ W/m^2.K$ for $h_{conv,o}$, and $5°C$ for $T_{\infty,o}$

$$\frac{1.2\left(T_1 - T_2\right)}{0.005} = 100\left(T_2 + 5\right)$$
$$T_1 - T_2 = 0.4167\left(T_2 + 5\right)$$

$$T_1 = T_2 + 0.4167\left(T_2 + 5\right) \quad \text{..... (2)}$$

Step 3 of 3

Solve equations (1) and (2)

Substitute $T_2 + 0.4167\left(T_2 + 5\right)$ for T_1 in equation (1)

$$15T_1 + 100T_2 = 1130$$
$$15\left(T_2 + 0.4167\left(T_2 + 5\right)\right) + 100T_2 = 1130$$
$$15T_2 + 6.2505T_2 + 31.252 + 100T_2 = 1130$$
$$T_2 = 9.0618°C$$

Substitute $9.0618°C$ for T_2 in equation (1).

$$15T_1 + 100T_2 = 1130$$
$$15T_1 + 100 \times 9.0618 = 1130$$
$$T_1 = 14.9213°C$$

Therefore, the temperature of the inner surface of the window is $\boxed{14.9213°C}$

Question: 17.024P

Consider a power transistor that dissipates 0.15 W of power in an environment at 30°C. The transistor is 0.4 cm long and has a diameter of 0.5 cm. Assuming heat to be transferred uniformly from all surfaces, determine (*a*) the amount of heat this resistor dissipates during a 24-h period, in kWh; (*b*) the heat flux on the surface of the transistor, in W/m2; and (*c*) the surface temperature of the resistor for a combined convection and radiation heat transfer coefficient of 18 W/m2·K.

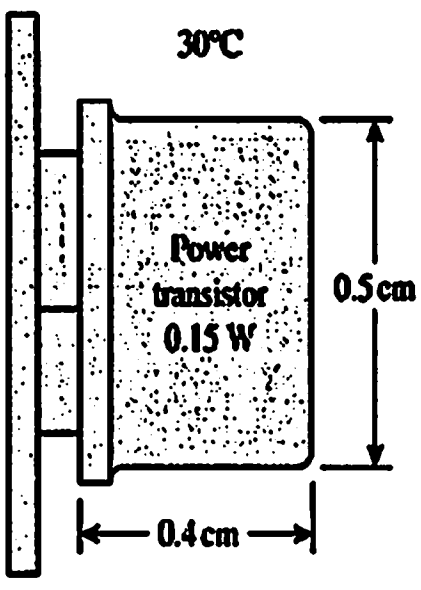


FIGURE P17–24

Step 1 of 5

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Step 2 of 5

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Step 3 of 5

a) The amount of heat dissipated by transistor during 24-h period can be determined as

$$Q = \dot{Q} \times t$$
$$= 0.15(\text{W}) \times 24(\text{h})$$
$$= 3.6\,\text{Wh}$$

Therefore amount of heat transferred is $\boxed{Q = 3.6\,\text{Wh}}$

Step 4 of 5

b) The heat flux on the surface of the transistor can be determined as follows,

$$\dot{q} = \frac{\dot{Q}}{A_s}$$

Where A_s is the surface area of the transistor from which heat is dissipating to the ambient air and it is calculated as

$$A_s = 2\frac{\pi D^2}{4} + \pi DL$$
$$= 2 \times \frac{\pi \times 0.005^2(\text{m}^2)}{4} + \pi \times 0.005(\text{m}) \times 0.004(\text{m})$$
$$= 0.0001021\text{m}^2$$

Therefore, the heat flux calculated as,

$$\dot{q} = \frac{0.15(\text{W})}{0.0001021(\text{m}^2)}$$
$$= 1469.12\,\text{W/m}^2$$

$\boxed{\dot{q} = 1469.12\,\text{W/m}^2}$

Step 5 of 5

The surface temperature of the transistor can be determined by applying the energy balance as follows,

$$\dot{E}_{in} + \dot{Q}_{gen} - \dot{E}_{out} = 0$$

$$0 + \dot{Q}_{gen} - h_{conv}A_s(T_s - T_\infty) - h_{rad}A_s(T_s - T_\infty) = 0$$

$$\dot{Q}_{gen} = (h_{conv} + h_{rad})A_s(T_s - T_\infty)$$
$$0.15(\text{W}) = 18(\text{W/m}^2\text{-K}) \times 0.0001021(\text{m}^2) \times (T_s - 303.15)$$
$$T_s = 384.77\,\text{K}$$
$$= 111.62°\text{C}$$

Hence, The surface temperature of the transistor is $\boxed{T_s = 111.62°\text{C}}$.

Question: 17.025P

A 12-cm × 18-cm circuit board houses on its surface 100 closely spaced logic chips, each dissipating 0.06 W in an environment at 40°C. The heat transfer from the back surface of the board is negligible. If the heat transfer coefficient on the surface of the board is 10 W/m2·K, determine (*a*) the heat flux on the surface of the circuit board, in W/m2; (*b*) the surface temperature of the chips; and (*c*) the thermal resistance between the surface of the circuit board and the cooling medium, in °C/W.

Step 1 of 5

Given data:

Area of circuit board $A_s = 12\text{cm} \times 18\text{cm}$

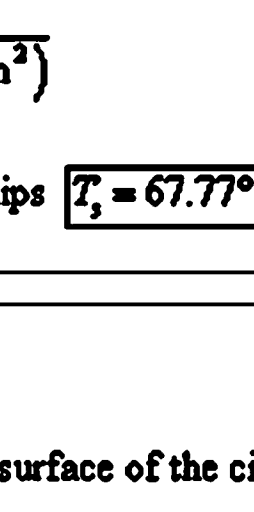
Dissipated heat $Q = 0.06\ \text{W}$

No of chips $n = 100$

Environment temperature $T_\infty = 40°\text{C}$

Surface heat transfer coefficient of the board $h = 10\text{W/m}^2 \cdot °\text{C}$

Step 2 of 5



Step 3 of 5

(*a*)

If steady operating conditions exist and heat transfer from the back surface of the board is negligible then

The heat flux on the surface of the circuit board is calculated by

$$A_s = 0.12\text{m} \times 0.18\text{m}$$

$$A_s = 0.0216\text{m}^2$$

$$\dot{q} = \frac{\dot{Q}}{A_s}$$

$$= \frac{(100 \times 0.06)\ \text{W}}{0.0216\ \text{m}^2}$$

$$= 277.77\ \text{W/m}^2$$

∴ The heat flux $\boxed{\dot{q} = 277.77\ \text{W/m}^2}$

Step 4 of 5

(*b*)

The surface temperature of the chips is

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$T_s = T_\infty + \frac{\dot{Q}}{hA_s}$$

$$= 40°\text{C} + \frac{(100 \times 0.06)\text{W}}{(10\ \text{W/m}^2.°\text{C})(0.0216\ \text{m}^2)}$$

$$= 67.77°\text{C}$$

∴ The surface temperature of the chips $\boxed{T_s = 67.77°\text{C}}$

Step 5 of 5

(*c*)

The thermal resistance between the surface of the circuit board and the cooling medium is calculated by

$$R_{conv} = \frac{1}{hA_s}$$

$$= \frac{1}{(10\ \text{W/m}^2 \cdot °\text{C})(0.0216\ \text{m}^2)}$$

$$= 4.629°\text{C/W}$$

∴ The thermal resistance $\boxed{R_{conv} = 4.629°\text{C/W}}$

Question: 17.026P

Consider a person standing in a room at 20°C with an exposed surface area of 1.7 m2. The deep body temperature of the human body is 37°C, and the thermal conductivity of the human tissue near the skin is about 0.3 W/m·K. The body is losing heat at a rate of 150 W by natural convection and radiation to the surroundings. Taking the body temperature 0.5 cm beneath the skin to be 37°C, determine the skin temperature of the person.

Step 1 of 2

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Step 2 of 2

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Question: 17.027P

A 1.0 m × 1.5 m double-pane window consists of two 4-mm-thick layers of glass (k= 0.78 W/m·K) that are the separated by a 5-mm air gap ($k$air = 0.025 W/m·K). The heat flow through the air gap is assumed to be by conduction. The inside and outside air temperatures are 20°C and −20°C, respectively, and the inside and outside heat transfer coefficients are 40 and 20 W/m2·K. Determine (*a*) the daily rate of heat loss through the window in steady operation and (*b*) the temperature difference across the largest thermal resistence.

Step 1 of 8

Given that:

Area of the window, $A = 1.0\text{m} \times 1.5\text{m}$

Thickness of glass layers, $L_1 = L_2 = 4\text{mm}$

Thickness of air gap, $L_3 = 5\text{mm}$

The thermal conductivity of the glass, $k_{glass} = 0.78\ \text{W/m} \cdot ^\circ\text{C}$

The thermal conductivity of the air, $k_{air} = 0.025\ \text{W/m} \cdot ^\circ\text{C}$

Inside temperature of the room, $T_{\infty 1} = 20^\circ\text{C}$

Outside temperature of the room, $T_{\infty 2} = 20^\circ\text{C}$

Step 2 of 8

(a) Consider heat transfer through the window is steady since the surface temperatures remain constant at the specified values and heat transfer by radiation is negligible.

The area of the window

$A = 1.0\ \text{m} \times 1.5\ \text{m}$

$A = 1.5\ \text{m}^2$

Step 3 of 8

The individual resistances are

$R_1 = R_{conv,1}$

$$R_{conv,1} = \frac{1}{h_1 A}$$

$$R_1 = \frac{1}{(40\ \text{W/m}^2 \cdot ^\circ\text{C})(1.5\ \text{m}^2)}$$

$R_1 = 0.01667^\circ\text{C/W}$

Step 4 of 8

$$R_{glass1} = \frac{L_1}{k_{glass} A}$$

$$R_{glass1} = \frac{0.004\ \text{m}}{0.78\ \text{W/m} \cdot ^\circ\text{C} \times 1.5}$$

$R_{glass1} = 0.00342\ ^\circ\text{C/W}$

$$R_{glass2} = \frac{L_2}{k_{glass} A}$$

$$R_{glass2} = \frac{0.004\ \text{m}}{0.78\ \text{W/m} \cdot ^\circ\text{C} \times 1.5\ \text{m}^2}$$

$R_{glass2} = 0.00342\ ^\circ\text{C/W}$

Step 5 of 8

$$R_{air} = \frac{L_{air}}{k_{air} A}$$

$$R_{air} = \frac{0.005\ \text{m}}{0.025\text{W/m} \cdot ^\circ\text{C} \times 1.5\ \text{m}^2}$$

$R_{air} = 0.133\ ^\circ\text{C/W}$

$R_2 = R_{conv,2}$

$$R_2 = \frac{1}{h_2 A}$$

$$R_2 = \frac{1}{20\text{W/m}^2 \cdot ^\circ\text{C} \times 1.5\ \text{m}^2}$$

$R_2 = 0.0333\ ^\circ\text{C/W}$

Step 6 of 8

$R_{total} = R_{conv,1} + R_{glass} + R_{air} + R_{glass} + R_{conv,2}$

$R_{total} = 0.01667 + 0.00342 + 0.00342 + 0.133 + 0.0333$

$R_{total} = 0.18981^\circ\text{C/W}$

Step 7 of 8

The steady rate of heat transfer through window glass is then

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$$

$$\dot{Q} = \frac{[20 - (-20)]^\circ\text{C}}{0.18981^\circ\text{C/W}}$$

$\dot{Q} = 210W$

∴ The heat transfer is $\boxed{\dot{Q} = 210W}$

Step 8 of 8

(b) Temperature difference across greatest thermal resistance can be determined by using thermal resistance of air which is maximum

$$\dot{Q} = \frac{\Delta t}{R_{air}}$$

$\Delta t = \dot{Q} R_{conv,1}$

$\Delta t = (210\ \text{W})(0.133\ ^\circ\text{C/W})$

$\Delta t = 28^\circ\text{C}$

∴ Temperature difference across greatest thermal resistance is $\boxed{\Delta t = 28^\circ\text{C}}$

Question: 17.028P

A wall is constructed of two layers of 0.6-in-thick sheetrock (k= 0.10 Btu/h·ft·°F), which is a plasterboard made of two layers of heavy paper separated by a layer of gypsum, placed 7 in apart. The space between the sheetrocks is filled with fiberglass insulation (k = 0.020 Btu/h·ft·°F), Determine (*a*) the thermal resistance of the wall and (*b*) its *R*-value of insulation in English units.

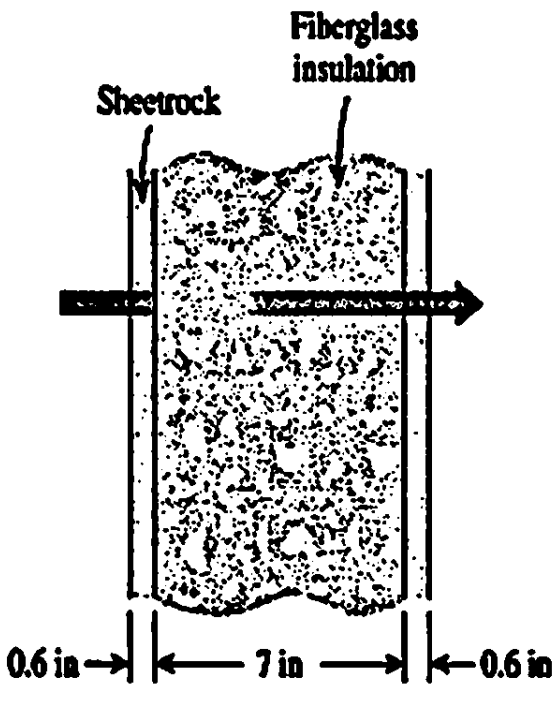
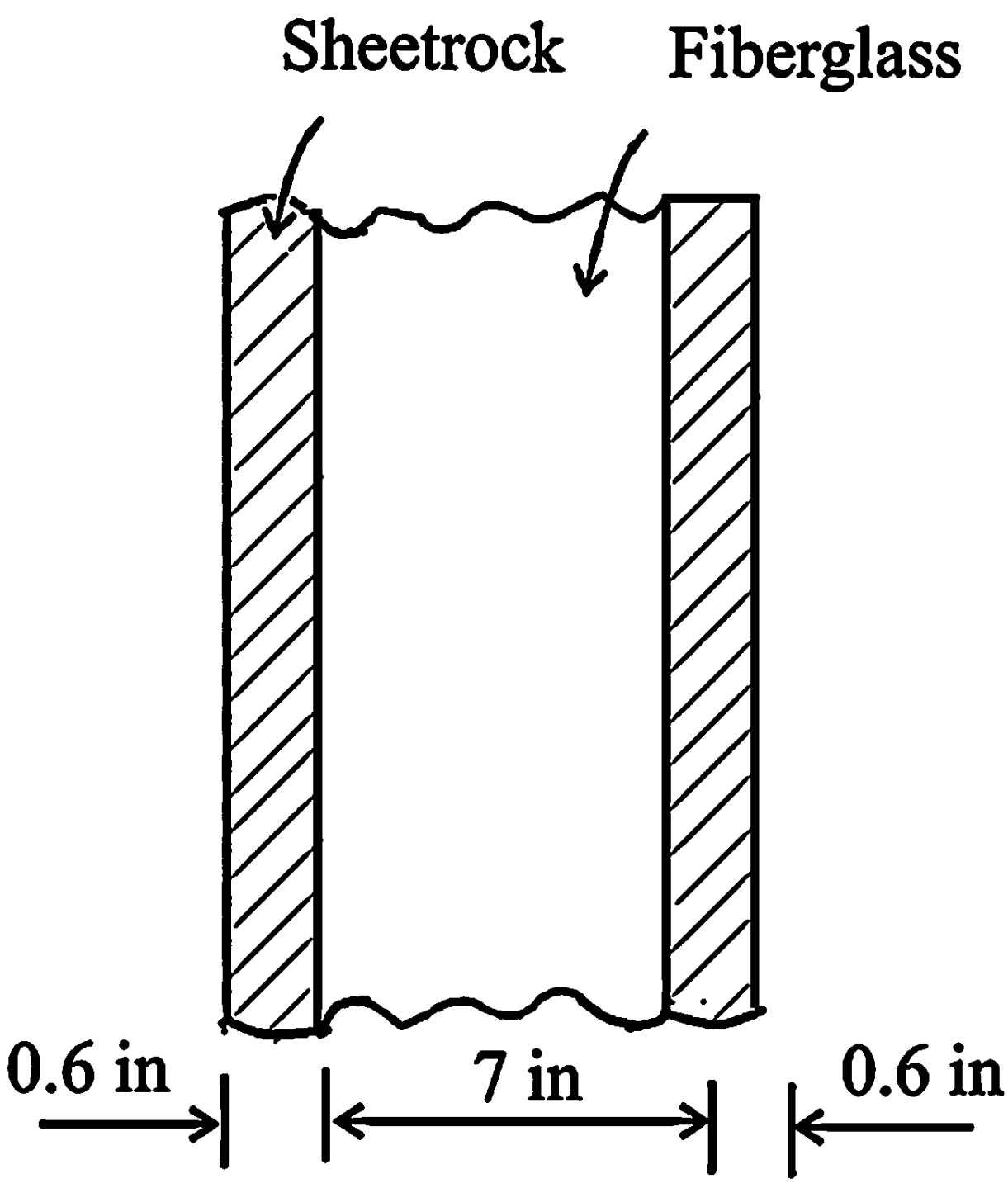


FIGURE P17–28E

Step 1 of 3

Schematic Diagram of construction of wall thickness:



Step 2 of 3

(a)

Calculate the thermal resistance of the wall

$$R_1 = \frac{L_1}{k_1}$$

Here,

Wall thickness is L_1 and L_3 respectively.

Thermal conductivity of the wall is k_1

Use the conversion factor to convert $1\,\text{in.}\,\text{to}\,\text{ft}$ of

$$1\,\text{ft} = 12\,\text{in}$$

Substitute $0.6\,\text{in.}\,\text{for}\,L_1, 0.10\,\text{Btu/h-ft}\cdot{}^\circ\text{F}\,\text{for}\,k_1$

$$R_1 = \frac{L_1}{k_1}$$
$$= \frac{0.6\,\text{in.}\times 1\,\text{ft}}{0.1\,\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}\times 12\,\text{in.}}$$
$$= 0.5{}^\circ\text{F}\cdot\text{h/Btu}$$

Therefore, the thermal resistance of the wall is $\boxed{0.5{}^\circ\text{F}\cdot\text{h/Btu}}$

Step 3 of 3

(b)

Calculate the *R*-value of the insulation is

$$R_2 = \frac{L_2}{k_2}$$

Here,

Thermal conductivity of the insulation is k_2

Thickness of the fiber glass Insulation is L_2

Use the conversion factor to convert $1\,\text{in.}\,\text{to}\,\text{ft}$ of

$$1\,\text{ft} = 12\,\text{in}$$

Substitute $7\,\text{in.}\,\text{for}\,L_2, 0.020\,\text{Btu/h-ft}\cdot{}^\circ\text{F}\,\text{for}\,k_2$

$$R_2 = \frac{L_2}{k_2}$$
$$= \frac{7\,\text{in.}\times 1\,\text{ft}}{0.020\,\text{Btu/hft}\cdot{}^\circ\text{F}\times 12\,\text{in.}}$$
$$= 29.167{}^\circ\text{F}\cdot\text{h/Btu}$$

Therefore, the insulation is *R*- value wall in English units is $\boxed{29.167{}^\circ\text{F}\cdot\text{h/Btu}}$

Question: 17.029P

The roof of a house consists of a 15-cm-thick concrete slab (k =2 W/m·K) that is 15 m wide and 20 m long. The convection heat transfer coefficients on the inner and outersurfaces of the roof are 5 and 12 W/m2·K, respectively. On a clear winter night, the ambient air is reported to be at 10°C, while the night sky temperature is 100 K. The house and the interior surfaces of the wall are maintained at a constant temperature of 20°C. The emissivity of both surfaces of the concrete roof is 0.9. Considering both radiation and convection heat transfers, determine the rate of heat transfer through the roof, and the inner surface temperature of the roof.

If the house is heated by a furnace burning natural gas with an efficiency of 80 percent, and the price of natural gas is $1.20/therm (1 therm = 105,500 kJ of energy content), determine the money lost through the roof that night during a 14-h period.

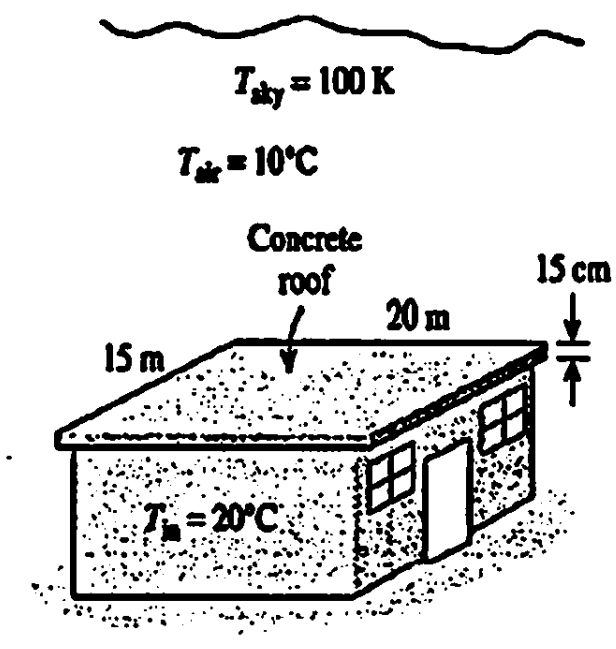


FIGURE P17–29

Step 1 of 9

Draw the schematic diagram of the house.

Step 2 of 9

Step 3 of 9

If steady operating conditions exist and the emissivity, thermal conductivity of the roof are constant and in steady operation, heat transfer from the room to the roof (by convection and radiation) must be equal to the heat transfer from the roof to the surroundings (by convection and radiation), that must be equal to the heat transfer through the roof by conduction that is,

$$\dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{room to roof, conv}} + \dot{Q}_{\text{rad}} = \dot{Q}_{\text{roof to surroundings, conv}} + \dot{Q}_{\text{rad}}$$

Step 4 of 9

Taking the inner and outer surface temperatures of the roof to be $T_{s,in}$ and $T_{s,out}$, respectively, the quantities above can be expressed as

$$\dot{Q}_{\text{room to roof, conv}} + \dot{Q}_{\text{rad}}$$
$$= h_i A\left(T_{room} - T_{s,in}\right) + \varepsilon A\sigma\left(T_{room}^4 - T_{s,in}^4\right)$$

Substitute $5\,\text{W/m}^2\cdot{}^\circ\text{C}$ for h_i , $300\,\text{m}^2$ for A , 20°C for T_{room} , $5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ for σ , 0.9 for ε , $293\,\text{K}$ for T_{room} in the terms containing fourth power of temperature and $\left(T_{s,in} + 273\,\text{K}\right)$ for $T_{s,in}$ in the terms containing fourth power of temperature.

$$= \left(5\ \text{W/m}^2\cdot{}^\circ\text{C}\right)\left(300\ \text{m}^2\right)\left(20 - T_{s,in}\right)^\circ\text{C}$$
$$+ (0.9)\left(300\ \text{m}^2\right)\left(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4\right)\left[(293\,\text{K})^4 - \left(T_{s,in} + 273\,\text{K}\right)^4\right]$$

$$\dot{Q}_{\text{room to roof, conv}} + \dot{Q}_{\text{rad}} = \left[1500\times\left(20 - T_{s,in}\right)\right]$$
$$+ \left(1.5309\times10^{-5}\left[(20+273\,\text{K})^4 - \left(T_{s,in} + 273\,\text{K}\right)^4\right]\right) \quad \text{..... (1)}$$

$$\dot{Q}_{\text{roof, cond}} = kA\frac{T_{s,in} - T_{s,out}}{L}$$

$$= \left(2\ \text{W/m}\cdot{}^\circ\text{C}\right)\left(300\ \text{m}^2\right)\frac{T_{s,in} - T_{s,out}}{0.15\ \text{m}}$$

$$= 4000\left(T_{s,in} - T_{s,out}\right)\ \text{W} \quad \text{..... (2)}$$

Step 5 of 9

Calculate the rate of heat transfer from room to surroundings by using the equation,

$$\dot{Q}_{\text{roof to surr, conv+rad}} = h_o A\left(T_{s,out} - T_{surr}\right) + \varepsilon A\sigma\left(T_{s,out}^4 - T_{sky}^4\right)$$

Substitute $12\ \text{W/m}^2\cdot{}^\circ\text{C}$ for h_0 , $300\,\text{m}^2$ for A , 0.9 for ε , $5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ for σ ,

10°C for T_{surr} , $\left(T_{s,out} + 273\,\text{K}\right)$ for $T_{s,out}$ in the terms containing fourth power of temperature and $100\,\text{K}$ for T_{sky} .

$$\dot{Q}_{\text{roof to surr, conv+rad}} = \left(12\ \text{W/m}^2\cdot{}^\circ\text{C}\right)\left(300\ \text{m}^2\right)\left(T_{s,out} - 10\right)^\circ\text{C}$$
$$+ 0.9\left(300\ \text{m}^2\right)\left(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4\right)\left[\left(T_{s,out} + 273\,\text{K}\right)^4 - (100\,\text{K})^4\right]$$

$$\dot{Q}_{\text{roof to surr, conv+rad}} = 3600\left(T_{s,out} - 10\right)^\circ\text{C} + 1.5309\times10^{-5}\left[\left(T_{s,out} + 273\,\text{K}\right)^4 - (100\,\text{K})^4\right] \quad \text{..... (3)}$$

Step 6 of 9

Equating equations (1) and (2)

$$4000\left(T_{s,in} - T_{s,out}\right) = 1500\left(20 - T_{s,in}\right)^\circ\text{C} + 1.5309\times10^{-5}\left[7.37\times10^9 - \left(T_{s,in} + 273\,\text{K}\right)^4\right]$$

$$4000\left(T_{s,in} - T_{s,out}\right) = 30000 - 1500T_{s,in} + 112827.33 - 1.5309\times10^{-5}\left(T_{s,in} + 273\,\text{K}\right)^4 \quad \text{..... (4)}$$

Equating equations (2) and (3)

$$4000\left(T_{s,in} - T_{s,out}\right) = 3600\left(T_{s,out} - 10\right)^\circ\text{C} + 1.5309\times10^{-5}\left[\left(T_{s,out} + 273\,\text{K}\right)^4 - (100\,\text{K})^4\right] \quad \text{..... (5)}$$

Step 7 of 9

Solving the equations (4) and (5) we get

$$T_{s,in} = 7.3^\circ\text{C}$$

$$T_{s,out} = -2.1^\circ\text{C}$$

Substituting values in the equation (2) we get

$$\dot{Q} = 37600\text{W}$$

Step 8 of 9

Calculate the total amount of natural gas consumption during a period by using the equation,

$$Q_{gas} = \frac{Q_{input}}{0.80}$$

$$= \frac{\dot{Q}\Delta t}{0.80}$$

Substitute $37.6\,\text{kW}$ for $\dot{Q}$ and $(14\times3600)\text{sec}$ for Δt .

$$Q_{gas} = \frac{(37.600\ \text{kJ/s})(14\times3600\ \text{s})}{0.80}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right)$$
$$= 22.453\ \text{therms}$$

Step 9 of 9

Calculate the money lost through the roof during that period is

$$\text{Money lost} = (22.453\ \text{therms})(\$1.20/\text{therm})$$
$$= 26.943$$

Therefore the money lost through the roof during that period is $\boxed{\$26.943}$.

Question: 17.030P

A 2-m × 1.5-m section of wall of an industrial furnace burning natural gas is not insulated, and the temperature at the outer surface of this section is measured to be 110°C. The temperature of the furnace room is 32°C, and the combined convection and radiation heat transfer coefficient at the surface of the outer furnace is 10 W/m2·K. It is proposed to insulate this section of the furnace wall with glass wool insulation (k = 0.038 W/m·K) in order to reduce the heat loss by 90 percent, Assuming the outer surface temperature of the metal section still remains at about 110°C, determine the thickness of the insulation that needs to be used.

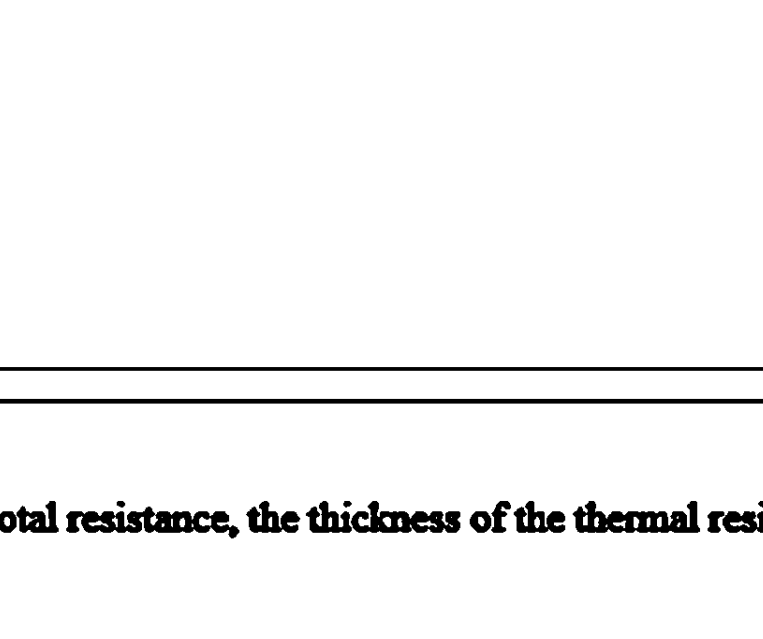
The furnace operates continuously and has an efficiency of 78 percent. The price of the natural gas is $1.10/therm (1 therm = 105,500 kJ of energy content). If the installation of the insulation will cost $250 for materials and labor, determine how long it will take for the insulation to pay for itself from the energy it saves.

Step 1 of 8

Loading solution. Please wait ...

Step 2 of 8

Diagram for thermal network:



Step 3 of 8

Without having any insulation the rate of heat transfer can be determined as

$$\dot{Q} = hA(T_s - T_\infty)$$
$$= 10 \times 3 \times (110 - 32)$$
$$= 2340\,\text{W}$$

Step 4 of 8

In order to reduce heat loss by 90%, the new heat transfer rate calculated as

$$\dot{Q} = 0.1 \times 2340\,(\text{W})$$
$$= 234\,\text{W}$$

The thermal resistance calculated as

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$
$$R_{total} = \frac{\Delta T}{\dot{Q}}$$
$$= \frac{(110 - 32)}{234}$$
$$= 0.3333\,\text{K/W}$$

Step 5 of 8

In order to have this total resistance, the thickness of the thermal resistance must be,

$$R_{total} = \frac{1}{hA} + \frac{L}{kA}$$
$$L = kA\left(R_{total} - \frac{1}{hA}\right)$$
$$L = 0.038 \times 3 \times \left(0.3333 - \frac{1}{10 \times 3}\right)$$
$$= 0.0342\,\text{m}$$

Therefore, thickness of the insulation is $\boxed{L = 0.0342\,\text{m}}$

Step 6 of 8

The amount of heat transfer rate saved by the insulation is 90% of the total heat transfer rate, therefore

$$\dot{Q} = 0.9 \times 2340\,(\text{W})$$
$$= 2106\,\text{W}$$
$$= 2.106\,\text{kJ/s}$$
$$= 2.106 \times 3600\,\text{kJ/h}$$
$$= 7851.6\,\text{kJ/h}$$

Therefore, the rate of heat transfer is $\boxed{\dot{Q} = 7851.6\,\text{kJ/h}}$

Step 7 of 8

Since the furnace operates continuously, the number of hours per year is $365 \times 24 = 8760\,\text{h/year}$ and the efficiency if the furnace is 78%, the amount of energy saved per year can be determined as

$$\text{Energy saved per year} = 7851.6\,(\text{kJ/h}) \times 8760\,(\text{h})$$
$$= 68780016\,\text{kJ}$$

Amount of energy saved in terms of therms can be calculated as

$$1\,\text{therm} = 105500\,\text{kJ}$$
$$\text{Energy saved per year} = 68780016\,(\text{kJ}) \times \frac{1\,\text{therm}}{105500\,(\text{kJ})}$$
$$= 652\,\text{therms}$$

Therefore, the money saved can be determined as

$$\text{money saved} = \text{Energy saved} \times \text{cost of energy}$$
$$= 652\,(\text{therms}) \times 1.1\,(\$/\text{therm})$$
$$= \$717.2\,\text{per year}$$

Therefore, amount of money saved is $\boxed{\text{money saved} = \$717.2\,\text{per year}}$

Step 8 of 8

cost of the insulation = $250

The payback period can be calculated as,

$$\text{Payback period}\,(t) = \frac{\text{insulation cost}}{\text{Money saved}}$$
$$= \frac{\$250}{\$717.2\,(\text{per year})}$$
$$= 0.35\,\text{years}$$
$$= 4.2\,\text{months}$$

Therefore, payback period is $\boxed{t = 4.2\,\text{months}}$

Question: 17.031P

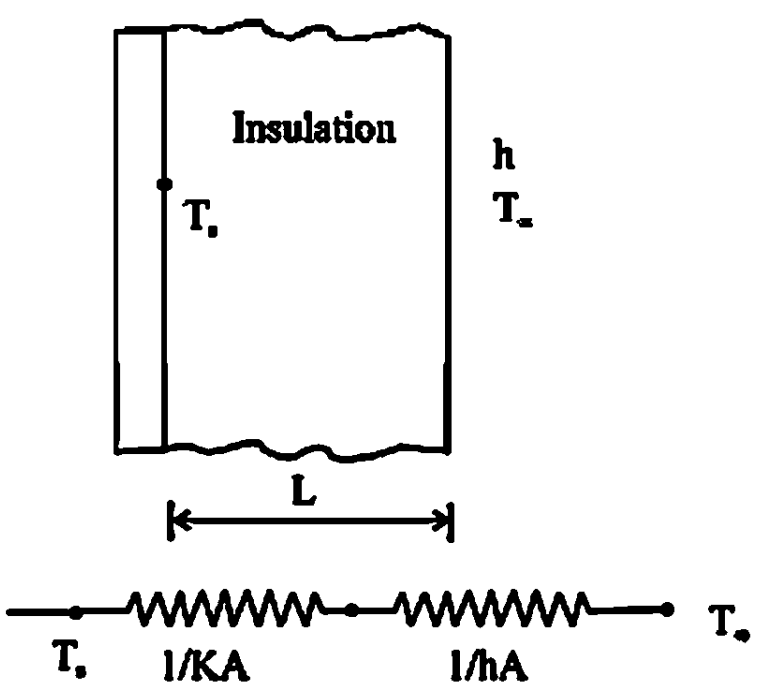
Repeat Prob. 17–30 for expanded perlite insulation assuming conductivity is $k = 0.052$ W/m·K.

Step 1 of 9

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Step 2 of 9

Diagram for thermal network:



Step 3 of 9

Without having any insulation the rate of heat transfer can be determined as

$$\dot{Q} = hA(T_s - T_\infty)$$
$$= 10 \times 3 \times (110 - 32)$$
$$= 2340\,\text{W}$$

Step 4 of 9

In order to reduce heat loss by 90%, the new heat transfer rate calculated as

$$\dot{Q} = 0.1 \times 2340$$
$$= 234\,\text{W}$$

The thermal resistance calculated as

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$
$$R_{total} = \frac{\Delta T}{\dot{Q}}$$
$$= \frac{(110 - 32)}{234}$$
$$= 0.3333\,\text{K/W}$$

Step 5 of 9

In order to have this total resistance, the thickness of the thermal resistance must be,

$$R_{total} = \frac{1}{hA} + \frac{L}{kA}$$
$$L = kA\left(R_{total} - \frac{1}{hA}\right)$$
$$L = 0.052 \times 3 \times \left(0.3333 - \frac{1}{10 \times 3}\right)$$
$$= 0.0468\,\text{m}$$

Therefore, thickness of the insulation, $\boxed{L = 0.0468\,\text{m}}$

Step 6 of 9

The amount of heat transfer rate saved by the insulation is 90% of the total heat transfer rate, therefore

$$\dot{Q} = 0.9 \times 2340$$
$$= 2106\,\text{W}$$
$$= 2.106\,\text{kJ/s}$$
$$= 2.106 \times 3600\,\text{kJ/h}$$
$$= 7851.6\,\text{kJ/h}$$

Step 7 of 9

Since the furnace operates continuously, the number of hours per year is $365 \times 24 = 8760\,\text{h/year}$ and the efficiency if the furnace is 78%, the amount of energy saved per year can be determined as,

Energy saved per year $= 7851.6\,(\text{kJ/h}) \times 8760\,(\text{h})$
$= 68780016\,\text{kJ}$

Amount of energy saved in terms of therms can be calculated as,

$1\,\text{therm} = 105500\,\text{kJ}$

$$\text{Energy saved per year} = 68780016\,(\text{kJ}) \times \frac{1\,\text{therm}}{105500\,(\text{kJ})}$$
$$= 652\,\text{therms}$$

Step 8 of 9

Therefore, the money saved can be determined as,

money saved = Energy saved × cost of energy
$= 652\,(\text{therms}) \times 1.1\,(\$/\text{therm})$
$= \$717.2\,\text{per year}$

Therefore, amount of money saved is $\boxed{\text{money saved} = \$717.2\,\text{per year}}$

Step 9 of 9

cost of the insulation $= \$250$

The payback period can be calculated as,

$$\text{Payback period}(t) = \frac{\text{insulation cost}}{\text{Money saved}}$$
$$= \frac{\$250}{\$717.2\,(\text{per year})}$$
$$= 0.35\,\text{years}$$
$$= 4.2\,\text{months}$$

Therefore, payback period is $\boxed{t = 4.2\,\text{months}}$

Question:

Reconsider Prob. 17–30. Using EES (or other) software, investigate the effect of thermal conductivity on the required insulation thickness. Plot the thickness of insulation as a function of the thermal conductivity of the insulation in the range of 0.02 W/m·K to 0.08 W/m·K, and discuss the results.

Step 1 of 11

Given data:

Area of the wall,

$A = 2 \times 1.5$

$= 3\text{m}^2$

Temperature of the outer surface of the furnace wall, $T_s = 110°\text{C}$

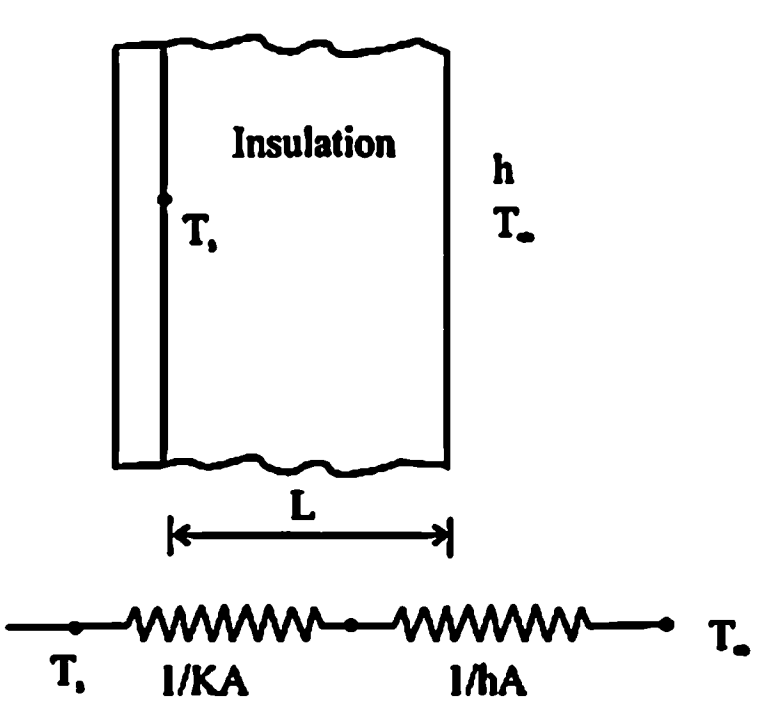
Temperature inside the furnace room, $T_\infty = 32°\text{C}$

Thermal conductivity of the glass insulation range, $k = 0.02 - 0.08\ \text{W/m-K}$

Combined Convection and radiation heat transfer coefficient, $h = 10\,\text{W/m}^2 \cdot \text{K}$

Step 2 of 11

Diagram for thermal network:



Step 3 of 11

Without having any insulation the rate of heat transfer can be determined as,

$\dot{Q} = hA(T_s - T_\infty)$

$= 10 \times 3 \times (110 - 32)$

$= 2340\,\text{W}$

Step 4 of 11

In order to reduce heat loss by 90%, the new heat transfer rate calculated as

$Q = 0.1 \times 2340(\text{W})$

$= 234\,\text{W}$

The thermal resistance calculated as,

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$

$$R_{total} = \frac{\Delta T}{\dot{Q}}$$

$$= \frac{(110 - 32)}{234}$$

$$= 0.3333\ \text{K/W}$$

Step 5 of 11

In order to have this total resistance, the thickness of the thermal resistance calculated and tabulated using Excel,

$$R_{total} = \frac{1}{hA} + \frac{L}{kA}$$

$$L = kA\left(R_{total} - \frac{1}{hA}\right)$$

$$L = k \times 3 \times \left(0.3333 - \frac{1}{10 \times 3}\right)$$

$$= 0.9k$$

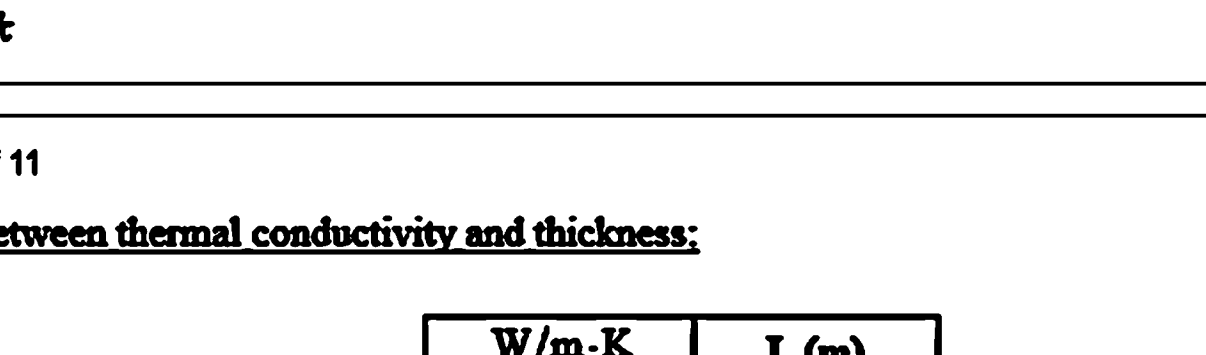
Step 6 of 11

Table between thermal conductivity and thickness:

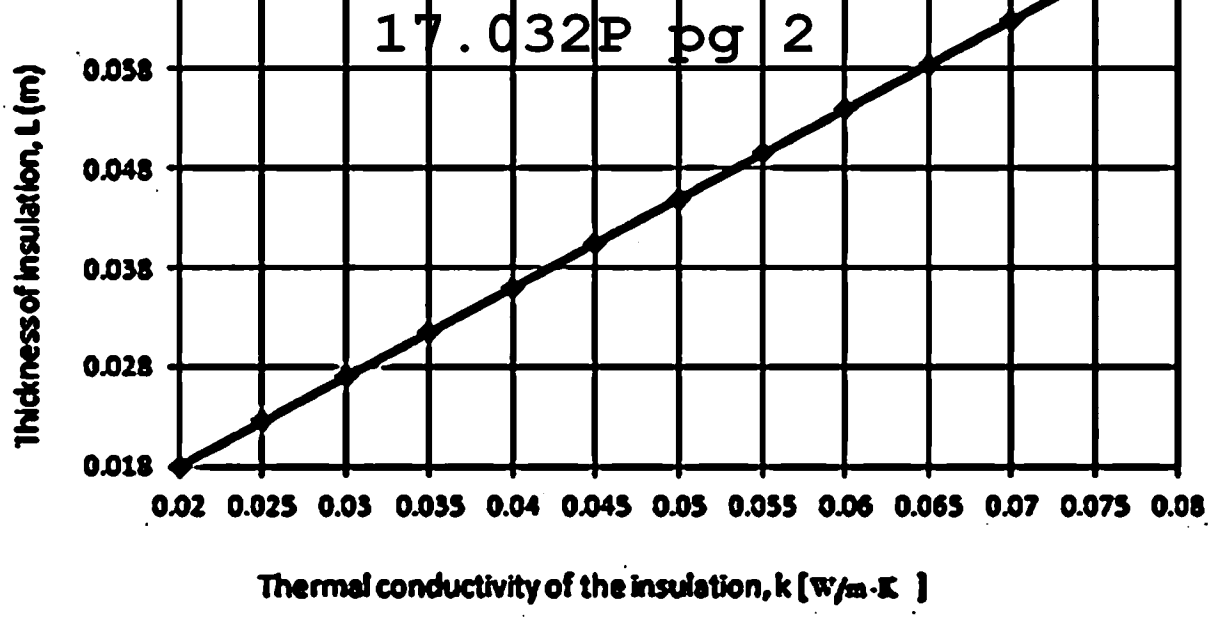
W/m-K	L (m)
0.02	0.018
0.025	0.0225
0.03	0.027
0.035	0.0315
0.04	0.036
0.045	0.0405
0.05	0.045
0.055	0.0495
0.06	0.054
0.065	0.0585
0.07	0.063
0.075	0.0675
0.08	0.072

Step 7 of 11

Plot between thermal conductivity and thickness:



17.032P pg 2



Step 8 of 11

From the above graph shows that thickness of the insulation is directly proportionally to the thermal conductivity. So amount of insulation added which will reduce the amount heat losses.

The amount of heat transfer rate saved by the insulation is 90% of the total heat transfer rate, therefore

$$\dot{Q} = 0.9 \times 2340\,(\text{W})$$
$$= 2106\,\text{W}$$
$$= 2.106\,\text{kJ/s}$$
$$= 2.106 \times 3600\,\text{kJ/h}$$
$$= 7851.6\,\text{kJ/h}$$

Step 9 of 11

Since the furnace operates continuously, the number of hours per year is $365 \times 24 = 8760\,\text{h/year}$ and the efficiency if the furnace is 78%, the amount of energy saved per year can be determined as

$$\text{Energy saved per year} = 7851.6\,(\text{kJ/h}) \times 8760\,(\text{h})$$
$$= 68780016\,\text{kJ}$$

Amount of energy saved in terms of therms can be calculated as

$$1\,\text{therm} = 105500\,\text{kJ}$$

$$\text{Energy saved per year} = 68780016\,(\text{kJ}) \times \frac{1\,\text{therm}}{105500\,(\text{kJ})}$$
$$= 652\,\text{therms}$$

Step 10 of 11

Therefore, the money saved can be determined as,

$$\text{money saved} = \text{Energy saved} \times \text{cost of energy}$$
$$= 652\,(\text{therms}) \times 1.1\,(\$/\text{therm})$$
$$= \$717.2\,\text{per year}$$

Therefore, amount of money saved is $\boxed{\text{money saved} = \$717.2\,\text{per year}}$

Step 11 of 11

$$\text{cost of the insulation} = \$250$$

The payback period can be calculated as,

$$\text{Payback period}\,(t) = \frac{\text{insulation cost}}{\text{Money saved}}$$
$$= \frac{\$250}{\$717.2\,(\text{per year})}$$
$$= 0.35\,\text{years}$$
$$= 4.2\,\text{months}$$

Therefore, payback period is $\boxed{t = 4.2\,\text{months}}$

Question: 17.033P

The wall of a refrigerator is constructed of fiberglass insulation (k = 0.035 W/m·K) sandwiched between two layers of 1-mm-thick sheet metal (k = 15.1 W/m·K). The refrigerated space is maintained at 2°C, and the average heat transfer coefficients at the inner and outer surfaces of the wall are 4 W/m2·K and 9 W/m2·K, respectively. The kitchen temperature averages 24°C. It is observed that condensation occurs on the outer surfaces of the refrigerator when the temperature of the outer surface drops to 20°C. Determine the minimum thickness of fiberglass insulation that needs to be used in the wall in order to avoid condensation on the outer surfaces.

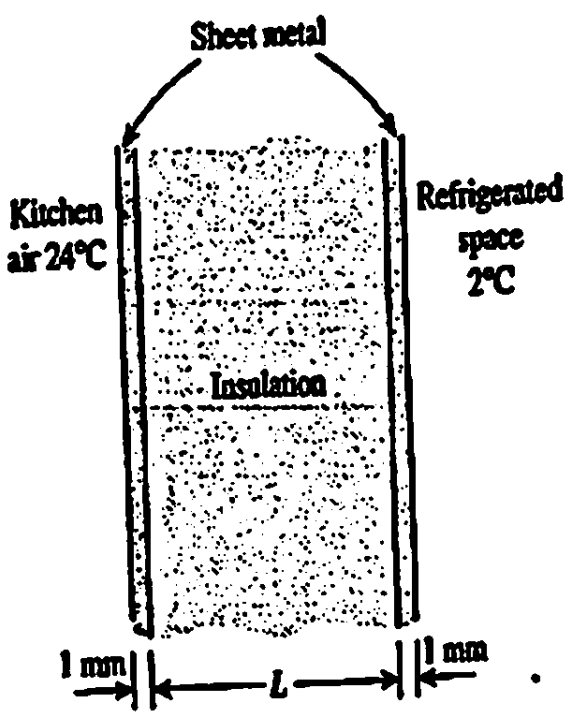


FIGURE P17-33

Step 1 of 5

Calculate the heat flux of the system is

$$\dot{q} = h(T_{room} - T_{s,out})$$

Here,

Temperature inside the kitchen room T_{room}

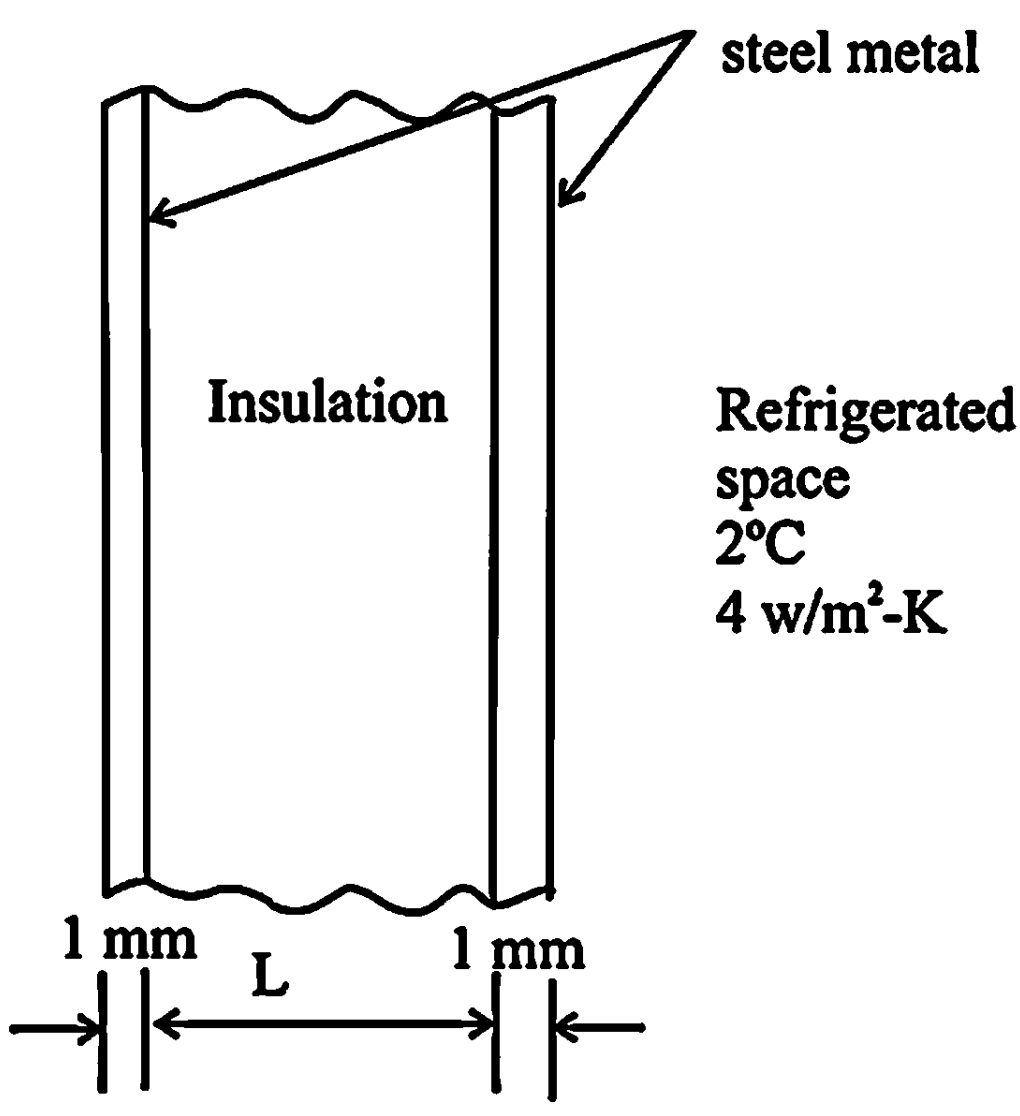
Temperature of the surface is $T_{s,out}$

Substitute $9\,\mathrm{W/m^2 \cdot K}$ for h_o, 24°C for T_{room}, 20°C for $T_{s,out}$

$$\begin{aligned}\dot{q} &= h(T_{room} - T_{s,out}) \\ &= 9(24-20) \\ &= 36\,\mathrm{W/m^2}\end{aligned}$$

Step 2 of 5

Schematic diagram of the wall of refrigerator system:



Step 3 of 5

Calculate the thermal resistance of the wall,

$$\dot{q} = \frac{\Delta T}{R_{total}}$$

$$R_{total} = \frac{\Delta T}{\dot{q}}$$

$$R_{total} = \frac{T_{room} - T_{refrig}}{\dot{q}}$$

Here,

Total thermal resistance is R_{total}

Temperature of the room is T_{room}

Temperature of the refrigeration is T_{refrig}

Substitute 25°C for T_{room}, 2°C for T_{refri}, $36\,\mathrm{W/m^2}$ for $\dot{q}$

$$\begin{aligned}R_{total} &= \frac{T_{room} - T_{refrig}}{\dot{q}} \\ &= \frac{(24-2)}{36} \\ &= 0.611\,\mathrm{m^2 \cdot K/W}\end{aligned}$$

Step 4 of 5

Write the thermal resistance expression in terms of heat transfer coefficient and wall coefficient

$$R_{total} = \frac{1}{h_o} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_i}$$

By rewrite the above equation, the thickness of the insulation

$$L_2 = k_2\left(R_{total} - \frac{1}{h_o} - 2\frac{L_1}{k_1} - \frac{1}{h_i}\right)$$

Here,

Thermal conductivity of the fiberglass insulation, k_2

Thermal conductivity of the sheet metal wall, k_1

Thicknesses of the sheet metal are L_1 and L_3

Heat transfer coefficient at the inner surface of the wall, h_i

Heat transfer coefficient at the outer surface of the wall, h_o

Step 5 of 5

Substitute $0.611\,\mathrm{m^2 \cdot K/W}$ for R_{total}, $9\,\mathrm{W/m^2 \cdot K}$ for h_o, $= 4\,\mathrm{W/m^2 \cdot K}$ for h_i, $15.1\,\mathrm{W/m \cdot K}$ for k_1, $0.035\,\mathrm{W/m \cdot K}$ for k_2,

$$\begin{aligned}L_2 &= k_2\left(R_{total} - \frac{1}{h_o} - 2\frac{L_1}{k_1} - \frac{1}{h_i}\right) \\ &= 0.035\left(0.611 - \frac{1}{9} - \frac{2\times 0.001}{15.1} - \frac{1}{4}\right) \\ &= 0.00874\,\mathrm{m}\end{aligned}$$

Therefore, thickness of insulation is $\boxed{0.00874\,\mathrm{m}}$

Question: 17.034P

Reconsider Prob. 17-33. Using EES(or other) software, investigate the effects of the thermal conductivities of the insulation material and the sheet metal on the thickness of the insulation. Let the thermal conductivity vary from 0.02 W/m·K to 0.08 W/m·K for insulation and 10W/m·K to 400 W/m·K for sheet metal. Plot the thickness of the insulation as the functions of the thermal conductivities of the insulation and the sheet metal, and discuss the results.

Step 1 of 5

Given data:

Thermal conductivity of the fiberglass insulation, $k_2 = 0.02 - 0.08 \text{W/m-K}$

Thermal conductivity of the sheet metal wall, $k_1 = k_3 = 10 - 400 \text{W/m-K}$

Thickness of the sheet metal, $L_1 = L_3 = 1\text{mm} = 0.001\text{m}$

Temperature inside the refrigerator, $T_{refrig} = 2^\circ\text{C}$

Temperature inside the kitchen room, $T_{room} = 28^\circ\text{C}$

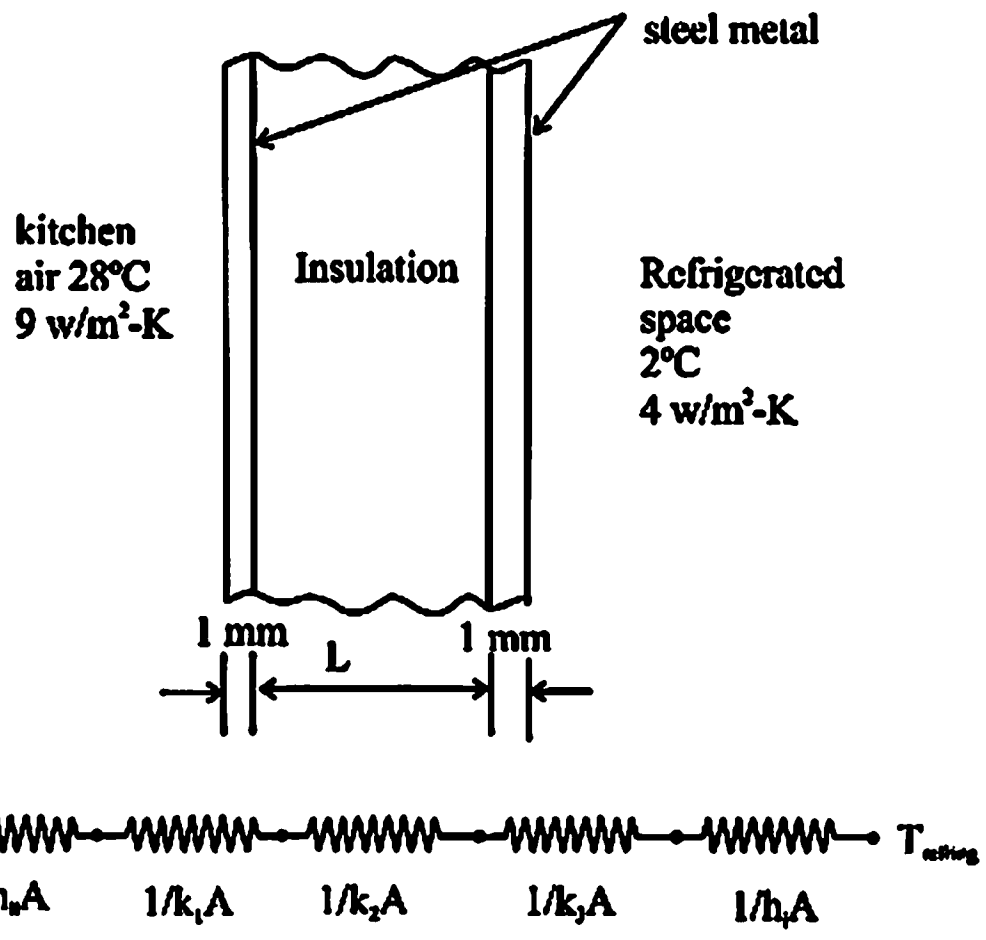
Heat transfer coefficient at the inner surface of the wall, $h_i = 4 \text{W/m}^2\text{-K}$

Heat transfer coefficient at the outer surface of the wall, $h_o = 9 \text{W/m}^2\text{-K}$

Temperature of the outer surface the refrigerator, $T_{s,out} = 20^\circ\text{C}$

Step 2 of 5

Network diagram:



Step 3 of 5

The minimum thickness of insulation can be determined by assuming the outer surface temperature of the refrigerator to be 20°C. In steady operation, the rate of heat transfer through the refrigerator wall is constant, and thus heat transfer/heat flux between the room and the refrigerated space is equal to the heat transfer/heat flux between the room and the outer surface of the refrigerator.

Therefore, the heat flux can be calculated as

$$\dot{q} = h(T_{room} - T_{s,out})$$
$$= 9(\text{W/m}^2\text{-K}) \times (25 - 20)$$
$$= 45 \text{W/m}^2$$

Step 4 of 5

By using the thermal resistance network approach, the total thermal resistance can be calculated as

$$\dot{q} = \frac{\Delta T}{R_{total}} \rightarrow R_{total} = \frac{\Delta T}{\dot{q}}$$

$$R_{total} = \frac{T_{room} - T_{refrig}}{\dot{q}}$$
$$= \frac{(25-2)}{45(\text{W/m}^2)}$$
$$= 0.511 \text{m}^2\text{-K/W}$$

The total thermal resistance can be expressed as

$$R_{total} = \frac{1}{h_o} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_i}$$

By rewriting the above equation for the insulation thickness

$$L_2 = k_2 \left(R_{total} - \frac{1}{h_o} - 2\frac{L_1}{k_1} - \frac{1}{h_i} \right)$$

Using the Microsoft Excel sheet the above equation solved by varying the thermal conductivities of insulation and sheet metal as follows

Step 5 of 5

The Variation of insulation thickness by varying the thermal conductivity of the insulation tabulated and plotted as follows,

$$L_2 = k_2 \left(R_{total} - \frac{1}{h_o} - 2\frac{L_1}{k_1} - \frac{1}{h_i} \right)$$
$$= k_2 \left(0.511(\text{m}^2\text{-K/W}) - \frac{1}{9(\text{W/m}^2\text{-K})} - 2 \times \frac{0.001(\text{m})}{15.1(\text{W/m-K})} - \frac{1}{4(\text{W/m}^2\text{-K})} \right)$$
$$= k_2 0.15\text{m}$$

Question: 17.035P

Heat is to be conducted along a circuit board that has a copper layer on one side. The circuit board is 15 cm long and 15 cm wide, and the thicknesses of the copper and epoxy layers are 0.1 mm and 1.2 mm, respectively. Disregarding heat transfer from side surfaces, determine the percentages of heat conduction along the copper (k = 386 W/m·K) and epoxy (k = 0.26 W/m·K) layers. Also determine the effective thermal conductivity of the board.

Step 1 of 8

Given data:

Area of circuit board $A = 15\text{cm} \times 15\text{cm}$

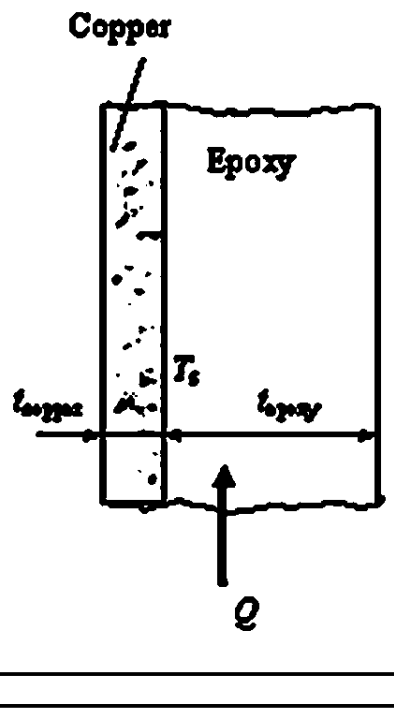
Thickness of copper layer $t_{copper} = 0.1\text{mm}$

Thickness of epoxy layer $t_{epoxy} = 1.2\text{mm}$

Thermal conductivity of copper $k_{Cu} = 386\text{W/m°C}$

Thermal conductivity of epoxy $k_{epoxy} = 0.26\text{W/m°C}$

Step 2 of 8



Step 3 of 8

If steady operating condition exists and heat transfer is one-dimensional since heat transfer from the side surfaces is disregarded then take the length in the direction of heat transfer to be L and the width of the board to be w. Then heat conduction along this two-layer board can be expressed as

$$\dot{Q} = \dot{Q}_{copper} + \dot{Q}_{epoxy}$$
$$= \left(kA\frac{\Delta T}{L}\right)_{copper} + \left(kA\frac{\Delta T}{L}\right)_{epoxy}$$
$$= \left[(kt)_{copper} + (kt)_{epoxy}\right] w\frac{\Delta T}{L}$$

Step 4 of 8

Heat conduction along an "equivalent" board of thickness $t = t_{copper} + t_{epoxy}$ and thermal conductivity k_{eff} can be expressed as

$$\dot{Q} = \left(kA\frac{\Delta T}{L}\right)_{board}$$
$$= k_{eff}\,(t_{copper} + t_{epoxy}) w\frac{\Delta T}{L}$$

Setting the two relations above equal to each other and solving for the effective conductivity gives

$$k_{eff}\left(t_{copper} + t_{epoxy}\right) = (kt)_{copper} + (kt)_{epoxy}$$
$$k_{eff} = \frac{(kt)_{copper} + (kt)_{epoxy}}{t_{copper} + t_{epoxy}}$$

Step 5 of 8

Note that heat conduction is proportional to kt.

Substituting, the fractions of heat conducted along the copper and epoxy layers as well as the effective thermal conductivity of the board are determined to be

$$(kt)_{copper} = (386\ \text{W/m°C})(0.0001\ \text{m})$$
$$= 0.0386\ \text{W/°C}$$
$$(kt)_{epoxy} = (0.26\ \text{W/m°C})(0.0012\ \text{m})$$
$$= 0.000312\ \text{W/°C}$$
$$(kt)_{total} = (kt)_{copper} + (kt)_{epoxy}$$
$$= 0.0386 + 0.000312$$
$$= 0.038912\ \text{W/°C}$$

Step 6 of 8

The percentage of heat conduction along the epoxy

$$f_{epoxy} = \frac{(kt)_{epoxy}}{(kt)_{total}}$$
$$= \frac{0.000312}{0.038912}$$
$$= 0.008$$
$$= 0.8\%$$

∴ The percentage of heat conduction $\boxed{f_{epoxy} = 0.8\%}$

Step 7 of 8

The percentage of heat conduction along the copper

$$f_{copper} = \frac{(kt)_{copper}}{(kt)_{total}}$$
$$= \frac{0.0386}{0.038912}$$
$$= 0.992$$
$$= 99.2\%$$

∴ The percentage of heat conduction $\boxed{f_{copper} = 99.2\%}$

Step 8 of 8

The effective thermal conductivity of the board

$$k_{eff} = \frac{(kt)_{copper} + (kt)_{epoxy}}{t_{copper} + t_{epoxy}}$$
$$k_{eff} = \frac{(386 \times 0.0001 + 0.26 \times 0.0012)\ \text{W/°C}}{(0.0001 + 0.0012)\ \text{m}}$$
$$k_{eff} = 29.9\text{W/m°C}$$

∴ The effective thermal conductivity $\boxed{k_{eff} = 29.9\text{W/m°C}}$

Question: 17.036P

A 0.05-in-thick copper plate (k = 223 Btu/h·ft·°F) is sandwiched between two 0.15-in-thick epoxy boards (k = 0.15 Btu/h·ft·°F) that are 7 in ×9 in in size. Determine the effective thermal conductivity of the board along its 9-in-long side. What fraction of the heat conducted along that side is, conducted through copper?

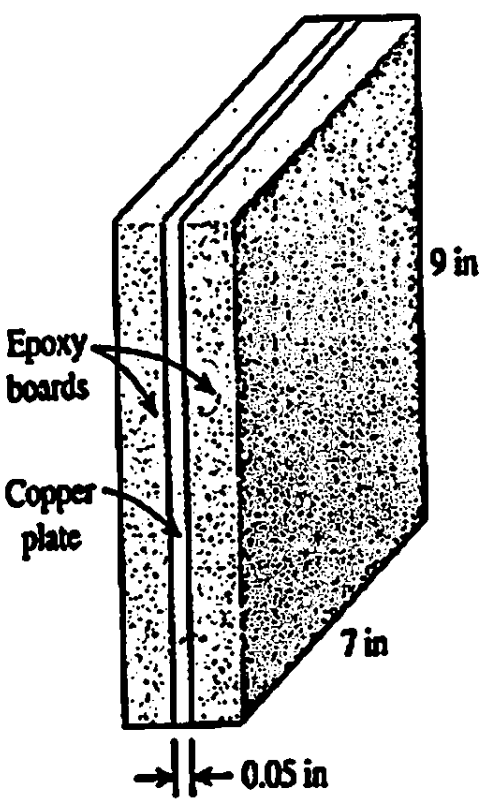


FIGURE P17–36E

Step 1 of 7

Given data:

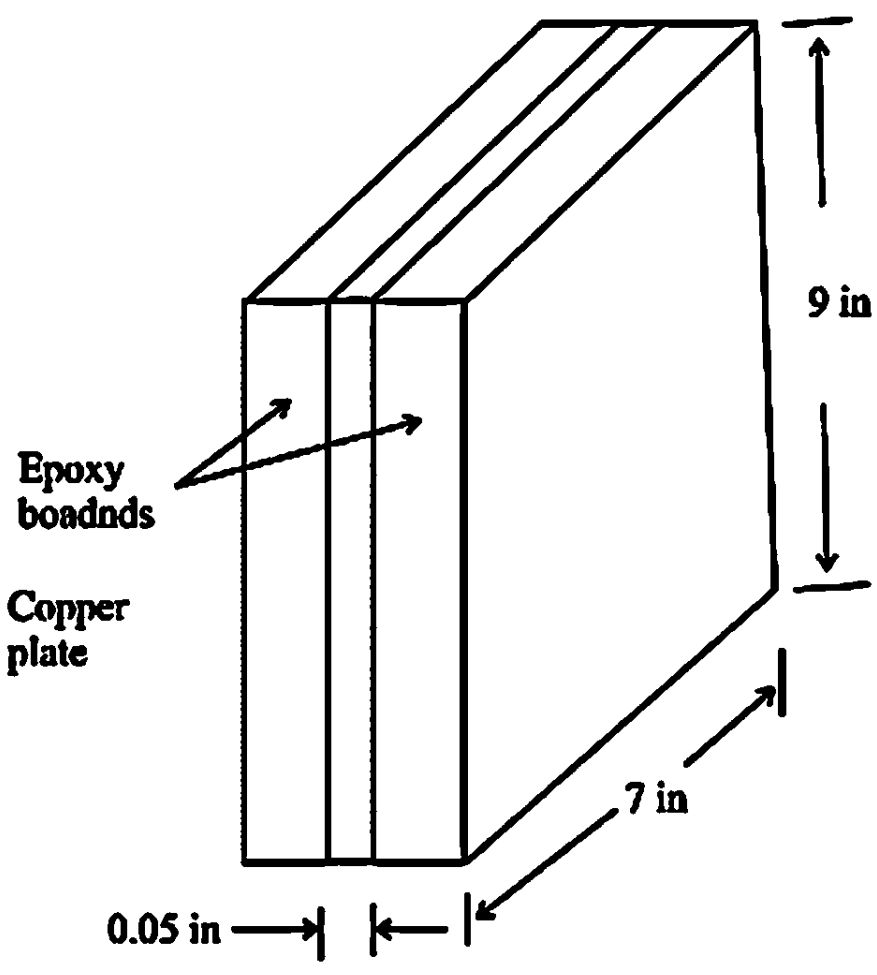
Thermal conductivity of the copper plate, $k = 223\text{Btu/h-ft-°F}$.

Thermal conductivity of the epoxy boards, $k = 0.15\text{Btu/h-ft-°F}$.

Thickness of the copper plate, $L = 0.05\text{in}$.

Thickness of the epoxy boards, $L = 0.15\text{in}$.

Step 2 of 7



Step 3 of 7

Take the length in the direction of heat transfer to be L, the width of the board to be w. and treat the two layers of epoxy as a single layer that is twice as thick. Then heat conduction along this two-layer plate can be expressed as

$$\begin{aligned}\dot{Q} &= \dot{Q}_{copper} + \dot{Q}_{epoxy} \\ &= \left(kA\frac{\Delta T}{L}\right)_{copper} + \left(kA\frac{\Delta T}{L}\right)_{epoxy} \\ &= \left(kwt\frac{\Delta T}{L}\right)_{copper} + \left(kwt\frac{\Delta T}{L}\right)_{epoxy} \\ &= \left[(kt)_{copper} + (kt)_{epoxy}\right]w\frac{\Delta T}{L}\end{aligned}$$

Step 4 of 7

The heat conduction happening along an equivalent plate thickness $t = t_{copper} + t_{epoxy}$ and the effective thermal conductivity k_{eff} can be expressed as

$$\begin{aligned}\dot{Q} &= \left(kA\frac{\Delta T}{L}\right)_{total} \\ &= \left(kwt\frac{\Delta T}{L}\right)_{total} \\ &= k_{eff}(t_{copper} + t_{epoxy})w\frac{\Delta T}{L}\end{aligned}$$

By equating the two equations the effective thermal conductivity can be expressed as

$$\dot{Q} = \left[(kt)_{copper} + (kt)_{epoxy}\right]w\frac{\Delta T}{L} = k_{eff}(t_{copper} + t_{epoxy})w\frac{\Delta T}{L}$$

$$k_{eff} = \frac{(kt)_{copper} + (kt)_{epoxy}}{(t_{copper} + t_{epoxy})}$$

Step 5 of 7

Heat conduction is proportional to kt, therefore the kt can be calculated for copper and epoxy as follows,

$$\begin{aligned}(kt)_{copper} &= 223(\text{Btu/h-ft-°F}) \times \frac{0.05}{12}(\text{ft}) \\ &= 0.93\text{Btu/h-°F}\end{aligned}$$

$$\begin{aligned}(kt)_{epoxy} &= 2 \times 0.15(\text{Btu/h-ft-°F}) \times \frac{0.15}{12}(\text{ft}) \\ &= 0.00375\text{Btu/h-°F}\end{aligned}$$

$$\begin{aligned}(kt)_{total} &= (kt)_{copper} + (kt)_{epoxy} \\ &= 0.93\text{Btu/h-°F} + 0.00375\text{Btu/h-°F} \\ &= 0.93375\text{Btu/h-°F}\end{aligned}$$

Step 6 of 7

The effective thermal conductivity can be calculated as

$$\begin{aligned}k_{eff} &= \frac{(kt)_{copper} + (kt)_{epoxy}}{(t_{copper} + t_{epoxy})} \\ &= \frac{0.93375(\text{Btu/h-°F})}{(0.05/12)(\text{ft}) + (2 \times 0.15/12)(\text{ft})} \\ &= 32.0143\text{Btu/h-ft-°F}\end{aligned}$$

Step 7 of 7

Fraction of the heat conducted along the length direction conducted through copper plate can be determined as

$$\begin{aligned}f_{copper} &= \frac{(kt)_{copper}}{(kt)_{total}} \\ &= \frac{0.93(\text{Btu/h-°F})}{0.93375(\text{Btu/h-°F})} \\ &= 0.996\end{aligned}$$

Therefore, Fraction of the heat conducted along the length direction conducted through copper plate is $\boxed{f_{copper} = 99.6\%}$.

Question: 17.037P

To defrost ice accumulated on the outer surface of an automobile windshield, warm air is blown over the inner surface of the windshield. Consider an automobile windshield with thickness of 5 mm and thermal conductivity of 1.4 W/m·K. The outside ambient temperature is −10°C and the convection heat transfer coefficient is 200 W/m2·K, while the ambient temperature inside the automobile is 25°C. Determine the value of the convection heat transfer coefficient for the warm air blowing over the inner surface of the windshield necessary to cause the accumulated ice to begin melting.

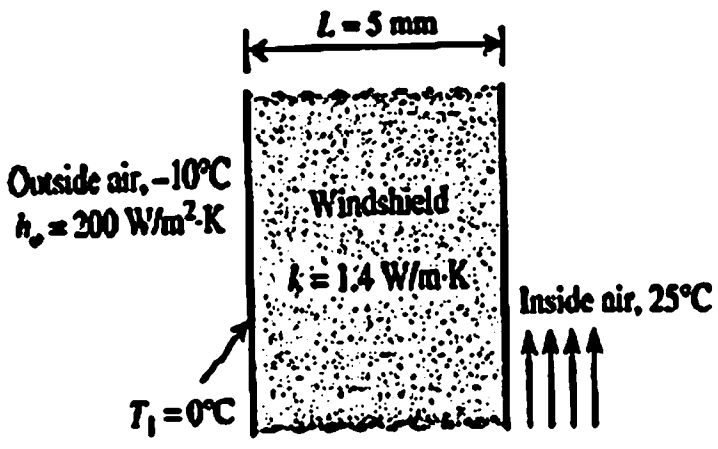


FIGURE P17–37

Step 1 of 5

Given data:

Thermal conductivity of the wind shield, $k = 1.4\text{W/m-K}$.

Thickness of the wind shield, $L = 5\text{mm} = 0.005\text{m}$.

Temperature of outside air, $T_{\infty,o} = -10\,^{\circ}\text{C}$.

Temperature of inside air, $T_{\infty,i} = 25\,^{\circ}\text{C}$.

Heat transfer coefficient at the outer surface of the wind shield, $h_o = 200\,\text{W/m}^2\text{-K}$.

Temperature of the outer surface the wind shield, $T_1 = 0\,^{\circ}\text{C}$.

Step 2 of 5

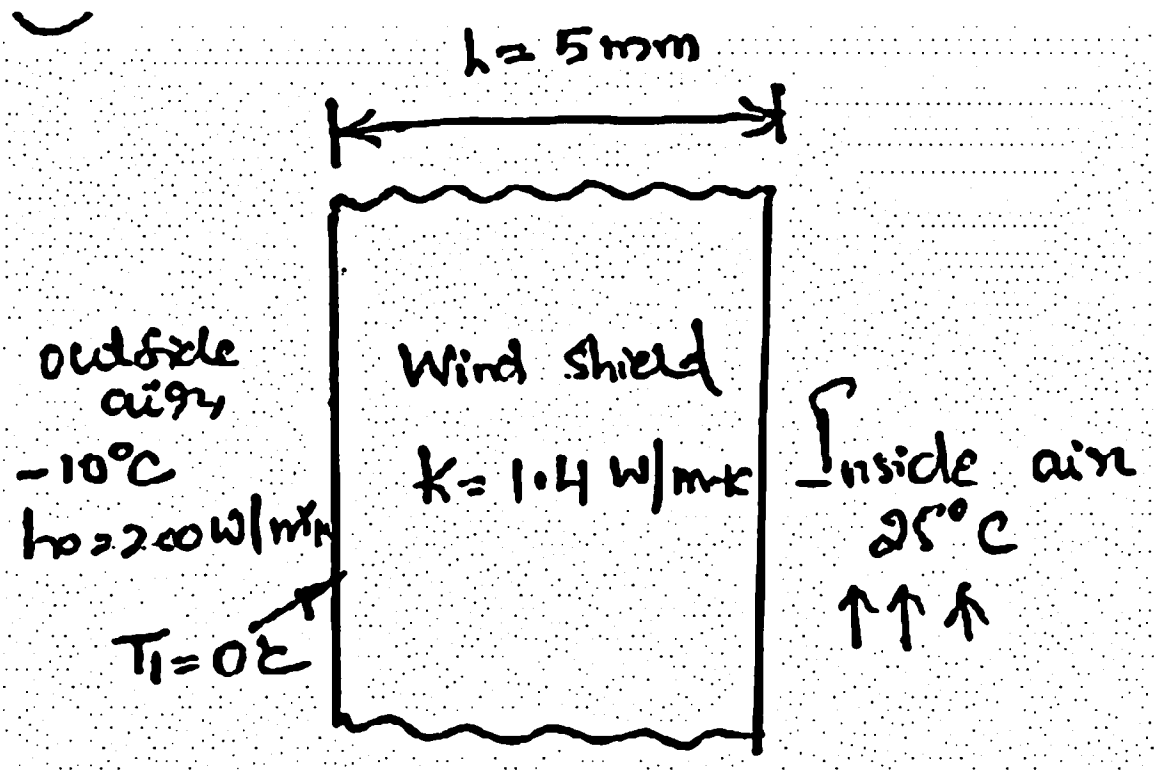


Figure 1 Schematic

Step 3 of 5

The temperature of the outer surface of the wind shield assumed to be $T_1 = 0\,^{\circ}\text{C}$, since the accumulated ice to begin melting. In order to determine the convective heat transfer coefficient for the warm air blowing over inner surface of the wind shield, the heat flux through the wind shield to be determined. Therefore, the heat flux through the wind shield can be calculated as

$$\dot{q} = h_o(T_1 - T_{\infty,o})$$
$$= 200\,(\text{W/m}^2\text{-K}) \times (0 - (-10))$$
$$= 2000\,\text{W/m}^2$$

Step 4 of 5

By using the thermal resistance network approach, the total thermal resistance can be calculated as

$$\dot{q} = \frac{\Delta T}{R_{total}}$$
$$\rightarrow R_{total} = \frac{\Delta T}{\dot{q}}$$
$$R_{total} = \frac{T_{\infty,i} - T_{\infty,o}}{\dot{q}}$$
$$= \frac{(25 - (-10))}{2000(\text{W/m}^2)}$$
$$= 0.0175\,\text{m}^2\text{-K/W}$$

Step 5 of 5

The total thermal resistance per unit area can be expressed as

$$R_{total} = \frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i}$$

By re writing the above equation, the convective heat transfer coefficient for the warm air blowing over inner surface of the wind shield can be expressed as

$$h_i = \frac{1}{\left(R_{total} - \left(\frac{1}{h_o} + \frac{L}{k}\right)\right)}$$
$$= \frac{1}{\left(0.0175(\text{m}^2\text{-K/W}) - \left(\frac{1}{200(\text{W/m}^2\text{-K})} + \frac{0.005(\text{K})}{1.4(\text{W/m-K})}\right)\right)}$$
$$= 112\,\text{W/m}^2\text{-K}$$

Therefore, convective heat transfer coefficient is

$$\boxed{h_i = 112\,\text{W/m}^2\text{-K}}$$

Question: 17.038P

An aluminum plate of 25 mm thick (k =235 W/m·K) is attached on a copper plate with thickness of 10 mm. The copper plate is heated electrically to dissipate a uniform heat flux of 5300 W/ m2.The upper surface of the aluminum plate is exposed to convection heat transfer in a condition such that the convection heat transfer coefficient is 67 W/m2·K and the surrounding room temperature is 20°C. Other surfaces of the two attached plates are insulated such that heat only dissipates through the upper surface of the aluminum plate. If the surface of the copper plate that is attached to the aluminum plate has a temperature of 100°C, determine the thermal contact conductance of the aluminum/copper interface.

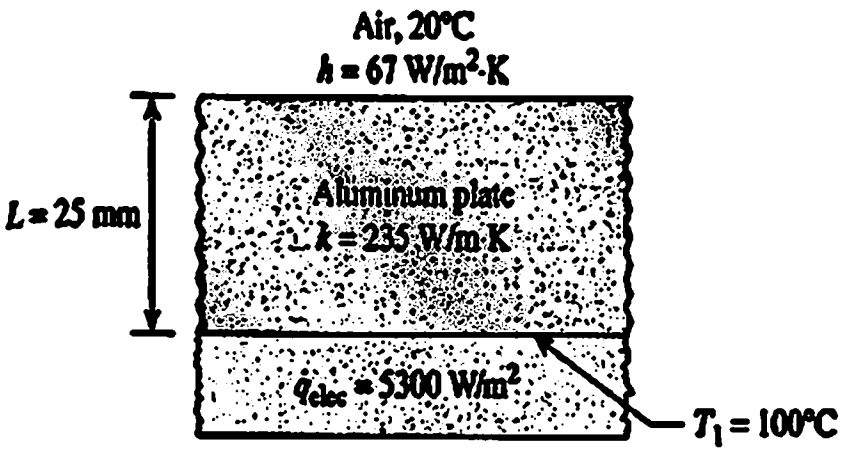


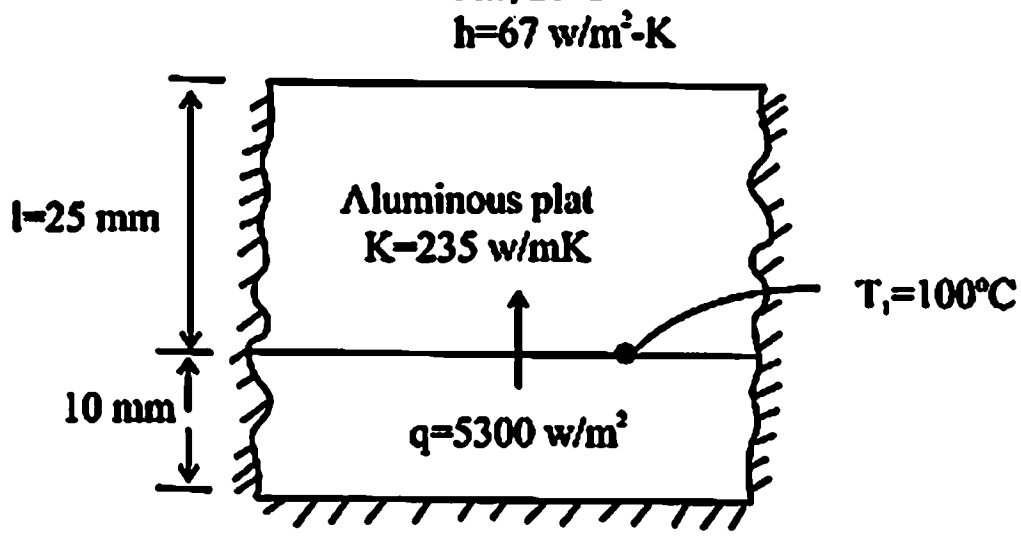
FIGURE P17–38

Step 1 of 5

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Step 2 of 5

Let us draw the diagram:



Step 3 of 5

By using the thermal resistance network approach, the total thermal resistance per unit area can be calculated as

$$\dot{q} = \frac{\Delta T}{R_{total}}$$

$$\rightarrow R_{total} = \frac{\Delta T}{\dot{q}}$$

$$R_{total} = \frac{T_i - T_\infty}{\dot{q}}$$

$$= \frac{(100-20)}{5300(\mathrm{W/m^2})}$$

$$= 0.0151 \mathrm{m^2\text{-}K/W}$$

Step 4 of 5

The total thermal resistance per unit area can be expressed as

$$R_{total} = R_{interface} + R_{plate} + R_{sub}$$

$$= R_{interface} + \frac{L}{k} + \frac{1}{h_\infty}$$

By re writing the above equation, the thermal resistance at the interface of the aluminum/copper plates can be expressed as

$$R_{interface} = R_{total} - \left(\frac{L}{k} + \frac{1}{h_i}\right)$$

$$= 0.0151(\mathrm{m^2\text{-}K/W}) - \left(\frac{0.025(\mathrm{m})}{235(\mathrm{W/m\text{-}K})} + \frac{1}{67(\mathrm{W/m^2\text{-}K})}\right)$$

$$= 6.8244 \times 10^{-5} \mathrm{m^2\text{-}K/W}$$

Step 5 of 5

Therefore, the thermal contact conductance of the aluminum/copper interface can be calculated as

$$\text{Thermal contact conductance} = \frac{1}{R_{interface}}$$

$$= \frac{1}{6.8244 \times 10^{-5}(\mathrm{m^2\text{-}K/W})}$$

$$= 14653.30 \mathrm{W/m^2\text{-}K}$$

Thermal contact conductance of the aluminum and copper interface is

$$\boxed{R_{cont} = 14.653 \mathrm{kW/m^2\text{-}K}}$$

Question: 17.039P

What is thermal contact resistance? How is it related to thermal contact conductance?

Step 1 of 2

When tow surfaces are pressed against each other the peaks form good material contact but the valleys form voids filled with air. As a result an interface contains numerous air gaps of varying sizes that act as insulation because of the low thermal conductivity of air. Thus an interface offers some resistance to heat transfer, and this resistance per unit interface area is called thermal contact resistance and it is denoted by R_c

$$R_c = \frac{1}{h_c} = \frac{\Delta T_{\text{interface}}}{\dot{Q}/A}$$

Step 2 of 2

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Question: 17.040P

Will the thermal contact resistance be greater for smooth or rough plain surfaces?

Step 1 of 1

The thermal contact resistance depends on the surface roughness and the material properties as well as the temperature and pressure at the interface and the type of fluid trapped at the interface

The thermal contact resistance will be greater for rough surfaces because an interface with rough surfaces will contain more air gaps whose thermal conductivity is low.

Question: 17.041P

A wall consists of two layers of insulation pressed against each other. Do we need to be concerned about the thermal contact resistance at the interface in a heat transfer analysis or can we just ignore it?

Step 1 of 1

The thermal contact resistance depends on the surface roughness and the material properties as well as the temperature and pressure at the interface and the type of fluid trapped at the interface. The thermal contact resistance is significant and can even dominate the heat transfer for good heat conductors such as metals, but can be disregarded for poor heat conductors such as insulators

An interface acts like a very thin layer of insulation, and thus the thermal contact resistance has significance only for highly conducting materials like metals. Therefore, the thermal contact resistance can be ignored for two layers of insulation pressed against each other.

Question: 17.042P

A plate consists of two thin metal layers pressed against each other. Do we need to be concerned about the thermal contact resistance at the interface in a heat transfer analysis or can we just ignore it?

Step 1 of 1

The thermal contact resistance depends on the surface roughness and the material properties as well as the temperature and pressure at the interface and the type of fluid trapped at the interface. The thermal contact resistance is significant and can even dominate the heat transfer for good heat conductors such as metals, but can be disregarded for poor heat conductors such as insulators. Therefore, the thermal contact resistance must be considered for two layers of metals pressed against each other.

Question: 17.043P

Consider two surfaces pressed against each other. Now the air at the interface is evacuated. Will the thermal contact resistance at the interface increase or decrease as a result?

Step 1 of 1

The thermal contact resistance is decrease with decreasing the roughness and increasing interface pressure. Heat transfer through the voids at an interface is by conduction and radiation. Evacuating the interface eliminates heat transfer by conduction, and thus increases the thermal contact resistance.

Question: 17.044P

Explain how the thermal contact resistance can be minimized.

Step 1 of 1

Thermal contact resistance can be minimized by
1) The thermal contact resistance is decrease with decreasing the roughness and increasing interface pressure.
2) Applying a thermally conducting liquid on the surfaces before they are pressed against each other,
3) By replacing the air at the interface by a better conducting gas such as helium or hydrogen,
5) By inserting a soft metallic foil such as tin, silver, copper, nickel, or aluminum between the two surfaces.

Question: 17.045P

The thermal contact conductance at the interface of two 1-cm-thick copper plates is measured to be 14,000 W/m2·K. Determine the thickness of the copper plate whose thermal resistance is equal to the thermal resistance of the interface between the plates.

Step 1 of 2

Obtain the thermal conductivity of the copper material from the table "properties of solid metal"

$$k = 401\,\mathrm{W/m \cdot K}$$

The thermal contact conductance, h_c is the inverse of thermal contact resistance, R_c and then the thermal contact resistance is,

$$R_c = \frac{1}{h_c}$$

Substitute $14,000\,\mathrm{W/m^2 \cdot K}$ for h_c

$$R_c = \frac{1}{14,000}$$
$$= 7.143 \times 10^{-5}\,\mathrm{m^2 \cdot K/W}$$

Step 2 of 2

The thermal resistance of the copper plates per unit area is,

$$R = \frac{L}{k}$$

Here L is the equivalent thickness of the copper plate and it can be determined by equating the thermal resistance of the copper plates and the thermal resistance of the interface between plates.

$$R = \frac{L}{k} = R_c$$

By rewriting the above equation for determining the equivalent thickness,

$$L = R_c k$$
$$= 7.143 \times 10^{-5} \times 401$$
$$= 0.02864\,\mathrm{m}$$
$$= 2.864\,\mathrm{cm}$$

Therefore, the thickness of the copper plate is $\boxed{L = 2.86\,\mathrm{cm}}$

Question: 17.046P

A 1-mm-thick copper plate (k = 386 W/m·K) is sandwiched between two 7-mm-thick epoxy boards (k = 0.26 W/m·K) that are 15 cm × 20 cm in size. If the thermal contact conductance on both sides of the copper plate is estimated to be 6000 W/m·K, determine the error involved in the total thermal resistance of the plate if the thermal contact conductances are ignored.

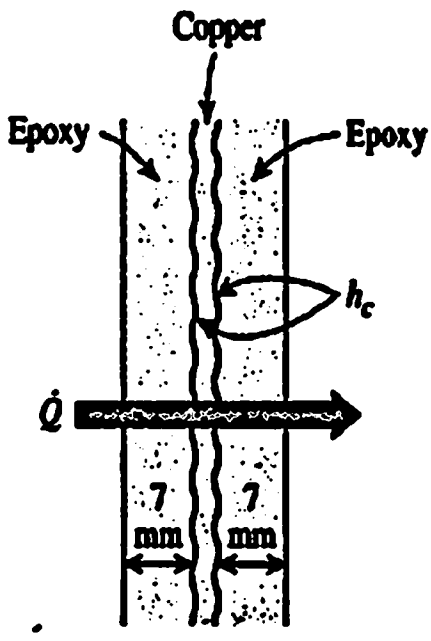


FIGURE P17–46

Step 1 of 5

Given data:

Thickness of the Epoxy plates, $L_1 = 7\text{mm} = 0.007\text{m}$.

Thickness of the copper plate, $L_2 = 1\text{mm} = 0.001\text{m}$.

Thermal conductivity of the copper plate, $k_c = 386\,\text{W/m-K}$.

Thermal conductivity of the epoxy boards, $k_e = 0.26\,\text{W/m-K}$.

Cross sectional area of the plates, $A = 0.15 \times 0.20\,\text{m}^2 = 0.03\text{m}^2$.

Thermal contact conductance of the copper plate, $h_c = 6000\,\text{W/m-K}$.

Step 2 of 5

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Step 3 of 5

Without considering the contact resistance:

By using the thermal network approach the total resistance of the network can be determined as

$$\begin{aligned} R_{total} &= R_{epoxy} + R_{copper} + R_{epoxy} \\ &= 2R_{epoxy} + R_{copper} \\ &= 2\frac{L_1}{k_e A} + \frac{L_2}{k_c A} \\ &= 2 \times \frac{0.007(\text{m})}{0.26(\text{W/m-K}) \times 0.03(\text{m}^2)} + \frac{0.001(\text{m})}{386(\text{W/m-K}) \times 0.03(\text{m}^2)} \\ &= 1.795\text{m-K/W} \end{aligned}$$

Step 4 of 5

With considering the contact resistance:

By using the thermal network approach the total resistance of the network can be determined as

$$\begin{aligned} R_{total} &= R_{epoxy} + R_c + R_{copper} + R_c + R_{epoxy} \\ &= 2R_{epoxy} + R_{copper} + 2R_c \\ &= 2\frac{L_1}{k_e A} + \frac{L_2}{k_c A} + 2\frac{1}{h_c A} \\ &= \left[2 \times \frac{0.007(\text{m})}{0.26(\text{W/m-K}) \times 0.03(\text{m}^2)} + \frac{0.001(\text{m})}{386(\text{W/m-K}) \times 0.03(\text{m}^2)} + 2 \times \frac{1}{6000(\text{W/m}^2\text{-K}) \times 0.03(\text{m}^2)} \right] \\ &= 1.806\text{m-K/W} \end{aligned}$$

Step 5 of 5

Therefore, the error associated in the total thermal resistance of the plate if the contact conductances are ignored can be calculated as

$$\begin{aligned} \text{Error} &= 1.806(\text{m-K/W}) - 1.795(\text{m-K/W}) \\ &= 0.011\,\text{m-K/W} \end{aligned}$$

% Error can be expressed as

$$\begin{aligned} \%\text{Error} &= \frac{Error}{R_{total}} \times 100 \\ &= \frac{0.011(\text{m-K/W})}{1.806\,(\text{m-K/W})} \times 100 \\ &= 0.61\% \end{aligned}$$

Therefore, Percentage of error involved is $\boxed{\%\text{Error} = 0.61\%}$

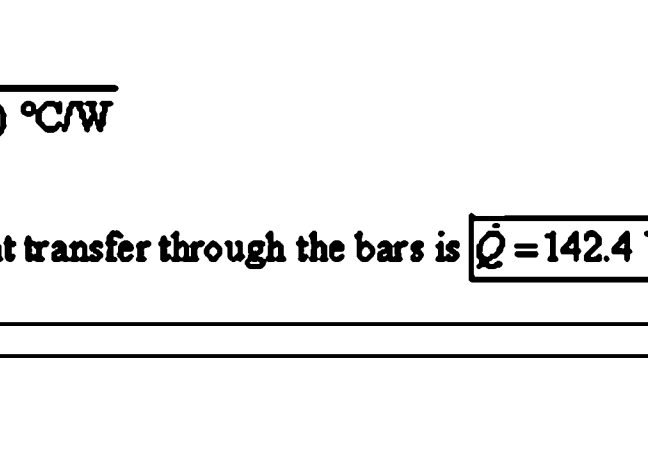
Question: 17.047P

Two 5-cm-diameter, 15-cm-long aluminum bars (k = 176 W/m·K) with ground surfaces are pressed against each other with a pressure of 20 atm. The bars are enclosed in an insulation sleeve and, thus, heat transfer from the lateral surfaces is negligible. If the top and bottom surfaces of the two-bar system are maintained at temperatures of 150°C and 20°C, respectively, determine (*a*) the rate of heat transfer along the cylinders under steady conditions and (*b*) the temperature drop at the interface.

Step 1 of 4

Given:
Diameter of aluminum bar = 5cm
Length of aluminum bar = 15cm
Thermal conductivity of aluminum k = 176 W/m·°C
The contact conductance at the interface of aluminum-aluminum plates for the case of ground surfaces and of 20 atm ≈ 2 MPa pressure is $h_c = 11{,}400\ \text{W/m}^2\cdot^\circ\text{C}$

Step 2 of 4



Step 3 of 4

(a)
The thermal resistance network in this case consists of two resistances.
Contact resistance

$$R_{contact} = \frac{1}{h_c A_c}$$

$$R_{contact} = \frac{1}{(11{,}400\ \text{W/m}^2\cdot^\circ\text{C})[\pi(0.05\ \text{m})^2/4]}$$

$$R_{contact} = 0.0447\ ^\circ\text{C/W}$$

Conduction resistance

$$R_{plate} = \frac{L}{kA}$$

$$R_{plate} = \frac{0.15\ \text{m}}{(176\ \text{W/m}\cdot^\circ\text{C})[\pi(0.05\ \text{m})^2/4]}$$

$$R_{plate} = 0.4341\ ^\circ\text{C/W}$$

Then the rate of heat transfer is determined to be

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$

$$\dot{Q} = \frac{\Delta T}{R_{contact} + 2R_{bar}}$$

$$\dot{Q} = \frac{(150-20)^\circ\text{C}}{(0.0447 + 2\times 0.4341)\ ^\circ\text{C/W}}$$

$$\dot{Q} = 142.4\ \text{W}$$

Therefore, the rate of heat transfer through the bars is $\boxed{\dot{Q} = 142.4\ \text{W}}$.

Step 4 of 4

(b)
The temperature drop at the interface is determined to be

$$\Delta T_{interface} = \dot{Q} R_{contact}$$

$$\Delta T_{interface} = (142.4\ \text{W})(0.0447\ ^\circ\text{C/W})$$

$$\boxed{\Delta T_{interface} = 6.4^0\text{C}}$$

Question: 17.048P

What are the two approaches used in the development of the thermal resistance network for two-dimensional problems?

Step 1 of 1

The two approaches used in development of the thermal resistance network in the x-direction for multi-dimensional problems are

(1) to assume any plane wall normal to the x-axis to be isothermal and

(2) to assume any plane parallel to the x-axis to be adiabatic.

Question: 17.049P

The thermal resistance networks can also be used approximately for multidimensional problems. For what kind of multidimensional problems will the thermal resistance approach give adequate results?

Step 1 of 1

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Question: 17.050P

When plotting the thermal resistance network associated with a heat transfer problem, explain when two resistances are in series and when they are in parallel.

Step 1 of 1

Parallel resistances indicate simultaneous heat transfer (such as convection and radiation on a surface). Series resistances indicate sequential heat transfer (such as two homogeneous layers of a wall).

Question: 17.051P

A typical section of a building wall is shown in Fig. P17-51. This section extends in and out of the page and is

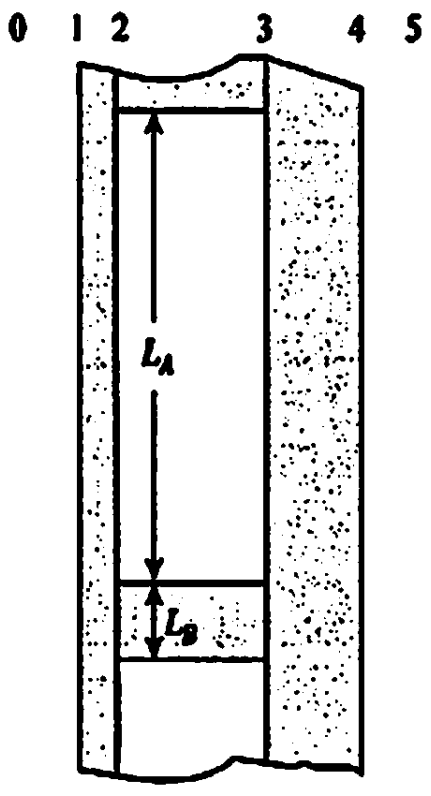
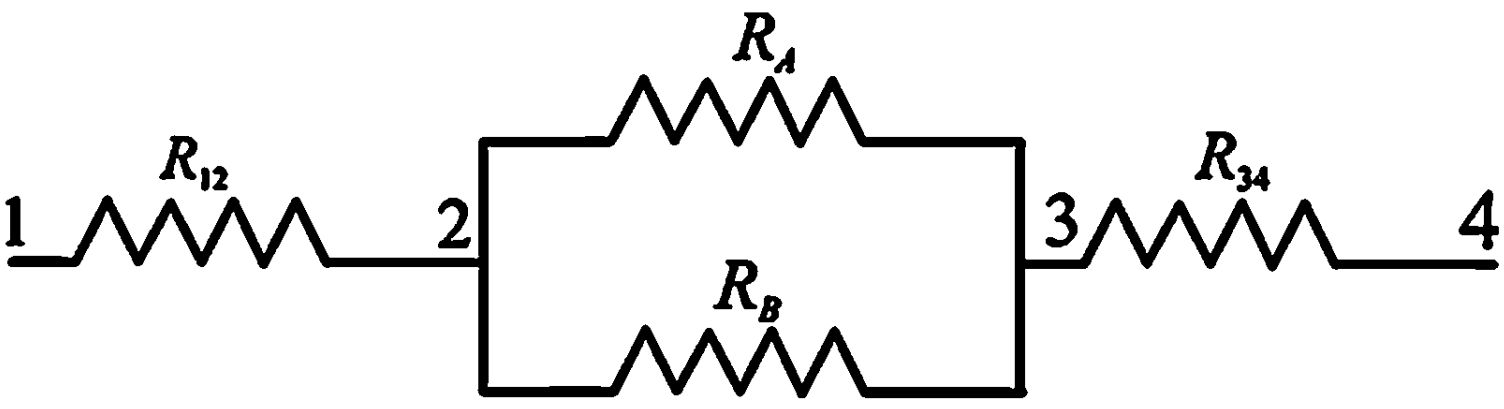


FIGURE P17–51

repeated in the vertical direction. The wall support members are made of steel (k = 50 W/m·K). The support members are 8 cm ($t23$) × 0.5 cm (LB). The remainder of the inner wall space is filled with insulation (k = 0.03 W/m·K) and measures 8 cm ($t23$) × 60 cm (LB). The inner wall is made of gypsum board (k= 0.5 W/m·K) that is 1 cm thick ($t12$) and the outer wall is made of brick (k = 1.0 W/m·K) that is 10 cm thick ($t34$). What is the average heat flux through this wall when $T1$ = 20°C and $T4$ = 35°C?

Step 1 of 4

The thermal network of the building wall is as shown in the diagram:



Step 2 of 4

By using the thermal network approach, the total resistance can be determined as follows:

$$R_{total} = R_{12} + R_{23} + R_{34}$$

$$= R_{12} + \frac{R_A R_B}{R_A + R_B} + R_{34}$$

$$= \frac{t_{12}}{k_i A_i} + \frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A} + \frac{t_B}{k_B A_B}} + \frac{t_{34}}{k_o A_o}$$

The cross sectional area of the inner or outer side of the building wall is as follows:

$$A_i = A_o = (L_A + L_B)w$$

$$A_A = L_A w$$

$$A_B = L_B w$$

Here, w is the width of the wall.

Step 3 of 4

The total resistance multiplies with $A_i = A_o = (L_A + L_B)w$ and it can be expressed as follows:

$$R_{total} A_i = \frac{t_{12}}{k_i} + A_i \left[\frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A} + \frac{t_B}{k_B A_B}}\right] + \frac{t_{34}}{k_o}$$

$$= \frac{t_{12}}{k_i} + (L_A + L_B)w \left(\frac{\left(\frac{t_A}{k_A L_A w}\right)\left(\frac{t_B}{k_B L_B w}\right)}{\frac{t_A}{k_A L_A w} + \frac{t_B}{k_B L_B w}}\right) + \frac{t_{34}}{k_o}$$

Assume, $w = 1\,\text{m}$, then the above equation is modified as follows:

$$R_{total} A_i = \frac{t_{12}}{k_i} + (L_A + L_B)\left(\frac{\left(\frac{t_A}{k_A L_A}\right)\left(\frac{t_B}{k_B L_B}\right)}{\frac{t_A}{k_A L_A} + \frac{t_B}{k_B L_B}}\right) + \frac{t_{34}}{k_o}$$

Substitute 0.01 m for t_{12} , $0.5\,\text{W/m}\cdot\text{K}$ for k_i , 0.005 m for L_B , 0.6 m for L_A , 0.08 m for t_A, t_B , $0.03\,\text{W/m}\cdot\text{K}$ for k_A , $50\,\text{W/m}\cdot\text{K}$ for k_B , 0.1 m for t_{34} and $1\,\text{W/m}\cdot\text{K}$ for k_o

$$R_{total} A_i = \frac{0.01}{0.5} + (0.6 + 0.005)\left(\frac{\left[\frac{0.08}{(0.03\times 0.6)}\right] \times \left[\frac{0.08}{(50\times 0.005)}\right]}{\left[\frac{0.08}{(0.03\times 0.6)}\right] + \left[\frac{0.08}{(50\times 0.005)}\right]}\right) + \frac{0.1}{1}$$

$$= 0.3006\,\text{m}^2\cdot\text{K/W}$$

Step 4 of 4

Calculate the average heat flux through the wall by using the following equation:

$$\frac{\dot{Q}}{A_i} = \frac{(T_4 - T_1)}{R_{total} A_i}$$

Here, T_1, T_4 are the extreme wall temperatures.

Substitute $35°\text{C}$ for T_4 and $20°\text{C}$ for T_1 .

$$\frac{\dot{Q}}{A_i} = \frac{35 - 20}{0.3006}$$

$$= 49.9\,\text{W/m}^2$$

Thus, the average heat flux through the wall is $\boxed{49.9\,\text{W/m}^2}$.

Question: 17.052P pg 1

A 4-m-high and 6-m-wide wall consists of a long 15-cm × 25-cm cross section of horizontal bricks (k = 0.72 W/m·K) separated by 3-cm-thick plaster layers (k = 0.22 W/m·K). There are also 2-cm-thick plaster layers on each side of the wall, and a 2-cm-thick rigid foam (k = 0.026 W/m·K) on the inner side of the wall. The indoor and the outdoor temperatures are 22°C and −4°C, and the convection heat transfer coefficients on the inner and the outer sides are $h1$ = 10 W/m2·K and $h2$ = 20 W/m2·K, respectively. Assuming one-dimensional heat transfer and disregarding radiation, determine the rate of heal transfer through the wall.

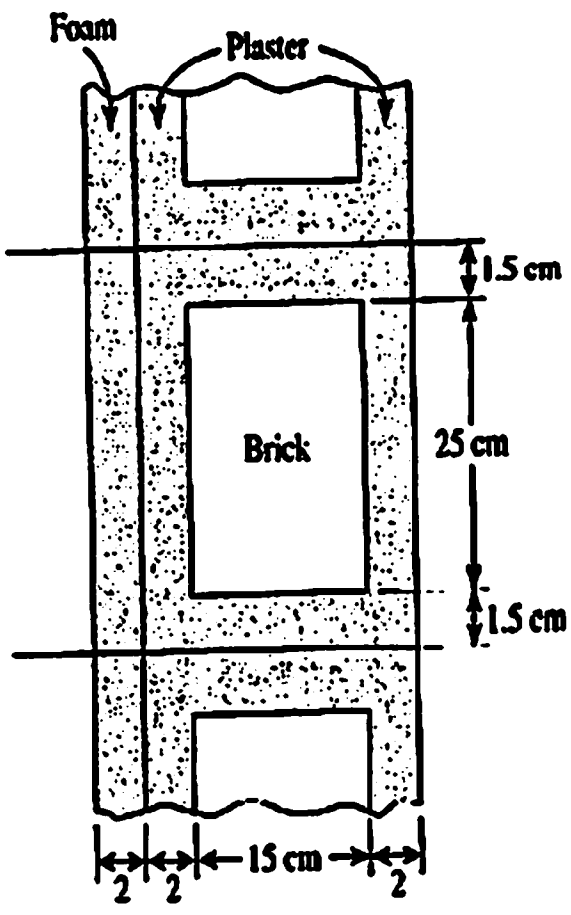


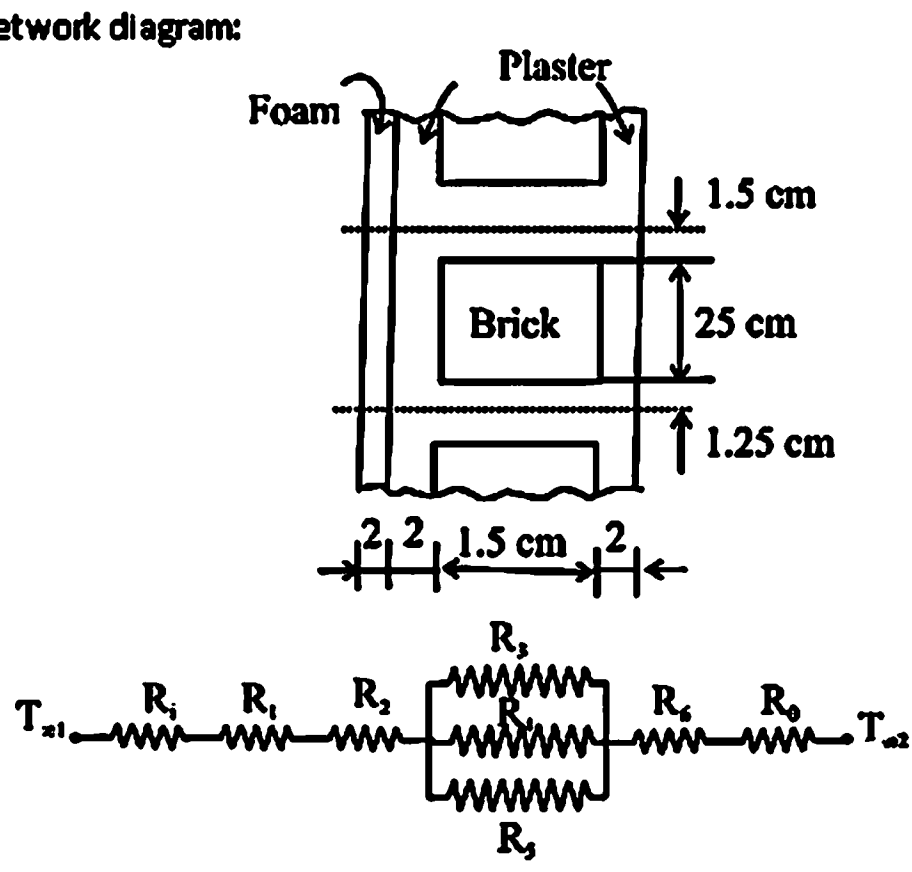
FIGURE P17–52

Step 1 of 7

Loading solution. Please wait ...

Step 2 of 7

Let us draw the network diagram:



Step 3 of 7

There is a pattern in the construction of the wall that repeats itself every 28-cm distance in the vertical direction. There is no variation in the horizontal direction. Therefore, consider a 1-m deep and 0.28 m high portion of the wall, since it is representative of the entire wall.

Assuming any cross section of the wall normal to the X-direction to be isothermal, the thermal resistance network for the representative section of the wall becomes as shown in the Fig above.

The individual resistances are evaluated as:

$$R_1 = R_{conv,1} = \frac{1}{h_1 A_1} = \frac{1}{10(\text{W/m}^2\text{-K}) \times (0.28 \times 1\text{m}^2)} = 0.357\text{K/W}$$

$$R_1 = R_{foam} = \frac{t_1}{k_1 A_1} = \frac{0.02(\text{m})}{0.026(\text{W/m-K}) \times (0.28 \times 1\text{m}^2)} = 2.75\text{K/W}$$

Step 4 of 7

$$R_2 = R_6 = R_{plaster,side} = \frac{t_2}{k_2 A_2} = \frac{0.02(\text{m})}{0.22(\text{W/m-K}) \times (0.28 \times 1\text{m}^2)} = 0.325\text{K/W}$$

$$R_3 = R_5 = R_{plaster,center} = \frac{t_3}{k_3 A_3} = \frac{0.15(\text{m})}{0.22(\text{W/m-K}) \times (0.015 \times 1\text{m}^2)} = 45.455\text{K/W}$$

Step 5 of 7

$$R_4 = R_{brick} = \frac{t_4}{k_4 A_4}$$

Step 5 of 7

$$R_4 = R_{brick}$$
$$= \frac{t_4}{k_4 A_4}$$
$$= \frac{0.15(\text{m})}{0.72(\text{W/m-K}) \times (0.25 \times 1\text{m}^2)}$$
$$= 0.8333\text{K/W}$$
$$R_o = R_{conv,2}$$
$$= \frac{1}{h_2 A_2}$$
$$= \frac{1}{20(\text{W/m}^2\text{-K}) \times (0.28 \times 1\text{m}^2)}$$
$$= 0.1786\text{K/W}$$

Step 6 of 7

The three resistances R_3, R_4 and R_5 in the middle are parallel, and their equivalent resistance is determined from

$$\frac{1}{R_{mid}} = \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5}$$
$$= \frac{1}{45.455(\text{K/W})} + \frac{1}{0.8333(\text{K/W})} + \frac{1}{45.455(\text{K/W})}$$
$$= 1.244\text{W/K}$$

This gives

$$R_{mid} = \frac{1}{1.244(\text{W/K})}$$
$$= 0.804\text{K/W}$$

Now all the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_2 + R_{mid} + R_6 + R_o$$
$$= 0.357(\text{K/W}) + 2.75(\text{K/W}) + 0.325(\text{K/W}) + 0.804(\text{K/W}) + 0.325(\text{K/W}) + 0.1786(\text{K/W})$$
$$= 4.7396\text{K/W}$$

Step 7 of 7

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{22 - (-4)}{4.7396(\text{K/W})}$$
$$= 5.5\text{W}$$

5.5 W Of heat is transferring through per 0.28m^2 surface area

And the total heat flux through the wall can be computed as

$$\dot{q} = \frac{\dot{Q}}{A}$$
$$= \frac{5.5(\text{W})}{0.28(\text{m}^2)}$$
$$= 19.6\text{W/m}^2$$

The total area of the wall is $A = 4 \times 6 = 24\text{m}^2$. Then the rate of heat transfer through the entire wall becomes

$$\dot{Q}_{total} = 19.6(\text{W/m}^2) \times 24(\text{m}^2)$$
$$= 470.4\text{ W}$$

Therefore maximum heat transfer is $\boxed{\dot{Q}_{total} = 470.4\text{ W}}$.

Question: 17.053P pg 1

Reconsider Prob. 17-52. Using EES (or other) software, plot the rate of heat transfer through the wall as a function of the thickness of the rigid foam in the range of 1 cm to 10 cm. Discuss the results.

Step 1 of 9

Given data:

Cross sectional are of the wall, $A = 4 \times 6 = 24\text{m}^2$

Thickness of the foam, $t_1 = 1-10\text{cm} = 0.01-0.1\text{m}$

Thickness of the plaster, $t_2 = t_6 = 2\text{cm} = 0.02\text{m}$

Thickness of the plaster between two bricks, $t_3 = t_5 = 15\text{cm} = 0.15\text{m}$

Thickness of the brick, $t_4 = 15\text{cm} = 0.15\text{m}$

Length of the foam, $L_1 = 28\text{cm}=0.28\text{m}$

Length of the plaster, $L_2 = L_6 = 28\text{cm}=0.28\text{m}$

Length of the plaster between two bricks, $L_3 = L_5 = 1.5\text{cm}=0.015\text{m}$

Length of the bricks, $L_4 = 25\text{cm}=0.25\text{m}$

Thermal conductivity of the foam, $k_1 = 0.026\ \text{W/m-K}$

Thermal conductivity of the plaster, $k_2 = k_3 = k_5 = k_6 = 0.22\ \text{W/m-K}$

Thermal conductivity of the brick, $k_4 = 0.72\ \text{W/m-K}$

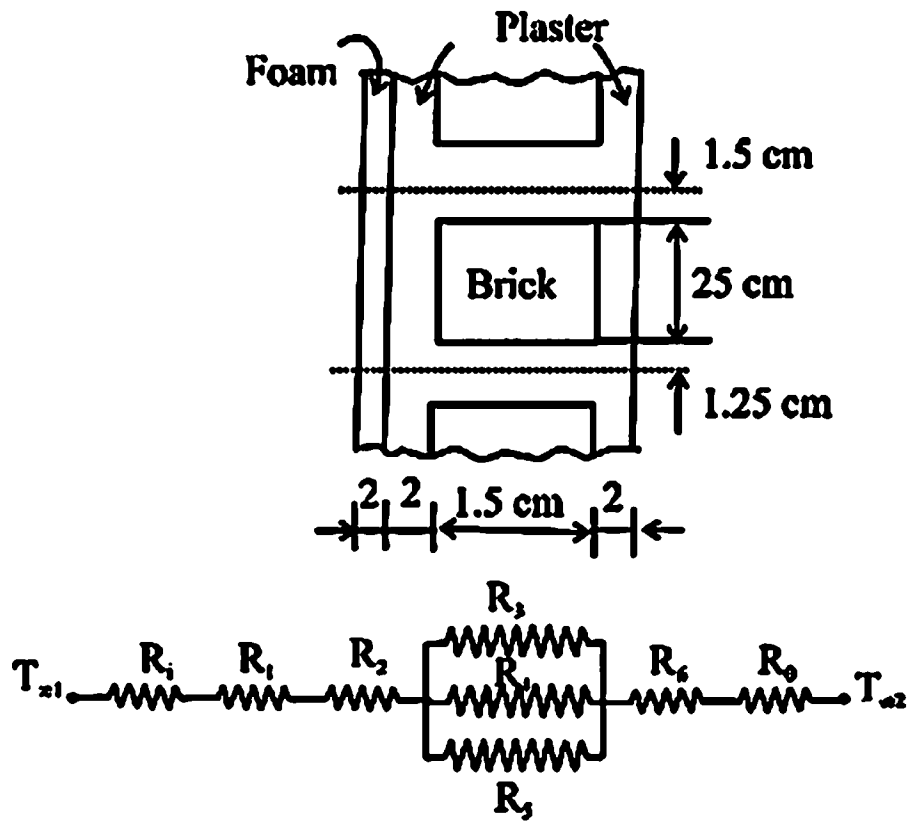
Indoor temperature, $T_{\infty,1} = 22^\circ\text{C}$

Outdoor temperature, $T_{\infty,2} = -4^\circ\text{C}$

Heat transfer coefficient on the inner wall, y

Heat transfer coefficient on the outer wall, $h_2 = 20\ \text{W/m}^2\text{-K}$

Step 2 of 9



Step 3 of 9

There is a pattern in the construction of the wall that repeats itself every 28-cm distance in the vertical direction. There is no variation in the horizontal direction. Therefore, consider a 1-m deep and 0.28 m high portion of the wall, since it is representative of the entire wall.

Assuming any cross section of the wall normal to the X-direction to be isothermal, the thermal resistance network for the representative section of the wall becomes as shown in the Fig above.

The individual resistances are evaluated as:

$$R_1 = R_{conv,1}$$

$$= \frac{1}{h_1 A_1}$$

$$= \frac{1}{10\,(\text{W/m}^2\text{-K}) \times (0.28 \times 1\text{m}^2)}$$

$$= 0.357\,\text{K/W}$$

$$R_1 = R_{foam}$$

$$= \frac{t_1}{k_1 A_1}$$

$$= \frac{t_1}{0.026\,(\text{W/m-K}) \times (0.28 \times 1\text{m}^2)}$$

$$= 137.4 t_1\,\text{K/W}$$

Step 4 of 9

$$R_2 = R_6$$

$$= R_{plaster,side}$$

$$= \frac{t_2}{k_2 A_2}$$

$$= \frac{0.02(\text{m})}{0.22\,(\text{W/m-K}) \times (0.28 \times 1\text{m}^2)}$$

$$= 0.325\,\text{K/W}$$

$$R_3 = R_5$$

$$= R_{plaster,center}$$

$$= \frac{t_3}{k_3 A_3}$$

$$= \frac{0.15(\text{m})}{0.22\,(\text{W/m-K}) \times (0.015 \times 1\text{m}^2)}$$

$$= 45.455\,\text{K/W}$$

Step 5 of 9

$$R_4 = R_{brick}$$

$$= \frac{t_4}{k_4 A_4}$$

$$= \frac{0.15(\text{m})}{0.72\,(\text{W/m-K}) \times (0.25 \times 1\text{m}^2)}$$

$$= 0.8333\,\text{K/W}$$

$$R_y = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$= 0.8333\,\text{K/W}$$

$$R_o = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$= \frac{1}{20\,(\text{W/m}^2\text{-K}) \times (0.28 \times 1\text{m}^2)}$$

$$= 0.1786\,\text{K/W}$$

Step 6 of 9

The three resistances R_3, R_4 and R_5 in the middle are parallel, and their equivalent resistance is determined from

$$\frac{1}{R_{mid}} = \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5}$$

$$= \frac{1}{45.455(\text{K/W})} + \frac{1}{0.8333(\text{K/W})} + \frac{1}{45.455(\text{K/W})}$$

$$= 1.244\,\text{W/K}$$

This gives

$$R_{mid} = \frac{1}{1.244(\text{W/K})}$$

$$= 0.804\,\text{K/W}$$

Now all the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_2 + R_{mid} + R_6 + R_o$$

$$= 0.357(\text{K/W}) + 137.4t_1(\text{K/W}) + 0.325(\text{K/W}) + 0.804(\text{K/W}) + 0.325(\text{K/W}) + 0.1786(\text{K/W})$$

$$= 2 + 137.4t_1\,\text{K/W}$$

Step 7 of 9

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$

$$= \frac{22 - (-4)}{2 + 137.4t_1(\text{K/W})}$$

$$= \frac{26}{2 + 137.4t_1(\text{K/W})}$$

This amount heat is transferring through per 0.28m^2 surface area

And the total heat flux through the wall can be computed as

$$\dot{q} = \frac{\dot{Q}}{A}$$

$$= \frac{26/[2 + 137.4t_1(\text{K/W})]}{0.28(\text{m}^2)}$$

$$= \frac{92.86}{[2 + 137.4t_1(\text{K/W})]}\,\text{W/m}^2$$

Step 8 of 9

The total area of the wall is $A = 4 \times 6 = 24\text{m}^2$. Then the rate of heat transfer through the entire wall becomes

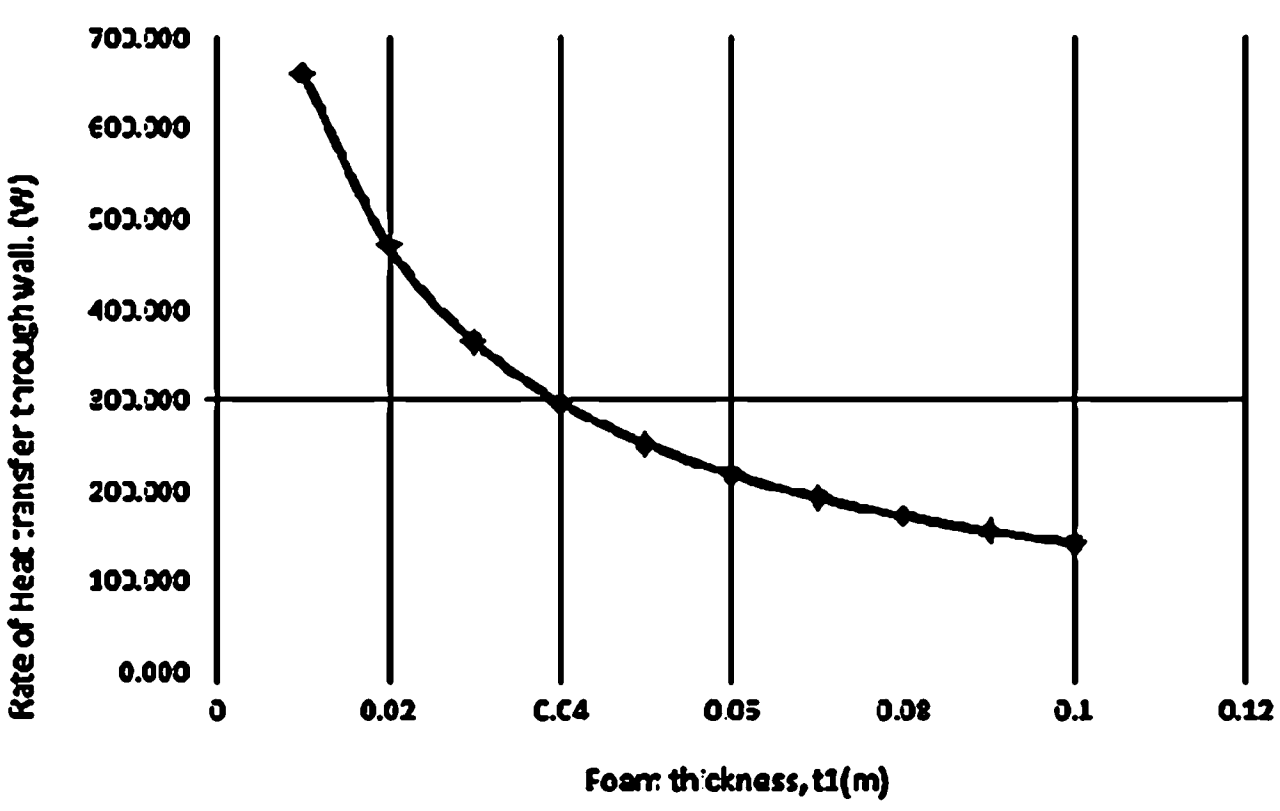
$$\dot{Q}_{total} = \frac{92.86}{[2 + 137.4t_1(\text{K/W})]}(\text{W/m}^2) \times 24(\text{m}^2)$$

$$= \frac{2228.57}{[2 + 137.4t_1(\text{K/W})]}\,\text{W}$$

The rate heat transfer through the wall computed by varying the foam thickness from 1 cm to 10 cm and this calculation performed using the Excel spread sheet; the computed values are tabulated and plotted as shown in below

foam thickness (m)	Rate of Heat transfer (W)
0.01	660.513
0.02	469.371
0.03	364.027
0.04	297.301
0.05	251.248
0.06	217.549
0.07	191.821
0.08	171.534
0.09	155.128
0.1	141.586

Step 9 of 9

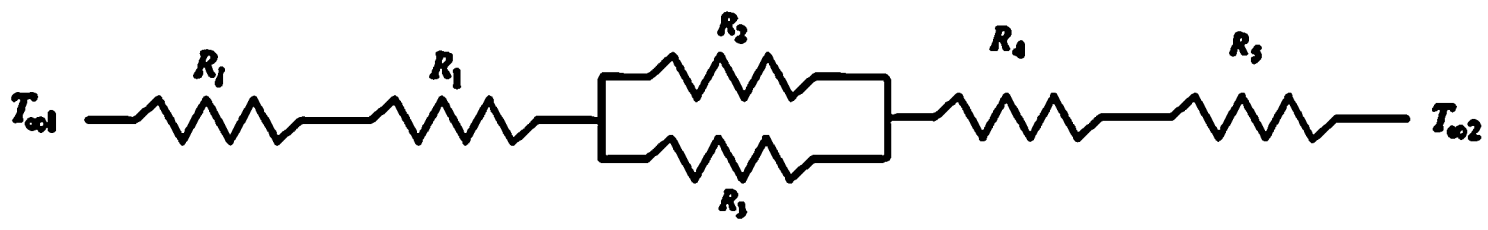


Question:

A 12-m-long and 5-m-high wall is constructed of two layers of 1-cm-thick sheetrock (k =0.17 W/m·K) spaced 16 cm by wood studs (k = 0.11 W/m·K) whose cross section is 16 cm × 5 cm. The studs are placed vertically 60 cm apart, and the space between them is filled with fiberglass insulation (k =0.034 W/m·K). The house is maintained at 20°C and the ambient temperature outside is−9°C. Taking the heat transfer coefficients at the inner andouter surfaces ofthe house to be 8.3 and 34 W/m2·K. respectively, determine (*a*) the thermal resistance of the wall considering a representative section of it and (*b*) the rate of heat transfer through the wall

Step 1 of 7

Draw the thermal resistance network for the representative section of the wall:



Step 2 of 7

(a)

Consider the home is $1\ \text{m}$ deep.

Find the individual resistances as shown.

Calculate the convectional thermal resistance at inner surface of house by using the following relation.

$$R_i = \frac{1}{h_i A}$$
$$= \frac{1}{h_i \times (d \times l)}$$

Here, h_i is the heat transfer coefficient at inner surface of home and A is surface area.

Substitute $8.3\ \text{W/m}^2\cdot^\circ\text{C}$ for h_i, 1m for d and $65\ \text{cm} = 0.65\ \text{m}$ for l

$$R_i = \frac{1}{8.3 \times (1 \times 0.65)}$$
$$= 0.1854\ ^\circ\text{C/W}$$

Calculate the conductive thermal resistance of sheetrock by using the following relation.

$$R_1 = R_4$$
$$= \frac{L}{k_{\text{sheetrock}} A}$$
$$= \frac{L}{k_{\text{sheetrock}} \times A}$$

Here, k is coefficient of thermal conductivity and L is length of the room.

Substitute $0.01\ \text{m}$ for L, $0.17\ \text{W/m}^\circ\text{C}$ for $k_{\text{sheetrock}}$ and $0.65\,\text{m}^2$ for A.

$$R_1 = R_4$$
$$= \frac{0.01}{0.17 \times 0.65}$$
$$= 0.0905\ ^\circ\text{C/W}$$

Step 3 of 7

Calculate the conductive thermal resistance of studs by using the following relation.

$$R_2 = \frac{L}{k_{stud} A_{stud}}$$
$$= \frac{L}{k \times A_{stud}}$$

Here, k_{stud} is coefficient of thermal conductivity of stud and A_{stud} is area of the stud.

Substitute $12\ \text{m}$ for L, $0.11\ \text{W/m}^\circ\text{C}$ for k_{stud} and $0.12\,\text{m} \times 0.05\,\text{m} = 0.60\ \text{m}^2$ for A_{stud}.

$$R_2 = \frac{0.12}{0.11 \times 0.05}$$
$$= 21.82\ ^\circ\text{C/W}$$

Calculate the conductive thermal resistance of fiber glass by using the following relation.

$$R_3 = \frac{L}{k_{\text{fiber glass}} A}$$
$$= \frac{L}{k_{\text{fiber glass}} \times A}$$

Here, $k_{\text{fiber glass}}$ is coefficient of thermal conductivity of fiber glass.

Substitute $0.12\ \text{m}$ for L, $0.034\,\text{W/m}^\circ\text{C}$ for $k_{\text{fiber glass}}$ and $0.65\ \text{m}^2$ for A.

$$R_3 = \frac{0.12}{0.037 \times 0.65}$$
$$= 5.882\ ^\circ\text{C/W}$$

Step 4 of 7

Calculate the convectional thermal resistance at outer surface of house by using the following relation.

$$R_o = \frac{1}{h_o A}$$
$$= \frac{1}{h_o \times (d \times l)}$$

Here, h_o is the heat transfer coefficient at outer surface of home.

Substitute $34\ \text{W/m}^2\cdot^\circ\text{C}$ for h_o, $0.65\ \text{m}^2$ for A.

$$R_o = \frac{1}{34 \times 0.65}$$

Substitute $34\ \text{W/m}^2\cdot^\circ\text{C}$ for h_o , $0.65\ \text{m}^2$ for A.

$$R_o = \frac{1}{34\times 0.65}$$
$$= 0.0452\ ^\circ\text{C/W}$$

Step 5 of 7

Calculate the resultant thermal resistance of stud and fiber glass.

$$\frac{1}{R_{sheetrock}} = \frac{1}{R_2} + \frac{1}{R_3}$$

Substitute $21.82\ ^\circ\text{C/W}$ for R_2 and $5.882\ ^\circ\text{C/W}$ for R_3.

$$\frac{1}{R_{sheetrock}} = \frac{1}{21.82} + \frac{1}{5.882}$$
$$R_{sheetrock} = 4.633\ ^\circ\text{C/W}$$

Step 6 of 7

Find the total thermal resistance by sung the following relation.

$$R_{total} = R_i + R_1 + R_{sheetrock} + R_4 + R_o$$

Substitute $0.1854\ ^\circ\text{C/W}$ for R_i , $0.0905\ ^\circ\text{C/W}$ for R_1, $4.633\ ^\circ\text{C/W}$ for $R_{sheetrock}$, $0.0905\ ^\circ\text{C/W}$ for R_4 , and $0.0452\ ^\circ\text{C/W}$ for R_o .

$$R_{total} = 0.1854 + 0.0905 + 4.633 + 0.0905 + 0.0452$$
$$= 5.0446\ ^\circ\text{C/W}$$

Therefore, thermal resistance of wall is $\boxed{5.0446\ ^\circ\text{C/W}}$.

Calculate the rate of heat transfer through the wall section.

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{total}}$$

Here, T_1 is the temperature of house and $T_{\infty 2}$ is the temperature of outer surface.

Substitute 20°C for T_1 , -9°C for $T_{\infty 2}$, and $5.0446\ ^\circ\text{C/W}$ for R_{total} .

$$\dot{Q} = \frac{20-(-9)}{5.0446}$$
$$= \frac{29}{5.0446}$$
$$= 5.748\ \text{W}$$

Step 7 of 7

(b)

Calculate the heat transfer through entire wall.

$$Q_{total} = \dot{Q}\times\left(\frac{L\times D}{d\times l}\right)$$

Here, D is height of the wall.

Substitute $5.748\ \text{W}$ for $\dot{Q}$, $12\ \text{m}$ for L , 5m for D , 1m for d , and $0.65\ \text{m}$ for l .

$$Q_{total} = 5.748\times\left(\frac{12\times 5}{1\times 0.65}\right)$$
$$= 530.65\ \text{W}$$

Therefore, the rate of heat transfer through the wall is $\boxed{530.65\ \text{W}}$.

Question: 17.055P

A 10-cm-thick wall is to be constructed with2.5 long wood studs (k =0.11 W/m·K) that have a crosssection of 10 cm × 10 cm. At some point the builder ran out ofthose studs and started using pairs of 2.5-m-long wood studsthat have a cross section of 5 cm × 10 cm nailed to eachother in stead. The manganese steel nails (k =50 W/m·K) are 10 cm long and have a diameter of 0.4 cm. A total of 50 nailsare used to connect the two studs, which are mounted to the wall such that the nails cross the wall. The temperature difference between the inner and outer surfaces of the wall is 8°C. Assuming the thermal contact resistance between the two layers tobenegligible, determine the rate of heat transfer (*a*) through a solid stud and (*b*) through a stud pair of equal length andwidth nailed toeach other. (*c*) Also determine the effective conductivity of the nailed stud pair.

Step 1 of 5

Given:
Thickness of wall = 10cm
Length of studs = 2.5m
Thermal conductivity for wood studs = 0.11 W/m·°C
Thermal conductivity for manganese steel nails = 50 W/m·°C
Cross section of studs=5cmX 10cm
Length of manganeses steel = 10cm
Diameter of manganeses steel = 0.4cm

Step 2 of 5

If the heat transfer is steady since there is no indication of change with time and heat transfer can be approximated as being one-dimensional since it is predominantly in the x direction

Step 3 of 5

(a) The thermal resistance and heat transfer rate through the solid stud are

$$R_{stud} = \frac{L}{kA} = \frac{0.1\text{ m}}{(0.11\text{ W/m.°C})(0.25\text{ m}^2)} = 3.636\text{ °C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8°C}{3.636\text{ °C/W}}$$

$$\boxed{\dot{Q} = 2.2\text{ W}}$$

Step 4 of 5

(b) The thermal resistances of stud pair and nails are in parallel

$$A_{nails} = 50\frac{\pi D^2}{4} = 50\left[\frac{\pi(0.004\text{ m})^2}{4}\right] = 0.000628\text{ m}^2$$

$$R_{nails} = \frac{L}{kA} = \frac{0.1\text{ m}}{(50\text{ W/m.°C})(0.000628\text{ m}^2)} = 3.18\text{ °C/W}$$

$$R_{stud} = \frac{L}{kA} = \frac{0.1\text{ m}}{(0.11\text{ W/m.°C})(0.25 - 0.000628\text{ m}^2)} = 3.65\text{ °C/W}$$

$$\frac{1}{R_{total}} = \frac{1}{R_{stud}} + \frac{1}{R_{nails}} = \frac{1}{3.65} + \frac{1}{3.18} \longrightarrow R_{total} = 1.70\text{ °C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8°C}{1.70\text{ °C/W}}$$

$$= 4.7\text{ W}$$

$$\boxed{\dot{Q} = 4.7\text{ W}}$$

Step 5 of 5

(c) The effective conductivity of the nailed stud pair can be determined from

$$\dot{Q} = k_{eff} A\frac{\Delta T}{L}$$

$$k_{eff} = \frac{\dot{Q}L}{\Delta TA} = \frac{(4.7\text{ W})(0.1\text{ m})}{(8°C)(0.25\text{ m}^2)}$$

$$k_{eff} = 0.235\text{ W/m°C}$$

$$\boxed{k_{eff} = 0.235\text{ W/m°C}}$$

Question: 17.056P pg 1

A 10-in-thick., 30-ft-long, and 10-ft-high wall is to be constructed using 9-in-long solid bricks (k = 0.40 Btu/h·ft·°F) of cross section 7 in × 7 in, or identical size bricks withnine square air holes (k = 0.015 Btu/h·ft·°F) that are 9 inlong and have a cross section of 1.5 in× 1.5 in . There isa 0.5- in-thick plaster layer (k = 0.10 Btu/h·ft·°F) between twoadjacent bricks on all four sides and on both sides of the wall. The house is maintained at 80°F and the ambient temperature outside is35°F. Taking the heat transfer coefficients at the inner andouter surfaces of the wall to be 1.5 and 6 Btu/h·ft2·°F. respectively, determine the rate of heat transfer through the wall constructed of(*a*) solid bricks and (*b*) bricks with air holes.

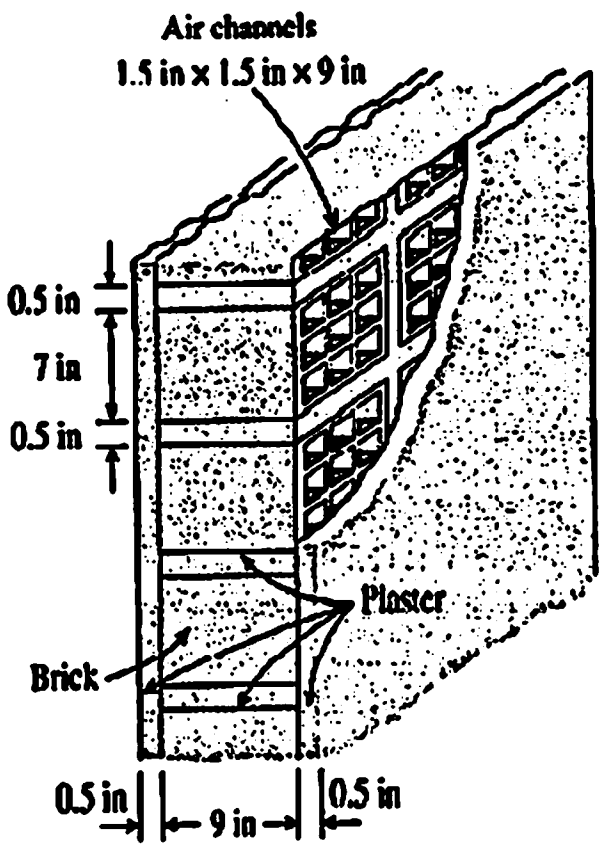


FIGURE P17–56E

Step 1 of 12

Given data:

Cross sectional are of the wall, $A = 30 \times 10\,\text{ft}^2 = 300\,\text{ft}^2$

Thickness of the plaster, $t_1 = t_5 = 0.5\text{in}$

Thickness of the plaster between two bricks, $t_2 = t_4 = 9\text{in}$

Thickness of the brick, $t_3 = 9\text{in}$

Length of the plaster, $L_1 = L_5 = 7.5\text{in}$

Length of the plaster between two bricks, $L_2 = L_4 = 0.25\text{in}$

Length of the bricks, $L_3 = 7\text{in}$

Thermal conductivity of the plaster, $k_1 = k_2 = k_4 = k_5 = 0.1\,\text{Btu/h-ft-°F}$

Thermal conductivity of the brick, $k_3 = 0.4\,\text{Btu/h-ft-°F}$

Indoor temperature, $T_{\infty,1} = 80°\text{F}$

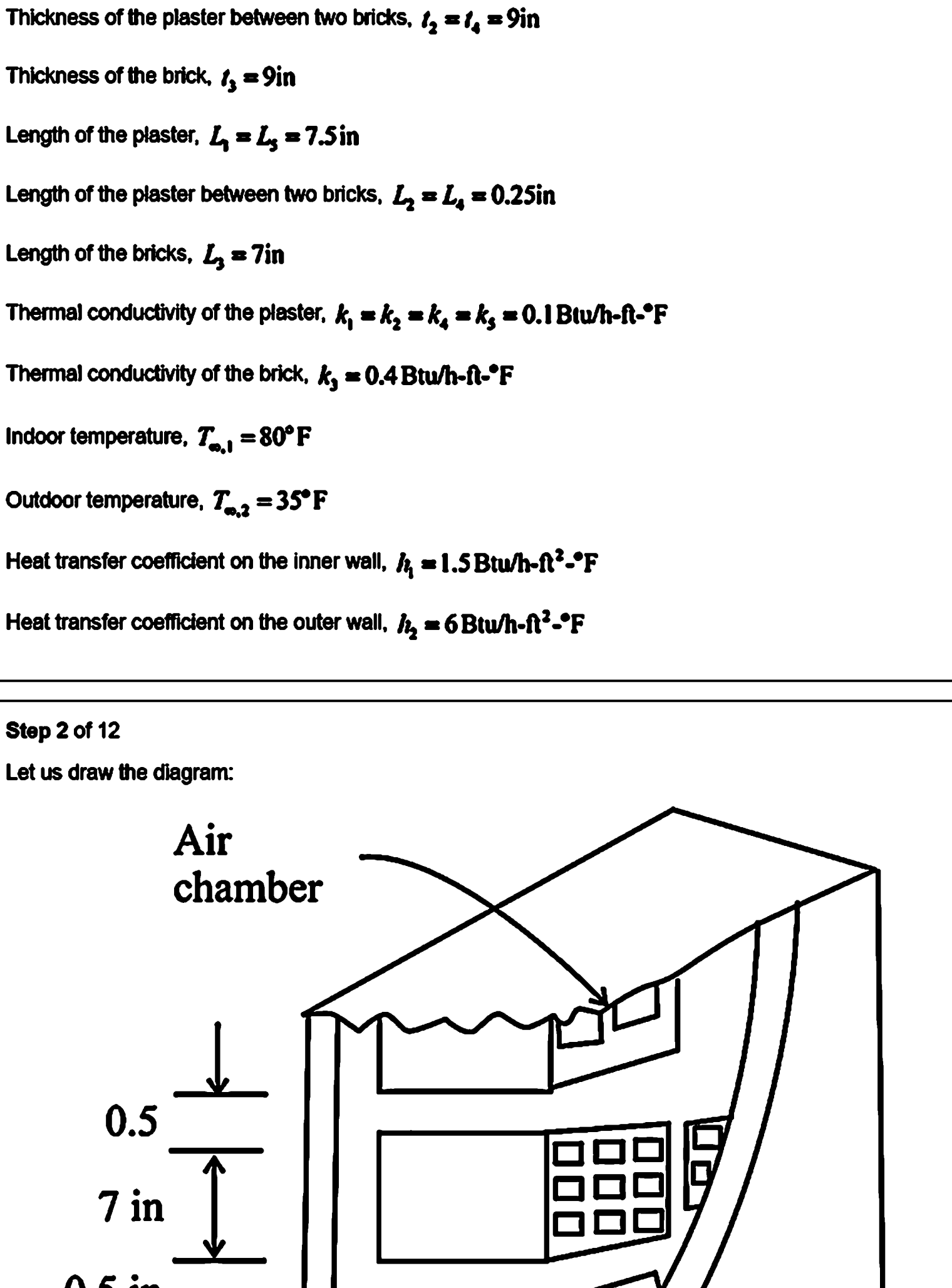
Outdoor temperature, $T_{\infty,2} = 35°\text{F}$

Heat transfer coefficient on the inner wall, $h_1 = 1.5\,\text{Btu/h-ft}^2\text{-°F}$

Heat transfer coefficient on the outer wall, $h_2 = 6\,\text{Btu/h-ft}^2\text{-°F}$

Step 2 of 12

Let us draw the diagram:



Step 3 of 12

Step 3 of 12

Step 4 of 12

With Solid Bricks:

There is a pattern in the construction of the wall that repeats itself every 7.5-in distance in the vertical direction. The representative surface area of the wall is $A = \frac{7.5}{12} \times \frac{7.5}{12} \text{ft}^2 = 0.3906 \text{ft}^2$

Assuming any cross section of the wall normal to the X-direction to be isothermal, the thermal resistance network for the representative section of the wall becomes as shown in the Fig above.

The individual resistances are evaluated as:

$$R_i = R_{conv,1}$$
$$= \frac{1}{h_1 A_1}$$
$$= \frac{1}{1.5(\text{Btu/h-ft}^2\text{-}^\circ\text{F}) \times 0.3906(\text{ft}^2)}$$
$$= 1.7067 \text{ h-}^\circ\text{F/Btu}$$

Step 5 of 12

$$R_1 = R_5$$
$$= R_{plaster,side}$$
$$= \frac{t_1}{k_1 A_1}$$
$$= \frac{0.5/12(\text{ft})}{0.1(\text{Btu/h-ft-}^\circ\text{F}) \times 0.3906(\text{ft}^2)}$$
$$= 1.0667 \text{ h-}^\circ\text{F/Btu}$$

$$R_2 = R_4$$
$$= R_{plaster,center}$$
$$= \frac{t_2}{k_2 A_2}$$
$$= \frac{9/12(\text{ft})}{0.1(\text{Btu/h-ft-}^\circ\text{F}) \times (7.5/12 \times 0.25/12 \text{ft}^2)}$$
$$= 576 \text{ h-}^\circ\text{F/Btu}$$

$$R_3 = R_{brick}$$
$$= \frac{t_3}{k_3 A_3}$$
$$= \frac{9/12(\text{ft})}{0.4(\text{Btu/h-ft-}^\circ\text{F}) \times (7/12 \times 7/12 \text{ft}^2)}$$
$$= 5.51 \text{ h-}^\circ\text{F/Btu}$$

$$R_o = R_{conv,2}$$
$$= \frac{1}{h_2 A_2}$$
$$= \frac{1}{6(\text{Btu/h-ft}^2\text{-}^\circ\text{F}) \times 0.3906(\text{ft}^2)}$$
$$= 0.4267 \text{ h-}^\circ\text{F/Btu}$$

Step 6 of 12

The three resistances R_2, R_3 and R_4 in the middle are parallel, and their equivalent resistance is determined from

$$\frac{1}{R_{mid}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4}$$
$$= \frac{1}{576(\text{h-}^\circ\text{F/Btu})} + \frac{1}{5.51(\text{h-}^\circ\text{F/Btu})} + \frac{1}{576(\text{h-}^\circ\text{F/Btu})}$$
$$= 0.185 \text{ Btu/h-}^\circ\text{F}$$

This gives

$$R_{mid} = \frac{1}{0.185(\text{Btu/h-}^\circ\text{F})}$$
$$= 5.406 \text{ h-}^\circ\text{F/Btu}$$

Now all the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_{mid} + R_5 + R_o$$
$$= \left\{ \begin{array}{l} 1.7067(\text{h-}^\circ\text{F/Btu}) + 1.0667(\text{h-}^\circ\text{F/Btu}) + 5.406(\text{h-}^\circ\text{F/Btu}) \\ \qquad + 1.0667(\text{h-}^\circ\text{F/Btu}) + 0.4267(\text{h-}^\circ\text{F/Btu}) \end{array} \right\}$$
$$= 9.674 \text{ h-}^\circ\text{F/Btu}$$

Step 7 of 12

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(80 - 35)^\circ\text{F}}{9.674(\text{h-}^\circ\text{F/Btu})}$$
$$= 4.652 \text{ Btu/h}$$

4.652 Btu/h Of heat is transferring through per 0.3906ft^2 surface area

And the total heat flux through the wall can be computed as

$$\dot{q} = \frac{\dot{Q}}{A}$$
$$= \frac{4.652(\text{Btu/h})}{0.3906(\text{ft}^2)}$$

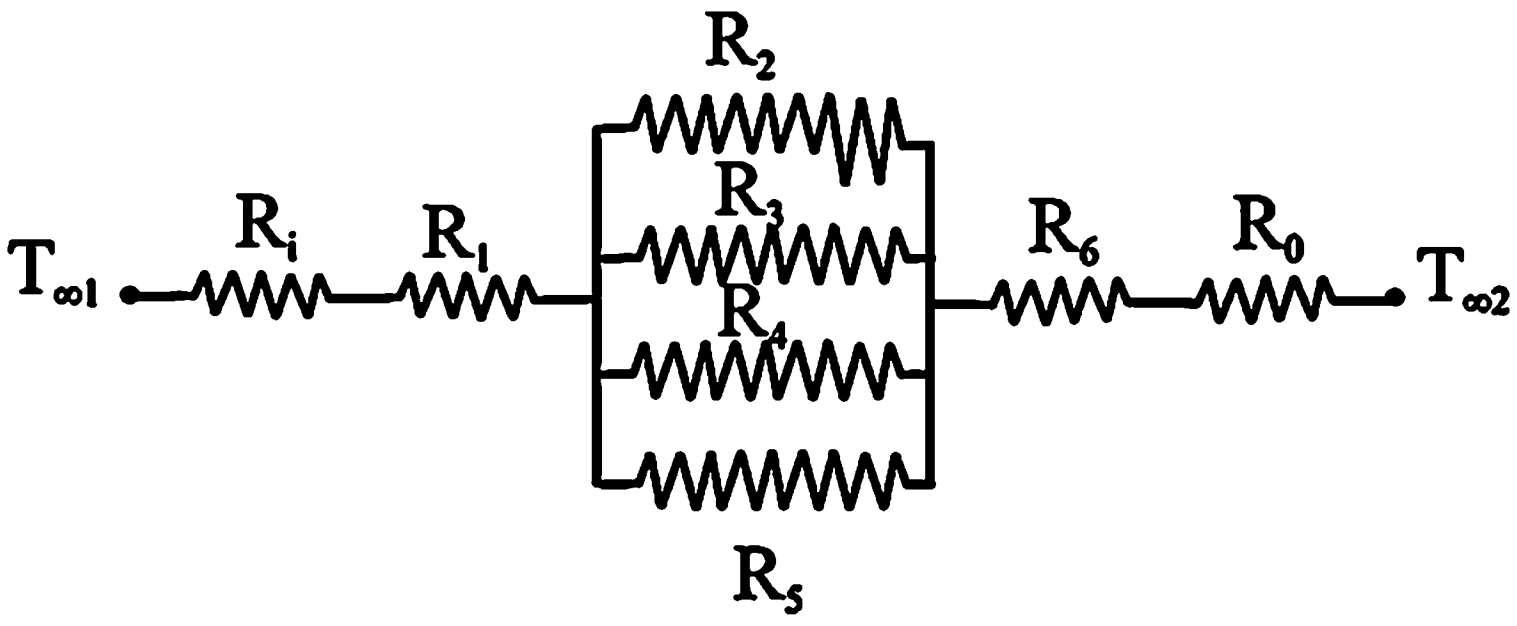
$$\dot{q} = \frac{\dot{Q}}{A}$$
$$= \frac{4.652(\text{Btu/h})}{0.3906(\text{ft}^2)}$$
$$= 12\,\text{Btu/ft}^2\text{-h}$$

The total area of the wall is $A = 30 \times 10 = 300\text{ft}^2$. Then the rate of heat transfer through the entire wall becomes

$$\dot{Q}_{total} = 12(\text{Btu/ft}^2\text{-h}) \times 300(\text{ft}^2)$$
$$= 3600\,\text{Btu/h}$$

$$\boxed{\dot{Q}_{total} = 3600\,\text{Btu/h}}$$

Step 8 of 12



Step 9 of 12

Bricks with square air holes:

Area occupied by Air holes in each brick can be calculated as

$$A_{airholes} = 9 \times (1.5/12 \times 1.5/12)\text{ft}^2$$
$$= 0.140625\text{ft}^2$$

The area occupied by the brick can be calculated as

$$A_{brick} = (7/12 \times 7/12)\text{ft}^2 - 0.140625\text{ft}^2$$
$$= 0.19965\text{ft}^2$$

The individual resistances are evaluated as

$$R_i = R_{conv,1}$$
$$= \frac{1}{h_1 A_1}$$
$$= \frac{1}{1.5(\text{Btu/h-ft}^2\text{-°F}) \times 0.3906(\text{ft}^2)}$$
$$= 1.7067\,\text{h-°F/Btu}$$

$$R_1 = R_6$$
$$= R_{plaster,side}$$
$$= \frac{l_1}{k_1 A_1}$$
$$= \frac{0.5/12(\text{ft})}{0.1(\text{Btu/h-ft-°F}) \times 0.3906(\text{ft}^2)}$$
$$= 1.0667\,\text{h-°F/Btu}$$

$$R_2 = R_5$$
$$= R_{plaster,center}$$
$$= \frac{l_2}{k_2 A_2}$$
$$= \frac{9/12(\text{ft})}{0.1(\text{Btu/h-ft-°F}) \times (7.5/12 \times 0.25/12\text{ft}^2)}$$
$$= 576\,\text{h-°F/Btu}$$

Step 10 of 12

$$R_3 = R_{brick}$$
$$= \frac{l_3}{k_3 A_3}$$
$$= \frac{9/12(\text{ft})}{0.4(\text{Btu/h-ft-°F}) \times (0.19965\text{ft}^2)}$$
$$= 9.391\,\text{h-°F/Btu}$$

$$R_4 = R_{airholes}$$
$$= \frac{l_4}{k_4 A_4}$$
$$= \frac{9/12(\text{ft})}{0.015(\text{Btu/h-ft-°F}) \times (0.140625\text{ft}^2)}$$
$$= 355.55\,\text{h-°F/Btu}$$

$$R_o = R_{conv,2}$$
$$= \frac{1}{h_2 A_2}$$
$$= \frac{1}{6(\text{Btu/h-ft}^2\text{-°F}) \times 0.3906(\text{ft}^2)}$$
$$= 0.4267\,\text{h-°F/Btu}$$

Step 11 of 12

The four resistances R_2, R_3, R_4 and R_5 in the middle are parallel, and their equivalent resistance is determined from

$$\frac{1}{R} = \frac{1}{R} + \frac{1}{R} + \frac{1}{R} + \frac{1}{R}$$

The four resistances R_2, R_3, R_4 and R_5 in the middle are parallel, and their equivalent resistance is determined from

$$\begin{aligned}\frac{1}{R_{mid}} &= \frac{1}{R_2}+\frac{1}{R_3}+\frac{1}{R_4}+\frac{1}{R_5}\\ &= \frac{1}{576(\text{h-°F/Btu})}+\frac{1}{9.391(\text{h-°F/Btu})}+\frac{1}{355.55(\text{h-°F/Btu})}+\frac{1}{576(\text{h-°F/Btu})}\\ &= 0.1065\,\text{Btu/h-°F}\end{aligned}$$

This gives

$$\begin{aligned}R_{mid} &= \frac{1}{0.1065(\text{Btu/h-°F})}\\ &= 9.4\,\text{h-°F/Btu}\end{aligned}$$

Now all the resistances are in series, and the total resistance is

$$\begin{aligned}R_{total} &= R_i + R_1 + R_{mid} + R_6 + R_o\\ &= \left[\begin{array}{l}1.7067(\text{h-°F/Btu})+1.0667(\text{h-°F/Btu})+9.4(\text{h-°F/Btu})\\ \qquad\qquad +1.0667(\text{h-°F/Btu})+0.4267(\text{h-°F/Btu})\end{array}\right]\\ &= 13.657\,\text{h-°F/Btu}\end{aligned}$$

Step 12 of 12

Then the steady rate of heat transfer through the wall becomes

$$\begin{aligned}\dot{Q} &= \frac{(T_{\infty,1}-T_{\infty,2})}{R_{total}}\\ &= \frac{(80-35)°\text{F}}{13.657(\text{h-°F/Btu})}\\ &= 3.3\,\text{Btu/h}\end{aligned}$$

$3.3\,\text{Btu/h}$ Of heat is transferring through per 0.3906ft^2 surface area

And the total heat flux through the wall can be computed as

$$\begin{aligned}\dot{q} &= \frac{\dot{Q}}{A}\\ &= \frac{3.3(\text{Btu/h})}{0.3906(\text{ft}^2)}\\ &= 8.436\,\text{Btu/ft}^2\text{-h}\end{aligned}$$

The total area of the wall is $A = 30\times 10 = 300\text{ft}^2$. Then the rate of heat transfer through the entire wall becomes

$$\begin{aligned}\dot{Q}_{total} &= 8.436(\text{Btu/ft}^2\text{-h})\times 300(\text{ft}^2)\\ &= 2530.8\,\text{Btu/h}\end{aligned}$$

$$\boxed{\dot{Q}_{total} = 2530.8\,\text{Btu/h}}$$

Question: 17.057P pg 1

Consider a 5-m-high, 8-m-long, and 0.22-m-thick wall whose representative cross section is as given in the figure. The thermal conductivities of various materials used, in W/m·K, are $kA = kF = 2, kB = 8$, $kc = 20$, $kD = 15$, and $kE = 35$. The left and right surfaces of the wall are maintained at uniform temperatures of 300°C and 100°C, respectively. Assuming heat transfer through the wall to be one-dimensional, determine (*a*) the rate of heat transfer through the wall; (*b*) the temperature at the point where the sections *B*, *D*, and *E* meet; and (*c*) the temperature drop across the section *F*. Disregard any contact resistances at the interfaces.

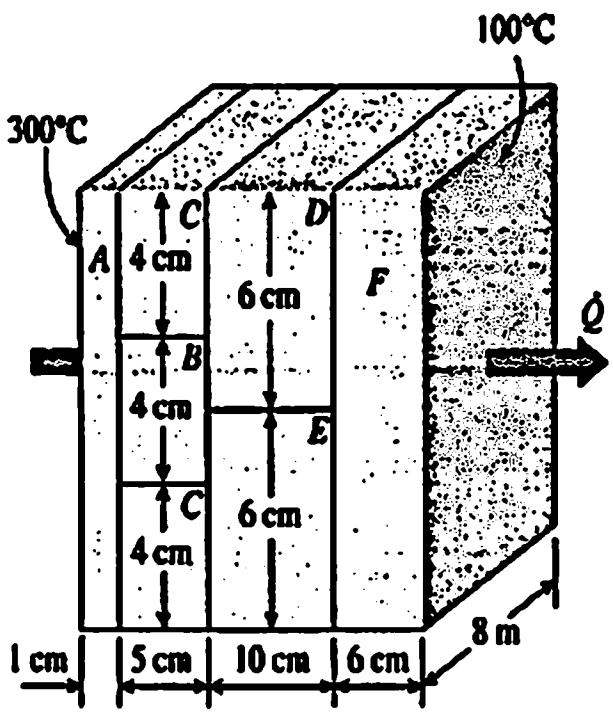
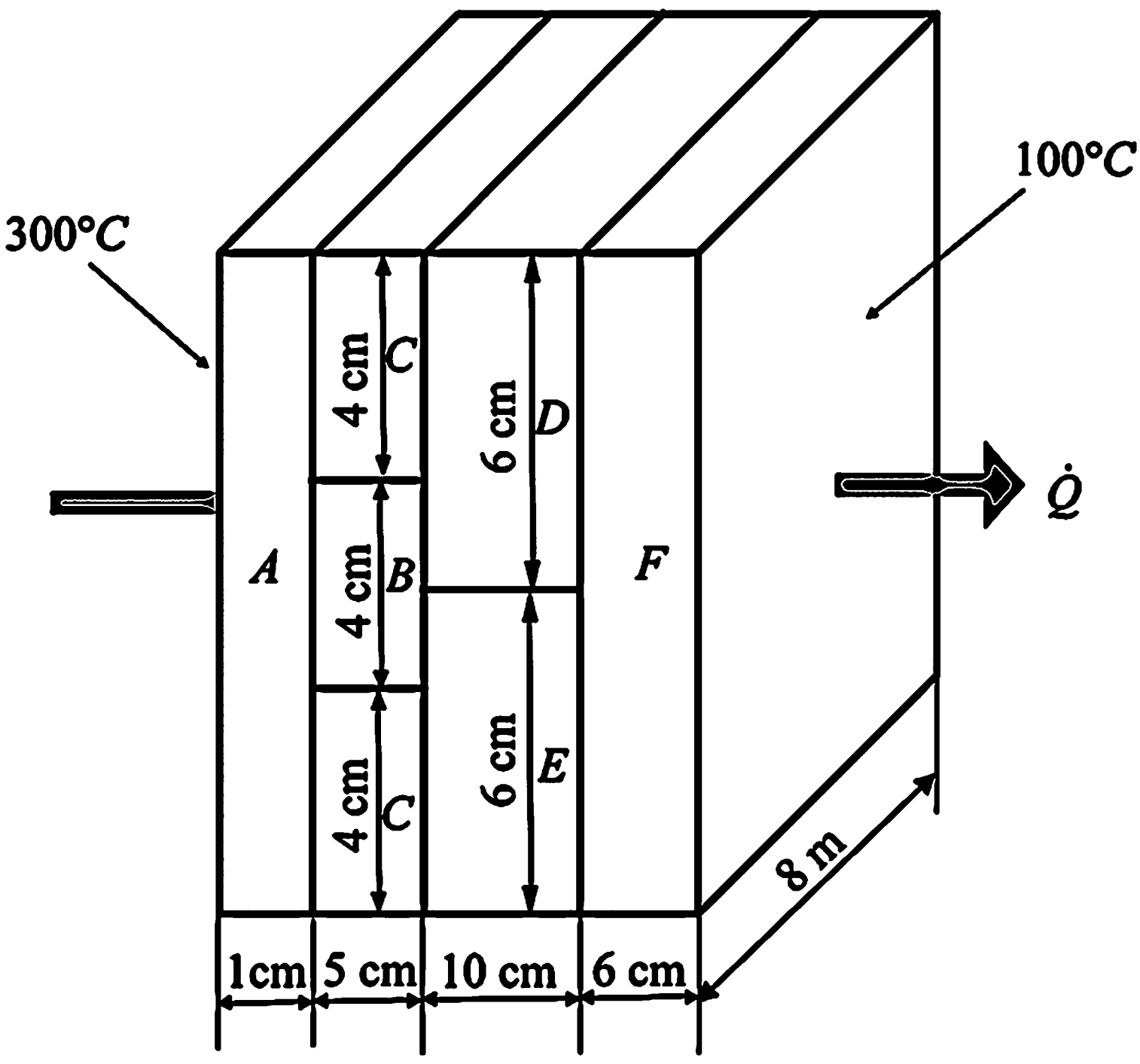


FIGURE P17–57

Step 1 of 13

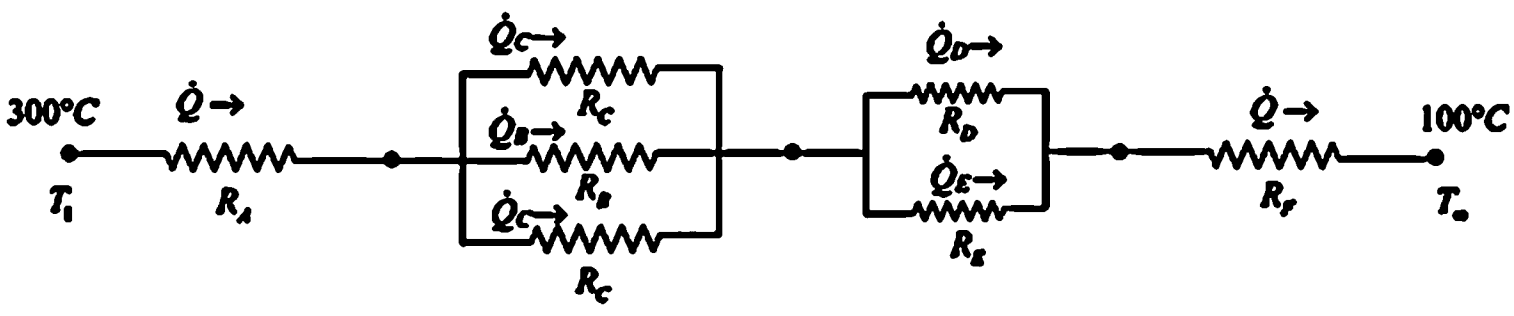
a)

Draw the cross section of $0.22\ \text{m}$ thick plane wall shown as follows:



Step 2 of 13

Draw the thermal resistance network due to conduction as shown below.



Step 3 of 13

Calculate the thermal resistance in wall A due to conduction, by using the equation as shown below.

$$R_A = \left(\frac{L_A}{k_A A_A}\right)$$

Here, the thickness of the wall A is L_A, thermal conductivity of wall A is k_A and the area normal to the direction of heat flow through A is A_A.

Substitute $0.01\ \text{m}$ for L_A, $(0.12\times 8)\ \text{m}^2$ for A_A and $2\ \text{W/m.°C}$ for k_A.

$$R_A = \frac{0.01\ \text{m}}{(2\ \text{W/m.°C})(0.12\times 8\ \text{m}^2)}$$
$$= 0.0052\ \text{°C/W}$$

Step 4 of 13

Calculate the thermal resistance in wall B due to conduction, by using the equation as shown below.

$$R_B = \left(\frac{L_B}{k_B A_B}\right)$$

Here, the thickness of the wall B is L_B, thermal conductivity of wall B is k_B and the area normal to the direction of heat flow through B is A_B.

Substitute $0.05\ \text{m}$ for L_B, $(0.04\times 8)\ \text{m}^2$ for A_B and $8\ \text{W/m.°C}$ for k_B.

$$R_B = \frac{0.05\ \text{m}}{(8\ \text{W/m.°C})(0.04\times 8\ \text{m}^2)}$$
$$= 0.01953\ \text{°C/W}$$

$$R_B = \frac{0.05\ \text{m}}{(8\ \text{W/m.°C})(0.04\times 8\ \text{m}^2)}$$

17.057P pg 2

$$= 0.01953\ \text{°C/W}$$

Step 5 of 13

Calculate the thermal resistance in wall C due to conduction, by using the equation as shown below.

$$R_C = \left(\frac{L_C}{k_C A_C}\right)$$

Here, the thickness of the wall C is L_C, thermal conductivity of wall C is k_C and the area normal to the direction of heat flow through C is A_C.

Substitute $0.05\ \text{m}$ for L_C, $(0.04\times 8)\ \text{m}^2$ for A_C and $20\ \text{W/m.°C}$ for k_C.

$$R_C = \frac{0.05\ \text{m}}{(20\ \text{W/m.°C})(0.04\times 8\ \text{m}^2)}$$

$$= 0.00781\ \text{°C/W}$$

Step 6 of 13

Calculate the thermal resistance in wall D due to conduction, by using the equation as follows:

$$R_D = \left(\frac{L_D}{k_D A_D}\right)$$

Here, the thickness of the wall D is L_D, thermal conductivity of wall D is k_D and the area normal to the direction of heat flow through D is A_D.

Substitute $0.1\ \text{m}$ for L_D, $(0.06\times 8)\ \text{m}^2$ for A_D and $15\ \text{W/m.°C}$ for k_D.

$$R_D = \frac{0.1\ \text{m}}{(15\ \text{W/m.°C})(0.06\times 8\ \text{m}^2)}$$

$$= 0.01388\ \text{°C/W}$$

Calculate the thermal resistance in wall E due to conduction, by using the equation as follows:

$$R_E = \left(\frac{L_E}{k_E A_E}\right)$$

Here, the thickness of the wall E is L_E, thermal conductivity of wall E is k_E and the area normal to the direction of heat flow through E is A_E.

Substitute $0.1\ \text{m}$ for L_E, $(0.06\times 8)\ \text{m}^2$ for A_E and $35\ \text{W/m.°C}$ for k_E.

$$R_E = \frac{0.1\ \text{m}}{(35\ \text{W/m.°C})(0.06\times 8\ \text{m}^2)}$$

$$= 0.00595\ \text{°C/W}$$

Step 7 of 13

Calculate the thermal resistance in wall F due to conduction, by using the equation as shown below.

$$R_F = \left(\frac{L_F}{k_F A_F}\right)$$

Here, the thickness of the wall F is L_F, thermal conductivity of the wall F is k_F and the area normal to the direction of heat flow through F is A_F.

Substitute $0.06\ \text{m}$ for L_F, $(0.12\times 8)\ \text{m}^2$ for A_F and $2\ \text{W/m.°C}$ for k_F.

$$R_F = \frac{0.06\ \text{m}}{(2\ \text{W/m.°C})(0.12\times 8\ \text{m}^2)}$$

$$= 0.03125\ \text{°C/W}$$

Step 8 of 13

The three resistances R_C, R_B and R_C in the middle are parallel.

Calculate the equivalent resistance of R_C, R_B and R_C, by using the equation as shown below.

$$\frac{1}{R_{mid,1}} = \frac{1}{R_C} + \frac{1}{R_B} + \frac{1}{R_C}$$

Substitute the known values in the above equation.

$$\frac{1}{R_{mid,1}} = \frac{1}{0.00781} + \frac{1}{0.01953} + \frac{1}{0.00781}$$

$$\frac{1}{R_{mid,1}} = 307.2852$$

$$R_{mid,1} = 0.003254\ \text{°C/W}$$

Step 9 of 13

The two resistances R_D and R_E in the middle are parallel.

Calculate the equivalent resistance of R_D and R_E by using the equation as shown below.

$$\frac{1}{R_{mid,2}} = \frac{1}{R_D} + \frac{1}{R_E}$$

Substitute the known values in the above equation.

$$\frac{1}{R_{mid,2}} = \frac{1}{0.01388} + \frac{1}{0.00595}$$

$$\frac{1}{R_{mid,2}} = 240.1133$$

$$R_{mid,2} = 0.004164$$

$$\frac{1}{R_{mid,2}} = 240.1133$$

$$R_{mid,2} = 0.004164$$

Step 10 of 13

Calculate the total thermal resistance.

$$R_{total} = R_A + R_{mid,1} + R_{mid,2} + R_F$$

Substitute the known values in the above equation.

$$R_{total} = 0.0052 + 0.003254 + 0.004164 + 0.03125$$
$$= 0.0438\ ^\circ\text{C/W}$$

Step 11 of 13

Calculate the rate of heat transfer by using the equation as shown below.

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{total}}$$

Here, the temperature of the left surface is T_1 and the temperature of the right surface is T_∞.

Substitute $300\ ^\circ\text{C}$ for T_1, $100\ ^\circ\text{C}$ for T_∞ and $0.0438\ ^\circ\text{C/W}$ for R_{total}.

$$\dot{Q} = \frac{(300 - 100)^\circ\text{C}}{0.0438\ ^\circ\text{C/W}}$$
$$= 4566.21\ \text{W}$$
$$= 4.56\ \text{kW}\ (\text{for a } 0.12\ \text{m} \times 8\ \text{m section})$$

Calculate the rate of heat transfer through entire wall $(\text{for } 5\ \text{m} \times 8\ \text{m section})$.

$$\dot{Q}_{total} = 4.56\ \text{kW} \times \frac{5\ \text{m}}{0.12\ \text{m}}$$
$$= 190\ \text{kW}$$
$$= 1.9 \times 10^5\ \text{W}$$

Therefore, the rate of heat transfer through the wall is $\boxed{1.9 \times 10^5\ \text{W}}$.

b)

Calculate the total thermal resistance between left surface and the point where the sections *B*, *D*, and *E* meet.

$$R_{total} = R_A + R_{mid,3}$$

Substitute the known values in the above equation.

$$R_{total} = 0.0052 + 0.003254$$
$$= 0.008454\ ^\circ\text{C/W}$$

Step 12 of 13

Calculate the temperature at the point where the sections *B*, *D*, and *E* meet.

$$\dot{Q} = \frac{T_1 - T}{R_{total}}$$
$$T_1 - T = \dot{Q} R_{total}$$
$$T = T_1 - \dot{Q} R_{total}$$

Substitute $300\ ^\circ\text{C}$ for T_1, $4566.21\ \text{W}$ for $\dot{Q}$ and $0.008454\ ^\circ\text{C/W}$ for R_{total}.

$$T = 300^\circ\text{C} - (4566.21\ \text{W} \times 0.008454\ ^\circ\text{C/W})$$
$$= 300 - 38.60$$
$$= 261.40^\circ C$$

Therefore, the temperature at *B*, *D*, and *E* interface is $\boxed{261.40^\circ\text{C}}$.

Step 13 of 13

c)

Calculate the temperature drop across the section *F* by using the equation as shown below.

$$\dot{Q} = \frac{\Delta T}{R_F}$$
$$\Delta T = \dot{Q} R_F$$

Substitute $4566.21\ \text{W}$ for $\dot{Q}$ and $0.03125\ ^\circ\text{C/W}$ for R_F.

$$\Delta T = \dot{Q} R_F$$
$$= (4566.21\ \text{W}) \times (0.03125\ ^\circ\text{C/W})$$
$$= 142.70^\circ\text{C}$$

Therefore, the temperature drop across the section *F* is $\boxed{142.70^\circ\text{C}}$.

Question: 17.058P pg 1

Repeat Prob. 17-57 assuming that the thermal contact resistance at the interfaces *D-F* and *E-F* is 0.00012 m2·°C/W.

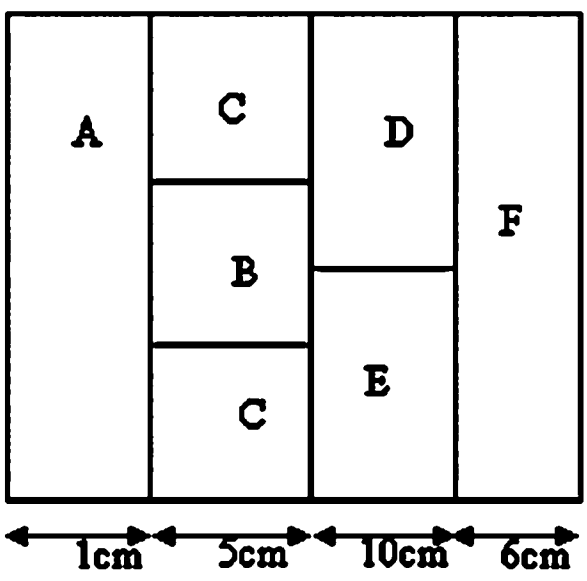
Step 1 of 9

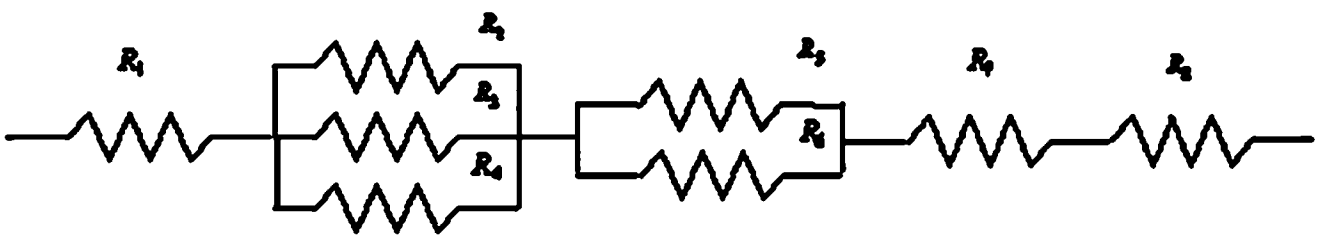
Given:
Length of the wall =8m
Thickness of the wall = .22m
Height of the wall = 5m
Bricks length = 9in
Thermal conductivity for A = 2 W/m·°C.
Thermal conductivity for B = 8 W/m·°C.
Thermal conductivity for C = 20 W/m·°C.
Thermal conductivity for D = 15 W/m·°C.
Thermal conductivity for E = 35 W/m·°C.
Thermal conductivity for F = 2 W/m·°C.
The thermal contact resistence at the interfaces D-F and E-F is $0.00012 m^2$
Left surface temperature = $300^0 C$
Right surface temperature = $100^0 C$

Step 2 of 9

If the heat transfer is steady since there is no indication of change with time and heat transfer through the wall is one-dimensionalthe then the thermal resistance network and the individual thermal resistances are

Step 3 of 9





Step 4 of 9

(a) The thermal resistance network and the individual thermal resistances are

$$R_1 = R_A = \left(\frac{L}{kA}\right)_A$$
$$= \frac{0.01\,\text{m}}{(2\ \text{W/m.°C})(0.12\ \text{m}^2)}$$
$$= 0.04\ \text{°C/W}$$
$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C$$
$$= \frac{0.05\,\text{m}}{(20\ \text{W/m.°C})(0.04\ \text{m}^2)}$$
$$= 0.06\ \text{°C/W}$$
$$R_3 = R_B = \left(\frac{L}{kA}\right)_B$$
$$= \frac{0.05\,\text{m}}{(8\ \text{W/m.°C})(0.04\ \text{m}^2)}$$
$$= 0.16\ \text{°C/W}$$

Step 5 of 9

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D$$
$$= \frac{0.1\,\text{m}}{(15\ \text{W/m.°C})(0.06\ \text{m}^2)}$$
$$= 0.11\ \text{°C/W}$$
$$R_6 = R_E = \left(\frac{L}{kA}\right)_E$$
$$= \frac{0.1\,\text{m}}{(35\ \text{W/m.°C})(0.06\ \text{m}^2)}$$
$$= 0.05\ \text{°C/W}$$
$$R_7 = R_F = \left(\frac{L}{kA}\right)_F$$
$$= \frac{0.06\,\text{m}}{(2\ \text{W/m.°C})(0.12\ \text{m}^2)}$$
$$= 0.25\ \text{°C/W}$$
$$R_8 = \frac{0.00012\ \text{m}^2\text{°C/W}}{0.12\ \text{m}^2}$$
$$= 0.001\ \text{°C/W}$$
$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4}$$
$$= \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06}$$
$$R_{mid,1} = 0.025\ \text{°C/W}$$

Step 6 of 9

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6}$$
$$= \frac{1}{0.11} + \frac{1}{0.05}$$
$$R_{mid,2} = 0.034\ \text{°C/W}$$
$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 + R_8$$

$$= \frac{1}{0.11} + \frac{1}{0.05}$$

$$R_{mid,2} = 0.034\ ^\circ\text{C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 + R_8$$
$$= 0.04 + 0.025 + 0.034 + 0.25 + 0.001$$
$$= 0.350\ ^\circ\text{C/W}$$

Step 7 of 9

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$$
$$= \frac{(300 - 100)^\circ\text{C}}{0.350\ ^\circ\text{C/W}}$$
$$= 571\ \text{W (for a 0.12 m} \times \text{1 m section)}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (571\ \text{W})\frac{(5\ \text{m})(8\ \text{m})}{0.12\ \text{m}^2}$$
$$\dot{Q}_{total} = 1.90 \times 10^5\ \text{W}$$
$$\boxed{\dot{Q}_{total} = 1.90 \times 10^5\ \text{W}}$$

Step 8 of 9

(b) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1}$$
$$= 0.04 + 0.025$$
$$= 0.065\ ^\circ\text{C/W}$$

Then the temperature at the point where The sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}}$$
$$T = T_1 - \dot{Q}R_{total}$$
$$= 300^\circ\text{C} - (571\ \text{W})(0.065\ ^\circ\text{C/W})$$
$$= 263^\circ C$$
$$\boxed{T = 263^\circ C}$$

Step 9 of 9

(c) The temperature drop across the section F can be determined from

$$\dot{Q} = \frac{\Delta T}{R_7}$$
$$\Delta T = \dot{Q}R_7$$
$$= (571\ \text{W})(0.25\ ^\circ\text{C/W})$$
$$= 143^\circ\text{C}$$
$$\boxed{\Delta T = 143^\circ\text{C}}$$

Question: 17.059P pg 1

Clothing made of several thin layers of fabric with trapped air in between, often called ski clothing, is commonly used in cold climates because it is light, fashionable, and a very effective thermal insulator. So it is no surprise that such clothing has largely replaced thick and heavy old-fashioned coats.

Consider a jacket made of five layers of 0.15-mm-thick synthetic fabric (k = 0.13 W/m·K) with 1.5-mm-thick air space (k = 0.026 W/m·K) between the layers. Assuming the inner surface temperature of the jacket to be 25°C and the surface area to be 1.25 m2, determine the rate of heat loss through the jacket when the temperature of the outdoors is 0°C and the heat transfer coefficient at the outer surface is 25 W/m2. K.

What would your response be if the jacket is made of a single layer of 0.75-mm-thick synthetic fabric? What should be the thickness of a wool fabric (k = 0.035 W/m·K) if the person is to achieve the same level of thermal comfort wearing a thick wool coat instead of a five-layer ski jacket?



FIGURE P17–59

Step 1 of 8

Given data:

Thickness of the jacket, $t_1 = 0.15\text{mm} = 0.00015\text{m}$

Thickness of the airspace, $t_2 = 1.5\text{mm} = 0.0015\text{m}$

Thermal conductivity of the jacket, $k_1 = k_3 = k_5 = k_7 = k_9 = 0.13\,\text{W/m-K}$

Thermal conductivity of the air, $k_2 = k_4 = k_6 = 0.026\,\text{W/m-K}$

Inner surface temperature of the jacket, $T_1 = 25^\circ\text{C}$

Outdoor temperature, $T_{\infty,2} = 0^\circ\text{C}$

Heat transfer coefficient on the outer surface, $h_2 = 25\,\text{W/m}^2\text{-K}$

Surface area of the Jacket, $A = 1.25\text{m}^2$

Step 2 of 8



Step 3 of 8

With five layered synthetic fabric:

The individual resistances are evaluated as:

$$R_{fabric} = R_1 = R_3 = R_5 = R_7 = R_9$$

$$= \frac{t_1}{k_1 A_1}$$

$$= \frac{0.00015(\text{m})}{0.13(\text{W/m-K}) \times 1.25(\text{m}^2)}$$

$$= 0.0009231\text{K/W}$$

$$R_{air} = R_2 = R_4 = R_6 = R_8$$

$$= \frac{t_2}{k_2 A_2}$$

$$= \frac{0.0015(\text{m})}{0.026(\text{W/m-K}) \times 1.25(\text{m}^2)}$$

$$= 0.04615\text{K/W}$$

$$R_o = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$= \frac{1}{25(\text{W/m}^2\text{-K}) \times 1.25(\text{m}^2)}$$

$$= 0.032\text{K/W}$$

Step 4 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = 5R_{fabric} + 4R_{air} + R_o$$

$$= 5 \times 0.0009231(\text{K/W}) + 4 \times 0.04615(\text{K/W}) + 0.032(\text{K/W})$$

$$= 0.2212\text{K/W}$$

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$

$$= \frac{(25-0)}{0.2212(\text{K/W})}$$

$$= 113\text{W}$$

$$\boxed{\dot{Q}_{total} = 113\text{W}}$$

Step 5 of 8

Step 6 of 8

If the jacket is made of a single layer of 0.75 mm thick synthetic fabric, the rate of heat transfer would be calculated as

$$R_{fabric} = \frac{t_1}{k_1 A_1}$$

$$= \frac{0.00075(\text{m})}{0.13(\text{W/m-K}) \times 1.25(\text{m}^2)}$$

$$= 0.0004615\text{K/W}$$

$$R_o = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$0.13(\text{W/m-K})\times 1.25(\text{m}^2)$$

$$= 0.0004615\text{K/W}$$

$$R_o = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$= \frac{1}{25(\text{W/m}^2\text{-K})\times 1.25(\text{m}^2)}$$

$$= 0.032\text{K/W}$$

Step 7 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = R_{fabric} + R_o$$

$$= 0.0004615\,(\text{K/W}) + 0.032\,(\text{K/W})$$

$$= 0.0366\text{K/W}$$

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$

$$= \frac{(25-0)}{0.0366\,(\text{K/W})}$$

$$= 682.77\text{W}$$

Since the Jacket is made with single layer, the heat transfer rate increased by 6 times, this is due to the fact that the jacket with air space having high resistance or low conductance to conduct heat across the jacket.

$$\boxed{\dot{Q}_{total} = 682.77\text{W}}$$

Step 8 of 8

In order to have the same level of thermal comfort with wool coat, the thickness of the jacket can be calculated as follows,

$$R_{total} = R_{wool} + R_o = 0.2212\text{K/W}$$

$$R_{wool} = \frac{t}{kA}$$

$$= \frac{t}{0.035(\text{W/m-K})\times 1.25(\text{m}^2)}$$

$$R_o = R_{conv,2}$$

$$= \frac{1}{h_2 A_2}$$

$$= \frac{1}{25(\text{W/m}^2\text{-K})\times 1.25(\text{m}^2)}$$

$$= 0.032\text{K/W}$$

$$R_{wool} + R_o = 0.2212\text{K/W}$$

$$\frac{t}{0.035\,(\text{W/m-K})\times 1.25(\text{m}^2)} + 0.032\,(\text{K/W}) = 0.2212\text{K/W}$$

$$t = 0.0082775\text{m}$$

Therefore, the thickness of the wool coat, $t = 0.0082775\text{m}$

$$\boxed{t = 8.2775\text{mm}}$$

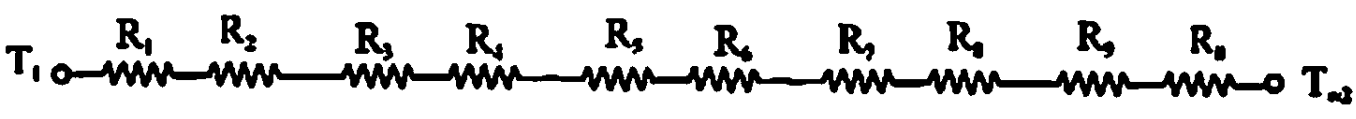
Question: 17.060P

Repeat Prob. 17-59 assuming the layers of the jacket are made of cotton fabric (k = 0.06 W/m·K).

Step 1 of 8

Loading solution. Please wait ...

Step 2 of 8



Step 3 of 8

With five layered synthetic fabric:
The individual resistances are evaluated as:

$$R_{fabric} = R_1 = R_3 = R_5 = R_7 = R_9$$

Step 4 of 8

All the resistances are in series, and the total resistance is

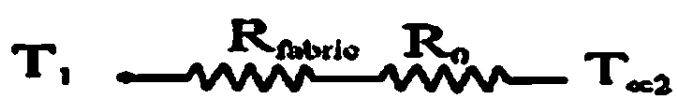
$$R_{total} = 5R_{fabric} + 4R_{air} + R_o$$
$$= 5\times 0.002(\text{K/W}) + 4\times 0.04615(\text{K/W}) + 0.032(\text{K/W})$$
$$= 0.2266\,\text{K/W}$$

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$
$$= \frac{(25-0)}{0.2266\,(\text{K/W})}$$
$$= 110.33\,\text{W}$$

$$\boxed{\dot{Q}_{total} = 113\,\text{W}}$$

Step 5 of 8



Step 6 of 8

If the jacket is made of a single layer of 0.75 mm thick synthetic fabric, the rate of heat transfer would be calculated as

$$R_{fabric} = \frac{t_1}{k_1 A_1}$$
$$= \frac{0.00075(\text{m})}{0.06\,(\text{W/m-K})\times 1.25(\text{m}^2)}$$
$$= 0.01\,\text{K/W}$$

$$R_o = R_{conv,2}$$
$$= \frac{1}{h_2 A_2}$$
$$= \frac{1}{25(\text{W/m}^2\text{-K})\times 1.25(\text{m}^2)}$$
$$= 0.032\,\text{K/W}$$

Step 7 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = R_{fabric} + R_o$$
$$= 0.01(\text{K/W}) + 0.032(\text{K/W})$$
$$= 0.042\,\text{K/W}$$

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q} = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$
$$= \frac{(25-0)}{0.042\,(\text{K/W})}$$
$$= 595.24\,\text{W}$$

Since the Jacket is made with single layer, the heat transfer rate increased by 6 times, this is due to the fact that the jacket with air space having high resistance or low conductance to conduct heat across the jacket.

$$\boxed{\dot{Q}_{total} = 595.24\,\text{W}}$$

Step 8 of 8

In order to have the same level of thermal comfort with wool coat, the thickness of the jacket can be calculated as follows,

$$R_{total} = R_{wool} + R_o = 0.2266\,\text{K/W}$$

$$R_{wool} = \frac{t}{kA}$$
$$= \frac{t}{0.035(\text{W/m-K})\times 1.25(\text{m}^2)}$$

$$R_o = R_{conv,2}$$
$$= \frac{1}{h_2 A_2}$$
$$= \frac{1}{25(\text{W/m}^2\text{-K})\times 1.25(\text{m}^2)}$$
$$= 0.032\,\text{K/W}$$

$$R_{wool} + R_o = 0.2266\,\text{K/W}$$

$$\frac{t}{0.035\,(\text{W/m-K})\times 1.25(\text{m}^2)} + 0.032\,(\text{K/W}) = 0.2266\,\text{K/W}$$

$$t = 0.00851375\,\text{m}$$

Therefore, the thickness of the wool coat, $t = 0.00851375\,\text{m}$

$$\boxed{t = 8.51375\,\text{mm}}$$

Question: 17.061P

In an experiment to measure convection heat transfer coefficients, a very thin metal foil of very low emissivity (e.g., highly polished copper) is attached on the surface of a slab of material with very low thermal conductivity. The other surface of the metal foil is exposed to convection heat transfer by flowing fluid over the foil surface. This setup diminishes heat conduction through the slab and radiation on the metal foil surface, while heat convection plays the prominent role. The slab on which the metal foil is attached to has a thickness of 25 mm and a thermal conductivity of 0.023 W/m·K. In a condition where the surrounding room temperature is 20°C, the metal foil is heated electrically with a uniform heat flux of 5000 W/m2. If the bottom surface of the slab is 20°C and the metal foil has an emissivity of 0.02, determine.

(a) The convection heat transfer coefficient if air is flowing over the metal foil and the surface temperature of the foil is 150°C.

(b) The convection heat transfer coefficient if water is flowing over the metal foil and the surface temperature of the foil is 30°C.

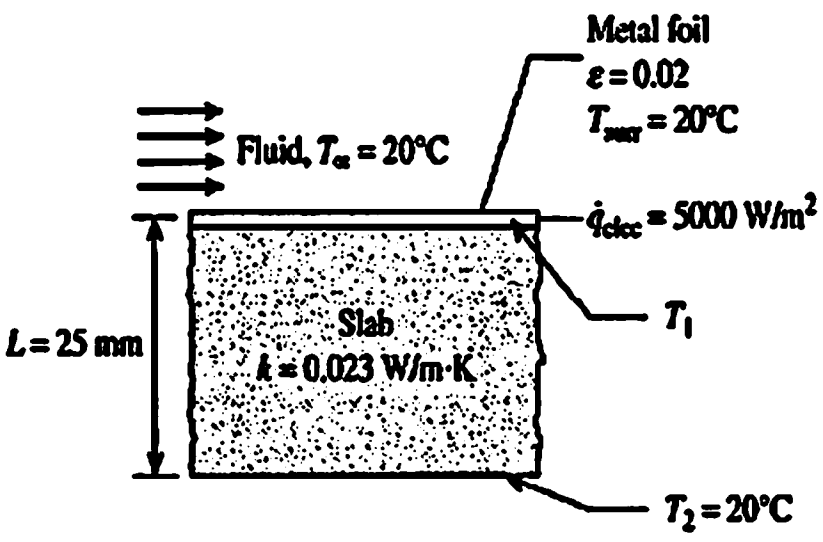


FIGURE P17–61

Step 1 of 3

The foil is heated electrically with a uniform flux of $5000\ W/m^2$. The electrical energy supplied is lost in three ways. They are conduction through the slab, convection from the foil and radiation from the foil.

Foil setup results in diminishing of heat conduction through the slab and radiation from the metal foil. Thus, only convection from the foil helps in conservation of energy.

$\dot{q}_{elec} = \dot{Q}_{conv}$ ..... (1)

Write the formula for rate of heat transferred by convection from metal foil.

$\dot{Q}_{conv} = h(T_s - T_\infty)$

Here, convection heat transfer coefficient is h.

Substitute $h(T_s - T_\infty)$ for $\dot{Q}_{conv}$ in equation (1).

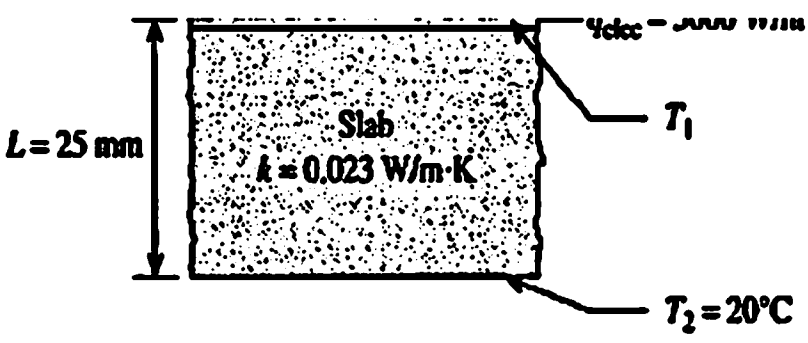


FIGURE P17–61

Step 1 of 3

The foil is heated electrically with a uniform flux of $5000\ W/m^2$. The electrical energy supplied is lost in three ways. They are conduction through the slab, convection from the foil and radiation from the foil.

Foil setup results in diminishing of heat conduction through the slab and radiation from the metal foil. Thus, only convection from the foil helps in conservation of energy.

$\dot{q}_{elec} = \dot{Q}_{conv}$ ..... (1)

Write the formula for rate of heat transferred by convection from metal foil.

$\dot{Q}_{conv} = h(T_s - T_\infty)$

Here, convection heat transfer coefficient is h.

Substitute $h(T_s - T_\infty)$ for $\dot{Q}_{conv}$ in equation (1).

Calculate the convection heat transfer coefficient when water is flowing over the surface of the metal foil at $30\ °C$ by using the relation in equation (2) as shown.

$\dot{q}_{elec} = h(T_s - T_\infty)$

Substitute $5000\ W/m^2$ for $\dot{q}_{elec}$, $(30+273)\ K$ for T_s , and $(20+273)\ K$ for T_∞ in equation (2).

$$5000 = h[(30+273)-(20+273)]$$
$$5000 = h(30-20)$$
$$h = \frac{5000}{10}$$
$$= 500\ W/m^2 \cdot K$$

Therefore, the convection heat transfer coefficient when water is flowing over the surface of the metal foil at $30\ °C$ is $\boxed{500\ W/m^2 \cdot K}$.

Question: 17.062P

Consider a 10-in × 12-in epoxy glass laminate (k = 0.10 Btu/h·ft·°F) whose thickness is 0.05 in. In order to reduce the thermal resistance across its thickness, cylindrical copper fillings (k = 223 Btu/h·ft·°F) of 0.02 in diameter are to be planted throughout the board, with a center-to-center distance of 0.06 in. Determine the new value of the thermal resistance of the epoxy board for heat conduction across its thickness as a result of this modification.

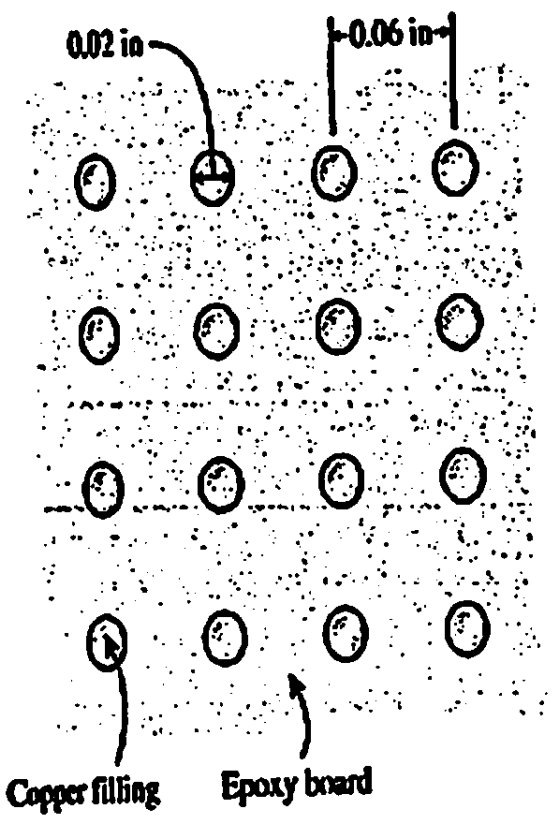


FIGURE P17–62E

Step 1 of 5

Given data:

Cross sectional are of the laminate, $A = 10 \times 12 \text{in}^2 = 120 \text{in}^2$

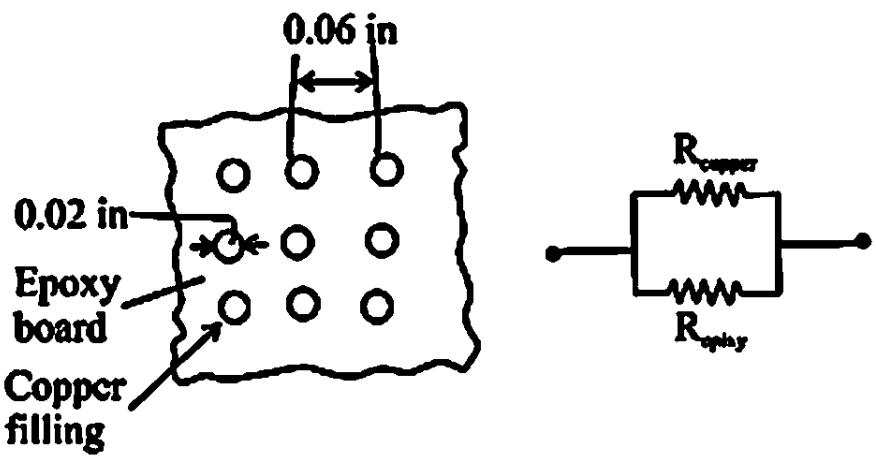
Thickness of the epoxy board, $t_1 = 0.05 \text{in}$

Thermal conductivity of the epoxy, $k_1 = 0.1 \text{Btu/h-ft-}^\circ\text{F}$

Thermal conductivity of the copper, $k_2 = 223 \text{Btu/h-ft-}^\circ\text{F}$

Diameter of the copper fillings, $d = 0.02 \text{in}$

Step 2 of 5



Step 3 of 5

The thermal resistances of copper fillings and the epoxy board are in parallel. The number of copper fillings in the board and the area they comprise are

$$A_{total} = \frac{10}{12} \times \frac{12}{12} \text{ft}^2$$
$$= 0.8333 \text{ft}^2$$

Number of copper fillings can be determined as

$$n_{copper} = \frac{0.8333(\text{ft}^2)}{0.06/12 \times 0.06/12(\text{ft}^2)}$$
$$= 33333$$

Area of the copper fillings can be determined as

$$A_{copper} = n\frac{\pi}{4} \times d^2 \text{ft}^2$$
$$= 33333 \times \frac{\pi}{4} \times \left(\frac{0.02}{12}\right)^2 \text{ft}^2$$
$$= 0.07272 \text{ ft}^2$$

Area of the epoxy can be calculated as

$$A_{epoxy} = A_{total} - A_{copper}$$
$$= (0.8333 - 0.07272)\text{ft}^2$$
$$= 0.7605 \text{ft}^2$$

Step 4 of 5

The Thermal resistances are evaluated to be

$$R_{copper} = \frac{t_2}{k_2 A_{copper}}$$
$$= \frac{0.05/12\,(\text{ft})}{223(\text{Btu/h-ft-}^\circ\text{F}) \times 0.07272(\text{ft}^2)}$$
$$= 0.000257 \text{h-}^\circ\text{F/Btu}$$

$$R_{epoxy} = \frac{t_1}{k_1 A_{epoxy}}$$
$$= \frac{0.05/12(\text{ft})}{0.1(\text{Btu/h-ft-}^\circ\text{F}) \times 0.7605(\text{ft}^2)}$$
$$= 0.0548 \text{h-}^\circ\text{F/Btu}$$

Step 5 of 5

Then the thermal resistance of the entire epoxy board becomes

$$\frac{1}{R_{board}} = \frac{1}{R_{copper}} + \frac{1}{R_{epoxy}}$$
$$= \frac{1}{0.000257(\text{h-}^\circ\text{F/Btu})} + \frac{1}{0.0548(\text{h-}^\circ\text{F/Btu})}$$
$$= 3909.3 \text{Btu/h-}^\circ\text{F}$$

Therefore the epoxy board resistance is

$$R_{board} = \frac{1}{3909.3(\text{Btu/h-}^\circ\text{F})}$$
$$= 0.000256 \text{h-}^\circ\text{F/Btu}$$

$$\boxed{R_{board} = 0.000256 \text{h-}^\circ\text{F/Btu}}$$

Question: 17.063P

What is an infinitely long cylinder? When is it proper to treat an actual cylinder as being infinitely long , and when is it not?

Step 1 of 1

When the diameter of cylinder is very small compared to its length, it can be treated as an indefinitely long cylinder.
Cylindrical rods can also be treated as being infinitely long when dealing with heat transfer at locations far from the top or bottom surfaces. However, it is not proper to use this model when finding temperatures near the bottom and the top of the cylinder.

Question: 17.064P

Can the thermal resistance concept be used for a solid cylinder or sphere in steady operation? Explain.

Step 1 of 1

No. In steady-operation the temperature of a solid cylinder or sphere does not change in radial direction (unless there is heat generation).

Question: 17.065P

Consider a short cylinder whose top and bottom surfaces are insulated. The cylinder is initially at a uniform temperature *Ti* and is subjected to convection from its side surface to a medium at temperature T_∞, with a heat transfer coefficient of *h*. Is the heat transfer in this short cylinder one- or two-dimensional? Explain.

Step 1 of 1

Heat transfer in this short cylinder is one-dimensional since there will be no heat transfer in the axial and tangential directions.

Question: 17.066P pg 1

An 8-m-internal-diameter spherical tank made of 1.5-cm-thick stainless steel (k =15 W/m·K) is used to store iced water at 0°C. The tank is located in a room whose temperature is 25°C. The walls of the room are also at 25° C. The outer surface of the tank is black (emissivity ε = 1), and heat transfer between the outer surface of the tank and the surroundings is by natural convection and radiation. The convection heat transfer coefficients at the inner and the outer surfaces of the tank are 80 W/m2·K and 10 W/m2·K, respectively. Determine (*a*) the rate of heat transfer to the iced water in the tank and (*b*) the amount of ice at 0°C that melts during a 24-h period. The heat of fusion of water at atmospheric sure is *hif*=333.7 kJ/kg.

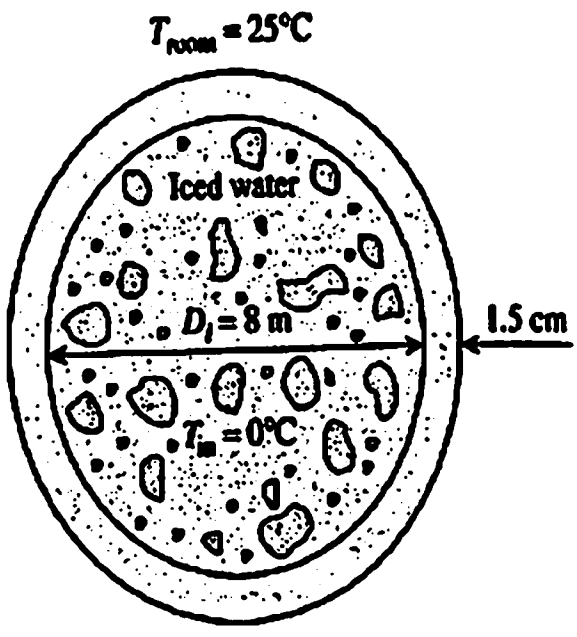


FIGURE P17–66

Step 1 of 8

Calculate the outer diameter of the spherical tank

$$D_o = D_i + 2t$$

Here, D_o is the outer diameter of the spherical tank, D_i is the inner diameter of the spherical tank and t is the thickness of the spherical tank.

Substitute $8\,\text{m}$ for D_i, $0.015\,\text{m}$ for t

$$\begin{aligned} D_o &= D_i + 2t \\ &= 8 + (2 \times 0.015) \\ &= 8.03\,\text{m} \end{aligned}$$

Step 2 of 8

Calculate the inside area of the spherical tank

$$\begin{aligned} A_i &= \pi D_i^2 \\ &= \pi (8\,\text{m})^2 \\ &= 201.06\,\text{m}^2 \end{aligned}$$

Calculate the outside area of the spherical tank

$$\begin{aligned} A_o &= \pi D_o^2 \\ &= \pi (8.03\,\text{m})^2 \\ &= 202.57\,\text{m}^2 \end{aligned}$$

Step 3 of 8

If the heat transfer is steady since the specified thermal conditions at the boundaries do not change with time and the heat transfer is one-dimensional since there is If the heat transfer is steady since the specified thermal conditions at the boundaries do not change with time and the heat transfer is one-dimensional since there thermal symmetry about the midpoint and assume the outer surface temperature T_2 to be 4°C after comparing convection heat transfer coefficients at the inner and the outer surfaces of the tank.

Use the conversion factor to convert the temperature from $1°\text{C to } 1\text{K}$

$$1\text{K} = 1°\text{C} + 273$$

Calculate the radiation heat transfer coefficient

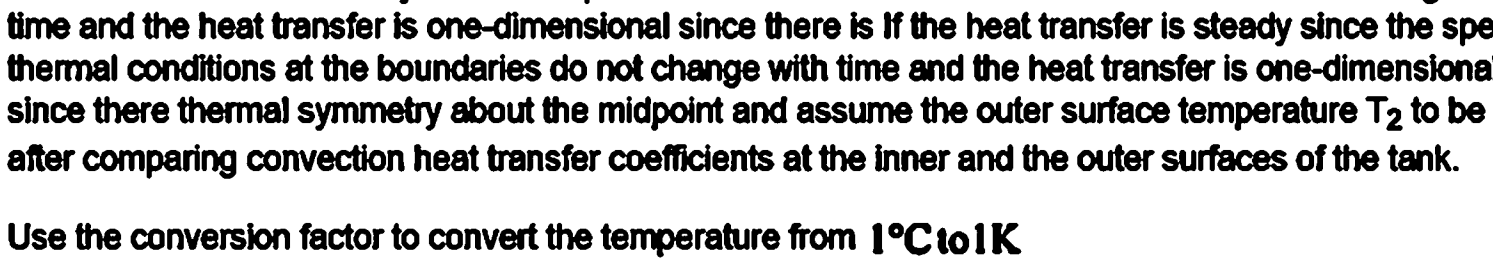
$$h_{rad} = \varepsilon\sigma\left(T_2^2 + T_{surr}^2\right)\left(T_2 + T_{surr}\right)$$

Here, σ is the Stefan Boltzmann constant, ε is the emissivity of the spherical tank, T_2 is the outer surface temperature of the tank and T_{surr} is the surrounding temperature

$$\begin{aligned} h_{rad} &= \varepsilon\sigma\left(T_2^2 + T_{surr}^2\right)\left(T_2 + T_{surr}\right) \\ &= 1\left(5.67 \times 10^{-8}\right)\left[(273+4)^2 + (273+25)^2\right]\left[(273+25) + (273+4)\right] \\ &= 1\left(5.67 \times 10^{-8}\right)(165533)(575) \\ &= 5.39 \end{aligned}$$

Step 4 of 8

Circuit diagram of spherical tank as shown below:



Step 5 of 8

Calculate the inner resistance of the spherical tank

$$R_{conv,1} = R_i = \frac{1}{h_i A_i}$$

Here, h_i is the inside heat transfer coefficient,

Substitute $201.06\,\text{m}^2$ for A_i, $80\,\text{W/m}^2 \cdot °\text{C}$

$$\begin{aligned} R_{conv,1} = R_i &= \frac{1}{h_i A_i} \\ &= \frac{1}{80(201.06)} \end{aligned}$$

$$R_{conv,i} = R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{80(201.06)}$$
$$= 0.0000622\ ^\circ C/W$$

Step 6 of 8

Calculate the resistance of the sphere

$$R_1 = R_{sphere}$$
$$= \frac{r_2 - r_1}{4\pi k r_1 r_2}$$

Substitute $4.015\,m$ for r_2, $4\,m$ for r_1, $15\,W/m\cdot^\circ C$ for k

$$R_1 = R_{sphere}$$
$$= \frac{r_2 - r_1}{4\pi k r_1 r_2}$$
$$= \frac{(4.015-4)\ m}{4\pi(15\ W/m.^\circ C)(4.015\ m)(4\ m)}$$
$$= 0.00000496\ ^\circ C/W$$

Calculate the outer convection resistance

$$R_{conv,o} = \frac{1}{h_o A_0}$$

Here, h_o is the outer heat transfer coefficient

Substitute $202.57\ m^2$ for A_o, $10\ W/m^2\cdot^\circ C$ for h_o

$$R_{conv,o} = \frac{1}{h_o A_0}$$
$$= \frac{1}{(10\ W/m^2.^\circ C)(202.57\ m^2)}$$
$$= 0.00049\ ^\circ C/W$$

Step 7 of 8

Calculate the radiation resistance

$$R_{rad} = \frac{1}{h_{rad} A_0}$$

Here, h_{rad} is the radiation heat transfer coefficient

Substitute $5.39\ W/m^2\cdot^\circ C$ for h_{rad},

$$R_{rad} = \frac{1}{h_{rad} A_0}$$
$$= \frac{1}{(5.57\ W/m^2\cdot^\circ C)(202.57\ m^2)}$$
$$= 0.000886\ ^\circ C/W$$

Calculate the equivalent resistance

$$\frac{1}{R_{eqv}} = \frac{1}{R_{conv,o}} + \frac{1}{R_{rad}}$$
$$= \frac{1}{0.00049} + \frac{1}{0.000886}$$
$$= 0.000316\ ^\circ C/W$$

Calculate the total resistance

$$R_{total} = R_{conv,i} + R_1 + R_{eqv}$$
$$= 0.0000622 + 0.00000496 + 0.000316$$
$$= 0.00038316\ ^\circ C/W$$

(a)

Calculate the steady rate of heat transfer to the iced water

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$$

Substitute $0.00038316\ ^\circ C/W$ for R_{total}

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}$$
$$= \frac{25-0}{0.00038316}$$
$$= 65246.894\,W$$
$$= 65.246894\,kW$$

Therefore, Steady rate of heat transfer to the iced water is $\boxed{65246.894\,W}$

Step 8 of 8

(b)

Calculate the total amount of heat transfer during a 24-hour period and the amount of ice that will melt during this period

$$Q = \dot{Q}\Delta t$$
$$= 65.246894(24\times 3600)$$
$$= 5.637\times 10^6\ kJ$$

Calculate the amount of ice melted

$$m_{ice} = \frac{Q}{h_{if}}$$
$$= \frac{5.637\times 10^6\ kJ}{333.7\ kJ/kg}$$
$$= 16892.41kg$$

$$m_{ice} = \frac{Q}{h_{if}}$$

$$= \frac{5.637 \times 10^6 \text{ kJ}}{333.7 \text{ kJ/kg}}$$

$$= 16892.41 \text{kg}$$

Therefore, the amount of ice melted is **16892.41kg**

Question: 17.067P

Steam at 280°C flows in a stainless steel pipe (k=15 W/m·K) whose inner and outer diameters are 5 cm and; 5.5 cm, respectively. The pipe is covered with 3-cm-thick glass wool insulation (k= 0.038 W/m· K). Heat is lost to the surroundings at 5°C by natural convection and radiation, with a combined natural convection and radiation heat transfer coefficient of 22 W/m2. K. Taking the heat transfer coefficient inside the pipe to be 80 W/m2. K, determine the rate of heat loss from the steam per unit length of the pipe. Also determine the temperature drops across the pipe shell and the insulation.

Step 1 of 8

Given data:

Inner diameter of the steel pipe, $d = 5\,\text{cm} = 0.05\,\text{m}$

Outer diameter of the steel pipe, $D = 5.5\,\text{cm} = 0.055\,\text{m}$

Thickness of the insulation, $t = 3\,\text{cm} = 0.03\,\text{m}$

Thermal conductivity of the steel pipe, $k_1 = 15\,\text{W/m-K}$

Thermal conductivity of the insulation, $k_2 = 0.038\,\text{W/m-K}$

Inside steam temperature of the pipe, $T_{\infty,1} = 280°\text{C}$

Surroundings temperature, $T_{\infty,2} = 5°\text{C}$

Combined heat transfer coefficient on the outer surface, $h_o = 22\,\text{W/m}^2\text{-K}$

Heat transfer coefficient on the inner surface, $h_i = 80\,\text{W/m}^2\text{-K}$

Step 2 of 8

Let us draw the net work diagram:



Step 3 of 8

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi dL$$
$$= \pi \times 0.05 \times 1$$
$$= 0.1571\,\text{m}^2$$
$$A_o = \pi D_o L$$
$$= \pi \times (0.055 + 0.06) \times 1$$
$$= 0.3613\,\text{m}^2$$

Step 4 of 8

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{80 \times 0.1571}$$
$$= 0.08\,\text{K/W}$$

$$R_1 = R_{pipe}$$
$$= \frac{\ln(r_2/r_1)}{2\pi k_1 L}$$
$$= \frac{\ln(2.75/2.5)}{2\pi \times 15 \times 1}$$
$$= 0.00101\,\text{K/W}$$

Step 5 of 8

$$R_2 = R_{insulation}$$
$$= \frac{\ln(r_3/r_2)}{2\pi k_2 L}$$
$$= \frac{\ln(5.75/2.75)}{2\pi \times 0.038 \times 1}$$
$$= 3.09\,\text{K/W}$$

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{22 \times 0.3613}$$
$$= 0.1258\,\text{K/W}$$

Step 6 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_2 + R_o$$
$$= 0.08 + 0.00101 + 3.09 + 0.1256$$
$$= 3.2968\,\text{K/W}$$

Then the steady rate of heat loss from the steam per unit length becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(280° - 5°)}{3.2968}$$
$$= 83.413\,\text{W}$$

Step 7 of 8

The temperature drop across the steel pipe can be determined as

$$\dot{Q} = \frac{\Delta T_{pipe}}{R_{pipe}}$$

By rewriting the above equation we would get

$$\Delta T_{pipe} = \dot{Q} R_{pipe}$$
$$= 83.413 \times 0.00101$$
$$= 0.08425\,\text{K}$$

Step 8 of 8

The temperature drop across the insulation can be determined as

$$\dot{Q} = \frac{\Delta T_{insulation}}{R_{insulation}}$$

By rewriting the above equation we would get

$$\Delta T_{insulation} = \dot{Q} R_{insulation}$$
$$= 83.413 \times 3.09$$
$$= 257.75\,\text{K}$$

Therefore the temperature drop is $\boxed{\Delta T_{insulation} = 257.75\,\text{K}}$

Question: 17.068P

Reconsider Prob. 17-67. Using EES (or other) software, investigate the effect of the thickness of the insulation on the rate of heat loss from the steam and the temperature drop across the insulation layer. Let the insulation thickness vary from 1 cm to 10 cm. Plot the rate of heat loss and the temperature drop as a function of insulation thickness, and discuss the results.

Step 1 of 9

Given data:

Inner diameter of the steel pipe, $d = 5\,\text{cm} = 0.05\,\text{m}$

Outer diameter of the steel pipe, $D = 5.5\,\text{cm} = 0.055\,\text{m}$

Thickness of the insulation, $t = (1-10)\,\text{cm}$

Thermal conductivity of the steel pipe, $k_1 = 15\,\text{W/m-K}$

Thermal conductivity of the insulation, $k_2 = 0.038\,\text{W/m-K}$

Inside steam temperature of the pipe, $T_{\infty,1} = 280°\text{C}$

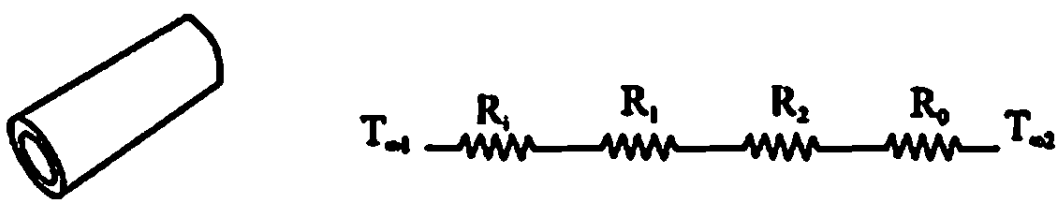
Surroundings temperature, $T_{\infty,2} = 5°\text{C}$

Combined heat transfer coefficient on the outer surface, $h_o = 22\,\text{W/m}^2\text{-K}$

Heat transfer coefficient on the inner surface, $h_i = 80\,\text{W/m}^2\text{-K}$

Step 2 of 9

Thermal network diagram:



Step 3 of 9

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi dL$$
$$= \pi \times 0.05 \times 1$$
$$= 0.1571\,\text{m}^2$$
$$A_o = \pi D_o L$$
$$= \pi \times (0.055 + 2t) \times 1$$
$$= \pi \times (0.055 + 2t)\,\text{m}^2$$

Step 4 of 9

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{80 \times 0.1571}$$
$$= 0.08\,\text{K/W}$$
$$R_1 = R_{pipe}$$
$$= \frac{\ln(r_2/r_1)}{2\pi k_1 L}$$
$$= \frac{\ln(2.75/2.5)}{2\pi \times 15 \times 1}$$
$$= 0.00101\,\text{K/W}$$

Step 5 of 9

$$R_2 = R_{insulation}$$
$$= \frac{\ln(r_3/r_2)}{2\pi k_2 L}$$
$$= \frac{\ln[(2.75+t)/2.75]}{2\pi \times 0.038 \times 1}$$
$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{22 \times \pi \times (0.055 + 2t)}$$

Step 6 of 9

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_2 + R_o$$
$$= 0.08 + 0.00101 + \frac{\ln[(2.75+t)/2.75]}{2\pi \times 0.038 \times 1} + \frac{1}{22 \times \pi \times (0.055 + 2t)}$$

Therefore the heat loss from the steam can be computed by varying the insulation thickness from 1-10 cm using the Excel spread sheet. The heat lost and the temperature drop across the insulation tabulated and plotted as a function of insulation thickness as shown in the below figures.

Step 7 of 9

Heat lost from the steam calculated as

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(280° - 5°)}{R_{total}}$$

The temperature drop across the insulation can be determined as

$$\dot{Q} = \frac{\Delta T_{insulation}}{R_{insulation}}$$

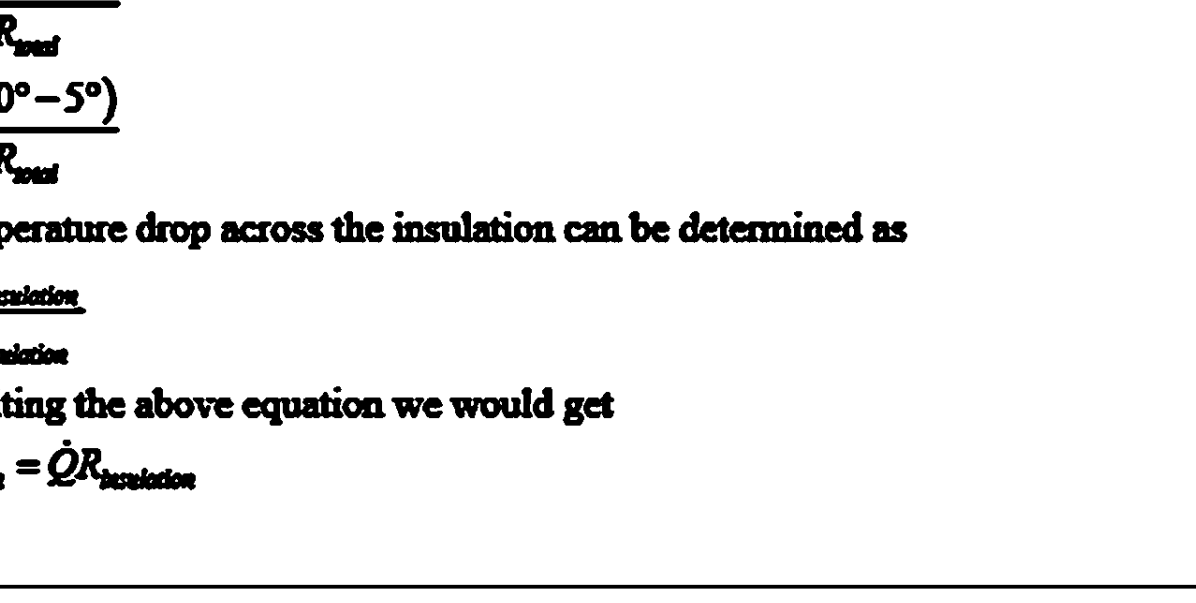
By rewriting the above equation we would get

$$\Delta T_{insulation} = \dot{Q} R_{insulation}$$

Step 8 of 9

Thickness (m)	A_o (m2)	$R_{insulation}$(K/W)	R_o(K/W)	R_{total}(K/W)	Heat loss(W)	ΔT_{ins} (K)
0.01	0.236	1.299	0.193	1.573	174.831	227.109
0.02	0.298	2.289	0.152	2.522	109.023	249.564
0.03	0.361	3.089	0.126	3.296	83.432	257.744
0.04	0.424	3.761	0.107	3.949	69.637	261.895
0.05	0.487	4.339	0.093	4.514	60.924	264.378
0.06	0.550	4.848	0.083	5.011	54.875	266.018
0.07	0.613	5.301	0.074	5.456	50.402	267.177
0.08	0.675	5.710	0.067	5.858	46.943	268.038
0.09	0.738	6.082	0.062	6.225	44.176	268.701
0.1	0.801	6.425	0.057	6.562	41.906	269.227

Step 9 of 9



Question: 17.069P

A 50-m-long section of a steam pipe whose outer diameter is 10 cm passes through an open space at 15°C. The average temperature of the outer surface of the pipe is measured to be 150°C. If the combined heal transfer coefficient on the outer surface of the pipe is 20 W/m2. K, determine (*a*) the rate of heat loss from the steam pipe; (*b*) the annual cost of this energy lost if steam is generated in a natural gas furnace that has an efficiency of 75 percent and the price of natural gas is $0.52/therm (1 therm = 105,500 kJ); and (*c*) the thickness of fiberglass insulation (*k*= 0.035 W/m·K) needed in order to save 90 percent of the heat lost. Assume the pipe temperature to remain constant at 150°C.

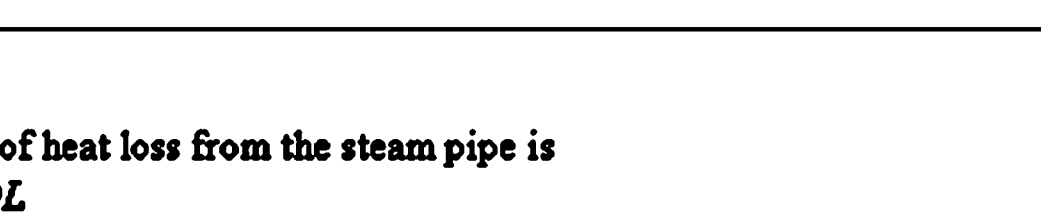
Step 1 of 6

Given: -
Length of steam pipe = 50m
Outer diameter of pipe = 10cm
Combined heat transfer coefficient = $20 W/m^{2o} C$
Pipe temperature = 150°C
Thermal conductivity of fiberglass insulation = 0.035 W/m °C

Step 2 of 6

If the heat transfer is steady since there is no indication of any change with time and the heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction

Step 3 of 6



Step 4 of 6

(*a*) The rate of heat loss from the steam pipe is

$$A_s = \pi DL$$
$$= \pi(0.1\text{ m})(50\text{ m})$$
$$= 15.71\text{ m}^2$$
$$\dot{Q}_{conv} = h_o A(T_s - T_{air})$$
$$= (20\text{ W/m}^2.^\circ\text{C})(15.71\text{ m}^2)(150-15)^\circ\text{C}$$
$$= 42412\text{ W}$$

The rate of heat loss from the steam pipe is **42412 W**

Step 5 of 6

(*b*) The amount of heat loss per year is

$$Q = \dot{Q}\Delta t$$
$$= (42.412\text{ kJ/s})(365\times24\times3600\text{ s/yr})$$
$$= 1.337\times10^9\text{ kJ/yr}$$

The amount of gas consumption from the natural gas furnace that has an efficiency of 75% is

$$Q_{gas} = \frac{1.337\times10^9\text{ kJ/yr}}{0.75}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right)$$
$$= 16{,}903\text{ therms/yr}$$

The annual cost of this energy lost is
Energy cost = (Energy used)(Unit cost of energy)
=(16,903 therms/yr)($0.52/therm)
= $8790/yr

The annual cost of this energy lost is **$8790/yr**

Step 6 of 6

(*c*) In order to save 90% of the heat loss and thus to reduce it to 0.1×42,412 = 4241 W, the thickness of insulation needed is determined from

$$\dot{Q}_{insulated} = \frac{T_s - T_{air}}{R_o + R_{insulation}}$$
$$= \frac{T_s - T_{air}}{\frac{1}{h_o A_o} + \frac{\ln(r_2/r_1)}{2\pi kL}}$$

Substituting and solving for r_2, we get

$$4241\text{ W} = \frac{(150-15)^\circ\text{C}}{\frac{1}{(20\text{ W/m}^2.^\circ\text{C})[(2\pi r_2(50\text{ m})]} + \frac{\ln(r_2/0.05)}{2\pi(0.035\text{ W/m.}^\circ\text{C})(50\text{ m})}}$$

$$r_2 = 0.0692\text{ m}$$

Then the thickness of insulation becomes

$$t_{insulation} = r_2 - r_1$$
$$= 6.92-5$$
$$= 1.92\text{cm}$$

The thickness of insulation is **1.92cm**

Question: 17.070P pg 1

Consider a 1.5-m-high electric hot-water heater that has a diameter of 40 cm and maintains the hot water at 60°C. The tank is located in a small room whose average temperature is 27°C, and the heat transfer coefficients on the inner and outer surfaces of the heater are 50 and 12 W/m2 · K, respectively. The tank is placed in another 46-cm-diameter sheet metal tank of negligible thickness, and the space between the two tanks is filled with foam insulation (k= 0.03 W/m · K). The thermal resistances of the water tank and the outer thin sheet metal shell are very small and can be neglected. The price of electricity is $0.08/kWh, and the home owner pays $280 a year for water heating. Determine the fraction of the hot-water energy cost of this household that is due to the heat loss from the tank.

Hot-water tank insulation kits consisting of 3-cm-thick fiberglass insulation (k =0.035 W/m · K) large enough to wrap the entire tank are available in the market for about $30. If such an insulation is installed on this water tank by the home owner himself, how long will it take for this additional insulation to pay for itself? *Answers;*15.2 percent, 21 months

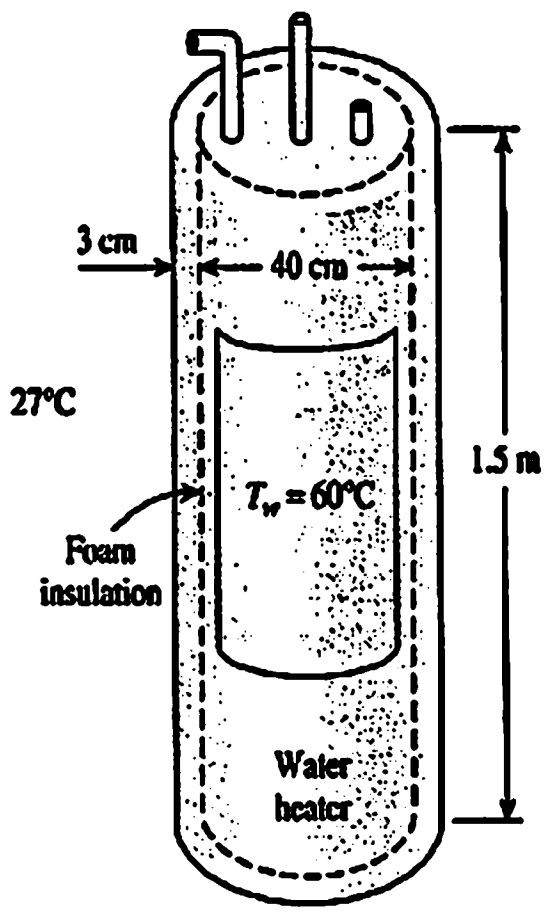


FIGURE P17–70

Step 1 of 13

Given data:

Height of the tank, $L = 1.5\text{m}$

Diameter of the heater, $d_1 = 40\text{cm} = 0.4\text{m}$

Diameter of the sheet metal tank, $d_2 = 46\text{cm} = 0.46\text{m}$

Heat transfer coefficient on the inner surface of the heater, $h_i = 50\,\text{W/m}^2\cdot\text{K}$

Heat transfer coefficient on the outer surface of the heater, $h_0 = 12\,\text{W/m}^2\cdot\text{K}$

Thermal conductivity of the foam insulation, $k_{foam} = 0.03\,\text{W/m}\cdot\text{K}$

Inside water temperature, $T_w = 60°\text{C}$

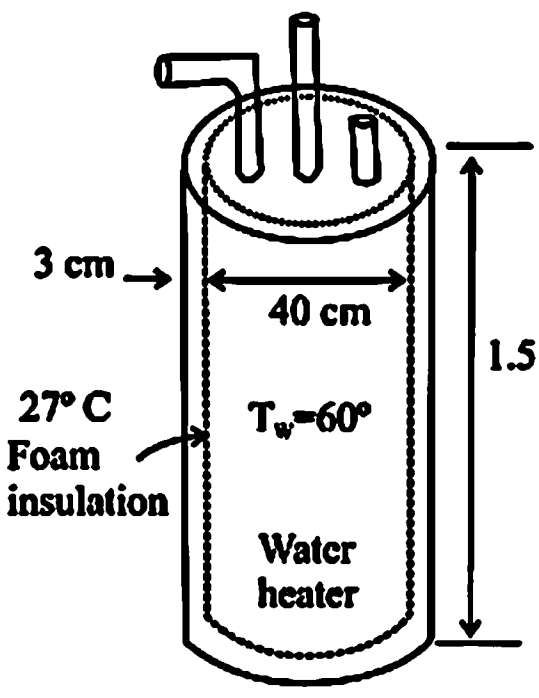
Surroundings temperature, $T_{\infty,2} = 27°\text{C}$

Price of electricity = $0.08/kWh

Amount of money paid by house owner = $280/year

Step 2 of 13

Figure representing the water heater





Step 3 of 13

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi d_1 L$$
$$= \pi \times 0.4 \times 1.5$$
$$= 1.885\ \text{m}^2$$
$$A_0 = \pi d_2 L$$
$$= \pi \times 0.46 \times 1.5$$
$$= 2.1677\ \text{m}^2$$

Step 4 of 13

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{50 \times 1.885}$$
$$= 0.01061\text{K/W}$$
$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi k_{foam} L}$$
$$= \frac{\ln(0.23/0.2)}{2\pi \times 0.03 \times 1.5}$$
$$= 0.4943\text{K/W}$$
$$R_0 = \frac{1}{h_0 A_0}$$
$$= \frac{1}{12 \times 2.1677}$$
$$= 0.038443\text{K/W}$$

Step 5 of 13

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{foam} + R_0$$

Step 5 of 13

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{foam} + R_o$$
$$= 0.01061 + 0.4943 + 0.038443$$
$$= 0.54335\,K/W$$

Then the steady rate of heat loss from the steam per unit length becomes

$$\dot{Q} = \frac{(T_w - T_{\infty,2})}{R_{total}}$$
$$= \frac{(60° - 27°)}{0.54335}$$
$$= 60.734\,W$$

$$\boxed{\dot{Q} = 60.734\,W}$$

Step 6 of 13

The amount and cost of heat loss per year are

$$Q = \dot{Q}\Delta t$$
$$= \frac{60.734}{1000} \times (365 \times 24\,h/year)$$
$$= 532.03\,kW\,h/year$$

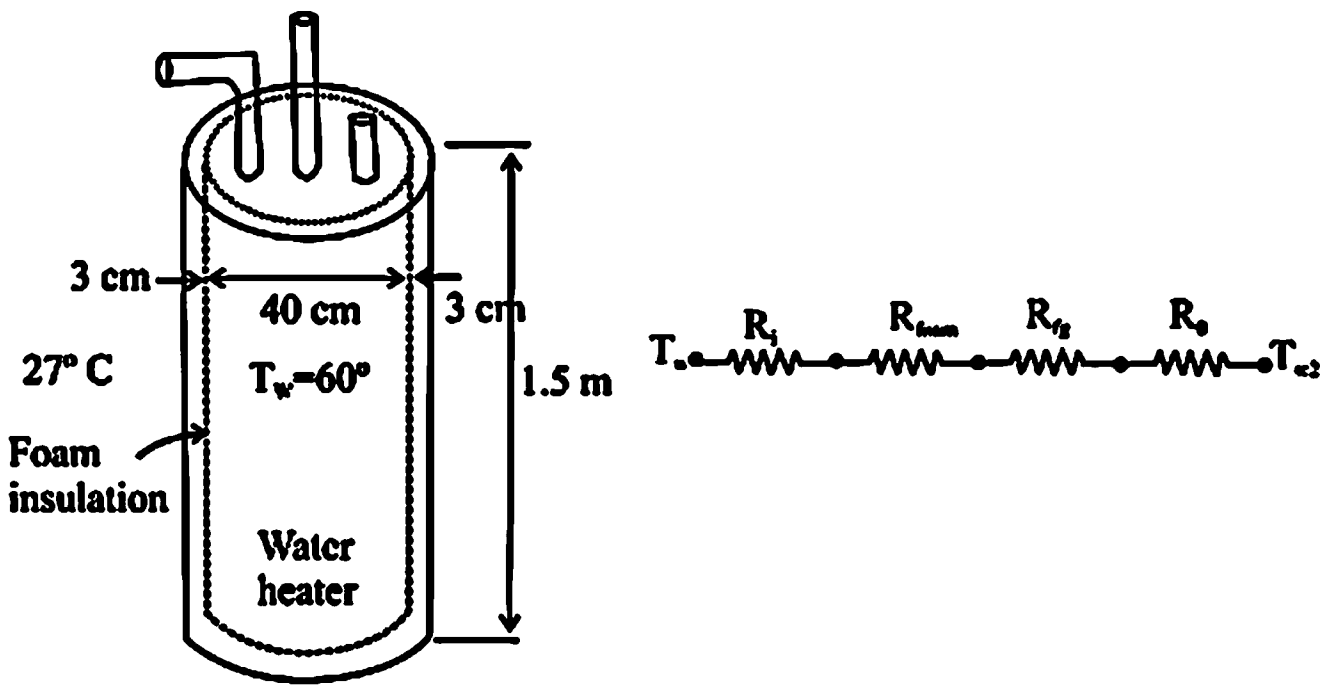
$$\text{Cost of Energy} = \text{Amount of Energy} \times \text{Unit cost}$$
$$= 532.03\,(kW\,h/year) \times (\$0.08/kWh)$$
$$= \$42.562/year$$

Step 7 of 13

Therefore, the fraction of the hot water energy cost of this household that is due to the heat loss from the tank calculated as

$$f = \frac{\text{cost of energy due to heat loss}}{\text{Amount paid annuallyfor water heating}}$$
$$= \frac{\$42.562/year}{\$280/year}$$
$$= 0.152$$
$$= 15.2\%$$

Step 8 of 13



Step 9 of 13

If 3 cm thick fiber glass insulation is used to wrap the entire tank, the individual resistances becomes

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi d_1 L$$
$$= \pi \times 0.4 \times 1.5$$
$$= 1.885\,m^2$$
$$A_o = \pi d_3 L$$
$$= \pi \times 0.52 \times 1.5$$
$$= 2.4504\,m^2$$

Step 10 of 13

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{50 \times 1.885}$$
$$= 0.01061 K/W$$
$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi k_{foam} L}$$
$$= \frac{\ln(0.23/0.2)}{2\pi \times 0.03 \times 1.5}$$
$$= 0.4943 K/W$$

Step 11 of 13

$$R_{fiberglass} = \frac{\ln(r_3/r_2)}{2\pi k_{fiberglass} L}$$
$$= \frac{\ln(0.26/0.23)}{2\pi \times 0.035 \times 1.5}$$
$$= 0.3716 K/W$$
$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{12 \times 2.4504}$$
$$= 0.034 K/W$$

Step 12 of 13

All the resistances are in series, and the total resistance is

$= 0.034\,\text{K/W}$

Step 12 of 13

All the resistances are in series, and the total resistance is

$$\begin{aligned} R_{total} &= R_i + R_{foam} + R_{fiberglass} + R_o \\ &= 0.01061 + 0.4943 + 0.3716 + 0.034 \\ &= 0.91052\,\text{K/W} \end{aligned}$$

Then the steady rate of heat loss from the steam per unit length becomes

$$\begin{aligned} \dot{Q} &= \frac{(T_w - T_{\infty,2})}{R_{total}} \\ &= \frac{(60° - 37°)}{0.91052} \\ &= 36.2431\,\text{W} \end{aligned}$$

The energy saving is

$$\begin{aligned} \text{saving} &= 60.734 - 36.2431 \\ &= 24.5\,\text{W} \end{aligned}$$

Step 13 of 13

The time necessary for this additional insulation to pay for its cost of \$30 is then determined to be

$$\text{cost of extra insulation} = \text{saving} \times \text{time period} \times \text{unit cost}$$

$$\$30 = \frac{24.5}{1000} \times \text{time period} \times (\$0.08/\text{kWh})$$

$$\begin{aligned} \text{time period} &= 15306.12245\,\text{h} \\ &= 15306.12245(\text{h}) \times \left(\frac{1\,\text{year}}{24(\text{h}) \times 365} \right) \\ &= 1.7472\,\text{years} \\ &= 21\,\text{months} \end{aligned}$$

Therefore the payback period is $\boxed{\text{payback period} = 21\,\text{months}}$

Question: 17.071P

Reconsider Prob. 17-70. Using EES (or other) software, plot the fraction of energy cost of hot water due to the heat loss from the tank as a function of the hot-water temperature in the range of 40°C to 90°C. Discuss the results.

Step 1 of 8

Given data:

Height of the tank, $L = 1.5\,\text{m}$

Diameter of the heater, $d_1 = 40\text{cm} = 0.4\text{m}$

Diameter of the sheet metal tank, $d_2 = 46\text{cm} = 0.46\text{m}$

Heat transfer coefficient on the inner surface of the heater, $h_i = 50\,\text{W/m}^2\text{-K}$

Heat transfer coefficient on the outer surface of the heater, $h_0 = 12\,\text{W/m}^2\text{-K}$

Thermal conductivity of the foam insulation, $k_{foam} = 0.03\text{W/m-K}$

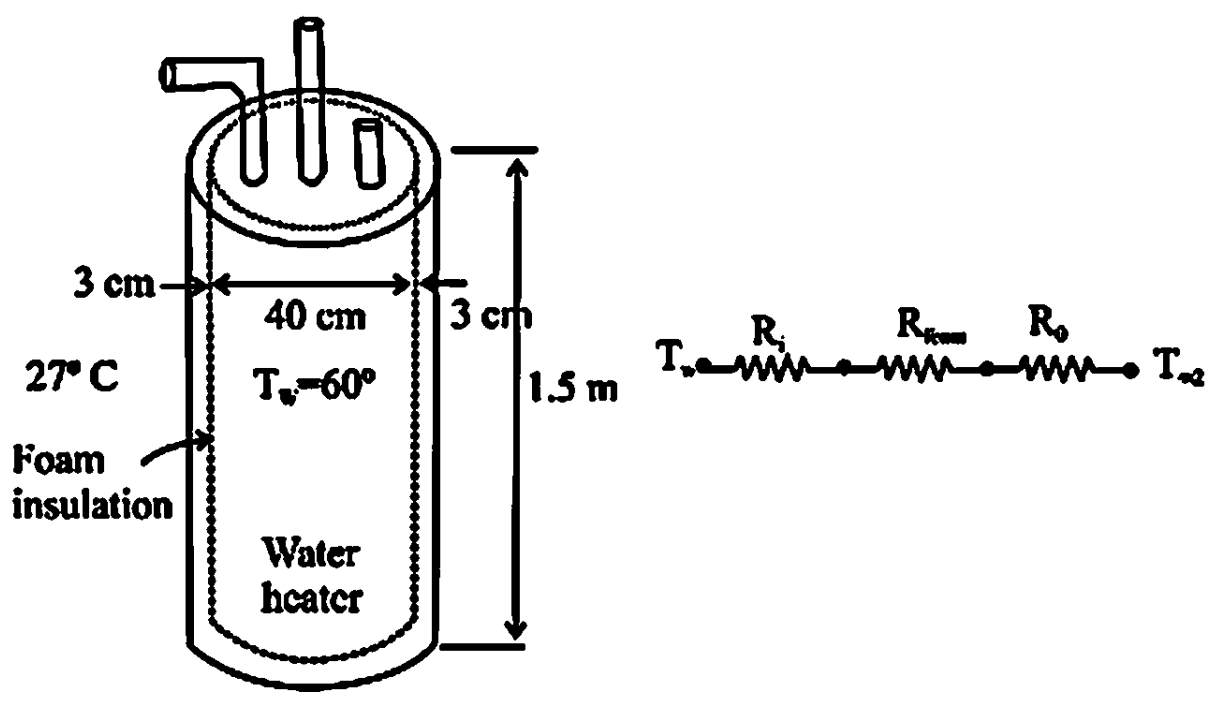
Inside water temperature, $T_w = 40 - 90^\circ\text{C}$

Surroundings temperature, $T_{\infty,2} = 27^\circ\text{C}$

Price of electricity = \$0.08/kWh

Amount of money paid by house owner = \$280/year

Step 2 of 8



Step 3 of 8

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi d_1 L$$
$$= \pi \times (0.4\text{m}) \times (1.5\text{m})$$
$$= 1.885\,\text{m}^2$$
$$A_0 = \pi d_2 L$$
$$= \pi \times (0.46\text{m}) \times (1.5\text{m})$$
$$= 2.1677\,\text{m}^2$$

Step 4 of 8

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{50(\text{W/m}^2\text{-K}) \times 1.885(\text{m}^2)}$$
$$= 0.01061\text{K/W}$$
$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi k_{foam} L}$$
$$= \frac{\ln(0.23/0.2)}{2\pi \times 0.03(\text{W/m-K}) \times 1.5(m)}$$
$$= 0.4943\text{K/W}$$
$$R_0 = \frac{1}{h_0 A_0}$$
$$= \frac{1}{12(\text{W/m}^2\text{-K}) \times 2.1677(\text{m}^2)}$$
$$= 0.038443\text{K/W}$$

Step 5 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{foam} + R_0$$
$$= 0.01061(\text{K/W}) + 0.4943(\text{K/W}) + 0.038443(\text{K/W})$$
$$= 0.54335\text{K/W}$$

Then the steady rate of heat loss from the steam per unit length becomes

$$\dot{Q} = \frac{(T_w - T_{\infty,2})}{R_{total}}$$
$$= \frac{(T_w - 37)}{0.54335(\text{K/W})}$$

Step 6 of 8

The amount and cost of heat loss per year are

$$Q = \dot{Q}\Delta t$$
$$= \frac{[(T_w - 37)/0.54335]}{1000}(\text{kW}) \times (365 \times 24\text{h/year})$$

Cost of Energy = Amount of Energy × Unit cost

$$= \frac{[(T_w - 37)/0.54335]}{1000} \times (365 \times 24)(\text{kWh/year}) \times (\$0.08/\text{kWh})$$

Therefore, the fraction of the hot water energy cost of this household that is due to the heat loss from the tank calculated as

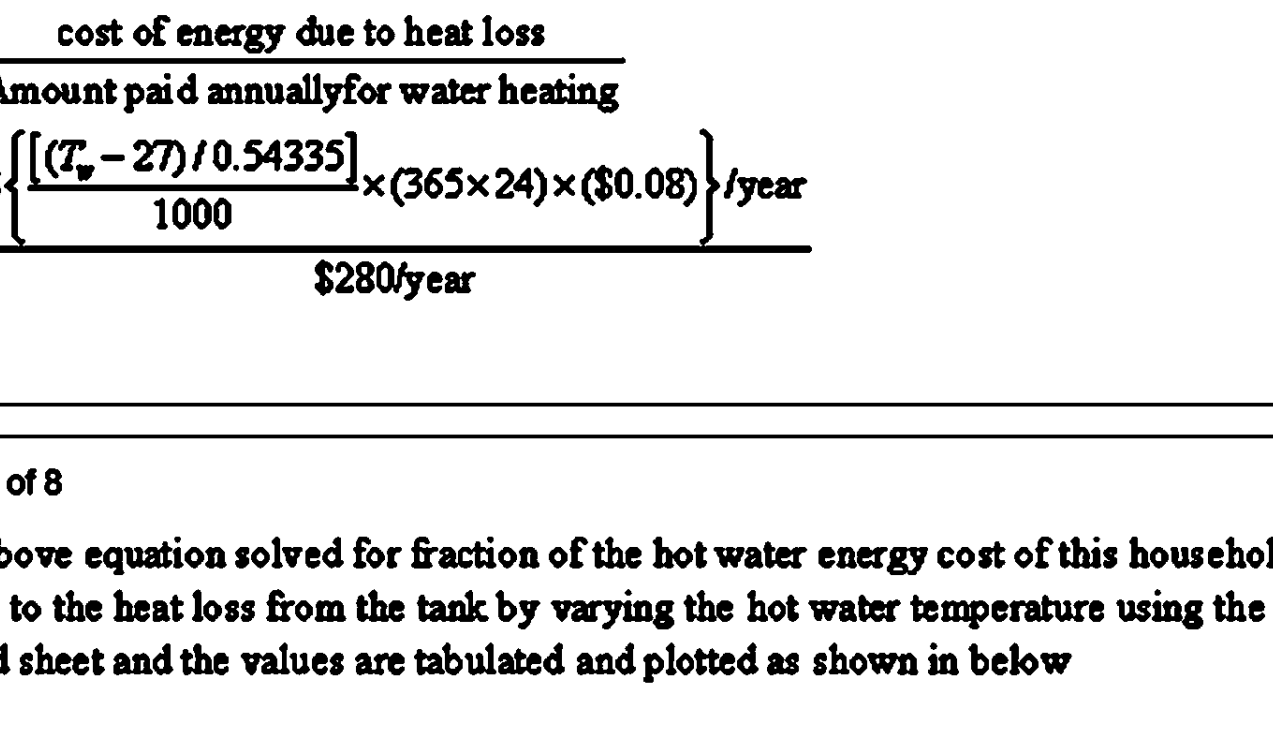
$$f = \frac{\text{cost of energy due to heat loss}}{\text{Amount paid annuallyfor water heating}}$$
$$= \frac{\$\left\{\frac{[(T_w - 27)/0.54335]}{1000} \times (365 \times 24) \times (\$0.08)\right\}/\text{year}}{\$280/\text{year}}$$

Step 7 of 8

The above equation solved for fraction of the hot water energy cost of this household that is due to the heat loss from the tank by varying the hot water temperature using the Excel spread sheet and the values are tabulated and plotted as shown in below

Hot water Temperature, T_w(°C)	f (%)
40	5.988
45	8.292
50	10.595
55	12.898
60	15.201
65	17.504
70	19.808
75	22.111
80	24.414
85	26.717
90	29.021

Step 8 of 8



Question: 17.072P

Chilled water enters a thin-shelled. 4-cm-diameter, 200-m-long pipe at 7°C at a rate of 0.98 kg/s and leaves at 8°C. The pipe is exposed to ambient air at 30°C with a heat transfer coefficient of 9 W/m2· K. If the pipe is to be insulated with glass wool insulation (k= 0.05 W/m · K) in order to decrease the temperature rise of water to 0.25°C, determine the required thickness of the insulation.

Step 1 of 6

Mean temperature of the pipe,

$$T_w = \frac{T_i + T_e}{2}$$

Substitute 7°C for T_i and 8°C for T_e,

$$T_w = \frac{T_i + T_e}{2} = \frac{8+7}{2} = 7.5°C$$

Heat transfer from the pipe without any insulation,

$$Q_{old} = \dot{m} c_p (T_e - T_i)$$

Here,

Flow rate of chilled water is $\dot{m}$

Substitute 0.98kg/s for $\dot{m}$, $T_\infty = 30°C$ and 4.18 for c_p,

$$Q_{old} = \dot{m} c_p (T_e - T_i) = 0.98 \times 4.18(8-7) = 4.0964 kW = 4096.4 W$$

Step 2 of 6

Heat transfer from the pipe can also be calculated as,

$$Q_{old} = \frac{T_\infty - T_w}{R_{total}}$$

Substitute 4096.4 W for Q_{old}, 30°C for T_∞ and 7.5°C for T_w,

$$Q_{old} = \frac{T_\infty - T_w}{R_{total}}$$

$$4096.4 = \frac{30 - 7.5}{R_{total}}$$

$$R_{total} = 0.00549 K/W$$

The convection resistance from the outer surface of the pipe,

$$R_o = \frac{1}{h_o A_o} = \frac{1}{h_o \times \pi DL} = \frac{1}{9 \times \pi \times 0.04 \times 200} = 0.00442 K/W$$

Step 3 of 6

The inside convective resistance and conduction resistance due to pipe thickness,

$$R_1 = R_{total} - R_o$$

Substitute 0.00549 K/W for R_{total} and 0.00442 K/W for R_o,

$$R_1 = R_{total} - R_o = 0.00549 - 0.00442 = 0.00107 K/W$$

If temperature rise of water is $\Delta T = 0.25°C$ then the rate of heat transfer with new insulation is,

$$Q_{new} = \dot{m} c_p \Delta T = 0.98 \times 4.18 \times 0.25 = 1.0241 kW = 1024.1 W$$

Step 4 of 6

The total thermal resistance with insulation can be calculated as,

$$Q_{new} = \frac{T_\infty - T_w}{R_{total,new}}$$

Substitute 1024.1 W for Q_{new}, 30°C for T_∞ and 7.125°C for T_w,

$$Q_{new} = \frac{T_\infty - T_w}{R_{total,new}}$$

$$1024.1 = \frac{30 - \left(\frac{7+7.25}{2}\right)}{R_{total,new}}$$

$$R_{total,new} = 0.0223 K/W$$

Step 5 of 6

The total thermal resistance with insulation can be written as

$$R_{total,new} = R_1 + R_{o,new} + R_{ins}$$

$$R_{total,new} = R_1 + \frac{1}{h_o A_{o,new}} + \frac{\ln\left(\frac{D_2}{D_1}\right)}{2\pi kL}$$

$$0.0223 = 0.00107 + \frac{1}{9 \times \pi D_2 \times 200} + \frac{\ln \frac{D_2}{0.04}}{2\pi \times 0.05 \times 200}$$

$$0.02123 = \frac{1}{5654.866 D_2} + \frac{\ln \frac{D_2}{0.04}}{62.83}$$

By solving the above equation using any equation solver we get $D_2 = 0.140267 m$

Step 6 of 6

Then the thickness of the insulation can be calculated as,

$$t = \frac{D_2 - D_1}{2} = \frac{0.140267 - 0.04}{2} = 0.0501 m$$

$$\boxed{t = 5.01 cm}$$

Therefore, the thickness of the insulation is 5.01cm

Question: 17.073P

Superheated steam at an average temperature 200°C is transported through a steel pipe (k= 50 W/m · K, $D0$ = 8.0 cm, Di =6.0 cm, and L= 20.0 m). The pipe is insulated with a 4-cm thick layer of gypsum plaster (k =0.5 W/m · K). The insulated pipe is placed horizontally inside a warehouse where the average air temperature is 10°C. The steam and the air heat transfer coefficients are estimated to be 800 and 200 W/m2· K, respectively. Calculate (*a*) the daily rate of heat transfer from the superheated steam, and (*b*) the temperature on the outside surface of the gypsum plaster insulation.

Step 1 of 5

Temperature of superheated steam $T_s = 200°C$

Thermal conductivity of the steel pipe $k_{steel} = 50\,W/m \cdot K$

Outer diameter of the steel pipe $D_o = 8.0\,cm = 0.08\,m$

Inner diameter of the steel pipe $D_i = 6.0\,cm = 0.06\,m$

Length of the steel pipe $L = 20.0\,m$

Thickness of the gypsum plaster $t_{gyp} = 4\,cm = 0.04\,m$

Thermal conductivity of the gypsum plaster $k_{gyp} = 0.5\,W/m \cdot K$

Average air temperature $T_{air} = 10°C$

Heat transfer coefficient of the steam $h_i = 800\,W/m^2.K$

Heat transfer coefficient of the air $h_o = 200\,W/m^2.K$

Step 2 of 5

The thermal resistance network from superheated steam to ambient air can be represented as,



Step 3 of 5

Step 4 of 5

The individual resistances can be calculated as;

Convection resistance due to superheated steam,

$$R_i = \frac{1}{h_i A}$$
$$= \frac{1}{h_i \times \pi D_i L}$$
$$= \frac{1}{800 \times \pi \times 0.06 \times 20}$$
$$= 0.00033\,K/W$$

Conduction resistance due to thickness of the steel pipe is

$$R_{steel} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi k_{steel} L}$$
$$= \frac{\ln\left(\frac{0.08}{0.06}\right)}{2 \times \pi \times 50 \times 20}$$
$$= 0.0000457\,K/W$$

Conduction resistance due to thickness of the insulation

$$R_{ins} = \frac{\ln\left(\frac{D_o + 2t}{D_o}\right)}{2\pi k_{gyp} L}$$
$$= \frac{\ln\left(\frac{0.08 + 2 \times 0.04}{0.08}\right)}{2 \times \pi \times 0.5 \times 20}$$
$$= 0.01103\,K/W$$

Convection resistance due to air,

$$R_o = \frac{1}{h_o A}$$
$$= \frac{1}{h_o \times \pi (D_o + 2t) L}$$
$$= \frac{1}{200 \times \pi \times (0.08 + 2 \times 0.04) \times 20}$$
$$= 0.000497\,K/W$$

Step 5 of 5

Therefore, the total thermal resistance can be determined as

$$R_{total} = R_i + R_{steel} + R_{ins} + R_o$$
$$= 0.00033 + 0.0000457 + 0.01103 + 0.000497$$
$$= 0.0119027\,K/W$$

Rate of heat transfer from the superheated steam can be calculated as

$$\dot{Q} = \frac{T_s - T_{air}}{R_{total}}$$
$$= \frac{200 - 10}{0.0119027}$$
$$= 15962.76\,W$$

$$\boxed{\dot{Q} = 15.962\,kW}$$

(b)Assume temperature on the outside surface of the gypsum plaster insulation is T_{ins}

The rate of heat transfer on the outer surface of the pipe is

$$\dot{Q} = \frac{T_{ins} - T_{air}}{R_o}$$
$$15962.76 = \frac{T_{ins} - 10}{0.000497}$$
$$T_{ins} = 17.933°C$$

Therefore, temperature of the outside surface of the gypsum plaster insulation is $\boxed{T_{ins} = 17.933°C}$

Question: 17.074P

Steam at 450°F is flowing through a steel pipe (k = 8.7 Btu/h · ft·°F) whose inner and outer diameters are 3.5 in and 4.0 in, respectively, in an environment at 55°F. The pipe is insulated with 2-in-thick fiberglass insulation (k = 0.020 Btu/h · ft · °F). If the heat transfer coefficients on the inside and the outside of the pipe are 30 and 5 Btu/h·ft2·°F, respectively, determine the rate of heat loss from the steam per foot length of the pipe. What is the error involved in neglecting the thermal resistance of the steel pipe in calculations?

Step 1 of 11

Given: -

Inner diameter of pipe $D_i = 3.5\text{in}$

Outer diameter of pipe $D_o = 4.0\text{in}$

Thickness of the insulation $t = 2\text{in}$

Thermal conductivity of fiberglass $k_{fiber\ glass} = 0.020\ \text{Btu/h·ft·°F}$

Thermal conductivity of steel $k_{steel} = 8.7\ \text{Btu/h·ft·°F}$

Heat transfer coefficient on inside of the pipe $h_i = 30\ \text{Btu/h.ft}^2$

Convection heat transfer coefficient outside pipe $h_o = 5\ \text{Btu/h.ft}^2$

Ambient temperature $T_\infty = 55^0\text{F}$

Steam temperature $T = 450^0\text{F}$

Step 2 of 11

If the heat transfer is steady since there is no indication of any change with time and the heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction

The inner and outer surface areas of the insulated pipe per foot length of the pipe are

$$A_i = \pi D_i L$$
$$= \pi(3.5/12)(1)$$
$$A_i = 0.916\ \text{ft}^2$$

$$A_o = \pi(D_o + 2t)L$$
$$= \pi(8/12)(1)$$
$$A_o = 2.094\ \text{ft}^2$$

Step 3 of 11

Thermal resistance circuit of this present system is



Step 4 of 11

The convection resistances on inside of the pipe is

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{(30)\times(0.916)}$$
$$R_i = 0.0363\ \text{h}\cdot{}^\circ\text{F/Btu}$$

Step 5 of 11

Conduction resistance from inner surface to outer surface of the pipe

$$R_1 = R_{pipe} = \frac{\ln(r_o/r_i)}{2\pi k_{steel} L}$$
$$= \frac{\ln(2/1.75)}{2\pi(8.7)\times(1)}$$
$$R_1 = R_{pipe} = 0.00244\ \text{h}\cdot{}^\circ\text{F/Btu}$$

Step 6 of 11

Conduction resistance from outer surface to insulation of the pipe

$$R_2 = R_{insulation}$$
$$= \frac{\ln(r_3/r_o)}{2\pi k_2 L}$$
$$= \frac{\ln((r_o+t)/r_o)}{2\pi k_2 L}$$
$$= \frac{\ln(4/2)}{2\pi(0.020)\times(1)}$$
$$R_2 = R_{insulation} = 5.516\ \text{h}\cdot{}^\circ\text{F/Btu}$$

Step 7 of 11

Convection resistance from outer surface of the pipe to environment

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{(5)\times(2.094)}$$
$$R_o = 0.096\ \text{h}\cdot{}^\circ\text{F/Btu}$$

Step 8 of 11

Total resistance is

$$R_{total} = R_i + R_1 + R_2 + R_o$$
$$= 0.0363 + 0.00244 + 5.516 + 0.096$$
$$R_{total} = 5.6507\ \text{h}\cdot{}^\circ\text{F/Btu}$$

Step 9 of 11

Then the steady rate of heat loss from the steam per ft. pipe length becomes

$$\dot{Q} = \frac{T - T_\infty}{R_{total}}$$
$$= \frac{(450-55)}{5.6507}$$
$$\dot{Q} = 69.9028\ \text{Btu/h}$$

Step 10 of 11

If the thermal resistance of the steel pipe is neglected, the new value of total thermal resistance will be

$$R_{total,1} = R_i + R_2 + R_o$$
$$= 0.0363 + 5.516 + 0.096$$
$$R_{total,1} = 5.648\ \text{h°F/Btu}$$

Step 11 of 11

Then the percentage error involved in neglecting the thermal resistance of the steel pipe is

$$\%error = \frac{R_{total} - R_{total,1}}{R_{total}}$$
$$\%error = \frac{(5.6507 - 5.648)}{5.6507}\times 100$$
$$\boxed{\%error = 0.0477\%}$$

Error percentage is not significant

Question: 17.075P pg 1

Hot water at an average temperature of 90°C is flowing through a 15-m section of a cast iron pipe (k = 52 W/m · K) whose inner and outer diameters are 4 cm and 4.6 cm, respectively. The outer surface of the pipe, whose emissivity is 0.7, is exposed to the cold air at 10°C in the basement, with a heat transfer coefficient of 15 W/m2 · K. The heat transfer coefficient at the inner surface of the pipe is 120 W/ m2 · K. Taking the walls of the basement to be at 10°C also, determine the rate of heat loss from the hot water. Also, determine the average velocity of the water in the pipe if the temperature of the water drops by 3°C as it passes through the basement.

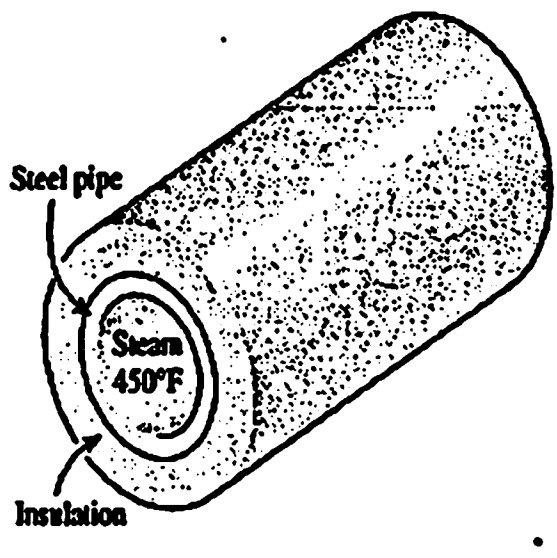
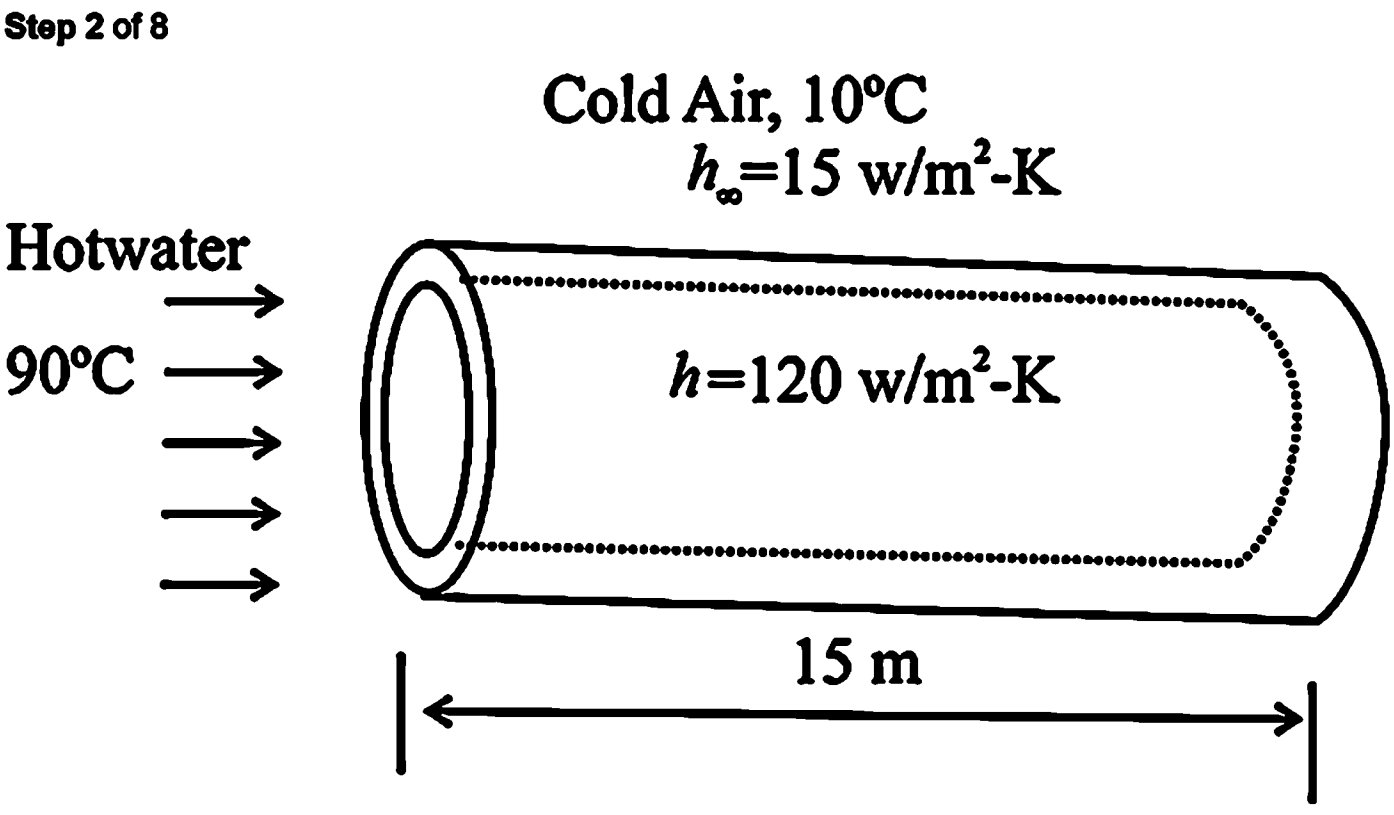


FIGURE P17–75

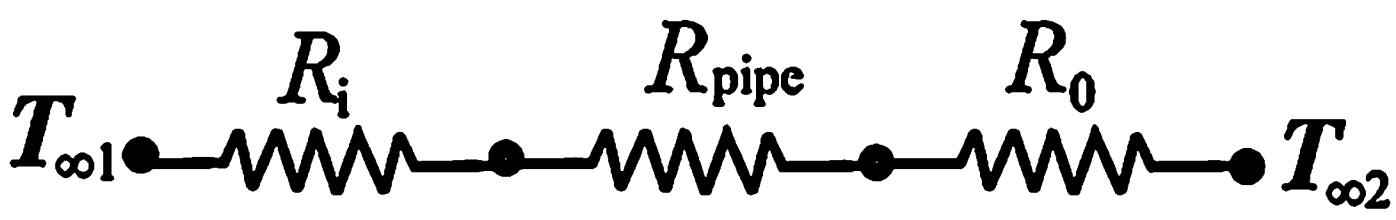
Step 1 of 8

Draw the schematic representation of the given system as shown below:

Step 2 of 8



Draw the thermal resistance circuit from inner surface to outer surface of the pipe as shown below:



Step 3 of 8

Find the inner surface area of the insulated pipe per unit length,

$$A_i = \pi d_1 L$$

Here, d_1 is inner diameter of the pipe, L is length of the pipe.

Substitute $0.04\,\text{m}$ for d_1 and 15 m for L

$$A_i = \pi \times 0.04 \times 15$$
$$= 1.885\,\text{m}^2$$

Find the outer surface area of the insulate pipe per unit length,

$$A_o = \pi d_2 L$$

Here, d_2 is outer diameter of the pipe.

Substitute $0.046\,\text{m}$ for d_2

$$A_o = \pi \times 0.046 \times 15$$
$$= 2.1677\,\text{m}^2$$

Step 4 of 8

Find convective thermal resistance due to hot water flowing through the pipe,

$$R_i = \frac{1}{h_i A_i}$$

Here, h_i is heat transfer coefficient at the inner surface of the pipe

Substitute $120\,\text{W/m}^2 \cdot \text{K}$ for h_i

$$R_i = \frac{1}{120 \times 1.885}$$
$$= 0.004421\,\text{K/W}$$

Find conductive resistance of the pipe by using the equation,

$$R_{pipe} = \frac{\ln(r_2 / r_1)}{2\pi k_{pipe} L}$$

Here, k_{pipe} is thermal conductivity of the pipe.

Substitute $52\,\text{W/m} \cdot \text{K}$ for k

$$R_{pipe} = \frac{\ln(0.023/0.02)}{2\pi \times 52 \times 15}$$
$$= 0.00002852\,\text{K/W}$$

Step 5 of 8

The outer surface temperature of the pipe will be little below the water temperature. Assume the outer surface temperature of the pipe to be 70°C (we will check this assumption later),

Find the radiation heat transfer coefficient by using the equation,

Step 5 of 8

The outer surface temperature of the pipe will be little below the water temperature. Assume the outer surface temperature of the pipe to be 70°C (we will check this assumption later),

Find the radiation heat transfer coefficient by using the equation,

$$h_{rad} = \varepsilon\sigma(T_2^2 + T_{surr}^2)(T_2 + T_{surr})$$

Here, ε is emissivity, σ is Stefan Boltzmann constant, T_2 is outer surface temperature and T_{surr} is surrounding temperature.

Substitute 0.7 for ε, $5.67\times10^{-8}\,\mathrm{W/m^2\cdot K^4}$ for σ, $70+273=343\,\mathrm{K}$ for T_2 and $10°\mathrm{C}=283\,\mathrm{K}$ for T_{surr}

$$\begin{aligned} h_{rad} &= 0.7\times5.67\times10^{-8}\times\left[343.15^2+283.15^2\right]\times\left[343.15+283.15\right] \\ &= 4.92\,\mathrm{W/m^2\cdot K} \end{aligned}$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$h_{combined} = h_{rad} + h_{conv}$$

Here, h_{conv} is convection heat transfer coefficient

Substitute $15\,\mathrm{W/m^2\cdot K}$ for h_{conv}

$$\begin{aligned} h_{combined} &= 4.92+15 \\ &= 19.92\,\mathrm{W/m^2\cdot K} \end{aligned}$$

Find convective thermal resistance from outer surface of the pipe due to cold air,

$$\begin{aligned} R_o &= \frac{1}{h_{combined}A_o} \\ &= \frac{1}{19.92\times2.1677} \\ &= 0.02315\,\mathrm{K/W} \end{aligned}$$

All the resistances are in series, and the total resistance becomes,

$$\begin{aligned} R_{total} &= R_i + R_{pipe} + R_o \\ &= 0.004421+0.00002852+0.02315 \\ &= 0.0276\,\mathrm{K/W} \end{aligned}$$

Find the steady rate of heat loss from the hot water by using the equation,

$$\dot{Q} = \frac{(T_{\infty,1}-T_{\infty,2})}{R_{total}}$$

Here, $T_{\infty,1}$ is hot water temperature and $T_{\infty,2}$ is cold air temperature

Substitute $90°\mathrm{C}$ for $T_{\infty,1}$ and $10°\mathrm{C}$ for $T_{\infty,2}$

$$\begin{aligned} \dot{Q} &= \frac{90-10}{0.0276} \\ &= 2898.5\,\mathrm{W} \end{aligned}$$

Therefore, the steady rate of heat loss from the hot water pipe is, $\boxed{2898.5\,\mathrm{W}}$

Step 6 of 8

Find the mass flow rate of water for a temperature drop of $3°\mathrm{C}$ by using the equation,

$$\dot{Q} = \dot{m}c_p\Delta T$$

Here, c_p is specific heat of water

Substitute $4180\,\mathrm{J/kg\cdot K}$ for c_p

$$\begin{aligned} \dot{m} &= \frac{\dot{Q}}{c_p\Delta T} \\ &= \frac{2898.5}{4180\times3} \\ &= 0.2311\,\mathrm{kg/s} \end{aligned}$$

Step 7 of 8

Find the velocity of water in the pipe by using the equation,

$$\dot{m} = \rho AV$$

Here, ρ is density of water

Substitute $1000\,\mathrm{kg/m^3}$ for ρ

$$\begin{aligned} V &= \frac{\dot{m}}{\rho A} \\ &= \frac{\dot{m}}{\rho\times\frac{\pi}{4}d_1^2} \\ &= \frac{0.2311}{1000\times\frac{\pi}{4}\times0.04^2} \\ &= 0.184\,\mathrm{m/s} \end{aligned}$$

Therefore, the velocity of water in the pipe, $\boxed{0.184\,\mathrm{m/s}}$

Step 8 of 8

Verify the outer surface temperature is correct or not

The outer surface temperature of the pipe by using the equation,

$$\dot{Q} = \frac{(T_{\infty,1}-T_s)}{R_i + R_{pipe}}$$

$$2898.5 = \frac{(90-T_s)}{0.004421+0.00002852}$$

$T_s = 77.18°\mathrm{C}$

$$\dot{Q} = \frac{(T_{\infty,1} - T_s)}{R_1 + R_{pipe}}$$

$$2898.5 = \frac{(90 - T_s)}{0.004421 + 0.00002852}$$

$$T_s = 77.1°C$$

This is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

Question: 17.076P

Repeat Prob. 17-75 for a pipe made of copper (k = 386 W/m · K) instead of cast iron.

Step 1 of 8

Given data:

Average temperature of hot water, $T_w = 90°C$

Length of the pipe, $L = 15m$

Thermal conductivity of the copper, $k_{copper} = 386 W/m\text{-}K$

Inner diameter of the pipe, $d_1 = 4cm = 0.04m$

Outer diameter of the pipe, $d_2 = 4.6cm = 0.046m$

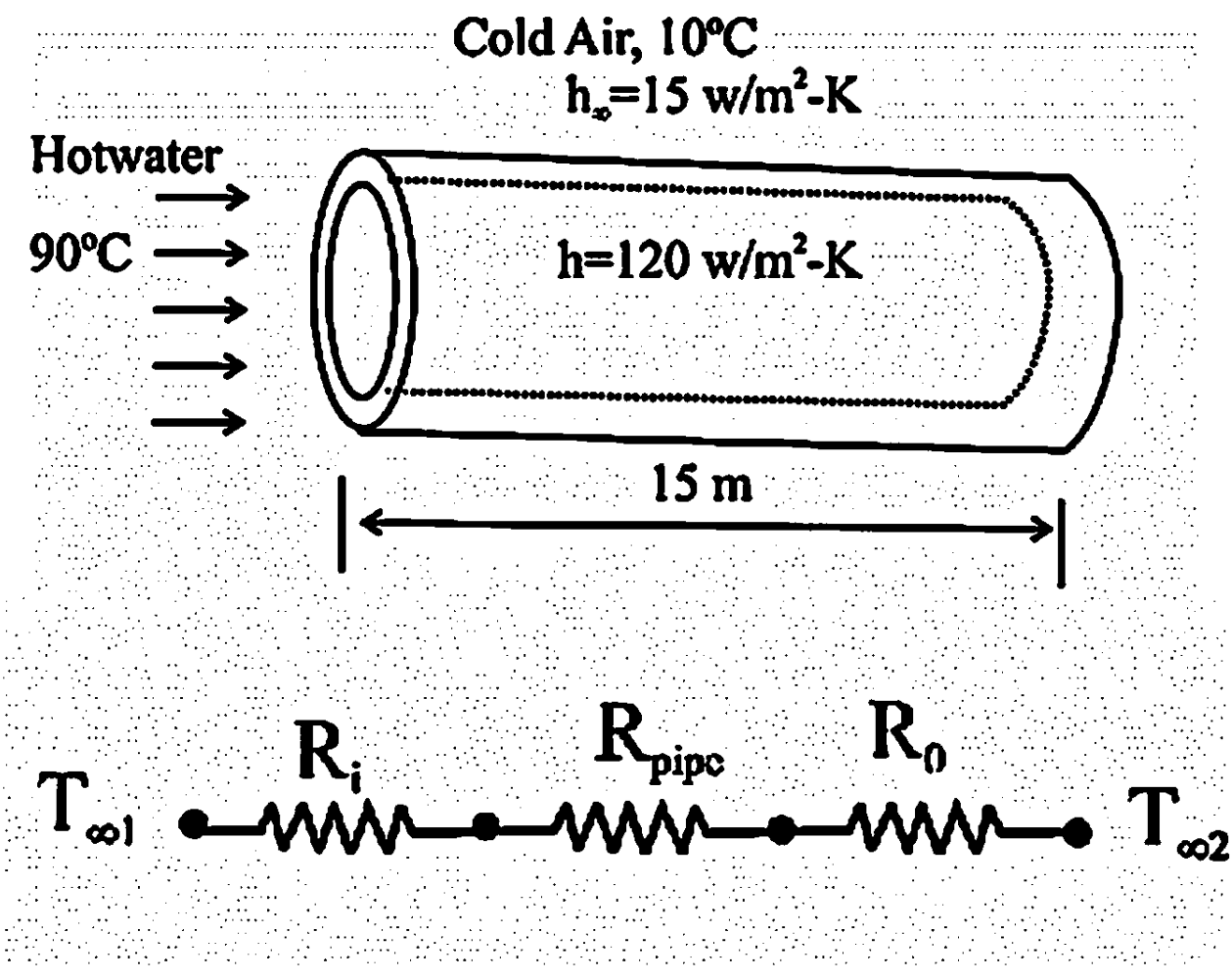
Emissivity of the outer surface, $\varepsilon = 0.7$

Cold air temperature in the basement, $T_\infty = 10°C$

Heat transfer coefficient on the outer surface of the pipe, $h_0 = 15 W/m^2\text{-}K$

Heat transfer coefficient on the inner surface of the heater, $h_i = 120 W/m^2\text{-}K$

Step 2 of 8



Step 3 of 8

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi d_1 L$$
$$= \pi \times (0.4m) \times (1.5m)$$
$$= 1.885\ m^2$$
$$A_o = \pi d_2 L$$
$$= \pi \times (0.46m) \times (1.5m)$$
$$= 2.1677\ m^2$$

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{120(W/m^2\text{-}K) \times 1.885(m^2)}$$
$$= 0.004421 K/W$$
$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k_{pipe} L}$$
$$= \frac{\ln(0.23/0.2)}{2\pi \times 386(W/m\text{-}K) \times 15(m)}$$
$$= 0.000003842 K/W$$

Step 4 of 8

The outer surface temperature of the pipe will be somewhat below the water temperature. Assuming the outer surface temperature of the pipe to be 70°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be

$$h_{rad} = \varepsilon\sigma(T_2^2 + T_{surr}^2)(T_2 + T_{surr})$$
$$= 0.7 \times 5.67 \times 10^{-8}(W/m^2\text{-}K^4) \times \left[(343.15K)^2 + (283.15K)^2\right] \times \left[(343.15K) + (283.15K)\right]$$
$$= 4.92\ W/m^2\text{-}K$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$h_{combined} = h_{rad} + h_{conv}$$
$$= (4.92 + 15)\ W/m^2\text{-}K$$
$$= 19.92\ W/m^2\text{-}K$$
$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{19.92(W/m^2\text{-}K) \times 2.1677(m^2)}$$
$$= 0.02315 K/W$$

Step 5 of 8

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{pipe} + R_o$$
$$= 0.0044221(K/W) + 0.000003842(K/W) + 0.02315(K/W)$$
$$= 0.0275 K/W$$

The steady rate of heat loss from the hot water pipe becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(90 - 10)}{0.0275(K/W)}$$
$$= 2901.2 W$$

$$\boxed{\dot{Q} = 2901.2 W}$$

Step 6 of 8

For a temperature drop of 3°C, the mass flow rate of water and the average velocity of water must be

$$\dot{Q} = \dot{m} C_p \Delta T$$

Therefore, the mass flow rate of the water can be expressed as

$$\dot{m} = \frac{\dot{Q}}{C_p \Delta T}$$
$$= \frac{2901.2(W)}{4180(J/kg\text{-}K) \times (3K)}$$
$$= 0.2313 kg/s$$

$$\boxed{\dot{m} = 0.2311 kg/s}$$

Step 7 of 8

Velocity if the water in the pipe can be expressed as

$$V = \frac{\dot{m}}{\rho A}$$
$$= \frac{0.23113(kg/s)}{1000(kg/m^3) \times \frac{\pi}{4} \times 0.04^2(m^2)}$$
$$= 0.1841 m/s$$

$$\boxed{V = 0.1841 m/s}$$

Step 8 of 8

Discussion:

The outer surface temperature of the pipe calculated as

$$\dot{Q} = \frac{(T_{\infty,1} - T_s)}{R_i + R_{pipe}}$$
$$2901.2 W = \frac{(90 - T_s)}{0.004421(K/W) + 0.000003842(K/W)}$$
$$T_s = 77.16°C$$

this is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

Question: 17.077P

Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper pipes (k = 223 Btu/h · ft · °F) of inner diameter 0.4 in and outer diameter 0.6 in at an average temperature of 70°F. The heat of vaporization of water at 100°F is 1037 Btu/lbm. The heat transfer coefficients are 2400 Btu/h · ft · °F on the steam side and 35 Btu/h·ft2·°F on the water side. Determine the length of the tube required to condense steam at a rate of 250 lbm/h. tower.- 2380 ft

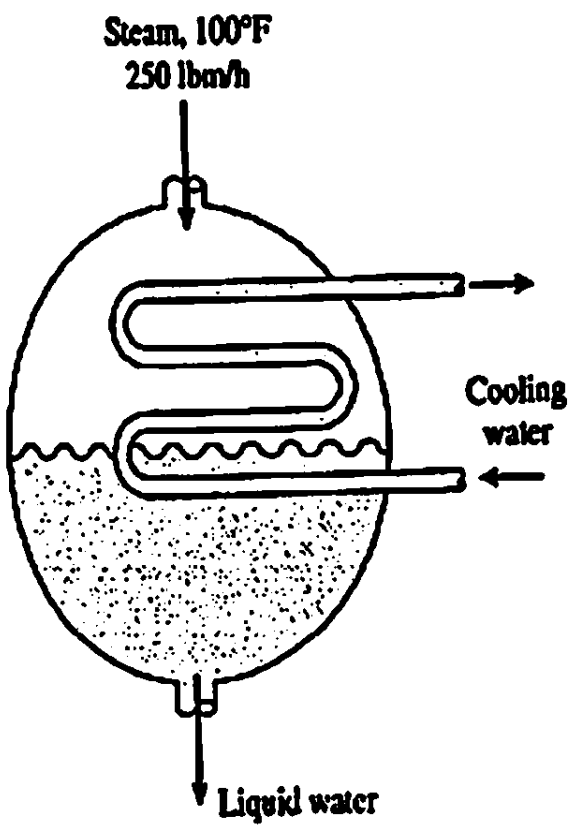


FIGURE P17–77E

Step 1 of 6

Given data:

Turbine exit temperature, $T_{\infty 1} = 100^{\circ}\text{F}$

Cooling water temperature, $T_{\infty 2} = 70^{\circ}\text{F}$

Mass flow rate of the steam, $\dot{m}_s = 250\text{lbs/h}$

Thermal conductivity of the copper pipe, $k_{copper} = 223\text{Btu/h-ft-}^{\circ}\text{F}$

Heat of vaporization of water, $h_{fg} = 1037\text{ Btu/lbm @}100^{\circ}\text{F}$

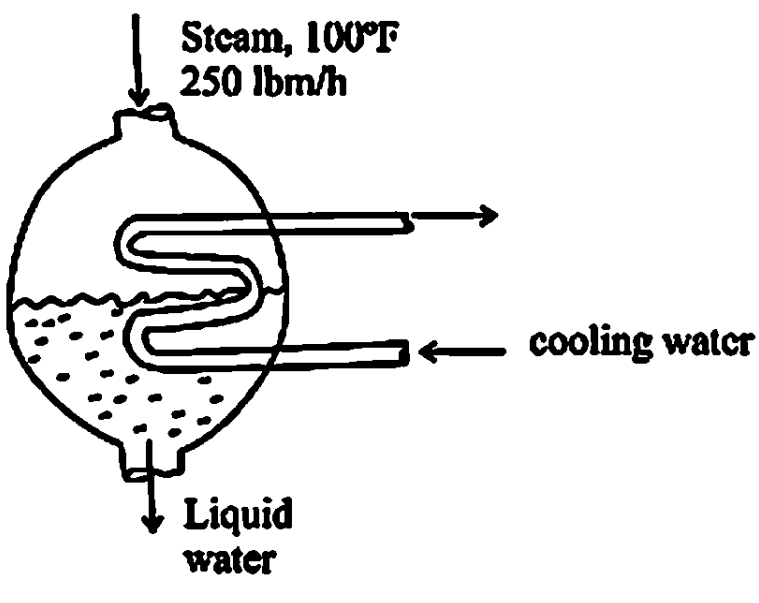
Inner diameter of the pipe, $d_1 = 0.4\text{in}$

Outer diameter of the pipe, $d_2 = 0.6\text{in}$

Heat transfer coefficient on the steam side, $h_o = 2400\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}$

Heat transfer coefficient on the water side, $h_i = 35\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}$

Step 2 of 6



Step 3 of 6

The inner and outer surface areas of the cooling pipe per unit length are calculated as

$$A_i = \pi d_1 L$$
$$= \pi \times (\frac{0.4}{12}\text{ft}) \times (1\text{ft})$$
$$= 0.10472\text{ ft}^2$$
$$A_o = \pi d_2 L$$
$$= \pi \times (\frac{0.6}{12}\text{ft}) \times (1\text{ft})$$
$$= 0.15708\text{ ft}^2$$

Step 4 of 6

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{35(\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}) \times 0.10472\ (\text{ft}^2)}$$
$$= 0.2728\text{h-}^{\circ}\text{F/Btu}$$
$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k_{copper} L}$$
$$= \frac{\ln(0.6/0.4)}{2\pi \times 223(\text{Btu/h-ft-}^{\circ}\text{F}) \times 1(\text{ft})}$$
$$= 0.00028\text{h-}^{\circ}\text{F/Btu}$$
$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{2400(\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}) \times 0.15708\ (\text{ft}^2)}$$
$$= 0.0026\text{h-}^{\circ}\text{F/Btu}$$

Step 5 of 6

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{pipe} + R_o$$
$$= 0.2728\,(\text{h-}^{\circ}\text{F/Btu}) + 0.000289\,(\text{h-}^{\circ}\text{F/Btu}) + 0.0026\,(\text{h-}^{\circ}\text{F/Btu})$$
$$= 0.27541\text{h-}^{\circ}\text{F/Btu}$$

The steady rate of heat rate per unit length of the tube can be determined as

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(100-70)^{\circ}\text{F}}{0.27541(\text{h-}^{\circ}\text{F/Btu})}$$
$$= 108.9285\text{Btu/h}$$

Step 6 of 6

The total rate of heat transfer required for condensing the steam at a rate of 250 lbm/h and the length of the tube required is determined to be

$$\dot{Q}_{total} = \dot{m}_s h_{fg}$$
$$= 250\,(\text{lbm/h}) \times 1037(\text{Btu/lbm})$$
$$= 259250\text{Btu/h}$$

$$\text{Tube length (L)} = \frac{\dot{Q}_{total}}{\dot{Q}}$$
$$= \frac{259250(\text{Btu/h})}{108.9285(\text{Btu/h})}$$
$$= 2380\text{ft}$$

Therefore, tube length is $\boxed{L = 2380\text{ft}}$

Question: 17.078P

Repeat Prob. 17-77E, assuming that a 0.01-in-thick layer of mineral deposit (k = 0.5 Btu/h·ft·°F) has formed on the inner surface of the pipe.

Step 1 of 6

Given data:

Turbine exit temperature, $T_{\infty 1} = 100^\circ F$

Cooling water temperature, $T_{\infty 2} = 70^\circ F$

Mass flow rate of the steam, $\dot{m}_s = 250 lbs/h$

Thickness of mineral deposit, $L = 0.01 in$

Thermal conductivity of the mineral deposit, $k_{deposit} = 0.5 Btu/h\text{-}ft\text{-}^\circ F$

Thermal conductivity of the copper pipe, $k_{copper} = 223 Btu/h\text{-}ft\text{-}^\circ F$

Heat of vaporization of water, $h_{fg} = 1037\ Btu/lbm\ @100^\circ F$

Inner diameter of the pipe, $d_1 = 0.4 in$

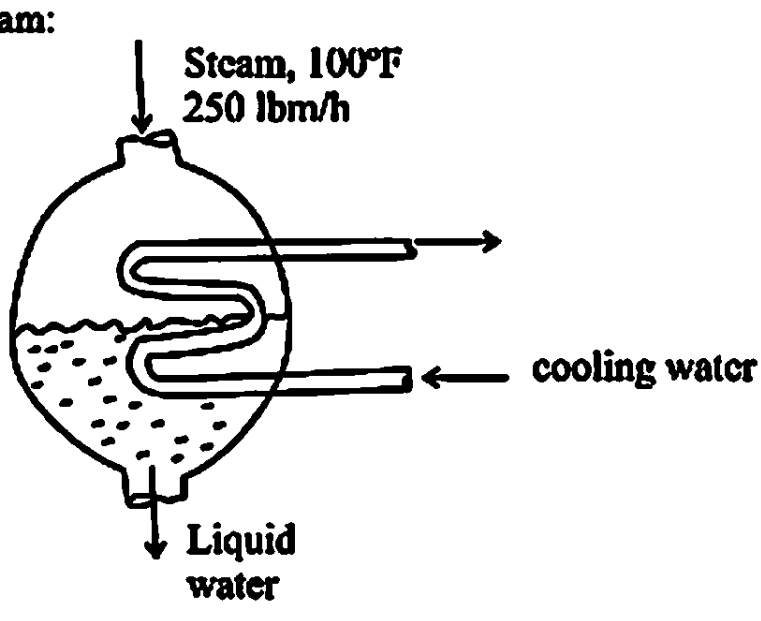
Outer diameter of the pipe, $d_2 = 0.6 in$

Heat transfer coefficient on the steam side, $h_o = 2400 Btu/h\text{-}ft^2\text{-}^\circ F$

Heat transfer coefficient on the water side, $h_i = 35 Btu/h\text{-}ft^2\text{-}^\circ F$

Step 2 of 6

Let us draw the diagram:



Step 3 of 6

The inner and outer surface areas of the cooling pipe per unit length are calculated as

$$A_i = \pi d_1 L$$

$$= \pi \times \left(\frac{0.4 - 0.02}{12} ft\right) \times (1 ft)$$

$$= 0.0995\ ft^2$$

$$A_o = \pi d_2 L$$

$$= \pi \times \left(\frac{0.6}{12} ft\right) \times (1 ft)$$

$$= 0.157\ ft^2$$

Step 4 of 6

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$

$$= \frac{1}{35\,(Btu/h\text{-}ft^2\text{-}^\circ F) \times 0.0995\ (ft^2)}$$

$$= 0.287\ h\text{-}^\circ F/Btu$$

$$R_{pipe} = \frac{\ln(r_2 / r_1)}{2\pi k_{copper} L}$$

$$= \frac{\ln(0.3/0.2)}{2\pi \times 223 (Btu/h\text{-}ft\text{-}^\circ F) \times 1 (ft)}$$

$$= 0.00028\ h\text{-}^\circ F/Btu$$

$$R_{deposit} = \frac{\ln(r_1 / r_0)}{2\pi k_{deposit} L}$$

$$= \frac{\ln(0.2/0.19)}{2\pi \times 0.5 (Btu/h\text{-}ft\text{-}^\circ F) \times 1 (ft)}$$

$$= 0.01632\ h\text{-}^\circ F/Btu$$

$$R_o = \frac{1}{h_o A_o}$$

$$= \frac{1}{2400\,(Btu/h\text{-}ft^2\text{-}^\circ F) \times 0.15708\ (ft^2)}$$

$$= 0.0026\ h\text{-}^\circ F/Btu$$

Step 5 of 6

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{pipe} + R_{deposit} + R_o$$

$$= 0.287\,(h\text{-}^\circ F/Btu) + 0.00028 (h\text{-}^\circ F/Btu) + 0.01632 + 0.0026 (h\text{-}^\circ F/Btu)$$

$$= 0.3062\ h\text{-}^\circ F/Btu$$

The steady rate of heat rate per unit length of the tube can be determined as

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$

$$= \frac{(100 - 70)^\circ F}{0.3062\,(h\text{-}^\circ F/Btu)}$$

$$= 97.975 Btu/h$$

Step 6 of 6

The total rate of heat transfer required for condensing the steam at a rate of 250 lbm/h and the length of the tube required is determined to be

$$\dot{Q}_{total} = \dot{m}_s h_{fg}$$

$$= 250\,(lbm/h) \times 1037 (Btu/lbm)$$

$$= 259250\ Btu/h$$

$$\text{Tube length}\,(L) = \frac{\dot{Q}_{total}}{\dot{Q}}$$

$$= \frac{259250 (Btu/h)}{97.975 (Btu/h)}$$

$$= 2646\ ft$$

Therefore, tube length is $\boxed{L = 2646 ft}$

Question: 17.079P pg 1

Reconsider Prob. 17-77E. Using EES (or other) software, investigate the effects of the thermal conductivity of the pipe material and the outer diameter of the pipe on the length of the tube required. Let the thermal conductivity vary from 10 Btu/h·ft·°F to 400 Btu/h·ft·°F and the outer diameter from 0.5 in to 1.0 in. Plot the length of the tube as functions of pipe conductivity and the outer pipe diameter, and discuss the results.

Step 1 of 10

Given data:

Turbine exit temperature, $T_{w1} = 100°F$

Cooling water temperature, $T_{w2} = 70°F$

Mass flow rate of the steam, $\dot{m}_s = 250 lbs/h$

Thickness of mineral deposit, $L = 0.01 in$

Thermal conductivity of the copper pipe, $k_{copper} = 10 - 400 Btu/h\text{-}ft\text{-}°F$

Heat of vaporization of water, $h_{fg} = 1037 Btu/lbm \ @100°F$

Inner diameter of the pipe, $d_1 = 0.4 in$

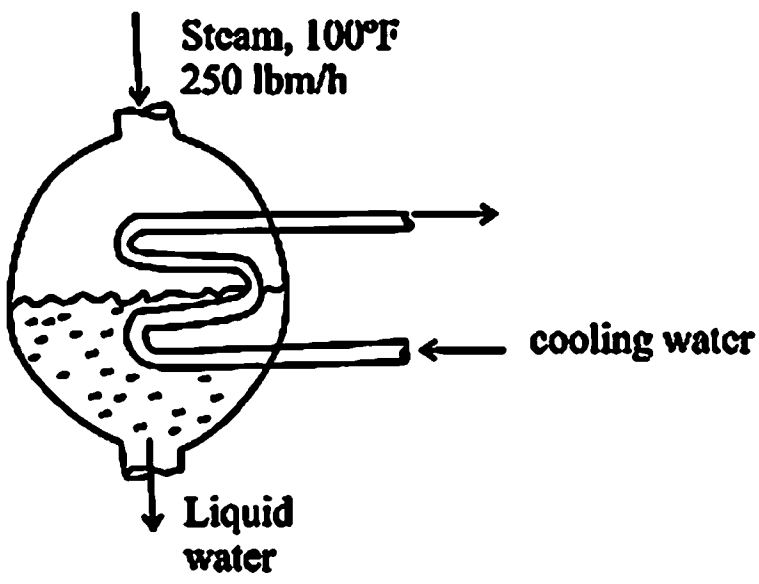
Outer diameter of the pipe, $d_2 = 0.5 - 1 in$

Heat transfer coefficient on the steam side, $h_o = 2400 Btu/h\text{-}ft^2\text{-}°F$

Heat transfer coefficient on the water side, $h_i = 35 Btu/h\text{-}ft^2\text{-}°F$

Step 2 of 10

Let us draw the diagram:



Step 3 of 10

The inner and outer surface areas of the cooling pipe per unit length are calculated as

$$A_i = \pi d_1 L$$

$$= \pi \times \left(\frac{0.4 - 0.02}{12} ft \right) \times (1 ft)$$

$$= 0.0995 \ ft^2$$

$$A_o = \pi d_2 L$$

$$= \pi \times d_2 \times (1 ft)$$

Step 4 of 10

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$

$$= \frac{1}{35 (Btu/h\text{-}ft^2\text{-}°F) \times 0.0995 \ (ft^2)}$$

$$= 0.287 \ h\text{-}°F/Btu$$

$$R_{pipe} = \frac{\ln(r_2 / r_1)}{2\pi k_{copper} L}$$

$$= \frac{\ln\left(\frac{d_2 / 2}{0.2} \right)}{2\pi \times k_{copper} (Btu/h\text{-}ft\text{-}°F) \times 1 (ft)}$$

$$R_o = \frac{1}{h_o A_o}$$

$$= \frac{1}{2400 (Btu/h\text{-}ft^2\text{-}°F) \times \pi \times d_2 \times 1 \ (ft^2)}$$

Step 5 of 10

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_{pipe} + R_{deposit} + R_o$$

$$= 0.287 \ (h\text{-}°F/Btu) + \frac{\ln\left(\frac{d_2 / 2}{0.2} \right)}{2\pi \times k_{copper} \times 1} (h\text{-}°F/Btu) + \frac{1}{2400 \times \pi \times d_2 \times 1} (h\text{-}°F/Btu)$$

The steady rate of heat rate per unit length of the tube can be determined as

$$\dot{Q} = \frac{(T_{w,1} - T_{w,2})}{R_{total}}$$

$$= \frac{(100 - 70)°F}{R_{total} \ (h\text{-}°F/Btu)}$$

Step 6 of 10

The total rate of heat transfer required for condensing the steam at a rate of 250 lbm/h and the length of the tube required is determined to be

$$\dot{Q}_{total} = \dot{m}_s h_{fg}$$

$$= 250 \ (lbm/h) \times 1037 (Btu/lbm)$$

$$= 259250 \ Btu/h$$

Step 7 of 10

$$\text{Tube length} = \frac{\dot{Q}_{total}}{\dot{Q}}$$

Step 7 of 10

$$\text{Tube length} = \frac{\dot{Q}_{total}}{\dot{Q}}$$

$$= \frac{259250(\text{Btu/h})}{\dot{Q}(\text{Btu/h})}$$

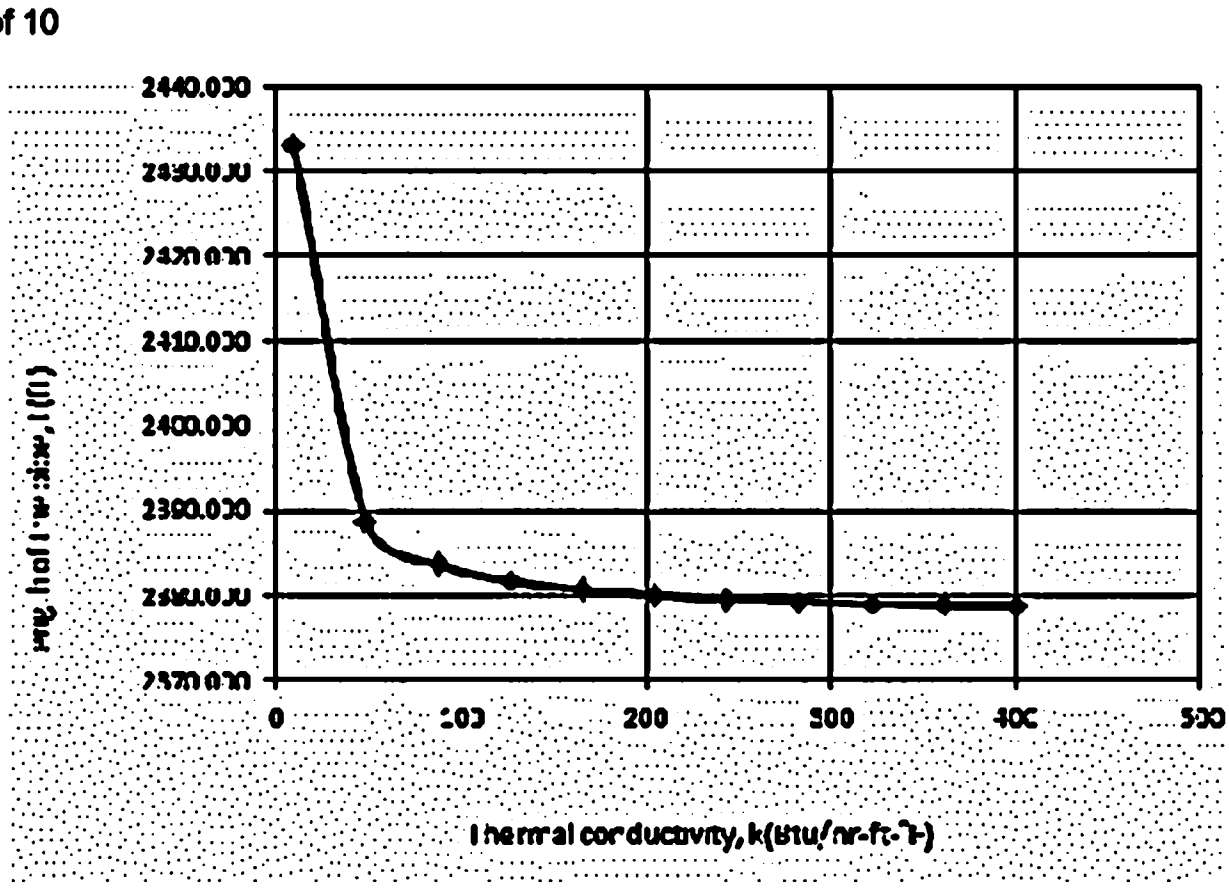
The length of the pipe calculated by solving the above equations using the Excel spread sheet by varying the thermal conductivity of the pipe material form $10-400$Btu/h-ft-°F and the pipe outer diameter from $0.5-1$in .

Step 8 of 10

Length of pipe required by Variation of Pipe thermal conductivity:

k (Btu/hr-ft-°F)	L (ft)
10	2558.4
49	2514
88	2509
127	2507
166	2506
205	2505.3
244	2504.9
283	2504.6
322	2504.4
361	2504.2
400	2504

Step 9 of 10



Step 10 of 10

Length of pipe required by Variation of Pipe outer diameter:

Outer Diameter (in)	L (ft)
0.5	2383.7
0.55	2381.8
0.6	2380.3
0.65	2379
0.7	2378
0.75	2377.1
0.8	2376.3
0.85	2375.7
0.9	2375.1
0.95	2374.7
1	2374.3

Question: 17.080P

A 2.2-mm-diameter and 14-m-long electric wire is tightly wrapped with a 1-mm-thick plastic cover whose thermal conductivity is k = 0.15 W/m·K. Electrical measurements indicate that a current of 13 A passes through the wire and there is a voltage drop of 8 V along the wire. If the insulated wire is exposed to a medium at T_∞ = 30°C with a heat transfer coefficient of h = 24 W/m2·K, determine the temperature at the interface of the wire and the plastic cover in steady operation. Also determine if doubling the thickness of the plastic cover will increase or decrease this interface temperature.

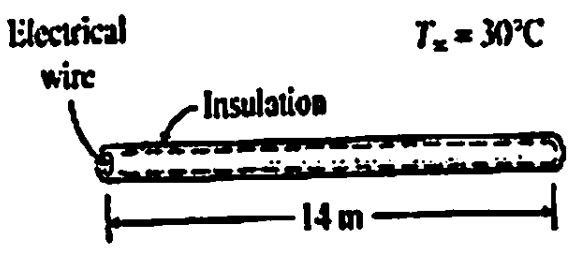


FIGURE P17–80

Step 1 of 8

Given data:

Length of the wire, $L = 14\,\text{m}$

Thermal conductivity of the plastic, $k_{plastic} = 0.15\,\text{W/m-K}$

Diameter of the wire, $d_1 = 2.2\,\text{mm}$

Thickness of the insulation, $t = 1\,\text{mm}$

Electrical current, $I = 13\,\text{A}$

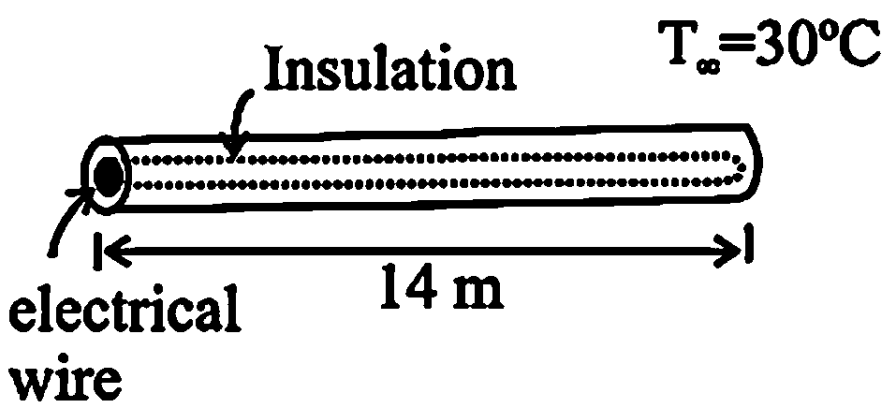
Voltage drop, $V = 8\,\text{V}$

Temperature of the environment, $T_\infty = 30°\text{C}$

Heat transfer coefficient of the exposed medium, $h_0 = 24\,\text{W/m}^2\text{-K}$

Step 2 of 8

Schematic representation of the given system,



Step 3 of 8

Using the Ohms Law, the heat generated inside the pipe,

$$\dot{Q} = IV$$
$$= 13 \times 8$$
$$= 104\,\text{W}$$

Outer surface area of the insulate pipe,

$$A_o = \pi d_2 L$$
$$= \pi \times 0.0042 \times 14$$
$$= 0.1847\,\text{m}^2$$

Step 4 of 8

The individual thermal resistances,

$$R_{plastic} = \frac{\ln(r_2/r_1)}{2\pi k_{plastic} L}$$
$$= \frac{\ln(2.1/1.1)}{2\pi \times 0.15 \times 14}$$
$$= 0.049\,\text{K/W}$$

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{24 \times 0.1847}$$
$$= 0.22559\,\text{K/W}$$

Step 5 of 8

All the resistances are in series, and the total resistance,

$$R_{total} = R_{plastic} + R_o$$
$$= 0.049 + 0.22559$$
$$= 0.2746\,\text{K/W}$$

The steady rate of heat loss from the hot water pipe,

$$\dot{Q} = \frac{(T_{int} - T_\infty)}{R_{total}}$$
$$= \frac{T_{int} - 30}{0.2746}$$

The temperature at the interface of the wire,

$$T_{int} = \dot{Q} R_{total} + T_\infty$$
$$= 104 \times 0.2746 + 30$$
$$= 58.56\,°\text{C}$$

Therefore the interface temperature of the wire, $\boxed{T_{int} = 58.56\,°\text{C}}$

Step 6 of 8

By doubling the thickness of the plastic cover the interface temperature:

Since outer diameter changed, the outer surface area,

$$A_o = \pi d_2 L$$
$$= \pi \times 0.0062 \times 14$$
$$= 0.27269\,\text{m}^2$$

The individual thermal resistances,

$$R_{plastic} = \frac{\ln(r_2/r_1)}{2\pi k_{plastic} L}$$
$$= \frac{\ln(3.1/1.1)}{2\pi \times 0.15 \times 14}$$
$$= 0.078523\,\text{K/W}$$

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{24 \times 0.27269}$$
$$= 0.152798\,\text{K/W}$$

Step 7 of 8

All the resistances are in series, and the total resistance,

$$R_{total} = R_{plastic} + R_o$$
$$= 0.078523 + 0.152798$$
$$= 0.23132\,\text{K/W}$$

The steady rate of heat loss from the hot water pipe,

$$\dot{Q} = \frac{(T_{int} - T_\infty)}{R_{total}}$$
$$= \frac{T_{int} - 30}{0.23132}$$

Step 8 of 8

The temperature at the interface of the wire,

$$T_{int} = \dot{Q} R_{total} + T_\infty$$
$$= 104 \times 0.2313 + 30$$
$$= 54\,°\text{C}$$

Therefore the interface temperature of the wire, $\boxed{T_{int} = 54\,°\text{C}}$

Hence, the interface temperature is decreased due to increase in thickness of the plastic cover.

Question: 17.081P

Ina pharmaceutical plant, a copper pipe (*kc*= 400 W/m · K) with inner diameter of 20 mm and wall thicknessof 2.5 mmis used for carrying liquid oxygen to a storage tank. The liquid oxygen flowing in the pipe has an average temperature of-200°C and a convection heat transfer coefficient of 120 W/m2·K.The condition surrounding the pipe has an ambient air temperature of 20°C and a combined heat transfer coefficient of 20 W/m2-K. Ifthe dew point is 10° C, determine the thickness of the insulation (*ki* -0.05 W/ m·K)around the copper pipe to avoid condensation on the outer surface. Assumethermal contact resistance is negligible.

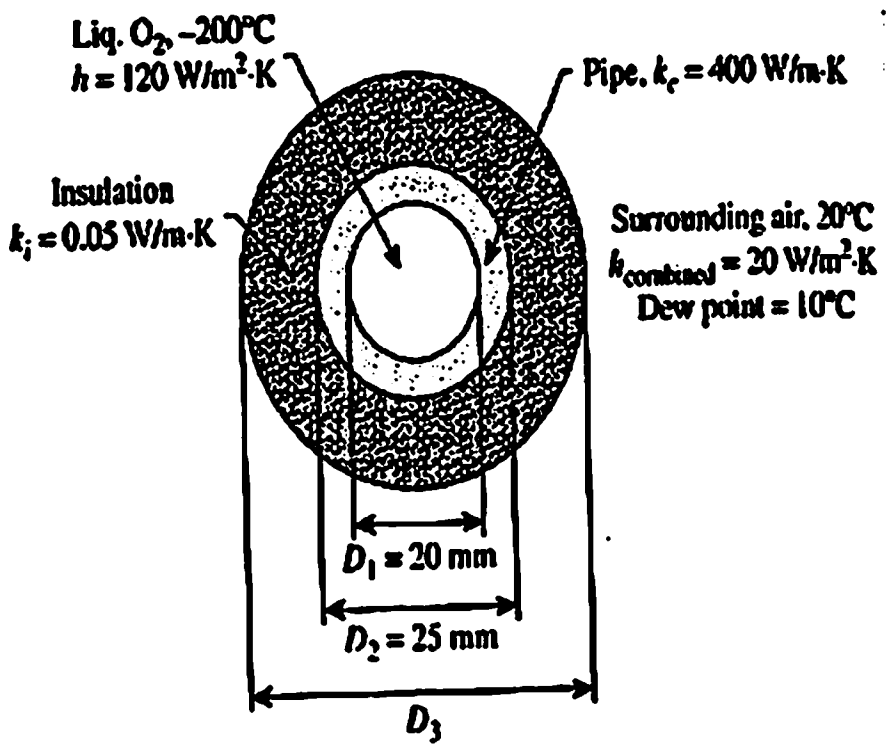


FIGURE P17-81

Step 1 of 3

Neglect the thermal contact resistance.

Calculate the overall rate of heat transfer through the pipe per unit length and insulation using the relation,

$$\begin{aligned}\dot{Q}_r &= U\Delta T \\ &= UA(T_\infty - T_i)\end{aligned}$$

Here, combined heat transfer coefficient is U , total area is A , surrounding temperature is T_∞ , and inside temperature is T_i .

Write the formula for total surface area per unit length of the system as shown.

$$A = \pi D_3 L$$

Substitute $20\ W/m^2 \cdot K$ for U , πD_3 for A , $(20+273)\ K$ for T_∞ , $(-200+273)\ K$ for T_i .

$$\begin{aligned}\dot{Q}_r &= 20 \times \pi D_3 \times [(20+273)-(-200+273)] \\ &= 20 \times 220 \times \pi D_3 \\ &= 4400\pi D_3\ W\end{aligned}$$

Step 2 of 3

Mode of heat transfer takes place as convection from surroundings to insulation, conduction through insulation, conduction through pipe and convection from fluid.

Overall heat transfer rate is equal to heat transfer due to convection from surroundings to insulation, conduction through insulation, conduction through pipe and convection from fluid.

Calculate the thickness of insulation to avoid condensation as shown.

$$\dot{Q}_r = \frac{T_\infty - T_i}{\frac{1}{h(\pi D_1)} + \frac{\ln(D_2/D_1)}{\pi L_1 k_1} + \frac{\ln(D_3/D_2)}{\pi L_2 k_2}} \quad \text{..... (1)}$$

Here, conduction through insulation is R_{cond2} , and conduction through pipe is R_{cond3} , and convection from fluid is R_{conv1} , thickness of pipe is L_1 , insulation thickness is L_2 , conduction coefficient of pipe is $k1$, conduction coefficient of insulation is k_2 , overall diameter is D_3 , diameter of pipe is D_1 , and outer diameter of the pipe is D_2 .

Step 3 of 3

Calculate the thickness of insulation to avoid condensation as shown.

Substitute $4400\pi D_3\ W$ for $\dot{Q}_r$, $120\ W/m^2 \cdot K$ for h , $(10+273)\ K$ for T_∞ , $(-200+273)\ K$ for T_i , $400\ W/m \cdot K$ for k_1 , $0.05\ W/m \cdot K$ for k_2 , $20\times10^{-3}\ m$ for D_1 , $(25\times10^{-3}+L_2)$ for D_3 , and $2.5\times10^{-3}\ m$ for L_1 in equation (1).

$$4400\pi(25\times10^{-3}+L_2)^2 = \frac{(10+273)-(-200+273)}{\left[\frac{1}{120(\pi\times20\times10^{-3}\times2.5\times10^{-3})} + \frac{\ln(25\times10^{-3}/20\times10^{-3})}{\pi\times2.5\times10^{-3}\times400} + \frac{\ln(25\times10^{-3}+L_2/25\times10^{-3})}{\pi L_2\times0.05}\right]}$$

Solve the equation and obtain the value of L_2 .

$$L_2 = 28.73\ mm$$

Therefore, thickness of insulation to prevent condensation is $\boxed{28.73\ mm}$.

Question: 17.082P

What is the critical radius of insulation? Howis it defined for a cylindrical layer?

Step 1 of 1

In a cylindrical pipe or a spherical shell, the additional insulation increases the conduction resistance of insulation, but decreases the convection resistance of the surface because of the increase in the outer surface area. Due to these opposite effects, a critical radius of insulation is defined as the outer radius that provides maximum rate of heat transfer.

For a cylindrical layer, it is defined as

$r_{cr} = k/h$

Where k is the thermal conductivity of insulation and

h is the external convection heat transfer coefficient.

Question: 17.083P

Consider an insulated pipe exposed to the atmosphere. Will the critical radius of insulation be greater on calm days or on windy days? Why?

Step 1 of 1

For a cylindrical pipe, the critical radius of insulation is defined as

$r_{cr} = k/h$.

On windy days, the external convection heat transfer coefficient is greater compared to calm days. Therefore critical radius of insulation will be greater on calm days.

Question: 17.084P

A pipe is insulated to reduce the heat loss fromit. However, measurements indicate that the rate of heat loss has increased instead of decreasing. Can the measurements be right?

Step 1 of 1

Yes, the measurements can be right. If the radius of insulation is less than critical radius of insulation of the pipe, then the rate of heat loss has increased instead of decreasing

Question: 17.085P

Consider a pipe at a constant temperature whose radius is greater than the critical radius of insulation. Someone claims that the rate of heat loss from the pipe has increased when someinsulation is added to the pipe. Is this claim valid?

Step 1 of 1

The rate of heat transfer from the cylinder increases with the addition of insulation and reaches a maximum when radius of the cylinder reaches to critical radius. The rate of heat transfer decreases when radius of the cylinder is greater than critical radius.
So the claim is wrong.

Question: 17.086P

A pipe is insulated such that the outer radius of the insulation is less than the critical radius. Now the insulation is taken off. Will the rate of heat transfer from the pipe increase or decrease for the same pipe surface temperature?

Step 1 of 1

When the outer radius is less than critical radius then rate of heat transfer from the cylinder increases

In the given problem the outer radius is less than critical radius then if the insulation is removed the rate of heat transfer from the pipe decreases

Question: 17.087P

A 0.083-in-diameter electrical wire at 90°F is covered by 0.02-in-thick plastic insulation (k = 0.075 Btu/h·ft·°F). The wire is exposed to a medium at 50°F, with a combined convection and radiation heat transfer coefficient of 2.5 Btu/h·ft2·°F. Determine if the plastic insulation on the wire will increase or decrease heat transfer from the wire.

Step 1 of 3

Given: -

Diameter of electric wire D = 0.083in

Thickness of plastic cover t= 0.02in

Thermal conductivity of plastic insulation $k = 0.075$ Btu/h·ft·°F.

Combined convection and radiation heat transfer coefficient h= 2.5 Btu/h.ft^2.°F

Ambient temperature $T_\infty = 50^0$F

Step 2 of 3

Outer diameter of the electric wire with plastic insulation

$D_0 = D + 2t$

$D_0 = 0.083 + 2\times 0.02$

$D_0 = 0.123$in

$$r_0 = \frac{0.123}{2}$$

$r_0 = 0.0615$in

Step 3 of 3

Critical radius of plastic insulation for a cylindrical body is

$$r_{cr} = \frac{k}{h}$$

$$= \frac{0.075}{2.5}$$

$= 0.03$ ft

$= 0.03\times 12$

$r_{cr} = 0.36$ in

$r_0 < r_{cr}$

Since the outer radius of the wire with insulation is smaller than critical radius of insulation, plastic insulation will increase heat transfer from the wire.

$D_0 = 0.083 + 2\times 0.02$

$D_0 = 0.123$in

0.123

Question: 17.088P

Repeat Prob. 17–87E, assuming a thermal contact resistance of 0.001 h·ft2·°F/Btu at the interface of the wire and the insulation.

Step 1 of 8

Given: -

Diameter of electric wire D = 0.083in

Thickness of plastic cover t= 0.02in

Thermal conductivity of plastic insulation $k = 0.075$ Btu/h·ft·°F.

Combined convection and radiation heat transfer coefficient h= 2.5 Btu/h.ft^2.°F

Ambient temperature $T_\infty = 50^0$F

Diameter of electric wire = 0.083in

Thermal contact resistance $h_c = 0.001$h.ft^2.^{0}F/Btu

Step 2 of 8

Outer diameter of the electric wire with plastic insulation

$D_0 = D + 2t$

$D_0 = 0.083 + 2\times 0.02$

$D_0 = 0.123$in

$$r_0 = \frac{0.123}{2}$$

$r_0 = 0.0615$in

Step 3 of 8

If the heat transfer from the wire is steady since there is no indication of any change with time and the heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction then for without insulation, the total thermal resistance is (per ft length of the wire)

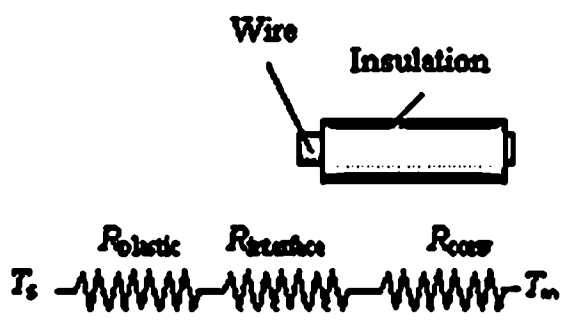
$$R_{tot} = R_{conv} = \frac{1}{h_o A_o}$$

$$= \frac{1}{h_0 \times \pi D \times L}$$

$$= \frac{1}{(2.5)[\pi(0.083/12)(1)]}$$

$R_{tot} = R_{conv} = 18.4$ h.°F/Btu

Step 4 of 8



Step 5 of 8

Thermal resistance due to convection from outer surface

$$R_{conv} = \frac{1}{h_o A_o}$$

$$= \frac{1}{h_o \times \pi D_0 L}$$

$$= \frac{1}{(2.5)[\pi(0.123/12)(1)]}$$

$R_{conv} = 12.42$ h.°F/Btu

Step 6 of 8

Thermal resistance due to conduction in electric wire and the insulation

$$R_{plastic} = \frac{\ln(r_2/r_1)}{2\pi k L}$$

$$= \frac{\ln(0.123/0.083)}{2\pi(0.075)(1)}$$

$R_{plastic} = 0.835$ h.°F/Btu

Step 7 of 8

Thermal resistance at the interface of the wire and insulation

$$R_{interface} = \frac{h_c}{A_c}$$

$$= \frac{h_c}{\pi D L}$$

$$= \frac{0.001}{[\pi(0.083/12)(1)]}$$

$R_{interface} = 0.046$ h.°F/Btu

Step 8 of 8

Total thermal resistance due to insulation

$R_{total} = R_{conv} + R_{plastic} + R_{interface}$

$= 12.42 + 0.835 + 0.046$

$R_{total} = 13.30$ h.°F/Btu

Since the total thermal resistance decreases with insulation, plastic insulation will increase heat transfer from the wire. The thermal contact resistance appears to have negligible effect in this case.

Question: 17.089P

A 4-mm-diameter spherical ball at 50°C is covered by a 1-mm-thick plastic insulation (k = 0.13 W/m·K). The ball is exposed to a medium at 15°C. with a combined convection and radiation heat transfer coefficient of 20 W/m2·K. Determine if the plastic insulation on the ball will help or hurt heat transfer from the ball.

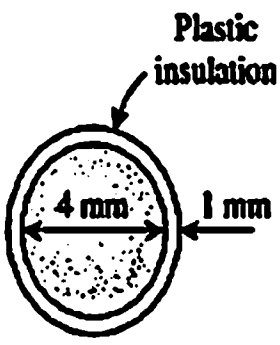


FIGURE P17–89

Step 1 of 4

Given data:

Diameter of the spherical ball, $d = 4\,\text{mm}$

Thermal conductivity of the plastic insulation, $k_{insulation} = 0.13\,\text{W/m-K}$

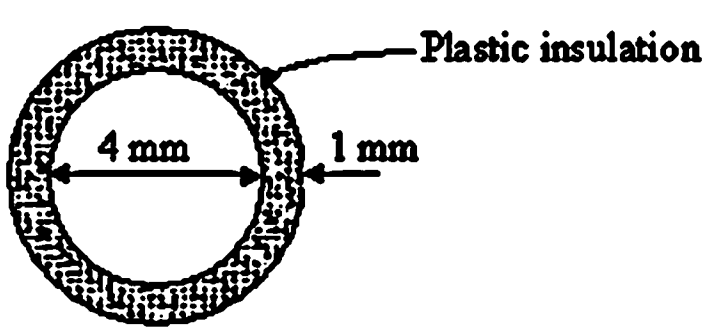
Thickness of the insulation, $t = 1\text{mm}$

Temperature of the ball, $T_{ball} = 50°\text{C}$

Temperature of the environment, $T_\infty = 15°\text{C}$

Combined convection and radiation heat transfer coefficient $h_0 = 20\,\text{W/m}^2\text{-K}$

Step 2 of 4



Step 3 of 4

Outer diameter of the spherical shell is

$$D_o = D_i + 2t$$
$$D_o = 4 + 2\times 1$$
$$D_o = 6\,\text{mm}$$

$$r_0 = \frac{D_0}{2}$$
$$r_0 = \frac{6}{2}$$
$$r_0 = 3\text{mm}$$

The critical radius of insulation for a sphere can be determined as

$$r_{cr,sphere} = \frac{2k}{h}$$
$$= \frac{2\times 0.13(\text{W/m-K})}{20(\text{W/m}^2\text{-K})}$$
$$= 0.013\text{m}$$
$$= 13\text{mm}$$

Step 4 of 4

Since the critical radius of insulation depends on the thermal conductivity of the insulation $k_{insulation}$ and the external convection heat transfer coefficient h. The rate of heat transfer from the sphere increases with the addition of insulation for $r_2 < r_{cr,sphere}$, reaches maximum when $r_2 = r_{cr,sphere}$, and starts to decrease for $r_2 > r_{cr,sphere}$. Since in the current problem the outer radius of the insulation $r_2(3\text{mm}) < r_{cr,sphere}(13\text{mm})$. Hence the rate of heat transfer from the sphere increases with the addition of insulation.

Question: 17.090P

Reconsider Prob. 17-89. Using EES (or other) software, plot the rate of heat transfer from the ball as a function of the plastic insulation thickness in the range of 0.5 mm to 20 mm. Discuss the results.

Step 1 of 7

Given data:

Diameter of the spherical ball, $d_1 = 4\,\text{mm}$

Thermal conductivity of the plastic insulation, $k_{insulation} = 0.13\,\text{W/m}\cdot\text{K}$

Thickness of the insulation, $t = 0.5\,\text{to}\,20\,\text{mm}$

Temperature of the ball, $T_{ball} = 50°\text{C}$

Temperature of the environment, $T_\infty = 15°\text{C}$

Combined convection and radiation heat transfer coefficient medium, $h_0 = 20\,\text{W/m}^2\text{-K}$

Step 2 of 7

Schematic representation of the given system,



Step 3 of 7

The outer surface area of the sphere,

$$A_2 = \pi d_2^2$$
$$= \pi \times (0.004 + 2t)^2$$

The individual resistance of the given system:

Conduction resistance,

$$R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2}$$
$$= \frac{(0.002 + t) - 0.002}{4\pi \times 0.13 \times 0.002 \times (0.002 + t)}$$
$$= \frac{t}{0.00326 \times (0.002 + t)}$$

Convection resistance from the surface to the environment,

$$R_{conv} = \frac{1}{h_0 A_2}$$
$$= \frac{1}{20 \times \pi \times (0.004 + 2t)^2}$$

Step 4 of 7

The total thermal resistance of the given system,

$$R_{total} = R_{sphere} + R_{conv}$$
$$= \frac{t}{0.00326 \times (0.002 + t)} + \frac{1}{20 \times \pi \times (0.004 + 2t)^2}$$

Therefore, the rate of heat transfer from the sphere

$$\dot{Q} = \frac{T_{ball} - T_\infty}{R_{total}}$$
$$= \frac{(50 - 15)}{\frac{t}{0.00326 \times (0.002 + t)} + \frac{1}{20 \times \pi \times (0.004 + 2t)^2}}$$
$$= \frac{35}{\frac{t}{0.00326 \times (0.002 + t)} + \frac{1}{62.83 \times (0.004 + 2t)^2}}$$

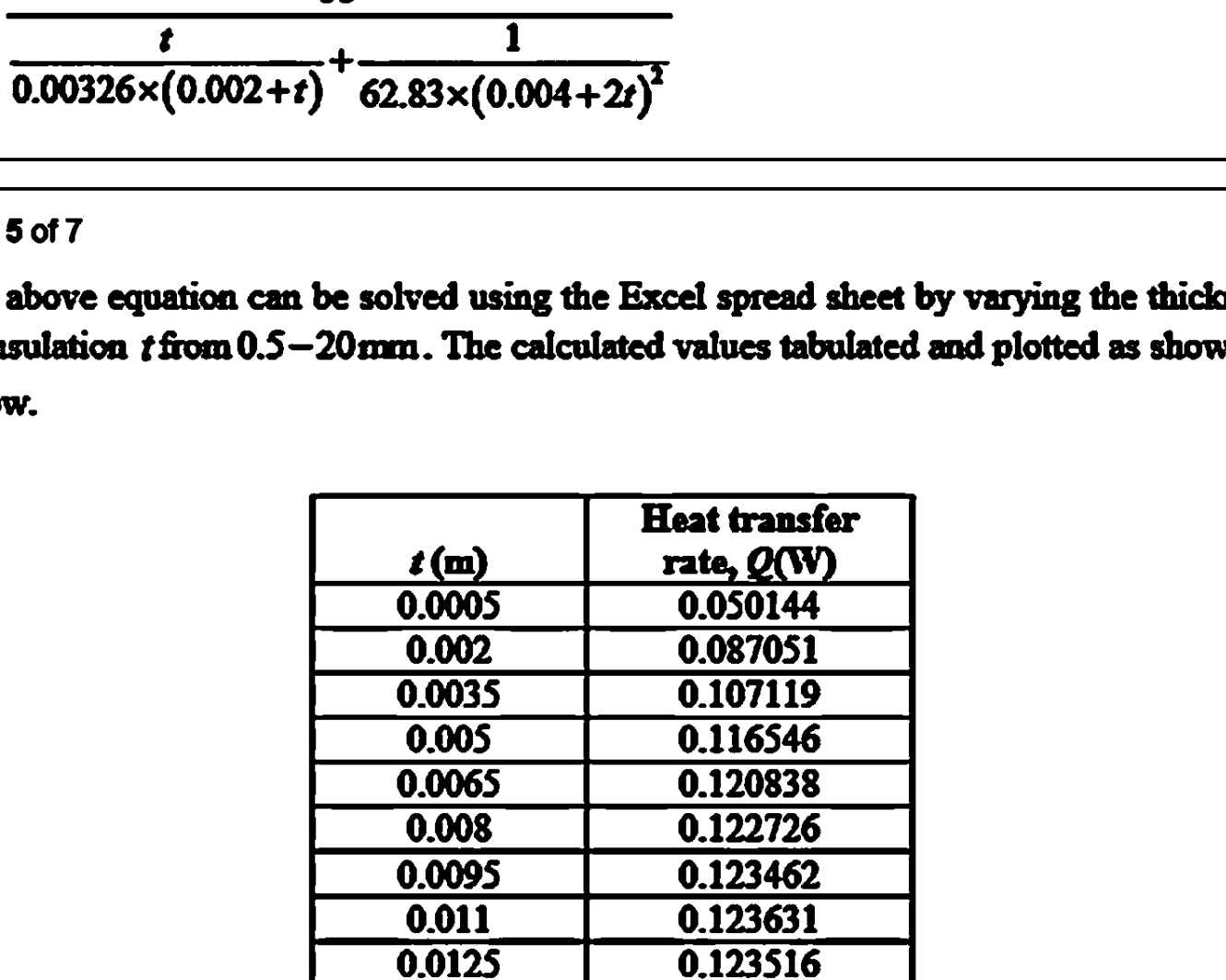
Step 5 of 7

The above equation can be solved using the Excel spread sheet by varying the thickness of insulation t from $0.5 - 20\,\text{mm}$. The calculated values tabulated and plotted as shown in below.

t (m)	Heat transfer rate, Q(W)
0.0005	0.050144
0.002	0.087051
0.0035	0.107119
0.005	0.116546
0.0065	0.120838
0.008	0.122726
0.0095	0.123462
0.011	0.123631
0.0125	0.123516
0.014	0.123262
0.0155	0.122943
0.017	0.1226
0.0185	0.122254
0.02	0.121916

Step 6 of 7

Graphical representation for heat transfer versus the thickness of the insulation,



Step 7 of 7

The critical radius of insulation for a sphere,

$$r_{cr,sphere} = \frac{2k}{h}$$
$$= \frac{2 \times 0.13}{20}$$
$$= 0.013\,\text{m}$$
$$= 13\,\text{mm}$$

Note that the critical radius of insulation depends on the thermal conductivity of the insulation $k_{insulation}$ and the external convection heat transfer coefficient h. From the plot it can be observed that the rate of heat transfer from the sphere increases with the addition of insulation for $r_2 < r_{cr,sphere}$, reaches maximum when $r_2 = r_{cr,sphere} = 13\,\text{mm}$, and starts to decrease for $r_2 > r_{cr,sphere}$. Thus, insulating the pipe may actually increasing the rate of heat transfer from the pipe instead of decreasing it when $r_2 < r_{cr,sphere}$.

Question: 17.091P

Hot air is to be cooled as it is forced to flow through the tubes exposed to atmospheric air. Fins are to be added in order to enhance heat transfer. Would you recommend attaching the fins inside or outside the tubes? Why? When would you recommend attaching fins both inside and outside the tubes?

Step 1 of 1

Fins should be attached to the outside since the heat transfer coefficient inside the tube will be higher due to forced convection. Fins should be added to both sides of the tubes when the convection coefficients at the inner and outer surfaces are comparable in magnitude.

Question: 17.092P

What is the reason for the widespread use of fins on surfaces?

Step 1 of 1

Fins increase the rate of heat transfer from the surface by increasing the heat transfer surface area.

Question: 17.093P

What is the difference between the fin effectiveness and the fin efficiency?

Step 1 of 1

The fin efficiency is defined as the ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature, and its value is between 0 and 1. Fin effectiveness is defined as the ratio of heat transfer rate from a finned surface to the heat transfer rate from the same surface if there were no fins, and its value is expected to be greater than 1.

Question: 17.094P

The fins attached to a surface are determined to have an effectiveness of 0.9. Do you think the rate of heat transfer from the surface has increased or decreased as a result of the addition of these fins?

Step 1 of 1

Fin effectiveness value is 0.9 (smaller than 1) this indicates that the heat transferred from the surface is decreased and hence the fins here were acting as thermal insulation.

Question: 17.095P

Explain how the fins enhance heat transfer from a surface. Also, explain how the addition of fins may actually decrease heat transfer from a surface.

Step 1 of 1

Fins enhance heat transfer from a surface by increasing heat transfer surface area for convection heat transfer. However, adding too many fins on a surface can suffocate the fluid and retard convection, and thus it may cause the overall heat transfer coefficient and heat transfer to decrease.

Question: 17.096P

How does the overall effectiveness of a finned surface differ from the effectiveness of a single fin?

Step 1 of 1

Effectiveness of a single fin is the ratio of the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the fin base area. The overall effectiveness of a finned surface is defined as the ratio of the total heat transfer from the finned surface to the heat transfer from the same surface if there were no fins.

Question: 17.097P

Hot water is to be cooled as it flows through the tubes exposed to atmospheric air. Fins are to be attached in order to enhance heat transfer. Would you recommend attaching the fins inside or outside the tubes? Why?

Step 1 of 1

Fins should be attached on the outside (surface exposed to air) since the convection heat transfer coefficient is lower on the air side than it is on the water side.

Question: 17.098P

Consider two finned surfaces that are identical except that the fins on the first surface are formed by casting or extrusion, whereas they are attached to the second surface afterwards by welding or tight fitting. For which case do you think the fins will provide greater enhancement in heat transfer? Explain.

Step 1 of 1

Welding or tight fitting introduces thermal contact resistance at the interface, and thus retards heat transfer. Therefore, the fins formed by casting or extrusion will provide greater enhancement in heat transfer.

Question: 17.099P

The heat transfer surface area of a fin is equal to the sum of all surfaces of the fin exposed to the surrounding medium, including the surface area of the fin tip. Under what conditions can we neglect heat transfer from the fin tip?

Step 1 of 1

If the fin is too long, the temperature of the fin tip will approach the surrounding temperature and we can neglect heat transfer from the fin tip. Also, if the surface area of the fin tip is very small compared to the total surface area of the fin, heat transfer from the tip can again be neglected.

Question: 17.100P

Does the (*a*) efficiency and (*b*) effectiveness of a fin increase or decrease as the fin length is increased?

Step 1 of 2

(a)
Increasing the length of a fin decreases its efficiency because for longer fins it is difficult to maintain entire surface at base temperature.

Step 2 of 2

(b)
Increasing the length of a fin increases its effectiveness because it enhances heat transfer by increasing the surface area.

Question: 17.101P

Two pin fins are identical, except that the diameter of one of them is twice the diameter of the other. For which fin is the (*a*) fin effectiveness and (*b*) fin efficiency higher? Explain.

Step 1 of 2

(a)

Fin with large diameter has more efficiency because more heat can reach the fin tip of a large diameter fin when compared to that of a fin with small diameter.

Step 2 of 2

(b)

Fin with smaller diameter is of higher effectiveness. This is due to the reason that for small diameter fins the surface area available per unit volume is relatively larger than that of the fins of large diameter.

Question: 17.102P

Two plate fins of constant rectangular cross section are identical, except that the thickness of one of them is twice the thickness of the other. For which fin is the (*a*) fin effectiveness and (*b*) fin efficiency higher? Explain.

Step 1 of 2

(a)

The thicker fin will have higher efficiency, because for thicker fin the tendency to maintain base temperature is more when compared to that of the thinner fin.

Step 2 of 2

(b)

The thinner fin will have higher effectiveness, because the thinner fin has relatively more surface area per unit volume when compared to that of the thicker fin.

Question: 17.103P

Two finned surfaces are identical, except that the convection heat transfer coefficient of one of them is twice that of the other. For which finned surface is the (*a*) fin effectiveness and (*b*) fin efficiency higher? Explain.

Step 1 of 2

(a)

The fin with the lower convective heat transfer coefficient will have the higher effectiveness because the effectiveness of fin is inversely proportional to the convective heat transfer coefficient.

Step 2 of 2

(b)

The fin with the lower convective heat transfer coefficient will also have the higher efficiency because the fin efficiency is directly proportional to the effectiveness of fin

Question: 17.104P

The case-to-ambient thermal resistance of a power transistor that has a maximum power rating of 15 W is given to be 25°C/W. If the case temperature of the transistor is not to exceed 80°C, determine the power at which this transistor can be operated safely in an environment at 35°C.

Step 1 of 3

Given data:

Maximum power rating of the transistor, $P = 15\text{W}$

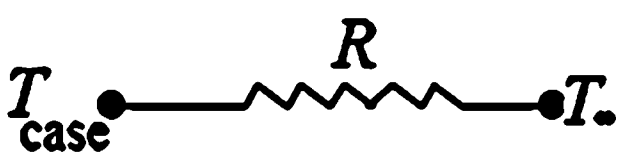
Case-to-ambient thermal resistance of a power transistor, $R = 25°\text{C/W}$

Case temperature of the transistor, $T_{case} = 80°\text{C}$

Ambient temperature, $T_\infty = 35°\text{C}$

Step 2 of 3

Assume temperature of the transistor is uniform throughout the surface and steady state conditions exist.



Step 3 of 3

The power at which this transistor can be operated safely can be determined as

$$\dot{Q} = \frac{(T_{case} - T_\infty)}{R}$$

$$= \frac{(80-35)(°\text{C})}{25(°\text{C/W})}$$

$$= 1.8\,\text{W}$$

$$\boxed{\dot{Q} = 1.8\,\text{W}}$$

Question: 17.105P

A 4-mm-diameter and (0-cm-long aluminum fin (k =237 W/m·K) is attached to a surface. If the heat transfer coefficient is 12 W/m2·K, determine the percent error in the rate of heat transfer from the fin when the infinitely long fin assumption is used instead of the adiabatic fin tip assumption.

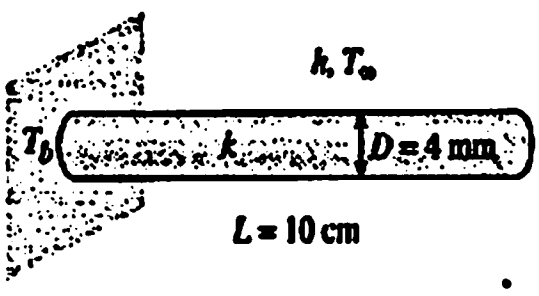


FIGURE P17–105

Step 1 of 2

The rate of heat transfer for a fin with insulated tip is as shown below:

$$\dot{Q}_{long} = \sqrt{hpkA_C}(T_b - T_\infty)$$

Here, h is heat transfer coefficient; k is thermal conductivity of the fin, p is perimeter of the fin, A_c is cross sectional area of the fin, T_b is base temperature of the fin and T_∞ is ambient air temperature.

The rate of heat transfer for a fin of infinitely long fin is as shown below:

$$\dot{Q}_{adiabatic} = \sqrt{hpkA_C}(T_b - T_\infty)\tanh(mL)$$

Calculate the value of m is as shown below,

$$m = \sqrt{\frac{hp}{kA_c}}$$
$$= \sqrt{\frac{h\pi D}{k\frac{\pi D^2}{4}}}$$
$$= \sqrt{\frac{4h}{kD}}$$

Substitute $12\,\mathrm{W/m^2 \cdot K}$ for h, $237\,\mathrm{W/m \cdot K}$ for k and $0.004\,\mathrm{m}$ for D

$$m = \sqrt{\frac{4\times 12}{237\times 0.004}}$$
$$= 7.1156\,\mathrm{m^{-1}}$$

Step 2 of 2

Find the percent error of the heat transfer as shown below,

$$\%error = \frac{\dot{Q}_{long} - \dot{Q}_{adiabatic}}{\dot{Q}_{adiabatic}} \times 100$$
$$= \frac{\sqrt{hpkA_C}(T_b - T_\infty) - \sqrt{hpkA_C}(T_b - T_\infty)\tanh(mL)}{\sqrt{hpkA_C}(T_b - T_\infty)\tanh(mL)} \times 100$$
$$= \left[\frac{1}{\tanh(mL)} - 1\right] \times 100$$

Here, L is length of the fin

Substitute $10\,\mathrm{cm} = 0.1\,\mathrm{m}$ for L

$$\%error = \left[\frac{1}{\tanh(7.116\times 0.1)} - 1\right] \times 100$$
$$= 0.6348 \times 100$$
$$= 63.48\%$$

Therefore, the percent error in the rate of heat transfers when the infinitely long fin assumption is used instead of the adiabatic fin tip assumption **63.48%**

Question: 17.106P

Consider a very long rectangular fin attached to a flat surface such that the temperature at the end of the fin is essentially that of the surrounding air, i.e. 20°C Its width is 5.0 cm; thickness is 1.0 mm; thermal conductivity is 200 W/m· K; and base temperature is 40°C The heat transfer coefficient is 20 W/m2·K. Estimate the fin temperature at a distance of 5.0 cm from the base and the rate of heat loss from the entire fin.

Step 1 of 3

The temperature distribution equation for long rectangular fin is

$$\frac{T-T_\infty}{T_b-T_\infty}=e^{-mx}$$

Here, T is the temperature of fin at any point, T_b is base temperature, T_∞ is ambient temperature and x is length of the fin.

Calculate the value of m by using the formula

$$m=\sqrt{\frac{hp}{kA_C}}$$
$$=\sqrt{\frac{h\times(2w+2t)}{k\times wt}}$$

Here, h is heat transfer coefficient, p is perimeter of the rectangular fin, k is thermal conductivity of aluminum fin, A_c is cross sectional area of the fin, w is width of the fin and t is thickness of the fin.

Substitute $20\,\mathrm{W/m^2\cdot K}$ for h, $0.05\,\mathrm{m}$ for w, $0.001\,\mathrm{m}$ for t

$$m=\sqrt{\frac{20\times(2\times0.05+2\times0.001)}{200\times(0.05\times0.001)}}$$
$$=\sqrt{\frac{20\times0.102}{200\times0.00005}}$$
$$=14.28\,\mathrm{m^{-1}}$$

Step 2 of 3

Now find the temperature at a distance x from the base by using equation (1)

$$\frac{T-T_\infty}{T_b-T_\infty}=e^{-mx}$$

Substitute $20°\mathrm{C}$ for T_∞, $40°\mathrm{C}$ for T_b and $0.05\,\mathrm{m}$ for x

$$\frac{T-20}{40-20}=e^{-(14.3)(0.05)}$$
$$=\sqrt{\frac{h\times(2w+2t)}{k\times wt}}$$

Here, h is heat transfer coefficient, p is perimeter of the rectangular fin, k is thermal conductivity of aluminum fin, A_c is cross sectional area of the fin, w is width of the fin and t is thickness of the fin.

Substitute $20\,\mathrm{W/m^2\cdot K}$ for h, $0.05\,\mathrm{m}$ for w, $0.001\,\mathrm{m}$ for t

$$m=\sqrt{\frac{20\times(2\times0.05+2\times0.001)}{200\times(0.05\times0.001)}}$$
$$=\sqrt{\frac{20\times0.102}{200\times0.00005}}$$
$$=14.28\,\mathrm{m^{-1}}$$

Step 2 of 3

Now find the temperature at a distance x from the base by using equation (1)

Question: 17.107P

A DC motor delivers mechanical power to a rotating stainless steel shaft (k = 15.1 W/m·K) with a length of 25 cm and a diameter of 25 mm. In a surrounding with ambient air temperature of 20°C and convection heat transfer coefficient of 25 W/m2·K, the surface area of the motor housing that is exposed to the ambient air is 0.075 m2. The motor uses 300 W of electrical power and delivers 55% of it as mechanical power to rotate the stainless steel shaft. If the tip of the stainless steel shaft has a temperature of 22°C, determine the surface temperature of the motor housing. Assume the base temperature of the shaft is equal to the surface temperature of the motor housing. *Answer:*87.7°C.

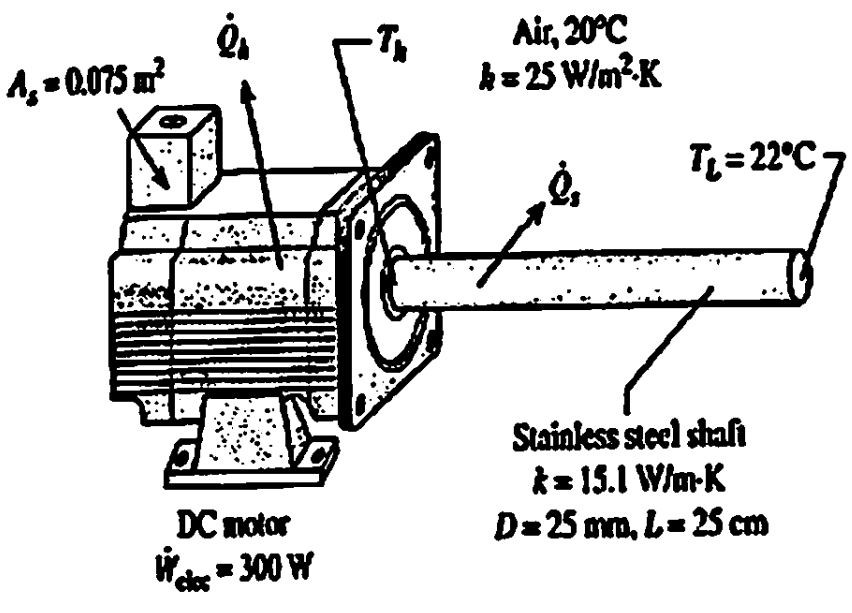


FIGURE P17–107

Step 1 of 5

Given data:

Diameter of the shaft, $D = 25\,\text{mm}$

Length of the shaft, $L = 25\,\text{cm}$

Thermal conductivity of the stainless steel shaft, $k = 15.1\,\text{W/m.K}$

Ambient air temperature, $T_\infty = 20\,^\circ\text{C}$

Convection heat transfer coefficient of ambient air, $h = 25\,\text{W/m}^2\text{.K}$

Surface area of the motor, $A_s = 0.075\,\text{m}^2$

Electrical power of the motor, $P = 300\,\text{W}$

Percentage of mechanical power delivered by the motor, $\eta = 55\%$

Temperature at the tip of the stainless shaft, $T_L = 22\,^\circ\text{C}$

Step 2 of 5

Since the motor transmitting 55 % of the power to rotate the shaft, remaining 45 % power will be converted into heat energy and that will be lost through the motor surface and the shaft.

Therefore the heat generated during transmission can be expressed as

$$\dot{Q}_{gen} = (1-\eta)P$$
$$= (1-0.55)\times 300(\text{W})$$
$$= 135\,\text{W}$$

The total heat generated in the motor will be lost to the surroundings can be expressed as

$$\dot{Q}_{gen} = \dot{Q}_b + \dot{Q}_s \qquad \ldots\ldots (1)$$

Where the heat transfer from the motor surface is $\dot{Q}_b$ with the constant surface temperature of T_b and it can be expressed as

$$\dot{Q}_b = hA_s(T_b - T_\infty)$$
$$= 25(\text{W/m}^2\text{-K})\times 0.075(\text{m}^2)\times(T_b - T_\infty)$$
$$= 1.875(T_b - T_\infty)$$

Step 3 of 5

The heat transfer from the shaft $\dot{Q}_s$ with base temperature same as the motor surface temperature and considering the shaft as an extended fin by fixing the fin tip temperature at $T_L = 22\,^\circ\text{C}$ can be expressed as

$$\dot{Q}_s = \sqrt{hpkA_c}(T_b - T_\infty)\frac{\cosh mL - \left[(T_L - T_\infty)/(T_b - T_\infty)\right]}{\sinh mL}$$

Where

$$\sqrt{hpkA_c} = \sqrt{h\times\pi D\times k\times\frac{\pi}{4}D^2}$$
$$= \sqrt{25(\text{W/m}^2\text{-K})\times\pi\times 0.025(\text{m})\times 15.1(\text{W/m-K})\times\frac{\pi}{4}\times 0.025^2(\text{m}^2)}$$
$$= 0.12064$$

$$mL = \sqrt{\frac{hp}{kA_c}}L$$
$$= \sqrt{\frac{h\times\pi D}{k\times\frac{\pi}{4}D^2}}L$$
$$= \sqrt{\frac{25(\text{W/m}^2\text{-K})\times\pi\times 0.025(\text{m})}{15.1(\text{W/m-K})\times\frac{\pi}{4}\times 0.025^2(\text{m}^2)}}0.25(\text{m})$$
$$= 4.06894$$

Step 4 of 5

Then substituting the known values in heat transfer from the shaft

$$\dot{Q}_s = \sqrt{hpkA_c}(T_b - T_\infty)\frac{\cosh mL - \left[(T_L - T_\infty)/(T_b - T_\infty)\right]}{\sinh mL}$$
$$= 0.12064\times(T_b - T_\infty)\frac{\cosh(4.06894) - \left[(22-20)/(T_b - T_\infty)\right]}{\sinh(4.06894)}$$
$$= 0.12071\times(T_b - T_\infty) - 0.008252$$

Step 5 of 5

Therefore, the surface temperature of the motor can be determined from equation (1)

$$\dot{Q}_{gen} = \dot{Q}_b + \dot{Q}_s$$
$$135 = \dot{Q}_b + \dot{Q}_s$$
$$= 1.875(T_b - T_\infty) + 0.12071\times(T_b - T_\infty) - 0.008252$$
$$= 1.875\times(T_b - 20) + 0.12071\times(T_b - 20) - 0.008252$$
$$135 = (T_b - 20)\times 1.9957 - 0.008252$$
$$(T_b - 20) = 67.649$$
$$\boxed{T_b = 87.65\,^\circ\text{C}}$$

Therefore the surface temperature of the motor housing is $T_b = 87.65\,^\circ\text{C}$

Question: 17.108P pg 1

Two 4-m-long and 0.4-cm-thick cast iron (k = 52 W/m·K) steam pipes of outer diameter 10 cm are connected to each other through two 1-cm-thick flanges of outer diameter 18cm. The steam flows inside the pipe at an average temperature of 200°C with a heat transfer coefficient of 180 W/m2·K. The outer surface of the pipe is exposed to an ambient at I2°C, with a heat transfer coefficient of 25 W/m2·K. (*a*) Disregarding the flanges, determine the average outer surface temperature of the pipe, (*b*) Using this temperature for the base of the flange and treating the flanges as the fins, determine the fin efficiency and the rate of heat transfer from the flanges, (*c*) What length of pipe is the flange section equivalent to for heat transfer purposes?

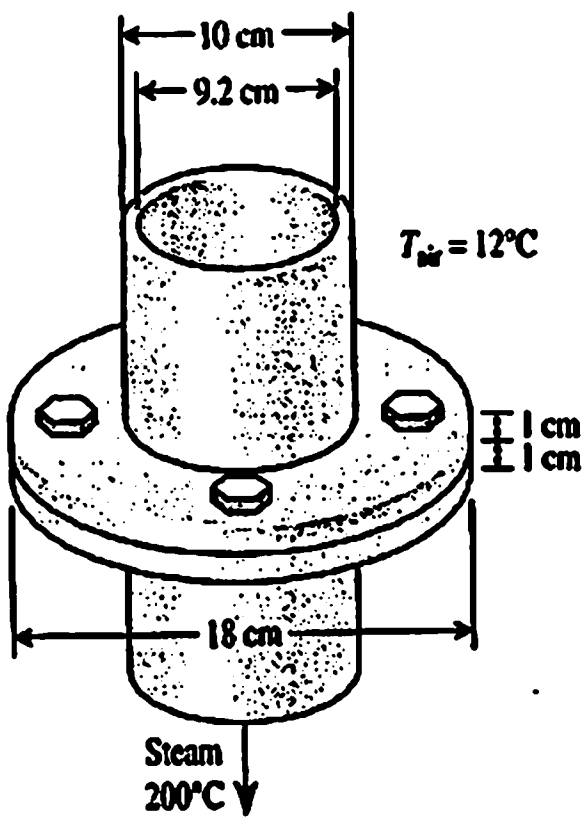


FIGURE P17–108

Step 1 of 10

Given data:

Outer diameter of the pipe, $d_2 = 10\,\text{cm}$

Inner diameter of the pipe, $d_1 = 9.2\,\text{cm}$

Length of the two pipes, $L = 4\,\text{m}$

Thickness of the pipe, $t_p = 0.4\,\text{cm}$

Thermal conductivity of the pipe, $k = 52\,\text{W/m-K}$

Thickness of the flange, $t_{f1} = 1\,\text{cm}$

Outer diameter of the flange, $D_f = 18\,\text{cm}$

Temperature of steam inside the pipe, $T_i = 200°\text{C}$

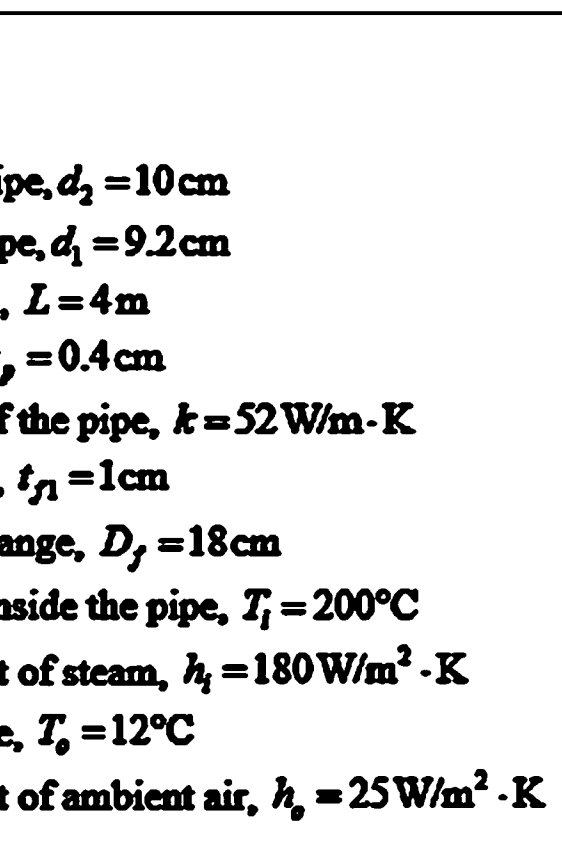
Heat transfer coefficient of steam, $h_i = 180\,\text{W/m}^2\text{-K}$

Ambient air temperature, $T_o = 12°\text{C}$

Heat transfer coefficient of ambient air, $h_o = 25\,\text{W/m}^2\text{-K}$

Step 2 of 10

Given diagram:



Step 3 of 10

a)

The inner and outer surface areas of the pipe are

$$A_i = \pi d_1 L$$
$$= \pi \times (0.092\text{m}) \times (8\text{m})$$
$$= 2.3122\,\text{m}^2$$
$$A_o = \pi d_2 L$$
$$= \pi \times (0.1\text{m}) \times (8\text{m})$$
$$= 2.5132\,\text{m}^2$$

The individual thermal resistances are evaluated as:

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{180(\text{W/m}^2\cdot\text{K}) \times 2.3122(\text{m}^2)}$$
$$= 0.002403\,\text{K/W}$$
$$R_1 = \frac{\ln(r_2/r_1)}{2\pi kL}$$
$$= \frac{\ln(0.05/0.046)}{2\pi \times 52(\text{W/m-K}) \times 8(\text{m})}$$
$$= 0.0000319\,\text{K/W}$$
$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{25(\text{W/m}^2\cdot\text{K}) \times 2.5132(\text{m}^2)}$$
$$= 0.015916\,\text{K/W}$$

Step 4 of 10

All the resistances are in series, and the total resistance will be

$$R_{total} = R_i + R_1 + R_o$$
$$= 0.002403(\text{K/W}) + 0.0000319(\text{K/W}) + 0.015916(\text{K/W})$$
$$= 0.018351\,\text{K/W}$$

Step 4 of 10

All the resistances are in series, and the total resistance will be

$$R_{total} = R_i + R_1 + R_o$$
$$= 0.002403(\text{K/W}) + 0.0000319(\text{K/W}) + 0.015916(\text{K/W})$$
$$= 0.018351\text{K/W}$$

The steady rate of heat loss from the hot water pipe becomes

$$\dot{Q} = \frac{(T_i - T_o)}{R_{total}}$$
$$= \frac{(200-12)}{0.018351(\text{K/W})}$$
$$= 10244.67331\text{W}$$

Step 5 of 10

The surface temperature of the pipe can be determined as

$$\dot{Q} = \frac{(T_s - T_o)}{R_o}$$

Therefore, the surface temperature can be expressed as

$$T_s = T_o + \dot{Q} R_o$$
$$= 12 + 10244.67331 \times 0.015916$$
$$= 175.05°\text{C}$$

$$\boxed{T_s = 175.05°\text{C}}$$

Step 6 of 10

b)

Since convection heat transfer only occurs from one surface of a flange, the connected flanges may be modeled as a single annular fin of thickness $t_f = 2t_{f1} = 2\,\text{cm}$

Hence,

$$r_{2c} = \frac{d_f}{2} + \frac{t_f}{2}$$
$$= \frac{0.18}{2} + \frac{0.02}{2}$$
$$= 0.1\text{m}$$

$$A_{fin} = 2\pi\left(r_{2c}^2 - r_1^2\right)$$
$$= 2\pi \times (0.1^2 - 0.05^2)$$
$$= 0.047123\text{m}^2$$

$$L_c = L + \frac{t_f}{2}$$
$$= \frac{d_f - d_o}{2} + \frac{t_f}{2}$$
$$= \frac{0.18-0.10}{2} + \frac{0.02}{2}$$
$$= 0.05\text{m}$$

Step 7 of 10

And

$$A_p = L_c t_f$$
$$= 0.05(\text{m}) \times 0.02(\text{m})$$
$$= 0.001\text{m}^2$$

$$\xi = L_c^{3/2}\left(h/kA_p\right)^{1/2}$$
$$= (0.05\text{m})^{3/2} \times \left(\frac{25(\text{W/m}^2\text{-K})}{52(\text{W/m-K}) \times 0.001(\text{m}^2)}\right)^{1/2}$$
$$= 0.2451$$

$$\frac{r_{2c}}{r_1} = \frac{r_{2c}}{d_o/2}$$
$$= \frac{0.1}{0.05}$$
$$= 2$$

From the diagram efficiency of annular fins of constant thickness the efficiency of the fin at $\xi = 0.2451$ and $\frac{r_{2c}}{r_1} = 2$ is

$$\eta_{fin} = 0.945$$

$$\boxed{\eta_{fin} = 94.5\%}$$

Step 8 of 10

Heat transfer from the flanges can be expressed as

$$\dot{Q}_{flanges} = \eta_{fin} h A_{fin} (T_s - T_\infty)$$
$$= 0.945 \times 25(\text{W/m}^2\text{-K}) \times 0.047123(\text{m}^2) \times (175.05 - 12)$$
$$= 181.52\,\text{W}$$

$$\boxed{\dot{Q}_{flanges} = 181.52\,\text{W}}$$

Therefore the fin efficiency is $\eta_{fin} = 94.5\%$ and rate of heat transfer from the flange is

$$\dot{Q}_{flanges} = 181.52\,\text{W}$$

Step 9 of 10

c)

The equivalent length of pipe at the flange section for heat transfer can be calculated by assuming that the heat lost from the flanges is same as the heat lost from the pipe for the equivalent length

$$\dot{Q} = \frac{(T_i - T_o)}{R_{total}}$$

$$181.52\,\text{W} = \frac{(200-12)}{R_{total}}$$

$$R_{total} = 1.0357\text{K/W}$$

$$R_{total} = R_i + R_1 + R_o$$

$$R_1 = R_{total} - (R_i + R_o)$$
$$= 1.0357 - [0.002403(\text{K/W}) + 0.015916(\text{K/W})]$$
$$= 1.01738\text{K/W}$$

$R_{total} = R_i + R_1 + R_o$

$R_1 = R_{total} - (R_i + R_o)$

$= 1.0357 - [0.002403(\text{K/W}) + 0.015916(\text{K/W})]$

$= 1.01738\text{K/W}$

$$R_1 = \frac{\ln(r_2/r_1)}{2\pi k L_{eq}}$$

Step 10 of 10

The equivalent pipe length can be expressed as

$$L_{eq} = \frac{\ln(r_2/r_1)}{2\pi k R_1}$$

$$L_{eq} = \frac{\ln(0.05/0.046)}{2\pi \times 52(\text{W/m-K}) \times 1.01738(\text{K/W})}$$

$= 0.000251\text{m}$

$= 0.251\text{mm}$

$\boxed{L_{eq} = 0.251\text{mm}}$

Therefore the equivalent length of the pipe for heat transfer purposes is $L_{eq} = 0.251\text{mm}$

Question: 17.109P

A 40-W power transistor is to be cooled by attaching it to one of the commercially available heat sinks shown in Table 17-6. Select a heat sink that will allow the case temperature of the transistor not to exceed 90°C in the ambient air at 20°C.

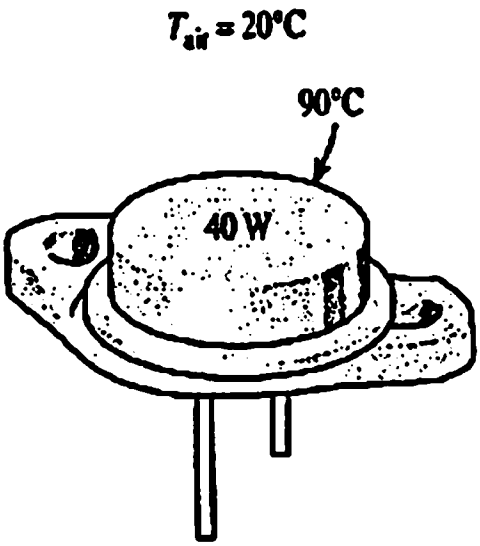


FIGURE P17–109

Step 1 of 2

Given
Power of transistor $(Q) = 40W$
Maximum allowable temperature of transistor $= 90°C$
Ambient air temperature $= 20°C$
Let thermal resistance of heat sink be R

Step 2 of 2

Rate of heat transfer is given by

$$\dot{Q} = \frac{\Delta T}{R}$$

$$R = \frac{T_{transistor} - T_{\infty}}{\dot{Q}}$$

$$R = \frac{(90 - 20)°C}{40\ W}$$

$$R = 1.75\,°C/W$$

Hence the thermal resistance for the heat sink which is selected for this purpose should be below 1.75 °C/W. From table of heat sinks for electronic devices (Table 10-6) HS5030 and HS6115 can be selected in both horizontal and vertical position and HS6071 can be selected in vertical position.

Question: 17.110P

A 25-W power transistor is to be cooled by attaching it to one of the commercially available heat sinks shown in Table 17-6. Select a heat sink that will allow the case temperature of the transistor not to exceed 55°C in the ambient air at 18°C

Step 1 of 3

Given data:

Power transistor $Q = 25\ \text{W}$

Transistor temperature, $T_{\text{transistor}} = 55°\text{C}$

Ambient temperature, $T_{\infty} = 18°\text{C}$

Step 2 of 3

Thermal resistance of heat sink be R

Rate of heat transfer is given by

$$\dot{Q} = \frac{\Delta T}{R}$$

$$R = \frac{T_{\text{transistor}} - T_{\infty}}{\dot{Q}}$$

$$R = \frac{(55-18)°\text{C}}{25\ \text{W}}$$

$$R = 1.48\ °\text{C/W}$$

∴ The thermal resistance $\boxed{R = 1.48°\text{C/W}}$

Step 3 of 3

To select a heat sink from the combined natural convection and radiation thermal resistance of various heat sink table such that temperature of case not to exceed 55°C when ambient air at 18°C

The thermal resistance for the heat sink which is selected for the purpose should be below 1.48 °C/W. From above mentioned table HS6115 can be selected in both horizontal and vertical position and HS6071 can be selected in vertical position.

Question: 17.111P pg 1

A plane wall with surface temperature of 350°C is attached with straight rectangular fins (k= 235 W/m·K). The fins are exposed to an ambient air condition of 25°C and the convection heat transfer coefficient is 154 W/m2·K. Each fin has a length of 50 mm, a base of 5 mm thick and a width of 100 mm. Determine the efficiency, heat transfer rate, and effectiveness of each fin. using (*a*) Table 17–3 and (*b*) Figure 17-43.

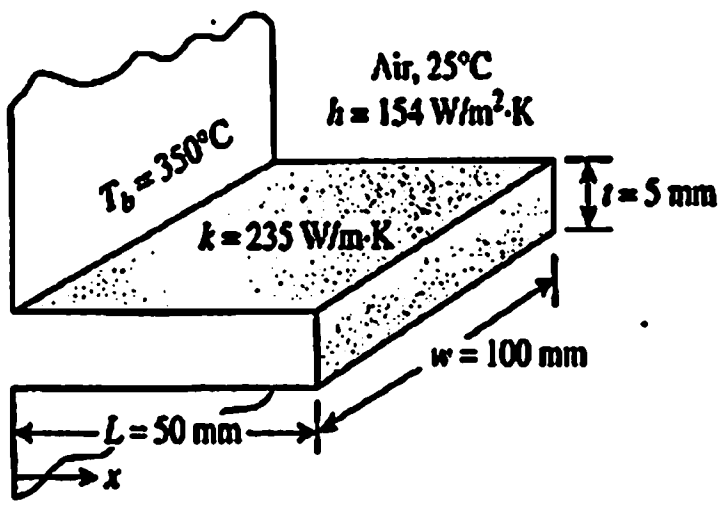


FIGURE P17–111

Step 1 of 7

Length of the fin, $L = 50\,\text{mm}$

Thickness of the fin, $t = 5\text{mm}$

Width of the fin, $w = 100\text{mm}$

Thermal conductivity of the fin, $k = 235\,\text{W/m·K}$

Surface temperature of the wall, $T_b = 350°\text{C}$

Ambient air temperature, $T_\infty = 25°\text{C}$

Convection heat transfer coefficient of ambient air, $h_\infty = 154\,\text{W/m}^2\text{·K}$

Step 2 of 7

a)

From efficiency and surface areas of common fin configuration table

For a straight rectangular fins the efficiency of the fin can be determined as

$$\eta_{fin} = \frac{\tanh mL_c}{mL_c}$$

Where m and L_c can be calculated as

$$m = \sqrt{\frac{2h}{kt}}$$

$$= \sqrt{\frac{2\times154}{235\times0.005}}$$

$$= 16.19035\,(1/\text{m})$$

$$L_c = L + \frac{t}{2}$$

$$= 0.05 + \frac{0.005}{2}$$

$$= 0.0525\,(\text{m})$$

Therefore, the efficiency of the fin can be calculated as

$$\eta_{fin} = \frac{\tanh(16.19035\times0.0525)}{16.19035\times0.0525}$$

$$= 0.813$$

$$= 81.3\%$$

The efficiency of the fin is $\boxed{81.3\%}$

Step 3 of 7

The fin surface area can be expressed as

$$A_{fin} = 2wL_c$$

$$= 2\times0.1\times0.0525$$

$$= 0.0105\,\text{m}^2$$

The rate of heat transfer from the fin can be expressed as

$$\dot{Q}_{fin} = \eta_{fin} h_\infty A_{fin}(T_b - T_\infty)$$

$$= 0.813\times154\times0.0105\times(350-25)$$

$$= 427.252\,\text{W}$$

The rate of heat transfer from the fin is $\boxed{427.252\text{W}}$

Step 4 of 7

The effectiveness of the fin can be expressed as

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{without\ fin}}$$

$$= \frac{\dot{Q}_{fin}}{h_\infty A_b(T_b - T_\infty)}$$

$$= \frac{\text{Heat tarnsfer from the fin of surface area } A_{fin}}{\text{Heat transfer from the fin base area } A_b}$$

Where A_b represents the cross sectional area of the fin at the base and

And $\dot{Q}_{without\ fin}$ represents the rate of heat transfer from the base area if no fins are attached to the surface

Base area A_b can be expressed as

$$A_b = wt$$

$$= 0.1\times0.005$$

$$= 0.0005\,\text{m}^2$$

Therefore, the effectiveness of the fin can be calculated as

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{without\ fin}}$$

$$= \frac{\dot{Q}_{fin}}{h_\infty A_b(T_b - T_\infty)}$$

$$= \frac{427.252}{154\times0.0005\times(350-25)}$$

$$= 17.073$$

$$= \frac{Q_{fin}}{h_w A_b (T_b - T_\infty)}$$

$$= \frac{427.252}{154 \times 0.0005 \times (350 - 25)}$$

$$= 17.073$$

The effectiveness of the fin is $\boxed{17.073}$

Step 5 of 7

b)

Efficiency of straight fins of rectangular profile

$$L_c = L + \frac{t}{2}$$

$$= 0.05 + \frac{0.005}{2}$$

$$= 0.0525\,\text{m}$$

$$A_p = L_c t$$

$$= 0.0525 \times 0.005$$

$$= 0.0002625\,\text{m}^2$$

$$\xi = L_c^{3/2} \left(h / k A_p \right)^{1/2}$$

$$= (0.0525\text{m})^{3/2} \times \left(\frac{154}{235 \times 0.0002625} \right)^{1/2}$$

$$= 0.601$$

Efficiency of straight fins of rectangular profile, the efficiency of the fin at $\xi = 0.601$ is

$\eta_{fin} = 0.81$

$\boxed{\eta_{fin} = 81\%}$

Step 6 of 7

The rate of heat transfer from the fin can be expressed as

$$\dot{Q}_{fin} = \eta_{fin} h_w A_{fin} (T_b - T_\infty)$$

$$= 0.81 \times 154 \times 0.0105 \times (350 - 25)$$

$$= 425.675\,\text{W}$$

$\boxed{\dot{Q}_{fin} = 425.675\text{W}}$

Step 7 of 7

The effectiveness of the fin can be expressed as

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{without\,fin}}$$

$$= \frac{\dot{Q}_{fin}}{h_w A_b (T_b - T_\infty)}$$

$$= \frac{\text{Heat tarnsfer from the fin of surface area } A_{fin}}{\text{Heat transfer from the fin base area } A_b}$$

Where A_b represents the cross sectional area of the fin at the base

And $\dot{Q}_{without\,fin}$ represents the rate of heat transfer from the base area if no fins are attached to the surface

Therefore, the effectiveness of the fin can be calculated as

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{without\,fin}}$$

$$= \frac{\dot{Q}_{fin}}{h_w A_b (T_b - T_\infty)}$$

$$= \frac{425.675}{154 \times 0.0005 \times (350 - 25)}$$

$$= 17.0099$$

$\boxed{\varepsilon_{fin} = 17.0099}$

Question: 17.112P pg 1

Steam in a heating system flows through tubes whose outer diameter is 5 cm and whose walls are maintained at a temperature of 130°C. Circular aluminum alloy 2024-T6 fins (k =186 W/m·K) of outer diameter 6 cm and constant thickness 1 mm are attached to the tube. The space between the fins is 3 mm, and thus there are 250 fins per meter length of the tube. Heat is transferred to the surrounding air at $T\infty$ =25°C, with a heat transfer coefficient of 40 W/m2·K. Determine the increase in heat transfer from the tube per meter of its length as a result of adding fins.

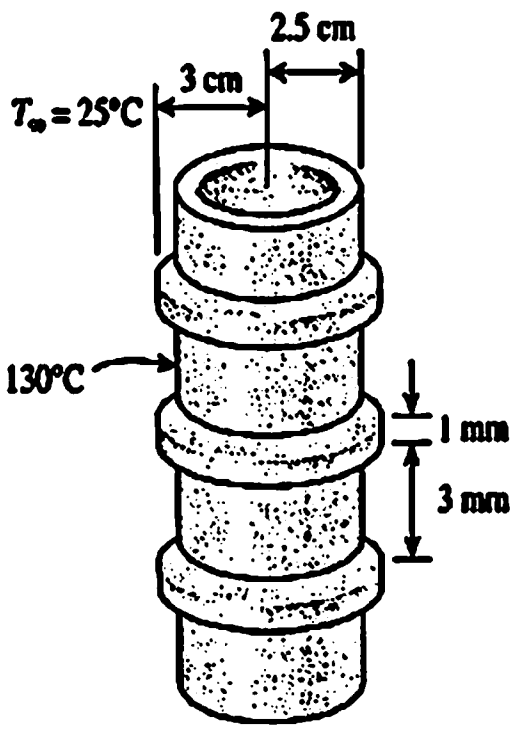


FIGURE P17–112

Step 1 of 7

Given data:

Outer diameter of tubes $D_1 = 5$ cm

Temperature of walls $T_b = 130°C$

Thermal conductivity of 2024-T6 fins $k = 186$ W/m·°C

Outer diameter of fins $D_f = 6$ cm

Thickness of fins $t = 1$ mm

Space between fins $s = 3$mm

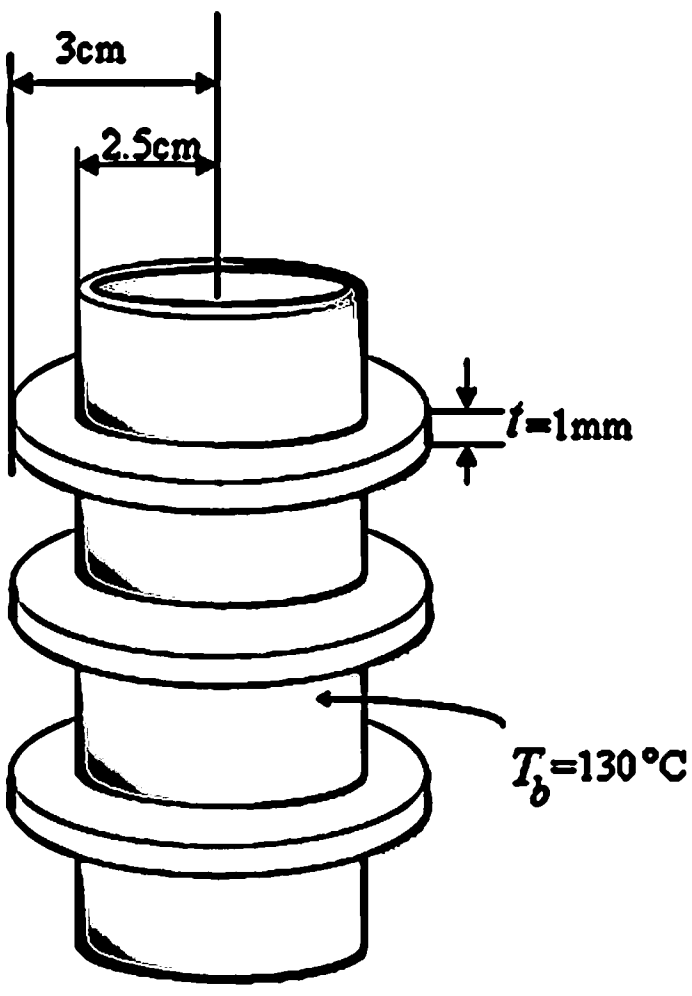
Ambient air temperature, $T_\infty = 25°C$

Heat transfer coefficient of ambient air, $h_\infty = 40 W/m^2 \cdot K$

Number of fins per meter length, $n = 250$

Step 2 of 7

Given diagram:



Step 3 of 7

The efficiency of the fin can be calculated by calculating these parameters

$$r_{2c} = \frac{D_f}{2} + \frac{t_f}{2}$$
$$= \frac{0.06}{2} + \frac{0.001}{2}$$
$$= 0.0305\,\text{m}$$
$$A_{fin} = 2\pi\left(r_{2c}^2 - r_1^2\right)$$
$$= 2\pi \times (0.0305^2 - 0.025^2)$$
$$= 0.00191794\,\text{m}^2$$
$$L_c = L + \frac{t_f}{2}$$
$$= \frac{D_f - D_1}{2} + \frac{t_f}{2}$$
$$= \frac{0.06 - 0.05}{2} + \frac{0.001}{2}$$
$$= 0.0055\,\text{m}$$

Step 4 of 7

And

$$A_p = L_c t_f$$
$$= 0.0055 \times 0.001$$
$$= 0.0000055\,\text{m}^2$$
$$\xi = L_c^{3/2}\left(h/kA_p\right)^{1/2}$$
$$= (0.0055\text{m})^{3/2} \times \left(\frac{40}{186 \times 0.0000055}\right)^{1/2}$$
$$= 0.0806559$$
$$\frac{r_{2c}}{r_1} = \frac{r_{2c}}{D_1/2}$$
$$= \frac{0.0305}{0.025}$$
$$= 1.22$$

From the graph efficiency of annular fins at $\xi = 0.0806559$ and $\frac{r_{2c}}{r_1} = 1.22$

$$= \frac{0.0305}{0.025}$$
$$= 1.22$$

From the graph efficiency of annular fins at $\xi = 0.0806559$ and $\frac{r_{2c}}{r_1} = 1.22$

$$\eta_{fin} = 0.97$$

Step 5 of 7

The rate of heat transfer from the fin can be expressed as

$$\dot{Q}_{fin} = \eta_{fin} h A_{fin} (T_s - T_\infty)$$
$$= 0.97 \times 40 \times 0.00191794 \times (130 - 25)$$
$$= 7.8136\,\text{W}$$

Step 6 of 7

For 250 fins the total heat transfer rate from the fins can be expressed as

$$\dot{Q}_{fins} = n \times \dot{Q}_{fin}$$
$$= 250 \times 7.81368$$
$$= 1953.42\,\text{W}$$

The heat transfer from the base area if no fins are attached to the surface can be calculated as

$$\dot{Q}_{no\,fin} = h_\infty A_b (T_b - T_\infty)$$
$$A_b = 2\pi r_1 t$$
$$= 2 \times \pi \times 0.025 \times 0.001$$
$$= 0.0001571\,\text{m}^2$$

$$\dot{Q}_{no\,fin} = h_\infty A_b (T_b - T_\infty)$$
$$= 40 \times 0.0001571 \times (130 - 25)$$
$$= 0.65982\,\text{W}$$

Step 7 of 7

For 250 fins, the total heat transfer rate from the fins base area becomes

$$\dot{Q}_{no\,fins} = n \times \dot{Q}_{no\,fin}$$
$$= 250 \times 0.65982$$
$$= 164.955\,\text{W}$$

Therefore, the increase in heat transfer due to addition of fins can be calculated as

$$\Delta\dot{Q}_{increase} = \dot{Q}_{fins} - \dot{Q}_{no\,fins}$$
$$= 1953.42 - 164.955$$
$$= 1788.465\,\text{W}$$

$$\boxed{\Delta\dot{Q}_{increase} = 1788.5\,\text{W}}$$

Question: 17.113P

Consider a stainless steel spoon (k =8.7Btu/h·ft·°F) partially immersed in boiling water at 200°F in a kitchen at 75°F. The handle of the spoon has a cross section of 0.08 in × 0.5 in, and extends 7 in in the air from the free surface of the water. If the heat transfer coefficient at the exposed surfaces of the spoon handle is 3 Btu/h·ft2·°F, determine the temperature difference across the exposed surface of the spoon handle. State your assumptions.

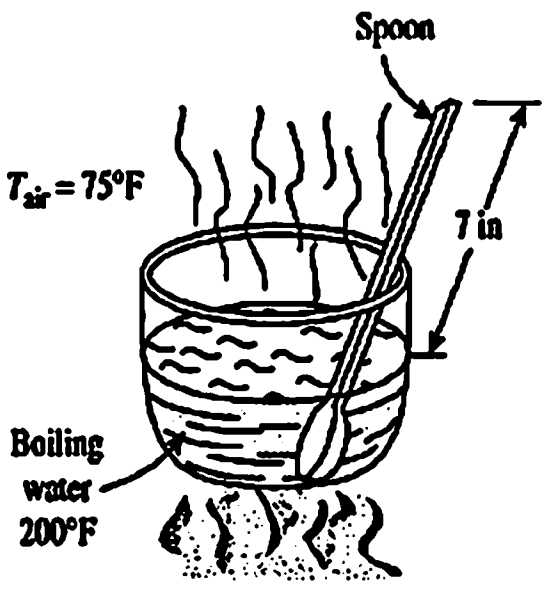


FIGURE P17–113E

Step 1 of 6

Given data:

Cross section area of spoon $A = 0.008\text{ in} \times 0.5\text{ in}$

Temperature of boiling water $T_b = 200°F$

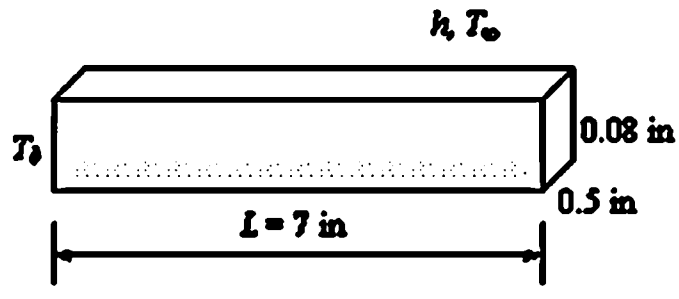
Temperature of kitchen $T_\infty = 75°F$

Heat transfer coefficient $h = 3\text{ Btu/h}\cdot\text{ft}^2\cdot°F$

Thermal conductivity of spoon $k = 8.7\text{ Btu/h/ft}\cdot°F$

Spoon extends 7 in from the surface of water

Step 2 of 6



Step 3 of 6

The temperature variation of spoon can be given by

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=\frac{\cosh m(L-x)}{\cosh mL}$$

p = perimeter

$$= 2\left(\frac{0.5}{12}\text{ft}+\frac{0.08}{12}\text{ft}\right)$$

$$= 0.0967\text{ ft}$$

$$A_c = \left(\frac{0.5}{12}\text{ft}\right)\left(\frac{0.08}{12}\text{ft}\right)$$

$$= 0.000278\text{ft}^2$$

Step 4 of 6

Therefore the fin parameter

$$m = \sqrt{\frac{hp}{kA_c}}$$

$$= \sqrt{\frac{(3\text{ Btu/h.ft}^2.°F)(0.0967\text{ ft})}{(8.7\text{ Btu/h.ft.°F})(0.000278\text{ ft}^2)}}$$

$$= 10.95\text{ ft}^{-1}$$

Step 5 of 6

Now $x = L$

$L = 7/12$

$L = 0.583$ ft

$$T(L) = T_\infty + (T_b - T_\infty)\frac{\cosh m(L-L)}{\cosh mL}$$

$$= 75°F + (200-75)\frac{\cosh 0}{\cosh(10.95\times 0.583)}$$

$$= 75°F + (200-75)\frac{1}{296}$$

$$= 75.4°F$$

Step 6 of 6

Therefore, the temperature difference across the exposed surface of the spoon handle is

$$\Delta T = T_b - T_{tip}$$

$$= (200-75.4)°F$$

$$= 124.6°F$$

∴ The temperature difference $\boxed{\Delta T = 124.6°F}$

Question: 17.114P

Repeat Prob. 17-113E for a silver spoon (k =247 Btu/h·ft·°F).

Step 1 of 6

Given data:

Cross section area of spoon $A = 0.08\text{ in} \times 0.5\text{ in}$

Temperature of boiling water $T_b = 200°\text{F}$

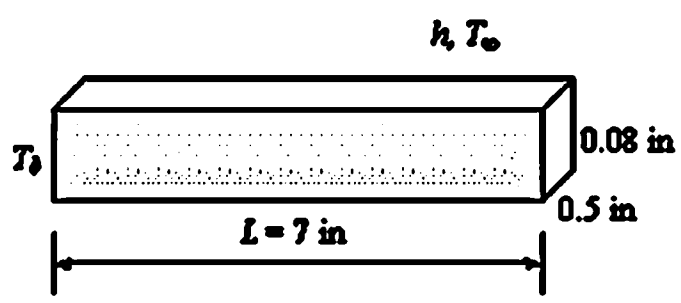
Temperature of kitchen $T_\infty = 75°\text{F}$

Heat transfer coefficient $h = 3\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$

Thermal conductivity of spoon $k = 247\text{ Btu/h/ft}\cdot°\text{F}$

Spoon extends 7 in from the surface of water

Step 2 of 6



Step 3 of 6

Assuming the spoon handle is acts as a fin

The temperature variation of spoon is be given by the equation

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=\frac{\cosh m(L-x)}{\cosh mL}$$

p = perimeter

$$=2\left(\frac{0.5}{12}\text{ft}+\frac{0.08}{12}\text{ft}\right)$$

$$=0.0967\text{ ft}$$

$$A_c=\left(\frac{0.5}{12}\text{ft}\right)\left(\frac{0.08}{12}\text{ft}\right)$$

$$=0.000278\text{ft}^2$$

Step 4 of 6

Therefore the fin parameter

$$m=\sqrt{\frac{hp}{kA_c}}$$

$$=\sqrt{\frac{(3\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(0.0967\text{ft})}{(247\text{ Btu/h}\cdot\text{ft}\cdot°\text{F})(0.000278\text{ ft}^2)}}$$

$$=2.055\text{ ft}^{-1}$$

Step 5 of 6

Now $x = L$

$L = 7/12$

$L = 0.583$ ft

$$T(L)=T_\infty+(T_b-T_\infty)\frac{\cosh m(L-L)}{\cosh mL}$$

$$=75°\text{F}+(200-75)\frac{\cosh 0}{\cosh(2.055\times0.583)}$$

$$=75°\text{F}+(200-75)\frac{1}{1.81}$$

$$=144.1°\text{F}$$

Step 6 of 6

Therefore, the temperature difference across the exposed surface of the spoon handle is

$\Delta T = T_b - T_{tip}$

$$=(200-144.1)°\text{F}$$

$$=55.9°\text{F}$$

∴ The temperature difference $\boxed{\Delta T = 55.9°\text{F}}$

Question: 17.115P pg 1

Reconsider Prob. 17 −113E. Using EES (or other) software, investigate the effects of the thermal conductivity of the spoon material and the length of its extension in the air on the temperature difference across the exposed surface of the spoon handle. Let the thermal conductivity vary from 5 Btu/h·ft·°F to 225 Btu/h·ft·°F and the length from 5 in to 12 in. Plot the temperature difference as the functions of thermal conductivity and length, and discuss the results.

Step 1 of 5

To solve this question using EES software open EES software

Then FILE->New

A dailogbox will open type the following in dailog box

"GIVEN"

"k_spoon=8.7 [Btu/h-ft-F], parameter to be varied"

T_w=200 "[F]"

T_infinity=75 "[F]"

A_c=0.08/12*0.5/12 "[ft^2]"

L=7 "[in], parameter to be varied"

h=3 "[Btu/h-ft^2-F]"

Step 2 of 5

"ANALYSIS"

p=2*(0.08/12+0.5/12)

a=sqrt((h*p)/(k_spoon*A_c))

(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))

x=L "for tip temperature"

DELTAT=T_w-T_tip

Then from Tables -> New parametric table

A dailog box will open

Select no of runs = 20

Select variables in equations as k_{spoon} and T press ADD and then press ok

A table will appear put values of k_{spoon} as 5,16.58,28.16....225 in run 1to 20

Then from Calculate -> Solve table

Solution will appear

k_{spoon} [Btu/h.ft.F]	T [F]
5	124.9
16.58	122.6
28.16	117.8
39.74	112.5
51.32	107.1
62.89	102
74.47	97.21
86.05	92.78
97.63	88.69
109.2	84.91
120.8	81.42
132.4	78.19
143.9	75.19
155.5	72.41
167.1	69.82
178.7	67.4
190.3	65.14
201.8	63.02
213.4	61.04
225	59.17

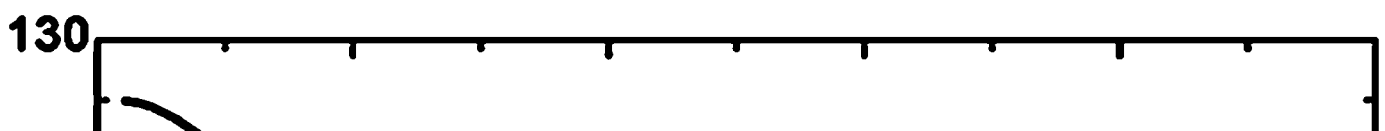
Step 3 of 5

Then from Plot -> New plot window -> X-Y Plot

Select X-axis k_{spoon} and Y-axis T

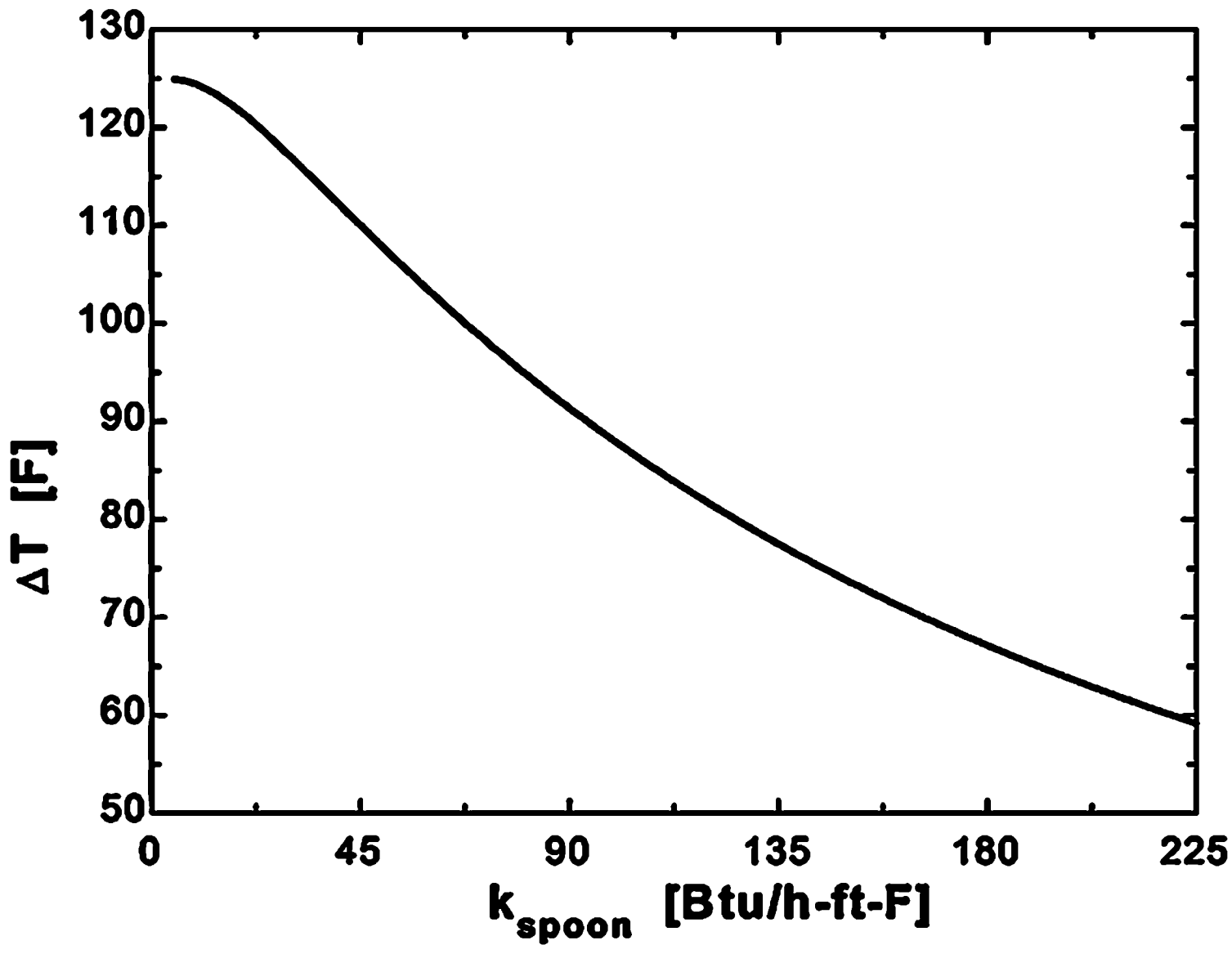
Press OK

Graph will be plotted.



Press OK

Graph will be plotted.



Step 4 of 5

Edit the text in dialog box

"GIVEN"

k_spoon=8.7" [Btu/h-ft-F], parameter to be varied"

T_w=200 "[F]"

T_infinity=75 "[F]"

A_c=0.08/12*0.5/12 "[ft^2]"

"L=7 "[in], parameter to be varied"

h=3 "[Btu/h-ft^2-F]"

"ANALYSIS"

p=2*(0.08/12+0.5/12)

a=sqrt((h*p)/(k_spoon*A_c))

(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))

x=L "for tip temperature"

DELTAT=T_w-T_tip

Then from Tables -> New parametric table

A dailog box will open

Select no of runs = 20

Select variables in equations as L and T press ADD and then press ok

A table will appear put values of L as 5,16.58,28.16....225 in run 1to 20

Then from Calculate -> Solve table

Solution will appear

L [Btu/h.ft.F]	T [F]
5	122.4
5.5	123.4
6	124
6.5	124.3
7	124.6
7.5	124.7
8	124.8
8.5	124.9
9	124.9
9.5	125
10	125
10.5	125
11	125
11.5	125
12	125

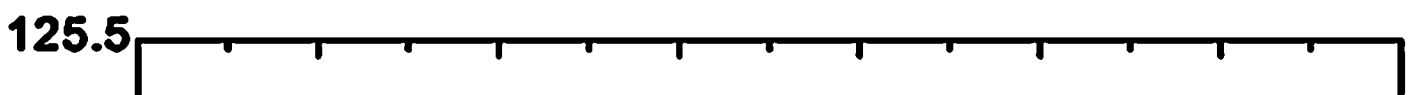
Step 5 of 5

Then from Plot -> New plot window -> X-Y Plot

Select X-axis L and Y-axis T

Press OK

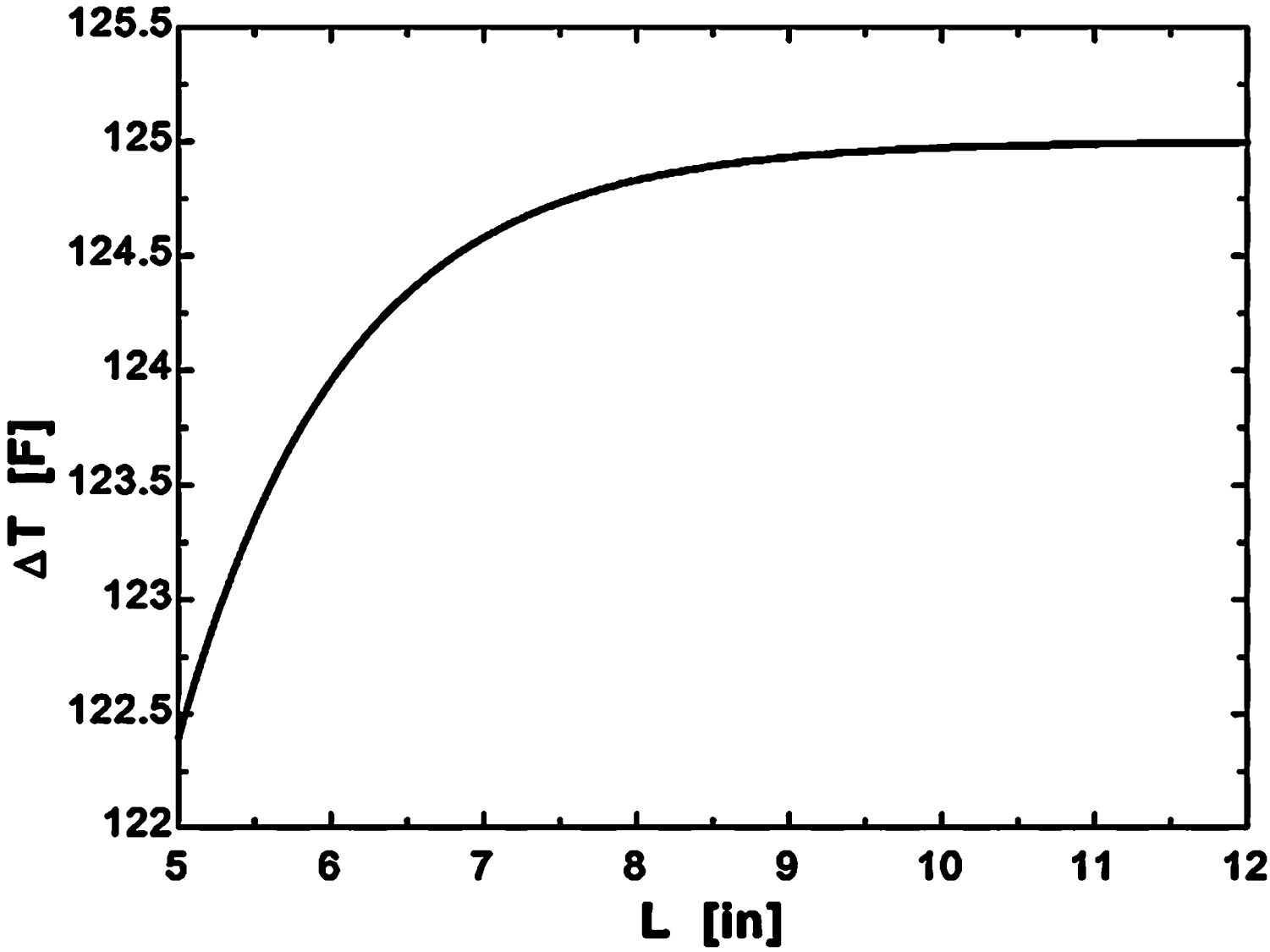
Graph will be plotted.



Press OK

17.115P pg 3

Graph will be plotted.



Question: 17.116P pg 1

A 0.4-cm-thick, 12-cm-high, and 18-cm-long circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.04 W. The board is impregnated with copper fillings and has an effective thermal conductivity of 30 W/m·K. All the heat generated in the chips is conducted across the circuit board and is dissipated from the back side of the board to a medium at 40°C, with a heat transfer coefficient of 52 W/m2·K. (*a*) Determine the temperatures on the two sides of the circuit board, (*b*) Now a 0.2-cm-thick, 12-cm-high, and 18- cm-long aluminum plate (k =237 W/m·K) with 864 2-cm- long aluminum pin fins of diameter 0.25 cm is attached to the back side of the circuit board with a 0.02-cm-thick epoxy adhesive (k = 1.8 W/m·K). Determine the new temperatures on the two sides of the circuit board.

Step 1 of 11

Given data:

Height of the circuit board, $H = 0.12\,\text{cm}$

Length of the circuit board, $L = 18\text{cm}$

Thickness of the board, $t = 0.4\,\text{cm}$

Thermal conductivity of the circuit board, $k = 30\,\text{W/m}\cdot\text{K}$

Heat generated by each chip, $\dot{Q} = 0.04\,\text{W}$

Ambient temperature, $T_\infty = 40°\text{C}$

Heat transfer coefficient of ambient medium, $h_\infty = 52\,\text{W/m}^2\cdot\text{K}$

Number of logic chips, $n = 80$

Step 2 of 11

a)

Total heat dissipated by chips

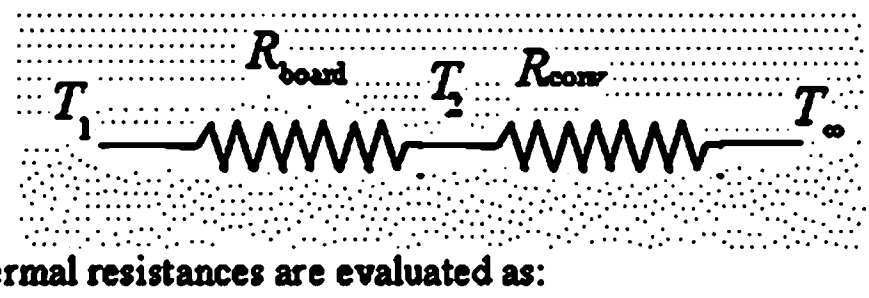
$$\dot{Q} = n \times Q$$
$$= 80 \times 0.04$$
$$= 3.2\ \text{W}$$

The surface area of the circuit board can be calculated as

$$A = HL$$
$$= 0.12 \times 0.18$$
$$= 0.0216\ \text{m}^2$$

Step 3 of 11

The thermal resistance of the circuit board is



The individual thermal resistances are evaluated as:

The thermal resistance due to conduction

$$R_{board} = \frac{t}{kA}$$
$$= \frac{0.004}{30 \times 0.0216}$$
$$= 0.00617284\,\text{K/W}$$

The thermal resistance due to convection

$$R_{conv} = \frac{1}{h_\infty A}$$
$$= \frac{1}{52 \times 0.0216}$$
$$= 0.8903134\,\text{K/W}$$

Since all the resistances are in series, and the total resistance is

$$R_{total} = R_{board} + R_{conv}$$
$$= 0.00617284 + 0.8903134$$
$$= 0.8964862\,\text{K/W}$$

Step 4 of 11

The temperatures on both sides of the circuit board are calculated by

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{total}}$$
$$T_1 = T_\infty + \dot{Q}R_{total}$$
$$= 40 + (3.2)(0.8964862)$$
$$= 42.868°\text{C}$$

The temperature on one side of the board $\boxed{T_1 = 42.868°\text{C}}$

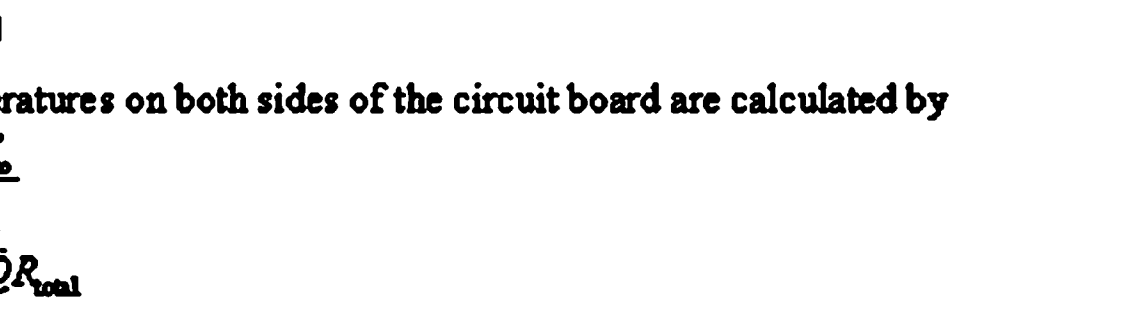
Step 5 of 11

$$\dot{Q} = \frac{T_1 - T_2}{R_{board}}$$
$$T_2 = T_1 - \dot{Q}R_{board}$$
$$= 42.868 - (3.2)(0.00617284)$$
$$= 42.848°\text{C}$$

The temperature on the other side of the board $\boxed{T_2 = 42.848°\text{C}}$

Step 6 of 11

b) The thermal resistance of the circuit board with new resistances becomes



Step 7 of 11

The cross sectional areas of the fins are constant, the efficiency of the circular fins for tip condition assumed to be adiabatic, can be determined as

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

Step 7 of 11

The cross sectional areas of the fins are constant, the efficiency of the circular fins for tip condition assumed to be adiabatic, can be determined as

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

Where

$$m = \sqrt{\frac{hp}{kA_c}}$$

$$= \sqrt{\frac{52 \times \pi \times 0.0025}{237 \times \frac{\pi}{4} \times (0.0025)^2}}$$

$$= 18.7365$$

$$mL = 18.7365 \times 0.02$$

$$= 0.37473$$

Therefore, the efficiency of the fin becomes

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

$$= \frac{\tanh (0.37473)}{0.37473}$$

$$= 0.9557$$

$$= 95.57\%$$

Step 8 of 11

Thermal resistance due to circuit board

$$R_{board} = \frac{t}{kA}$$

$$= \frac{0.004}{30 \times 0.0216}$$

$$= 0.00617284 \text{ K/W}$$

Thermal resistance due to thick epoxy adhesive

$$R_{epoxy} = \frac{t}{kA}$$

$$= \frac{0.0002}{1.8 \times 0.0216}$$

$$= 0.005144 \text{ K/W}$$

Thermal resistance due to aluminum plate

$$R_{alu} = \frac{t}{kA}$$

$$= \frac{0.002}{237 \times 0.0216}$$

$$= 0.00039 \text{ K/W}$$

Step 9 of 11

Area of the finned surface

$$A_{finned} = \eta_{fin} n \pi D L$$

$$= 0.9557 \times 864 \pi (0.0025)(0.02)$$

$$= 0.1297 \text{ m}^2$$

Area of the unfinned surface

$$A_{unfinned} = 0.0216 - 864 \frac{\pi D^2}{4}$$

$$= 0.0216 - 864 \times \frac{\pi (0.0025)^2}{4}$$

$$= 0.0174 \text{ m}^2$$

$$A_{total,with fins} = A_{finned} + A_{unfinned}$$

$$= 0.1297 + 0.0174$$

$$= 0.1471 \text{m}^2$$

Step 10 of 11

$$R_{conv} = \frac{1}{hA_{total,with fins}}$$

$$= \frac{1}{(52)(0.1471)}$$

$$= 0.13073 \text{K/W}$$

$$R_{total} = R_{board} + R_{epoxy} + R_{aluminum} + R_{conv}$$

$$= 0.00617284 + 0.005144 + 0.00039 + 0.13073$$

$$= 0.14243 \text{K/W}$$

Step 11 of 11

Temperatures on both sides of the circuit board is

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{total}}$$

$$T_1 = T_{\infty 2} + \dot{Q} R_{total}$$

$$= 40 + (3.2)(0.14243)$$

$$= 40.45^\circ\text{C}$$

The temperature on one side of the board $\boxed{T_1 = 40.45^\circ\text{C}}$

Question: 17.117P pg 1

Repeat Prob. 17-116 using a copper plate with copper fins (ft= 386 W/m·K) instead of aluminum ones.

Step 1 of 11

Given data:

Height of the circuit board, $H = 0.12\,\text{cm}$

Length of the circuit board, $L = 18\text{cm}$

Thickness of the board, $t = 0.4\,\text{cm}$

Thermal conductivity of the circuit board, $k = 30\,\text{W/m}\cdot\text{K}$

Heat generated by each chip, $\dot{Q} = 0.04\,\text{W}$

Ambient temperature, $T_\infty = 40°\text{C}$

Heat transfer coefficient of ambient medium, $h_\infty = 52\,\text{W/m}^2\cdot\text{K}$

Number of logic chips, $n = 80$

Step 2 of 11

a)

Total heat dissipated by chips

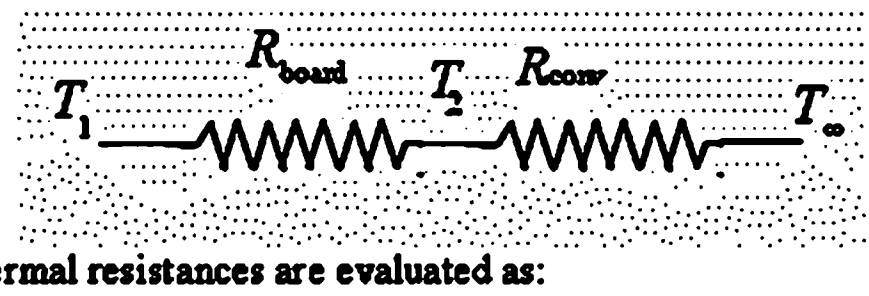
$$\dot{Q} = n \times Q$$
$$= 80 \times 0.04$$
$$= 3.2\ \text{W}$$

The surface area of the circuit board can be calculated as

$$A = HL$$
$$= 0.12 \times 0.18$$
$$= 0.0216\,\text{m}^2$$

Step 3 of 11

The thermal resistance of the circuit board is



The individual thermal resistances are evaluated as:

The thermal resistance due to conduction

$$R_{board} = \frac{t}{kA}$$
$$= \frac{0.004}{30 \times 0.0216}$$
$$= 0.00617284\,\text{K/W}$$

The thermal resistance due to convection

$$R_{conv} = \frac{1}{h_\infty A}$$
$$= \frac{1}{52 \times 0.0216}$$
$$= 0.8903134\,\text{K/W}$$

Since all the resistances are in series, and the total resistance is

$$R_{total} = R_{board} + R_{conv}$$
$$= 0.00617284 + 0.8903134$$
$$= 0.8964862\,\text{K/W}$$

Step 4 of 11

The temperatures on both sides of the circuit board are calculated by

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{total}}$$
$$T_1 = T_\infty + \dot{Q}R_{total}$$
$$= 40 + (3.2)(0.8964862)$$
$$= 42.868°\text{C}$$

The temperature on one side of the board $\boxed{T_1 = 42.868°\text{C}}$

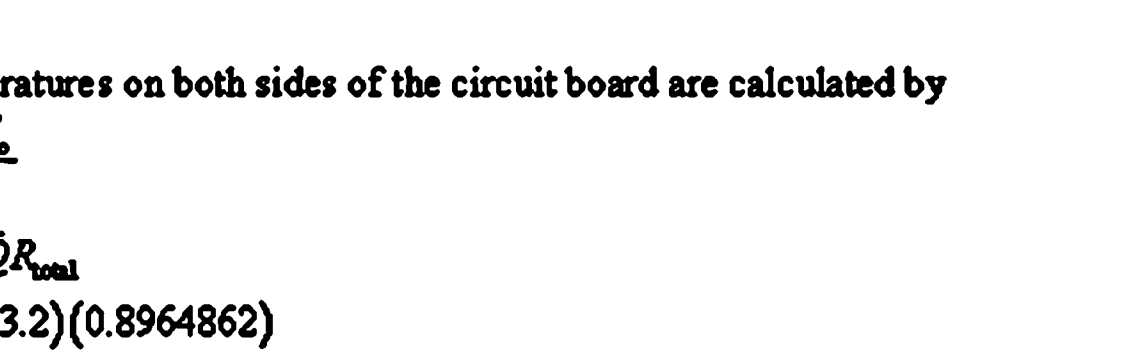
Step 5 of 11

$$\dot{Q} = \frac{T_1 - T_2}{R_{board}}$$
$$T_2 = T_1 - \dot{Q}R_{board}$$
$$= 42.868 - (3.2)(0.00617284)$$
$$= 42.848°\text{C}$$

The temperature on the other side of the board $\boxed{T_2 = 42.848°\text{C}}$

Step 6 of 11

b) The thermal resistance of the circuit board with new resistances becomes



Step 7 of 11

The cross sectional areas of the fins are constant, the efficiency of the circular fins for tip condition assumed to be adiabatic, can be determined as

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

Where

$$m = \sqrt{\frac{hp}{kA_c}}$$
$$= \sqrt{\frac{52 \times \pi \times 0.0025}{}}$$

Where

$$m = \sqrt{\frac{hp}{kA_c}}$$

$$= \sqrt{\frac{52 \times \pi \times 0.0025}{386 \times \frac{\pi}{4} \times (0.0025)^2}}$$

$$= 14.681$$

$$mL = 14.681 \times 0.02$$

$$= 0.29362$$

Therefore, the efficiency of the fin becomes

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

$$= \frac{\tanh (0.29362)}{0.29362}$$

$$= 0.9722$$

$$= 97.22\%$$

Step 8 of 11

Thermal resistance due to circuit board

$$R_{board} = \frac{t}{kA}$$

$$= \frac{0.004}{30 \times 0.0216}$$

$$= 0.00617284\,K/W$$

Thermal resistance due to thick epoxy adhesive

$$R_{epoxy} = \frac{t}{kA}$$

$$= \frac{0.0002}{1.8 \times 0.0216}$$

$$= 0.005144\,K/W$$

Thermal resistance due to copper plate

$$R_{cu} = \frac{t}{kA}$$

$$= \frac{0.002}{386 \times 0.0216}$$

$$= 0.000239\,K/W$$

Step 9 of 11

Area of the finned surface

$$A_{finned} = \eta_{fin} n \pi D L$$

$$= 0.9722 \times 864 \pi (0.0025)(0.02)$$

$$= 0.13194\ m^2$$

Area of the unfinned surface

$$A_{unfinned} = 0.0216 - 864 \frac{\pi D^2}{4}$$

$$= 0.0216 - 864 \times \frac{\pi (0.0025)^2}{4}$$

$$= 0.0174\ m^2$$

$$A_{total with fins} = A_{finned} + A_{unfinned}$$

$$= 0.13194 + 0.0174$$

$$= 0.1493 m^2$$

Step 10 of 11

$$R_{conv} = \frac{1}{hA_{total with fins}}$$

$$= \frac{1}{(52)(0.1493)}$$

$$= 0.1288 k/W$$

$$R_{total} = R_{board} + R_{epoxy} + R_{cu} + R_{conv}$$

$$= 0.00617284 + 0.005144 + 0.000239 + 0.1288$$

$$= 0.140355 K/W$$

Step 11 of 11

Temperatures on both sides of the circuit board is

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{total}}$$

$$T_1 = T_{\infty 2} + \dot{Q} R_{total}$$

$$= 40 + (3.2)(0.140355)$$

$$= 40.449°C$$

The temperature on one side of the board $\boxed{T_1 = 40.449°C}$

Question: 17.118P

A hot surface at 100°C is to be cooled by attaching 3-cm-long, 0.25-cm-diameter aluminum pin fins (k = 237 W/m·K) to it, with a center-to-center distance of 0.6 cm. The temperature of the surrounding medium is 30°C, and the heat transfer coefficient on the surfaces is 35 W/m2· K. Determine the rate of heat transfer from the surface for a 1 -m × 1 -m section of the plate. Also determine the overall effectiveness of the fins.

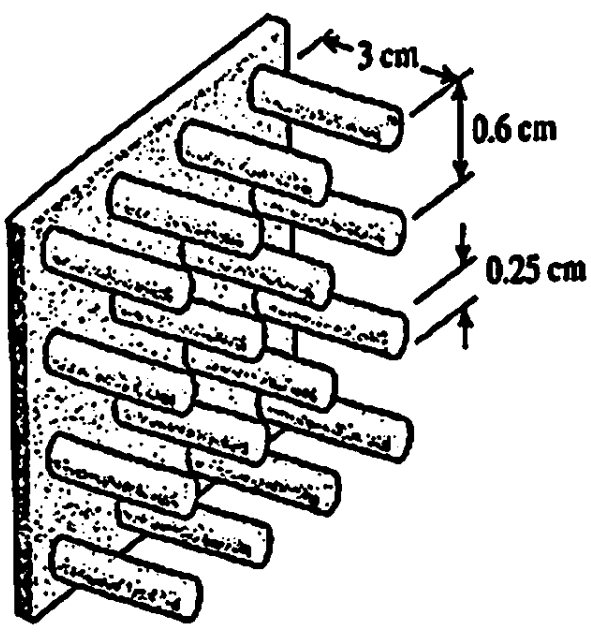


FIGURE P17–118

Step 1 of 10

Given data:

Temperature of surface $= 100°C$

Length of fins $L = 3\text{cm}$

Diameter of fins $D = 0.25\text{cm}$

Thermal conductivity of fins $k = 237\ \text{W/m} \cdot °C$

Centre to centre distance $= 0.6\ \text{cm}$

Temperature of the hot surface $T_s = 100°C$

Temperature of surrounding medium $T_\infty = 30°C$

Heat transfer coefficient $h = 35\ \text{W/m}^2 \cdot °C$

Step 2 of 10

Efficiency of the fins can be calculated by

Fin parameter

$$m = \sqrt{\frac{hp}{kA_c}}$$

$$= \sqrt{\frac{h\pi D}{k\pi D^2/4}}$$

$$= \sqrt{\frac{4h}{kD}}$$

$$= \sqrt{\frac{4(35\ \text{W/m}^2 \cdot °C)}{(237\ \text{W/m} \cdot °C)(0.0025\ \text{m})}}$$

$$= 15.37\ \text{m}^{-1}$$

Step 3 of 10

The fin efficiency is given by

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL}$$

$$= \frac{\tanh(15.37\ \text{m}^{-1} \times 0.03\ \text{m})}{15.37\ \text{m}^{-1} \times 0.03\ \text{m}}$$

$$= 0.935$$

Total number of fins

$$n = \frac{1\ \text{m}^2}{(0.006\ \text{m})(0.006\ \text{m})}$$

$$= 27,777$$

Step 4 of 10

Surface area of fins

$$A_{\text{fin}} = 27777\left[\pi DL + \frac{\pi D^2}{4}\right]$$

$$= 27777\left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4}\right]$$

$$= 6.68\ \text{m}^2$$

Step 5 of 10

Un finned surface area

$$A_{\text{unfinned}} = 1 - 27777\left(\frac{\pi D^2}{4}\right)$$

$$= 1 - 27777\left[\frac{\pi(0.0025)^2}{4}\right]$$

$$= 0.86\ \text{m}^2$$

Step 6 of 10

Heat transfer through finned surface

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}} \dot{Q}_{\text{finmax}}$$

$$= \eta_{\text{fin}} h A_{\text{fin}} (T_s - T_\infty)$$

$$= 0.935\left(35\ \text{W/m}^2 \cdot °C\right)\left(6.68\ \text{m}^2\right)(100-30)°C$$

$$= 15,302.2\ \text{W}$$

Step 7 of 10

Heat transfer through un finned surface

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}} (T_s - T_\infty)$$

$$= \left(35\ \text{W/m}^2 \cdot °C\right)\left(0.86\ \text{m}^2\right)(100-30)°C$$

$$= 2107\ \text{W}$$

Step 8 of 10

Total heat transfer from the plate with fins

$$\dot{Q}_{\text{total fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}}$$

$$= 15,300 + 2107$$

$$= 1.74 \times 10^4\ \text{W}$$

$$= 17.4\ \text{kW}$$

Step 9 of 10

Rate of heat transfer when there are no fins

$$A_{\text{without fin}} = (1\ \text{m})(1\ \text{m})$$

$$= 1\ \text{m}^2$$

$$\dot{Q}_{\text{without fin}} = hA_{\text{no fin}} (T_s - T_\infty)$$

$$= \left(35\ \text{W/m}^2.°C\right)\left(1\ \text{m}^2\right)(100-30)°C$$

$$= 2450\ \text{W}$$

∴ The rate of heat transfer $\boxed{\dot{Q} = 2450\ \text{W}}$

Step 10 of 10

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{without fin}}}$$

$$= \frac{17,400}{2450}$$

$$= 7.10$$

∴ The overall effectiveness of the fin $\boxed{\varepsilon_{\text{fin}} = 7.10}$

Question: 17.119P

Repeat Prob. 17-118 using copper fins (k =386 W/m·K) instead of aluminum ones.

Step 1 of 10

Given data:
Temperature of surface $= 100°C$
Length of fins $L = 3\text{cm}$
Diameter of fins $D = 0.25\text{cm}$
Thermal conductivity of fins $k = 386\ \text{W/m} \cdot °C$
Centre to centre distance $= 0.6\ \text{cm}$
Temperature of the hot surface $T_s = 100°C$
Temperature of surrounding medium $T_\infty = 30°C$
Heat transfer coefficient $h = 35\ \text{W/m}^2 \cdot °C$

Step 2 of 10

Efficiency of the fins can be calculated by
Fin parameter

$$m = \sqrt{\frac{hp}{kA_c}}$$
$$= \sqrt{\frac{h\pi D}{k\pi D^2/4}}$$
$$= \sqrt{\frac{4h}{kD}}$$
$$= \sqrt{\frac{4(35\ \text{W/m}^2 \cdot °C)}{(386\ \text{W/m} \cdot °C)(0.0025\ \text{m})}}$$
$$= 12.04\ \text{m}^{-1}$$

Step 3 of 10

The fin efficiency is given by

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL}$$
$$= \frac{\tanh(12.04\ \text{m}^{-1} \times 0.03\ \text{m})}{12.04\ \text{m}^{-1} \times 0.03\ \text{m}}$$
$$= 0.959$$

Step 4 of 10

Total number of fins

$$n = \frac{1\ \text{m}^2}{(0.006\ \text{m})(0.006\ \text{m})}$$
$$= 27777$$

Surface area of fins

$$A_{\text{fin}} = 27777\left[\pi DL + \frac{\pi D^2}{4}\right]$$
$$= 27777\left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4}\right]$$
$$= 6.68\ \text{m}^2$$

Step 5 of 10

Un finned surface area

$$A_{\text{unfinned}} = 1 - 27777\left(\frac{\pi D^2}{4}\right)$$
$$= 1 - 27777\left[\frac{\pi(0.0025)^2}{4}\right]$$
$$= 0.86\ \text{m}^2$$

Step 6 of 10

Heat transfer through finned surface

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}} \dot{Q}_{\text{finmax}}$$
$$= \eta_{\text{fin}} h A_{\text{fin}} (T_s - T_\infty)$$
$$= 0.959(35\ \text{W/m}^2 \cdot °C)(6.68\ \text{m}^2)(100 - 30)°C$$
$$= 15{,}700\ \text{W}$$

Step 7 of 10

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}} (T_s - T_\infty)$$
$$= (35\ \text{W/m}^2 °C)(0.86\ \text{m}^2)(100 - 30)°C$$
$$= 2107\ \text{W}$$

Step 8 of 10

Total heat transfer from the plate with fins

$$\dot{Q}_{\text{total fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}}$$
$$= 15{,}700 + 2107$$
$$= 1.78 \times 10^4\ \text{W}$$
$$= 17.8\text{kW}$$

Step 9 of 10

Rate of heat transfer when there are no fins

$$A_{\text{no fin}} = 1 \times 1$$
$$= 1\ \text{m}^2$$
$$\dot{Q}_{\text{no fin}} = hA_{\text{no fin}} (T_s - T_\infty)$$
$$= 35\ \text{W/m}^2 \cdot °C \times 1\ \text{m}^2 \times (100 - 30)°C$$
$$= 2450\ \text{W}$$

∴ The rate of heat transfer $\boxed{\dot{Q} = 2450\ \text{W}}$

Step 10 of 10

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{without fin}}}$$
$$= \frac{17800}{2450}$$
$$= 7.27$$

∴ The overall effectiveness of the fin $\boxed{\varepsilon_{\text{fin}} = 7.27}$

Question: 17.120P pg 1

Reconsider Prob. 17-118. Using EES (or other) software, investigate the effect of the center-to-center distance of the fins on the rate of heat transfer from the surface and the overall effectiveness of the fins. Let the center-to-center distance vary from 0.4 cm to 2.0 cm. Plot the rate of heat transfer and the overall effectiveness as a function of the center-to-center distance, and discuss the results.

Step 1 of 5

To solve this question using EES software open EES software

Then FILE->New

A dailogbox will open type the following in dailog box

"GIVEN"

"k_spoon=8.7 [Btu/h-ft-F], parameter to be varied"

T_w=200 "[F]"

T_infinity=75 "[F]"

A_c=0.08/12*0.5/12 "[ft^2]"

L=7" [in], parameter to be varied"

h=3 "[Btu/h-ft^2-F]"

Step 2 of 5

"ANALYSIS"

p=2*(0.08/12+0.5/12)

a=sqrt((h*p)/(k_spoon*A_c))

(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))

x=L "for tip temperature"

DELTAT=T_w-T_tip

Then from Tables -> New parametric table

A dailog box will open

Select no of runs = 20

Select variables in equations as k_{spoon} and T press ADD and then press ok

A table will appear put values of k_{spoon} as 5,16.58,28.16....225 in run 1to 20

Then from Calculate -> Solve table

Solution will appear

k_{spoon} [Btu/h.ft.F]	T [F]
5	124.9
16.58	122.6
28.16	117.8
39.74	112.5
51.32	107.1
62.89	102
74.47	97.21
86.05	92.78
97.63	88.69
109.2	84.91
120.8	81.42
132.4	78.19
143.9	75.19
155.5	72.41
167.1	69.82
178.7	67.4
190.3	65.14
201.8	63.02
213.4	61.04
225	59.17

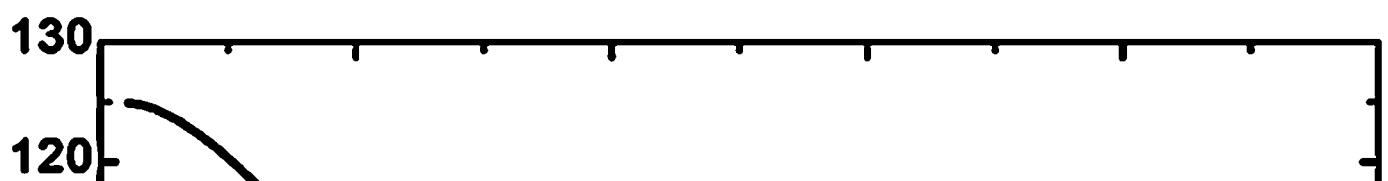
Step 3 of 5

Then from Plot -> New plot window -> X-Y Plot

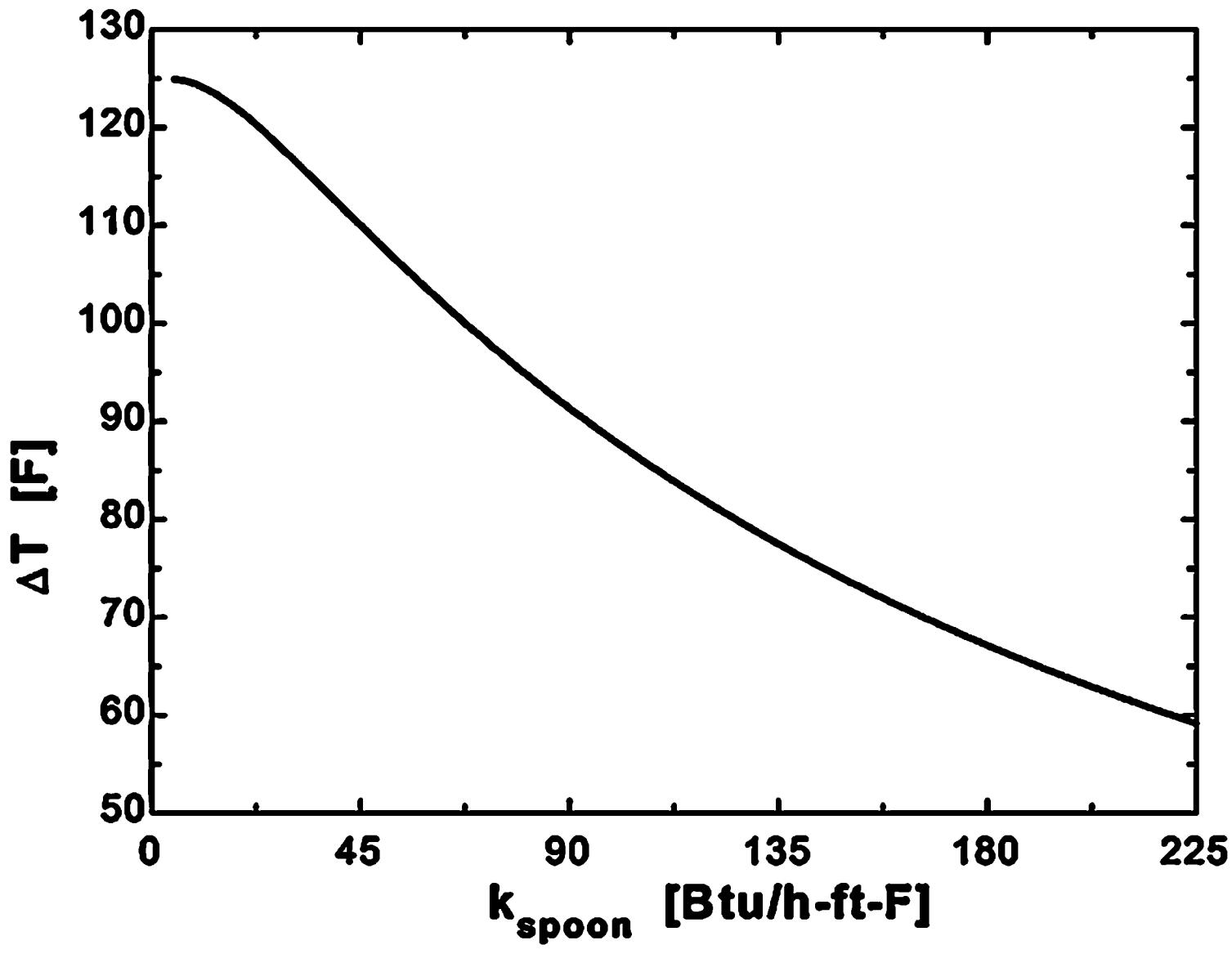
Select X-axis T_s and Y-axis h

Press OK

Graph will be plotted.



Graph will be plotted.



Step 4 of 5

Edit the text in the dialog box

"GIVEN"

k_spoon=8.7 "[Btu/h-ft-F], parameter to be varied"

T_w=200 "[F]"

T_infinity=75 "[F]"

A_c=0.08/12*0.5/12 "[ft^2]"

"L=7 [in], parameter to be varied"

h=3 "[Btu/h-ft^2-F]"

"ANALYSIS"

p=2*(0.08/12+0.5/12)

a=sqrt((h*p)/(k_spoon*A_c))

(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))

x=L "for tip temperature"

DELTAT=T_w-T_tip

Then from Tables -> New parametric table

A dailog box will open

Select no of runs = 15

Select variables in equations as L and T press ADD and then press ok

A table will appear put values of L as 5,5.5,6....12 in run 1to 15

Then from Calculate -> Solve table

Solution will appear

L [Btu/h.ft.F]	T [F]
5	122.4
5.5	123.4
6	124
6.5	124.3
7	124.6
7.5	124.7
8	124.8
8.5	124.9
9	124.9
9.5	125
10	125
10.5	125
11	125
11.5	125
12	125

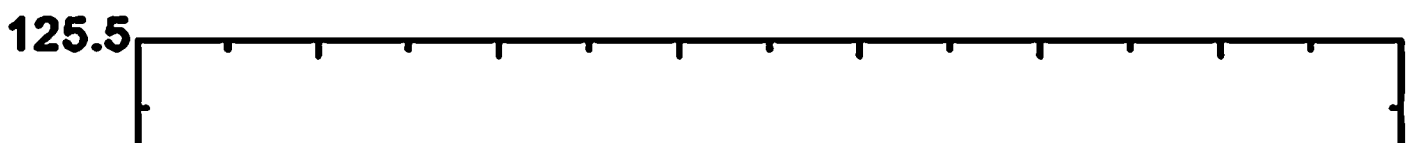
Step 5 of 5

Then from Plot -> New plot window -> X-Y Plot

Select X-axis L and Y-axis T

Press OK

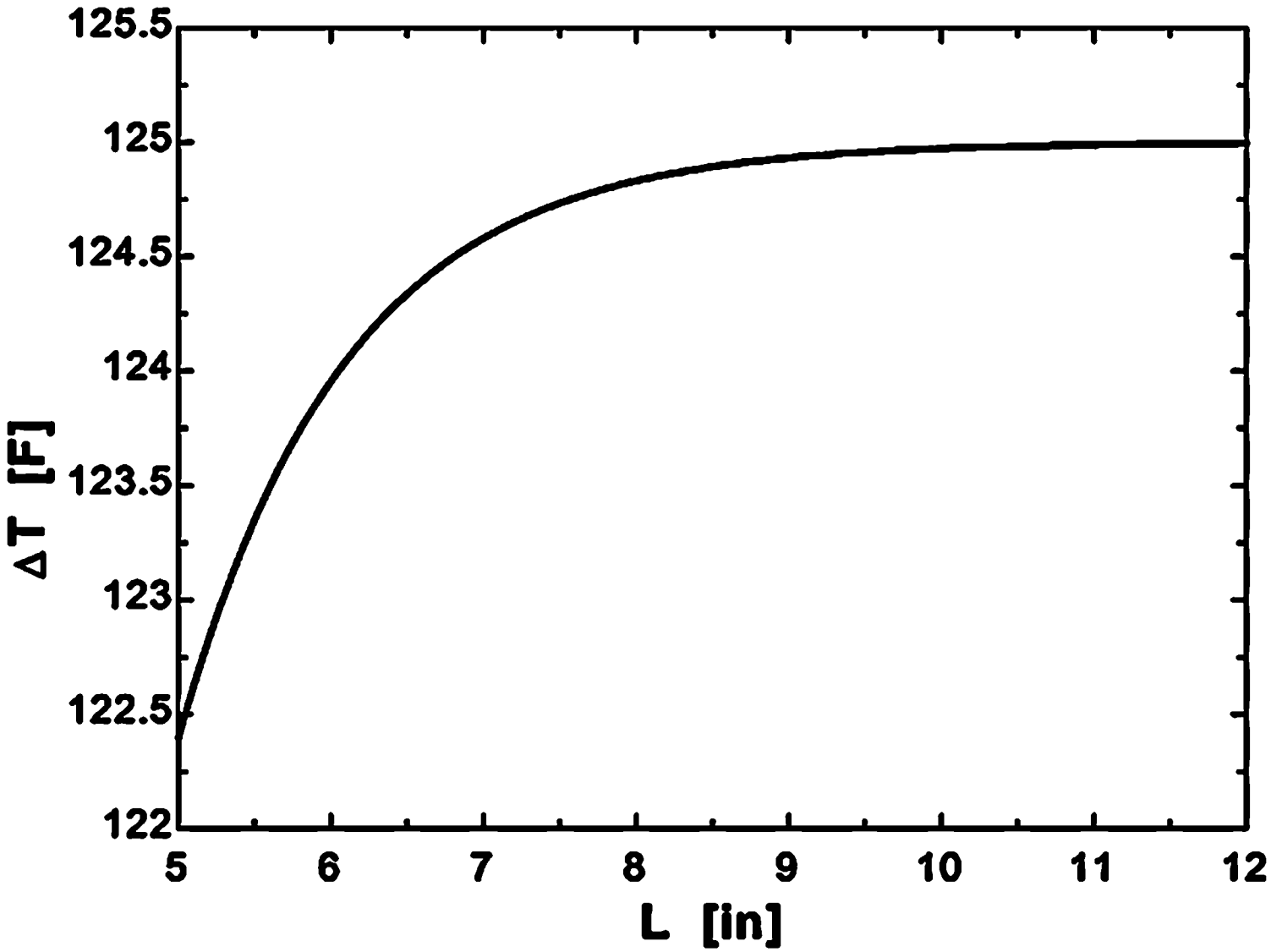
Graph will be plotted.



Press OK

Graph will be plotted.

17.120P pg 3



Question: 17.121P pg 1

Circular cooling fins of diameter D = 1 mm and length L= 30 mm, made of copper (k = 380 W/m·K), are used to enhance heat transfer from a surface that is maintained at temperature $T\infty$ = 132°C. Each rod has one end attached to this surface (x = 0), while the opposite end (x = L) is joined to a second surface, which is maintained at $Ts2$ =0°C. The air flowing between the surfaces and the rods is also at $T\infty$ = 0°C, and the convection coefficient is h = 100 W/m2·K.

(*a*) Express the function $\theta(x) = T(x) - T\infty$ along a fin, and calculate the temperature at $x = L/2$.

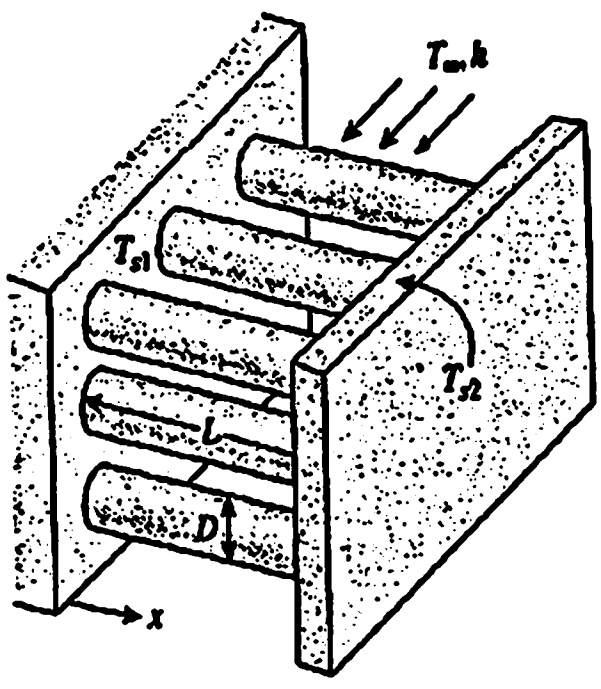


FIGURE P17–121

(*b*) Determine the rate of heal transferred from the hot surface through each fin and the fin effectiveness. Is the use of fins justified? Why?

(*c*) What is the total rate of heat transfer from a 10-cm by 10-cm section of the wall, which has 625uniformly distributed fins? Assume the same convection coefficient for the fin and for the unfinned wall surface.

Step 1 of 14

Given data:

Diameter of the fin, $D = 1\,\text{mm}$

Length of the fin, $L = 30\,\text{mm}$

Thermal conductivity of the copper plate, $k = 380\,\text{W/m}\cdot\text{K}$

Surface temperature of the copper plate, $T_{s1} = 132°\text{C}$

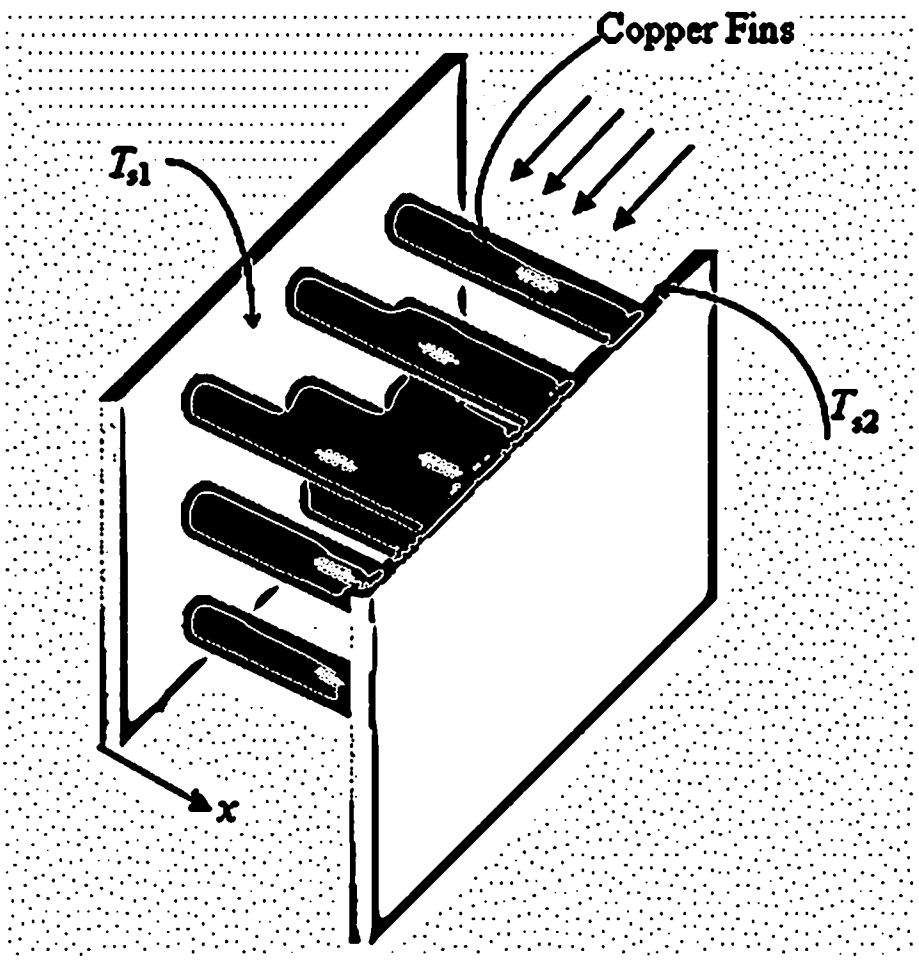
Surface temperature of the other plate, $T_{s2} = 0°\text{C}$

Ambient temperature, $T_\infty = 0°\text{C}$

Heat transfer coefficient of ambient medium, $h_\infty = 100\,\text{W/m}^2\cdot\text{K}$

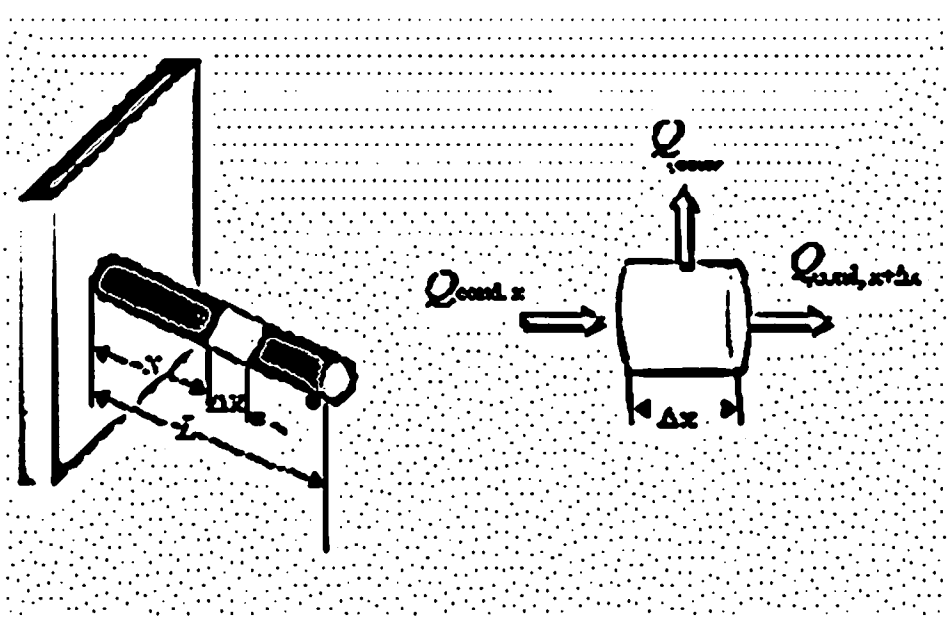
Number of fins, $n = 625$

Step 2 of 14



Step 3 of 14

a) Consider elemental volume of fin



Step 4 of 14

The fin equation can be derived by considering an elemental volume of a fin at a location x having a length of Δx, cross sectional area of A_c and a perimeter p.

Under steady conditions, the energy balance for the elemental volume can be expressed as

$$\begin{pmatrix}\text{Rate of heat}\\ \text{conduction into the}\\ \text{element at } x\end{pmatrix} = \begin{pmatrix}\text{Rate of heat}\\ \text{conduction from the}\\ \text{element at } x+\Delta x\end{pmatrix} + \begin{pmatrix}\text{Rate of heat}\\ \text{convection from the}\\ \text{element}\end{pmatrix}$$

$$\dot{Q}_{cond,x} = \dot{Q}_{cond,x+\Delta x} + \dot{Q}_{conv}$$

Where heat transfer due to convection is

$$\dot{Q}_{conv} = hp\Delta x(T - T_\infty)$$

Substituting the above expression and dividing by Δx, we will get

$$\frac{\dot{Q}_{cond,x+\Delta x} - \dot{Q}_{cond,x}}{\Delta x} + hp(T - T_\infty) = 0$$

Taking the limit as $\Delta x \to 0$ gives

$$\frac{d\dot{Q}}{}$$

Substituting the above expression and dividing by Δx, we will get

$$\frac{\dot{Q}_{cond,x+\Delta x}-\dot{Q}_{cond,x}}{\Delta x}+hp(T-T_\infty)=0$$

Taking the limit as $\Delta x \to 0$ gives

$$\frac{d\dot{Q}_{cond}}{dx}+hp(T-T_\infty)=0$$

From Fourier law of heat conduction we have

$$\dot{Q}_{cond}=-kA_c\frac{dT}{dx}$$

Substituting this relation gives the differential equation governing heat transfer in fins,

$$\frac{d}{dx}\left(kA_c\frac{dT}{dx}\right)-hp(T-T_\infty)=0$$

Step 5 of 14

For the constant cross sectional area and the constant thermal conductivity, the differential equation reduces to

$$\frac{d^2T}{dx^2}-\frac{hp}{kA_c}(T-T_\infty)=0$$

Let

$$\theta=T-T_\infty$$

$$\frac{d\theta}{dx}=\frac{dT}{dx}$$

$$\frac{d^2\theta}{dx^2}=\frac{d^2T}{dx^2}$$

And

$$m^2=\frac{hp}{kA_c}$$

Then the differential equation reduces to

$$\frac{d^2\theta}{dx^2}-m^2\theta=0$$

The general solution of the differential equation is

$$\theta(x)=C_1e^{mx}+C_2e^{-mx}$$

Step 6 of 14

In sufficiently long fin, the temperature of the fin at the tip approaches the environment temperature T_∞

At $x=L\to\infty$

$$\theta=T_\infty-T_\infty=0$$

Then

$$\theta(x)=C_1e^{mx}+C_2e^{-mx}$$

$$0=C_1e^{m\times\infty}+C_2e^{-m\times\infty}$$

$$C_1=0$$

At fin base $x=0$

Temperature at the base is T_b

$$\theta_b=T_b-T_\infty$$

Then

$$\theta_b=C_1+C_2$$

$$\theta_b=0+C_2$$

$$C_2=\theta_b$$

Then substituting the constants the general solution of the differential equation is

$$\theta(x)=C_1e^{mx}+C_2e^{-mx}$$

$$\theta(x)=\theta_be^{-mx}$$

Step 7 of 14

Therefore, for a very long fin the temperature distribution can be expressed as

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=e^{-mx}$$

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=e^{-x\sqrt{hp/kA_c}}$$

Therefore,

$$T(x)-T_\infty=(T_b-T_\infty)e^{-x\sqrt{hp/kA_c}}$$

$$\theta(x)=\theta_be^{-x\sqrt{hp/kA_c}}$$

$$\boxed{\theta(x)=\theta_be^{-x\sqrt{hp/kA_c}}}$$

Step 8 of 14

The temperature at $x=L/2$ can be determined as

$$m=\sqrt{\frac{hp}{kA_c}}$$

$$=\sqrt{\frac{h\pi D}{k\frac{\pi}{4}D^2}}$$

$$=\sqrt{\frac{4h}{kD}}$$

$$=\sqrt{\frac{100\times4}{380\times0.001}}$$

$$=32.44(1/\text{m})$$

The temperature on the fin at $x=L/2$ can be determined as

$$T(x)-T_\infty=(T_b-T_\infty)e^{-x\sqrt{hp/kA_c}}$$

$$T(L/2)=T_\infty+(T_b-T_\infty)e^{-\frac{L}{2}\sqrt{hp/kA_c}}$$

$$=0+(132-0)\times e^{-\frac{0.08}{2}\times32.44}$$

$$=81.142^\circ\text{C}$$

$$\boxed{T(L/2)=81.142^\circ\text{C}}$$

Therefore the temperature at $x=L/2$ is $T(L/2)=81.142^\circ\text{C}$

Step 9 of 14

$\boxed{T(L/2)=81.142^{\circ}C}$

Therefore the temperature at $x=L/2$ is $T(L/2)=81.142^{\circ}C$

Step 9 of 14

b) The rate of heat transfer from the entire fin can be determined using the Fourier's law of heat conduction

$$\dot{Q}_{long\,fin}=-kA_c\frac{dT}{dx}$$

$$=\sqrt{hpkA_c}(T_b-T_\infty)$$

$$=\sqrt{100\times\pi\times0.001\times380\times\frac{\pi}{4}\times(0.001)^2}\times(132-0)$$

$$=1.2782\,W$$

$\boxed{\dot{Q}_{long\,fin}=1.2782\,W}$

Step 10 of 14

The effectiveness of the fin can be expressed as

$$\varepsilon_{fin}=\frac{\dot{Q}_{fin}}{\dot{Q}_{no\,fin}}$$

$$=\frac{\dot{Q}_{fin}}{h_\infty A_b(T_b-T_\infty)}$$

Where A_b represents the cross sectional area of the fin at the base

And $\dot{Q}_{no\,fin}$ represents the rate of heat transfer from the base area if no fins are attached to the surface

Base area A_b can be expressed as

$$A_b=\frac{\pi}{4}D^2$$

$$=\frac{\pi}{4}\times(0.001)^2$$

$$=7.854\times10^{-7}\,m^2$$

Step 11 of 14

Therefore, the effectiveness of the fin can be calculated as

$$\varepsilon_{fin}=\frac{\dot{Q}_{fin}}{\dot{Q}_{no\,fin}}$$

$$=\frac{\dot{Q}_{fin}}{h_\infty A_b(T_b-T_\infty)}$$

$$=\frac{1.2782}{100\times7.854\times10^{-7}\times(132-0)}$$

$$=123.292$$

$\boxed{\varepsilon_{fin}=123.292}$

Since $\varepsilon_{fin}>1$ it indicates that fins are enhancing the heat transfer from the surface and it is justifiable.

The rate of heat transfer from the hot surface through each fin is $\dot{Q}_{long\,fin}=1.2782\,W$ and fin effectiveness is $\varepsilon_{fin}=123.292$

Step 12 of 14

c)

The total fin area can be determined as

$$A_{finned}=n\pi DL$$

$$=625\times\pi\times0.001\times0.03$$

$$=0.0589\,m^2$$

The unfinned area can be calculated as

$$A_{unfinned}=A-864\frac{\pi}{4}D^2$$

$$=(0.1\times0.1)-625\times\frac{\pi}{4}\times0.001^2$$

$$=0.0095091\,m^2$$

The total area exposed to the ambient medium can be calculated as

$$A_{total}=A_{finned}+A_{unfinned}$$

$$=0.0589+0.0095091$$

$$=0.0684\,m^2$$

Step 13 of 14

Thermal resistance due to convection

$$R_{conv}=\frac{1}{h_\infty A}$$

$$=\frac{1}{100\times0.0684}$$

$$=0.1462\,K/W$$

Step 14 of 14

The steady total rate of heat transfer from the wall can be expressed as

$$\dot{Q}_{total}=\frac{(T_{s1}-T_\infty)}{R_{total}}$$

$$=\frac{(132-0)}{0.1462}$$

$$=902.87\,W$$

$\boxed{\dot{Q}_{total}=902.87\,W}$

Therefore the total rate of heat transfer from a 10-cm by 10-cm section of the wall is

$\dot{Q}_{total}=902.87\,W$

$\boxed{\dot{Q}_{total} = 902.87\,\text{W}}$

Therefore the total rate of heat transfer from a 10-cm by 10-cm section of the wall is

$\dot{Q}_{total} = 902.87\,\text{W}$

Question: 17.122P

Steam in a heating system flows through tubes whoseouter diameter is 3 cm and whose walls are maintained at a temperature of 120T. Circular aluminum alloy fins (k = 180 W/m·K) of outer diameter 6 cm and constant thickness t = 2 mm are attached to the tube, as shown in Fig. P17–i22. The space between the fins is 3 mm, and thus there are 200 fins per meter length of the tube. Heat is transferred to the surrounding air at 25°C, with a combined heat transfer coefficient of60 W/m2·K. Determine the increase in heat transfer from the tube per meter of its length as a result of adding fins.

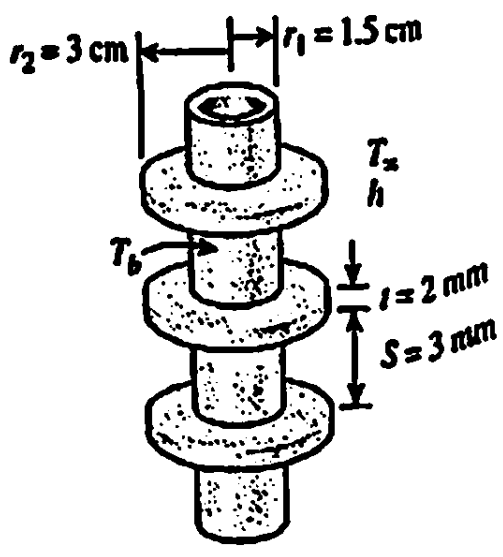


FIGURE P17–122

Step 1 of 6

Given data:

Outer radius of the pipe, $r_1 = 1.5\,\text{cm}$

Thickness of the fin, $t = 2\,\text{mm}$

Thermal conductivity of the fin material, $k = 180\,\text{W/m·K}$

Outer radius of the fin, $r_f = 3\,\text{cm}$

Temperature of the pipe at the outer surface, $T_b = 120°\text{C}$

Ambient air temperature, $T_\infty = 25°\text{C}$

Heat transfer coefficient of ambient air, $h_\infty = 60\,\text{W/m}^2\cdot\text{K}$

Number of fins per meter length of the tube, $n = 200$

Step 2 of 6

Efficiency of annular fins can be calculated by finding these parameters

$$r_{2c} = r_f + \frac{t_f}{2}$$
$$= 0.03 + \frac{0.002}{2}$$
$$= 0.031\,\text{m}$$
$$A_{fin} = 2\pi\left(r_{2c}^2 - r_1^2\right)$$
$$= 2\pi \times (0.031^2 - 0.015^2)$$
$$= 0.0046244\,\text{m}^2$$
$$L_c = L + \frac{t_f}{2}$$
$$= r_f - r_1 + \frac{t_f}{2}$$
$$= 0.03 - 0.015 + \frac{0.002}{2}$$
$$= 0.016\,\text{m}$$

Step 3 of 6

Hence,

$$A_p = L_c t_f$$
$$= 0.016 \times 0.002$$
$$= 0.000032\,\text{m}^2$$
$$\xi = L_c^{3/2}\left(h/kA_p\right)^{1/2}$$
$$= (0.016)^{3/2} \times \left(\frac{60}{180 \times 0.000032}\right)^{1/2}$$
$$= 0.20656$$
$$\frac{r_{2c}}{r_1} = \frac{0.031}{0.015}$$
$$= 2.0667$$

Step 4 of 6

From the graph efficiency of annular fins of constant thickness t at $\xi = 0.20656$ and $\frac{r_{2c}}{r_1} = 2.0667$ the efficiency are

$$\eta_{fin} = 0.955$$

The rate of heat transfer from the fin can be expressed as

$$\dot{Q}_{fin} = \eta_{fin} h A_{fin} (T_s - T_\infty)$$
$$= 0.955 \times 60 \times 0.0046244 \times (120 - 25)$$
$$= 25.173\,\text{W}$$

Since there are 200 fins per meter length of the tube then the total heat transfer rate from the fins can be expressed as

$$\dot{Q}_{fins} = n \times \dot{Q}_{fins}$$
$$= 200 \times 25.173$$
$$= 5034.6\,\text{W}$$

Step 5 of 6

The heat transfer from the base area if no fins are attached to the surface can be calculated as

$$\dot{Q}_{no\,fin} = h_\infty A_b (T_b - T_\infty)$$
$$A_b = 2\pi r_1 t$$
$$= 2 \times \pi \times 0.015 \times 0.002$$
$$= 0.0001885\,\text{m}^2$$
$$\dot{Q}_{no\,fin} = h_\infty A_b (T_b - T_\infty)$$
$$= 60 \times 0.0001885 \times (120 - 25)$$
$$= 1.0744\,\text{W}$$

For 200 fins, the total heat transfer rate from the fins base area becomes

$$\dot{Q}_{no\,fins} = n \times \dot{Q}_{no\,fin}$$
$$= 200 \times 1.0744$$
$$= 214.885\,\text{W}$$

Step 6 of 6

Therefore, the increase in heat transfer due to addition of fins per meter length of the tube can be calculated as

$$\Delta\dot{Q}_{increase} = \dot{Q}_{fins} - \dot{Q}_{no\,fins}$$
$$= 5034.6 - 214.885$$
$$= 4819.715\,\text{W}$$

$$\boxed{\Delta\dot{Q}_{increase} = 4819.715\,\text{W}}$$

Question: 17.123P

Steam is produced in the copper tubes (k = 223 Btu/h·ft·°F) of a heat exchanger at a temperature of 280°F by another fluid condensing on the outside surfaces of the tubes at 350°F. The inner and outer diameters of the tube are 1 in and 1.3 in, respectively. When the heat exchanger was new, the rate of heat transfer per foot length of the tube was 2 × 104 Btu/h. Determine the rate of heat transfer per foot length of the tube when a 0.01-in-thick layer of limestone (k = 1.7 Btu/h·°F) has formed on the inner surface of the tube after extended use.

Step 1 of 4

Given data:

Outer diameter of the pipe, $d_1 = 1.3\text{in}$

Inner diameter of the pipe, $d_2 = 1\text{in}$

Thickness of the limestone layer, $t = 0.01\text{in}$

Thermal conductivity of the copper tubes, $k = 223\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

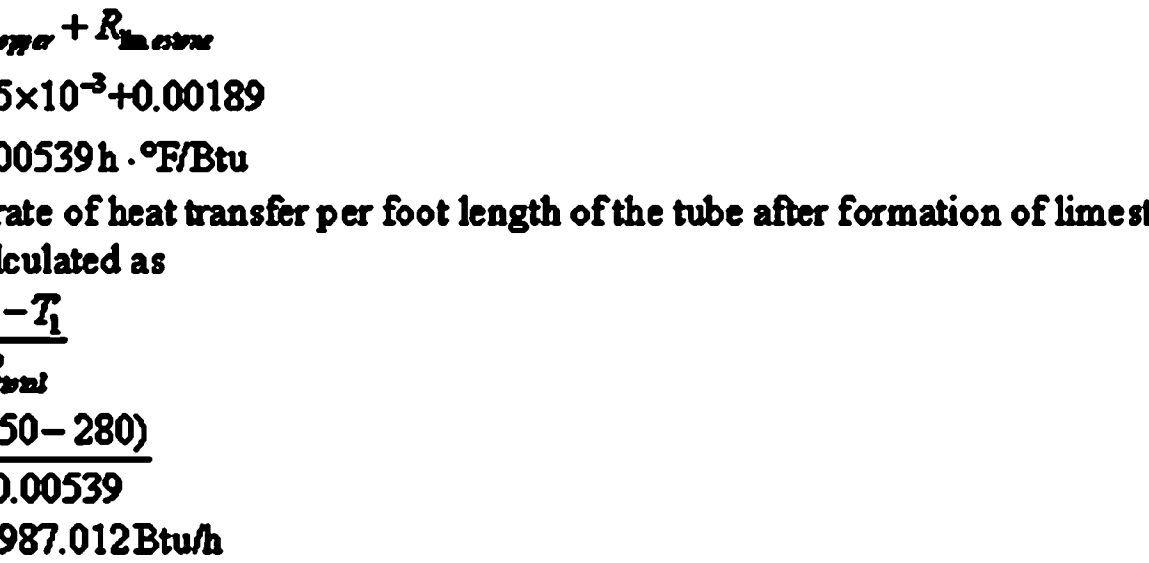
Thermal conductivity of the limestone, $k = 1.7\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

Temperature of the steam in copper tubes, $T_1 = 280^\circ\text{F}$

Condensing fluid temperature, $T_2 = 350^\circ\text{F}$

The rate of heat transfer per foot length of the tube, $\dot{Q} = 2\times10^4\ \text{Btu/h}$

Step 2 of 4



Step 3 of 4

The total thermal resistance of the new heat exchanger can be determined using the Fourier law of heat conduction equation, which is given by

$$\dot{Q}_{copper} = \frac{T_2 - T_1}{R_{copper}}$$

Therefore, the total new resistance can be calculated as

$$\begin{aligned} R_{copper} &= \frac{T_2 - T_1}{\dot{Q}_{new}} \\ &= \frac{(350-280)}{2\times10^4} \\ &= 3.5\times10^{-3}\ \text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

Step 4 of 4

If there is 0.01 in thick layer of limestone forms on the inner surface of the copper pipes, the total value of thermal resistance and heat transfer rate are determined to be

$$\begin{aligned} R_{limestone} &= \frac{\ln(r_2/r_3)}{2\pi kL} \\ &= \frac{\ln[0.5/(0.5-0.01)]}{2\times\pi\times1.7\times1} \\ &= 0.00189\ \text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

Therefore, the total resistance becomes

$$\begin{aligned} R_{total} &= R_{copper} + R_{limestone} \\ &= 3.5\times10^{-3} + 0.00189 \\ &= 0.00539\ \text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

The new rate of heat transfer per foot length of the tube after formation of limestone layer can be calculated as

$$\begin{aligned} \dot{Q}_{new} &= \frac{T_2 - T_1}{R_{total}} \\ &= \frac{(350-280)}{0.00539} \\ &= 12987.012\ \text{Btu/h} \end{aligned}$$

$$\boxed{\dot{Q}_{new} = 12987.012\ \text{Btu/h}}$$

Since the limestone layer forming on the inner surface of the copper pipes it will change the inner surface area of the pipe and thus the internal convection resistance slightly and hence reduced heat transfer to the steam formation.

Question: 17.124P

Repeat Prob. 17-123E, assuming that a 0.01-in-thick limestone layer has formed on both the inner and outer surfaces of the tube.

Step 1 of 5

Given data:

Outer diameter of the pipe, $d_1 = 1.3\text{in}$

Inner diameter of the pipe, $d_2 = 1\text{in}$

Thickness of the limestone layer on both sides of the pipe, $t = 0.01\text{in}$

Thermal conductivity of the copper tubes, $k = 223\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

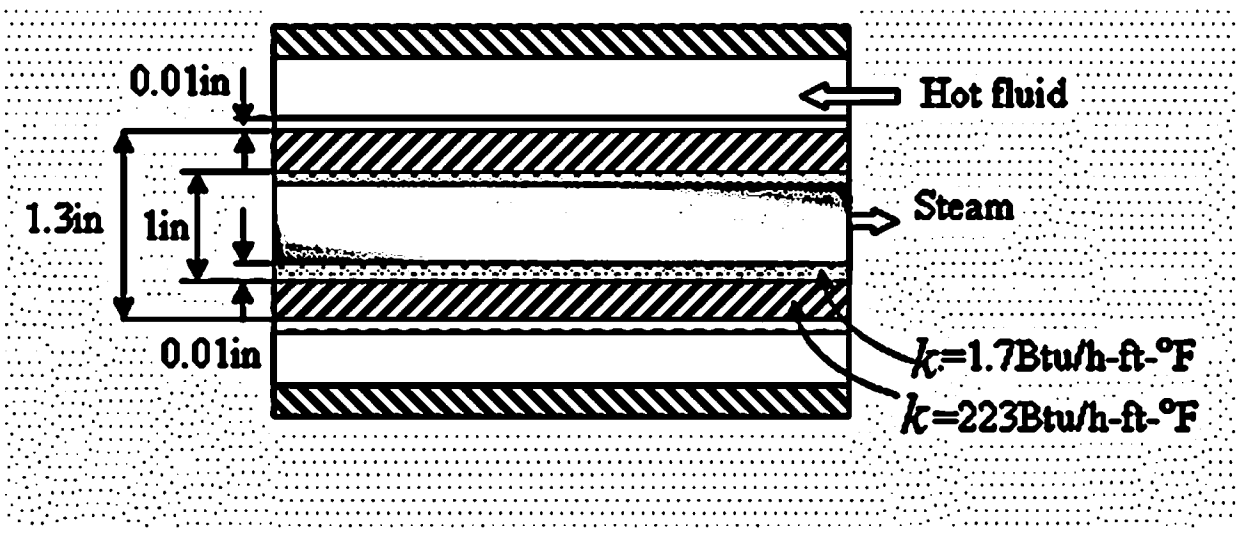
Thermal conductivity of the limestone, $k = 1.7\,\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

Temperature of the steam in copper tubes, $T_1 = 280^\circ\text{F}$

Condensing fluid temperature, $T_2 = 350^\circ\text{F}$

The rate of heat transfer per foot length of the tube, $\dot{Q} = 2\times10^4\ \text{Btu/h}$

Step 2 of 5



Step 3 of 5

The total thermal resistance of the new heat exchanger can be determined using the Fourier law of heat conduction equation, which is given by

$$\dot{Q}_{copper} = \frac{T_2 - T_1}{R_{copper}}$$

Therefore, the total new resistance can be calculated as

$$\begin{aligned} R_{copper} &= \frac{T_2 - T_1}{\dot{Q}_{new}} \\ &= \frac{(350-280)}{2\times10^4} \\ &= 3.5\times10^{-3}\ \text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

Step 4 of 5

After 0.01 in thick layer of limestone forms on the inner and outer surfaces of the copper pipes, the total value of thermal resistance and heat transfer rate are determined to be

$$\begin{aligned} R_{limestone,i} &= \frac{\ln(r_2/r_i)}{2\pi kL} \\ &= \frac{\ln\left[0.5/(0.5-0.01)\right]}{2\times\pi\times1.7\times1} \\ &= 0.00189\,\text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

$$\begin{aligned} R_{limestone,o} &= \frac{\ln(r_o/r_1)}{2\pi kL} \\ &= \frac{\ln\left[(0.65+0.01)/0.65)\right]}{2\times\pi\times1.7\times1} \\ &= 1.42935\times10^{-3}\,\text{h-}^\circ\text{F/Btu} \end{aligned}$$

Therefore, the total resistance becomes

$$\begin{aligned} R_{total} &= R_{limestone,i} + R_{copper} + R_{limestone,o} \\ &= 0.00189 + 3.5\times10^{-3} + 1.42935\times10^{-3} \\ &= 0.00681\,\text{h}\cdot{}^\circ\text{F/Btu} \end{aligned}$$

Step 5 of 5

The new rate of heat transfer per foot length of the tube after formation of limestone layer can be calculated as

$$\begin{aligned} \dot{Q}_{new} &= \frac{T_2 - T_1}{R_{total}} \\ &= \frac{(350-280)}{0.00681} \\ &= 10279.0014\,\text{Btu/h} \end{aligned}$$

$$\boxed{\dot{Q}_{new} = 10279.0014\,\text{Btu/h}}$$

Since the limestone layer forming on the inner and outer surfaces of the copper pipes it will change the inner and outer surface areas of the pipe and thus the internal and external convection resistance slightly and hence reduced heat transfer to the steam formation.

Question: 17.125P

Hot water is flowing at an average velocity of 1.5 m/s through a cast iron pipe (k = 52 W/m·K) whose inner and outer diameters are 3 cm and 3.5 cm. respectively. The pipe passes through a 15-m-long section of a basement whose temperature is 15°C. If the temperature of the water drops from 70°C to 67°C as it passes through the basement and the heat transfer coefficient on the inner surface of the pipe is 400 W/m2-K. determine the combined convection and radiation heat transfer coefficient at the outer surface of the pipe.

Step 1 of 5

Given: -

Average velocity of water = 1.5 m/s

Thermal conductivity of cast iron pipe (k) = 52 W/m.°C

Inner diameter of pipe = 3 cm

Outer diameter of pipe = 3.5 cm

Pipe passes through section of basement 15 m long at temperature of T_2 =15°C

Temperature drop of water from 70°C to 67°C

Heat transfer coefficient h= 400 W/m^2°C

Step 2 of 5

Mass flow rate of water

$$\dot{m} = \rho V A_c$$
$$= (1000)(1.5)\left[\pi(0.03)^2/4\right]$$
$$\dot{m} = 1.06028 \text{ kg/s}$$

Heat transfer rate from the water can be calculated as

$$\dot{Q} = \dot{m} C_p \Delta T$$
$$= (1.06028)(4180)(70-67)$$
$$\dot{Q} = 13295.91 \text{ W}$$

Step 3 of 5

Thermal resistance of pipe and for convection through pipe is

$$R_p = \frac{\ln(r_2/r_1)}{2\pi kL}$$
$$= \frac{\ln(1.75/1.5)}{2\pi(52)\times(15)}$$
$$R_p = 0.000031 \text{ °C/W}$$

$$R_c = \frac{1}{h_i A_i}$$
$$= \frac{1}{(400)[\pi(0.03)(15)]}$$
$$R_c = 0.001768 \text{ °C/W}$$

$$R_{combine} = \frac{1}{h_{combine} A_o}$$
$$= \frac{1}{h_{combine}[\pi(0.035)(15)]}$$
$$= \frac{1}{h_{combine}[1.649]} \text{°C/W}$$

Step 4 of 5

Total thermal resistance can be calculated as

$$R_{total} = R_p + R_c + R_{combine}$$
$$= 0.000031 + 0.001768 + \frac{1}{h_{combine}[1.649]}$$
$$R_{total} = 0.001799 + \frac{1}{h_{combine}[1.649]} \text{°C/W}$$

Step 5 of 5

Mean temperature difference of water is

$$T_{AMTD} = \frac{70+67}{2}$$
$$T_{AMTD} = 68.5^0\text{C}$$

Rate of heat transfer can be expressed as

$$\dot{Q} = \frac{T_{AMTD} - T_2}{R_{total}}$$
$$13295.91 = \frac{(68.5-15)}{\left(0.001799 + \frac{1}{h_{combine}[1.649]}\right)}$$
$$\left(0.001799 + \frac{1}{h_{combine}[1.649]}\right) = 0.004023$$
$$\frac{1}{h_{combine}[1.649]} = 0.002224$$
$$\boxed{h_{combine} = 272.67 \text{ W/m}^2.^0\text{C}}$$

Question: 17.126P

Newly formed concrete pipes are usually cured first overnight by steam in a curing kiln maintained at a temperature of 45°C before the pipes are cured for several days outside. The heat and moisture to the kiln is provided by steam (lowing in a pipe whose outer diameter is 12 cm. During a plant inspection, it was noticed that the pipe passes through a g-m section that is completely exposed to the ambient air before it reaches the kiln. The temperature measurements indicate that the average temperature of the outer surface of the steam pipe is 90°C when the ambient temperature is 8°C. The combined convection and radiation heat transfer coefficient at the outer surface of the pipe is estimated to be 35 W/m2-K. Determine the amount of heat lost from the steam during a 10-h curing process that night.

Steam is supplied by a gas-fired steam generator that has an efficiency of 85 percent, and the plant pays $ 1.20/therm of natural gas (1 therm = 105,500 kJ). If the pipe is insulated and 90 percent of the heat loss is saved as a result, determine the amount of money this facility will save a year as a result of insulating the steam pipes. Assume that the concrete pipes are cured 110 nights a year. State your assumptions.

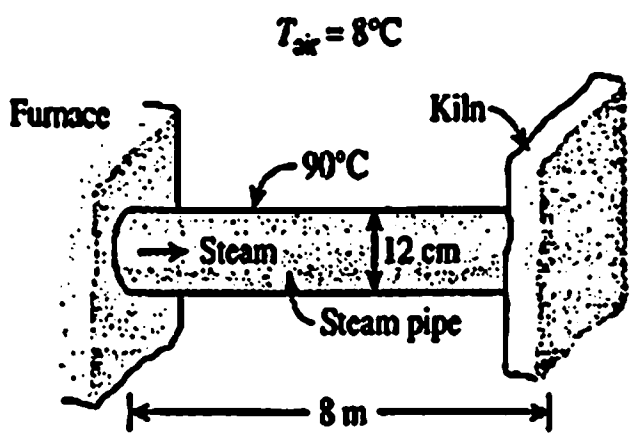


FIGURE P17–126

Step 1 of 3

Given data:

Diameter of the pipe, $D = 12\,\text{cm}$

Temperature of the kiln, $T_i = 45°C$

Length of the pipe, $L = 8\,\text{m}$

Temperature of the pipe at the outer surface, $T_s = 90°C$

Ambient air temperature, $T_\infty = 8°C$

Combined convection and radiation Heat transfer coefficient of air, $h_\infty = 35\,\text{W/m}^2\cdot\text{K}$

Efficiency of the steam generator, $\eta = 85\%$

Unit cost of fuel \$1.20/ therm

1therm = 105,500kJ

Step 2 of 3

Assumptions:

1. Heat transfer through the pipe is steady and one-dimensional.
2. Thermal conductivities are constant.
3. The furnace operates continuously.
4. The given heat transfer coefficients accounts for the radiation effects.
5. The temperatures of the pipe surface and the surroundings are representative of annual average during operating hours.
6. The plant operates 110 days a year.

Step 3 of 3

The surface area of the steam pipe can be determined as

$$\begin{aligned} A_s &= \pi DL \\ &= \pi \times 0.12 \times 8 \\ &= 3.016\,\text{m}^2 \end{aligned}$$

The rate of heat transfer from the pipe can be determined as

$$\begin{aligned} \dot{Q} &= hA_s(T_s - T_\infty) \\ &= 35 \times 3.016 \times (90 - 8) \\ &= 8655.92\,\text{W} \end{aligned}$$

The amount of heat loss during a 10-hour period can be calculated as

$$\begin{aligned} Q &= \dot{Q}\Delta t \\ &= 8655.92 \times 10 \times 3600 \\ &= 3.116 \times 10^5\,\text{kJ (per day)} \end{aligned}$$

Question: 17.127P

Consider an 18-cm × 18-cm multilayer circuit board dissipating 27 W of heat. The board consists of four layers of 0.2-mm-thick copper (k =386 W/m·K) and three layers of 1.5-mm-thick epoxy glass (k = 0.26 W/m·K) sandwiched together, as shown in the figure. The circuit board is attached to a heat sink from both ends, and the temperature of the board at those ends is 35°C. Heat is considered to be uniformly generated in the epoxy layers of the board at a rate of 0.5 W per 1 -cm × 18-cm epoxy laminate strip (or 3.5 W per 1-cm × 18-cm strip of the board). Considering only a portion of the board because of symmetry, determine the magnitude and location of the maximum temperature that occurs in the board. Assume heat transfer from the top and bottom faces of the board to be negligible.

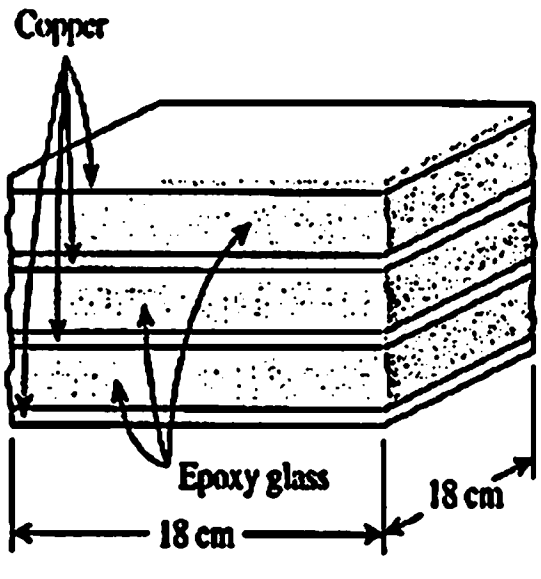


FIGURE P17–127

Step 1 of 3

Given: -

Area of circuit board = 18 cm × 18 cm

Heat dissipated by board Q= 27 W

Number of layers of copper = 4

Number of layers of epoxy glass = 3

Temperature of heat sink T_2= 35C.

Thickness of copper layer = 0.2 mm

Thickness of epoxy layer = 1.5 mm

Thermal conductivity of copper (k_{copper}) = 386 W/m°C

Thermal conductivity (k_{epoxy}) = 0.26 W/m°C

Step 2 of 3

Effective thermal conductivity of circuit board can be calculated as

$$(kt)_{copper} = 4[(386)(0.0002)]$$

$$(kt)_{copper} = 0.3088 \text{ W/°C}$$

$$(kt)_{epoxy} = 3[(0.26)(0.0015)]$$

$$(kt)_{epoxy} = 0.00117 \text{ W/°C}$$

$$k_{eff} = \frac{(kt)_{copper} + (kt)_{epoxy}}{t_{copper} + t_{epoxy}}$$

$$= \frac{(0.3088 + 0.00117)}{[4(0.0002) + 3(0.0015)]}$$

$$k_{eff} = 58.48 \text{ W/m.°C}$$

Step 3 of 3

Maximum temperature can be calculated as

$$A = 0.18[4(0.0002) + 3(0.0015)]$$

$$A = 0.000954 \text{ m}^2$$

$$\dot{Q} = \frac{k_{eff} A}{L}(T_{max} - T_2)$$

Because of the symmetry of the plane taking $\dot{Q} = \dot{Q}/2$ and $L = L/2$

$$T_{max} = T_2 + \frac{\dot{Q}L}{k_{eff} A}$$

$$T_{max} = 35 + \frac{(27/2)(0.18/2)}{(58.48)(0.000954)}$$

$$\boxed{T_{max} = 56.8^0\text{C}}$$

Question: 17.128P

The surface temperature of a 3-in-diameter baked potato is observed to drop from 300°F to 200°F in 5 min in an environment at 70°F. Determine the average heat transfer coefficient between the potato and its surroundings. Using this heat transfer coefficient and the same surface temperature, determine how long it will take for the potato to experience the same temperature drop if it is wrapped completely in a 0.12-in-thick towel (k =0.035 Btu/h·ft·°F), You may use the properties of water for potato.

Step 1 of 11

Given: -

Diameter of potato $d = 3\text{in}$

Radius of potato $r = 1.5\text{in}$

Temperature drop of baked potato from 300°F to 200°F

Time taken for temperature drop of potato is t = 5 minutes

Temperature of surrounding $T_\infty = 70°\text{F}$

Thermal conductivity of towel (k) = 0.035 Btu/h·ft·°F

Thickness of towel $t = 0.12\text{in}$

Step 2 of 11

Taking properties of water for potato

Density (ρ) = 62.2 lbm/ft^3

Specific heat at constant pressure (C_p) = 0.998 Btu/lbm·°F

Step 3 of 11

Assuming constant average temperature of potato

Mass of potato will be

$$m = \rho V$$

$$= \rho \frac{4}{3}\pi r^3$$

$$= (62.2)\frac{4}{3}\pi(1.5/12)^3$$

$$m = 0.5089\ \text{lbm}$$

Step 4 of 11

Heat lost from potato

$$Q = mC_p\Delta T$$

$$= (0.5089)(0.998)(300\text{-}200)$$

$$Q = 50.8\ \text{Btu}$$

Step 5 of 11

Heat transfer rate

$$\dot{Q} = \frac{Q}{\Delta t}$$

$$= \frac{50.8\ \text{Btu}}{(5/60\ \text{h})}$$

$$\dot{Q} = 609.6\ \text{Btu/h}$$

Step 6 of 11

Coefficient of heat transfer between surrounding and potato is

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$h = \frac{\dot{Q}}{A_s(T_s - T_\infty)}$$

$$= \frac{609.6}{\pi(3/12)^2(250-70)}$$

$$\boxed{h = 17.248\ \text{Btu/h.ft}^2.°\text{F}}$$

Step 7 of 11

If the potato is wrapped completely by 0.12 in thick towel then

Outer diameter will be $d_2 = 3+0.12+0.12 = 3.24\text{in}$

Or outer radius will be $r_2 = (1.5+0.12)\text{in}$

Thermal resistance due to conduction is

$$R_t = \frac{r_2 - r_1}{4\pi k r_1 r_2}$$

$$= \frac{[(1.5+0.12)/12]-(1.5/12)}{4\pi(0.035)[(1.5+0.12)/12](1.5/12)}$$

$$= \frac{[0.135]-(0.125)}{4\pi(0.035)[0.135](0.125)}$$

$$R_t = 1.3473\ \text{h°F/Btu}$$

Step 8 of 11

Thermal resistance due to convection from outer surface

$$R_c = \frac{1}{hA}$$

$$= \frac{1}{(17.248)\pi\left[(3+0.24)/12\right]^2}$$

$$R_c = 0.2531\ \text{h.°F/Btu}$$

Step 9 of 11

Total thermal resistance due to conduction and convection

$$R_{total} = R_t + R_c$$

$$= 1.3473+0.2531$$

$$R_{total} = 1.6004\ \text{h°F/Btu}$$

Step 10 of 11

Rate of heat transfer

$$\dot{Q} = \frac{T_s - T_\infty}{R_{total}}$$

Where average surface temperature of potato will be

$$T_s = \left(\frac{300+200}{2}\right)$$

$$T_s = 250^0\text{F}$$

Step 11 of 11

Then

$$\dot{Q} = \frac{(250-70)}{1.6004}$$

$$= 112.47\ \text{Btu/h}$$

Time taken for the potato to experience same temperature drop when wrapped in 0.12-in thick towel

$$\Delta t = \frac{Q}{\dot{Q}}$$

$$= \frac{50.8}{112.47}$$

$$= 0.452\ \text{h}$$

$$\boxed{\Delta t = 27.12\ \text{min}}$$

Question: 17.129P

Repeat Prob. 17-128E assuming there is a 0.02-in-thick air space (k = 0.015 Btu/h·ft·°F) between the potato and the towel.

Step 1 of 12

Given: -

Diameter of potato $d = 3\text{in}$

Radius of potato $r = 1.5\text{in}$

Temperature drop of baked potato from 300°F to 200°F

Time taken for temperature drop of potato is t = 5 minutes

Temperature of surrounding $T_\infty = 70°\text{F}$

Thermal conductivity of towel (k_{towel}) = 0.035 Btu/h·ft·°F

Thickness of towel $t = 0.12\text{in}$

Thermal conductivity of air $k_{air} = 0.015\text{Btu/h.ft.}^0\text{F}$

Thickness of air space $t_{air} = 0.02\text{in}$

Step 2 of 12

Taking properties of water for potato

Density (ρ) = 62.2 lbm/ft^3

Specific heat at constant pressure (C_p) = 0.998 Btu/lbm·°F

Step 3 of 12

Assuming constant average temperature of potato

Mass of potato will be

$$m = \rho V$$
$$= \rho\frac{4}{3}\pi r^3$$
$$= (62.2)\frac{4}{3}\pi(1.5/12)^3$$
$$m = 0.5089\text{ lbm}$$

Step 4 of 12

Heat lost from potato

$$Q = mC_p\Delta T$$
$$= (0.5089)(0.998)(300\text{-}200)$$
$$Q = 50.8\text{ Btu}$$

Step 5 of 12

Heat transfer rate

$$\dot{Q} = \frac{Q}{\Delta t}$$
$$= \frac{50.8\text{ Btu}}{(5/60\text{ h})}$$
$$\dot{Q} = 609.6\text{ Btu/h}$$

Step 6 of 12

Coefficient of heat transfer between surrounding and potato is

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$h = \frac{\dot{Q}}{A_s(T_s - T_\infty)}$$
$$= \frac{609.6}{\pi(3/12)^2(250-70)}$$
$$h = 17.248\text{Btu/h.ft}^2\text{.°F}$$

Step 7 of 12

Total thermal resistance for the system is

Outer radius of the air gap is $r_2 = (1.5+0.02)\text{in}$

Thermal resistance due to air gap will be

$$R_{air} = \frac{r_2 - r_1}{4\pi k r_1 r_2}$$
$$= \frac{[(1.50+0.02)/12] - (1.50/12)}{4\pi(0.015)[(1.50+0.02)/12](1.50/12)}$$
$$= \frac{0.001666}{0.00298}$$
$$R_{air} = 0.55906\text{ h.°F/Btu}$$

Step 8 of 12

Outer radius of the towel is $r_3 = (1.5+0.02+0.12)\text{in}$

Then thermal resistance between air gap and towel will be

$$R_t = \frac{r_3 - r_2}{4\pi k r_2 r_3}$$
$$= \frac{[(1.52+0.12)/12] - (1.52/12)}{4\pi(0.035)[(1.52+0.12)/12](1.52/12)}$$
$$= \frac{0.01}{0.007613}$$
$$R_t = 1.31354\text{ h°F/Btu}$$

Step 9 of 12

Convection resistance from the outer surface of the towel

$$R_c = \frac{1}{hA}$$
$$= \frac{1}{(17.248)\pi\left[(3+0.02+0.02+0.12+0.12)/12\right]^2}$$
$$R_c = 0.247\text{ h.°F/Btu}$$

Step 10 of 12

Then total thermal resistance will be

$$R_{total} = R_{air} + R_t + R_c$$
$$= 0.55906 + 1.31354 + 0.247$$
$$R_{total} = 2.1196\text{ h°F/Btu}$$

Step 11 of 12

Rate of heat transfer

$$\dot{Q} = \frac{T_s - T_\infty}{R_{total}}$$

Where average surface temperature of potato will be

$$T_s = \left(\frac{300+200}{2}\right)$$
$$T_s = 250^0\text{F}$$

Then

$$\dot{Q} = \frac{T_s - T_\infty}{R_{total}}$$
$$= \frac{(250-70)}{2.1196}$$
$$\dot{Q} = 84.92\text{ Btu/h}$$

Step 12 of 12

Time taken for the potato to experience same temperature drop when wrapped in 0.12-in thick towel and there is a 0.02 in thick air space

$$\Delta t = \frac{Q}{\dot{Q}}$$
$$= \frac{50.8}{84.92}$$
$$= 0.598\text{ h}$$
$$\boxed{\Delta t = 35.88\text{min}}$$

Question: 17.130P

A 6-m-wide, 2.8-m-high wall is constructed of one layer of common brick (k = 0.72 W/m·K) of thickness 20 cm, one inside layer of light-weight plaster (k = 0.36 W/m·K) of thickness 1 cm. and one outside layer of cement based covering (k = 1.40 W/m·K) of thickness 2 cm. The inner surface of the wall is maintained at 23°C while the outer surface is exposed to outdoors at 8°C with a combined convection and radiation heat transfer coefficient of 17 W/m2 ·K. Determine the rate of heat transfer through the wall and temperature drops across the plaster, brick, covering, and surface-ambient air.

Step 1 of 12

Given: -

Thickness of brick layer is $L_2 = 20\,\text{cm}$

Thickness of plaster layer $L_1 = 1\text{cm}$

Thickness of cement based covering $L_3 = 2\,\text{cm}$

Thermal conductivity of plaster $k_1 = 0.36\ \text{W/m}\cdot{}^\circ\text{C}$

Thermal conductivity of brick $k_2 = 0.72\ \text{W/m}\cdot{}^\circ\text{C}$

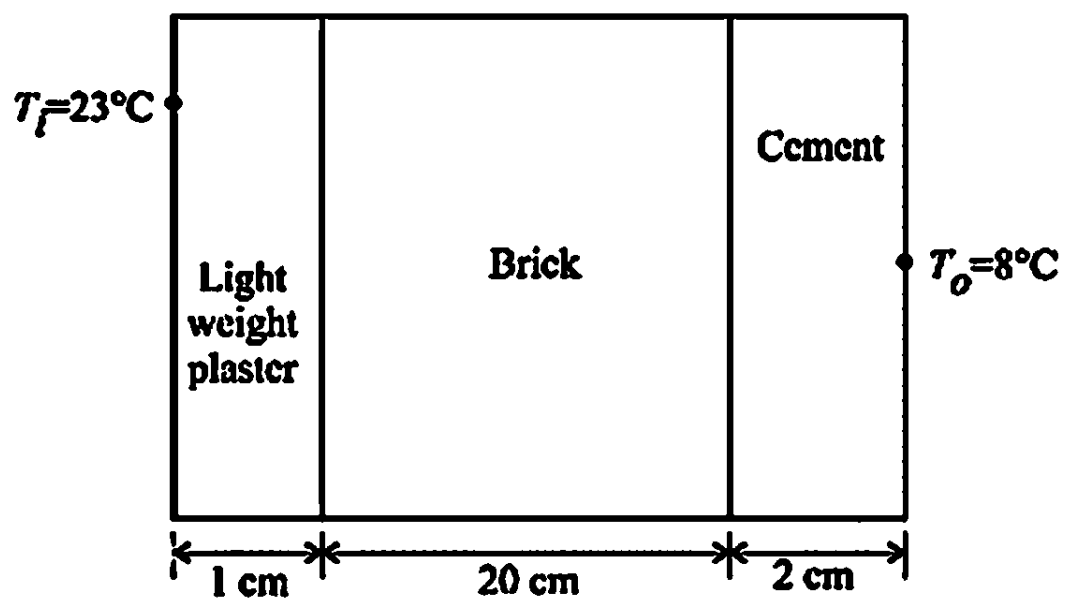
Thermal conductivity of cement covering $k_3 = 1.40\ \text{W/m}\cdot{}^\circ\text{C}$

Inner surface temperature of the wall $T_i = 23^0\,\text{C}$

Outer surface temperature of the wall $T_o = 8^0\,\text{C}$

Combined convection and radiation heat transfer coefficient $h = 17\,\text{W/m}^2.{}^0\text{C}$

Step 2 of 12



Step 3 of 12

Area of heat transfer

$A = (6)\times(2.8)$

$A = 16.8\ \text{m}^2$

Thermal resistance in case of plaster layer

$$R_1 = \frac{L_1}{k_1 A}$$

$$= \frac{0.01}{(0.36)(16.8)}$$

$R_1 = 0.00165\ ^\circ\text{C/W}$

Step 4 of 12

Thermal resistance in case of brick

$$R_2 = \frac{L_2}{k_2 A}$$

$$= \frac{0.20}{(0.72)(16.8)}$$

$R_2 = 0.01653\ ^\circ\text{C/W}$

Step 5 of 12

Thermal resistance in case of cement based coating

$$R_3 = \frac{L_3}{k_3 A}$$

$$= \frac{0.02}{(1.4)(16.8)}$$

$R_3 = 0.00085\ ^\circ\text{C/W}$

Step 6 of 12

Thermal resistance due to convection

$$R_{conv} = \frac{1}{hA}$$

$$= \frac{1}{(17)(16.8)}$$

$R_{conv} = 0.00350^\circ\text{C/W}$

Step 7 of 12

Then total thermal resistance will be

$R_t = R_1 + R_2 + R_3 + R_{conv}$

$= 0.00165 + 0.01653 + 0.00085 + 0.00350$

$R_t = 0.02253\ ^\circ\text{C/W}$

Step 8 of 12

Heat transfer rate for the entire wall can be calculated as

$$\dot{Q} = \frac{T_i - T_o}{R_t}$$

$$= \frac{(23-8)}{0.02253}$$

$$\boxed{\dot{Q} = 665.778\,\text{W}}$$

Step 9 of 12

Temperature drop across the plaster is

$\Delta T_1 = \dot{Q} R_1$

$= (665.778)(0.00165)$

$$\boxed{\Delta T_1 = 1.098^0\,\text{C}}$$

Step 10 of 12

Temperature drop across the brick is

$\Delta T_2 = \dot{Q} R_2$

$= (665.778)(0.01653)$

$$\boxed{\Delta T_2 = 11.0053^0\,\text{C}}$$

Step 11 of 12

Temperature drop across the cement covering

$\Delta T_3 = \dot{Q} R_3$

$= (665.778)(0.00085)$

$$\boxed{\Delta T_3 = 0.565^0\,\text{C}}$$

Step 12 of 12

Temperature drop across surface ambient air

$\Delta T_{conv} = \dot{Q} R_{conv}$

$= (665.778)(0.00350)$

$$\boxed{\Delta T_{conv} = 2.33^0\,\text{C}}$$

Question: 17.131P pg 1

Reconsider Prob. 17-130. It is desired to insulate the wall in order to decrease the heat loss by 90 percent. For the same inner surface temperature, determine the thickness of insulation and the outer surface temperature if the wall is insulated with (*a*) polyurethane foam (k = 0.025 W/m·K) and (*b*) glass fiber (k = 0.036 W/m·K).

Step 1 of 11

Given data:

Cross sectional area of the wall, $A = 6 \times 2.8 = 16.8 \text{ m}^2$

Thickness of the light weight plaster, $t_1 = 1\text{cm} = 0.01\text{m}$

Thickness of the common brick, $t_2 = 20\text{ cm} = 0.2\text{m}$

Thickness of the cement covering, $t_3 = 2\text{cm} = 0.02\text{m}$

Thermal conductivity of the light weight plaster, $k_1 = 0.36 \text{ W/m} \cdot \text{K}$

Thermal conductivity of the common brick, $k_2 = 0.72 \text{ W/m} \cdot \text{K}$

Thermal conductivity of the cement covering, $k_3 = 1.4 \text{ W/m} \cdot \text{K}$

Thermal conductivity of the polyurethane foam, $k_4 = 0.025 \text{ W/m} \cdot \text{K}$

Thermal conductivity of the glass fiber, $k_4 = 0.036 \text{ W/m} \cdot \text{K}$

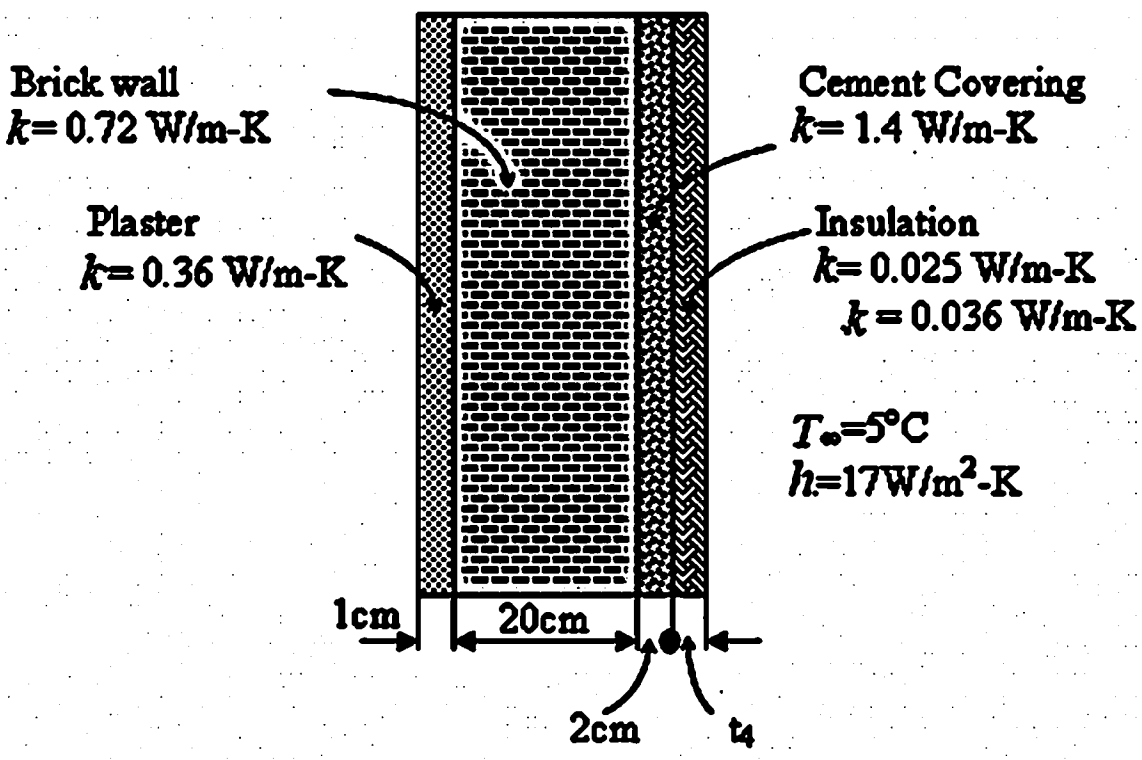
Inner surface temperature of the wall, $T_1 = 23°\text{C}$

Outdoor temperature, $T_{\infty,2} = 8°\text{C}$

Heat transfer coefficient on the outer wall, $h_o = 17 \text{ W/m}^2 \cdot \text{K}$

Step 2 of 11

The schematic diagram of the concrete wall and the thermal resistances can be represented as follows.



Step 3 of 11

Without any insulation, the heat lost by the wall to the ambient air can be determined as follows.

The individual resistances are evaluated in order to find out the thicknesses and surface temperatures of different layers of the wall.

Thermal resistance due to light weight plaster:

$$R_1 = R_{plaster} = \frac{t_1}{k_1 A_1} = \frac{0.01}{0.36 \times 16.8} = 0.0016534 \text{ K/W}$$

Thermal resistance due to common brick:

$$R_2 = R_{brick} = \frac{t_2}{k_2 A_2} = \frac{0.2}{0.72 \times 16.8} = 0.016534 \text{ K/W}$$

Step 4 of 11

Thermal resistance due to the cement based covering:

$$R_3 = R_{cement} = \frac{t_3}{k_3 A_3} = \frac{0.02}{1.4 \times 16.8} = 0.00085034 \text{ K/W}$$

Convection resistance from the outer surface of the wall:

$$R_o = R_{conv,o} = \frac{1}{h_o A} = \frac{1}{17 \times 16.8} = 0.0035014 \text{ K/W}$$

Step 5 of 11

As all the resistances are in series, the total resistance is given by,

$$R_{total} = R_1 + R_2 + R_3 + R_o = 0.0016534 + 0.016534 + 0.00085034 + 0.0035014 = 0.02254 \text{ K/W}$$

Step 6 of 11

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q}_t = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$

Step 6 of 11

Then the steady rate of heat transfer through the wall becomes

$$\dot{Q}_t = \frac{(T_1 - T_{\infty,2})}{R_{total}}$$
$$= \frac{23-8}{0.02254}$$
$$= 665.483\,\text{W}$$

Step 7 of 11

It is desired to insulate the wall in order to decrease the heat loss by 90%. Hence only 10% of total heat allowed.

Therefore, the allowable heat transfer through the wall with insulation can be calculated as

$$\dot{Q} = 0.1 \times \dot{Q}_t$$
$$= 0.1 \times 665.483$$
$$= 66.5483\,\text{W}$$

The total resistance can be determined as

$$R_{total} = \frac{(T_1 - T_{\infty,2})}{\dot{Q}}$$
$$= \frac{23-8}{66.5483}$$
$$= 0.2254\ \text{K/W}$$

Step 8 of 11

a) Using the polyurethane foam as an insulation material, the thickness of the insulation can be determined as,

$$R_4 = R_{total} - (R_1 + R_2 + R_3 + R_o)$$
$$= 0.2254 - 0.02254$$
$$= 0.20286\ \text{K/W}$$

where R_4 represents the thermal resistance across the insulation and the thickness of insulation can be determined as,

$$R_4 = \frac{t_4}{Ak_4}$$
$$t_4 = R_4 k_4 A_4$$
$$= 0.20286 \times 0.025 \times 16.8$$
$$= 0.0852\,\text{m}$$
$$= 8.52\,\text{cm}$$

$\boxed{t_4 = 8.52\,\text{cm}}$

Step 9 of 11

The outer surface temperature can be calculated using the heat rate equation.

$$\dot{Q} = \frac{(T_2 - T_\infty)}{R_o}$$

By rewriting the heat rate equation, we have,

$$T_2 = T_\infty + R_o \dot{Q}$$
$$= 8 + 0.0035014 \times 66.5483$$
$$= 8.233^\circ\text{C}$$

$\boxed{T_2 = 8.233^\circ\text{C}}$

Therefore if the insulation is polyurethane foam, then the thickness of insulation and the outer surface temperature are $t_4 = 8.52\,\text{cm}$ and $T_2 = 8.233^\circ\text{C}$ respectively.

Step 10 of 11

b)

Using glass fiber as an insulation material, the thickness of the insulation can be determined as follows.

$$R_4 = R_{total} - (R_1 + R_2 + R_3 + R_o)$$
$$= 0.2254 - 0.02254$$
$$= 0.20286\ \text{K/W}$$

where R_4 represents the thermal resistance across the insulation.

The thickness of insulation is given by,

$$t_4 = R_4 k_4 A_4$$
$$= 0.20286 \times 0.036 \times 16.8$$
$$= 0.1226\,\text{m}$$
$$= 12.26\,\text{cm}$$

$\boxed{t_4 = 12.26\,\text{cm}}$

Step 11 of 11

The outer surface temperature can be calculated using the heat rate equation

$$\dot{Q} = \frac{(T_2 - T_\infty)}{R_o}$$

By rewriting the heat rate equation, we have,

$$T_2 = T_\infty + R_o \dot{Q}$$
$$= 8 + 0.0035014 \times 66.5483$$
$$= 8.233^\circ\text{C}$$

$\boxed{T_2 = 8.233^\circ\text{C}}$

Therefore if the insulation is Glass fiber then the thickness of insulation and the outer surface temperature are $t_4 = 12.26\,\text{cm}$ and $T_2 = 8.233^\circ\text{C}$.

Question: 17.132P pg 1

A 0.2-cm-thick, 10-cm-high, and 15-cm-long circuit board houses electronic components on one side that dissipate a total of 15 W of heat uniformly. The board is impregnated with conducting metal fillings and has an effective thermal conductivity of 12 W/m·K. All the heat generated in the components is conducted across the circuit board and is dissipated from the back side of the board to a medium at 37°C, with a heat transfer coefficient of 45 W/m2·K. (*a*) Determine the surface temperatures on the two sides of the circuit board. (*b*) Now a 0.1-cm-thick, 10-cm-high, and 15-cm-long aluminum plate (k= 237 W/m·K) with 20 0.2-cm-thick. 2-cm-long, and 15-cm-wide aluminum fins of rectangular profile are

attached to the back side of the circuit board with a 0.03-cm.-thick epoxy adhesive (k = 1.8 W/m·K). Determine the new temperatures on the two sides of the circuit board.

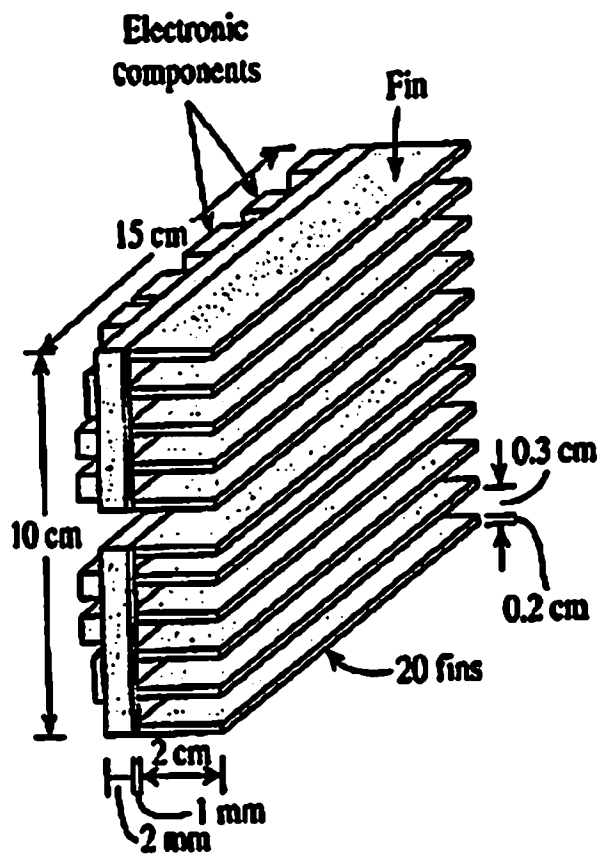


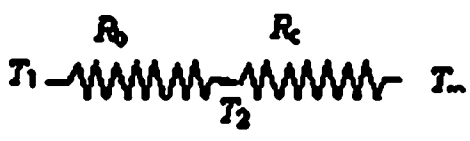
FIGURE P17–132

Step 1 of 12

Given: -
Dimensions of circuit board 0.2 cm-thick 10cm-high 15 cm-long
Heat dissipated Q= 15 W
Effective thermal conductivity of board k= 12 W/m.°C
Heat transfer coefficient h= 45 W/m^2.°C
Temperature of the medium $T_\infty = 37\,^\circ C$

Step 2 of 12

(*a*)
Total thermal resistance for the board can be calculated as follows:



Thermal resistance due to conduction in the board

$$R_b = \frac{L}{kA}$$
$$= \frac{0.002}{(12)(0.1)(0.15)}$$
$$R_b = 0.011\ ^\circ C/W$$

Thermal resistance due to convection from surface of the board to medium

$$R_t = \frac{1}{hA}$$
$$= \frac{1}{(45)\times(0.1)\times(0.15)}$$
$$R_t = 1.481\ ^\circ C/W$$

Total thermal resistance
$$R_t = R_b + R_t$$
$$= 0.011+1.481$$
$$R_t = 1.492\ ^\circ C/W$$

Step 3 of 12

The rate of heat transfer from the board is

$$\dot{Q} = \frac{T_1 - T_\infty}{R_t}$$
$$T_1 = T_\infty + \dot{Q}R_t$$
$$= 37 + (15)(1.492)$$
$$\boxed{T_1 = 59.38^0\,C}$$

Temperature on other surface of the board can be calculated as follows

$$\dot{Q} = \frac{T_1 - T_2}{R_b}$$
$$T_2 = T_1 - \dot{Q}R_b$$
$$= 59.38 - (15)(0.011)$$
$$\boxed{T_2 = 59.215^0\,C}$$

Step 4 of 12

(*b*)
Given: -
Dimensions of aluminum plate 0.1 cm-thick, 10 cm-high and 15 cm-long
Total number of fins N= 20
Dimensions of fins 0.2 cm-thick, 2 cm-long and 15 cm-wide
Thermal conductivity of aluminum $k_{al} = 237$ W/m. °C
Thermal conductivity of epoxy adhesive $k_{epoxy} = 1.8$ W/m. °C



Step 5 of 12

Efficiency of rectangular fins can be calculated as follows

$$m = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}}$$
$$\sqrt{\frac{2h}{}}$$

Efficiency of rectangular fins can be calculated as follows

$$m = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}}$$

$$= \sqrt{\frac{2h}{kt}}$$

$$= \sqrt{\frac{2(45)}{(237)(0.002)}}$$

$$m = 13.78 \text{ m}^{-1}$$

$$\eta_{fin} = \frac{\tanh mL}{mL}$$

$$= \frac{\tanh(13.78 \times 0.02)}{13.78 \times 0.02}$$

$$\eta_{fin} = 0.975$$

Step 6 of 12

Area of finned surface can be calculated as

$$A_f = N \times 2w \times L_c$$

Where L_c the characteristic length and characteristic length for rectangular fins is

$$L_c = L + \frac{t}{2}$$

Step 7 of 12

Then area of finned surfaces will be

$$A_f = (20)2w\left(L + \frac{t}{2}\right)$$

$$= (20)2(0.15)\left(0.02 + \frac{0.002}{2}\right)$$

$$A_f = 0.126 \text{ m}^2$$

Surface area of unfinned surface

$$A_{uf} = (0.1)(0.15) - 20(0.002)(0.15)$$

$$A_{uf} = 0.009 \text{ m}^2$$

Step 8 of 12

Efficiency of the fin is defined as the ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature

Then

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin\,max}}$$

$$\dot{Q}_{fin} = \eta_{fin}\dot{Q}_{fin\,max}$$

$$= \eta_{fin}hA_f(T_b - T_\infty)$$

Heat transfer rate from the unfinned surface is

$$\dot{Q}_{uf} = hA_{uf}(T_b - T_\infty)$$

Then total heat transfer rate is

$$\dot{Q}_t = \dot{Q}_{uf} + \dot{Q}_{fin}$$

$$= hA_{uf}(T_b - T_\infty) + \eta_{fin}hA_f(T_b - T_\infty)$$

$$= h(T_b - T_\infty)(\eta_{fin}A_f + A_{uf})$$

Step 9 of 12

Temperature of base for finned surface can be calculated as

$$T_b = T_\infty + \frac{\dot{Q}_t}{h(\eta_{fin}A_f + A_{uf})}$$

$$= 37 + \frac{15}{(45)[(0.975)(0.126) + (0.0090)]}$$

$$T_b = 39.528^0\text{C}$$

Step 10 of 12

Temperature of surface on two side of the board can be calculated as follows:

$$R_{al} = \frac{L}{kA}$$

$$= \frac{0.001}{(237)(0.1)(0.15)}$$

$$R_{Al} = 0.00028 \text{ °C/W}$$

$$R_{ep} = \frac{L}{kA}$$

Since thickness of the epoxy layer is $L = 0.03\text{cm} = 0.0003\text{m}$

$$R_{ep} = \frac{0.0003}{(1.8)(0.1)(0.15)}$$

$$R_{ep} = 0.0111 \text{ °C/W}$$

Step 11 of 12

Heat transfer rate is

$$\dot{Q} = \frac{T_1 - T_{base}}{R_{aluminum} + R_{epoxy} + R_{board}}$$

$$15 = \frac{(T_1 - 39.5)}{(0.00028 + 0.0111 + 0.011)}$$

$$T_1 = 39.5 + (15)(0.02238)$$

$$\boxed{T_1 = 39.8357^0\text{C}}$$

Step 12 of 12

$$\dot{Q} = \frac{T_1 - T_2}{R_{board}}$$

$$T_2 = T_1 - \dot{Q}R_{board}$$

$$= 39.8357 - (15)(0.011)$$

$$\boxed{T_2 = 39.67^0\text{C}}$$

17.132P pg 3

$$T_2 = T_1 - \dot{Q}R_{board}$$
$$= 39.8357 - (15)(0.011)$$

$$\boxed{T_2 = 39.67^0\text{C}}$$

Question: 17.133P pg 1

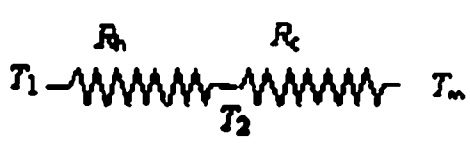
Repeat Prob, 17-132 using a copper plate with copper fins (k = 386 W/m·K) instead of aluminum ones.

Step 1 of 12

Given: -
Dimensions of circuit board 0.2 cm-thick 10cm-high 15 cm-long
Heat dissipated Q= 15 W
Effective thermal conductivity of board k= 12 W/m.°C
Heat transfer coefficient h= 45 W/m^2.°C
Temperature of the medium = 37 °C

Step 2 of 12

(*a*)
Total thermal resistance for the board can be calculated as follows:



Thermal resistance due to conduction in the board

$$R_b = \frac{L}{kA}$$
$$= \frac{0.002}{(12)(0.1)(0.15)}$$
$$R_b = 0.011\ ^\circ C/W$$

Thermal resistance due to convection from surface of the board to medium

$$R_c = \frac{1}{hA}$$
$$= \frac{1}{(45)\times(0.1)\times(0.15)}$$
$$R_c = 1.481\ ^\circ C/W$$

Total thermal resistance

$$R_t = R_b + R_c$$
$$= 0.011+1.481$$
$$R_t = 1.492\ ^\circ C/W$$

Step 3 of 12

The rate of heat transfer from the board is

$$\dot{Q} = \frac{T_1 - T_\infty}{R_t}$$
$$T_1 = T_\infty + \dot{Q}R_t$$
$$= 37 + (15)(1.492)$$
$$\boxed{T_1 = 59.38^0\,C}$$

Step 4 of 12

Temperature on other surface of the board can be calculated as follows

$$\dot{Q} = \frac{T_1 - T_2}{R_b}$$
$$T_2 = T_1 - \dot{Q}R_b$$
$$= 59.38 - (15)(0.011)$$
$$\boxed{T_2 = 59.215^0\,C}$$

Step 5 of 12

(*b*)
Given: -
Dimensions of aluminum plate 0.1 cm-thick, 10 cm-high and 15 cm-long
Total number of fins N= 20
Dimensions of fins 0.2 cm-thick, 2 cm-long and 15 cm-wide
Thermal conductivity of copper k_{Cu} = 386 W/m. °C
Thermal conductivity of epoxy adhesive k_{epoxy} = 1.8 W/m. °C



Step 6 of 12

Efficiency of rectangular fins can be calculated as follows

$$m = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}}$$
$$= \sqrt{\frac{2h}{kt}}$$
$$= \sqrt{\frac{2(45)}{(386)(0.002)}}$$
$$m = 10.79\ m^{-1}$$

$$\eta_{fin} = \frac{\tanh mL}{mL}$$
$$= \frac{\tanh(10.79\times 0.02)}{10.79\times 0.02}$$
$$\eta_{fin} = 0.9847$$

Step 7 of 12

Area of finned surface can be calculated as
$$A_f = N\times 2w\times L_c$$
Where L_c the characteristic length and characteristic length for rectangular fins is

$$L_c = L + \frac{t}{2}$$

Then area of finned surfaces will be

$$A_f = (20)2w\left(L + \frac{t}{2}\right)$$
$$= (20)2(0.15)\left(0.02 + \frac{0.002}{2}\right)$$
$$A_f = 0.1290\ m^2$$

Then area of finned surfaces will be

$$A_f = (20)2w\left(L+\frac{t}{2}\right)$$

$$= (20)2(0.15)\left(0.02+\frac{0.002}{2}\right)$$

$$A_f = 0.126\ \text{m}^2$$

Surface area of unfinned surface

$$A_{uf} = (0.1)(0.15) - 20(0.002)(0.15)$$

$$A_{uf} = 0.009\ \text{m}^2$$

Step 8 of 12

Efficiency of the fin is defined as the ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature

Then

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{finmax}}$$

$$\dot{Q}_{fin} = \eta_{fin}\dot{Q}_{finmax}$$

$$= \eta_{fin}hA_f(T_b - T_\infty)$$

Heat transfer rate from the unfinned surface is

$$\dot{Q}_{uf} = hA_{uf}(T_b - T_\infty)$$

Then total heat transfer rate is

$$\dot{Q}_t = \dot{Q}_{uf} + \dot{Q}_{fin}$$

$$= hA_{uf}(T_b - T_\infty) + \eta_{fin}hA_f(T_b - T_\infty)$$

$$= h(T_b - T_\infty)(\eta_{fin}A_f + A_{uf})$$

Step 9 of 12

Temperature of base for finned surface can be calculated as

$$T_b = T_\infty + \frac{\dot{Q}_t}{h(\eta_{fin}A_f + A_{uf})}$$

$$= 37 + \frac{15}{(45)[(0.9847)(0.126)+(0.0090)]}$$

$$T_b = 39.505^0\text{C}$$

Step 10 of 12

Temperature of surface on two side of the board can be calculated as follows:

$$R_{Cu} = \frac{L}{kA}$$

$$= \frac{0.001}{(386)(0.1)(0.15)}$$

$$R_{Cu} = 0.000172\ ^\circ\text{C/W}$$

$$R_{ep} = \frac{L}{kA}$$

Since thickness of the epoxy layer is $L = 0.03\text{cm} = 0.0003\text{m}$

$$R_{ep} = \frac{0.0003}{(1.8)(0.1)(0.15)}$$

$$R_{ep} = 0.0111\ ^\circ\text{C/W}$$

Step 11 of 12

Heat transfer rate is

$$\dot{Q} = \frac{T_1 - T_{base}}{R_{Cu} + R_{epoxy} + R_{board}}$$

$$15 = \frac{(T_1 - 39.505)}{(0.000172 + 0.0111 + 0.011)}$$

$$T_1 = 39.505 + (15)(0.02227)$$

$$\boxed{T_1 = 39.839^0\text{C}}$$

Step 12 of 12

$$\dot{Q} = \frac{T_1 - T_2}{R_{board}}$$

$$T_2 = T_1 - \dot{Q}R_{board}$$

$$= 39.839 - (15)(0.011)$$

$$\boxed{T_2 = 39.674^0\text{C}}$$

Question: 17.134P

Consider two identical people each generating 60 w of metabolic heat steadily while doing sedentary work, and dissipating it by convection and perspiration. The first person is wearing clothes made of 1-mm-thick leather (k = 0.159 W/m·K) that covers half of the body while the second one is wearing clothes made of 1-mm-thick synthetic fabric (k = 0.13 W/m·K) that covers the body completely. The ambient air is at 30°C, the heat transfer coefficient at the outer surface is 15 W/m2·K, and the inner surface temperature of the clothes can be taken to be 32°C. Treating the body of each person as a 25-cm-diameter, 1.7-m-long cylinder, determine the fractions of heat lost from each person by perspiration.

Step 1 of 4

Given: - Heat generated by both people = 60 W

Thickness of cloth of person one = 1mm

Thermal conductivity of leather = 0.159 W/m°C

Thickness of cloth of second person = 1mm

Thermal conductivity of synthetic fibre = 0.13 W/m°C

Temperature of ambient air = 30°C

Heat transfer coefficient = 15 W/m^2.°C

Area of heat transfer for both persons

$$A_1 = \pi DL/2$$
$$= \pi(0.25\text{ m})(1.7\text{ m})/2 = 0.6675\text{ m}^2$$
$$A_2 = 2A_1$$
$$= 2\times 0.6675 = 1.335\text{ m}^2$$

Step 2 of 4

Calculating thermal resistance from first person body

$$R_1 = \frac{L}{kA}$$
$$= \frac{0.001\text{ m}}{(0.159\text{ W/m.°C})(0.6675\text{ m}^2)} = 0.00942\text{ °C/W}$$
$$R_c = \frac{1}{hA}$$
$$= \frac{1}{(15\text{ W/m}^2\text{.°C})(0.6675\text{ m}^2)} = 0.09988\text{ °C/W}$$
$$R_t = R_1 + R_c$$
$$= 0.00942 + 0.09988 = 0.10930\text{ °C/W}$$

Step 3 of 4

Rate of sensible heat transfer through the first person

$$\dot{Q}_c = \frac{T_i - T_o}{R_t}$$
$$= \frac{(32-30)°\text{C}}{0.10930°\text{C/W}} = 18.3\text{ W}$$
$$\dot{Q}_s = \frac{T_i - T_o}{R_c}$$
$$= \frac{(32-30)°\text{C}}{0.09988°\text{C/W}} = 20.0\text{ W}$$
$$\dot{Q}_{sensible} = \dot{Q}_c + \dot{Q}_s$$
$$= 18.3 + 20 = 38.3\text{ W}$$

Fraction of heat lost by respiration can be calculated as

$$f = \frac{\dot{Q}_r}{\dot{Q}_t}$$
$$= \frac{\dot{Q}_t - \dot{Q}_{sensible}}{\dot{Q}_t}$$
$$= \frac{60-38.3}{60} = 0.362$$

Step 4 of 4

Now calculating fraction of heat lost by respiration for second person

T_2

$$R_s = \frac{L}{kA}$$
$$= \frac{0.001\text{ m}}{(0.13\text{ W/m.°C})(1.335\text{ m}^2)} = 0.00576\text{ °C/W}$$
$$R_c = \frac{1}{hA}$$
$$= \frac{1}{(15\text{ W/m}^2\text{.°C})(1.335\text{ m}^2)} = 0.04994\text{ °C/W}$$
$$R_t = R_s + R_c$$
$$= 0.00576 + 0.04994 = 0.05570\text{ °C/W}$$
$$\dot{Q}_{sensible} = \frac{T_i - T_o}{R_t}$$
$$= \frac{(32-30)°\text{C}}{0.05570°\text{C/W}} = 35.9\text{ W}$$

$$f = \frac{\dot{Q}_r}{\dot{Q}_t}$$
$$= \frac{\dot{Q}_t - \dot{Q}_{sensible}}{\dot{Q}_t} = \frac{60-35.9}{60}$$
$$= 0.402$$

Question: 17.135P pg 1

A 4-m-high and 6-m-long wall is constructed of two large 0.8-cm-thick steel plates (k = 15 W/m·K) separated by 1-cm-thick and 22-cm wide steel bars placed 99 cm apart. The remaining space between the steel plates is filled with fiberglass insulation (k = 0.035 W/m·K). If the temperature difference between the inner and the outer surfaces of the walls is 22°C, determine the rate of heat transfer through the wall. Can we ignore the steel bars between the plates in heat transfer analysis since they occupy only 1 percent of the heat transfer surface area?

Step 1 of 10

Given data:

Area of the steel wall, $A_1 = 6 \times 4 = 24 \text{ m}^2$

Thickness of the steel plates, $t_{12} = t_{34} = 0.8 \text{ cm} = 0.008 \text{ m}$

Thickness of the steel bar, $t_{23} = 22\text{cm} = 0.22\text{m}$

Thickness of the fiber glass insulation, $t_{23} = 22\text{cm} = 0.22\text{m}$

Length of the fiber glass insulation, $L_A = 99 \text{ cm} = 0.99 \text{ m}$

Length of the steel plate, $L_B = 1\text{cm} = 0.01\text{m}$

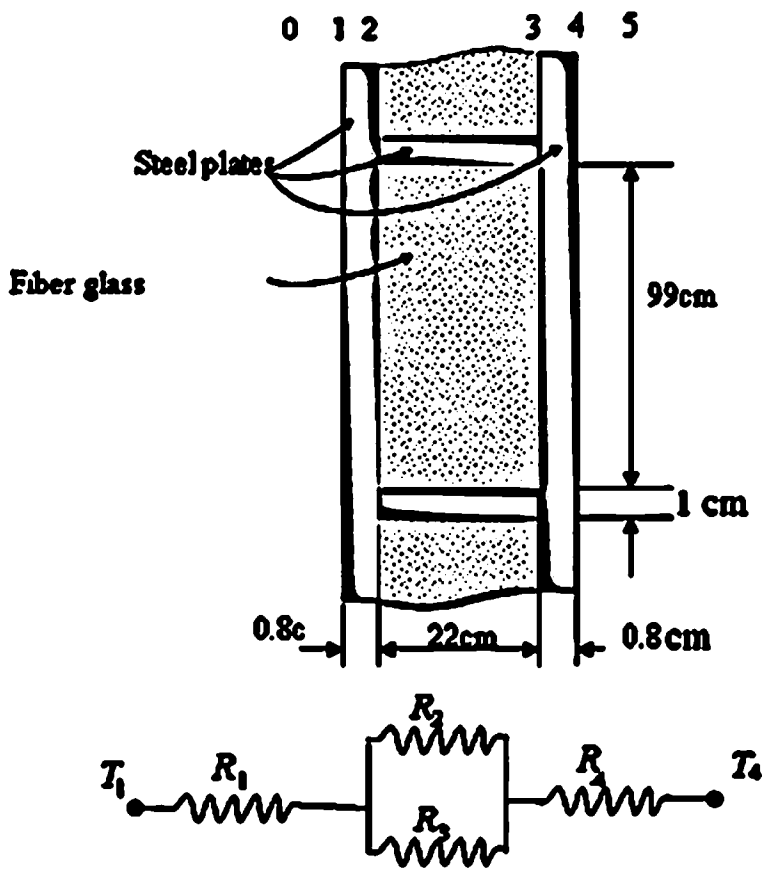
Thermal conductivity of the steel plates, $k_{steel} = 15\,\text{W/m}\cdot\text{K}$

Thermal conductivity of the insulation, $k_{insulation} = 0.035\,\text{W/m}\cdot\text{K}$

Temperature difference between inner and outer surfaces of the wall, $\Delta T = 22°\text{C}$

Step 2 of 10

The schematic diagram of the steel plates and thermal network is as shown below:



Step 3 of 10

By using the thermal network approach the total resistance of the network can be determined as

$$R_{total} = R_1 + R_{23} + R_4$$

$$= R_1 + \frac{R_2 R_3}{R_2 + R_3} + R_4$$

The individual thermal resistance formulae are as follows,

$$R_1 = \frac{t_{12}}{k_{steel} A_i}$$

$$R_2 = \frac{t_{23}}{k_{insulation} A_A}$$

$$R_3 = \frac{t_{23}}{k_{steel} A_B}$$

$$R_{23} = \frac{R_2 R_3}{R_2 + R_3}$$

$$= \frac{\left(\frac{t_{23}}{k_{insulation} A_A}\right)\left(\frac{t_{23}}{k_{steel} A_B}\right)}{\frac{t_{23}}{k_{insulation} A_A} + \frac{t_{23}}{k_{steel} A_B}}$$

$$R_4 = \frac{t_{34}}{k_{steel} A_o}$$

Step 4 of 10

Then total thermal resistance becomes,

$$R_{total} = \frac{t_{12}}{k_{steel} A_i} + \frac{\left(\frac{t_{23}}{k_{insulation} A_A}\right)\left(\frac{t_{23}}{k_{steel} A_B}\right)}{\frac{t_{23}}{k_{insulation} A_A} + \frac{t_{23}}{k_{steel} A_B}} + \frac{t_{34}}{k_{steel} A_o}$$

Where cross sectional areas are

$$A_i = A_o = (L_A + L_B) w$$

$$A_A = L_A w$$

$$A_B = L_B w$$

Where, w represents the width of all walls

Step 5 of 10

The total resistance multiplies with $A_i = A_o = (L_A + L_B) w$ and it can be expressed as

$$R_{total} A_i = \frac{t_{12}}{k_{steel}} + A_i \left[\frac{\left(\frac{t_{23}}{k_{insulation} A_A}\right)\left(\frac{t_{23}}{k_{steel} A_B}\right)}{\frac{t_{23}}{k_{insulation} A_A} + \frac{t_{23}}{k_{steel} A_B}} \right] + \frac{t_{34}}{k_{steel}}$$

$$= \frac{t_{12}}{k_{steel}} + (L_A + L_B) w \left[\frac{\left(\frac{t_{23}}{k_{insulation} A_A}\right)\left(\frac{t_{23}}{k_{steel} A_B}\right)}{\frac{t_{23}}{k_{insulation} A_A} + \frac{t_{23}}{k_{steel} A_B}} \right] + \frac{t_{34}}{k_{steel}}$$

$$= \frac{t_{12}}{k_{steel}} + (L_A + L_B) \left[\frac{\left(\frac{t_{23}}{k_{insulation} L_A}\right)\left(\frac{t_{23}}{k_{steel} L_B}\right)}{\frac{t_{23}}{k_{insulation} L_A} + \frac{t_{23}}{k_{steel} L_B}} \right] + \frac{t_{34}}{k_{steel}}$$

$$= \frac{t_{12}}{k_{steel}} + (L_A + L_B)w\left[\frac{1}{\frac{t_{23}}{k_{insulation}A_A} + \frac{t_{23}}{k_{steel}A_B}}\right] + \frac{t_{34}}{k_{steel}}$$

$$= \frac{t_{12}}{k_{steel}} + (L_A + L_B)\left(\frac{\left(\frac{t_{23}}{k_{insulation}L_A}\right)\left(\frac{t_{23}}{k_{steel}L_B}\right)}{\frac{t_{23}}{k_{insulation}L_A} + \frac{t_{23}}{k_{steel}L_B}}\right) + \frac{t_{34}}{k_{steel}}$$

Step 6 of 10

Then substituting the known values in the above equation,

$$R_{total}A_i = \frac{0.008}{15} + 1\times\left(\frac{\left[\frac{0.22}{(0.035\times0.99)}\right]\times\left[\frac{0.22}{(15\times0.01)}\right]}{\left[\frac{0.22}{(0.035\times0.99)}\right]+\left[\frac{0.22}{(15\times0.01)}\right]}\right) + \frac{0.008}{15}$$

$$= 0.00053 + \frac{9.712}{7.0158} + 0.000533$$

$$= 1.3853\,\text{m}^2\cdot\text{K/W}$$

Step 7 of 10

The average heat flux through the wall can be determined as

$$q = \frac{\dot{Q}}{A_i}$$

$$= \frac{(T_4 - T_1)}{R_{total}A_i}$$

$$= \frac{22}{1.3853}$$

$$= 15.88\,\text{W/m}^2$$

The total heat transfer through the composite wall can be determined as

$$\dot{Q} = Aq$$

$$= 24\times15.88$$

$$= 381.12\,\text{W}$$

Therefore, the total heat transfer through the composite wall is $\boxed{\dot{Q} = 381.12\,\text{W}}$

Step 8 of 10

Let us calculate the total heat transfer through the wall by neglecting the steel bars between the plates,

Cross sectional areas are

$$A_i = A_o = L_A w$$

$$A_A = L_A w$$

Then total thermal resistance becomes,

$$R_{total}A_i = \frac{t_{12}}{k_{steel}} + \frac{t_{23}}{k_{insulation}} + \frac{t_{34}}{k_{steel}}$$

$$= \frac{0.008}{15} + \frac{0.22}{0.035} + \frac{0.008}{15}$$

$$= 6.286\,\text{m}^2\cdot\text{K/W}$$

Step 9 of 10

The average heat flux through the wall can be determined as

$$q = \frac{\dot{Q}}{A_i}$$

$$= \frac{(T_4 - T_1)}{R_{total}A_i}$$

$$= \frac{22}{6.286}$$

$$= 3.499\,\text{W/m}^2$$

The total heat transfer through the composite wall can be determined as

$$\dot{Q} = Aq$$

$$= 24\times3.499$$

$$= 83.976\ \text{W}$$

Therefore, the total heat transfer through the composite wall is $\boxed{\dot{Q} = 83.976\ \text{W}}$

Step 10 of 10

Since we neglected the steel bars in between two steel plates, the total heat transfer rate drastically reduced and the error is to be

$$\%\text{Error} = \frac{(381.12 - 83.976)}{381.12}\times100$$

$$= 77.96\%$$

Therefore, though we ignored 1% of the total area, the error is significant and hence we cannot ignore the steel bars for calculation. 77% of total heat transfer occurs through the steel bars across the wall despite the negligible space that they occupy. The connecting bars are served as "thermal bridges".

Question: 17.136P

Cold conditioned air at 12°C is flowing inside a l·5-cm-thick square aluminum (k = 237 W/m·K) duct ofinner cross section 22 cm × 22 cm at a mass flow rate of 0.8 kg/s. The duct is exposed to air at 33°C with a combined convection-radiation heal transfer coefficient of 13 W/nr-K. The convection heat transfer coefficient at the inner surface is 75 W/m2-K. If thc air temperature in the duct should not increase by more than l°C determine the maximum length of the duct.

Step 1 of 8

Given: -

Temperature of cold conditioned air $T_i = 12°C$

Thickness of square aluminum duct $L_{Al} = 15$ cm

Thermal conductivity of aluminum $k_{Al} = 237\ W/m^0C$

Inner cross-section of aluminum duct is 22 cm × 22 cm

Mass flow rate of cold conditioned air $\dot{m} = 0.8$ kg/s

Temperature of air surrounding duct $T_\infty = 33°C$

Combined convection and radiation heat transfer coefficient on outer surface $h_o = 13\ W/m^2.°C$

Convection heat transfer coefficient for inner surface $h_i = 75\ W/m^2.°C$

Step 2 of 8

The thermal resistance circuit is



Step 3 of 8

Inner and outer surface area for unit length of the duct can be calculated as follows

$$A_i = 4a_1$$
$$= 4(0.22)$$
$$A_i = 0.88\ m^2$$
$$A_o = 4a_2$$
$$= 4(0.22+0.03)$$
$$A_o = 1.0\ m^2$$

Step 4 of 8

Thermal resistances for the system can be calculated as follows

$$R_{inner} = \frac{1}{h_1 A_i}$$
$$= \frac{1}{(75)(0.88)}$$
$$R_{inner} = 0.01515°C/W$$

$$R_{Al} = \frac{L_{Al}}{kA}$$
$$R_{Al} = \frac{L_{Al}}{k(A_i + A_o)/2}$$
$$= \frac{0.015}{(237)[(0.88+1)/2]}$$
$$R_{Al} = 0.0000673\ °C/W$$

$$R_{outer} = \frac{1}{h_2 A_o}$$
$$= \frac{1}{(13)(1.0)}$$
$$R_{outer} = 0.07692°C/W$$

Step 5 of 8

Total thermal resistance will be

$$R_t = R_{inner} + R_{Alm} + R_{outer}$$
$$= 0.01515+0.0000673+0.07692$$
$$R_t = 0.09213\ °C/W$$

Step 6 of 8

Rate of heat loss per length of duct

$$\dot{Q} = \frac{T_\infty - T_i}{R_t}$$
$$= \frac{(33-12)}{0.09213}$$
$$\dot{Q} = 227.938\ W/m$$

Step 7 of 8

If inside temperature of air in the duct increases by 1°C, rate of heat gained by inside air can be calculated as follows

$$\dot{Q}_t = \dot{m} C_p \Delta T$$
$$= (0.8)(1006)(1)$$
$$\dot{Q}_t = 804\ W$$

Step 8 of 8

Maximum length of duct can be calculated as

$$L = \frac{\dot{Q}_{total}}{\dot{Q}}$$
$$= \frac{804}{227.938}$$
$$\boxed{L = 3.527\ m}$$

Question: 17.137P pg 1

When analyzing heat transfer through windows, it is important to consider the frame as well as the glass area. Consider a 2-m-wide, 1.5-m-high wood-framed window with 85 percent of the area covered by 3-mm-thick single-pane glass (k = 0.7 W/m·K). The frame is 5 cm thick, and is made of pine wood (k = 0.12 W/m·K). The heat transfer coefficient is 7 W/m2·K inside and 13 W/m2·K outside. Thc room is maintained at 24°C, and the outdoor temperature is 40°C. Determine the percent error involved in heat transfer when the window is assumed to consist of glass only.

Step 1 of 14

Given: -

Width of window = 2-m

Height of window =1.5 m

Inside heat transfer coefficient $h_i = 7\ \text{W/m}^2\cdot^\circ\text{C}$

Outer heat transfer coefficient $h_o = 13\ \text{W/m}^2\cdot^\circ\text{C}$

Thermal conductivity of glass $k_{glass} = 0.7\ \text{W/m}\cdot^\circ\text{C}$

Thermal conductivity of pine wood $k_{pinewood} = 0.12\ \text{W/m}\cdot^\circ\text{C}$

Thickness of the glass $L_g = 3\text{mm}$

Room air temperature $T_{\infty 1} = 24^0\text{C}$

Outside air temperature $T_{\infty 2} = 40^0\text{C}$

Step 2 of 14

Only 85% of the area covered by thick single pane glass

Heat transfer area of glass and wood

$A_g = 0.85(1.5)(2)$

$A_g = 2.55\ \text{m}^2$

$A_w = 0.15(1.5)(2)$

$A_w = 0.45\ \text{m}^2$

Step 3 of 14

Thermal resistance due to convection from inside air and inner surface of the glass

$$R_{ig} = \frac{1}{h_i A_g}$$

$$= \frac{1}{(7)(2.55)}$$

$R_{ig} = 0.05602^\circ\text{C/W}$

Thermal resistance due to conduction inside the glass

$$R_g = \frac{L_g}{k_g A_g}$$

$$= \frac{0.003}{(0.7)(2.55)}$$

$R_g = 0.00168\ ^\circ\text{C/W}$

Thermal resistance due to convection from outside air and outer surface of the glass

$$R_{og} = \frac{1}{h_0 A_g}$$

$$= \frac{1}{(13)(2.55)}$$

$R_{og} = 0.03017^\circ\text{C/W}$

Step 4 of 14

Total resistance in the glass

$R_{tg} = R_{ig} + R_g + R_{og}$

$= 0.05602 + 0.00168 + 0.03017$

$R_{tg} = 0.08787\ ^\circ\text{C/W}$

Step 5 of 14

Thermal resistance due to convection from inside air and inner surface of the pine wood

$$R_{iw} = \frac{1}{h_i A_w}$$

$$= \frac{1}{(7)(0.45)}$$

$R_{iw} = 0.31746^\circ\text{C/W}$

Thermal resistance due to conduction inside the pine wood

$$R_w = \frac{L_w}{k_w A_w}$$

$$= \frac{0.05}{(0.12)(0.45)}$$

$R_w = 0.92593\ ^\circ\text{C/W}$

Thermal resistance due to convection from outside air and outer surface of the pine wood

$$R_{ow} = \frac{1}{h_o A_w}$$

$$= \frac{1}{(13)(0.45)}$$

$R_{ow} = 0.17094^\circ\text{C/W}$

Step 6 of 14

Total resistance in the pine wood

$R_{tw} = R_{iw} + R_w + R_{ow}$

$= 0.31746 + 0.92593 + 0.17094$

$R_{tw} = 1.41433\ ^\circ\text{C/W}$

Step 7 of 14

Heat transfer rate through glass

$$\dot{Q}_g = \frac{T_{\infty 2} - T_{\infty 1}}{R_{tg}}$$

$$= \frac{(40-24)}{0.08787}$$

Heat transfer rate through glass

$$\dot{Q}_g = \frac{T_{\infty 2} - T_{\infty 1}}{R_{tg}}$$
$$= \frac{(40-24)}{0.08787}$$
$$\dot{Q}_g = 182.1\ \text{W}$$

Step 8 of 14

Heat transfer rate through pine wood

$$\dot{Q}_w = \frac{T_{\infty 2} - T_{\infty 1}}{R_{tw}}$$
$$= \frac{(40-24)}{1.41433}$$
$$\dot{Q}_w = 11.3\ \text{W}$$

Step 9 of 14

Total heat transfer rate through glass and pine wood

$$\dot{Q}_t = \dot{Q}_g + \dot{Q}_w$$
$$= 182.1 + 11.3$$
$$\dot{Q}_t = 193.4\ \text{W}$$

Step 10 of 14

Heat transfer area when there is only glass

$$A_g = (1.5)(2)$$
$$A_g = 3.0\ \text{m}^2$$

Step 11 of 14

Thermal resistance for the system can be calculated as follows
Thermal resistance due to convection from inside air and inner surface of the glass

$$R_{ig} = \frac{1}{h_i A_g}$$
$$= \frac{1}{(7)(3.0)}$$
$$R_{ig} = 0.04762°\text{C/W}$$

Thermal resistance due to conduction inside the glass

$$R_g = \frac{L_g}{k_g A_g}$$
$$= \frac{0.003}{(0.7)(3.0)}$$
$$R_g = 0.001428\ °\text{C/W}$$

Thermal resistance due to convection from outside air and outer surface of the glass

$$R_{og} = \frac{1}{h_0 A_g}$$
$$= \frac{1}{(13)(3.0)}$$
$$R_{og} = 0.02564°\text{C/W}$$

Step 12 of 14

Total thermal resistance if there is only glass

$$R_{tg} = R_{ig} + R_g + R_{og}$$
$$= 0.04762 + 0.001428 + 0.02564$$
$$R_{tg} = 0.07468\ °\text{C/W}$$

Step 13 of 14

Heat transfer rate

$$\dot{Q}_g = \frac{T_{\infty 2} - T_{\infty 1}}{R_{tg}}$$
$$= \frac{(40-24)}{0.07468}$$
$$\dot{Q}_g = 214.247\ \text{W}$$

Step 14 of 14

Percentage change for heat transfer rate with glass and wood and with only glass can be calculated as follows

$$\%\ \text{Error} = \frac{\dot{Q}_g - \dot{Q}_t}{\dot{Q}_t}$$
$$= \frac{214.247 - 193.4}{193.4} \times 100$$
$$\boxed{\%\text{Error} = 10.779\%}$$

Question: 17.138P pg 1

Steam at 260°C is flowing inside a steel pipe (k = 61 W/m·K) whose inner and outer diameters are 10 cm and 12 cm, respectively, in an environment at 20°C. Thc heal transfer coefficients inside and outside the pipe are 120 W/m2·K and 14 W/m2·K. respectively. Determine (*a*) the thickness of the insulation (k = 0.038 W/m·K) needed to reduce the heat loss by 95 percent and (*b*) the thickness of the insulation needed to reduce the exposed surface temperature of insulated pipe to 40°C for safety reasons.

Step 1 of 10

Given data:

Inner diameter of the steel pipe, $d_1 = 10\text{cm} = 0.1\text{m}$

Outer diameter of the steel pipe, $d_2 = 12\text{cm} = 0.12\text{m}$

Thermal conductivity of the steel pipe, $k_1 = 61\,\text{W/m}\cdot\text{K}$

Thermal conductivity of the insulation, $k_2 = 0.038\,\text{W/m}\cdot\text{K}$

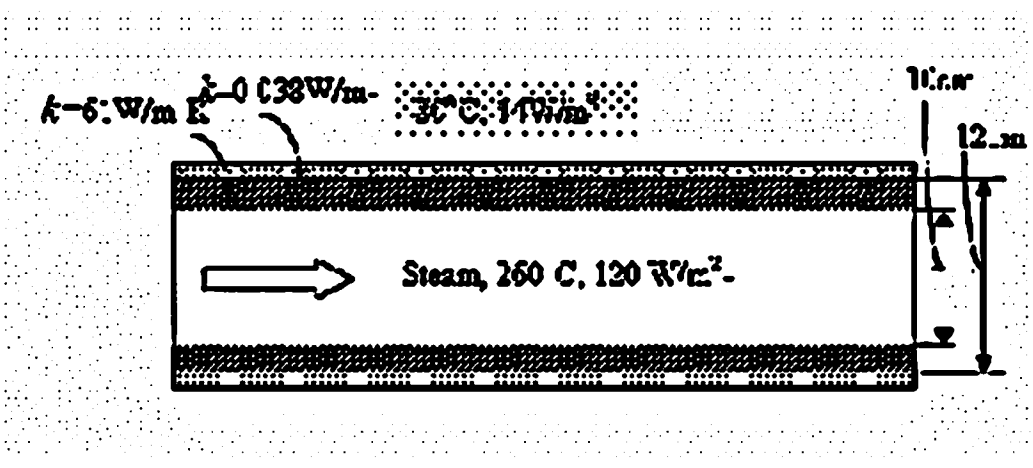
Inside steam temperature of the pipe, $T_{\infty,1} = 260°\text{C}$

Surroundings temperature, $T_{\infty,2} = 20°\text{C}$

Heat transfer coefficient on the outer surface, $h_o = 14\,\text{W/m}^2\cdot\text{K}$

Heat transfer coefficient on the inner surface, $h_i = 120\,\text{W/m}^2\cdot\text{K}$

Step 2 of 10



Step 3 of 10

The inner and outer surface areas of the insulate pipe per unit length are

$$A_i = \pi d_1 L$$
$$= \pi\times(0.1)\times(1)$$
$$= 0.31416\,\text{m}^2$$
$$A_o = \pi d_2 L$$
$$= \pi\times 0.12\times(1)$$
$$= 0.377\,\text{m}^2$$

The individual thermal resistances are evaluated as:

Convection resistance from steam to inner surface of the pipe

$$R_i = \frac{1}{h_i A_i}$$
$$= \frac{1}{120\times 0.31416}$$
$$= 0.0265\,\text{K/W}$$

Conduction resistance in the steel pipe

$$R_1 = R_{pipe}$$
$$= \frac{\ln(r_2/r_1)}{2\pi kL}$$
$$= \frac{\ln(0.06/0.05)}{2\pi\times 61\times 1}$$
$$= 0.0004757\,\text{K/W}$$

Convection resistance from the outer surface of the pipe

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{14\times 0.377}$$
$$= 0.189466\,\text{K/W}$$

Step 4 of 10

All the resistances are in series, and the total resistance is

$$R_{total} = R_i + R_1 + R_o$$
$$= 0.0265 + 0.0004757 + 0.189466$$
$$= 0.21644\,\text{K/W}$$

Then the steady rate of heat loss from the steam per unit length becomes

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{260-20}{0.21644}$$
$$= 1108.852\,\text{W}$$

$$\boxed{\dot{Q} = 1108.852\ \text{W}}$$

Step 5 of 10

a) The thickness of insulation needed to reduce the heat loss by 95% can be determined as follows

The amount of heat lost with insulation can be calculated as

$$\dot{Q} = 1108.852\times(1-0.95)$$
$$= 55.4426\,\text{W}$$

Step 6 of 10

The total resistance can be determined as

$$R_{total} = \frac{(T_{\infty,1} - T_{\infty,2})}{\dot{Q}}$$
$$= \frac{260-20}{55.4426}$$
$$= 4.328\,\text{K/W}$$

Then the total thermal resistance with insulation becomes

$$= \frac{260-20}{55.4426}$$
$$= 4.328\,\text{K/W}$$

Then the total thermal resistance with insulation becomes

$$R_{total} = R_i + R_1 + R_{insulation} + R_o$$
$$R_{insulation} = R_{total} - (R_i + R_1 + R_o)$$
$$= 4.328 - 0.21644$$
$$= 4.11156\,\text{K/W}$$

Step 7 of 10

We know thermal resistance of the insulation can be written as

$$R_2 = R_{insulation}$$
$$4.11156 = \frac{\ln(r_3/r_2)}{2\pi k_2 L}$$
$$= \frac{\ln(r_3/0.06)}{2\pi \times 0.038 \times 1}$$
$$\ln(r_3/0.06) = 0.9816$$
$$r_3 = 0.16012\,\text{m}$$

Therefore the thickness of insulation can be calculated as

$$t_{insulation} = r_3 - r_2$$
$$= 0.16012 - 0.06$$
$$= 0.10012\,\text{m}$$
$$= 10.012\,\text{cm}$$

$$\boxed{t_{insulation} = 10.012\,\text{cm}}$$

Step 8 of 10

b) The thickness of insulation needed to reduce the exposed surface temperature of insulated pipe to 40°C, can be determined as

The amount of heat lost with insulation can be calculated as

$$\dot{Q} = \frac{(T_{\infty,1} - T_{\infty,2})}{R_{total}}$$
$$= \frac{(T_2 - T_{\infty,2})}{R_o}$$
$$\frac{(260-20)}{R_{total}(\text{K/W})} = \frac{(40-20)}{R_o(\text{K/W})}$$
$$R_{total} = R_o \times \frac{240}{20}$$
$$R_{total} = 12R_o$$

Where R_{total} and R_o can be represented as

$$R_o = \frac{1}{h_o A_o}$$
$$= \frac{1}{14 \times \pi \times d_3 \times 1}$$
$$= \frac{0.02274}{d_3}\,\text{K/W}$$
$$R_{total} = R_i + R_1 + R_{insulation} + R_o$$
$$= 0.0265 + 0.0004757 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1} + \frac{0.02274}{d_3}$$
$$R_{total} = 0.0269 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1} + \frac{0.02274}{d_3}$$

Step 9 of 10

Substituting $R_{total} = 12R_o$ in the above equation

$$R_{total} = 0.0269 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1} + \frac{0.02274}{d_3}$$
$$12R_o = 0.0269 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1} + \frac{0.02274}{d_3}$$
$$12 \times \frac{0.02274}{d_3} = 0.0269 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1} + \frac{0.02274}{d_3}$$
$$\frac{0.25014}{d_3} = 0.0269 + \frac{\ln(d_3/0.12)}{2\pi \times 0.038 \times 1}$$
$$\frac{0.25014}{d_3} = 0.0269 + \frac{\ln(d_3/0.12)}{0.2387}$$

By solving the above equation we can obtain the outer diameter of the insulated pipe, which is given by

$$d_3 = 0.1695\,\text{m}$$

Step 10 of 10

Therefore, the thickness of insulation becomes

$$t_{insulation} = r_3 - r_2$$
$$= \frac{d_3}{2} - \frac{d_2}{2}$$
$$= \frac{0.1695}{2} - \frac{0.12}{2}$$
$$= 0.08475 - 0.06$$
$$= 0.02475\,\text{m}$$
$$= 2.475\,\text{cm}$$

$$\boxed{t_{insulation} = 2.475\,\text{cm}}$$

Question: 17.139P

When the transportation of natural gas in a pipeline is not feasible for economic or other reasons, if is first liquefied at about -160°C, and then transported in specially insulated tanks placed in marine ships. Consider a 4-m-diameter spherical tank that is filled with liquefied natural gas (LNG) at - 160°C. The tank is exposed to ambient air at 24°C with a heat transfer coefficient of 22 W/m2·K. The tank is thin-shelled and its temperature can be taken to be the same as the LNG temperature. The tank is insulated with 5-cm-thick super insulation that has an effective thermal conductivity of 0.00008 W/m·K. Taking the density and the specific heat of LNG to be 425 kg/m3 and 3.475 kJ/kg·°C, respectively, estimate how long it will take for the LNG temperature to rise to -150°C

Step 1 of 10

Given:

Diameter of spherical tank $D_1 = 4$ m

Temperature of LNG $T_1 = -160°C$

Temperature of ambient air $T_\infty = 24°C$

Heat transfer coefficient of ambient air $h_o = 22\ W/m^{2.0}C$

Thickness of the insulation is $t = 5$ cm

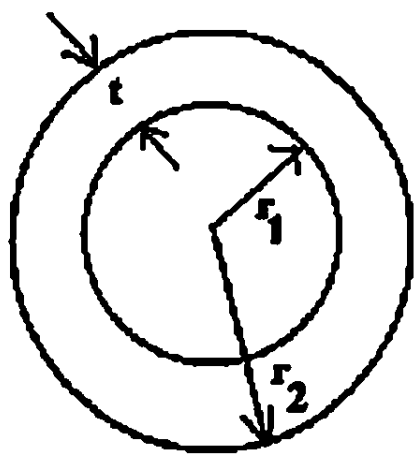
Density of LNG $\rho = 425\ kg/m^3$

Specific heat of LNG $C = 3.475\ kJ/kg.°C$

Thermal conductivity of super insulation $k = 0.00008\ W/m.°C$

Temperature rise of LNG is $T_2 = -150^0C$

Step 2 of 10



Step 3 of 10

Outer and inner surface areas of the spherical tank are

$$A_1 = \pi D_1^2$$
$$= \pi(4)^2$$
$$A_1 = 50.265\ m^2$$
$$A_2 = \pi D_2^2$$
$$= \pi(D_1 + 2t)^2$$
$$= \pi(4 + 2\times 0.05)^2$$
$$= \pi(4.1)^2$$
$$A_2 = 52.81\ m^2$$

Step 4 of 10

Volume of LNG tank is

$$V_1 = \frac{4}{3}\pi r_1^3$$
$$V_1 = \pi D_1^3 / 6$$
$$= \pi(4)^3 / 6$$
$$V_1 = 33.51\ m^3$$

Mass of LNG is

$$m = \rho V_1$$
$$= (425)(33.51)$$
$$m = 14241.75\ kg$$

Step 5 of 10

Thermal resistance for this spherical system due to conduction is

$$R_1 = \frac{r_2 - r_1}{4\pi k r_1 r_2}$$
$$= \frac{(2.05 - 2)}{4\pi(0.00008)(2)(2.05)}$$
$$R_1 = 12.13°C/W$$

Step 6 of 10

Thermal resistance due to convection from the outer surface to the atmospheric air

$$R_o = \frac{1}{h_o A_2}$$
$$= \frac{1}{(22)(52.81)}$$
$$R_o = 0.00086\ °C/W$$

Step 7 of 10

Total resistance becomes

$$R_t = R_o + R_1$$
$$= 0.00086 + 12.13$$
$$R_t = 12.13086°C/W$$

Step 8 of 10

Heat transfer rate for the LNG can be calculated as

$$\dot{Q} = \frac{T_\infty - T_1}{R_t}$$
$$= \frac{[24 - (-160)]}{12.13086}$$
$$\dot{Q} = 15.1679\ W$$

Step 9 of 10

Total amount of heat added to increase the temperature of LNG from -160°C to -150°C is

$$Q = mC\Delta T$$
$$= (14241.75)(3.475)[(-150) - (-160)]$$
$$Q = 494900.812\ kJ$$

Step 10 of 10

Time taken for LNG to rise from -160°C to -150°C is

$$\Delta t = \frac{Q}{\dot{Q}}$$
$$= \frac{494900.812}{0.0151679}$$
$$= 32628169.49\ s$$
$$= 32628169.49 \times \frac{1}{3600} \times \frac{1}{24}$$

$$\boxed{\Delta t = 377.6408\ days}$$

Question: 17.140P

A 15-cm × 20-cm hot surface at 85°C is to be cooled by attaching 4-cm-long aluminum (k = 237 W/m·K) fins of 2-mm × 2-mm square cross section. The temperature of surrounding medium is 25°C and the heat transfer coefficient on the surfaces can be taken to be 20 W/m2·K. If it is desired to triple the rate of heat transfer from the bare hot surface, determine the number of fins that needs to be attached.

Step 1 of 8

Given: -

Temperature of hot surface T_s = 85°C

Dimensions of hot surface is $15\,\text{cm} \times 20\,\text{cm}$

Length of fin L = 4 cm =0.04m

Area of cross-section of fin is 2 mm × 2 mm

Temperature of surrounding medium T_∞ = 25°C

Heat transfer coefficient on the surfaces h = 20 W/m^2.°C

Thermal conductivity of the aluminum fins k = 237 W/m°C

Effectiveness of the fin is 3

Step 2 of 8

As cross-section area of fins remains constant

We know

$$m = \sqrt{\frac{hp}{kA_c}}$$

$$= \sqrt{\frac{h \times 4a}{ka^2}}$$

Where a side of the fin

$$= \sqrt{\frac{4(20) \times (0.002)}{(237)(0.002)^2}}$$

$$m = 12.99\ \text{m}^{-1}$$

Step 3 of 8

Efficiency of the fin is

$$\eta_f = \frac{\tanh mL}{mL}$$

$$= \frac{\tanh(12.99\ \text{m}^{-1} \times 0.04\ \text{m})}{12.99\ \text{m}^{-1} \times 0.04\ \text{m}}$$

$$\eta_f = 0.9187$$

Step 4 of 8

Let number of fins will be n_f

Heat transfer area for finned surface is

$$A_f = n_f \times 4 \times (0.002)(0.04)$$

$$A_f = 0.00032 n_f\ \text{m}^2$$

Heat transfer area of unfinned surface is

$$A_{uf} = (0.15)(0.20) - n_f(0.002)(0.002)$$

$$A_{uf} = 0.03 - 0.000004 n_f\ \text{m}^2$$

Step 5 of 8

We know efficiency of the fin is ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature

$$\eta_f = \frac{\dot{Q}_f}{hA_f(T_s - T_\infty)}$$

$$\dot{Q}_f = \eta_f h A_f (T_s - T_\infty)$$

$$= 0.9187 \times (20) \times (0.00032 n_{fin})(85 - 25)$$

$$\dot{Q}_f = 0.35278 n_{fin}\ \text{W}$$

Step 6 of 8

The amount of heat transfer rate from unfinned fin is

$$\dot{Q}_{uf} = hA_{uf}(T_s - T_\infty)$$

$$= (20)(0.03 - 0.000004 n_f)(85 - 25)$$

$$\dot{Q}_{uf} = 36 - 0.0048 n_f\ \text{W}$$

Total rate of heat transfer

$$\dot{Q}_{total} = \dot{Q}_f + \dot{Q}_{uf}$$

$$\dot{Q}_{total} = 0.35278 n_{fin} + 36 - 0.0048 n_{fin}$$

$$\dot{Q}_{total} = (0.34798 n_{fin} + 36)\ \text{W}$$

Step 7 of 8

Area of the base surface without fin

$$A_{nf} = (0.15)(0.20)$$

$$A_{nf} = 0.03\ \text{m}^2$$

Heat transfer rate without fins

$$\dot{Q}_{nf} = hA_{nf}(T_b - T_s)$$

$$= (20)(0.03)(85 - 25)$$

$$\dot{Q}_{nf} = 36\ \text{W}$$

Step 8 of 8

Overall effectiveness of fins can be used to calculate total number of fins

We know effectiveness of fin is ratio rate of heat transfer with fin to rate of heat transfer without fin

$$\varepsilon_{fin} = \frac{\dot{Q}_f}{\dot{Q}_{nf}}$$

$$3 = \frac{0.34798 n_f + 36}{36}$$

$$n_f = 206.908$$

The total number of fins attached should be

$$\boxed{n_f = 207}$$

Question: 17.141P

Reconsider Prob. 17-140. Using EES (or other) software, plot the number of fins as a function of the increase in the heat loss by fins relative to no fin case (i.e., overall effectiveness of the fins) in the range of 1.5 to 5. Discuss the results. Is it realistic to assume the heal transfer coefficient to remain constant?

Step 1 of 3

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
A_surface=0.15*0.20 "[m^2]"
T_b=85 "[C]"
k=237 "[W/m-C]"
side=0.002  "[m]"
L=0.04 "[m]"
T_infinity=25 "[C]"
h=20 "[W/m^2-C]"
"epsilon_fin=3 parameter to be varied"

"ANALYSIS"
A_c=side^2
p=4*side
a=sqrt((h*p)/(k*A_c))
eta_fin=tanh(a*L)/(a*L)
A_fin=n_fin*4*side*L
A_unfinned=A_surface-n_fin*side^2
Q_dot_finned=eta_fin*h*A_fin*(T_b-T_infinity)
Q_dot_unfinned=h*A_unfinned*(T_b-T_infinity)
Q_dot_total_fin=Q_dot_finned+Q_dot_unfinned
Q_dot_nofin=h*A_surface*(T_b-T_infinity)
epsilon_fin=Q_dot_total_fin/Q_dot_nofin
```

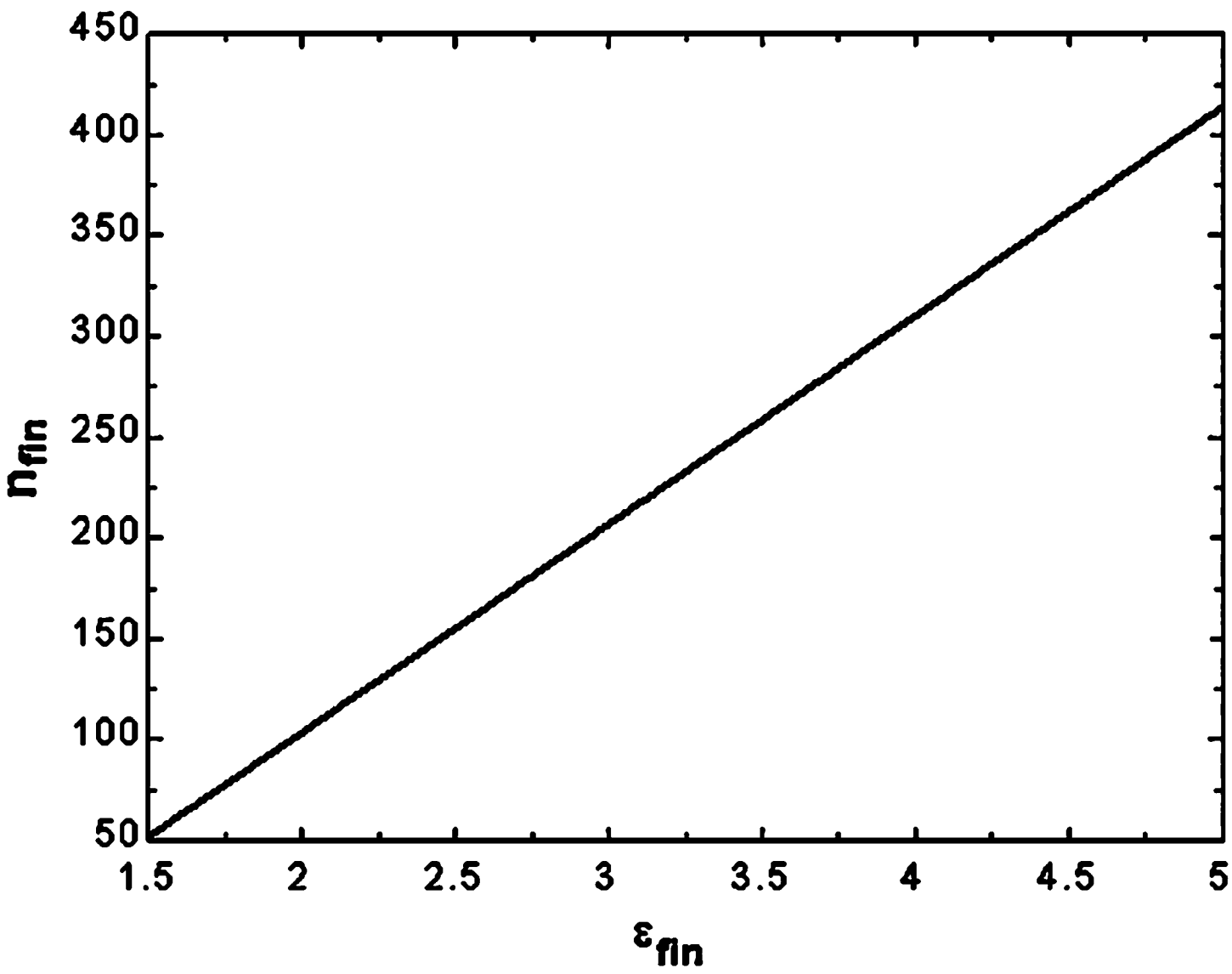
Step 2 of 3

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 15
Select variables in equations as ε_{fin} and n_{fin} press ADD and then press ok
A table will appear put values of ε_{fin} as 1.5,1.75,2....5 in run 1to 15
Then from Calculate -> Solve table
Solution will appear

ε_{fin}	n_{fin}
1.5	51.72
1.75	77.59
2	103.4
2.25	129.3
2.5	155.2
2.75	181
3	206.9
3.25	232.8
3.5	258.6
3.75	284.5
4	310.3
4.25	336.2
4.5	362.1
4.75	387.9
5	413.8

Step 3 of 3

Then from Plot -> New plot window -> X-Y Plot
Select X-axis ε_{fin} and Y-axis n_{fin}
Press OK
Graph will be plotted.



Question: 17.142P

An agitated vessel is used for heating 500 kg/min of an aqueous solution at 15°C by saturated steam condensing in the jacket outside the vessel. The vessel can hold 6200 kg of the aqueous solution. It is fabricated from 15-mm-thick sheet of 1.0 percent carbon steel (k = 43 W/m·K). and it provides a heat transfer area of 12.0 m. The heat transfer coefficient due to agitation is 5.5 kW/m2·K. while the steam condensation at 115°C in the jacket gives a heal transfer coefficient of 10.0 k W/m2·K. All properties of the aqueous solution are comparable to those of pure water. Calculate the temperature of the outlet stream in steady operation.

Step 1 of 4

Given data:

Mass flow rate of aqueous solution, $\dot{m} = 500\ \text{kg/min}$

Mass capacity of aqueous solution, $m = 6200\,\text{kg}$

Thickness of the steel sheet, $t = 15\,\text{mm}$

Heat transfer area of steel sheet, $A = 12\,\text{m}^2$

Thermal conductivity of the steel sheet, $k = 43\,\text{W/m-K}$

Temperature of aqueous solution, $T_{in} = 15°\text{C}$

Temperature of the steam, $T_{steam} = 115°\text{C}$

Heat transfer coefficient due to the agitation, $h_i = 5.5\,\text{kW/m}^2\text{-K}$

Heat transfer coefficient of jacket, $h_i = 10\,\text{kW/m}^2\text{-K}$

Step 2 of 4

The heat balance applied for the system under steady operation gives

$$\dot{m}c_p(T - T_{in}) = UA(T_{steam} - T)$$

We know specific heat of water is $C_p = 4180\,\text{J/kg-K}$

The thermal resistance due to convection from inside of the vessel is

$$R_i = \frac{1}{h_i}$$
$$= \frac{1}{5.5\times10^3}$$
$$= \frac{1}{5500}$$

The thermal resistance due to conduction because of thickness of vessel

$$R_1 = \frac{t}{k}$$
$$= \frac{0.015}{43}$$
$$= \frac{0.015}{43}$$

Step 3 of 4

Thermal resistance due to convection from outside of the vessel

$$R_o = \frac{1}{h_o}$$
$$= \frac{1}{10\times10^3}$$
$$= \frac{1}{10000}$$

Overall heat transfer coefficient, U can be calculated as

$$U = \frac{1}{\Sigma R}$$
$$= \frac{1}{R_i + R_1 + R_o}$$
$$= \frac{1}{\frac{1}{5500} + \frac{0.015}{43} + \frac{1}{10000}}$$
$$= 1585.652\,\text{W/m}^2\text{-K}$$

Step 4 of 4

Substituting the known values in heat balance equation, gives

$$\dot{m}c_p(T - T_{in}) = UA(T_{steam} - T)$$
$$\frac{500}{60}\times4180\times(T-15) = 1585.652\times12(115-T)$$
$$1.8306\times(T-15) = (115-T)$$
$$2.8306T = 142.459$$
$$T = 50.328°\text{C}$$

Therefore, the temperature of the outlet stream T in steady operation is $\boxed{T = 50.328°\text{C}}$

Question: 17.143P pg 1

A typical section of a building wall is shown in Fig. P17-143. This section extends in and out of the page and is repeated in the vertical direction. The wall support members are made of steel (k = 50 W/m·K).The support members are 8 cm ($t23$) × 0.5 cm (LB). The remainder of the inner wall space is filled with insulation (k = 0.03 W/m·K) and measures 8 cm ($t23$) × 60 cm (LB). The inner wall is made of gypsum board (k = 0.5 W/m·K) that is 1 cm thick ($t12$) and the outer wall is made of brick (K = 1.0W/m·K)that is 10 cm thick ($t34$). What is the temperature on the interior brick surface. 3, when $T1$ = 20°C and $T4$ = 35°C?

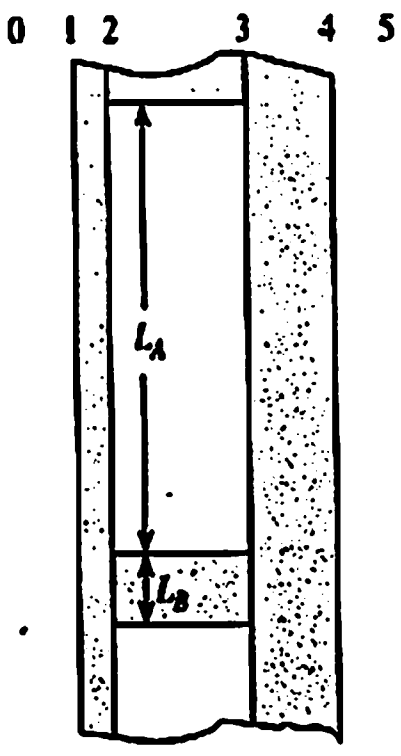
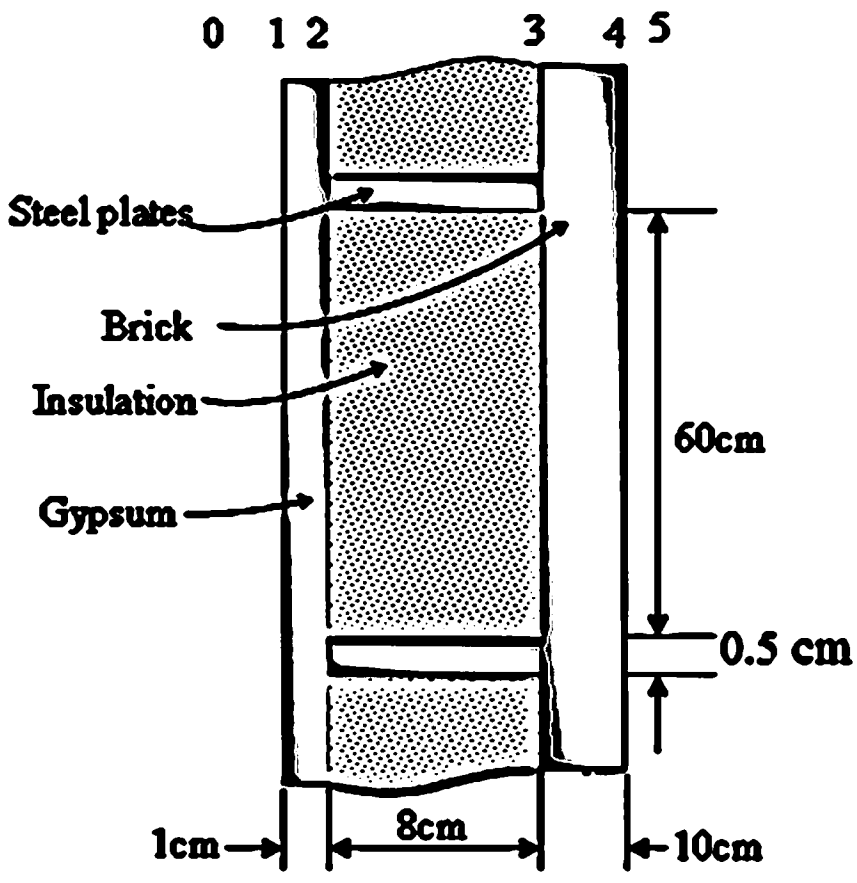


FIGURE P17–143

Step 1 of 10

Draw the schematic layout of building wall:



Step 2 of 10

Find the thermal resistance from 1 to 2:

$$R_{12} = \frac{t_{12}}{k_t A}$$

Here, R_{12} is the thermal resistance, t_{12} is the thickness of the portion, k_t is the thermal conductivity of gypsum board, A is the cross-sectional area of wall.

Find the thermal resistance of the insulation:

$$R_A = \frac{t_A}{k_A A_A}$$

Here, R_A is the thermal resistance, t_A is the thickness of the portion of insulation, k_A is the thermal conductivity of insulation, A_A is the cross-sectional area of insulation.

Step 3 of 10

Find the thermal resistance of steel:

$$R_B = \frac{t_B}{k_B A_B}$$

Here, R_B is the thermal resistance, t_B is the thickness of the portion of steel, k_B is the thermal conductivity of insulation, A_B is the cross-sectional area of steel.

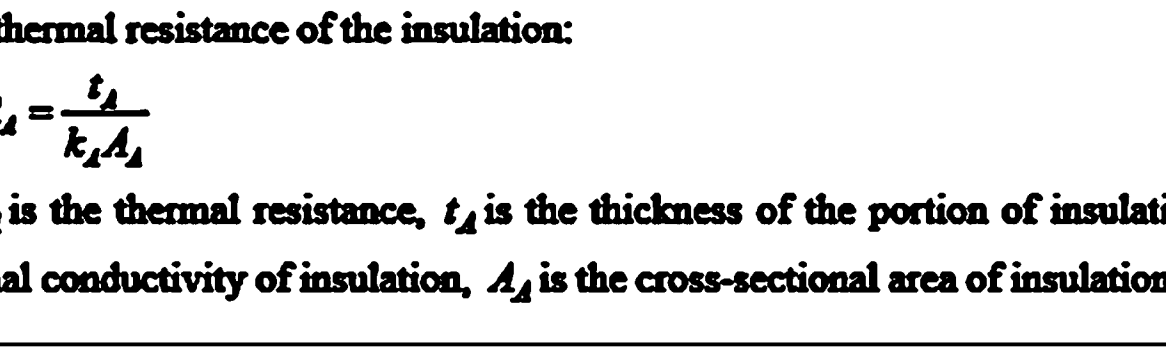
Find the thermal resistance from 3 to 4:

$$R_{34} = \frac{t_{34}}{k_o A}$$

Here, R_{34} is the thermal resistance, t_{34} is the thickness of the portion of brick, k_o is the thermal conductivity of brick, A is the cross-sectional area of wall.

Step 4 of 10

Draw the thermal network circuit of the building wall:



Step 5 of 10

By using the thermal network approach, find the total resistance:

$$R_{total} = R_{12} + \frac{R_A R_B}{R_A + R_B} + R_{34}$$

Substitute, $\frac{t_{12}}{k_t A}$ for R_{12}, $\frac{t_A}{k_A A_A}$ for R_A, $\frac{t_B}{k_B A_B}$ for R_B and $\frac{t_{34}}{k_o A}$ for R_{34}

$$R_{total} = \frac{t_{12}}{k_t A} + \frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A} + \frac{t_B}{k_B A_B}} + \frac{t_{34}}{k_o A}$$

17 143P pg 2

$$R_{total}=\frac{t_{12}}{k_i A}+\frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A}+\frac{t_B}{k_B A_B}}+\frac{t_{34}}{k_o A}$$

$$R_{total}\times A=\frac{t_{12}}{k_i}+A\left(\frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A}+\frac{t_B}{k_B A_B}}\right)+\frac{t_{34}}{k_o} \quad \text{..... (1)}$$

Step 6 of 10

Find the cross-sectional area of wall:

$$A=(L_A+L_B)w$$

Here, L_A is the length of the insulation filled, L_B is the length of steel and w is the width of the wall.

Find the cross-sectional area of insulation:

$$A_A=L_A w$$

Here, L_A is the length of the insulation filled and w is the width of the wall.

Find the cross-sectional area of steel:

$$A_B=L_B w$$

Here, L_B is the length of steel and w is the width of the wall.

Step 7 of 10

Substitute, $(L_A+L_B)w$ for A, $L_A w$ for A_A and $L_B w$ for A_B in equation (1)

$$R_{total}A=\frac{t_{12}}{k_i}+A\left[\frac{\left(\frac{t_A}{k_A A_A}\right)\left(\frac{t_B}{k_B A_B}\right)}{\frac{t_A}{k_A A_A}+\frac{t_B}{k_B A_B}}\right]+\frac{t_{34}}{k_o}$$

$$=\frac{t_{12}}{k_i}+(L_A+L_B)w\left(\frac{\left(\frac{t_A}{k_A L_A w}\right)\left(\frac{t_B}{k_B L_B w}\right)}{\frac{t_A}{k_A L_A w}+\frac{t_B}{k_B L_B w}}\right)+\frac{t_{34}}{k_o}$$

$$=\frac{t_{12}}{k_i}+(L_A+L_B)\left(\frac{\left(\frac{t_A}{k_A L_A}\right)\left(\frac{t_B}{k_B L_B}\right)}{\frac{t_A}{k_A L_A}+\frac{t_B}{k_B L_B}}\right)+\frac{t_{34}}{k_o}$$

Step 8 of 10

Substitute, $1\,\text{cm}=0.01\,\text{m}$ for t_{12}, $0.5\,\text{W/m-K}$ for k_i, $60\,\text{cm}=0.6\,\text{m}$ for L_A, $0.5\,\text{cm}=0.005\,\text{m}$ for L_B, $8\,\text{cm}=0.08\,\text{m}$ for t_A, $8\,\text{cm}=0.08\,\text{m}$ for t_B, $0.03\,\text{W/m-K}$ for k_A, $50\,\text{W/m-K}$ for k_B, $10\,\text{cm}=0.1\,\text{m}$ for t_{34} and $1\,\text{W/m-K}$ for k_o

$$R_{total}A=\frac{0.01}{0.5}+(0.6+0.005)\times\left(\frac{\left[\frac{0.08}{(0.03\times 0.6)}\right]\times\left[\frac{0.08}{(50\times 0.005)}\right]}{\left[\frac{0.08}{(0.03\times 0.6)}\right]+\left[\frac{0.08}{(50\times 0.005)}\right]}\right)+\frac{0.1}{1}$$

$$=0.3006\,\text{m}^2\text{-K/W}$$

Step 9 of 10

Find the average heat flux through the wall using the equation:

$$\frac{\dot{Q}}{A}=\frac{(T_4-T_1)}{R_{total}A}$$

Here, $\frac{\dot{Q}}{A}$ is the average heat flux, T_4 is the temperature of the wall at point 4, T_1 is the temperature of the wall at point 1.

Substitute, $35°\text{C}$ for T_4, $20°\text{C}$ for T_1 and $0.3006\,\text{m}^2\text{-K/W}$ for $R_{total}A$.

$$\frac{\dot{Q}}{A}=\frac{35-20}{0.3006}$$
$$=49.9\,\text{W/m}^2$$

Step 10 of 10

Find the temperature on the interior brick surface, 3, from the equation:

$$\dot{Q}=\frac{(T_4-T_3)}{R_{34}}$$

$$\dot{Q}=\frac{(T_4-T_3)}{\left(\frac{t_{34}}{k_o A}\right)}$$

$$\frac{\dot{Q}}{A}=\frac{(T_4-T_3)k_o}{t_{34}}$$

Substitute, $49.9\,\text{W/m}^2$ for $\frac{\dot{Q}}{A}$, $35°\text{C}$ for T_4, $10\,\text{cm}=0.1\,\text{m}$ for t_{34} and $1\,\text{W/m-K}$ for k_o

$$49.9=\frac{(35-T_3)1}{0.1}$$
$$T_3=30.01°\text{C}$$

Therefore, the temperature on the interior brick surface is $\boxed{30.01°\text{C}}$

Question: 17.144P

A total of 10 rectangular aluminum fins (k = 203 W/m·K) are placed on the outside flat surface of an electronic device. Each fin is 100 mm wide, 20 mm high and 4 mm thick. The fins are located parallel to each other at a center-to-center distance of 8 mm. The temperature at the outside surface of the electronic device is 72°C. The air is at 20°C, and the heat transfer coefficient is 80 W/m2·K. Determine *(a)* the rate of heat loss from the electronic device to the surrounding air and *(b)* the fin effectiveness.

Step 1 of 7

Given data:

Number of fins, $n=10$

Width of the fin, $w=100\,\text{mm}$

Height of the fin, $L=20\,\text{mm}$

Thickness of the fin, $t=4\,\text{mm}$

Thermal conductivity of the aluminum fin, $k=203\,\text{W/m}\cdot\text{K}$

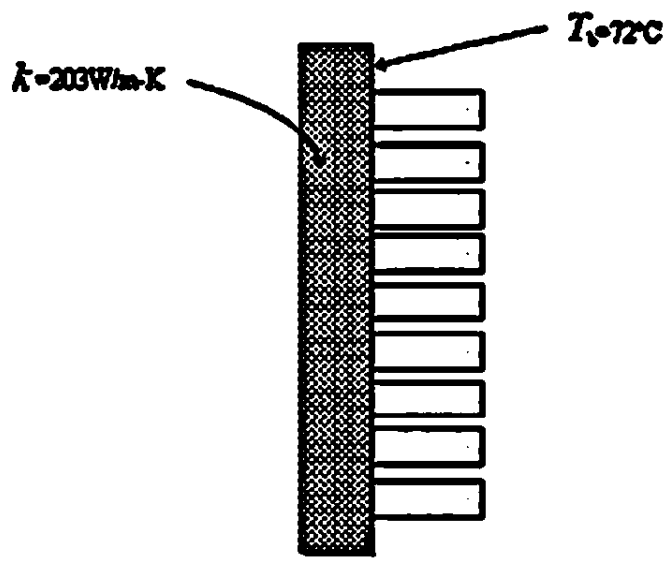
Outer surface temperature of the circuit board, $T_b=72°\text{C}$

Surroundings air temperature, $T_\infty=20°\text{C}$

Heat transfer coefficient on the outer surface, $h=80\,\text{W/m}^2\cdot\text{K}$

Step 2 of 7

The figure representing flat surface having 10 rectangular Aluminum fins,



Step 3 of 7

(a)

The fin efficiency can be determined by finding these parameters

$$\xi = L_c^{3/2}\sqrt{h/kA_p}$$
$$= L_c^{3/2}\sqrt{h/kL_ct}$$
$$= L_cL_c^{1/2}\sqrt{h/kL_ct}$$
$$= L_c\sqrt{h/kt}$$
$$= (L+t/2)\sqrt{h/kt}$$

By substituting the known values in the above expression

$$\xi = (0.02+0.004/2)\times\sqrt{\frac{80}{203\times0.004}}$$
$$= 0.2184$$

Step 4 of 7

From the graph efficiency of straight fins of rectangular profiles at $\xi=0.2184$, the efficiency of the fin is obtained as

$$\eta_{fin}=0.97$$

The surface area of the fin can be determined as

$$A_{fin}=2n(Lt+Lw)$$
$$=2\times10\times(0.02\times0.004+0.02\times0.1)$$
$$=0.0416\,\text{m}^2$$

Area at the base of the fin,

$$A_b=n(wt)$$
$$=10\times(0.1\times0.004)$$
$$=0.004\,\text{m}^2$$

Step 5 of 7

The rate of heat loss from the fin tip can be determined as

$$\dot{Q}_{fin}=\eta_{fin}hA_{fin}(T_b-T_\infty)$$
$$=0.97\times80\times0.0416\times(72-20)$$
$$=167.864\,\text{W}$$

The rate of heat loss from the base surface of the fin can be determined as

$$\dot{Q}_{base}=hA_b(T_b-T_\infty)$$
$$=80\times0.004\times(72-20)$$
$$=16.64\,\text{W}$$

Step 6 of 7

Total heat loss becomes,

$$\dot{Q}_{total}=\dot{Q}_{fin}+\dot{Q}_{base}$$
$$=167.864+16.64$$
$$=184.504\,\text{W}$$

Therefore, the total heat loss from the electronic device to the surrounding air is

$$\boxed{\dot{Q}_{total}=184.504\,\text{W}}$$

Step 7 of 7

(b)

The fin effectiveness can be determined as the ratio of the heat transfer rate with fin to the heat transfer rate without fin

$$\varepsilon_{fin}=\frac{\dot{Q}_{total}}{\dot{Q}_{base}}$$
$$=\frac{184.504}{16.64}$$
$$=11.088$$

Therefore, the fin effectiveness is $\boxed{\varepsilon_{fin}=11.088}$.

Question: 17.145P pg 1

One wall of a refrigerated warehouse is 10.0-m-high and 5.0-m-wide. The wall is made of three layers: 1.0-cm-thick aluminum (k = 200 W/m·K), 8.0-cm-thick fibreglass (k = 0.038 W/m·K), and 3.0-cm thick gypsum board (k = 0.48 W/m-K). The warehouse inside and outside temperatures are -10°C and 20°C, respectively, and (he average value of both inside and outside heat transfer coefficients is 40 W/m2·K.

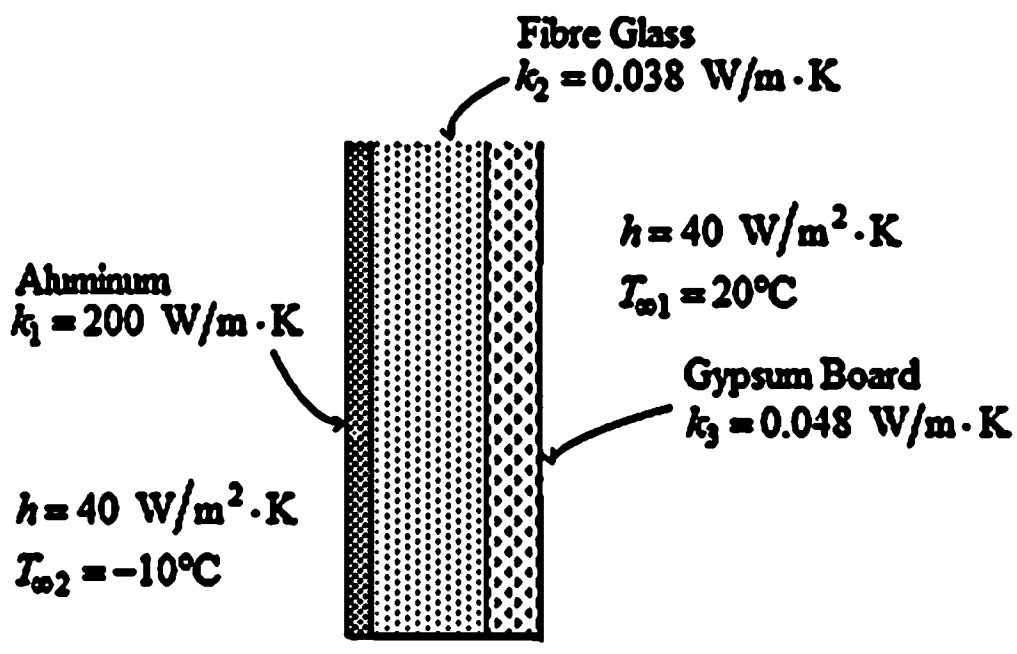
(a) Calculate the rate of heat transfer across the warehouse wall in steady operation.

(*b*) Suppose that 400 metal bolts (k = 43 W/m·K), each 2.0 cm in diameter and 12.0 cm long, are used to fasten (i.e., hold together) the three wall layers. Calculate the rate of heat transfer for the "bolted"wall,

(*c*) What is the percent change in the rate of heat transfer across the wall due to metal bolts?

Step 1 of 16

Draw the schematic diagram of the refrigerated warehouse:



Step 2 of 16

(a)

Find the cross-sectional area for the wall of a refrigerated warehouse:

$$A = L \times w$$

Here, A is the cross-sectional area, L is the length of the wall and w is the width of the wall.

Substitute, 10 m for L and 5 m for w.

$$A = 10 \times 5$$
$$= 50 \text{ m}^2$$

Step 3 of 16

Calculate the thermal resistances at entrance and exit:

$$R = \frac{1}{hA}$$

Here, R is the thermal resistance at entrance and exit, h is the heat transfer coefficient.

Substitute, 50 m^2 for A and 40 W/m^2·K for h.

$$R = \frac{1}{40 \times 50}$$
$$= 0.0005 \text{ K/W}$$

Step 4 of 16

Find the resistance due to the aluminum layer:

$$R_1 = \frac{t_1}{k_1 A}$$

Here, t_1 is the thickness of the aluminum layer, k_1 is the thermal conductivity of aluminum.

Substitute, 1cm = 0.01 m for t_1, 200 W/m·K for k_1 and 50 m^2 for A.

$$R_1 = \frac{0.01}{200 \times 50}$$
$$= 0.000001 \text{K/W}$$

Step 5 of 16

Find the resistance due to fiberglass:

$$R_2 = \frac{t_2}{k_2 A}$$

Here, t_2 is the thickness of the fiberglass layer, k_2 is the thermal conductivity of fiberglass.

Substitute, 8cm = 0.08 m for t_2, 0.038 W/m·K for k_2 and 50 m^2 for A.

$$R_2 = \frac{0.08}{0.038 \times 50}$$
$$= 0.042105 \text{K/W}$$

Step 6 of 16

Find the resistance due to gypsum board:

$$R_3 = \frac{t_3}{k_3 A}$$

Here, t_3 is the thickness of the gypsum board, k_3 is the thermal conductivity of gypsum.

Substitute, 3cm = 0.03 m for t_3, 0.48 W/m·K for k_3 and 50 m^2 for A.

$$R_3 = \frac{0.03}{0.48 \times 50}$$
$$= 0.00125 \text{K/W}$$

Step 7 of 16

Find the total resistance of the refrigerated warehouse wall:

$$R_{total} = R + R_1 + R_2 + R_3 + R$$

Substitute, 0.0005 K/W for R, 0.000001K/W for R_1, 0.042105K/W for R_2 and 0.00125K/W for R_3.

$$R_{total} = 0.0005 + 0.000001 + 0.042105 + 0.00125 + 0.0005$$
$$= 0.044356 \text{K/W}$$

Step 8 of 16

Find the total rate of heat transfer:

$$\dot{Q} = \frac{(T_{\infty 1} - T_{\infty 2})}{R_{total}}$$

Here, $T_{\infty 2}$ is the temperature inside the refrigerated warehouse and $T_{\infty 1}$ is the temperature

Step 8 of 16

Find the total rate of heat transfer:

$$\dot{Q}=\frac{(T_{\infty 1}-T_{\infty 2})}{R_{total}}$$

Here, $T_{\infty 2}$ is the temperature inside the refrigerated warehouse and $T_{\infty 1}$ is the temperature outside the refrigerated warehouse.

Substitute, $-10°C$ for $T_{\infty 2}$, $20°C$ for $T_{\infty 1}$ and $0.044356 K/W$ for R_{total}.

$$\dot{Q}=\frac{(20-(-10))}{0.044356}$$
$$=676.342\,W$$

Therefore, the total rate of heat transfer is $\boxed{676.342\,W}$

Step 9 of 16

(b)

To fasten the three layers, 400 metal bolts are used.

The rate of heat transfer through the wall with the consideration of bolts can be determine as follows,

Calculate the total cross-sectional area of the 400 bolts:

$$A_{bolt}=n\frac{\pi}{4}d^2$$

Here, n is the number of bolts and d is the diameter of the bolt.

Substitute, 400 for n and 2.0 cm = 0.02 m for d.

$$A_{bolt}=400\times\frac{\pi}{4}\times 0.02^2$$
$$=0.125664\,m^2$$

Step 10 of 16

Calculate the area of the wall without the bolts:

$$A_{wall}=A-A_{bolt}$$

Here, A is the area of the original area of the wall.

Substitute, $50\,m^2$ for A and $0.125664\,m^2$ for A_{bolt}

$$A_{wall}=50-\left(400\times\frac{\pi}{4}\times 0.02^2\right)$$
$$=49.87434\,m^2$$

Step 11 of 16

Calculate the total resistance of the wall:

$$R_{wall}=R+R_1+R_2+R_3+R$$
$$=\frac{1}{hA_{wall}}+\frac{t_1}{k_1A_{wall}}+\frac{t_2}{k_2A_{wall}}+\frac{t_3}{k_3A_{wall}}+\frac{1}{hA_{wall}}$$
$$=\frac{1}{A_{wall}}\left[\frac{1}{h}+\frac{t_1}{k_1}+\frac{t_2}{k_2}+\frac{t_3}{k_3}+\frac{1}{h}\right]$$

Substitute, $49.87434\,m^2$ for A_{wall}, $40\,W/m^2\cdot K$ for h, $1cm=0.01\,m$ for t_1, $200\,W/m\cdot K$ for k_1, $8cm=0.08\,m$ for t_2, $0.038\,W/m\cdot K$ for k_2, $3cm=0.03\,m$ for t_3, $0.48\,W/m\cdot K$ for k_3

$$R_{wall}=\frac{1}{49.87434}\left[\frac{1}{40}+\frac{0.01}{200}+\frac{0.08}{0.038}+\frac{0.03}{0.48}+\frac{1}{40}\right]$$
$$=0.044468\,K/W$$

Step 12 of 16

Find the heat transfer through the wall by using the equation:

$$\dot{Q}_1=\frac{(T_{\infty 1}-T_{\infty 2})}{R_{wall}}$$

Substitute, $0.044468\,K/W$ for R_{wall}, $20°C$ for $T_{\infty 1}$ and $-10°C$ for $T_{\infty 2}$

$$\dot{Q}_1=\frac{(20+10)}{0.044468(K/W)}$$
$$=674.642\,W$$

Step 13 of 16

Find the total resistance of the bolts:

$$R_{bolt}=\frac{t}{kA_{bolt}}+\left(2\times\frac{1}{hA_{bolt}}\right)$$

Substitute, $0.125664\,m^2$ for A_{bolt}, $40\,W/m^2\cdot K$ for h, $43\,W/m\cdot K$ for k.

$$R_{bolt}=\frac{0.12}{43\times 0.125664}+\left(2\times\frac{1}{40\times 0.125664}\right)$$
$$=0.0222+0.3978$$
$$=0.4201K/W$$

Step 14 of 16

Find the heat transfer through the bolts:

$$\dot{Q}_2=\frac{(T_{\infty 1}-T_{\infty 2})}{R_{bolt}}$$

Substitute, $0.4201\,K/W$ for R_{bolt}, $20°C$ for $T_{\infty 1}$ and $-10°C$ for $T_{\infty 2}$

$$\dot{Q}_2=\frac{(20+10)}{0.4201(K/W)}$$
$$=71.4126\,W$$

Step 15 of 16

Find the total heat transfer rate through wall with bolts:

$$\dot{Q}_{total}=\dot{Q}_1+\dot{Q}_2$$

Substitute, $674.642\,W$ for $\dot{Q}_1$ and $71.4126\,W$ for $\dot{Q}_2$

$$\dot{Q}_{total}=674.642+71.4126$$
$$=746.055\,W$$

Therefore, the total rate of heat transfer for the bolted wall is $\boxed{746.055\,W}$

It can be observed that, the heat transfer rate increased due to fixing of bolts in wall.

Step 16 of 16

It can be observed that, the heat transfer rate increased due to fixing of bolts in wall.

Step 16 of 16

(c)

Calculate the percentage change in the rate of heat transfer across the wall due to metal bolts:

$$\%\text{change} = \frac{\dot{Q}_{total} - \dot{Q}}{\dot{Q}} \times 100$$

Substitute, 746.055 W for $\dot{Q}_{total}$ and 676.342 W for $\dot{Q}$

$$\%\text{change} = \frac{746.055 - 676.342}{676.342} \times 100$$

$$= 10.31\%$$

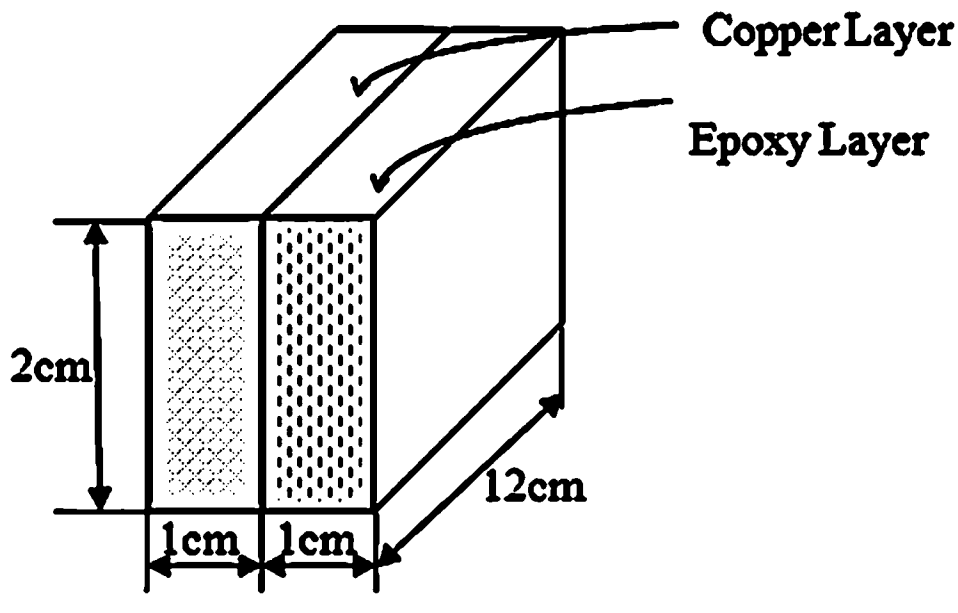
Therefore, by the addition of bolts, the heat transfer rate increases by $\boxed{10.31\%}$

Question: 17.146P pg 1

A 12-cm-long bar with a square cross-section, as shown in Fig. P17-146, consists of a 1-cm-thick copper layer (k = 380 W/m·K) and a 1-cm-thick epoxy composite layer (k = 0.4 W/m·K). Calculate the rate of heat transfer under a thermal driving force of 50°C, when thc direction of steady one-dimensional heat transfer is *(a)* from front to back (i.e., along its length), *(b)* from left to right, and *(c)* from top to bottom.

Step 1 of 7

Draw the figure as follows:



Step 2 of 7

(a)

Along the length direction, determine the rate of heat transfer as follows:

Write the equation for thermal resistance of copper bar as follows:

$$R_1 = \frac{L}{k_c A}$$

Here, L is the length of the bar, k_c is the thermal conductivity of the copper, and A is the area of the surface normal to the heat transfer.

Substitute 0.12 m for L, $380\ \text{W/m}\cdot\text{K}$ for k_c, and $(0.02\times0.01)\ \text{m}^2$ for A.

$$R_1 = \frac{0.12}{380\times0.02\times0.01}$$
$$= 1.57895\ \text{K/W}$$

Write the equation for epoxy layer bar as follows:

$$R_2 = \frac{L}{k_e A}$$

Here, k_e is the thermal conductivity of the epoxy layer bar.

$$R_2 = \frac{0.12}{0.4\times0.02\times0.01}$$
$$= 1500\ \text{K/W}$$

Step 3 of 7

Since the two resistances are in parallel, calculate the total resistance as follows:

$$R_{total} = \frac{R_1 R_2}{R_1 + R_2}$$

Substitute $1.57895\ \text{K/W}$ for R_1 and $1500\ \text{K/W}$ for R_2.

$$R_{total} = \frac{1.57895\times1500}{1.57895+1500}$$
$$= 1.5773\ \text{K/W}$$

Calculate the rate of heat transfer as follows:

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$

Here, ΔT is the temperature difference.

Substitute $50°\text{C}$ for ΔT and $1.5773\ \text{K/W}$ for R_{total}.

$$\dot{Q} = \frac{50}{1.5773(\text{K/W})}$$
$$= 31.7\ \text{W}$$

Therefore, the heat transfer from front to back is $\boxed{31.7\ \text{W}}$.

Step 4 of 7

(b)

From left to the right, calculate the rate of heat transfer as follows:

Calculate the thermal resistance of copper bar as follows:

$$R_1 = \frac{L}{k_c A}$$

Substitute 0.01 m for L, $380\ \text{W/m}\cdot\text{K}$ for k_c, and $(0.02\times0.12)\ \text{m}^2$ for A.

$$R_1 = \frac{0.01}{380\times0.02\times0.12}$$
$$= 0.011\ \text{K/W}$$

Calculate the thermal resistance of epoxy layer bar as follows:

$$R_2 = \frac{L}{k_e A}$$

Substitute 0.01 m for L, $0.4\ \text{W/m}\cdot\text{K}$ for k_e, and $(0.02\times0.12)\ \text{m}^2$ for A.

$$R_2 = \frac{0.01}{0.4\times0.02\times0.12}$$
$$= 10.4167\ \text{K/W}$$

Substitute 0.01 m for L, $0.4\ \text{W/m}\cdot\text{K}$ for k_1, and $(0.02\times0.12)\ \text{m}^2$ for A.

$$R_2 = \frac{0.01}{0.4\times0.02\times0.12}$$
$$= 10.4167\,\text{K/W}$$

Step 5 of 7

Since the two resistances are in series, calculate the total resistance as follows:

$$R_{total} = R_1 + R_2$$

Substitute $0.011\,\text{K/W}$ for R_1 and $10.4167\,\text{K/W}$ for R_2.

$$R_{total} = 0.011(\text{K/W}) + 10.4167(\text{K/W})$$
$$= 10.4276\,\text{K/W}$$

Calculate the rate of heat transfer as follows:

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$

Substitute $10.4276\,\text{K/W}$ for R_{total} and $50°\text{C}$ for ΔT.

$$\dot{Q} = \frac{50}{10.4276(\text{K/W})}$$
$$= 4.8\,\text{W}$$

Therefore, the rate of heat transfer is $\boxed{4.8\,\text{W}}$.

Step 6 of 7

From the top to bottom direction, calculate the rate of heat transfer as follows:

Calculate the thermal resistance of copper wire as follows:

$$R_1 = \frac{L}{k_c A}$$

Substitute 0.02 m for L, $380\ \text{W/m}\cdot\text{K}$ for k_c, and $(0.01\times0.12)\ \text{m}^2$ for A.

$$R_1 = \frac{0.02(\text{m})}{380(\text{W/m-K})\times0.01(\text{m})\times0.12(\text{m})}$$
$$= 0.04386\,\text{K/W}$$

Calculate the thermal resistance of epoxy layer as follows:

$$R_2 = \frac{L}{kA}$$

Substitute 0.02 m for L, $0.4\ \text{W/m}\cdot\text{K}$ for k_e, and $(0.01\times0.12)\ \text{m}^2$ for A.

$$R_2 = \frac{0.02(\text{m})}{0.4(\text{W/m-K})\times0.01(\text{m})\times0.12(\text{m})}$$
$$= 41.667\,\text{K/W}$$

Step 7 of 7

Since the two resistances are in parallel, calculate the total resistance as follows:

$$R_{total} = \frac{R_1 R_2}{R_1 + R_2}$$

Substitute $0.04386\,\text{K/W}$ for R_1 and $41.667\,\text{K/W}$ for R_2.

$$R_{total} = \frac{0.04386(\text{K/W})\times41.667(\text{K/W})}{0.04386(\text{K/W}) + 41.667(\text{K/W})}$$
$$= 0.04381\,\text{K/W}$$

Calculate the heat transfer rate as follows:

$$\dot{Q} = \frac{\Delta T}{R_{total}}$$

Substitute $50°\text{C}$ for ΔT and $0.04381\,\text{K/W}$ for R_{total}.

$$\dot{Q} = \frac{50}{0.04381(\text{K/W})}$$
$$= 1141.2\,\text{W}$$

Therefore, the heat transfer rate is $\boxed{1141.2\,\text{W}}$.

CHAPTER 18
SOLUTIONS

Question: 18.001P

What is lumped system analysis? When is it applicable?

Step 1 of 1

The bodies in which the internal temperature will be uniform throughout the heat transfer process are treated as "Lump". The temperature of the lump is expressed as the function of time and the calculation of the heat transfer with respect to this temperature is defined as "Lumped System analysis".

It is applicable when the Biot number (the ratio of conduction resistance within the body to convection resistance at the surface of the body) is less than or equal to 0.1.

$$Bi = \frac{hL_c}{K}$$

Therefore, lumped system analysis is applicable when Biot number is less than 0.1.

Question: 18.002P

For which solid is the lumped system analysis more likely to be applicable: an actual apple or a golden apple of the same size? Why?

Step 1 of 1

The lumped system analysis is more likely to be applicable for a golden apple than for an actual apple since the thermal conductivity is much larger and thus the Biot number is much smaller for gold.

Question: 18.003P

For which kind of bodies made of the same material is the lumped system analysis more likely to be applicable: slender ones or well-rounded ones of the same volume? Why?

Step 1 of 1

The lumped system analysis is more likely to be applicable to slender bodies than the well-rounded bodies since the characteristic length (ratio of volume to surface area) and thus the Biot number is much smaller for slender bodies.

Question: 18.004P

Consider heat transfer between two identical hot solid bodies and the air surrounding them. The first solid is being cooled by a fan while the second one is allowed to cool naturally. For which solid is the lumped system analysis more likely to be applicable? Why?

Step 1 of 1

The lumped system analysis is more likely to be applicable for the body cooled naturally.

As we know the Biot number is proportional to the convection heat transfer coefficient, and it is proportional to the air velocity. Therefore, the Biot number is more likely to be less than 0.1 for the case of natural convection. When Biot no is less than 0.1 then only we can apply lumped analysis.

Question: 18.005P

Consider heat transfer between two identical hot solid bodies and their environments. The first solid is dropped in a large container filled with water, while the second one is allowed to cool naturally in the air. For which solid is the lumped system analysis more likely to be applicable? Why?

Step 1 of 1

The lumped system analysis is more likely to be applicable for the body allowed to cool in the air since the Biot number is proportional to the convection heat transfer coefficient, which is larger in water than it is in air because of the larger thermal conductivity of water. Therefore, the Biot number is more likely to be less than 0.1 for the case of the solid cooled in the air. So solid which is cooled in natural air is more likely to be applicable.

Question: 18.006P

Consider a hot baked potato on a plate. The temperature of the potato is observed to drop by 4°C during the first minute. Will the temperature drop during the second minute be less than, equal to, or more than 4°C? Why?

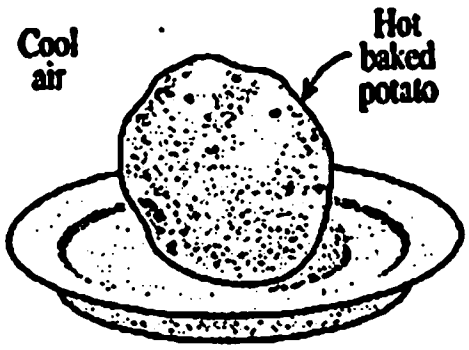


FIGURE P18-6C

Step 1 of 1

The temperature drop of the potato during the second minute will be less than 4°C, since the temperature of a body approaches the temperature of the surrounding

Question:

Consider a hot baked potato on a plate. The temperature of the potato is observed to drop by 4°C during

Question: 18.007P

Consider a potato being baked in an oven that is maintained at a constant temperature. The temperature of the potato is observed to rise by 5°C during the first minute. Will the temperature rise during the second minute be less than, equal to, or more than 5°C? Why?

Step 1 of 1

The temperature rise of the potato during the second minute will be less than 5°C since the temperature of a body approaches the temperature of the surrounding medium asymptotically, and thus it changes rapidly at the beginning, but slowly later on

Question: 18.008P

What is the physical significance of the Biot number? Is the Biot number more likely to be larger for highly conducting solids or poorly conducting ones?

Step 1 of 1

Biot number represents the ratio of conduction resistance within the body to convection resistance at the surface of the body.

The Biot number is more likely to be larger for poorly conducting solids since such bodies have larger resistances against heat conduction.

Question: 18.009P

Consider two identical 4-kg pieces of roast beef. The first piece is baked as a whole, while the second is baked after being cut into two equal pieces in the same oven. Will there be any difference between the cooking limes of the whole and cut roasts? Why?

Step 1 of 1

The heat transfer is proportional to the surface area. Two half pieces of the roast have a much larger surface area than the single piece and thus a higher rate of heat transfer. As a result, the two half pieces will cook much faster than the single large piece

Question: 18.010P

Consider a sphere and a cylinder of equal volume made of copper. Both the sphere and the cylinder are initially at the same temperature and are exposed to convection in the same environment. Which do you think will cool faster, the cylinder or the sphere? Why?

Step 1 of 2

The lumped parameter analysis yields the following equation:

$$\frac{(T-T_0)}{(T_\infty-T_0)}=\exp\left[-\frac{hA}{\rho Vc}t\right]$$

Here, T_0 is the initial temperature, T_∞ is the temperature of the ambient, T is the temperature at a prescribed time, h is the heat transfer coefficient, A is the surface area, ρ is the density of the material, V is the volume of the material, c is the specific heat and t is the time taken for heat transfer.

Differentiate the equation with respect to time.

$$\frac{dT}{dt}=(T_\infty-T_0)\left(-\frac{hA}{\rho Vc}\right)\exp\left[-\frac{hA}{\rho Vc}t\right]$$

$$\frac{dT}{dt}\propto\frac{A}{V}$$

Step 2 of 2

For a sphere:

$$\left(\frac{A}{V}\right)_{sphere}=\frac{4\pi r^2}{\frac{4}{3}\pi r^3}$$

$$\left(\frac{A}{V}\right)_{sphere}=\frac{3}{r}$$

For a cylinder:

$$\left(\frac{A}{V}\right)_{cylinder}=\frac{2\pi rh}{\pi r^2 h}$$

$$\left(\frac{A}{V}\right)_{cylinder}=\frac{2}{r}$$

It is obvious that:

$$\left(\frac{A}{V}\right)_{sphere}>\left(\frac{A}{V}\right)_{cylinder}$$

$$\left(\frac{dT}{dt}\right)_{sphere}>\left(\frac{dT}{dt}\right)_{cylinder}$$

Therefore, **the sphere will cool faster than the cylinder**.

Question: 18.011P

In what medium is the lumped system analysis more likely to be applicable: in water or in air? Why?

Step 1 of 1

Write the expression for Biot number.

$$Bi = \frac{\text{Conduction resistance}}{\text{Convection resistance}}$$

$$= \frac{\frac{L}{Ak}}{\frac{1}{hA}}$$

$$= \frac{hL}{k}$$

Here, h is convection heat transfer coefficient and k is thermal conductivity.

The lumped system analysis is applicable when Biot number is less than 0.1.

From the above expression, Biot number is proportional to convection heat transfer coefficient and inversely proportional to thermal conductivity. The convection coefficient is more in water than in air because of larger thermal conductivity of water.

Hence, the Biot number is less in case of solid cooled in the air when compared to solid cooled in the water.

Therefore, **the Biot number is much smaller in air.**

Question: 18.012P

To warm up some milk for a baby, a mother pours milk into a thin-walled glass whose diameter is 6 cm. The height of the milk in the glass is 7 cm. She then places the glass into a large pan filled with hot water at 70°C. The milk is stirred constantly, so that its temperature is uniform at all times. If the heat transfer coefficient between the water and the glass is 120 W/m2 -K, determine how long it will take for the milk to warm up from 3°C to 38°C. Take the properties of the milk to be the same as those of water. Can the milk in this case be treated as a lumped system? Why?

Step 1 of 5

Given data:

Diameter of the thin glass is $D = 6\text{cm}$.

Radius of the thin glass is $r = 3\text{cm}$.

Heat transfer coefficient between water and glass is $h = 120\ \text{W/m}^2 \cdot \text{K}$.

Temperature of the hot water is $T_\alpha = 70^\circ\text{C}$.

Initial temperature of the milk is $T_i = 3^\circ\text{C}$.

Final temperature of the milk is $T(t) = 38^\circ\text{C}$.

Step 2 of 5

Assumption:

We use lumped analysis for this problem as temperature of milk is assumed to remain constant always.

Thermal properties of milk are assumed to be as close as that of water and properties are constant.

Characteristic length is:

$$L_c = \frac{V}{A}$$

$$= \frac{\pi r^2 h}{(2\pi r h) + (2\pi r^2)}$$

$$= \frac{\pi \times 0.03^2 \times 0.07}{(2\pi \times 0.03 \times 0.07) + (2\pi \times 0.03^2)}$$

$$= 0.0105\text{m}$$

Step 3 of 5

Then, Biot number is

$$Bi = \frac{hL_c}{K}$$

$$= \frac{120 \times 0.0105}{0.607}$$

$$= 2.076$$

Hence Biot number is greater than 0.1, so lumped system analysis not applicable. But in this problem we are using the lumped system analysis because of temperature remains uniform due to constant stirring.

Step 4 of 5

Then,

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = e^{-bt}$$

Where

't' is the time taken for warming up.

Here:

$$b = \frac{h}{\rho C_p L_c}$$

$$= \frac{120}{998 \times 4182 \times 0.0105}$$

$$= 0.00274\ s^{-1}$$

Step 5 of 5

Substituting the values we get,

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = e^{-bt}$$

$$\frac{38 - 70}{3 - 70} = e^{-0.00274t}$$

$$\ln\left(\frac{38 - 70}{3 - 70}\right) = -0.00274t$$

$$t = 4.5\ \text{min}$$

Therefore, time taken for warming up milk is $\boxed{t = 4.5\ \text{min}}$

As milk is stirred constantly, the temperature of the milk remains constant almost all the time even if the Biot number is much larger. So lumped system analysis is quite appropriate for this problem.

Question: 18.013P

Repeat Prob. 18-12 for the case of water also being stirred, so that the heat transfer coefficient is doubled to 240W/m2 ·K.

Step 1 of 5

Given data:

Diameter of the glass, $D = 6\text{cm}$

Radius of the glass, $r = 3\text{cm}$

Heat transfer coefficient between water and glass, $h = 240\ \text{W/m}^2\cdot\text{K}$

Height of the milk in glass, $H = 7\text{cm}$

Temperature of hot water, $T_a = 70°\text{C}$

Initially temperature of the milk, $T_i = 3°\text{C}$

Final temperature of the milk, $T(t) = 38°\text{C}$

Step 2 of 5

Let us assume the temperature of the milk is $T = 25°\text{C}$

Characteristic length of the thin walled glass is given as

$$L_c = \frac{V}{A}$$

$$= \frac{\left(\pi r^2 H\right)}{\left(2\pi rH\right)+\left(2\pi r^2\right)}$$

$$= \frac{\pi\times 0.03^2\times 0.07}{\left(2\pi\times 0.03\times 0.07\right)+\left(2\pi\times 0.03^2\right)}$$

$$= 0.0105\text{m}$$

Step 3 of 5

From the table of "Properties of saturated water"

At $T = 25°\text{C}$

Specific heat of water is $c_p = 4182\text{kJ/kg}\cdot\text{K}$

Density of the water is $\rho = 998\text{kg/m}^3$

Thermal conductivity is $k = 0.607\ \text{W/m}\cdot\text{K}$

Then, Biot number of the process is

$$Bi = \frac{hL_c}{k}$$

$$= \frac{240\times L_c}{0.607}$$

$$= \frac{240\times 0.0105}{0.607}$$

$$= 4.15$$

In this problem lumped analysis can't be used because of Biot number is greater than 0.1 But as temperature remains uniform due to constant stirring process. We use the lumped system analysis.

Step 4 of 5

Then, lumped system analysis expression is given as

$$\frac{T(t)-T_a}{T_i-T_a} = e^{-bt}$$

Where

't' is the time taken for warming up.

Here:

$$b = \frac{h}{\rho c_p L_c}$$

$$= \frac{240}{998\times 4182\times 0.0105}$$

$$= 0.00548\text{s}^{-1}$$

Step 5 of 5

Substitute the known values in above expression, we get

$$\frac{T(t)-T_a}{T_i-T_a} = e^{-bt}$$

$$\frac{38-70}{3-70} = e^{-0.00548t}$$

$$\ln\left(\frac{38-70}{3-70}\right) = -0.00548t$$

$$t = 134.846\text{sec}$$

$$= 2.25\text{min}$$

Therefore, time taken for warming up is $\boxed{t = 2.25\ \text{min}}$.

As milk is stirred constantly, the temperature of the milk remains constant almost all the time even if the Biot number is much larger. So lumped system analysis is quite appropriate for this problem.

Question: 18.014P

Obtain a relation for the time required for a lumped system to reach the average temperature $\frac{1}{2}(Ti + T\infty)$ where Ti is the initial temperature and $T\infty$ is the temperature of the environment.

Step 1 of 3

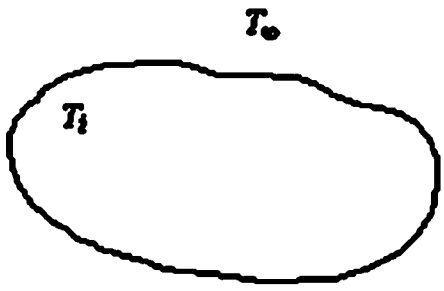
Initial temperature of lumped system $= T_i$

Environment temperature $= T_\infty$

Average temperature of lumped system $= \frac{(T_i + T_\infty)}{2}$

The relation for time period for a lumped system to reach the average temperature $\frac{(T_i + T_\infty)}{2}$ can be determined as

Step 2 of 3



Step 3 of 3

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{\frac{T_i + T_\infty}{2} - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{T_i - T_\infty}{2(T_i - T_\infty)} = e^{-bt}$$

$$\frac{1}{2} = e^{-bt}$$

$$-bt = -\ln 2$$

$$t = \frac{\ln 2}{b}$$

the time $\boxed{t = \frac{0.693}{b}}$

Question: 18.015P

The temperature of a gas stream is to be measured by a thermocouple whose junction can be approximated as a 1.2-mm-diameter sphere. The properties of the junction are k = 35 W/m·K, ρ = 8500 kg/m3 and cp = 320 J/kg·K, the heat transfer coefficient between the junction and the gas is h = 110 W/m2 ·K. Determine how long it will take for the thermocouple to read 99 percent of the initial temperature difference.

Step 1 of 5

Given data:

Diameter of the sphere, $D = 1.2\text{mm} = 0.0012\text{m}$

Heat transfer coefficient between junction and gas, $h = 110\ \text{W/m}^2\cdot\text{K}$

Reading of the thermocouple to initially temperature, 99%

Properties of junction are:

Thermal conductivity, $k = 35\ \text{W/m}\cdot\text{K}$

Density, $\rho = 8500\ \text{kg/m}^3$

Specific heat, $c_p = 320\ \text{J/kg}\cdot\text{K}$

Step 2 of 5

Characteristic length is of the thermocouple is given as

$$L_c = \frac{V}{A}$$
$$= \frac{\left(\frac{\pi D^3}{6}\right)}{\pi D^2}$$
$$= \frac{\pi \times \frac{0.0012^3}{6}}{\pi \times 0.0012^2}$$
$$= 0.0002\text{m}$$

Step 3 of 5

Then, Biot number of the process given as

$$Bi = \frac{hL_c}{K}$$
$$= \frac{110 \times L_c}{35}$$
$$= \frac{110 \times 0.0002}{35}$$
$$= 3.7 \times 10^{-3}$$

Here Biot number is less than the 0.1, therefore lumped system analysis is applicable.

Step 4 of 5

From the expression of lumped system analysis, we have

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = e^{-bt}$$

As the thermocouple reads 99 percent of the initial temperature difference, we put

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = 0.01$$

Here:

$$b = \frac{h}{\rho C_p L_c}$$
$$= \frac{110}{8500 \times 320 \times 0.0002}$$
$$= 0.2022\ s^{-1}$$

Step 5 of 5

Substituting the known values, we get

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = e^{-bt}$$
$$0.01 = e^{-0.2022t}$$
$$\ln(0.01) = -0.2022t$$
$$t = 22.8\ \text{sec}$$

Therefore, time taken to read the temperature is $\boxed{t = 22.8\ \text{sec}}$.

Question: 18.016P

In a manufacturing facility, 2-in-diameter brass balls (k = 64.1 Btu/h· ft·°F, ρ = 532 lbm/ft3 , and cp = 0.092 Btu/lbm·°F) initially at 250°F are quenched in a water bath at 120°F for a period of 2 min at a rate of 120 balls per minute . If the convection heat transfer coefficient is 42 Btu/h·ft2 ·F, determine (*a*) the temperature of the balls after quenching and (*b*) the rate at which heat needs to be removed from the water in order to keep its temperature constant at 120°F.

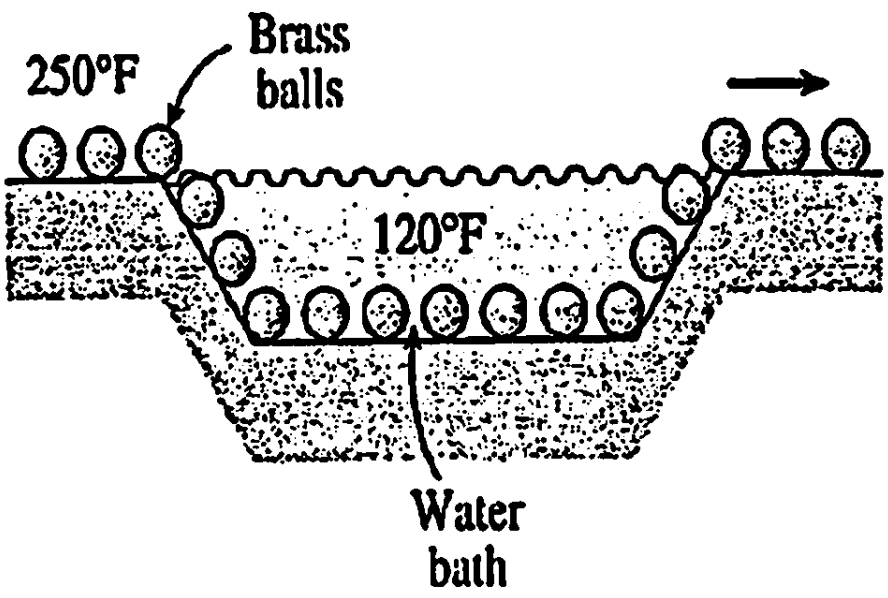


FIGURE PI8-16E

Step 1 of 5

Given:

Diameter of brass ball $D = 2\ in$

Thermal conductivity of brass $K = 64.1\ \frac{\text{Btu}}{\text{h.ft.°F}}$

Density of brass $\rho = 532\ \frac{\text{lbm}}{\text{ft}^3}$

Specific heat of brass $C_p = 0.092\ \frac{\text{Btu}}{\text{lbm.°F}}$

The balls are spherical in shape with a radius $r_o = 1\ \text{in}$

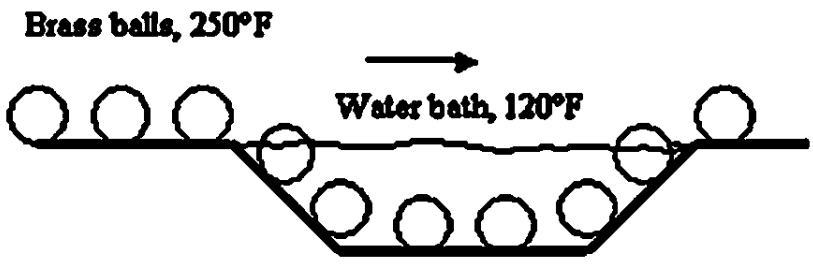
The initial temperature $T_i = 250°\text{F}$

The water bath temperature $T_\infty = 120°\text{F}$

Time $t = 120\ S$

Step 2 of 5

The thermal properties of the balls are constant. If the heat transfer coefficient is constant and uniform over the entire surface. Then the characteristic length and the Biot number for the brass balls are



Step 3 of 5

The characteristic length $L_c = \frac{V}{A_s}$

$$L_c = \frac{\pi D^3/6}{\pi D^2}$$

$$L_c = \frac{D}{6}$$

$$L_c = \frac{2/12\ \text{ft}}{6}$$

$$L_c = 0.02778\ \text{ft}$$

The Biot number $Bi = \frac{hL_c}{k}$

$$Bi = \frac{(42\ \text{Btu/h.ft}^2\text{.°F})(0.02778\ \text{ft})}{(64.1\ \text{Btu/h.ft.°F})}$$

$$Bi = 0.01820 < 0.1$$

Step 4 of 5

The lumped system analysis is applicable since Bi < 0.1. Then the temperature of the balls after quenching becomes

The value of exponent $b = \frac{hA_s}{\rho C_p V}$

$$b = \frac{h}{\rho C_p L_c}$$

$$b = \frac{42\ \text{Btu/h.ft}^2\text{.°F}}{(532\ \text{lbm/ft}^3)(0.092\ \text{Btu/lbm.°F})(0.02778\ \text{ft})}$$

$$b = 30.9\ \text{h}^{-1}$$

$$b = 0.00858\ s^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$\frac{T(t)-120}{250-120} = e^{-(0.00858\ s^{-1})(120\ s)}$$

$$T(t) = 166\ °\text{F}$$

Step 5 of 5

(*b*) The total amount of heat transfer from a ball during a 2-minute period is

The mass $m = \rho V$

$$m = \rho\frac{\pi D^3}{6}$$

$$m = (532\ \text{lbm/ft}^3)\frac{\pi(2/12\ \text{ft})^3}{6}$$

$$m = 1.290\ \text{lbm}$$

The heat transfer from a ball $Q = mC_p[T_i - T(t)]$

$$Q = (1.29\ \text{lbm})(0.092\ \text{Btu/lbm.°F})[250-166]°\text{F}$$

$$Q = 9.97\ \text{Btu}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{ball}$$

$$\dot{Q}_{total} = (120\ \text{balls/min}) \times (9.97\ \text{Btu})$$

$$\boxed{\dot{Q}_{total} = 1196\ \text{Btu/min}}$$

Therefore, heat must be removed from the water at a rate of 1196 Btu/min in order to keep its temperature constant at 120°F.

Question: 18.017P

Repeat Prob. 18-16E for aluminum balls.

Step 1 of 5

Calculate the characteristic length of the ball.

$$L_c = \frac{D}{6}$$

Here, D is the diameter of the ball.

Substitute 2 in. for D.

$$L_c = \frac{(2/12)\text{ ft}}{6}$$
$$= 0.02778\text{ ft}$$

Calculate the Biot number.

$$\text{Bi} = \frac{hL_c}{k}$$

Here, h is the heat transfer coefficient, k is the coefficient of thermal conductivity of brass, and L_c is the characteristic length of the ball.

Substitute $42\text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ for h, 0.02778 ft for L_c, and $64.1\text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$ for k.

$$\text{Bi} = \frac{(42)(0.02778)}{(64.1)}$$
$$= 0.0182$$

As $\text{Bi} < 0.1$, lumped system analysis is applicable.

Step 2 of 5

Calculate the value of exponent (b).

$$b = \frac{h}{\rho c_p L_c}$$

Here, ρ is the density of the brass ball, c_p is the specific heat at constant pressure.

Substitute 532 lbm/ft^3 for ρ, $0.092\text{ Btu/lbm}\cdot^\circ\text{F}$ for c_p, 0.02778 ft for L_c, and $42\text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ for h.

$$b = \frac{42}{(532)(0.092)(0.02778)}$$
$$= 30.889\text{ h}^{-1}$$
$$= 0.00858\text{ s}^{-1}$$

Step 3 of 5

Calculate the temperature of the balls after quenching using the following relation.

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

Here, T_i is the initial temperature of ball and T_∞ is the water bath temperature.

Substitute 0.00858 s^{-1} for b and 250°F for T_i, and 120°F for T_∞.

$$\frac{T(t)-120}{250-120} = e^{-(0.00858)(120)}$$
$$\frac{T(t)-120}{250-120} = 0.3571$$
$$T(t) = 166.43^\circ\text{F}$$

Therefore, the temperature of the balls after quenching is $\boxed{166.43^\circ\text{F}}$.

Step 4 of 5

(*b*)

Calculate the mass of each ball.

$$m = \rho V$$
$$= \rho\frac{\pi D^3}{6}$$

Substitute $532\frac{\text{lbm}}{\text{ft}^3}$ for ρ and 2 in. for D.

$$m = \left(532\text{ lbm/ft}^3\right)\frac{\pi(2/12\text{ ft})^3}{6}$$
$$= 1.2896\text{ lbm}$$

Step 5 of 5

Calculate the heat transfer from each ball.

$$Q_{\text{ball}} = mC_p\left[T_i - T(t)\right]$$

Substitute 1.2896 lbm for m, 250°F for T_i and 166.43°F for $T(t)$.

$$Q_{\text{ball}} = (1.2896)(0.092)(250-166.43)$$
$$= 9.915\text{ Btu}$$

Calculate the total rate of heat transfer from the balls to the water.

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}}Q_{\text{ball}}$$

Substitute 120 balls/min for $\dot{n}_{\text{ball}}$ and 9.915 Btu for Q_{ball}.

$$\dot{Q}_{\text{total}} = (120)\times(9.915)$$
$$= 1189.80\text{ Btu/min}$$

Therefore, the rate of heat needs to be removed from the water in order to keep its temperature constant is $\boxed{1189.80\text{ Btu/min}}$.

Question: 18.018P

Obtain relations for the characteristic lengths of a large plane wall of thickness 2L, a very long cylinder of radius *r0* , and a sphere of radius *r0* .

Step 1 of 4

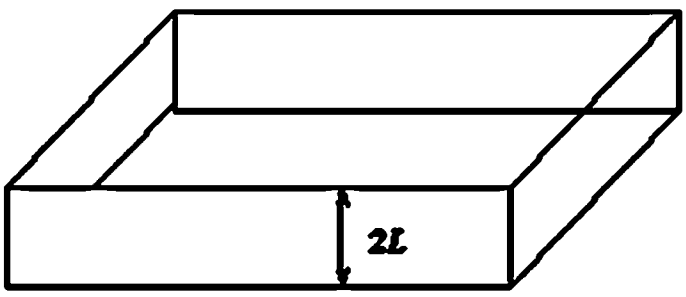
Thickness of large plane wall $= 2L$

Radius of long cylinder $= r$

Radius of sphere $= r_o$

Relations for the characteristic lengths of a large plane wall of thickness 2L, a very long cylinder of radius r_o and a sphere of radius r_o are

Step 2 of 4



The characteristic lengths of a large plane wall $L_{c,wall} = \dfrac{V}{A_{surface}}$

$$L_{c,wall} = \frac{2LA}{2A}$$

$$L_{c,wall} = L$$

Step 3 of 4



The characteristic lengths of a cylinder $L_{c,cylinder} = \dfrac{V}{A_{surface}}$

$$L_{c,cylinder} = \frac{\pi r_o^2 h}{2\pi r_o h}$$

$$L_{c,cylinder} = \frac{r_o}{2}$$

Step 4 of 4



The characteristic lengths of a sphere $L_{c,sphere} = \dfrac{V}{A_{surface}}$

$$L_{c,sphere} = \frac{\frac{4\pi r_o^3}{3}}{4\pi r_o^2}$$

$$L_{c,sphere} = \frac{r_o}{3}$$

Question: 18.019P

A long copper rod of diameter 2.0 cm is initially at a uniform temperature of 100°C. It is now exposed to an air stream at 20°C with a heat transfer coefficient of 200 W/m2 ·K. How long would it take for the copper road to cool to an average temperature of 25°C?

Step 1 of 4

Given:

Diameter of copper rod $D = 2\ \text{cm}$

Initial temperature of rod $T_i = 100°\text{C}$

Temperature of air stream $T_\infty = 20°\text{C}$

The average temperature $T(t) = 25°\text{C}$

Heat transfer coefficient $h = 200\ \dfrac{\text{W}}{\text{m}^2.°\text{C}}$

Step 2 of 4

From properties of solid metals take values

The thermal conductivity of copper $k = 401\ \text{W/m.°C}$

The density of copper $\rho = 8933\ \text{kg/m}^3$

The specific heat of the copper $C_p = 385\text{J/kg.°C}$

The thermal properties of the rod are constant. If the heat transfer coefficient is constant and uniform over the entire surface. Then the characteristic length and the Biot number for the copper rod are

Step 3 of 4

The characteristic length $L_c = \dfrac{V}{A_s}$

$$L_c = \frac{(\pi D^2/4)L}{\pi DL}$$

$$L_c = \frac{D}{4}$$

$$L_c = \frac{0.02m}{6}$$

$$L_c = 0.005m$$

The Biot number $Bi = \dfrac{hL_c}{k}$

$$Bi = \frac{(200W/\text{m}^2.°\text{C})(0.005m)}{(401\ \text{W/m.°C})}$$

$$Bi = 0.0025 < 0.1$$

Step 4 of 4

The lumped system analysis is applicable since Bi < 0.1. Then the temperature of the rod becomes

The value of exponent $b = \dfrac{hA}{\rho C_p V}$

$$b = \frac{h}{\rho C_p L_c}$$

$$b = \frac{200\ \text{W/m}^2.°\text{C}}{(8933\ \text{kg/m}^3)(385\ \text{J/kg.°C})(0.005\text{m})}$$

$$b = 0.01163\ s^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$\frac{25-20}{100-20} = e^{-(0.01163s^{-1})t}$$

$t = 238s$

$\boxed{t = 4.0\text{min}}$

Question:

Consider a sphere of diameter 5 cm, a cube of side length 5 cm, and a rectangular prism of dimension 4 cm × 5 cm × 6 cm, all initially at 0°C and all made of silver (k = 429 W/m2·K, ρ = 10,500 kg/m3 , Cp = 0.235 kJ/kg·K). Now all three of these geometries are exposed to ambient air at 33°C on all of their surfaces with a heat transfer coefficient of 12 W/m2 ·K. Determine how long it will take for the temperature of each geometry to rise to 25°C.

Step 1 of 12

For sphere:

Find the characteristic length for sphere using the equation:

$$L_c = \frac{V}{A}$$

$$= \frac{\left(\frac{\pi D^3}{6}\right)}{\pi D^2}$$

Here, V is the volume of the sphere, A is the area of the sphere and D is the diameter of the sphere.

Substitute, 5 cm = 0.05 m for D.

$$L_c = \frac{\left(\frac{\pi \times 0.05^3}{6}\right)}{\left(\pi \times 0.05^2\right)}$$

$$= 0.00833 \text{ m}$$

Step 2 of 12

Find the Biot number for sphere using the equation,

$$\text{Bi} = \frac{hL_c}{k}$$

Here, Bi is the Biot number, h is the convection heat transfer coefficient, L_c is the characteristic length and k is conduction heat transfer coefficient.

Substitute, 12 $\text{W/m}^2\cdot^\circ\text{C}$ for h, 0.00833 m for L_c and 429 $\text{W/m}\cdot^\circ\text{C}$ for k.

$$\text{Bi} = \frac{12 \times 0.00833}{429}$$

$$= 0.0002330$$

Here Biot number is less than 0.1, therefore lumped system analysis is applicable.

Step 3 of 12

Find the time period required to reach the temperature of 25 °C by using the equation,

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

Here, t is the time in second taken for heating, T_∞ is the temperature of the surroundings, T_i is the initial temperature, $T(t)$ is the temperature after t seconds and b is constant.

Find the constant b using the equation,

$$b = \frac{h}{\rho c_p L_c}$$

Here, ρ is the density of the sphere, c_p is the specific heat of the sphere at constant pressure.

Substitute, 12 $\text{W/m}^2\cdot^\circ\text{C}$ for h, 10500 kg/m^3 for ρ, 235 J/kg for c_p and 0.00833 m for L_c

$$b = \frac{12}{10500 \times 235 \times 0.00833}$$

$$= 0.00058 \text{ s}^{-1}$$

Step 4 of 12

Find the time period required to reach the temperature of 25 °C by,

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

Substitute, 25 °C for $T(t)$, 33°C for T_∞, 0°C for T_i and 0.00058 s^{-1} for b.

$$\frac{25 - 33}{0 - 33} = e^{-(0.00058)(t)}$$

$$t = 2443.22 \text{ s}\left|\frac{1 \text{ min}}{60 \text{ s}}\right|$$

$$= 40.72 \text{ min}$$

Therefore, in order to reach the temperature of 25°C, the sphere will take a time of

$\boxed{40.72 \text{ min.}}$

Step 5 of 12

For cube:

Find the characteristic length for cube by using the equation,

$$L_c = \frac{V}{A}$$

$$= \frac{L^3}{6L^2}$$

Here, V is the volume of cube, A is the surface area of cube and L is the length of the cube.

Substitute, 5 cm = 0.05 m for L.

$$L_c = \frac{0.05^3}{6 \times 0.05^2}$$

$$= 0.00833 \text{ m}$$

Step 6 of 12

Find the Biot number for cube using the equation,

$$\text{Bi} = \frac{hL_c}{k}$$

Substitute, 12 $\text{W/m}^2\cdot^\circ\text{C}$ for h, 0.00833 m for L_c and 429 $\text{W/m}\cdot^\circ\text{C}$ for k.

$$\text{Bi} = \frac{12 \times 0.00833}{429}$$

$$= 0.0002330$$

Here Biot number is less than 0.1, therefore lumped system analysis is applicable.

$$=\frac{429}{}$$
$$=0.0002330$$

Here Biot number is less than 0.1, therefore lumped system analysis is applicable.

Step 7 of 12

Find the time period required to reach the temperature of 25 °C by,

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt}$$

Here, t is the time in second taken for heating, T_∞ is the temperature of the surroundings, T_i is the initial temperature, $T(t)$ is the temperature after t seconds and b is constant.

Find the value of the constant b:

$$b=\frac{h}{\rho c_p L_c}$$

Substitute, 12 $W/m^2\cdot°C$ for h, 10500 kg/m^3 for ρ, 235 J/kg for c_p and 0.00833 m for L_c

$$b=\frac{12}{10500\times235\times0.00833}$$
$$=0.00058\,s^{-1}$$

Step 8 of 12

Find the time period required to reach the temperature of 25 °C by,

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt}$$

Substitute, 25 °C for $T(t)$, 33°C for T_∞, 0°C for T_i and 0.00058 s^{-1} for b.

$$\frac{25-33}{0-33}=e^{-(0.00058)\times(t)}$$
$$t=2443.22\text{ s}\left|\frac{1\text{ min}}{60\text{ s}}\right|$$
$$=40.72\text{ min}$$

Therefore, in order to reach the temperature of 25°C, the cube will take a time of $\boxed{40.72\,\text{min.}}$

Step 9 of 12

For rectangular prism:

Find the characteristic for rectangular prism by using the equation,

$$L_c=\frac{V}{A}$$
$$=\frac{L\times B\times H}{2(L\times B)+2(B\times H)+2(H\times L)}$$

Here, Here, V is the volume of rectangular prism, A is the surface area of rectangular prism, L is the length of rectangular prism, B is the breadth of the rectangular prism and H is the height of the rectangular prism.

Substitute, 4 cm = 0.04 m for L, 5 cm = 0.05 m for B, 6 cm = 0.06 m for H.

$$L_c=\frac{0.04\times0.05\times0.06}{2\times(0.04\times0.05)+2\times(0.04\times0.06)+2\times(0.05\times0.06)}$$
$$=0.0081\text{ m}$$

Step 10 of 12

Find the Biot number for rectangular prism by using the equation,

$$\text{Bi}=\frac{hL_c}{k}$$

Substitute, 12 $W/m^2\cdot°C$ for h, 0.0081 m for L_c and 429 $W/m\cdot°C$ for k.

$$\text{Bi}=\frac{12\times0.0081}{429}$$
$$=0.0002266$$

Here Biot number is less than 0.1, therefore lumped system analysis is applicable.

Step 11 of 12

Find the time period required to reach the temperature of 25 °C by using the equation,

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt}$$

Here, t is the time in second taken for heating, T_∞ is the temperature of the surroundings, T_i is the initial temperature, $T(t)$ is the temperature after t seconds and b is constant.

Find the constant b by using the equation,

$$b=\frac{h}{\rho c_p L_c}$$

Substitute, 12 $W/m^2\cdot°C$ for h, 10500 kg/m^3 for ρ, 235 J/kg for c_p and 0.0081 m for L_c

$$b=\frac{12}{10500\times235\times0.0081}$$
$$=0.0006\,s^{-1}$$

Step 12 of 12

Find the time period required to reach the temperature of 25 °C by using the equation,

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt}$$

Substitute, 25 °C for $T(t)$, 33°C for T_∞, 0°C for T_i and 0.0006 s^{-1} for b.

$$\frac{25-33}{0-33}=e^{-(0.0006)\times(t)}$$
$$t=39.36$$

Therefore, in order to reach the temperature of 25°C, the rectangular prism will take a time of $\boxed{39.36\,\text{min.}}$

Question: 18.021P

Consider a 800-W iron whose base plate is made of 0.5-cm-thick aluminum alloy 20218-T6 (p = 2770 kg/m3 , *cp* = 875 J/kg·K, α = 7.3 × 10-5 m2 /s). The base plate has a surface area of 0,03 m2 . Initially, the iron is in thermal equilibrium with the ambient air at 22°C. Taking the heat transfer coefficient at the surface of the base plate to be 12 W/m2 ·K and assuming 85 percent of the heat generated in the resistance wires is transferred to the plate, determine how long it will take for the plate temperature to reach 140°C. Is it realistic to assume the plate temperature to be uniform at all times?

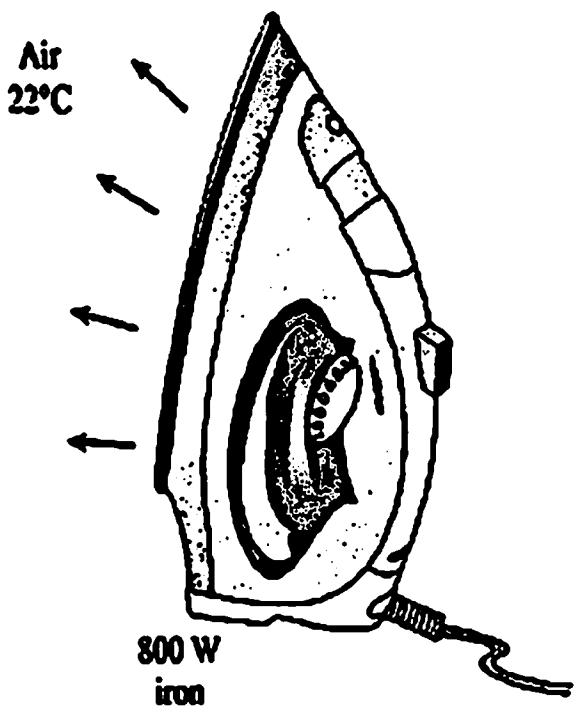


FIGURE P18-21

Step 1 of 5

Given data:

Capacity of the iron, $P = 800\,\text{W}$

Thickness of the aluminum, $t = 0.5\,\text{cm} = 0.005\,\text{m}$

Temperature of ambient air, $T = 22°\text{C}$

Heat transfer coefficient of the base plate, $h = 12\,\text{W/m}^2\cdot\text{K}$

Percentage of heat transferred to the plate, $\eta_t = 85\% = 0.85$

Center temperature of the plate is $T(t) = 140°\text{C}$

Area of the plate, $A = 0.03\,\text{m}^2$

Step 2 of 5

Properties of the iron:

Density, $\rho = 2770\,\text{kg/m}^3$

Specific heat, $c_p = 875\,\text{J/kg}\cdot\text{K}$

Thermal diffusivity, $\alpha = 7.3\times10^{-5}\,\text{m}^2/\text{s}$

Amount of heat going to iron,

$$P_{iron} = \eta \times P$$
$$= 0.85\times800$$
$$= 680\ \text{W}$$

Average temperature of the plate,

$$T_{avg} = \frac{140+22}{2}$$
$$= 81°\text{C}$$

Step 3 of 5

The rate of heat loss from plate,

$$Q_{loss} = hA\left(T_{avg} - T\right)$$
$$= 12\times0.03\times(81-22)$$
$$= 21.24\ \text{W}$$

Mass of the plate is

$$m = \rho A t$$
$$= 2770\times0.005\times0.03$$
$$= 0.4155\ \text{Kg}$$

Therefore, amount of heat available is

$$Q = P - Q_{loss}$$
$$= 680-21.24$$
$$= 658.76\,\text{W}$$

Step 4 of 5

Now if this temperature difference occurs over the span of time is equal to't', then an energy balance gives:

$$658.76t = mc_p(140-22)$$

Putting this data:

$$658.76t = mc_p(140-22)$$

$$658.76t = 0.4155\times875\times(140-22)$$

$$t = 65.12\ \text{sec}$$

Therefore, the required time is $\boxed{t = 65.12\ \text{sec}}$

Step 5 of 5

For this problem Biot Number becomes

$$Bi = \frac{hL_c}{K}$$
$$= \frac{12\times L_c}{177}$$

Now, characteristic length is:

$$L_c = t = 0.005\ \text{m}$$

Putting this into Biot number expression, we get:

$$Bi = \frac{hL_c}{K}$$
$$= \frac{12\times L_c}{117}$$
$$= \frac{12\times0.005}{117}$$
$$= 3.4\times10^{-4}$$

In this problem lumped analysis and as well as uniform base plate temperature can be used as Biot no is much smaller than 0.1.

Question: 18.022P

Reconsider Prob. 18-21. Using EES (or other) software, investigate the effects of the heat transfer coefficient and the final plate temperature on the time it will take for the plate to reach this temperature. Let the heat transfer coefficient vary from 5 W/m2 ·K to 25 W/m·K and the temperature from 30°C to 200°C. Plot the time as functions of the heat transfer coefficient and the temperature, and discuss the results.

Step 1 of 7

Given data:

Capacity of the iron, $Q = 800\,\text{W}$

Thickness of the base plate, $t = 0.5\,\text{cm}$

Cross sectional area of the base plate, $A = 0.03\,\text{m}^2$

Heat transfer coefficient between base plate and air, $h = 12\ \text{W/m}^2\cdot\text{K}$

Temperature of the ambient air, $T_a = 22°\text{C}$

Final temperature of the air, $T_f = 140°\text{C}$

Efficiency of the iron, $\eta = 0.85$

Step 2 of 7

Properties of the aluminum alloy 2024-T6:

$\rho = 2770\ \text{kg/m}^3$.

$c_p = 875\,\text{J/kg}\cdot\text{K}$.

$\alpha = 7.3\times10^{-5}\ \text{m}^2/\text{s}$.

Heat transfer coefficient between base plate and air is $h\ \text{W/m}^2\cdot\text{K}$.

Actual heat transferred at the base plate is

$$Q = 0.85\times800$$
$$= 680\ \text{W}$$

Mean film temperature is $T_{avg} = \dfrac{T_f+22}{2}$.

Step 3 of 7

The rate of heat loss from plate is $h\times0.03\times\left(\dfrac{T_f+22}{2}-22\right)$

So difference in heat transfer is $680-h\times0.03\times\left(\dfrac{T_f+22}{2}-22\right)$

Now if this temperature difference occurs over the span of time is equal to't', then an energy balance gives:

$$\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = mC_p\left(T_f-22\right)$$

$$\boxed{\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = 0.4155\times875\times\left(T_f-22\right)}$$

Step 4 of 7

Let us first consider $h = 12\ \text{W/m}^2\cdot\text{K}$.

So, it implies:

$$\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = 0.4155\times875\times\left(T_f-22\right)$$

$$\Rightarrow\left\{680-12\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = 0.4155\times875\times\left(T_f-22\right)$$

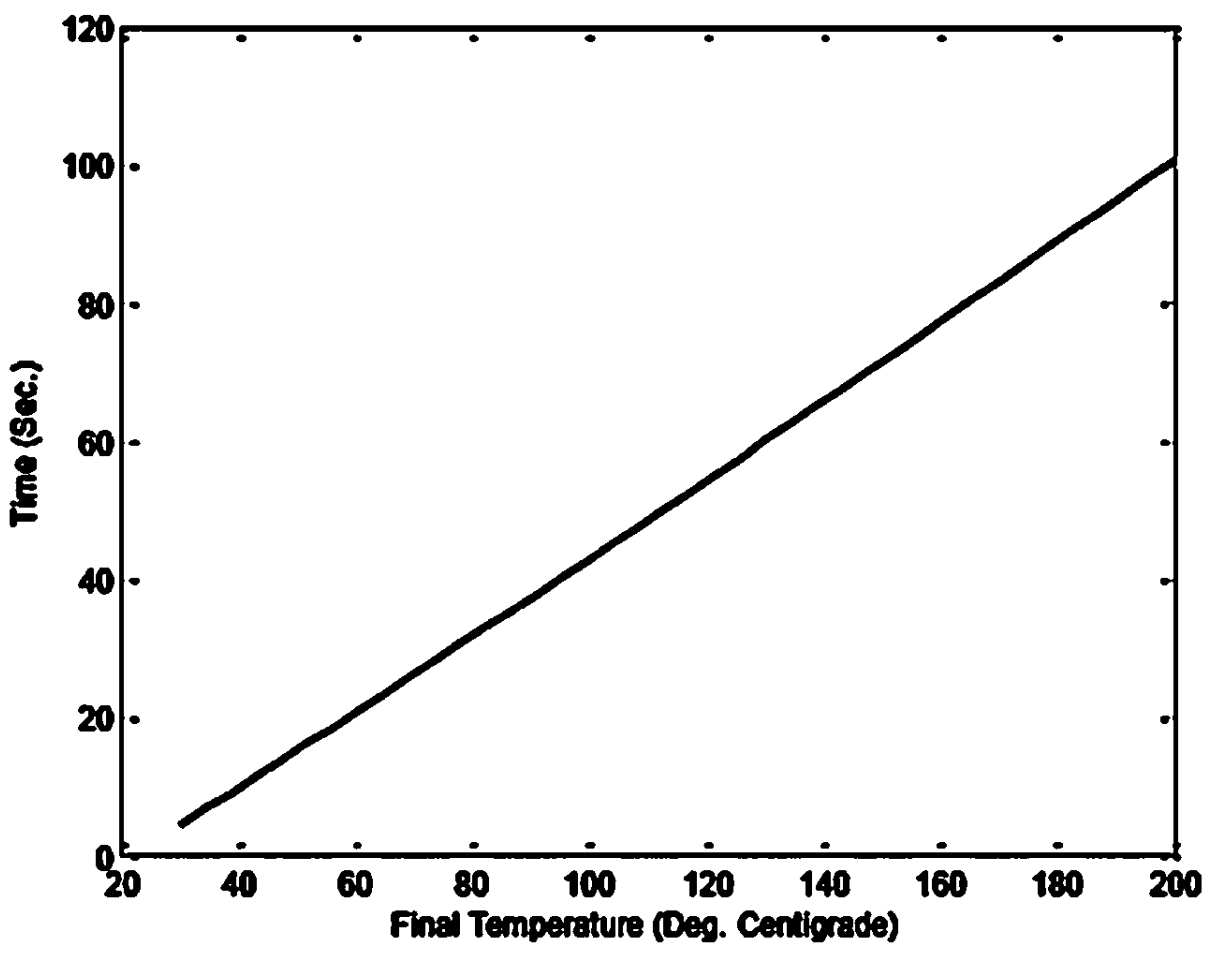
$$\Rightarrow\boxed{t = \frac{0.4155\times875\times\left(T_f-22\right)}{\left\{680-12\times0.03\times\left(\dfrac{T_f+22}{2}-22\right)\right\}}}$$

Step 5 of 7

This expression is used to plot variation of time as function of final temperature.

MATLAB Program:

```
clear all
clc
x=linspace(30,200,40);
y=0.4185*875*(x-22);
z=680-((((x+22)/2)-22)*12*0.03);
t=y./z;
plot(x,t)
```



Step 6 of 7

Again in the expression:

$$\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = mC_p\left(T_f-22\right)$$

$$\boxed{\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = 0.4155\times875\times\left(T_f-22\right)}$$

Let us consider: $T_f = 140^{\circ}\text{C}$.

So, it implies:

$$\left\{680-h\times0.03\times\left(\frac{T_f+22}{2}-22\right)\right\}t = 0.4155\times875\times\left(T_f-22\right)$$

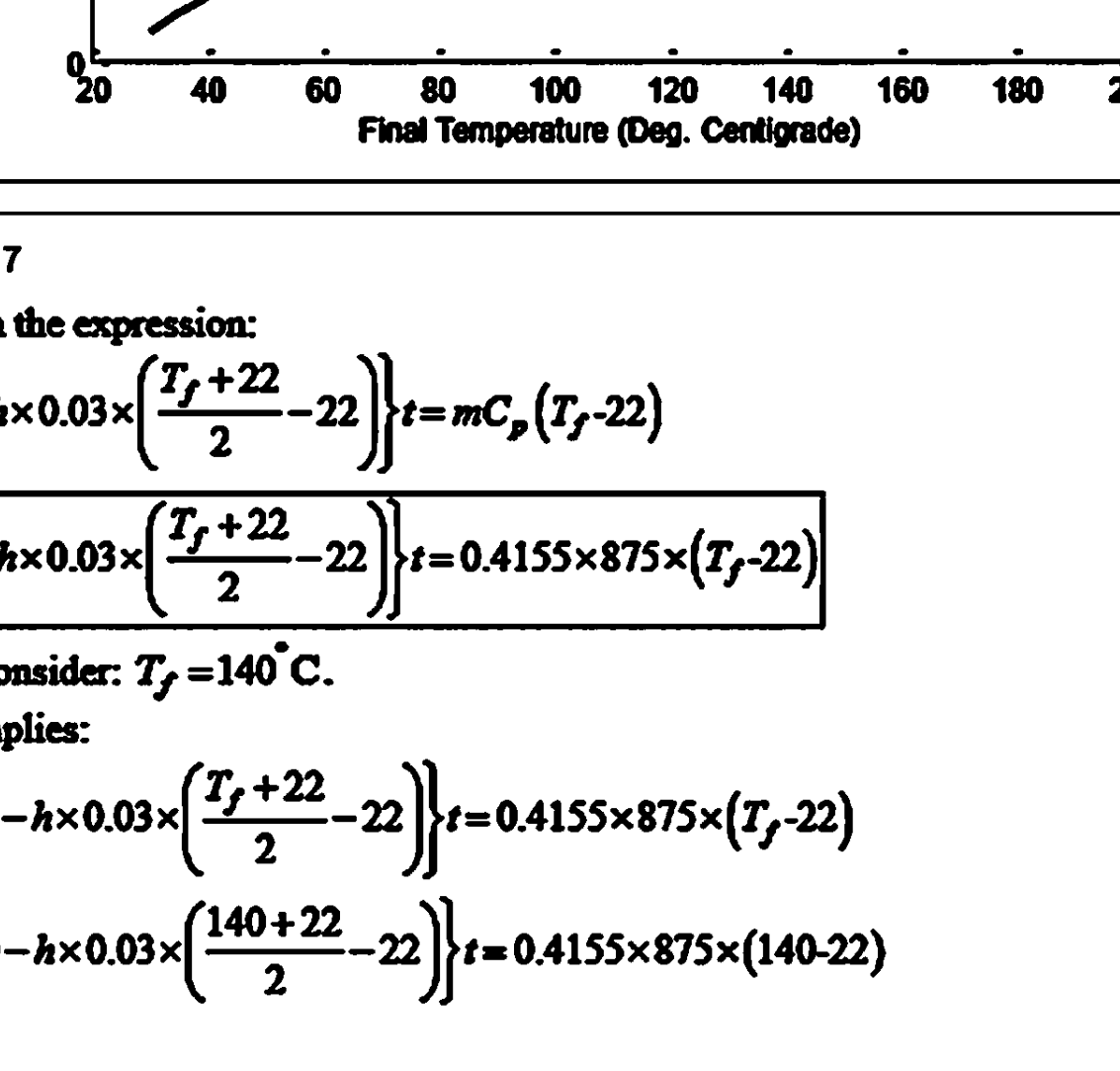
$$\Rightarrow\left\{680-h\times0.03\times\left(\frac{140+22}{2}-22\right)\right\}t = 0.4155\times875\times(140-22)$$

Step 7 of 7

This expression is used to plot variation of time as function of heat transfer coefficient.

MATLAB Program:

```
clear all
clc
x=linspace(5,25,40);
y=0.4185*875*(140-22);
z=680-((((140+22)/2)-22)*x*0.03);
t=y./z;
plot(x,t)
```



Question: 18.023P

A people is found dead at 5pm in a room whose temperature is 20°C. The temperature of the body is measured to be 25°C when found, and the heat transfer coefficient is estimated to be 8 W/m2 ·K. Modeling the body as a 30-cm-diameter, 1.70-m-long cylinder and using the lumped system analysis as a rough approximation, estimate the time of death of that person.

Step 1 of 5

Calculate the radius of the cylinder.

$$r = \frac{D}{2}$$

Here, D is the diameter of the cylinder.

Substitute 30 cm for D.

$$\begin{aligned} r &= \frac{30}{2} \\ &= 15\ \text{cm} \\ &= 15\times10^{-2}\ \text{m} \end{aligned}$$

Step 2 of 5

Calculate the characteristics length of the person as follows:

$$\begin{aligned} L_c &= \frac{V}{A} \\ &= \frac{\left(\frac{\pi}{4}D^2L\right)}{\left(2\pi rL+2\pi r^2\right)} \end{aligned}$$

Here, L is the length of the cylinder (Body assumed as cylinder), r is the radius of the cylinder, D is the diameter of the cylinder.

Substitute $15\times10^{-2}\ \text{m}$ for r, 30 cm for D, 1.70 m for L.

$$\begin{aligned} L_c &= \frac{\left(\frac{\pi\times\left(30\times10^{-2}\right)^2}{4}\right)(1.7)}{\left(2\pi\times15\times10^{-2}\ \text{m}\times1.7\right)+\left(2\pi\times\left(15\times10^{-2}\ \text{m}\right)^2\right)} \\ &= 0.0689\ \text{m} \end{aligned}$$

Step 3 of 5

Consider the average temperature of human body is to be $(37°\text{C})$.

The average human body is 72 percent water by mass, and thus assume the body to have the properties of water at the average temperature of $\left(\frac{37+25}{2}\right)=31°\text{C}$.

Obtain the properties of water from table A-9 "Saturated properties of water" at $T_{av}=31°\text{C}$

$$k = 0.617\ \text{W/m}$$
$$\rho = 996\ \text{kg/m}^3$$
$$c_p = 4718\ \text{J/kg}\cdot\text{K}$$

Calculate the Biot number.

$$\text{Bi} = \frac{hL_c}{k}$$

Here, h is the heat transfer coefficient, k is the thermal conductivity.

Substitute $0.617\ \text{W/m}$ for k, $0.0689\ \text{m}$ for L_c and $8\ \text{W/m}^2$ for h.

$$\begin{aligned} \text{Bi} &= \frac{hL_c}{k} \\ &= \frac{8\times0.0689}{0.617} \\ &= 0.89>1 \end{aligned}$$

Thus, lumped system analysis is not applicable.

Step 4 of 5

Calculate the time constant as follows:

$$b = \frac{h}{\rho c_p L_c}$$

Here, h is the heat transfer coefficient, ρ is the density, and c_p is the specific heat capacity at constant pressure.

Substitute $8\ \text{W/m}^2$ for h, $996\ \text{kg/m}^3$ for ρ, $4718\ \text{J/kg}\cdot\text{K}$ for c_p, $0.0689\ \text{m}$ for L_c.

$$\begin{aligned} b &= \frac{8}{996\times4178\times0.0689} \\ &= 2.79\times10^{-5}\ \text{s}^{-1} \end{aligned}$$

Calculate the time required to reach the temperature of $25°\text{C}$ as follows:

$$\frac{T(t)-T_a}{T_i-T_a} = e^{-bt}$$

Here, time taken for cooling is t, b is Biot constant, T_a is the ambient temperature, T_i is the initial temperature of the body of that person, $T(t)$ is the final temperature of the body after death.

Step 5 of 5

Substituting $25°\text{C}$ for $T(t)$, $20°\text{C}$ for T_a, $\left(2.79\times10^{-5}\ \text{s}^{-1}\right)$ for b, $37°\text{C}$ for T_i.

$$\begin{aligned} \frac{T(t)-T_a}{T_i-T_a} &= e^{-bt} \\ \frac{25-20}{37-20} &= e^{-\left(2.79\times10^{-5}\right)(t)} \\ 0.294 &= e^{-\left(2.79\times10^{-5}t\right)} \\ \ln 0.294 &= -\left(2.79\times10^{-5}t\right) \\ -1.224 &= -\left(2.79\times10^{-5}t\right) \\ t &= \frac{2.79\times10^{-5}}{1.224} \\ &= 43870 \\ &= 12.2\ \text{hours} \end{aligned}$$

Therefore, the time of death of the person is $\boxed{12.2\,\text{hour}}$.

Question: 18.024P

Springs in a suspension system of automobiles are made of steel rods heated and wound into coils while ductile. Consider steel rods (ρ = 7832 kg/m3 , cp = 434 J/kg·K. and k = 63.9 W/m·K) with diameter of 2.5 cm and length of 1.27 m. The steel rods are heated in an oven with a uniform convection heat transfer coefficient of 20 W/m2 ·K. The steel rods were heated from an initial temperature of 20°C to the desired temperature of 450°C before being wound into coils. Determine the ambient temperature in the oven, if the steel rods were to be heated to the desired temperature within 10 minutes.

Step 1 of 3

Characteristic length of the steel rod is given by,

$$L_C = \frac{V}{A}$$

Here,
Volume of the steel rod is V
Area of the steel rod is A

$$L_C = \frac{V}{A}$$
$$= \frac{\left(\frac{\pi}{4}D^2L\right)}{(\pi DL)}$$
$$= \frac{D}{4}$$

Now substitute 0.025 m for D,

$$L_C = \frac{0.025}{4}$$
$$= 6.25\times10^{-3}\ \text{m}$$

Step 2 of 3

Then, Biot number is

$$Bi = \frac{hL_c}{k}$$

Here,
Heat transfer coefficient is h
Thermal conductivity of the steel rod is k

$$Bi = \frac{hL_c}{k}$$
$$= \frac{20\times6.25\times10^{-3}}{63.9}$$
$$= 0.001956$$

Since Biot number is less than the 0.1, therefore lumped system analysis is applicable.

Step 3 of 3

From the equation of lumped system analysis

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{\frac{-hA\tau}{\rho c_p V}}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{\frac{-h\tau}{\rho c_p L_C}}$$

Substitute 450°C for $T(t)$, 20°C for T_i, 20 W/m^2·K for h, 600 s for τ and 7832 kg/m^3 for ρ

$$\frac{450-T_\infty}{20-T_\infty} = e^{-\left(\frac{20\times600}{7832\times434\times6.25\times10^{-3}}\right)}$$
$$\frac{450-T_\infty}{20-T_\infty} = 0.568$$
$$\boxed{T_\infty = 1016°C}$$

Therefore, temperature of the fluid is 1016°C.

Question: 18.025P

Steel rods (ρ = 7832 kg/m3 , cp = 434 J/kg·K. and k = 63.9 W/m2 ·K) are heated in a furnace to 850°C and then quenched in a water bath at 50°C for a period of 40 seconds as part of a hardening process. The convection heat transfer coefficient is 650 W/m2 ·K. If the steel rods have diameter of 40 mm and length of 2 m, determine their average temperature when they are taken out of the water bath.

Step 1 of 3

Calculate the characteristic length of the steel rod using the relation below

$$\begin{aligned} L_c &= \frac{V}{A} \\ &= \frac{\left(\frac{\pi}{4}D^2L\right)}{(\pi DL)} \\ &= \frac{D}{4} \end{aligned}$$

Here, volume of the steel rod is V , Area of the steel rod is A and diameter of the steel rod is D

Substitute $0.04\,\text{m}$ for D in the above equation

$$\begin{aligned} L_c &= \frac{0.04}{4} \\ &= 0.01\,\text{m} \end{aligned}$$

Step 2 of 3

The relation for the Biot number is as follows

$$Bi = \frac{hL_c}{k}$$

Here, heat transfer coefficient is h and thermal conductivity of the steel rod is k

Substitute $650\,\text{W/m}^2\cdot\text{K}$ for h, $63.9\,\text{W/m}\cdot\text{K}$ for k and 0.01

$$\begin{aligned} Bi &= \frac{hL_c}{k} \\ &= \frac{650\times 0.01}{63.9} \\ &= 0.1 \end{aligned}$$

Since the Biot number, $Bi \le 0.1$, the lumped system analysis is applicable.

Substitute $0.04\,\text{m}$ for D in the above equation

$$\begin{aligned} L_c &= \frac{0.04}{4} \\ &= 0.01\,\text{m} \end{aligned}$$

Step 2 of 3

The relation for the Biot number is as follows

$$Bi = \frac{hL_c}{k}$$

Here, heat transfer coefficient is h and thermal conductivity of the steel rod is k

Substitute $650\,\text{W/m}^2\cdot\text{K}$ for h, $63.9\,\text{W/m}\cdot\text{K}$ for k and 0.01

$$\begin{aligned} Bi &= \frac{hL_c}{k} \\ &= \frac{650\times 0.01}{63.9} \\ &= 0.1 \end{aligned}$$

Since the Biot number, $Bi \le 0.1$, the lumped system analysis is applicable.

Question: 18.026P

A brick of 203 × 102 × 57 mm in dimension is being burned in a kiln to 1100°C, and then allowed to cool in a room with ambient air temperature of 30°C and convection heat transfer coefficient of 5 W/m2 ·K. If the brick has properties of ρ = 1920 kg/m3 , cp = 790 J/kg·K, and k = 0.90 W/m2 ·K, determine the time required to cool the brick to a temperature difference of 5°C from the ambient air temperature.

Step 1 of 3

Find the volume of the brick.

$$V = 203 \times 102 \times 57$$
$$= 1180242 \text{ mm}^3$$

Find the surface area of the brick.

$$A = 2\left[(203 \times 102) + (203 \times 57) + (102 \times 57)\right]$$
$$= 76182 \text{ mm}^2$$

Find the characteristic length of the steel rod.

$$L_C = \frac{V}{A}$$

Substitute 1180242 mm^3 for V and 76182 mm^2 for A .

$$L_C = \frac{1180242}{76182}$$
$$= 15.49 \text{ mm}$$
$$= 0.01549 \text{ m}$$

Step 2 of 3

Find the Biot number using the equation.

$$Bi = \frac{hL_c}{k}$$

Here, h is the heat transfer coefficient and k is the thermal conductivity.

Substitute $5 \text{ W/m}^2 \cdot \text{K}$ for h , $0.90 \text{ W/m} \cdot \text{K}$ for k and 0.01549 m for L_c .

$$Bi = \frac{5 \times 0.01549}{0.90}$$
$$= 0.0860$$

Since, the Biot number is less than or equal to 0.1, therefore lumped system analysis is applicable.

Step 3 of 3

Use the equation of lumped system analysis and find the time taken to brick reach the temperature.

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{\frac{-hA\tau}{\rho c_p V}}$$
$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{\frac{-h\tau}{\rho c_p L_c}}$$

Here, ρ is the density of the steel rod, c_p is the specific heat at constant pressure, T_∞ is the ambient temperature, $T(t)$ is center temperature of the steel rod and τ is the time.

Substitute 1920 kg/m^3 for ρ , 30°C for T_∞ , 5°C for $(T(t) - T_\infty)$, 1100°C for $T(t)$, $790 \text{ J/kg} \cdot \text{K}$ for c_p and 0.01549 m for L_C .

$$\left(\frac{5}{1100 - 30}\right) = e^{-\left(\frac{5 \times \tau}{1920 \times 790 \times 0.01549}\right)}$$
$$\ln\left(\frac{5}{1100 - 30}\right) = \ln\left(e^{-\left(\frac{5 \times \tau}{1920 \times 790 \times 0.01549}\right)}\right)$$
$$-5.365976 = -\left(\frac{5 \times \tau}{1920 \times 790 \times 0.01549}\right)$$
$$\tau = 25214.97 \text{ sec}$$
$$\tau = 7 \text{ h}$$

Therefore, the time taken to brick reach the temperature of 1100°C is $\boxed{7 \text{ h}}$.

Question:

18.027P

Carbon steel balls (ρ = 7833 kg/m3 , k = 54 W/m·K. cp = 0.465 kJ/kg·°C, and α = 1.474 × 10-6 m/s) 8 mm in diameter are annealed by heating them first to 900°C in a furnace and then allowing them to cool slowly to 100°C in ambient air at 35°C. If the average heat transfer coefficient is 75 W/m2 ·K, determine how long the annealing process will take. If 2500 balls are to be annealed per hour, determine the total rate of heat transfer from the balls to the ambient air.

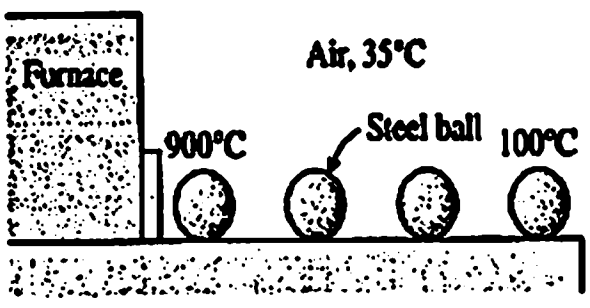


FIGURE P18-27

Step 1 of 5

Given:

Diameter of carbon ball, D_c = 1.2 cm

Question:

Carbon steel balls (ρ = 7833 kg/m3 , k = 54 W/m·K. cp = 0.465 kJ/kg·°C, and α = 1.474 × 10-6 m/s) 8 mm in diameter are annealed by heating them first to 900°C in a furnace and then allowing them to cool slowly to 100°C in ambient air at 35°C. If the average heat transfer coefficient is 75 W/m2 ·K, determine how long the annealing process will take. If 2500 balls are to be annealed per hour, determine the total rate of heat transfer from the balls to the ambient air.

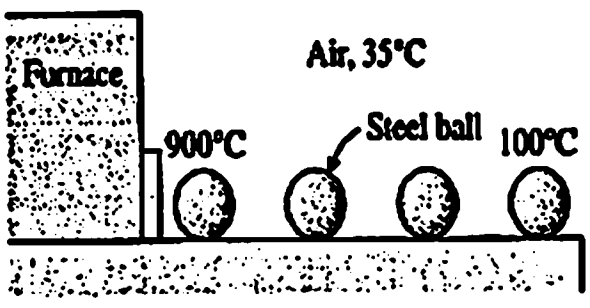


FIGURE P18-27

Step 1 of 5

Given:

Diameter of carbon ball, D_c = 1.2 cm

$$\frac{A_s}{}$$

$$L_c = \frac{\pi D^3/6}{\pi D^2}$$

$$L_c = \frac{D}{6}$$

$$L_c = \frac{0.008\ \text{m}}{6}$$

$$L_c = 0.0013\ \text{m}$$

The Biot number $Bi = \frac{hL_c}{k}$

$$Bi = \frac{(75\ \text{W/m}^2.^\circ\text{C})(0.0013\ \text{m})}{(54\ \text{W/m.}^\circ\text{C})}$$

$$Bi = 0.0018 < 0.1$$

Step 4 of 5

Therefore, the lumped system analysis is applicable. Then the time for the annealing process is determined to be

The value of exponent $b = \frac{hA_s}{\rho C_p V}$

$$b = \frac{h}{\rho C_p L_c}$$

$$b = \frac{75\ \text{W/m}^2.^\circ\text{C}}{(7833\ \text{kg/m}^3)(465\ \text{J/kg.}^\circ\text{C})(0.0013\ \text{m})}$$

$$b = 0.01584\ s^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$\frac{100-35}{900-35} = e^{-(0.01584\ s^{-1})t}$$

t=163 s

t=2.7 min

Step 5 of 5

The amount of heat transfer from a single ball is

The mass $m = \rho V$

$$m = \rho\frac{\pi D^3}{6}$$

$$m = (7833\ \text{kg/m}^3)\frac{\pi(0.008\ \text{m})^3}{6}$$

$$m = 0.0021\ \text{kg}$$

The heat transfer from a ball

$Q = mC_p[T_f - T_i]$

$Q = (0.0021\ \text{kg})(465\ \text{J/kg.}^\circ\text{C})(900-100)^\circ\text{C}$

$Q = 781\ \text{J}$

$Q = 0.781\ \text{kJ}$ (per ball)

Then the total rate of heat transfer from the balls to the ambient air becomes

$\dot{Q} = \dot{n}_{ball} Q$

$\dot{Q} = (2500\ \text{balls/h}) \times (0.781\ \text{kJ/ball})$

$\dot{Q} = 1{,}953\ \text{kJ/h}$

$\dot{Q}$=543 W

Question: 18.028P

Reconsider Prob. 18-27. Using EES (or other) software, investigate the effect of the initial temperature of the balls on the annealing time and the total rate of heat transfer. Let the temperature vary from 500°C to 1000°C. Plot the time and the total rate of heat transfer as a function of the initial temperature, and discuss the results.

Step 1 of 3

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box
"GIVEN"
D=0.008 "[m]"
"T_i=900 [C], parameter to be varied"
T_f=100 "[C]"
T_infinity=35 "[C]"
h=75 "[W/m^2-C]"
n_dot_ball=2500 "[1/h]"
"PROPERTIES"

Question:

Reconsider Prob. 18-27. Using EES (or other) software, investigate the effect of the initial temperature of the balls on the annealing time and the total rate of heat transfer. Let the temperature vary from 500°C to 1000°C. Plot the time and the total rate of heat transfer as a function of the initial temperature, and discuss the results.

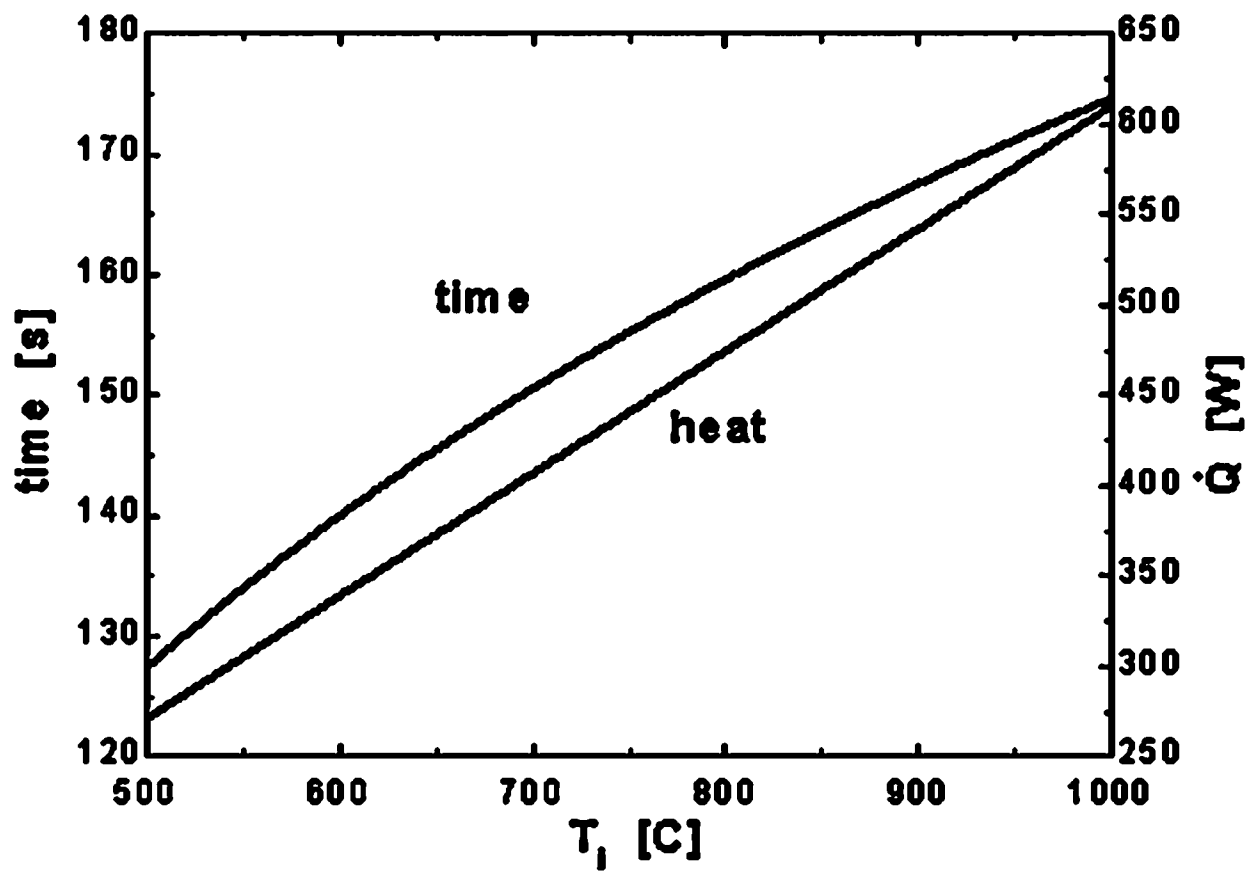
Step 1 of 3

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box
"GIVEN"
D=0.008 "[m]"
"T_i=900 [C], parameter to be varied"
T_f=100 "[C]"
T_infinity=35 "[C]"
h=75 "[W/m^2-C]"
n_dot_ball=2500 "[1/h]"
"PROPERTIES"

500	127.4	271.2
550	134	305.1
600	140	339
650	145.5	372.9
700	150.6	406.9
750	155.3	440.8
800	159.6	474.7
850	163.7	508.6
900	167.6	542.5
950	171.2	576.4
1000	174.7	610.3

Step 3 of 3

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_i and Y-axis time
Repeat procedure for T_i and Q
Press OK
Graph will be plotted.



Question: 18.029P

An electronic device dissipating 18 W has a mass of 20 g, a specific heat of 850 J/kg·K, and a surface area of 4 cm2 . The device is lightly used, and it is on for 5 min and then off for several hours, during which it cools to the ambient temperature of 25°C. Taking the heat transfer coefficient to be 12 W/m2 -K, determine the temperature of the device at the end of the 5-min operating period. What would your answer be if the device were attached to an aluminum heat sink having a mass of 200 g and a surface area of 80 cm2 ? Assume the device and the heat sink to be nearly isothermal.

Step 1 of 3

Calculate the rate of heat loss from the electronic device.

$$Q = hA_s\left(\frac{T+T_\infty}{2}\right)\Delta t$$

Here, heat transfer coefficient is h, surface area of the device is A_s and time is t.

Substitute $12\ \mathrm{W/m^2\cdot K}$ for h, $4\times10^{-4}\ \mathrm{m^2}$ for A_s, $25°\mathrm{C}$ for T_∞ and $5\times60\ \mathrm{sec}$ for Δt.

$$Q = 12\times4\times10^{-4}\times\left(\frac{T+25}{2}\right)\times300$$
$$= 0.72(T+25)\ \mathrm{J}$$

Step 2 of 3

Write the energy balance on the device.

$$\Delta E_{device} = E_{in} - E_{out} + E_{generation}$$
$$mC_p\Delta T_{device} = -\dot{Q}_{out}\Delta t + \dot{E}_{generation}\Delta t \quad \text{..... (1)}$$
$$mC_p(T-T_\infty) = \dot{E}_{generation}\Delta t - hA_s\left(\frac{T+T_\infty}{2}\right)\Delta t$$

Here, mass of the electronic device is m and specific heat of the electronic device is c_p.

Substitute $0.72(T+25)$ for Q, $0.02\ \mathrm{kg}$ for m, $850\ \mathrm{J/kg\cdot K}$ for c_p, $300\ \mathrm{sec}$ for Δt and $18\ \mathrm{W}$ for E_g in equation (1).

$$18\times300 - 0.72(T+25) = 0.02\times850\times(T-25)$$
$$5400 - 0.72T - 18 = 17T - 425$$
$$17.72T = 5807$$
$$T = 327.7°\mathrm{C}$$

Therefore, the temperature of the electronic device after 5 min time is $\boxed{327.7°\mathrm{C}}$.

Step 3 of 3

Calculate the temperature of the electronic device when it is attached to an aluminum heat sink having mass $0.2\ \mathrm{kg}$.

$$\dot{E}_{generation}\Delta t - hA_s\left(\frac{T+T_\infty}{2}\right)\Delta t = (m+m_a)c_p(T-T_\infty) \quad \text{..... (2)}$$

Here, mass of the electronic device is m, mass of aluminum heat sink is m_a and specific heat of the electronic device is c_p.

Substitute $0.72(T+25)$ for Q, $0.02\ \mathrm{kg}$ for m, $0.2\ \mathrm{kg}$ for m_a, $850\ \mathrm{J/kg\cdot K}$ for c_p, $300\ \mathrm{sec}$ for Δt and $18\ \mathrm{W}$ for E_g in equation (2).

$$18\times300 - 0.72(T+25) = (0.02+0.2)\times850\times(T-25)$$
$$5400 - 0.72T - 18 = 187T - 4675$$
$$187.72T = 10057$$
$$T = 53.57°\mathrm{C}$$

Therefore, the temperature of electronic device with an aluminum heat sink attachment is $\boxed{53.57°\mathrm{C}}$.

Question: 18.030P

An egg is to be cooked to a certain level of doneness by being dropped into boiling water. Can the cooking time be shortened by turning up the heat and bringing water to a more rapid boiling?

Step 1 of 2

Assume the temperature of the egg at room temperature $(T_R = 30°C)$ is cooked in boiling water which is at $T_f = 80°C$, the rate of heat transfer from the boiling water to the egg is more when compared to the egg boiling in the water at room temperature. Now calculate the heat transfer rate of the egg boiling hot water at temperature

Consider the mass of the egg as m and the specific heat of water as c_p. The quantity of heat transfer rate for the egg boiling in the water at 100°C is given by

$$\begin{aligned} Q_{boil} &= mc_p(\Delta T) \\ &= mc_p(T_f - T_R) \\ &= mc_p(80-30) \\ &= 50mc_p \end{aligned}$$

Step 2 of 2

Now consider the egg is at temperature $(T_{egg} = 24°C)$ and water is at room temperature $T_R = 30°C$ then the rate of heat transfer from water to the egg on boiling is given by

$$\begin{aligned} Q_{room} &= mc_p(\Delta T) \\ &= mc_p(T_R - T_{egg}) \\ &= mc_p(30-24) \\ &= 6mc_p \end{aligned}$$

From the above obtained heat transfer rate on comparing $Q_{boil} > Q_{room}$
The time required for cooking the egg in boiled water is less when compared to the water at room temperature.

The cooking of the egg depends on the following parameters

1. size of the egg
2. temperature of the egg before cooking
3. heat transfer coefficient

Question: 18.031P

What is an infinitely long cylinder? When is it proper to treat an actual cylinder as being infinitely long, and when is it not? For example, is it proper to use this model when finding the temperatures near the bottom or top surfaces of a cylinder? Explain.

Step 1 of 1

A cylinder whose diameter is small relative to its length is called infinitely long cylinder. When the diameter and length of the cylinder are comparable, it is not proper to treat the cylinder as being infinitely long. It is also not proper to use this model when finding the temperatures near the bottom or top surfaces of a cylinder since heat transfer at those locations can be two-dimensional.

Question: 18.032P

Can the transient temperature charts in Fig. 18-15 for a plane wall exposed to convection on both sides be used for a plane wall with one side exposed to convection while the other side is insulated? Explain.

Step 1 of 1

Yes. A plane wall whose one side is insulated is equivalent to a plane wall that is twice as thick and is exposed to convection from both sides. The midplane in the latter case will behave like an insulated surface because of thermal symmetry.

Question: 18.033P

Why are the transient temperature charts prepared using nondimensionalized quantities such as the Biot and Fourier numbers instead of the actual variables such as thermal conductivity and time?

Step 1 of 1

The solution for determination of the one-dimensional transient temperature distribution involves many variables that make the graphical representation of the results impractical. In order to reduce the number of parameters, some variables are grouped into dimensionless quantities.

Question: 18.034P

What is the physical significance of the Fourier number? Will the Fourier number for a specified heat transfer problem double when the time is doubled?

Step 1 of 1

The Fourier number is a measure of heat conducted through a body relative to the heat stored. Thus a large value of Fourier number indicates faster propagation of heat through body. Since Fourier number is proportional to time, doubling the time will also double the Fourier number.

Question: 18.035P

How can we use the transient temperature charts when the surface temperature of the geometry is specified instead of the temperature of the surrounding medium and the convection heat transfer coefficient?

Step 1 of 1

This case can be handled by setting the heat transfer coefficient h to infinity ∞ since the temperature of the surrounding medium in this case becomes equivalent to the surface temperature.

Question: 18.036P

A body at an initial temperature of T_i is brought into a medium at a constant temperature of T_∞ . How can you determine the maximum possible amount of heat transfer between the body and the surrounding medium?

Step 1 of 1

Loading solution. Please wait ...

Question: 18.037P

The Biot number during a heat transfer process between a sphere and its surroundings is determined to be 0.02. Would you use lumped system analysis or the transient temperature charts when determining the midpoint temperature of the sphere? Why?

Step 1 of 1

When the Biot number is less than 0.1, the temperature of the sphere will be nearly uniform at all times. Therefore, it is more convenient to use the lumped system analysis in this case.

Question: 18.038P

A long cylindrical wood log (k = 0.17 W/m2·K and α = 1.28 × 10-7 m2 /s) is 10 cm in diameter and is initially at a uniform temperature of 25°C. It is exposed to hot gases at 600°C in a fireplace with a heat transfer coefficient of 13.6 W/m2 ·K on the surface. If the ignition temperature of the wood is 420°C, determine how long it will be before the log ignites.

Step 1 of 2

Obtain the constants λ_1 and A_1 corresponding to the Biot number, $\mathrm{Bi}=4$ from the table "coefficient used for in dimensional approximation of transient one dimensional heat conduction in cylinder",

$$\lambda_1 = 1.9081$$

$$A_1 = 1.4698$$

Calculate the Fourier number by using the equation,

$$\frac{T(r_o,t)-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Here,

Temperature of hot gases T_∞

Wooden log temperature T_i

Ignition temperature of wood, $T(r_o,t)$

Substitute $420°C$ for $T(r_o,t)$, $600°C$ for T_∞, $25°C$ for T_i . 1.4698 for A_1. 1.9081 for λ_1

$$\frac{T(r_o,t)-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{420-600}{25-600} = (1.4698)e^{-(1.9081)^2 \tau}$$

$$0.313 = 1.4698e^{-(1.9081)^2 \tau}$$

$$0.21295 = e^{-(1.9081)^2 \tau}$$

$$\tau = 0.424$$

Step 2 of 2

Calculate the length of time before the log ignites, by using the dimensionless number, Fourier number,

$$\tau = \frac{\alpha t}{r_o^2}$$

$$t = \frac{\tau r_o^2}{\alpha}$$

Here,

Time taken to reach the log final temperature is t and Thermal diffusivity is α

Substitute 0.424 for τ, $1.28\times10^{-7}\ \mathrm{m^2/s}$ for α, $0.05\,\mathrm{m}$ for r_0

$$t = \frac{\tau r_o^2}{\alpha}$$

$$= \frac{0.424\times0.05^{\,2}}{1.28\times10^{-7}}$$

$$= 8281.25\ \mathrm{s}$$

$$= \frac{8281.25}{3600}$$

$$= 2.3003\,\mathrm{hr}$$

Therefore, the length of time before the log ignites is $\boxed{2.300\,\mathrm{hr}}$

Question: 18.039P

An experiment is to be conducted to determine heat transfer coefficient on the surfaces of tomatoes that are placed in cold water at 7°C. The tomatoes (k = 0.59 W/m-K, α= 0.141 × 10-6 m2 /s, ρ = 999 kg/m3 , cp = 3.99 kJ/kg-K) with an initial uniform temperature of 30°C are spherical in shape with a diameter of 8 cm. After a period of 2 hours, the temperatures at the center and the surface of the tomatoes are measured to be 10.0°C and 7.1°C, respectively. Using analytical one-term approximation method (not the Heisler charts), determine the heat transfer coefficient and the amount of heat transfer during this period if there are eight such tomatoes in water.

Step 1 of 5

Calculate the radius of the tomato.

$$r_o = \frac{D}{2}$$

Here, D is the diameter of the tomato.

Substitute 0.08 m for D.

$$r_o = \frac{0.08\ \text{m}}{2}$$
$$= 0.04\ \text{m}$$

Calculate the Fourier number by using the relation,

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, τ is the Fourier number, α is the thermal diffusivity of the material, and t is the time.

Substitute $0.141\times10^{-6}\ \text{m}^2/\text{s}$ for α , $2\times3600\ \text{s}$ for t, and $0.04\ \text{m}$ for r_o .

$$\tau = \frac{0.141\times10^{-6}\ \text{m}^2/\text{s}(2\times3600\ \text{s})}{(0.04\ \text{m})^2}$$
$$= 0.6345$$

As the Fourier number is more than 0.2, the one term solution is applicable.

Step 2 of 5

Calculate the value of λ_1 from the ratio of dimensionless temperatures at the surface and the center of the tomatoes.

$$\frac{\theta_{s,sph}}{\theta_{o,sph}} = \frac{\left(\frac{T_s - T_\infty}{T_i - T_\infty}\right)}{\left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)}$$
$$= \frac{T_s - T_\infty}{T_o - T_\infty}$$
$$= \frac{A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1)}{\lambda_1}}{A_1 e^{-\lambda_1^2\tau}}$$
$$= \frac{\sin(\lambda_1)}{\lambda_1}$$

Here, $\theta_{s,sph}$ is the dimensionless temperature at the surface, $\theta_{o,sph}$ is the dimensionless temperature at the center of the surface, T_s is the surface temperature of the tomato, T_∞ is the temperature of the cold water, T_i is the initial temperature of the tomato, T_o is the temperature at the center of the tomato, A_1 and λ_1 are the constants.

Substitute $7.1°\text{C}$ for T_s , $7°\text{C}$ for T_∞ , and $10°\text{C}$ for T_o .

$$\frac{T_s - T_\infty}{T_o - T_\infty} = \frac{\sin(\lambda_1)}{\lambda_1}$$
$$\frac{7.1°\text{C} - 7°\text{C}}{10°\text{C} - 7°\text{C}} = \frac{\sin(\lambda_1)}{\lambda_1}$$
$$\frac{0.1}{3} = \frac{\sin(\lambda_1)}{\lambda_1}$$
$$0.03334 = \frac{\sin\left(\lambda_1 \times \frac{180}{\pi}\right)}{\lambda_1}$$

From the calculation,

$$\lambda_1 = 3.0401.$$

Step 3 of 5

Obtain the Biot number from the table 4-2 corresponding to the value of λ_1 (3.0401) for sphere by interpolation.

$$\text{Bi} = 30 + \frac{40-30}{3.0632-3.0372}(3.0401-3.0372)$$
$$= 30 + 1.11$$
$$= 31.11$$

Calculate the convective heat transfer coefficient by using the relation,

$$\text{Bi} = \frac{hr_o}{k}$$
$$h = \frac{\text{Bi}\times k}{r_o}$$

Here, h is the convective heat transfer coefficient, and k is the thermal conductivity of the tomato.

Substitute 31.11 for Bi , 0.04 m for r_o , and $0.59\ \text{W/m}\cdot\text{K}$ for k.

$$h = \frac{31.1\times0.59\ \text{W/m}\cdot\text{K}}{0.04\ \text{m}}$$
$$= 458.95\ \text{W/m}^2\cdot\text{K}$$

Therefore, the heat transfer coefficient is $\boxed{458.95\ \text{W/m}^2\cdot\text{K}}$.

Step 4 of 5

Calculate the mass of eight tomatoes by using the relation,

$$m = 8\rho V$$
$$= 8\rho\frac{\pi D^3}{6}$$

Here, ρ is the density of the tomato, and D is the diameter of the tomato.

Substitute $999\ \text{kg/m}^3$ for ρ , and $0.08\ \text{m}$ for D.

$$m = (8\times999)\frac{\pi\times0.08^3}{6}$$
$$= 2.1425\ \text{kg}$$

Calculate the maximum heat transfer rate by using the relation,

$$Q_{max} = mc_p(T_i - T_\infty)$$

Substitute $2.1425\ \text{kg}$ for m, $3.99\ \text{kJ/kg}\cdot\text{K}$ for c_p , $30+273\ \text{K}$ for T_i , and $7+273\ \text{K}$ for T_∞ .

$$Q_{max} = 2.1425\times3.99[(30+273)-(7+273)]$$
$$= 196.62\ \text{kJ}$$

Step 5 of 5

Calculate the actual amount of heat transferred by using the relation,

$$\left(\frac{Q}{Q_{max}}\right)_{sph} = 1 - 3\theta_{o,sph}\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{\lambda_1^3}$$
$$= 1 - 3\left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{\lambda_1^3}$$

Here, Q is the actual amount of heat transferred.

Substitute $196.62\ \text{kJ}$ for Q_{max} , $7°\text{C}$ for T_∞ , $10°\text{C}$ for T_o , $30°\text{C}$ for T_i , and 3.0401 for λ_1 .

$$\left(\frac{Q}{196.62\ \text{kJ}}\right)_{sph} = 1 - 3\left(\frac{10°\text{C} - 7°\text{C}}{30°\text{C} - 7°\text{C}}\right)\frac{\sin3.0401 - 3.0401\cos3.0401}{3.0401^3}$$
$$\frac{Q}{196.62\ \text{kJ}} = 0.9564$$
$$Q = 188.04\ \text{kJ}$$

Therefore, the amount of heat transferred during the period of 2 hours is $\boxed{188.04\ \text{kJ}}$.

Question: 18.040P

In an annealing process, a 50-mm-thick stainless steel plate (p = 8238 kg/m3 , cp = 468 J/kg·K, k = 13.4 W/m·K, and α = 3.48 × 10-6 m2 /s) was reheated in a furnace from an initial uniform temperature of 230°C. The ambient temperature inside the furnace is at a uniform temperature of 1000°C and has a convection heat transfer coefficient of 215 W/m2 ·K. If the entire stainless steel plate is to be heated to at least 600°C, determine the time that the plate should be heated in the furnace using (*a*) Table 18-2 (*b*) the Heisler chart (Figure 18-15).

Step 1 of 5

Calculate the Biot number by using the relation,

$$\mathrm{Bi} = \frac{hL}{k}$$

Here, h is the convection heat transfer coefficient, and k is the thermal conductivity of the brass plate.

Substitute $215\ \mathrm{W/m^2 \cdot K}$ for h, $0.025\ \mathrm{m}$ for L . $13.4\ \mathrm{W/m \cdot K}$ for k.

$$\mathrm{Bi} = \frac{(215\ \mathrm{W/m^2 \cdot K})(0.025\ \mathrm{m})}{13.4\ \mathrm{W/m \cdot K}}$$
$$= 0.4$$

Obtain the constant λ_1 and A_1 from table 4-2 at $\mathrm{Bi} = 0.4$ (For plane wall).

$$\lambda_1 = 0.5932$$

$$A_1 = 1.0580$$

Step 2 of 5

Calculate the Fourier number by using the dimensionless temperature relation,

$$\theta_{0,\mathrm{wall}} = \frac{T_o - T_\infty}{T_i - T_\infty}$$
$$= A_1 e^{-\lambda_1^2 \tau}$$

Here, T_o is the center plane temperature of the plate after heating, T_∞ is the ambient temperature of the furnace, T_i is the initial temperature of the plate, A_1 and λ_1 are the constants, and τ is the Fourier number.

Substitute $600°\mathrm{C}$ for T_o . $1000°\mathrm{C}$ for T_∞ . $230°\mathrm{C}$ for T_i . 0.5932 for λ_1 , and 1.0580 for A_1 .

$$\frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$
$$\frac{600°\mathrm{C} - 1000°\mathrm{C}}{230°\mathrm{C} - 1000°\mathrm{C}} = (1.0580) e^{-(0.5932)^2 \tau}$$
$$0.52 = (1.0580) e^{-(0.5932)^2 \tau}$$
$$0.491 = e^{-(0.5932)^2 \tau}$$
$$\tau = 2.021$$

Step 3 of 5

Calculate the time that the plate should be heated in the furnace by using the relation,

$$\tau = \frac{\alpha t}{L^2}$$
$$t = \frac{\tau L^2}{\alpha}$$

Here, t is the time that plate should be heated, α is the thermal diffusivity, and L is the critical length.

Substitute 2.021 for τ , $0.025\ \mathrm{m}$ for L , and $3.48 \times 10^{-6}\ \mathrm{m^2/s}$ for α .

$$t = \frac{2.021 \times (0.025\ \mathrm{m})^2}{3.48 \times 10^{-6}\ \mathrm{m^2/s}}$$
$$= 362.97\ \mathrm{s}$$

Therefore, the time that the plate should be heated in the furnace is $\boxed{362.97\ \mathrm{s}}$.

Step 4 of 5

b)

Calculate the inverse of the Biot number.

$$\frac{1}{\mathrm{Bi}} = \frac{1}{0.4}$$
$$= 2.5$$

Calculate the dimensionless temperature of the plate by using the relation,

$$\theta_{0,\mathrm{wall}} = \frac{T_o - T_\infty}{T_i - T_\infty}$$

Substitute $600°\mathrm{C}$ for T_o , and $1000°\mathrm{C}$ for T_∞ . $230°\mathrm{C}$ for T_i .

$$\theta_{0,\mathrm{wall}} = \frac{600°\mathrm{C} - 1000°\mathrm{C}}{230°\mathrm{C} - 1000°\mathrm{C}}$$
$$= 0.52$$

Step 5 of 5

Obtain the Fourier number from the Heisler chart (Figure 4-16 (*a*)) corresponding to the value of 2.5 for $\frac{1}{\mathrm{Bi}}$ and 0.52 for $\theta_{0,\mathrm{wall}}$.

$$\tau = 2.1$$

Calculate the time that the plate should be heated in the furnace by using the relation,

$$t = \frac{\tau L^2}{\alpha}$$

Substitute 2.1 for τ , $0.025\ \mathrm{m}$ for L , and $3.48 \times 10^{-6}\ \mathrm{m^2/s}$ for α .

$$t = \frac{2.1 \times (0.025\ \mathrm{m})^2}{3.48 \times 10^{-6}\ \mathrm{m^2/s}}$$
$$= 377.15\ \mathrm{s}$$

Therefore, the time that the plate should be heated in the furnace is $\boxed{377.15\ \mathrm{s}}$.

Question: 18.041P

An ordinary egg can be approximated as a 5.5- cm diameter sphere whose properties are roughly k = 0.6 W/m·k and α = 0.14 × 10-6 m2 /s. The egg is initially at a uniform temperature of 4°C and is dropped into boiling water at 97°C. Taking the convection heat transfer coefficient to be h = 1400 W/m2 ·K, determine how long it will take for center of the egg to reach 70°C.

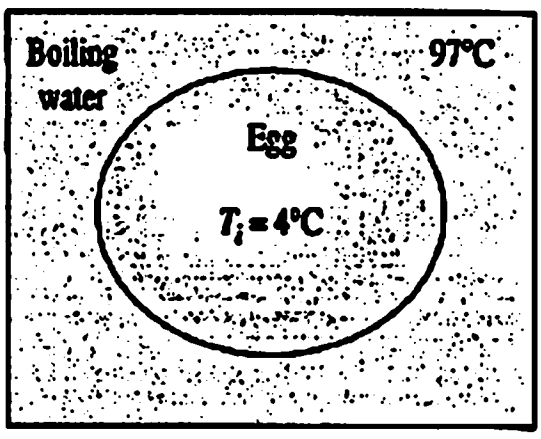


FIGURE P 18-41

Step 1 of 4

Given data:

Diameter of egg $D = 5.5$ cm.

Thermal conductivity of egg $k = 0.6\ W/m^2 \cdot K$.

Thermal diffusivity of egg $\alpha = 0.14 \times 10^{-6}\ m^2/s$.

Initial temperature of egg $T_i = 4°C$.

Temperature of boiling water $T_\infty = 97°C$.

Final temperature of egg $T(t) = 70°C$.

Heat transfer coefficient $h = 1400\ W/m^2 \cdot K$.

Egg is spherical in shape with a radius of $r_0 = 2.75$ cm.

Step 2 of 4

If the heat conduction in the egg is one-dimensional because of symmetry about the midpoint and the thermal properties of the egg are constant

The Biot number for this process is

$$Bi = \frac{hr_o}{k}$$

$$= \frac{(1400\ W/m^2.°C)(0.0275\ m)}{(0.6\ W/m.°C)}$$

$$= 64.2$$

Since Biot number is greater than 0.1, and thus lumped system analysis is not applicable.

The coefficient λ_1 and A_1 for a cylinder corresponding to the Bi number are

From the coefficient used for in dimensional approximation of transient one dimensional heat conduction in cylinder is

For this Biot number, we get $\lambda = 3.0877$, $A = 1.9969$

Step 3 of 4

Then one term approximation is

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = Ae^{-\lambda^2 \tau}$$

$$\frac{70 - 97}{4 - 97} = 1.9969e^{-3.0877^2 \tau}$$

$$\tau = 0.202$$

This is above the value of 0.2. Therefore, the one-term approximate solution be used.

Step 4 of 4

Then the length of time will take for center of the egg to reach 70°C is determined from the Fourier number to be

$$t = \frac{\tau r_o^2}{\alpha}$$

$$= \frac{0.2 \times 0.0275^2}{0.14 \times 10^{-6}}$$

$$= 18\ min$$

Therefore, the time needed for the centre of egg to reach 70° C is $\boxed{18\ min}$.

Question: 18.042P

Reconsider Prob. 18-41. Using EES (or other) software, investigate the effect of the final center temperature of the egg on the time it will take for the center to reach this temperature. Let the temperature vary from 50°C to 95°C. Plot the time versus the temperature, and discuss the results.

Step 1 of 4

Given data:

Diameter of egg, $D = 5.5$ cm

Thermal conductivity of egg, $k = 0.6\,W/m \cdot K$

Thermal diffusivity of egg, $\alpha = 0.14 \times 10^{-6}\ m^2/s$

Initial temperature of egg, $T_i = 8°C$

Temperature of boiling water, $T_\infty = 97°C$

Final temperature of egg, $T_o = 70°C$

Heat transfer coefficient, $h = 1400\ W/m^2 \cdot K$

Egg is spherical in shape with a radius, $r_0 = 2.75$ cm

Step 2 of 4

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
D=0.055 "[m]"
T_i=8 "[C]"
"T_o=70 [C], parameter to be varied"
T_infinity=97 "[C]"
h=1400 "[W/m^2-C]"

"PROPERTIES"
k=0.6 "[W/m-C]"
alpha=0.14E-6 "[m^2/s]"

"ANALYSIS"
Bi=(h*r_o)/k
r_o=D/2
"From biot number Table corresponding to this Bi number, we read"
lambda_1=1.9969
A_1=3.0863
(T_o-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau)
time=(tau*r_o^2)/alpha*Convert(s, min)
```

Step 3 of 4

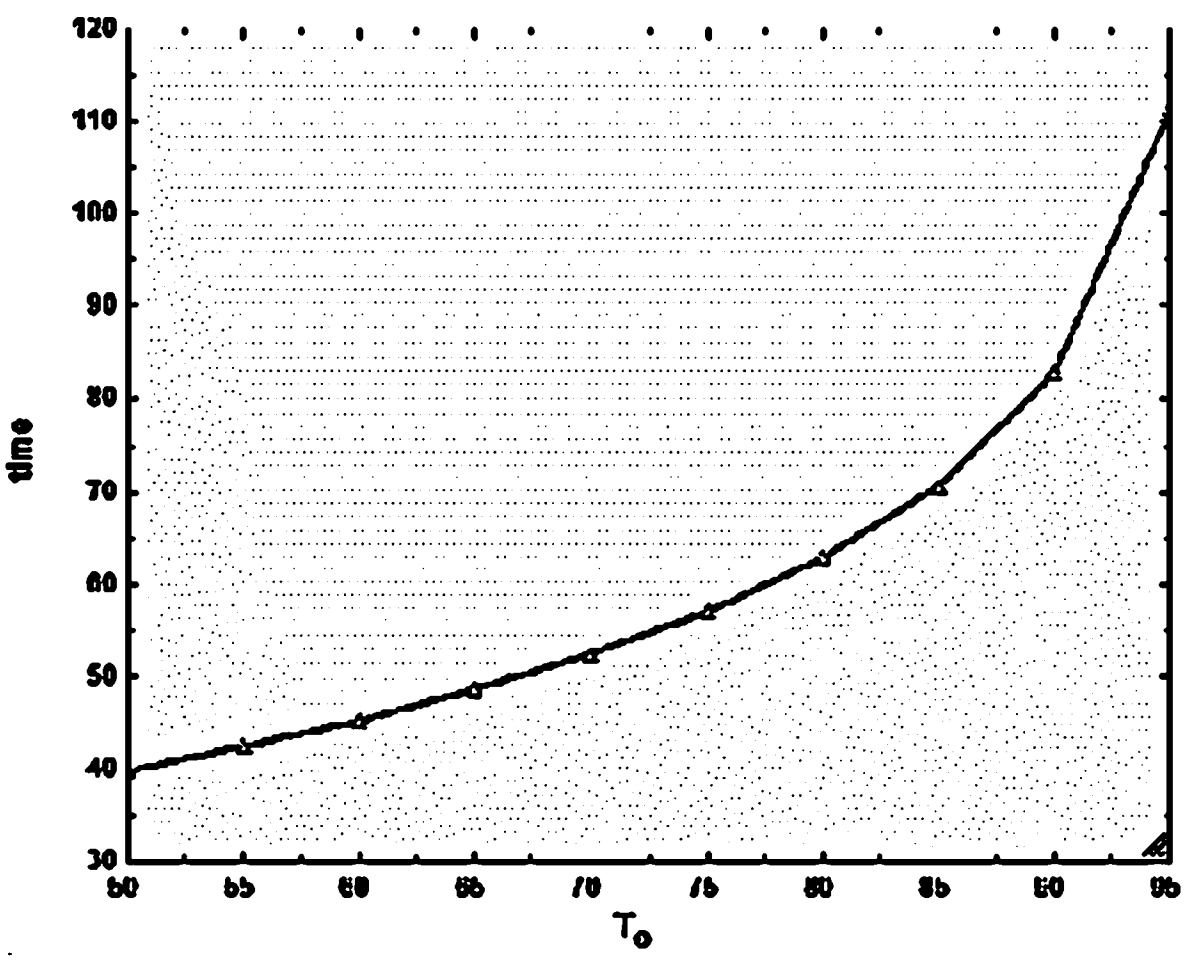
Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 10
Select variables in equations as T_o and time press ADD and then press ok
A table will appear put values of T_o as 50,55,60....95 in run 1to 10
Then from Calculate -> Solve table
Solution will appear

Table 1

1..10	time	T_o
Run 1	39.86	50
Run 2	42.4	55
Run 3	45.26	60
Run 4	48.54	65
Run 5	52.38	70
Run 6	57	75
Run 7	62.82	80
Run 8	70.68	85
Run 9	82.85	90
Run 10	111.1	95

Step 4 of 4

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_o and Y-axis time
Press OK
Graph will be plotted.



Question: 18.043P

In a production facility, 3-cm-thick large brass plates (k - 110 W/m2 ·K, ρ - 8530 kg/m3 , $c\rho$ = 380 J/kg-K. and α = 33.9 × 10-6 m2 /s) that are initially at a uniform temperature of 25°C are heated by passing them through an oven maintained at 700°C. The plates remain in the oven for a period of 10 min. Taking the convection heat transfer coefficient to be h = 80 W/m2 ·K, determine the surface temperature of the plates when they come out of the oven.

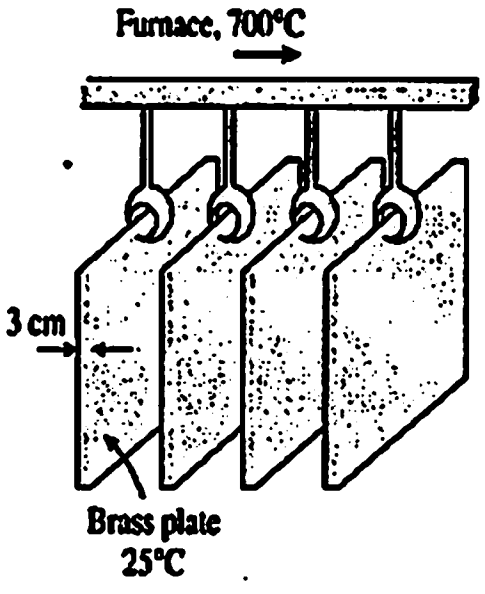


FIGURE P 18-43

Step 1 of 3

Given:

Thickness of brass plate $t = 3$ cm

Thermal conductivity of brass $k = 110$ W/m.°C

Density of brass $\rho = 8530$ kg/m^3

Specific heat of brass $C_p = 380$ J/kg.°C

Thermal diffusivity of brass $\alpha = 33.9\times10^{-6}$ m^2/s

Temperature of oven $T_\infty = 700$°C

The initial temperature $T_i = 25$°C

Plate remain in the oven $t = 10$ min

Step 2 of 3

Heat conduction in the plate is one-dimensional since the plate is large relative to its thickness and there is thermal symmetry about the center plane.

The thermal properties of the plate are constant.

The heat transfer coefficient is constant and uniform over the entire surface.

The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

The Biot number for this process $Bi = \dfrac{hL}{k}$

$$Bi = \frac{(80\ \text{W/m}^2.^\circ\text{C})(0.015\ \text{m})}{(110\ \text{W/m.}^\circ\text{C})}$$

$$Bi = 0.0109$$

The constants λ_1 and A_1 corresponding to this Biot number are, from 11-2 tables

The interpolation method used to find the

$$\lambda_1 = 0.1039 \quad \text{and} \quad A_1 = 1.0018$$

The Fourier number $\tau = \dfrac{\alpha t}{L^2}$

$$\tau = \frac{(33.9\times10^{-6}\ \text{m}^2/\text{s})(10\ \text{min}\times60\ \text{s/min})}{(0.015\ \text{m})^2}$$

$$\tau = 90.4 > 0.2$$

Step 3 of 3

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the temperature at the surface of the plates becomes

$$\theta(L,t)_{wall} = \frac{T(x,t)-T_\infty}{T_i-T_\infty}$$

$$\theta(L,t)_{wall} = A_1 e^{-\lambda_1^2\tau}\cos(\lambda_1 L/L)$$

$$\theta(L,t)_{wall} = (1.0018)e^{-(0.1039)^2(90.4)}\cos(0.1039)$$

$$\theta(L,t)_{wall} = 0.378$$

$$\frac{T(L,t)-700}{25-700} = 0.378$$

$$\boxed{T(L,t) = 445\ ^\circ\text{C}}$$

Question:

Reconsider Prob. 18-43. Using EES (or other) software, investigate the effects of the temperature of the oven and the heating time on the final surface temperature of the plates. Let the oven temperature vary from 500°C to 900°C and the time from 2 min to 30 min. Plot the surface temperature as the functions of the oven temperature and the time, and discuss the results.

Step 1 of 12

Loading solution. Please wait ...

Step 2 of 12

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box
"GIVEN"
L=0.03/2 "[m]"

Question:

Reconsider Prob. 18-43. Using EES (or other) software, investigate the effects of the temperature of the oven and the heating time on the final surface temperature of the plates. Let the oven temperature vary from 500°C to 900°C and the time from 2 min to 30 min. Plot the surface temperature as the functions of the oven temperature and the time, and discuss the results.

Step 1 of 12

Loading solution. Please wait ...

Step 2 of 12

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box
"GIVEN"
L=0.03/2 "[m]"

Step 5 of 12

Solution will appear

T_∞ [C]	T_L [C]
500	321.6
525	337.2
550	352.9
575	368.5
600	384.1
625	399.7
650	415.3
675	430.9
700	446.5
725	462.1
750	477.8
775	493.4
800	509
825	524.6
850	540.2
875	555.8
900	571.4

Step 6 of 12

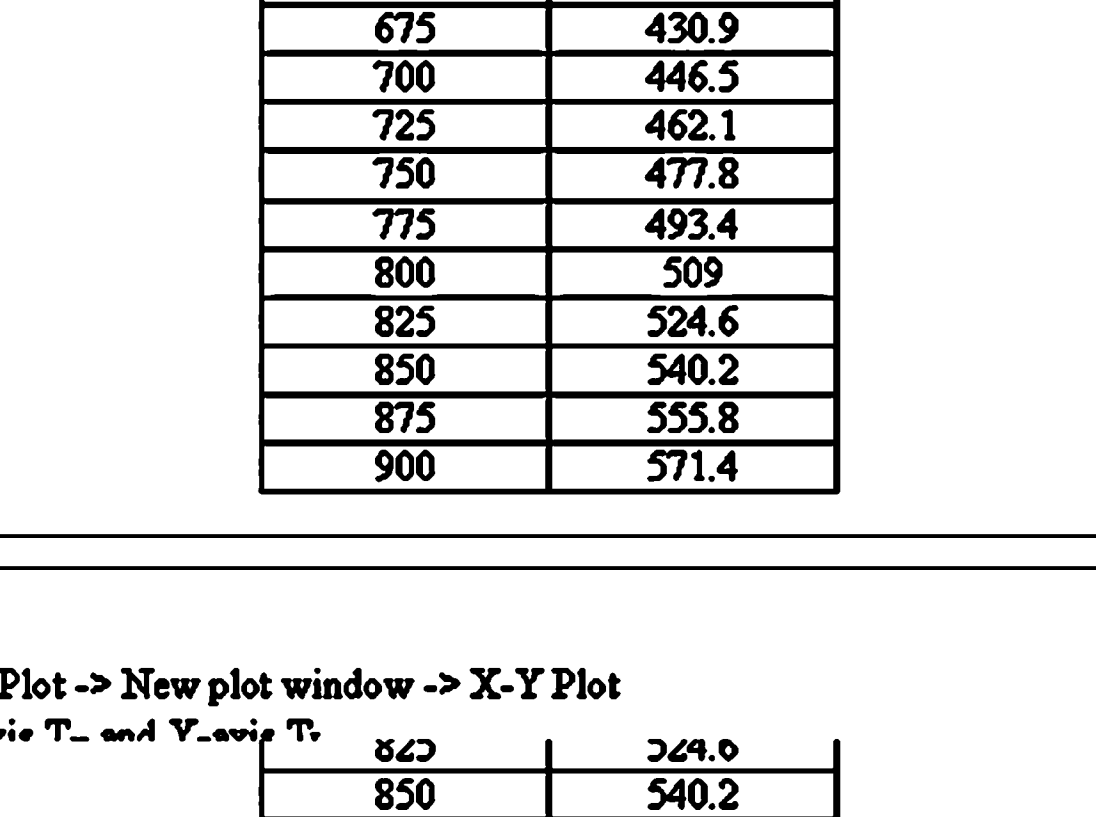
Then from Plot -> New plot window -> X-Y Plot
Select Y_axis T_ and V_axis T.

875	555.8
900	571.4

Step 6 of 12

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_∞ and Y-axis T_L
Press OK
Graph will be plotted.

Step 7 of 12



Step 8 of 12

In the dialog box change code as below

Step 8 of 12

In the dialog box change code as below
"GIVEN"
L=0.03/2 "[m]"
T_i=25 "[C]"
T_infinity=700 "[C], parameter to be varied"
"time=10 [min]" "parameter to be varied"
h=80 "[W/m^2-C]"
"PROPERTIES"
k=110 "[W/m-C]"
alpha=33.9E-6 "[m^2/s]"
"ANALYSIS"
Bi=(h*L)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1=0.1039
A_1=1.0018
tau=(alpha*time*Convert(min, s))/L^2
(T_L-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau)*Cos(lambda_1*L/L)

Step 9 of 12

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 11
Select variables in equations as time and T_L press ADD and then press ok
A table will appear put values of time as 0.1,0.12,0.14....0.3 in run 1to 11
Then from Calculate -> Solve table

Step 10 of 12

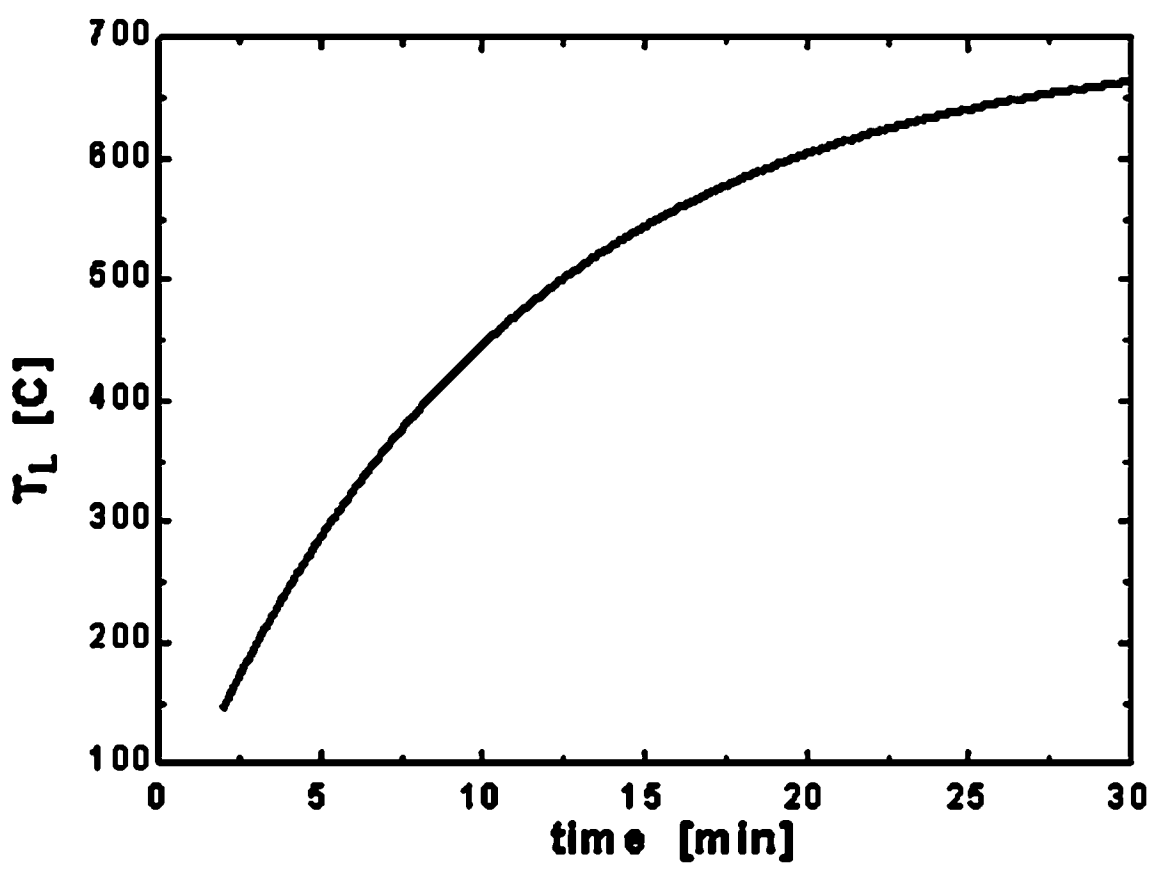
Solution will appear

time [min]	T_L [C]
2	146.7
4	244.8
6	325.5
8	391.9
10	446.5
12	491.5
14	528.5
16	558.9
18	583.9
20	604.5
22	621.4
24	635.4
26	646.8
28	656.2
30	664

Step 11 of 12

Then from Plot -> New plot window -> X-Y Plot
Select X-axis time and Y-axis T_L
Press OK
Graph will be plotted.

Step 12 of 12



Question: 18.045P

A long 35-cm-diameter cylindrical shaft made of stainless steel 304 (k = 14.9 W/m2·K, ρ = 7900 kg/m3 , cp = 477 J/kg·K, and α = 3.95 × 10-6 m2 /s) comes out of an oven at a uniform temperature of 500°C. The shaft is then allowed to cool slowly in a chamber at 150°C with an average convection heat transfer coefficient of h = 60 W/m2 ·K. Determine the temperature at the center of the shaft 20 min after the start of the cooling process. Also, determine the heat transfer per unit length of the shaft during this time period.

Step 1 of 7

Calculate the Biot number by using the equation as shown below.

$$\text{Bi} = \frac{hr_o}{k}$$

Here, the heat transfer coefficient is h, the radius is r_o and the thermal conductivity is k.

Substitute $60\ \text{W/m}^2\cdot\text{K}$ for h, $0.175\ \text{m}$ for r_o and $14.9\ \text{W/m}\cdot\text{K}$ for k.

$$\text{Bi} = \frac{(60\ \text{W/m}^2\cdot\text{K})(0.175\ \text{m})}{(14.9\ \text{W/m}\cdot\text{K})}$$
$$= 0.705$$

Since $Bi > 0.1$, and thus lumped system analysis is not applicable.

Step 2 of 7

From the coefficient used for in dimensional approximation of transient one dimensional heat conduction in cylinder,

Obtain the values of the coefficients λ and A for a cylinder corresponding to this Biot number.

$$\lambda = 1.0935\ ,\ A = 1.1558$$

Calculate the Fourier number by using the equation as shown below.

$$t = \frac{\tau r_o^2}{\alpha}$$

Here, the thermal diffusivity is α.

Step 3 of 7

Substitute $(20\times 60)\text{sec}$ for t and $3.95\times10^{-6}\ \text{m}^2/\text{s}$ for α.

$$20\times 60 = \frac{\tau\times 0.175^2}{3.95\times10^{-6}}$$
$$\tau = 0.155$$

This is very close to the value of 0.2.

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable with an error of more than 2 percent.

Step 4 of 7

Calculate the temperature at the center of the shaft by using the equation as shown below.

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}$$

Substitute the known values in the above equation.

$$\frac{T(t)-150}{500-150} = 1.1558e^{-(1.0935)^2\times0.155}$$
$$T(t) = 486°\text{C}$$

Step 5 of 7

Therefore, the temperature at the center of the shaft is $\boxed{486°\text{C}}$.

Determine the maximum heat can be transferred from the cylinder per meter of its length.

Calculate the mass of the stainless steel cylinder per meter of its length.

$$m = \rho V$$

Step 4 of 7

Calculate the temperature at the center of the shaft by using the equation as shown below.

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}$$

Substitute the known values in the above equation.

$$\frac{T(t)-150}{500-150} = 1.1558e^{-(1.0935)^2\times0.155}$$
$$T(t) = 486°\text{C}$$

Step 5 of 7

Therefore, the temperature at the center of the shaft is $\boxed{486°\text{C}}$.

Determine the maximum heat can be transferred from the cylinder per meter of its length.

Calculate the mass of the stainless steel cylinder per meter of its length.

$$m = \rho V$$

1	0.440
1.1	0.4709
1.0935	x

$$\frac{1.0935-1}{1.1-1} = \frac{x-0.440}{0.4709-0.440}$$
$$\frac{0.0935}{0.1} = \frac{x-0.440}{0.0309}$$
$$x = 0.4689$$

Step 7 of 7

Calculate the actual heat transfer by using the equation as shown below.

$$\left(\frac{Q}{Q_{max}}\right)_{cyl} = 1-2\left(\frac{T_0-T_\infty}{T_i-T_\infty}\right)\frac{J_1(\lambda_1)}{\lambda_1}$$

Substitute the known values in the above equation.

$$\frac{Q}{126882} = 1-2\left(\frac{486-150}{500-150}\right)\times\frac{0.4689}{1.0935}$$
$$Q = 22420\ \text{kJ}$$

Therefore, heat transfer per unit length is $\boxed{Q = 22420\ \text{kJ}}$

Question: 18.046P

Reconsider Prob. 18-45. Using EES (or other) software, investigate the effect of the cooling time on the final center temperature of the shaft and the amount of heat transfer. Let the time vary from 5 min to 60 min. Plot the center temperature and the heat transfer as a function of the time, and discuss the results.

Step 1 of 6

Given data:

Diameter of cylindrical shaft, $D = 35$ cm

Thermal conductivity of stainless steel at room temperature, $k = 14.9\ \text{W/m} \cdot \text{K}$

Density of stainless steel at room temperature, $\rho = 7900\ \text{kg/m}^3$

Specific heat of stainless steel at room temperature, $c_p = 477\ \text{J/kg} \cdot \text{K}$

Thermal diffusivity of stainless steel at room temperature, $\alpha = 3.95 \times 10^{-6}\ \text{m}^2/\text{s}$

Temperature of cylindrical shaft, $T_i = 500°\text{C}$

Heat transfer coefficient, $h = 60\ \text{W/m}^2.\text{K}$

Cooling Temperature, $T_\infty = 150°\text{C}$

Step 2 of 6

The procedure for developing solution by using EES software:

Open the new equations window EES software and type all the information which is used to solve the problem

For developing table:

1) Click on table followed by new parametric table and add the variables, which are varying variable and required answer variable, then one table will form with that variables

2) Click on the variable which is varying and enter initial value and final value.

3) For getting the required answer variable values corresponding to the varying variable values click on the green color button and then the table will form with required answers

For developing graph:

1) Click on plot followed by new plot window followed by x-y plot.

2) Select the x-axis variable and y-axis variable and then press ok, then the graph will form with the given values.

Step 3 of 6

The equations which are using in EES software for developing the solution are given in equations window is as shown below

File Edit Search Options Calculate Tables Plots Windows He

```
"GIVEN"
r_o=0.35/2 "[m]"
T_i=500 "[C]"
T_infinity=150 "[C]"
h=60 "[W/m^2-C]"
"time=20 [min], parameter to be varied"


"PROPERTIES"
k=14.9 "[W/m-C]"
rho=7900 "[kg/m^3]"
C_p=477 "[J/kg-C]"
alpha=3.95E-6 "[m^2/s]"


"ANALYSIS"
Bi=(h*r_o)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1=1.0935
A_1=1.1558
J_1=0.4709 "From Table corresponding to lambda_1"
tau=(alpha*time*Convert(min, s))/r_o^2
(T_o-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau)


L=1 "[m], 1 m length of the cylinder is considered"
V=pi*r_o^2*L
m=rho*V
Q_max=m*C_p*(T_i-T_infinity)*Convert(J, kJ)
Q/Q_max=1-2*(T_o-T_infinity)/(T_i-T_infinity)*J_1/lambda_1
```

Step 4 of 6

Select no of runs = 12

Select variables in equations as time and T_0 press ADD and then press ok

A table will appear, put values of time as 5,10,15....60 in run 1 to 12

Repeat procedure for time and Q

Then from Calculate -> Solve table

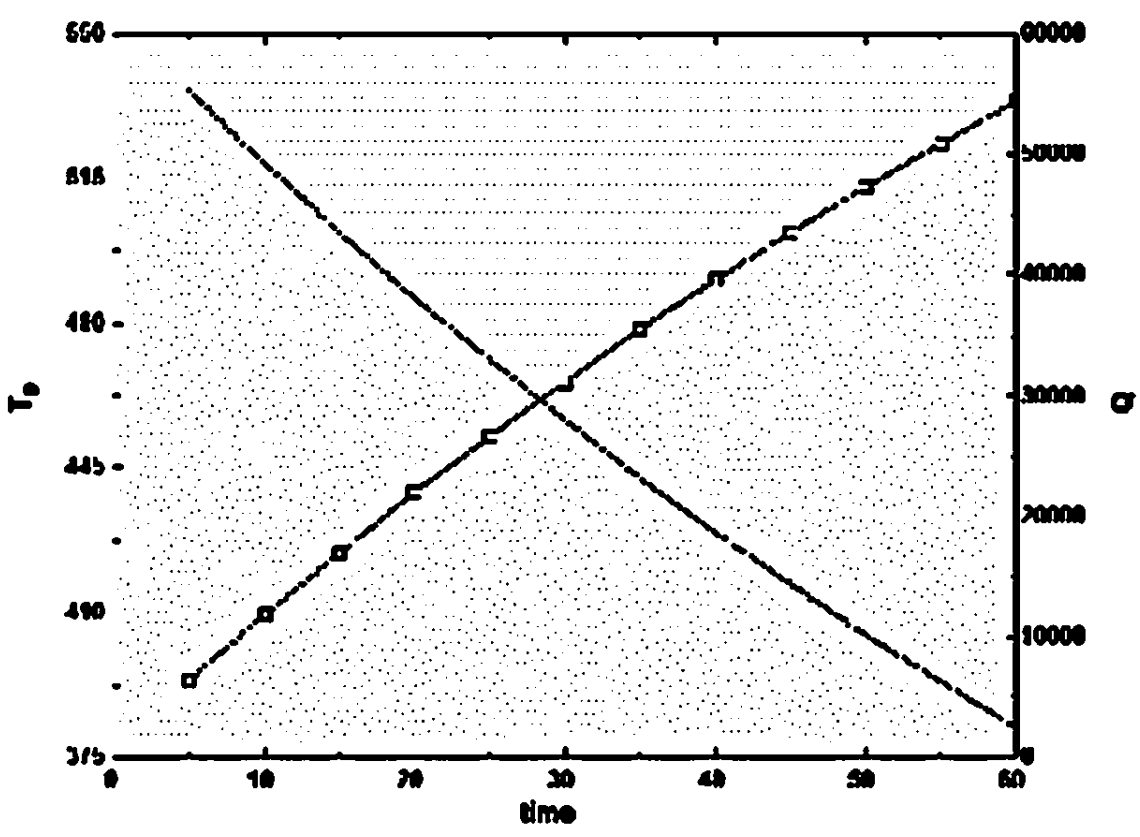
Step 5 of 6

The EES software is developing the table for Q, T_0 corresponding to the time values is as shown below,

1..12	Q	time	T_0
Run 1	6288	5	536.2
Run 2	11741	10	518.8
Run 3	16947	15	502.1
Run 4	21918	20	486.2
Run 5	26665	25	471
Run 6	31196	30	456.5
Run 7	35523	35	442.6
Run 8	39654	40	429.4
Run 9	43599	45	416.8
Run 10	47365	50	404.7
Run 11	50961	55	393.2
Run 12	54394	60	382.2

Step 6 of 6

The EES software is developing the graph between T_0 and time is as shown below,



Question: 18.047P

Long cylindrical AISI stainless steel rods (k = 7.74 Btu/h·ft·°F and α = 0.135 ft2 /h) of 4-in-diameter are heat treated by drawing them at a velocity of 7 ft/min through a 21-ft-long oven maintained at 1700°F. The heat transfer coefficient in the oven is 20 Btu/h·ft2 ·°F. If the rods enter the oven at 70°F, determine their centerline temperature when they leave.

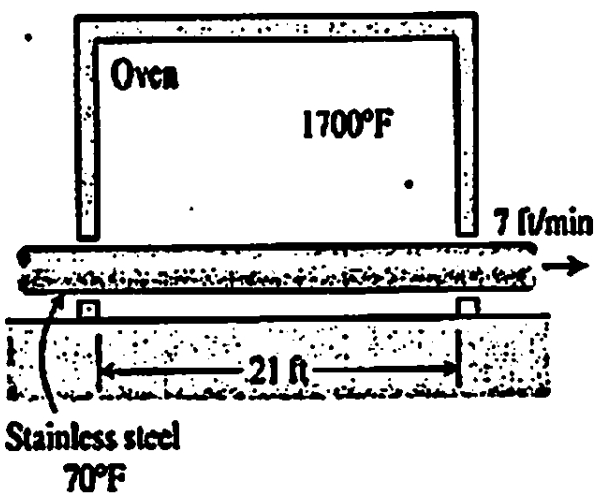


FIGURE P18-47E

Step 1 of 6

Assumptions:

Long cylindrical steel rods are heat-treated in an oven.

Heat conduction in the rods is one-dimensional since the rods are long and they have thermal symmetry about the center line.

The heat transfer coefficient is constant and uniform over the entire surface.

The Fourier number is > 0.2 so that the one-term approximate solutions are applicable.

Step 2 of 6

Find the time the steel rod stays in the oven by using the equation,

$$t = \frac{L}{V}$$

Here, L is length of the steel rod in an oven; V is velocity of steel rod drawing from the oven.

Substitute 21 ft for L, $7\,\text{ft/min}$ for V

$$t = \frac{21\,\text{ft}}{7\,\text{ft/min}}$$
$$= 3\,\text{min}$$
$$= \frac{3}{60}\,\text{h}$$

Step 3 of 6

Find the Fourier number by using the expression,

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, α is thermal diffusivity of the rod, r_o is radius of the rod.

Substitute $0.135\,\text{ft}^2/\text{h}$ for α and $(2/12)\,\text{ft}$ for r_o

$$\tau = \frac{(0.135\ \text{ft}^2/\text{h})(3/60\ \text{h})}{(2/12\ \text{ft})^2}$$
$$= 0.243$$

Here the Fourier is greater than 0.2, the one-term approximate solutions are applicable.

Step 4 of 6

Find the Biot number by using the expression,

$$\text{Bi} = \frac{hr_o}{k}$$

Here, h is heat transfer coefficient; k is thermal conductivity of the rod.

Substitute $20\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for h, $7.74\,\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$ for k

$$\text{Bi} = \frac{(20\ \text{Btu/h.ft}^2.{}^\circ\text{F})(2/12\ \text{ft})}{(7.74\ \text{Btu/h.ft.}{}^\circ\text{F})}$$
$$= 0.4306$$

Step 5 of 6

Obtain the constants λ_1 and A_1 corresponding to this Biot number from table "Coefficients used in the one term approximate solution of transient one dimensional heat conduction in cylinders" by using interpolation,

$$\lambda_1 = 0.8516 + (0.9408 - 0.8516)\left(\frac{0.4306 - 0.4}{0.5 - 0.4}\right)$$
$$= 0.8788$$

$$A_1 = 1.0931 + (1.1143 - 1.0931)\left(\frac{0.4306 - 0.4}{0.5 - 0.4}\right)$$
$$= 1.0995$$

Step 6 of 6

Find the temperature at the center of the rod, T_0 by using the equation,

$$\frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Here, T_∞ is temperature of the oven, T_i is initial temperature of the steel rod that enters in to the oven

Substitute $1700°\text{F}$ for T_∞ and $70°\text{F}$ for T_i

$$\frac{x - 1700}{70 - 1700} = (1.0995)e^{-(0.8788)^2(0.243)}$$
$$T_0 = 214.474°\text{F}$$

Therefore, the center line temperature of the rod when it leaves from the oven is $\boxed{214.474°\text{F}}$

Question: 18.048P

In a meat processing plant, 2-cm-thick steaks (k = 0.45 W/m·K and α = 0.91 × 10-7 m2 /s) that are initially at 25°C are to be cooled by passing them through a refrigeration room at -11°C. The heat transfer coefficient on both sides of the steaks is 9 W/m2 ·K. If both surfaces of the steaks are to be cooled to 2°C, determine how long the steaks should be kept in the refrigeration room.

Step 1 of 4

Given:

Thickness of steaks $t = 2$ cm

Thermal conductivity of stainless steel at room temperature $k = 0.45$ W/m.°C

Thermal diffusivity of stainless steel at room temperature $\alpha = 0.91\times10^{-7}$ m^2/s

Initial temperature of meat $T_i = 25°C$

Temperature of refrigeration room $T_\infty = -11°C$

Heat transfer coefficient on both side of steak $h = 9$ W/m^2.°C

Step 2 of 4

Steaks are cooled by passing them through a refrigeration room.

Heat conduction in the steaks is one-dimensional since the steaks are large relative to their thickness and there is thermal symmetry about the center plane. The thermal properties of the steaks are constant. The heat transfer coefficient is constant and uniform over the entire surface.

The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Step 3 of 4

The Biot number $Bi = \dfrac{hL}{k}$

$$Bi = \frac{(9\ \text{W/m}^2.°\text{C})(0.01\ \text{m})}{(0.45\ \text{W/m.°C})}$$

$$Bi = 0.200$$

The constants λ_1 and A_1 corresponding to this Biot number are, from 11-2 tables

$$\lambda_1 = 0.4328 \quad \text{and} \quad A_1 = 1.0311$$

The Fourier number

$$\frac{T(L,t)-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 L/L)$$

$$\frac{2-(-11)}{25-(-11)} = (1.0311)e^{-(0.4328)^2 \tau} \cos(0.4328)$$

$$\tau = 5.601 > 0.2$$

Step 4 of 4

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Then the length of time for the steaks to be kept in the refrigerator is determined to be

$$t = \frac{\tau L^2}{\alpha}$$

$$t = \frac{(5.601)(0.01\ \text{m})^2}{(0.91\times10^{-7}\ \text{m}^2/\text{s})}$$

$$t = 6155\ \text{s}$$

$$\boxed{\text{t=102.6 min}}$$

Question: 18.049P

A student calculates that the total heat transfer from a spherical copper ball of diameter 18 cm initially at 200°C and its environment at a constant temperature of 25°C during the first 20 min of cooling is 3150 kJ. Is this result reasonable? Why?

Step 1 of 3

Given data:

Diameter of the spherical ball is $D = 18\text{cm} = 0.18\text{ m}$

Initially temperature of the ball is $T_i = 200°\text{C}$

Ambient temperature is $T_\infty = 25°\text{C}$

Step 2 of 3

From the properties of solid metal

For copper,

Density, $\rho = 8933\ \text{kg/m}^3$

Specific heat, $c_p = 0.385\ \text{KJ/kg-K}$

Mass of cylinder is given by

$$m = \rho V$$

$$= 8933 \times \frac{\pi \times 0.18^3}{8}$$

$$= 20.45\ \text{Kg}$$

Step 3 of 3

Maximum heat transfer from copper ball is

$$Q = mc_p(T_i - T_\infty)$$

$$= mc_p(200 - 25)$$

$$= 20.45 \times 0.385 \times (200 - 25)$$

$$= 1378.39\ \text{kJ}$$

Therefore, the student's result is not reasonable as 3150 KJ is much greater than maximum heat transfer data i.e. 1378.39 kJ .

Question: 18.050P pg 1

In *Betty Crocker's Cookbook*, it is stated that it takes 2 h 45 min to roast a 3.2-kg rib initially at 4.5°C "rare" in an oven maintained at 163°C. It is recommended that a meat thermometer be used to monitor the cooking, "and the rib is considered rare done when the thermometer inserted into the center of the thickest part of the meat registers 60°C. The rib can be treated as a homogeneous spherical object with the properties ρ = 1200 kg/m3 , cp = 4.1 kJ/kg·K, k = 0.45 W/m2·K, and α = 0.91× 10 -7 m2 /s. Determine (*a*) the heat transfer coefficient at the surface of the rib; (*b*) the temperature of the outer surface of the rib when it is done; and (*c*) the amount of heat transferred to the rib. (*d*) Using the values obtained, predict how long it will take to roast this rib to "medium" level, which occurs when the innermost temperature of the rib reaches 71°C. Compare your result to the listed value of 3 h 20 min.

If the roast rib is to be set on the counter for about 15 min before it is sliced, it is recommended that the rib be taken out of the oven when the thermometer registers about 4°C below the indicated value because the rib will continue cooking even after it is taken out of the oven. Do you agree with this recommendation?

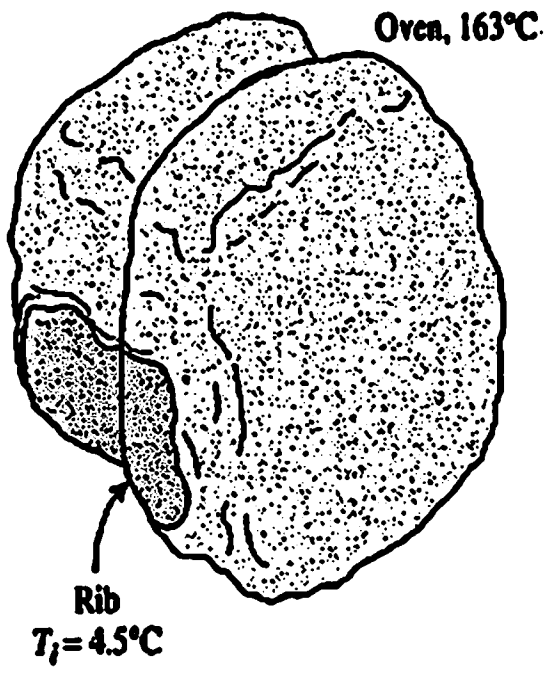


FIGURE P 18-50

Step 1 of 10

(a)

Calculate the volume of the roast.

$$V = \frac{m}{\rho}$$

Here, V is the volume of the roast, m is the mass of the roast, and ρ is the density of the roast.

Substitute 3.2 kg for m and 1200kg/m^3 for ρ .

$$V = \frac{3.2}{1200}$$
$$= 0.002667 \text{m}^3$$

Step 2 of 10

Calculate the radius of the roast.

$$V = \frac{4}{3}\pi r_o^3$$

$$r_o = \left(\frac{3V}{4\pi}\right)^{\frac{1}{3}}$$

Here, r_o is the radius of the roast.

Substitute 0.002667m^3 for V.

$$r_o = \left(\frac{3 \times 0.002667}{4\pi}\right)^{\frac{1}{3}}$$
$$= 0.08603 \text{m}$$

Step 3 of 10

Convert the time from hours to seconds.

$$t = 2\text{hr}45\text{min}$$

Calculate the Fourier number.

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, τ is the Fourier number, α is the thermal expansion coefficient, and t is the time.

Substitute $0.91 \times 10^{-7} \text{m}^2/\text{s}$ for α , 2 hr 45 min for t, 0.08603 m for r_o .

$$\tau = \frac{0.91 \times 10^{-7} \text{m}^2/\text{s} \times 2\text{hr}45\text{min}}{(0.08603\text{m})^2}$$

$$\tau = \frac{(0.91 \times 10^{-7}\ \text{m}^2/\text{s})(2 \times 3600 + 45 \times 60)\text{s}}{(0.08603\ \text{m})^2}$$
$$= 0.1217$$

Which is somewhat below the value of 0.2, therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent.

Step 4 of 10

Write the one-term solution for sphere.

$$\theta_{0,\text{sph}} = A_1 e^{-\lambda_1^2 \tau}$$
$$\frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Here, T_0 is the temperature of the roast at the center, T_i is the initial temperature, and T_∞ is the temperature in oven, A_1 and λ_1 are the constants.

Substitute $60°\text{C}$ for T_0 , $4.5°\text{C}$ for T_i , $163°\text{C}$ for T_∞ , and 0.1217 for τ .

$$\frac{60 - 163}{4.5 - 163} = A_1 e^{-\lambda_1^2 (0.1217)} \quad \text{..... (1)}$$
$$0.65 = A_1 e^{-\lambda_1^2 (0.1217)}$$

$$\frac{60-163}{4.5-163} = A_1 e^{-\lambda_1^2(0.1217)} \quad \text{..... (1)}$$

$$0.65 = A_1 e^{-\lambda_1^2(0.1217)}$$

Step 5 of 10

Use trial and error method to obtain the values of A_1 and λ_1 .

Trail-1: Assume $\lambda_1 = 3.0372$ and $A_1 = 1.9898$.

Substitute 1.9898 for A_1 , 3.0372 for λ_1 in equation (1).

$$0.65 = A_1 e^{-\lambda_1^2(0.1217)}$$

$$0.65 = 1.9898 \times e^{-(3.0372)^2(0.1217)}$$

$$0.65 = 0.65$$

Here, $\text{LHS} = \text{RHS}$, therefore, take $\lambda_1 = 3.0372$ and $A_1 = 1.9898$.

Obtain the Biot number at $\lambda_1 = 3.0372$ and $A_1 = 1.9898$ from the table 11-2 "coefficients used in the one-term approximate solution of transient one dimensional heat conduction in plane walls, cylinders, and spheres" of the text book.

Biot number, $Bi = 30$.

Step 6 of 10

Calculate the heat transfer coefficient by using the following relation.

$$Bi = \frac{hr_o}{k}$$

$$h = \frac{kBi}{r_o}$$

Substitute $0.45\ \text{W/m.°C}$ for *k*, 30 for *Bi*, and 0.08603 m for r_o .

$$h = \frac{(0.45\ \text{W/m.°C})(30)}{(0.08603\ \text{m})}$$

$$= 156.922\ \text{W/m}^2\text{.°C}$$

Therefore, the heat transfer coefficient is, $\boxed{156.9\ \text{W/m}^2\text{.°C}}$.

Step 7 of 10

(b)

Calculate the temperature at the surface of the rib by using the following relation.

$$\theta(r_o,t)_{sph} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\frac{T(r_o,t)-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1)}{\lambda_1}$$

Here, $T(r_o,t)$ is the temperature at the surface of the rib.

Substitute $4.5°\text{C}$ for T_i , $163°\text{C}$ for T_∞ , 0.1217 for τ , 1.9898 for A_1 , 3.0372 for λ_1 , and 0.08603 m for r_o .

$$\frac{T(r_o,t)-163}{4.5-163} = (1.9898) e^{-(3.0372)^2(0.1217)} \frac{\sin(3.0372\ \text{rad})}{3.0372}$$

$$\frac{T(r_o,t)-163}{4.5-163} = 0.0222$$

$$T(r_o,t) = 159.5°\text{C}$$

Therefore, the temperature of the outer surface of the rib is, $\boxed{159.5\ °\text{C}}$.

Step 8 of 10

(c)

Calculate the maximum possible heat transfer.

$$Q_{max} = mC_p(T_\infty - T_i)$$

Here, Q_{max} is the maximum possible heat transfer and C_p is the constant pressure specific heat capacity.

Substitute 3.2 kg for *m*, $4.1\ \text{kJ/kg.°C}$ for C_p , $4.5°\text{C}$ for T_i , and $163°\text{C}$ for T_∞

$$Q_{max} = (3.2\ \text{kg})(4.1\ \text{kJ/kg.°C})(163-4.5)°\text{C}$$

$$= 2080\ \text{kJ}$$

Calculate the actual amount of heat transfer.

$$\frac{Q}{Q_{max}} = 1 - 3\theta_{o,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3}$$

Here, Q is the actual heat transfer.

Substitute 2080 kJ for Q_{max} , 0.65 for $\theta_{o,sph}$, and 3.0372 for λ_1 .

$$\frac{Q}{2080} = 1 - 3(0.65) \frac{\sin(3.0372) - (3.0372)\cos(3.0372)}{(3.0372)^3}$$

$$\frac{Q}{2080} = 0.783$$

$$Q = 1629\ \text{kJ}$$

Therefore, the actual amount of heat transfer is, $\boxed{1629\ \text{kJ}}$.

Step 9 of 10

(d)

Calculate the Fourier number at innermost temperature of rib is $71°\text{C}$.

$$\theta_{o,sph} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Calculate the Fourier number at innermost temperature of rib is 71°C .

18.0302 pg 3

$$\theta_{0,rib} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Substitute $71°C$ for T_0 , $4.5°C$ for T_i , $163°C$ for T_∞ , 1.9898 for A_1 , and 3.0372 for λ_1 .

$$\frac{71-163}{4.5-163} = (1.9898)e^{-(3.0372)^2 \tau}$$

$$\tau = 0.1336$$

Step 10 of 10

Calculate the cooking time for medium-done rib.

$$t = \frac{\tau r_o^2}{\alpha}$$

Here, t is the cooking time for medium-done rib.

Substitute 0.1336 for τ , 0.08603 m for r_o , and $0.91 \times 10^{-7}\ m^2/s$ for α .

$$t = \frac{(0.1336)(0.08603\ m)^2}{(0.91 \times 10^{-7}\ m^2/s)}$$

$$\approx 10866\ s$$

$$\approx 181\ min$$

$$\approx 3hr$$

Therefore, the cooking time for medium-done rib is, $\boxed{3hr}$.

Question: 18.051P

Repeat Prob. 18-50 for a roast rib that is to be "well-done" instead of "rare." A rib is considered to be well-done when its center temperature reaches 77°C, and the roasting in this case takes about 4 h 15 min.

Step 1 of 6

Loading solution. Please wait ...

Step 2 of 6

The rib is a homogeneous spherical object.
Heat conduction in the rib is one-dimensional because of symmetry about the midpoint.
The thermal properties of the rib are constant.
The heat transfer coefficient is constant and uniform over the entire surface.
The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).
(a) The heat transfer coefficient can be determined from
First the radius of the rib is determined to be
The mass $m = \rho V$

$$V = \frac{m}{\rho}$$

$$V = \frac{3.2\ \text{kg}}{1200\ \text{kg/m}^3}$$

$$V = 0.00267\ \text{m}^3$$

$$V = \frac{4}{3}\pi r_o^3$$

Step 3 of 6

The radius $r_o = \sqrt[3]{\frac{3V}{4\pi}}$

$$r_o = \sqrt[3]{\frac{3(0.00267\ \text{m}^3)}{4\pi}}$$

$$r_o = 0.08603\ \text{m}$$

The Fourier number $\tau = \frac{\alpha t}{r_o^2}$

$$\tau = \frac{(0.91\times10^{-7}\ \text{m}^2/\text{s})(4\times3600+15\times60)\text{s}}{(0.08603\ \text{m})^2}$$

$$\tau = 0.1881$$

Which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the one-term solution formulation can be written in the form

$$\theta_{0,\text{spl}} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$\theta_{0,\text{spl}} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{77-163}{4.5-163} = 0.54$$

$$0.54 = A_1 e^{-\lambda_1^2 (0.1881)}$$

It is determined from Biot number table by trial and error that this equation is satisfied when $Bi = 4.3$, which corresponds to $\lambda_1 = 2.4900$ and $A_1 = 1.7402$. Then the heat transfer coefficient can be determined from.

$$Bi = \frac{hr_o}{k}$$

$$h = \frac{kBi}{r_o}$$

$$h = \frac{(0.45\ \text{W/m.°C})(4.3)}{(0.08603\ \text{m})}$$

$$\boxed{h = 22.5\ \text{W/m}^2.\text{°C}}$$

Step 4 of 6

(b) The temperature at the surface of the rib is

$$\theta(r_o,t)_{\text{spl}} = \frac{T(r_o,t) - T_\infty}{T_i - T_\infty}$$

$$\theta(r_o,t)_{\text{spl}} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\theta(r_o,t)_{\text{spl}} = (1.7402) e^{-(2.49)^2 (0.1881)} \frac{\sin(2.49)}{2.49}$$

$$\frac{T(r_o,t) - 163}{4.5-163} = 0.132$$

$$\boxed{T(r_o,t) = 142.1\ \text{°C}}$$

Step 5 of 6

(c) The maximum possible heat transfer is

$$Q_{\max} = mC_p(T_\infty - T_i)$$

$$Q_{\max} = (3.2\ \text{kg})(4.1\ \text{kJ/kg.°C})(163-4.5)\text{°C}$$

$$Q_{\max} = 2080\ \text{kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{o,\text{spl}} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3}$$

$$\frac{Q}{Q_{\max}} = 1 - 3(0.543) \frac{\sin(2.49) - (2.49)\cos(2.49)}{(2.49)^3}$$

$$\frac{Q}{Q_{\max}} = 0.727$$

$$Q = 0.727 Q_{\max}$$

$$Q = (0.727)(2080\ \text{kJ})$$

$$\boxed{Q = 1512\ \text{kJ}}$$

Step 6 of 6

(d) The cooking time for medium-done rib is determined to be

$$\theta_{0,\text{spl}} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$\theta_{0,\text{spl}} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{71-163}{4.5-163} = (1.7402) e^{-(2.49)^2 \tau}$$

$$\tau = 0.177$$

$$t = \frac{\tau r_o^2}{\alpha}$$

$$t = \frac{(0.177)(0.08603\ \text{m})^2}{(0.91\times10^{-7}\ \text{m}^2/\text{s})}$$

$$t = 14{,}403\ \text{s}$$

$$t = 240.0\ \text{min}$$

$$\boxed{t = 4\ \text{hr}}$$

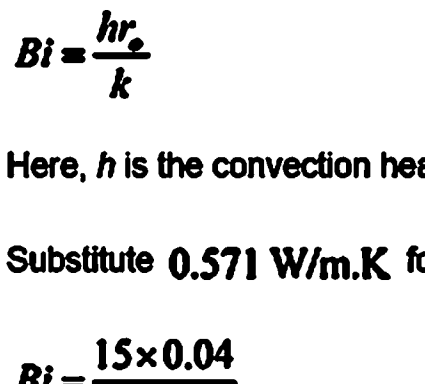
Question: 18.052P

Citrus fruits are very susceptible to cold weather, and extended exposure to subfreezing temperatures can destroy them. Consider an 8-cm-diameter orange that is initially at 15°C. A cold front moves in one night, and the ambient temperature suddenly drops to -6°C, with a heat transfer coefficient of 15 W/m2 ·K. Using the properties of water for the orange and assuming the ambient conditions to remain constant for 4 h before the cold front moves out, determine if any part of the orange will freeze that night.

Step 1 of 6

Draw the schematic diagram of the orange.

Step 2 of 6



Step 3 of 6

Obtain the properties of orange which are approximated by those of water at the average temperature of $5°C$ from the Table A-9, "Properties of saturated water" from the text book.

Thermal conductivity, $k = 0.571\ \text{W/m.K}$

Specific heat $C_p = 4205\ \text{J/kg.k}$

Calculate the thermal diffusivity if the orange by using the following relation:

$$\alpha = \frac{k}{\rho C_p}$$

Here, ρ is the density.

Substitute $0.571\ \text{W/m.K}$ for k, $1000\ \text{kg/m}^3$ for ρ, and $4205\ \text{J/kg.k}$ for C_p.

$$\alpha = \frac{0.571}{1000 \times 4205}$$
$$= 0.136 \times 10^{-6}\ \text{m}^2/\text{s}$$

Step 4 of 6

Calculate the Biot number.

$$Bi = \frac{hr_o}{k}$$

Here, h is the convection heat transfer coefficient, and r_o is the radius of the sphere.

Substitute $0.571\ \text{W/m.K}$ for k, $15\ \text{W/m}^2\text{.K}$ for h, and $0.04\ \text{m}$ for r_o.

$$Bi = \frac{15 \times 0.04}{0.571}$$
$$= 1.05$$
$$\approx 1$$

Obtain the constants λ_1 and A_1 corresponding to this Biot number:

$$\lambda_1 = 1.5708$$
$$A_1 = 1.2732$$

Step 5 of 6

Calculate the Fourier number.

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, t is the time.

Substitute $0.136 \times 10^{-6}\ \text{m}^2/\text{s}$ for α, $4\ \text{h}$ for t, and $0.04\ \text{m}$ for r_o.

$$\tau = \frac{(0.136 \times 10^{-6}\ \text{m}^2/\text{s})(4\ \text{h} \times \frac{3600\ \text{s}}{\text{h}})}{(0.04\ \text{m})^2}$$
$$= 1.224 > 0.2$$

Therefore the one-term approximate solution is applicable.

Step 6 of 6

Calculate the surface temperature of the oranges from the following relation:

$$\theta(r_o, t)_{sph} = \frac{T(r_o, t) - T_\infty}{T_i - T_\infty}$$
$$= A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

Substitute 1.5708 for λ_1, 1.2732 for A_1, and 1.224 for τ.

$$\theta(r_o, t)_{sph} = (1.2732) e^{-(1.5708)^2 (1.224)} \frac{\sin(1.5708\ \text{rad})}{1.5708}$$
$$\theta(r_o, t)_{sph} = 0.0396$$
$$\frac{T(r_o, t) - (-6)}{15 - (-6)} = 0.0396$$
$$T(r_o, t) = -5.2°C < 0°C$$

Therefore, the oranges will freeze.

Question: 18.053P

In a chicken processing plant, whole chickens averaging 5 lbm each and initially at 65°F are to be cooled in the racks of a large refrigerator that is maintained at 5°F. The entire chicken is to be cooled below 45°F, but the temperature of the chicken is not to drop below 35°F at any point during refrigeration. The convection heat transfer coefficient and thus the rate of heat transfer from the chicken can be controlled by varying the speed of a circulating fan inside, Determine the heat transfer-coefficient that will enable us to meet both temperature constraints while keeping the refrigeration time to a minimum. The chicken can be treated as a homogeneous spherical object having the properties ρ= 74.9 lbm/ft3 , *cp* = 0.98 Btu/lbm-°F, *k* - 0.26 BtuM· °F, and α = 0.0035 ft2 /h.

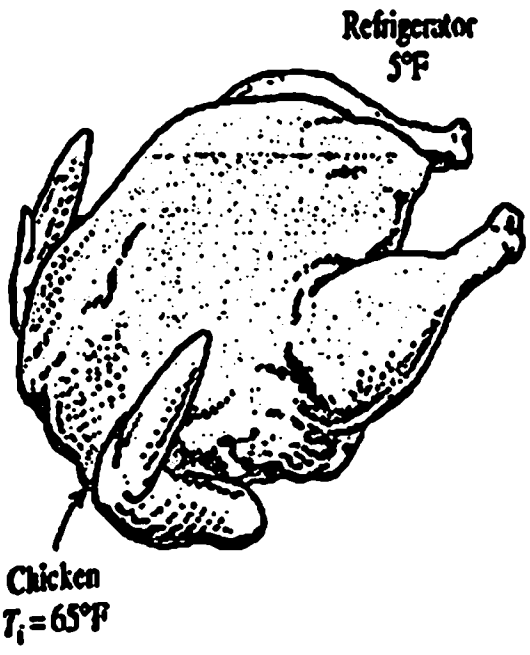


FIGURE P18-53E

Step 1 of 3

Given:

Initial temperature of chicken $T_i = 65°F$

Refrigerators racks temperature $T_\infty = 5°F$

Chicken cooled temperature $T_o = 45°F$

The chicken not to drop below temperature $T = 35°F$

The chicken is a homogeneous spherical object. Heat conduction in the chicken is one-dimensional because of symmetry about the midpoint. The thermal properties of the chicken are constant. The heat transfer coefficient is constant and uniform over the entire surface.

The thermal conductivity of the chicken $k = 0.26 \frac{\text{Btu}}{\text{h.ft.°F}}$

The density of the chicken $\rho = 74.9 \frac{\text{lbm}}{\text{ft}^3}$,

The sp.heat of the chicken $C_p = 0.98 \frac{\text{Btu}}{\text{lbm.°F}}$

The thermal diffusivity of the chicken $\alpha = 0.003598 \frac{\text{ft}^2}{\text{h}}$

Step 2 of 3

The radius of the chicken is determined to be

The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

The mass $m = \rho V$

$$V = \frac{m}{\rho}$$

$$V = \frac{5\ \text{lbm}}{74.9\ \text{lbm/ft}^3}$$

$$V = 0.06676\ \text{ft}^3$$

$$V = \frac{4}{3}\pi r_o^3$$

The radius $r_o = \sqrt[3]{\frac{3V}{4\pi}}$

$$r_o = \sqrt[3]{\frac{3(0.06676\ \text{ft}^3)}{4\pi}}$$

$$r_o = 0.2517\ \text{ft}$$

Step 3 of 3

we have

$$\frac{T - T_\infty}{T_o - T_\infty} = \frac{35 - 5}{45 - 5}$$

$$\frac{T - T_\infty}{T_o - T_\infty} = 0.75$$

$$\frac{r}{r_o} = \frac{r_o}{r_o} = 1$$

$$\frac{1}{Bi} = \frac{k}{hr_o} = 1.75$$

Then the heat transfer coefficients becomes

$$h = \frac{k}{1.75 r_o}$$

$$h = \frac{(0.26\ \text{Btu/h.ft.°F})}{1.75(0.2517\ \text{ft})}$$

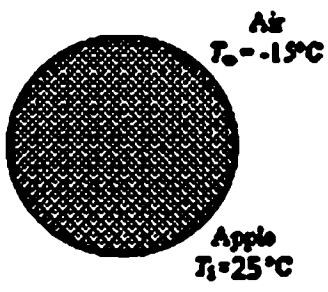
$$\boxed{h = 0.590 \frac{\text{Btu}}{\text{h.ft}^2\text{.°F}}}$$

Question: 18.054P

A person puts a few apples into the freezer at -15°C to cool them quickly for guests who are about to arrive. Initially, the apples are at a uniform temperature of 25°C, and the heat transfer coefficient on the surfaces is 8 W/m2 K. Treating the apples as 9-cm-diameter spheres and taking their properties to be ρ = 840 kg/m3 , cp = 3.81 kJ/kg·K. k = 0.418 W/m·K, and α = 1.3 × 10 7 m2 /s. determine the center and surface temperatures of the apples in 1 h. Also, determine the amount of heat transfer from each apple.

Step 1 of 5

Let us draw the diagram:



Step 2 of 5

A person puts apples into the freezer to cool them quickly. Heat conduction in the apples is one-dimensional because of symmetry about the midpoint. The thermal properties of the apples are constant. The Fourier number is > 0.2 so that the one-term approximate solutions (or the transient temperature charts) are applicable.

Characteristic c length of the s[here is

$$r_0 = \frac{9}{2}$$
$$= 4.5\text{cm}$$
$$= 0.045\text{m}$$

Then Biot number is

$$Bi = \frac{hL_c}{K}$$
$$= \frac{8 \times 0.045}{0.418}$$
$$= 0.861$$

Since Biot number is greater than 0.1, and thus lumped system analysis is not applicable. The coefficient λ_1 and A_1 for a cylinder corresponding to the Bi number are

From the coefficient used for in dimensional approximation of transient one dimensional heat conduction in sphere is

For this Biot number, we get: $\lambda = 1.476, A = 1.2390$

Step 3 of 5

Therefore, Fourier number is obtained to be:

$$t = \frac{\tau r^2}{\alpha}$$
$$3600 = \frac{\tau \times 0.045^2}{1.3 \times 10^{-7}}$$
$$\tau = 0.231$$

Then the temperature at the center of the apples is determined from the dimensional approximation

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$
$$\frac{T(t) + 15}{25 + 15} = 1.2392e^{-1.476^2 \times 0.231}$$
$$T(t) = 14.96^\circ\text{C}$$

So, the temperature at the centre of apple is $\boxed{T(t) = 14.96^\circ\text{C}}$.

Step 4 of 5

Now for temperature at the surface of apple we use the expression:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau} \frac{\sin(\lambda)}{\lambda}$$
$$\frac{T(t) + 15}{25 + 15} = 1.2392e^{-1.476^2 \times 0.231} \times \frac{\sin(1.476)}{1.476}$$
$$T(t) = 5.21^\circ\text{C}$$

Therefore, the temperature at the surface of apple is: $\boxed{T(t) = 5.21^\circ\text{C}}$.

Step 5 of 5

The maximum possible heat transfer is

$$m = \rho V$$
$$= \rho \frac{4}{3} \pi r_o^3$$
$$= (840\ \text{kg/m}^3)\left[\frac{4}{3}\pi(0.045\ \text{m})^3\right]$$
$$= 0.3206\ \text{kg}$$

Maximum heat transfer from apple is

Then the actual amount of heat transfer is determined from the fraction of heat transfer for sphere is

Therefore, heat transfer per unit length is 1.

Question: 18.055P

Reconsider Prob. 18-54. Using EES (or other) software, investigate the effect of the initial temperature of the apples on the final center and surface temperatures and the amount of heat transfer. Let the initial temperature vary from 2°C to 30°C. Plot the center temperature, the surface temperature, and the amount of heat transfer as a function of the initial temperature, and discuss the results.

Step 1 of 6

Loading solution. Please wait ...

Step 2 of 6

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
T_infinity=-15 "[C]"
"T_i=25 [C], parameter to be varied"
h=8 "[W/m^2-C]"
r_o=0.09/2 "[m]"
time=1*3600 "[s]"

"PROPERTIES"
k=0.418 "[W/m-C]"
rho=840 "[kg/m^3]"
C_p=3.81 "[kJ/kg-C]"
alpha=1.3E-7 "[m^2/s]"
```

Step 3 of 6

```
"ANALYSIS"
Bi=(h*r_o)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1=1.476
A_1=1.239
tau=(alpha*time)/r_o^2
(T_o-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau)

(T_r-T_infinity)/(T_i-T_infinity)=A_1*exp(-
lambda_1^2*tau)*(Sin((lambda_1*r_o/r_o)*180/3.14))/(lambda_1*r_o/r_o)

V=4/3*pi*r_o^3
m=rho*V
Q_max=m*C_p*(T_i-T_infinity)
Q/Q_max=1-3*(T_o-T_infinity)/(T_i-T_infinity)*(Sin(lambda_1*180/3.14)-
lambda_1*Cos(lambda_1*180/3.14))/lambda_1^3
```

Step 4 of 6

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 15
Select variables in equations as T_i and T_o press ADD and then press ok
A table will appear put values of T_i as 2,4,6....30 in run 1 to 15
Repeat procedure for T_i and T_r and T_i and Q
Then from Calculate -> Solve table

Step 5 of 6

Loading solution. Please wait ...

Step 6 of 6

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_i and Y-axis T_r, T_o and Q
Press OK
Graph will be plotted.

25

Run 10	20	11.21	2.679	17.16
Run 11	22	12.71	3.69	18.42
Run 12	24	14.21	4.7	19.72
Run 13	26	15.7	5.71	20.1
Run 14	28	17.2	6.72	21.08
Run 15	30	18.7	7.73	22.06

Step 6 of 6

Then from Plot -> New plot window -> X-Y Plot
Select X-axis T_i and Y-axis T_r, T_o and Q
Press OK
Graph will be plotted.

25

Question: 18.056P

A 9-cm-diameter potato (ρ = 1100 kg/m3 , cp = 3900 J/kg·K, k = 0.6 W/m·K, and α = 1.4 × 10-7 m2 /s) that is initially at a uniform temperature of 25°C is baked in an oven at 170°C until a temperature sensor inserted to the center of the potato indicates a reading of 70°C. The potato is then taken out of the oven and wrapped in thick J that almost no heat is lost from the baked potato. Assuming the heat transfer coefficient in the oven to be 40 W/m2 ·K determine (*a*) how long the potato is baked in the (*b*) the final equilibrium temperature of the potato after it is Wrapped.

Step 1 of 6

Given:

Diameter of potato $D = 8\text{cm}$

Thermal conductivity of the potato $k = 0.6\ \text{W/m.°C}$

Density of the potato $\rho = 1000\,\frac{\text{kg}}{\text{m}^3}$,

Specific heat of potato $C_p = 3.9\,\frac{\text{kJ}}{\text{kg.°C}}$

Thermal diffusivity of potato $\alpha = 1.4\times10^{-7}\,\frac{\text{m}^2}{S}$.

Initial temperature of potato $T_i = 25°\text{C}$

Temperature at the center of potato $T_o = 70°\text{C}$

Oven temperature of potato $T_\infty = 170°\text{C}$

Step 2 of 6

Heat conduction in the potato is one-dimensional because of symmetry about the midpoint.

The thermal properties of the potato are constant.

The Biot number $Bi = \frac{hr_o}{k}$

$$Bi = \frac{(25\ \text{W/m}^2.°\text{C})(0.04\ \text{m})}{(0.6\ \text{W/m.°C})}$$

$$Bi = 1.67$$

The constants λ_1 and A_1 corresponding to this Biot number are from 11-2 tables

Using interpolation method to find

$$\lambda_1 = 1.8777 \quad \text{and} \quad A_1 = 1.4113$$

Step 3 of 6

Then the Fourier number and the time period become

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$\theta_{0,sph} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{70-170}{25-170} = 0.69$$

$$\frac{70-170}{25-170} = (1.4113)e^{-(1.8777)^2 \tau}$$

$$\tau = 0.203 > 0.2$$

The baking time of the potatoes is determined to be

$$t = \frac{\tau r_o^2}{\alpha}$$

$$t = \frac{(0.203)(0.04\ \text{m})^2}{(1.4\times10^{-7}\ \text{m}^2/s)}$$

$$t = 2320\ s$$

$$\boxed{t=38.7\ \text{min}}$$

Step 4 of 6

(*b*)**The maximum amount of heat transfer is**

The mass $m = \rho V$

$$m = \rho\frac{4}{3}\pi r_o^3$$

$$m = (1100\ \text{kg/m}^3)\left[\frac{4}{3}\pi(0.04\ \text{m})^3\cdot\right]$$

$$m = 0.295\ \text{kg}$$

$$Q_{max} = mC_p(T_\infty - T_i)$$

$$Q_{max} = (0.295\ \text{kg})(3.900\ \text{kJ/kg.°C})(170-25)°\text{C}$$

$$Q_{max} = 166.76\ \text{kJ}$$

Step 5 of 6

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{max}} = 1 - 3\theta_{0,sph}\frac{\sin(\lambda_1) - \lambda_1\cos(\lambda_1)}{\lambda_1^3}$$

$$\frac{Q}{Q_{max}} = 1 - 3(0.69)\frac{\sin(1.8777) - (1.8777)\cos(1.8777)}{(1.8777)^3}$$

$$\frac{Q}{Q_{max}} = 0.525$$

$$Q = 0.525Q_{max}$$

$$Q = (0.525)(166.76\ \text{kJ})$$

$$\boxed{Q=87.5\ \text{kJ}}$$

Step 6 of 6

The final equilibrium temperature of the potato after it is wrapped is

$$Q = mC_p(T_{eqv} - T_i)$$

$$T_{eqv} = T_i + \frac{Q}{mC_p}$$

$$T_{eqv} = 25°\text{C} + \frac{87.5\ \text{kJ}}{(0.295\ \text{kg})(3.9\ \text{kJ/kg.°C})}$$

$$\boxed{T_{eqv}=101°\text{C}}$$

Question: 18.057P

White potatoes (k = 0.50 W/m·K and α = 0.13×10-6 m2 /s) that are initially at a uniform temperature of 20°C and have an average diameter of 6 cm are to be cooled by refrigerated air at 2°C flowing at a velocity of 3 m/s. The average heat transfer coefficient between the potatoes and the air is experimentally determined to be 19 W/m2 · K. Determine how long it will take for the center temperature of the potatoes to drop to 6°C. Also, determine if any part of the potatoes will experience chilling injury during this process.

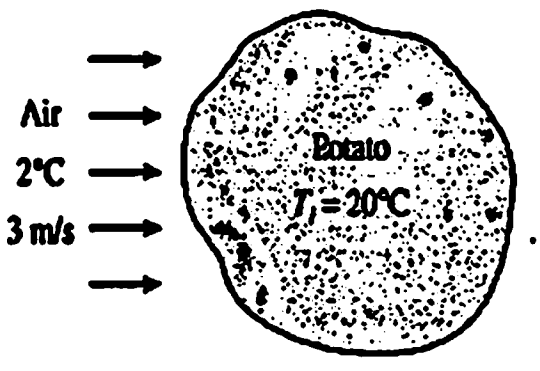


FIGURE P18-57

Step 1 of 4

Given data:

Thermal conductivity of potatoes, $k = 0.50\ \text{W/m}\cdot\text{K}$

Thermal diffusivity of potatoes, $\alpha = 0.13\times10^{-6}\ \text{m}^2/\text{s}$

Initial temperature of potatoes, $T_i = 20^\circ\text{C}$

Average heat transfer coefficient between the potato and air, $h = 19\ \text{W/m}^2\cdot\text{K}$

Center temperature of the potato, $T(t) = 6^\circ\text{C}$

Velocity of the air, $V = 3\ \text{m/s}$

Radius of the spherical potato, $r_o = 3\ \text{cm}$

Step 2 of 4

Heat conduction in the potato is one-dimensional in the radial direction because of the symmetry about the midpoint. The thermal properties of the potato are constant.
First we shall find the Biot number:

$$\text{Bi} = \frac{hr_0}{k}$$
$$= \frac{(19\ \text{W/m}^2.^\circ\text{C})(0.03\ \text{m})}{0.5\ \text{W/m.}^\circ\text{C}}$$
$$= \frac{0.57}{0.5}$$
$$= 1.14$$

Since Biot number is greater than 0.1, and thus lumped system analysis is not applicable. The coefficients λ_1 and A_1 for a cylinder corresponding to the Bi number are determined from the coefficient table used for in dimensional approximation of transient one dimensional heat conduction in sphere is $\lambda_1 = 1.635$ and $A_1 = 1.302$.

Step 3 of 4

The one dimensional approximation is

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}$$
$$\frac{6-2}{20-2} = 1.302e^{-1.635^2\tau}$$
$$\tau = 0.661$$

This is above the value of 0.2. Therefore, the one-term approximate solution must be used.

Then the length of time will take for center of the potato to reach 6°C is determined from the Fourier number to be

$$t = \frac{\tau r^2}{\alpha}$$
$$= \frac{0.661(0.03)^2}{0.13\times10^{-6}}$$
$$= 1.27\ \text{hour}$$

Therefore, the required time for cooling is $\boxed{1.27\ \text{hour}}$

Step 4 of 4

The lowest temperature during cooling will occur on the surface ($r/r_0 = 1$), and is determined to be

$$\frac{T(r)-T_\infty}{T_i-T_\infty} = A_1e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_0)}{\lambda_1 r/r_0}$$
$$\frac{T(r_0)-T_\infty}{T_i-T_\infty} = \theta_0\frac{\sin(\lambda_1 r_0/r_0)}{\lambda_1 r_0/r_0}$$
$$= \frac{T_0-T_\infty}{T_i-T_\infty}\frac{\sin(\lambda_1 r_0/r_0)}{\lambda_1 r_0/r_0}$$
$$\frac{T(r)-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}\frac{\sin(\lambda)}{\lambda}$$
$$\frac{T(r)-2}{20-2} = 1.302e^{-1.635^2\times0.661}\times\frac{\sin(1.635\text{rad})}{1.635}$$
$$T(r) = 6^\circ\text{C}$$

For potato chilling injury occurs at 6° C. As the obtained temperature is more than the threshold value, there will be no chilling injury.

Question: 18.058P

Layers of 23-cm-thick meat slabs (k = 0.47 W/m·K and $\alpha = 0.13 \times 10\text{-}6$ m2 /s) initially at a uniform temperature of 7° C are to be frozen by refrigerated air at -30°C flowing at a velocity of 1.4 m/s. The average heat transfer coefficient between the meat and the air is 20 W/m2 ·K. Assuming the size of the meat slabs to be large relative to their thickness, determine how long it will take for the center temperature ofthe slabs to drop to -18° C . Also, determine the surface temperature of the meat slab at that time.

Step 1 of 5

Consider the meat slabs as very large plane wall of half-thickness, L = 11.5cm.

Calculate the Biot number

$$\text{Bi} = \frac{hL}{k}$$

Here, Bi is the Biot number, h is the convection coefficient, k is the thermal conductivity, L is the half thickness of the slab.

Substitute $20\ \text{W/m}^2\text{.K}$ for h, $0.47\ \text{W/m.K}$ for k, and $0.115\ \text{m}$ for L.

$$\text{Bi} = \frac{(20\ \text{W/m}^2\text{.K})(0.115\ \text{m})}{0.47\ \text{W/m.K}}$$
$$= 4.89$$

Step 2 of 5

Consider the one term approximate of transient one dimensional heat conduction.

Determine the coefficient for one term approximate of transient one dimensional heat conduction for a plane wall with Biot number 4.89 using interpolation formula.

Linear interpolate the Coefficient values between 4 and 5 to get value at 4.89.

Calculate the coefficient, λ_1 .

$$\lambda_1 = 1{\cdot}2646 + [1{\cdot}3138 - 1{\cdot}2646]\left[\frac{4{\cdot}89 - 4}{5 - 4}\right]$$
$$\lambda_1 = 1.308388$$

Linear interpolate the Coefficient values between 4 and 5 to get value at 4.89.

Calculate the coefficient, A_1 .

$$A_1 = 1{\cdot}2287 + [1{\cdot}2403 - 1{\cdot}2287]\left[\frac{4{\cdot}89 - 4}{5 - 4}\right]$$
$$A_1 = 1.239024$$

Step 3 of 5

Calculate the Fourier number.

$$\frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Here, T_o is the temperature at the center of the slab after time t, T_i is the initial temperature of the slab, T_∞ is the ambient temperature, and τ is the Fourier number.

Substitute $-18°\text{C}$ for T_o , $-30°\text{C}$ for T_∞ , $7°\text{C}$ for T_i , 1.239024 for A_1 , and 1.308388 for λ_1 .

$$\frac{-18 - (-30)}{7 - (-30)} = 1.239024 e^{-(1.308388)^2 \tau}$$
$$\tau = 0.783$$

Since $\tau(0.783) > 0.2$ one term approximation method is valid.

Step 4 of 5

Calculate the time required to bring the temperature of the meat slab to $-18°\text{C}$.

$$\tau = \frac{\alpha t}{L^2}$$

Here, α is the thermal diffusivity of the meat.

Rearrange the formula to find time , t.

$$t = \frac{\tau L^2}{\alpha}$$

Substitute 0.783 for τ , $0.115\ \text{m}$ for L, and $0.13 \times 10^{-6}\ \text{m}^2/\text{s}$ for τ .

$$t = \frac{(0.783)(0.115\ \text{m})^2}{0.13 \times 10^{-6}\ \text{m}^2/\text{s}}$$
$$t = 796555.19\ \text{s}$$
$$t = 22.126\ \text{hrs}$$

Time taken to reach the center temperature $-18°\text{C}$ for the meat slab is $\boxed{22.126\ \text{h}}$.

Step 5 of 5

Calculate the surface temperature of the slab after time, $22.126\ \text{hrs}$.

Write the formula to find the temperature at a distance from center.

$$\frac{T(x) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \cos(\lambda_1 x / L)$$

Substitute L for x and $\left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)$ for $A_1 e^{-\lambda_1^2 \tau}$.

$$\frac{T(L) - T_\infty}{T_i - T_\infty} = \left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)\cos(\lambda_1 L / L)$$

$$(T(L) - T_\infty) = (T_0 - T_\infty)\cos(\lambda_1)$$
$$T(L) = T_\infty + (T_0 - T_\infty)\cos(\lambda_1)$$

Substitute $-18°\text{C}$ for T_o , $-30°\text{C}$ for T_∞ , and 1.308388 for λ_1 .

$$T(L) = -30 + (-18 - (-30))\cos(1.308388\text{rad}) \in$$
$$T(L) = -26.88°\text{C}$$

Thus, the temperature at the surface of the meat slab is $\boxed{-26.88°\text{C}}$.

Question: 18.059P

Layers of 6-in-thick meat slabs (k = 0.26 Btu/h·ft·° C and α = 1.4×10-6 ft2 /s) initially at a uniform temperature of 50° F arc cooled by refrigerated air at 23°F to a temperature of 36°F at their center in 12 h. Estimate the average heat transfer coefficient during this cooling process.

Step 1 of 4

Given:

Thickness of metal slab = 6in

Thermal conductivity of meat slabs are given to be $k = 0.26\ \text{Btu/h} \cdot \text{ft} \cdot {}^{\circ}\text{F}$

Thermal diffusivity of metal slab $\alpha = 1.4 \times 10^{-6}\ \text{ft}^2/\text{s}.$

Step 2 of 4

Center temperature of meat slabs is to be lowered to 36°F during 12-h of cooling.

The meat slabs can be approximated as very large plane walls of half-thickness L = 3-in.

Heat conduction in the meat slabs is one-dimensional because of symmetry about the centerplane.

Step 3 of 4

The Fourier number is τ > 0.2 so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

The average heat transfer coefficient during this cooling process is determined from the transient temperature charts for a flat plate as follows:

$$\tau = \frac{\alpha t}{L^2}$$

$$= \frac{(1.4\times10^{-6}\text{ft}^2/\text{s})(12\times3600\ \text{s})}{(3/12\ \text{ft})^2}$$

$$= 0.968$$

$$\frac{T_o - T_\infty}{T_i - T_\infty} = \frac{36-23}{50-23}$$

$$= 0.481$$

From the transient temperature charts for a flat plate above two values gives the

$$\frac{1}{Bi} = 0.7$$

Step 4 of 4

Therefore,

$$h = \frac{kBi}{L}$$

$$= \frac{(0.26\text{Btu/h.ft.}^{\circ}\text{F})(1/0.7)}{(3/12)\ \text{ft}}$$

$$= 1.5\ \text{Btu/h.ft}^2.^{\circ}\text{F}$$

Average heat transfer coefficient is $\boxed{1.5\ \text{Btu/h.ft}^2.^{\circ}\text{F}}$

Question: 18.060P

Oranges of 2.5-in-diameter (k = 0 26 Btu/h·ft·°F and α = 1.4 × 10-6 ft2 /s) initially at a uniform temperature of 78°F are to be cooled by refrigerated air at 25°F flowingat a velocity of 1ft/s. The average heat transfer coefficient between the oranges and the air is experimentally determined to be 4.6 Btu/h·ft2 ·°F. Determine how long it will take for the center temperature of the oranges to drop to 40°F. Also, determine if any pan of the oranges will freeze during this process.

Step 1 of 6

Given:

Diameter of orange $D = 2.5\ \text{in}$

Thermal conductivity of oranges $k = 0.26\ \text{Btu/h}\cdot\text{ft}\cdot{}^{\circ}\text{F}$

Thermal diffusivity of oranges $\alpha = 1.4\ 10^{-6}\ \text{ft}^2/\text{s}$

Initial temperature of orange $T_i = 78^0\text{F}$

Refrigerated air temperature $T_\infty = 25^0\text{F}$

Refrigerated air velocity $v = 1\text{ft/sec}$

Final temperature of orange $T_o = 40^0\text{F}$

Step 2 of 6

The oranges are spherical in shape with a radius of

$r_o = 1.25\ \text{in}$

$r_o = 0.1042\ \text{ft}$

The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

First we find the Biot number:

$$\text{Bi} = \frac{hr_0}{k}$$

$$= \frac{(4.6\ \text{Btu/h.ft}^2.^{\circ}\text{F})(1.25/12\ \text{ft})}{0.26\ \text{Btu/h.ft.}^{\circ}\text{C}}$$

$$= 1.843$$

Step 3 of 6

From the transient one dimensional heat conduction for a sphere,

$\lambda_1 = 1.9569$ And $A_1 = 1.447$

Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{40-25}{78-25} = 1.447 e^{-(1.9569)^2 \tau}$$

$\tau = 0.426$

Step 4 of 6

Which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_0^2}$$

$$t = \frac{\tau r_0^2}{\alpha}$$

$$= \frac{(0.426)(1.25/12\ \text{ft})^2}{1.4\times10^{-6}\ \text{ft}^2/\text{s}}$$

$$= 3302\ \text{s}$$

Time required dropping temperature is $\boxed{55.0\ \text{min}}$

Step 5 of 6

The lowest temperature during cooling will occur on the surface ($r/r_0 = 1$), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_0)}{\lambda_1 r / r_0}$$

$$\frac{T(r_0) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_0 / r_0)}{\lambda_1 r_0 / r_0}$$

$$= \frac{T_0 - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_0 / r_0)}{\lambda_1 r_0 / r_0}$$

Step 6 of 6

Substituting,

$$\frac{T(r_0) - 25}{78 - 25} = \left(\frac{40-25}{78-25}\right) \frac{\sin(1.9569\ \text{rad})}{1.9569}$$

$T(r_0) = 32.1^{\circ}\text{F}$

Which is above the freezing temperature of 31 °C for oranges. Therefore, no part of the oranges will freeze during this cooling process.

Question: 18.061P

A heated 6-mm-thick Pyroceram plate (ρ= 2600 kg/m3 , cp = 808 J/kg · K, k = 3.98 W/m2 · K, and α = 1.89 × 10-6 m2 /s) is being cooled in a room with air temperature of 25°C and convection heat transfer coefficient of 13.3 W/m2 ·K. The heated Pyroceram plate had an initial temperature of 500°C. and is allowed to cool for 286 seconds. If the mass of the Pyroceram plate is 10 kg, determine the heat transfer from the Pyroceram plate during the cooling process using (*a*) Table 18-2 and (*b*) Figure 18-15.

Step 1 of 6

Calculate the Biot number by using the relation,

$$\mathrm{Bi} = \frac{hL}{k}$$

Here, h is the convection heat transfer coefficient, and k is the thermal conductivity of the brass plate.

Substitute $13.3\ \mathrm{W/m^2 \cdot K}$ for h, $0.003\ \mathrm{m}$ for L . $3.98\ \mathrm{W/m \cdot K}$ k.

$$\mathrm{Bi} = \frac{(13.3\ \mathrm{W/m^2 \cdot K})(0.003\ \mathrm{m})}{3.98\ \mathrm{W/m \cdot K}}$$
$$= 0.01$$

Obtain the constant λ_1 and A_1 from table 4-2 at $\mathrm{Bi} = 0.01$ (For plane wall).

$$\lambda_1 = 0.0998$$

$$A_1 = 1.0017$$

Step 2 of 6

Calculate the Fourier number by using the relation,

$$\tau = \frac{\alpha t}{L^2}$$

Here, τ is the Fourier number, α is the thermal diffusivity of the material, t is the time, and L is the critical length of the plate.

Substitute $1.89\times10^{-6}\ \mathrm{m^2/s}$ for α , $286\ \mathrm{s}$ for t, and $0.003\ \mathrm{m}$ for L .

$$\tau = \frac{(1.89\times10^{-6}\ \mathrm{m^2/s})(286\ \mathrm{s})}{(0.003\ \mathrm{m})^2}$$
$$= 60.06$$

Step 3 of 6

Calculate the dimensionless temperature at the center of the plate by using the relation,

$$\theta_{0,\mathrm{wall}} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$
$$= A_1 e^{-\lambda_1^2 \tau}$$

Here, T_0 is the upper surface temperature of the plate, T_∞ is the temperature of the surrounding air, and T_i is the initial temperature of the plate.

Substitute 1.0017 for A_1 , 0.0998 for λ_1 , and 60.06 for τ .

$$\theta_{0,\mathrm{wall}} = 1.007 e^{-(0.0998)^2 60.06}$$
$$= 0.5507$$

Step 4 of 6

Calculate the maximum amount of heat transferred from the plate by using the relation,

$$Q_{\max} = mc_p(T_i - T_\infty)$$

Substitute 10 kg for m, $808\ \mathrm{J/kg \cdot K}$ for c_p , $(25+273)\mathrm{K}$ for T_∞ , and $(500+273)\mathrm{K}$ for T_i .

$$Q_{\max} = (10\ \mathrm{kg})(808\ \mathrm{J/kg \cdot K})((500+273)\mathrm{K} - (25+273)\mathrm{K})$$
$$= 3.838\times10^6\ \mathrm{J}$$

Step 5 of 6

Calculate the heat transfer from the Pyroceram plate during the cooling process by using the relation,

$$\left(\frac{Q}{Q_{\max}}\right)_{\mathrm{wall}} = 1 - \theta_{0,\mathrm{wall}} \frac{\sin(\lambda_1)}{\lambda_1}$$

Substitute $3.838\times10^6\ \mathrm{J}$ for $Q_{\max}$, 0.5507 for $\theta_{0,\mathrm{wall}}$, and 0.0998 for λ_1 .

$$\frac{Q}{3.838\times10^6\ \mathrm{J}} = 1 - (0.5507)\frac{\sin(0.0998^{\mathrm{rad}})}{0.0998}$$
$$= 1 - (0.5507)\frac{\sin\left(0.0998\times\frac{180^\circ}{\pi}\right)}{0.0998}$$
$$= 0.4502$$

$$Q = 0.4502(3.838\times10^6\ \mathrm{J})$$
$$= 1.728\times10^6\ \mathrm{J}$$

Therefore, the heat transferred from the Pyroceram plate during the cooling process is $\boxed{1.728\times10^6\ \mathrm{J}}$.

Step 6 of 6

b)
Calculate the value of the $\mathrm{Bi}^2\tau$.

Substitute 0.01 for Bi, and 60.06 for τ .

$$\mathrm{Bi}^2\tau = 0.01^2 \times 60.06$$
$$= 0.006$$

Obtain the ratio of heat transferred from the plate to the maximum amount of heat transferred from the plate $\left(\frac{Q}{Q_{\max}}\right)$ from figure 4-18 (*c*) corresponding to 0.01 of Bi and 0.006 of $\mathrm{Bi}^2\tau$.

$$\frac{Q}{Q_{\max}} = 0.45$$

Substitute $3.838\times10^6\ \mathrm{J}$ for $Q_{\max}$.

$$\frac{Q}{3.838\times10^6\ \mathrm{J}} = 0.45$$
$$Q = 1.7271\times10^6\ \mathrm{J}$$

Therefore, the heat transferred from the Pyroceram plate during the cooling process is $\boxed{1.7271\times10^6\ \mathrm{J}}$.

Question: 18.062P

A 10-cm thick aluminum plate ($\rho = 2702$ kg/m3 , $cp = 903$ J/kg · K, $k = 237$ W/m · K, and $\alpha = 97.1 \times 10\text{-}6$ m2 /s) is being heated in liquid with temperature of 500°C. The aluminum plate has a uniform initial temperature of 25°C. If the surface temperature of the aluminum plate is approximately the liquid temperature, determine the temperature at the center plane of the aluminum plate after 15 seconds of heating.

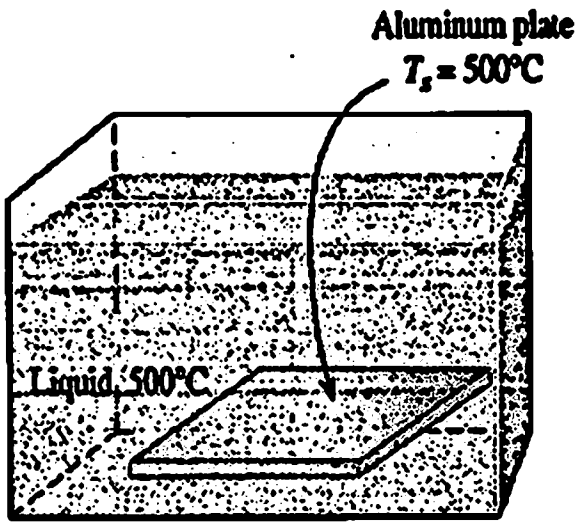


FIGURE P18-62

Step 1 of 3

As the surface temperature of the plate is equal to the liquid temperature, the convective heat transfer coefficient is infinitive.

$$h = \infty$$

Calculate the Biot number by using the relation,

$$\text{Bi} = \frac{hL}{k}$$

Here, L is the characteristic length, and k is the thermal conductivity of the plate material.

Substitute ∞ for h.

$$\text{Bi} = \frac{\infty L}{k}$$
$$= \infty$$

Obtain the constants for the plate from table 4-2 corresponding to the Biot number of ∞ for plane wall.

$$\lambda_1 = 1.5708$$

And, $A_1 = 1.2732$

Step 2 of 3

Calculate the Fourier number by using the relation,

$$\tau = \frac{\alpha t}{L^2}$$

Here, τ is the Fourier number, α is the thermal diffusivity of the material, t is the time, and L is the critical length of the plate.

Substitute $97.1\times10^{-6}\ \text{m}^2/\text{s}$ for α , $15\ \text{s}$ for t, and $0.05\ \text{m}$ for L .

$$\tau = \frac{97.1\times10^{-6}\ \text{m}^2/\text{s}(15\ \text{s})}{(0.05\ \text{m})^2}$$
$$= 0.5826 > 0.2$$

As the Fourier number is more than 0.2, the one term approximation method is used to solve the problem.

Step 3 of 3

Calculate the temperature of the plate at the center plane of the aluminum plate by using the relation,

$$\theta_{\text{o,sph}} = \left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)$$
$$= A_1 e^{-\lambda_1^2 \tau}$$

Further solve the equation as follows,

$$T_o - T_\infty = (T_i - T_\infty) A_1 e^{-\lambda_1^2 \tau}$$
$$T_o = T_\infty + (T_i - T_\infty) A_1 e^{-\lambda_1^2 \tau}$$

Here, T_o is the center plane temperature, T_∞ is the fluid temperature, and T_i is the initial temperature of the plate.

Substitute $25°C$ for T_i , $550°C$ for T_∞ , 1.2732 for A_1 , 1.5708 for λ_1 , and 0.5826 for τ .

$$T_o = 500°C + (25°C - 500°C)(1.2732) e^{-(1.5708)^2(0.5826)}$$
$$= 500°C + (25°C - 500°C)(1.2732) e^{-1.4375}$$
$$= 500°C + (25°C - 500°C)(1.2732) \times 0.2375$$
$$= 500°C - 143.64°C$$

$$T_o = 356.36°C$$

Therefore, the center plane temperature of the aluminum plate after 15 seconds of heating is **356.36°C** .

Question: 18.063P

A long iron rod (ρ = 7870 kg/m3 , cp = 447 J/kg · K, k = 80.2 W/m · K, and α = 23.1 × 10-6 m2 /s) with diameter of 25 mm is initially heated to a uniform temperature of 700°C. The iron rod is then quenched in a large water bath that is maintained at constant temperature of 50°C and convection heat transfer coefficient of 128 W/m2 ·K. Determine the time required for the iron rod surface temperature to cool to 200°C.

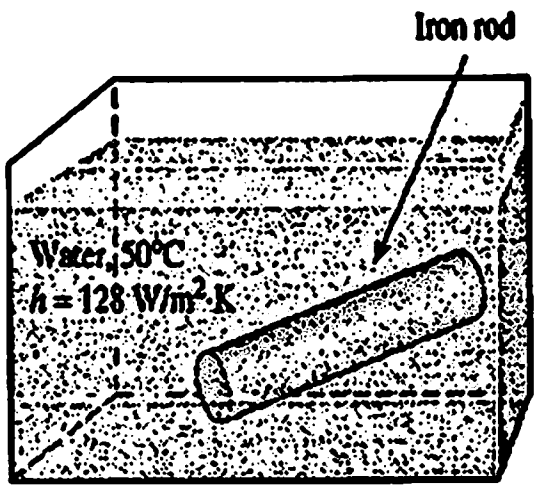


FIGURE P 18-63

Step 1 of 3

Determine the characteristic length of the long cylinder rod.

$$r_o = \frac{d}{2}$$

Here, diameter of the rod is d and characteristic length of the rod is r_o .

Substitute 25×10^{-3} m for d.

$$r_o = \frac{25\times10^{-3}}{2}$$
$$= 12.5\times10^{-3} \text{ m}$$

Calculate the Biot number of the rod and water system.

$$\text{Bi} = \frac{hr_o}{k}$$

Here, convective heat transfer coefficient is h , Biot number of the rod is Bi, and conduction coefficient of the rod is k .

Substitute $128 \text{ W/m}^2\cdot\text{K}$ for h, $80.2 \text{ W/m}\cdot\text{K}$ for k, and 12.5×10^{-3} m for r_o .

$$\text{Bi} = \frac{128\times12.5\times10^{-3}}{80.2}$$
$$= 0.02$$

Step 2 of 3

Consider the one term approximate of transient one dimensional heat conduction.

Take values of one term approximate coefficient for a cylinder from the table "Coefficients used in one term approximate solution of one dimensional transient heat conduction" recorded at Biot number 0.02.

$$\lambda_1 = 0.1995$$
$$A_1 = 1.005$$

Calculate the Fourier number.

$$\frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

Here, T_o is the temperature at the centre of the cylinder after time t, T_i is the initial temperature of the cylinder, T_∞ is the ambient temperature, and τ is the Fourier number.

Substitute $200°\text{C}$ for T_o , $50°\text{C}$ for T_∞ , $700°\text{C}$ for T_i , 1.005 for A_1 , and 0.1995 for λ_1 .

$$\frac{200-50}{700-50} = 1.005e^{-(0.1995)^2\tau}$$
$$\tau = 36.96$$

Since $\tau(36.96) > 0.2$ one term approximation method is valid.

Step 3 of 3

Calculate the time required to bring the temperature of the cylinder to $200°\text{C}$.

$$\tau = \frac{\alpha t}{L^2}$$

Here, α is the thermal diffusivity of the cylinder.

Rearrange the formula to find time , t.

$$t = \frac{\tau L^2}{\alpha}$$

Substitute 36.96 for τ , 12.5×10^{-3} m for r_o , and 23.1×10^{-6} m^2/s for τ .

$$t = \frac{(36.96)(12.5\times10^{-3}\text{ m})^2}{23.1\times10^{-6}\ \text{m}^2/\text{s}}$$
$$t = 250 \text{ s}$$

Thus, the time required to bring the temperature of the cylinder to $200°\text{C}$ is $\boxed{250 \text{ s}}$.

Question: 18.064P

A long Pyroceram rod (ρ = 2600 kg/m3 , cp = 808 J/kg · K, k = 3.98 W/m · K, and α = 1.89 × 10-6 m2 /s) with diameter of 10 mm has an initial uniform temperature of 1000°C. The Pyroceram rod is allowed to cool in ambient temperature of 25°C and convection heat transfer coefficient of 80 W/m2 ·K. If the Pyroceram rod is allowed to cool for 3 minutes, determine the temperature at the center of the rod using (a) Tabic 18-2 and (b) the Heisler chart (Figure 18-16).

Step 1 of 6

Calculate the Biot number for the heating process of pyroceram rod by using the relation,

$$\mathrm{Bi} = \frac{hL_c}{k} \quad \text{..... (1)}$$

Here, convective heat transfer coefficient is h , characteristic length of the rod is L_c , and conduction coefficient of the rod is k .

Calculate the characteristic length of the pyroceram rod by using the relation,

$$L_c = r_o$$

Here, radius of the rod is r_o .

Substitute $0.005\ \mathrm{m}$ for r_o .

$$L_c = 0.005\ \mathrm{m}$$

Step 2 of 6

Calculate the Biot number for this process as shown.

Substitute $0.005\ \mathrm{m}$ for L_c . $80\ \mathrm{W/m^2 \cdot K}$ for h , and $3.98\ \mathrm{W/m \cdot K}$ for k in equation (1).

$$\mathrm{Bi} = \frac{80 \times 0.005}{3.98}$$
$$= 0.10$$

Calculate the Fourier number for this process by using the formula,

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, diffusivity of pyroceram rod is α , and time for which the rod is heated is t .

Substitute $1.89 \times 10^{-6}\ \mathrm{m^2/s}$ for α . $(3 \times 60)\ \mathrm{s}$ for t , and $0.005\ \mathrm{m}$ for r_o .

$$\tau = \frac{1.89 \times 10^{-6} \times 3 \times 60}{(0.005)^2}$$
$$= 13.61$$

Step 3 of 6

(a).

Calculate the dimensionless temperature of the cylindrical rod by using the relation,

$$\theta_{0,cyl} = A_1 e^{-\lambda_1^2 \tau} \quad \text{..... (2)}$$

Obtain the values of λ_1 and A_1 for $\mathrm{Bi} = 0.10$ from table 4.2 as shown.

$$\lambda_1 = 0.4417$$
$$A_1 = 1.0246$$

Substitute 13.61 for τ . 1.0246 for A_1 , and 0.4417 for λ_1 in equation (2).

$$\theta_{0,cyl} = (1.0246) e^{-\left[(0.4417)^2 \times 13.61\right]}$$
$$= 0.072$$

Step 4 of 6

Calculate the temperature at the centre of the rod by using the relation,

$$\frac{T_0 - T_\infty}{T_i - T_\infty} = \theta_{0,cyl}$$

Here, the temperature at the centre of the rod is T_0 , surroundings temperature is T_∞ , and initial temperature of the rod is T_i .

Substitute 0.072 for $\theta_{0,cyl}$. $25\ ^\circ\mathrm{C}$ for T_∞ , and $1000\ ^\circ\mathrm{C}$ for T_i .

$$\frac{T_0 - 25}{1000 - 25} = 0.072$$
$$T_0 - 25 = 975 \times 0.072$$
$$T_0 = 25 + 70.2$$
$$= 95.2\ ^\circ\mathrm{C}$$

Therefore, the temperature at the centre of the rod is $\boxed{95.2\ ^\circ\mathrm{C}}$.

Step 5 of 6

(b).

Calculate the value $\frac{1}{\mathrm{Bi}}$ as shown.

$$\frac{1}{\mathrm{Bi}} = \frac{1}{0.10}$$
$$= 10$$

Obtain the value of dimensionless temperature at the centre, θ_0 corresponding to values $\frac{1}{\mathrm{Bi}} = 10$ and $\tau = 13.61$ from figure 4-17 as shown.

$$\theta_0 = 0.075$$

Step 6 of 6

Calculate the temperature at the centre of the rod as shown.

$$\frac{T_0 - T_\infty}{T_i - T_\infty} = \theta_{0,cyl}$$

Substitute 0.075 for $\theta_{0,cyl}$. $25\ ^\circ\mathrm{C}$ for T_∞ , and $1000\ ^\circ\mathrm{C}$ for T_i .

$$\frac{T_0 - 25}{1000 - 25} = 0.075$$
$$T_0 - 25 = 975 \times 0.075$$
$$T_0 = 25 + 73.125$$
$$= 98.125\ ^\circ\mathrm{C}$$

Therefore, the temperature at the centre of the rod is $\boxed{98.125\ ^\circ\mathrm{C}}$.

Question: 18.065P

A 2-cm-diameter plastic rod has a thermocouple inserted to measure temperature at the center of the rod. The plastic (ρ = 1190 kg/m3 , cp = 1465 J/kg · K, k = 0.19 W/m · K) was initially heated to a uniform temperature of 70°C, and allowed to be cooled in ambient air temperature of 25°C. After 1388 s of cooling, the thermocouple measured the temperature at the center of the rod to be 30°C. Determine the convection heat transfer coefficient for this process.

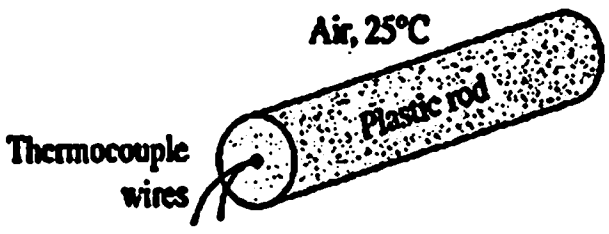


FIGURE P 18-65

Step 1 of 4

Diameter of the rod is, $D=2$ cm

Density is, $\rho=1190$ kg/m^3

Specific heat, $c_p=1465$ kJ/kg·K

Center temperature, $T=30$ °C

Initially temperature is, $T_i=70$ °C

Ambient temperature is, $T_\infty=25$ °C

Time taken for cooling, $t=1388$s

Step 2 of 4

Calculation of the convective heat transfer coefficient:

In order to calculate the convective heat transfer coefficient, let us calculate Biot number, Proper choice of characteristic length is important for this problem. At outer surface, $r=r_o$

Since $r=1$cm, $\frac{r}{r_o}=1$ and non-dimensional temperature as:

$$\frac{T-T_\infty}{T_i-T_\infty}=\frac{30-25}{70-25}$$

$$\frac{T-T_\infty}{T_i-T_\infty}=0.111$$

Step 3 of 4

And we know that,

$$\tau=\frac{\alpha t}{L^2} \qquad \text{..... (1)}$$

Where,

$$\alpha=\frac{k}{\rho c_p}$$

$$=\frac{(0.19)}{(1190\times1465)}$$

$$=1.08\times10^{-7}\ \text{m}^2/\text{s}$$

By substituting the above value in equation (1), we get,

$$\tau=\frac{\alpha t}{r^2}$$

$$=\frac{\left(1.08\times10^{-7}\right)(1388)}{(0.01)^2}$$

$$=1.51$$

From the transient temperature and heat transfer charts for long cylinder,

At $\frac{T-T_\alpha}{T_i-T_\alpha}=0.11$ and $\frac{r}{r_o}=1$, we have,

$$\frac{1}{\text{Bi}}=0.18$$

Step 4 of 4

We know that,

$$\text{Bi}=\frac{hr_o}{k}$$

$$\frac{1}{\text{Bi}}=\frac{k}{hr_o}$$

$$0.18=\frac{0.19}{h\times0.01}$$

$$\boxed{h=105.55\ \text{W/m}^2\cdot\text{K}}$$

Therefore, the convective heat transfer coefficient is obtained as, $\boxed{h=105.55\ \text{W/m}^2\cdot\text{K}}$

Question: 18.066P

Steel rods, 2 m in length and 60 mm in diameter, are being drawn through an oven that maintains a temperature of 800°C and convection heat transfer coefficient of 128 W/m2 ·K. The steel rods (ρ = 7832 kg/m3 , *cp* = 434 J/kg · K, *k* = 63.9 W/m · K, and α = 1.88 × 10-6 m2 /s) were initially in uniform temperature of 30°C. Using (*a*) Table 18-2 and (*b*) Figure 18-16, determine the amount of heat is transferred to the steel rod after 133 s of heating.

Step 1 of 5

Calculate the maximum heat transferred to the steel rod by using the relation,

..... (1)

Here, density of steel rod is , volume of the steel rod is , and specific heat at constant pressure is C_p , surrounding temperature is T_∞ , and initial temperature of the rod is T_i .

Calculate the volume of the steel rod by using the formula,

$$V = \pi r_o^2 L$$

Here, radius of the rod is r_o , and length of the rod is L .

Calculate the maximum heat transferred to the steel rod as shown.

Substitute $\pi r_o^2 L$ for V in equation (1).

$$Q_{max} = \rho \times \pi r_o^2 L \times C_p (T_\infty - T_i)$$

Substitute 2 m for L , 0.03 m for r_o , 7832 kg/m^3 for ρ , $434\text{ J/kg}\cdot\text{K}$ for C_p , 800 °C for T_∞ , and 30 °C for T_i .

$$\begin{aligned} Q_{max} &= 7832 \times \pi \times (0.03)^2 \times 2 \times 434 \times (800 - 30) \\ &= 1.48 \times 10^7\text{ J} \end{aligned}$$

Step 2 of 5

Calculate the Biot number of the steel rod by using the relation,

$$\text{Bi} = \frac{hL_c}{k} \quad \text{..... (2)}$$

Here, convective heat transfer coefficient is h , characteristic length of the rod is L_c , and conduction coefficient of the rod is k .

Calculate the characteristic length of the steel rod by using the relation,

$$L_c = r_o$$

Here, radius of the rod is r_o .

Substitute 0.03 m for r_o .

$$L_c = 0.03\text{ m}$$

Step 3 of 5

Calculate the Biot number for this process as shown.

Substitute 0.03 m for L_c , $128\text{ W/m}^2\cdot\text{K}$ for h , and $63.9\text{ W/m}\cdot\text{K}$ for k in equation (2).

$$\begin{aligned} \text{Bi} &= \frac{128 \times 0.03}{63.9} \\ &= 0.06 \end{aligned}$$

Calculate the Fourier number for this process by using the formula,

$$\tau = \frac{\alpha t}{r_o^2}$$

Here, diffusivity of steel rod is α , and time for which the rod is heated is t .

Substitute $18.8 \times 10^{-6}\text{ m}^2/\text{s}$ for α , 133 s for t , and 0.03 m for r_o .

$$\begin{aligned} \tau &= \frac{18.8 \times 10^{-6} \times 133}{(0.03)^2} \\ &= 2.778 \end{aligned}$$

Step 4 of 5

(a).

Calculate the dimensionless temperature of the cylindrical rod by using the relation,

$$\theta_{0,cyl} = A_1 e^{-\lambda_1^2 \tau} \quad \text{..... (3)}$$

Obtain the values of λ_1 and A_1 for $\text{Bi} = 0.06$ from table 4.2 as shown.

$$\lambda_1 = 0.3438$$
$$A_1 = 1.0148$$

Substitute 2.778 for τ , 1.0148 for A_1 , and 0.3438 for λ_1 in equation (3).

$$\begin{aligned} \theta_{0,cyl} &= (1.0148) e^{-\left[(0.3438)^2 \times 2.778\right]} \\ &= 0.7308 \end{aligned}$$

Obtain the value of $J(0.3438)$ from table 4.3 as shown.

$$J(0.3438) = 0.1694$$

Step 5 of 5

Calculate the heat transfer to a steel rod after 133 s by using the relation,

$$Q = Q_{max}\left[1 - 2\theta_{0,cyl}\frac{J_1(\lambda_1)}{\lambda_1}\right]$$

Substitute 0.1694 for $J(0.3438)$, 0.3438 for λ_1 , 0.7308 for $2\theta_{0,cyl}$, and $1.48 \times 10^7\text{ J}$ for Q_{max} .

$$\begin{aligned} Q &= 1.48 \times 10^7\left[1 - 2(0.7308)\left(\frac{0.1694}{0.3438}\right)\right] \\ &= 1.48 \times 10^7\left[1 - 0.72\right] \\ &= 4.14 \times 10^6\text{ J} \\ &= 4140\text{ kJ} \end{aligned}$$

Therefore, the heat transfer to a steel rod after 133 s is $\boxed{4140\text{ kJ}}$.

(b).

Calculate the value $\text{Bi}^2\tau$ as shown.

$$\begin{aligned} \text{Bi}^2\tau &= (0.06)^2 \times 2.778 \\ &= 0.0036 \times 2.778 \\ &= 0.01 \end{aligned}$$

Obtain the value of $\frac{Q}{Q_{max}}$ corresponding to value $\text{Bi}^2\tau = 0.01$ from figure 4-17 as shown.

$$\frac{Q}{Q_{max}} = 0.3 \quad \text{..... (4)}$$

Calculate the heat transfer to a steel rod after 133 s as shown.

Rearrange equation (4) as shown.

$$Q = 0.3Q_{max}$$

Substitute $1.48 \times 10^7\text{ J}$ for Q_{max} .

$$\begin{aligned} Q &= 0.3 \times 1.48 \times 10^7 \\ &= 4.44 \times 10^6\text{ J} \\ &= 4440\text{ kJ} \end{aligned}$$

Therefore, the heat transfer to a steel rod after 133 s is $\boxed{4440\text{ kJ}}$.

Question: 18.067P

What is a semi-infinite medium? Give examples of solid bodies that can be treated as semi-infinite mediums for heat transfer purposes.

Step 1 of 1

A semi-infinite medium is an idealized body which has a single exposed plane surface and extends to infinity in all directions. The earth and thick walls can be considered to be semi-infinite media.

Question: 18.068P

Under what conditions can a plane wall be treated as a semi-infinite medium?

Step 1 of 1

A thick plane wall can be treated as a semi-infinite medium if all we are interested in is the variation of temperature in a region near one of the surfaces for a time period during which the temperature in the mid section of the wall does not experience any change.

Question: 18.069P

Consider a hot semi-infinite solid at an initial temperature of Ti that is exposed to convection to a cooler medium at a constant temperature of $T\infty$ with a heat transfer coefficient of h. Explain how you can determine the total amount of heat transfer from the solid up to a specified time t 0

Step 1 of 1

The total amount of heat transfer from a semi-infinite solid up to a specified time t_0 can be determined by integration from

$$Q = \int_0^{t_0} Ah[T(0,t) - T_\infty]dt$$

Where the surface temperature T(0,t) is obtained from above equation by substituting $x = 0$.

Question: 18.070P

A thick wood slab (k = 0.17 W/m·K and α = 1.28 × 10-7 m2 /s) that is initially at a uniform temperature of 25°C is exposed to hot gases at 550°C for a period of 5 min. The heat transfer coefficient between the gases and the wood slab is 35 W/m2 ·K. If the ignition temperature of the wood is 450°C, determine if the wood will ignite.

Step 1 of 4

Given:

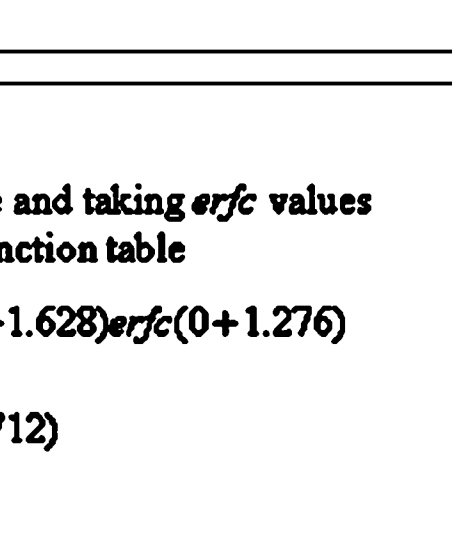
Initial temperature of wood slab $T_i = 25°C$

Temperature of hot gases $T_\infty = 550°C$

Thermal conductivity of the wood slab $k = 0.17 \dfrac{W}{m.°C}$

Thermal diffusivity of the wood slab $\alpha = 1.28 \times 10^{-7} \dfrac{m^2}{S}$

Step 2 of 4



Step 3 of 4

The wood slab is treated as a semi-infinite medium subjected to convection at the exposed surface. The thermal properties of the wood slab are constant. The heat transfer coefficient is constant and uniform over the entire surface.

The one-dimensional transient temperature distribution in the wood can be determined from

$$\frac{T(x,t)-T_i}{T_\infty-T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k}+\frac{h^2\alpha t}{k^2}\right)\left[erfc\left(\frac{x}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)\right]$$

Where

$$\frac{h\sqrt{\alpha t}}{k} = \frac{(35\ W/m^2.°C)\sqrt{(1.28\times10^{-7}\ m^2/s)(5\times60\ s)}}{0.17\ W/m.°C}$$

$$\frac{h\sqrt{\alpha t}}{k} = 1.276$$

$$\frac{h^2\alpha t}{k^2} = \left(\frac{h\sqrt{\alpha t}}{k}\right)^2$$

$$\frac{h^2\alpha t}{k^2} = 1.276^2$$

$$\frac{h^2\alpha t}{k^2} = 1.628$$

Step 4 of 4

Noting that $x = 0$ at the surface and taking $erfc$ values

From complementary error function table

$$\frac{T(x,t)-25}{550-25} = erfc(0) - \exp(0+1.628)erfc(0+1.276)$$

$$\frac{T(x,t)-25}{550-25} = 1-(5.0937)(0.0712)$$

$$\frac{T(x,t)-25}{550-25} = 0.630$$

Solving for $T(x, t)$ gives

$\boxed{T(x,t)=360°C}$

Which is less than the ignition temperature of 450°C. Therefore, the wood will not ignite.

Question: 18.071P

The soil temperature in the upper layers of the earth varies with the variations in the atmospheric conditions. Before a cold front moves in, the earth at a location is initially at a uniform temperature of 15°C. Then the area is subjected to a temperature of -8°C and high winds that resulted in a convection heat transfer coefficient of 40 W/m2 ·K on the earth's surface for a period of 10 h. Taking the properties of the soil at that location to be k = 0.9 W/m·K and α = 1.6 × 10- 5 m2 /s, determine the soil temperature at distances 0, 10, 20, and 50 cm from the earth's surface at the end of this 10-h period.

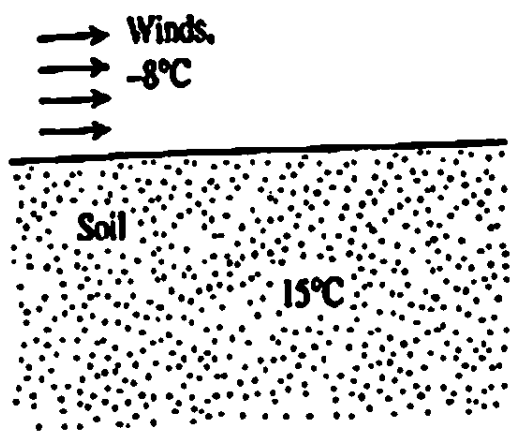


FIGURE P 18-71

Step 1 of 7

Given data:

Temperature of earth, $T_\infty = 15°C$

Temperature of operating area, $T_i = -8°C$

Convection heat transfer coefficient, $h = 40\ W/m^2 \cdot K$

Thermal conductivities of the soil, $k = 0.9\ W/m \cdot K$

Thermal diffusivity, $\alpha = 1.6 \times 10^{-5}\ m^2/s$

Step 2 of 7

An area is subjected to cold air for a 10-h period. The soil temperatures at distances 0, 10, 20, and 50 cm from the earth's surface are to be determined. The one-dimensional transient temperature distribution in the ground can be determined from

$$\frac{T(x,t)-T_i}{T_\infty - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k}+\frac{h^2\alpha t}{k^2}\right)\left[erfc\left(\frac{x}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)\right]$$

Where

$$\frac{h\sqrt{\alpha t}}{k} = \frac{(40\ W/m^2.°C)\sqrt{(1.6\times10^{-5}\ m^2/s)(10\times3600\ s)}}{0.9\ W/m.°C}$$

$$\frac{h^2\alpha t}{k^2} = \left(\frac{h\sqrt{\alpha t}}{k}\right)^2$$

$$\frac{h^2\alpha t}{k^2} = 33.7^2$$

$$\frac{h^2\alpha t}{k^2} = 1138$$

Step 3 of 7

Then we conclude that the last term in the temperature distribution relation above must be zero regardless of x despite the exponential term tending to infinity since $erfc(\xi) \to 0$ for $\xi > 4$ and the term has to remain less than 1 to have physically meaningful solutions. That is,

$$= \exp\left(\frac{hx}{k}+\frac{h^2\alpha t}{k^2}\right)\left[erfc\left(\frac{x}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)\right]$$

$$= \exp\left(\frac{hx}{k}+1138\right)\left[erfc\left(\frac{x}{2\sqrt{\alpha t}}+33.3\right)\right]$$

$$\cong 0$$

Therefore, the temperature distribution relation simplifies to

$$\frac{T(x,t)-T_i}{T_\infty - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Step 4 of 7

Then the temperature of the soil at distance of 0m is

At $x = 0$

$$\frac{T(x,t)-T_i}{T_\alpha - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{T(x,t)-15}{-8-15} = erfc(0)$$

From the complementary error function table

At $erfc(0)$ is 1

Then,

$$\frac{T(x,t)-15}{-8-15} = erfc(0)$$

$$\frac{T(x,t)-15}{-8-15} = 1$$

$$T(x,t) = -8°C$$

Therefore, the temperature of the soil at distance of 0 m is $\boxed{T(x,t) = -8°C}$

Step 5 of 7

The temperature of the soil at distance of x=10m is

At $x = 10\ cm$

$$\frac{T(x,t)-T_i}{T_\alpha - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{T(x,t)-15}{-8-15} = erfc\left(\frac{0.1}{2\sqrt{1.6\times10^{-5}\times3600\times10}}\right)$$

From the complementary error function table

At $erfc(0.066)$ is 0.9257

$$\frac{T(x,t)-15}{-8-15} = 0.9257$$

$$T(x,t) = -6.3°C$$

Therefore, the temperature of the soil at distance of 10m is $\boxed{T(x,t) = -6.3°C}$

Step 6 of 7

The temperature of the soil at distance of $x = 20\ cm$ is

At $x = 20\ cm$

$$\frac{T(x,t)-T_i}{T_\alpha - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{T(x,t)-15}{-8-15} = erfc\left(\frac{0.2}{2\sqrt{1.6\times10^{-5}\times3600\times10}}\right)$$

From the complementary error functions table

At $erfc(0.131)$ is 0.8519

$$\frac{T(x,t)-15}{-8-15} = 0.8519$$

$$T(x,t) = -4.6°C$$

Therefore, the temperature of the soil at distance of 10m is $\boxed{T(x,t) = -4.6°C}$

Step 7 of 7

The temperature of the soil at distance of x=20m is

At $x = 50\ cm$

$$\frac{T(x,t)-T_i}{T_\alpha - T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{T(x,t)-15}{-8-15} = erfc\left(\frac{0.5}{2\sqrt{1.6\times10^{-5}\times3600\times10}}\right)$$

From the complementary error functions table

At $erfc(0.329)$ is 0.6418

$$\frac{T(x,t)-15}{-8-15} = 0.6418$$

$$T(x,t) = 0.24°C$$

Therefore, the temperature of the soil at distance of 10m is $\boxed{T(x,t) = 0.24°C}$

Question: 18.072P

Reconsider Prob. 18-71. Using EES (or other) software, plot the soil temperature as a function of the distance from the earth's surface as the distance varies from 0 m to 1m, and discuss the results.

Step 1 of 4

Given data:

Temperature of earth $T_\infty = 15°C$.

Temperature of operating area $T_i = -8°C$.

Convection heat transfer coefficient $h = 40\ W/m^2 \cdot K$.

Thermal conductivities of the soil $k = 0.9\ W/m \cdot K$.

Thermal diffusivity $\alpha = 1.6 \times 10^{-5}\ m^2/s$.

Step 2 of 4

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
T_i=15 "[C]"
T_infinity=-8 "[C]"
h=40 "[W/m^2-C]"
time=10*3600 "[s]"
"x=0.1 [m], parameter to be varied"
"PROPERTIES"
k=0.9 "[W/m-C]"
alpha=1.6E-5 "[m^2/s]"
"ANALYSIS"
(T_x-T_i)/(T_infinity-T_i)=erfc(x/(2*sqrt(alpha*time)))-
exp((h*x)/k+(h^2*alpha*time)/k^2)*erfc(x/(2*sqrt(alpha*time))+(h*sqrt(alpha*time)/
k))
```

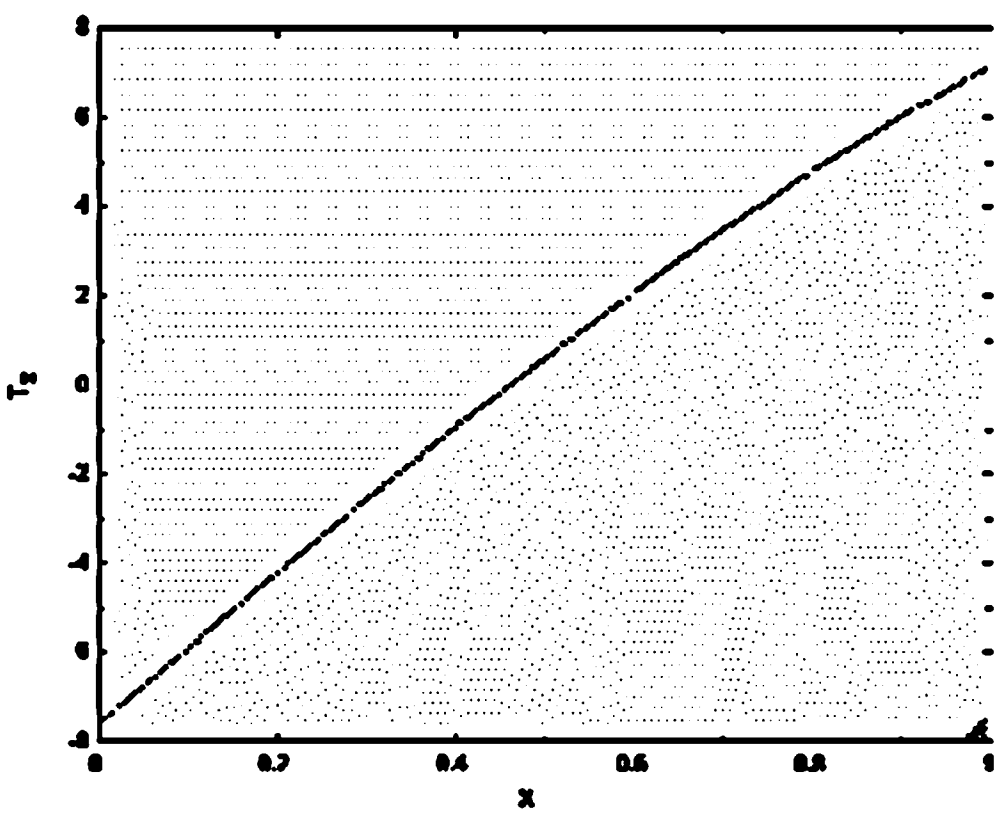
Step 3 of 4

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 21
Select variables in equations as x and T_x press ADD and then press ok
A table will appear put values of x as 0,0.05,0.1....1 in run 1 to 21
Then from Calculate -> Solve table
Solution will appear

1_21	x	T_x
Run 1	0	-7.611
Run 2	0.05	-6.762
Run 3	0.1	-5.911
Run 4	0.15	-5.064
Run 5	0.2	-4.224
Run 6	0.25	-3.391
Run 7	0.3	-2.569
Run 8	0.35	-1.758
Run 9	0.4	-0.96
Run 10	0.45	-0.1764
Run 11	0.5	0.5912
Run 12	0.55	1.342
Run 13	0.6	2.074
Run 14	0.65	2.786
Run 15	0.7	3.478
Run 16	0.75	4.149
Run 17	0.8	4.797
Run 18	0.85	5.423
Run 19	0.9	6.026

Step 4 of 4

Plot for Variation of Soil Temperature with Distance from Earth's Surface:



Question: 18.073P

A thick aluminum block initially at 20°C is subjected to constant heat flux of 4000 W/m2 by an electric resistance heater whose top surface is insulated. Determine how much the surface temperature of the block will rise after 30 minutes.

Step 1 of 2

Given:

Temperature of aluminum block $T_i = 20°C$

Heat flux $\dot{q}_s = 4000 \ \frac{W}{m^2}$

Time $t = 30 \text{ min}$

Step 2 of 2

This is conduction in transient mode where constant surface heat flux is applied to semi infinite medium, so surface temperature can be calculated as

$$T_s = T_i + \frac{\dot{q}_s}{k}\sqrt{\frac{4\alpha t}{\pi}}$$

$$T_s = 20°C + \frac{4000 W/m^2}{237 W/m.°C}\sqrt{\frac{4(9.71\times10^{-5} m^2/s)(30\times60s)}{\pi}}$$

$$T_s = 28.0°C$$

Rise in temperature for the surface will be

$$\Delta T_s = 28 - 20$$

$$\boxed{\Delta T_s = 8°C}$$

Question: 18.074P

A bare-footed person whose feet are at 32°C steps on a large aluminum block at 20°C. Treating both the feet and the aluminum block as semi-infinite solids, determine the contact surface temperature. What would your answer be if the person stepped on a wood block instead? At room temperature, the $\sqrt{k\rho c_p}$ value is 218 kJ/m2 ·°C for aluminum, 0.38 kJ/m2 ·°C for wood, and 1.1 kJ/m2 ·°C for human flesh.

Step 1 of 3

Given:

Temperature of feet at $T = 32°C$

Aluminium block at temperature $T_{Al} = 20°C$

The value of $\sqrt{(k\rho c_p)}$ for human flesh is $1.1 kJ/m^2 \cdot °C$

The value of $\sqrt{(k\rho c_p)}$ for Aluminium is $218 kJ/m^2 \cdot °C$

The value of $\sqrt{(k\rho c_p)}$ for wood is $0.38 kJ/m^2 \cdot °C$

Step 2 of 3

Contact of two semi infinite solids of different initial temperatures, the contact surface temperature can be determined as

$$T_s = \frac{\sqrt{(k\rho c_p)_A}T_{Ai} + \sqrt{(k\rho c_p)_B}T_{Bi}}{\sqrt{(k\rho c_p)_A} + \sqrt{(k\rho c_p)_B}}$$

Consider the surface A is human feet and surface B is Aluminium block and substituting the known values in the above equation,

$$T_s = \frac{1.1\times(32+273)+218\times(20+273)}{1.1+218}$$

$$= 293.06 K$$

$$= 20.06°C$$

Therefore, contact surface temperature, T_s between human feet and aluminium block is

$\boxed{20.06°C}$

Step 3 of 3

Consider the surface A is human feet and surface B is wood and substituting the known values in the surface temperature.

$$T_s = \frac{\sqrt{(k\rho c_p)_A}T_{Ai} + \sqrt{(k\rho c_p)_B}T_{Bi}}{\sqrt{(k\rho c_p)_A} + \sqrt{(k\rho c_p)_B}}$$

$$= \frac{1.1\times(32+273)+0.38\times(20+273)}{1.1+0.38}$$

Step 2 of 3

Contact of two semi infinite solids of different initial temperatures, the contact surface temperature can be determined as

$$T_s = \frac{\sqrt{(k\rho c_p)_A}T_{Ai} + \sqrt{(k\rho c_p)_B}T_{Bi}}{\sqrt{(k\rho c_p)_A} + \sqrt{(k\rho c_p)_B}}$$

Consider the surface A is human feet and surface B is Aluminium block and substituting the known values in the above equation,

$$T_s = \frac{1.1\times(32+273)+218\times(20+273)}{1.1+218}$$

Question: 18.075P

The walls of a furnace are made of 1.5-ft-thick concrete (k = 0.64 Btu/h·ft·°F and α = 0.023 ft2 /h). Initially, the furnace and the surrounding air are in thermal equilibrium at 70°F. The furnace is then fired, and the inner surfaces of the furnace are subjected to hot gases at 1800°F with a very large heat transfer coefficient. Determine how long it will take for the temperature of the outer surface of the furnace walls to rise to 70.1 °F.

Step 1 of 3

Given data:

Thickness of the wall, $x = 1.5$ ft.

Thermal conductivity of the concrete, $k = 0.64\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$

Thermal diffusivity, $\alpha = 0.023\ \text{ft}^2/\text{h}$

Temperature of outer furnace wall and air $T_i = 70\,^\circ\text{F}$

Inner temperature of furnace, $T_s = 1800\,^\circ\text{F}$

Final temperature of outer furnace temperature $T(x,t) = 70.1\,^\circ\text{F}$

Step 2 of 3

The temperature in the wall is affected by the thermal conditions at inner surfaces only and the convection heat transfer coefficient inside is given to be very large. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature of 1800°F.

The one-dimensional transient temperature distribution in the wall for that time period can be determined from

$$\frac{T(x,t)-T_i}{T_s-T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

By substituting the known values in the above equation, we get:

$$\frac{T(x,t)-T_i}{T_\alpha-T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{70.1-70}{1800-70} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$5.78\times10^{-5} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Step 3 of 3

From the complementary error function table,

By interpolating the values at $\text{erfc}(0.00008)$ and $\text{erfc}(0.00004)$, we get the error function at 5.78×10^{-5} as,

$$\text{erfc}(2.85) = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{x}{2\sqrt{\alpha t}} = 2.85$$

$$1.5 = 2.85\times2\times\sqrt{0.023t}$$

$$\sqrt{0.023t} = 0.26315$$

$$0.023t = 0.0692$$

$$t = 3\ \text{hour}$$

Therefore, the required time to reach the temperature, $\boxed{t = 3\ \text{hour}}$

$$\frac{x}{2\sqrt{\alpha t}} = 2.85$$

$$1.5 = 2.85\times2\times\sqrt{0.023t}$$

$$\sqrt{0.023t} = 0.26315$$

$$0.023t = 0.0692$$

$$t = 3\ \text{hour}$$

Therefore, the required time to reach the temperature, $\boxed{t = 3\ \text{hour}}$

Question: 18.076P

In areas where the air temperature remains below 0°C for prolonged periods of time, the freezing of water in underground pipes is a major concern. Fortunately, the soil remains relatively warm during those periods, and it takes weeks for the subfreezing temperatures to reach the water mains in the ground. Thus, the soil effectively serves as an insulation to protect the water from the freezing atmospheric temperatures in winter.

The ground at a particular location is covered with snow pack at -8°C for a continuous period of 60 days, and the average soil properties at that location are k = 0.35 W/m·K and α = 0.15 × 10-6 m2 /s. Assuming an initial uniform temperature of 8°C for the ground, determine the minimum burial depth to prevent the water pipes from freezing.

Step 1 of 3

The length of time the snow pack stays on the ground t is 60 days. Convert the length of time from days to seconds by using the conversion formula.

$$t = (60 \text{ days})(24 \text{ hr/days})(3600 \text{ s/hr})$$
$$= 5.184 \times 10^6 \text{ s}$$

Find the depth at which freezing occurs at 0C by using analytical solution,

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Step 1 of 3

The length of time the snow pack stays on the ground t is 60 days. Convert the length of time from days to seconds by using the conversion formula.

$$t = (60 \text{ days})(24 \text{ hr/days})(3600 \text{ s/hr})$$
$$= 5.184 \times 10^6 \text{ s}$$

Find the depth at which freezing occurs at 0C by using analytical solution,

0.02

$$\eta = 0.477$$

$$\frac{x}{1.7636} = 0.477$$

Step 3 of 3

The complementary error function of $\left(\frac{x}{1.7636}\right)$ is 0.477. Use the value to find x.

$$\frac{x}{1.7636} = 0.477$$
$$x = 0.8412 \text{ m}$$

Therefore, the minimum burial depth x required to prevent the water pipes from freezing is, **0.8412 m** .

Question: 18.077P

A highway made of asphalt is initially at a uniform temperature of 55°C. Suddenly the highway surfacetemperature is reduced to 25°C by rain. Determine the temperature at the depth of 3 cm from the highway surface andthe heat flux transferred from the highway after 60 minutes. Assume the highway surface temperature is maintained at 25°C.

Step 1 of 3

Obtain the values for properties of asphalt from Appendix table A-8 as shown.

$$k = 0.062 \text{ W/m} \cdot \text{K}$$
$$\rho = 2115 \text{ kg/m}^3$$
$$C_p = 920 \text{ J/kg} \cdot \text{K}$$

Calculate the thermal diffusivity of asphalt by using the relation,

$$\alpha = \frac{k}{\rho C_p}$$

Here, thermal conductivity of the asphalt material is k, density of the asphalt material is ρ, and specific heat of the material is C_p.

Substitute $0.062 \text{ W/m} \cdot \text{K}$ for k, 2115 kg/m^3 for ρ, and $920 \text{ J/kg} \cdot \text{K}$ for ρ.

$$\alpha = \frac{0.062}{2115 \times 920}$$
$$= 3.186 \times 10^{-8} \text{ m}^2/\text{s}$$

Step 2 of 3

Calculate the temperature at a depth of 3 cm from the highway surface after 60 min as shown.

$$\frac{T(x,i) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Here, initial temperature is T_i, surface temperature is T_s, depth from the highway surface is x, and time required for heat flux transfer is t.

Substitute 55 °C for T_i, 25 °C for T_s, 0.03 m for x, $3.186 \times 10^{-8} \text{ m}^2/\text{s}$ for α, and 3600 s for t.

$$\frac{T(0.03 \text{ m}, 3600 \text{ s}) - 55}{25 - 55} = \text{erfc}\left(\frac{0.03}{2\sqrt{3.186 \times 10^{-8} \times 3600}}\right) \quad \text{..... (1)}$$
$$T(0.03 \text{ m}, 3600 \text{ s}) - 55 = (-30)\text{erfc}(1.40)$$
$$T = 55 - 30[\text{erfc}(1.40)]$$

Obtain the value of $\text{erfc}(1.40)$ from table 4-4 as shown.

$$\text{erfc}(1.40) = 0.04772$$

Step 3 of 3

Calculate the temperature at a depth of 3 cm from the highway surface after 60 min as shown.

Substitute 0.04772 for $\text{erfc}(1.40)$ in equation (1).

$$T = 55 - 30[0.04772]$$
$$= 53.57 \text{ °C}$$

Therefore, the temperature at a depth of 3 cm from the highway surface after 60 min is $\boxed{53.57 \text{ °C}}$.

Calculate the heat flux transferred from the highway surface by using the relation,

$$\dot{q}_s = \frac{k(T_i - T_s)}{\sqrt{\pi \alpha t}}$$

Substitute 55 °C for T_i, 25 °C for T_s, $0.062 \text{ W/m} \cdot \text{K}$ for k, $3.186 \times 10^{-8} \text{ m}^2/\text{s}$ for α, and 3600 s for t.

$$\dot{q}_s = \frac{0.062(55 - 25)}{\sqrt{\pi \times 3.186 \times 10^{-8} \times 3600}}$$
$$= \frac{1.86}{18.98 \times 10^{-5}}$$
$$= 98 \text{ W/m}^2$$

Therefore, the heat flux transferred from the highway surface is $\boxed{98 \text{ W/m}^2}$.

Question: 18.078P

In a vacuum chamber, a thick slab is placed under an array of laser diodes with an output constant pulse *M* thermocouple is inserted inside the slab at 25 mm from the surface and the slab has an initial uniform temperature of20°C. The known properties of the slab are $k = 63.9$ W/m·K and $\alpha = 18.8 \times 10\text{-}6$ m2 /s. If the thermocouple measured temperature of 130°C after 30 s the slab surface has been exposed to the laser pulse, determine (*a*) the amount of J energy per unit surface area directed on the slab surface and (*b*) the thermocouple reading after 60 s has elapsed

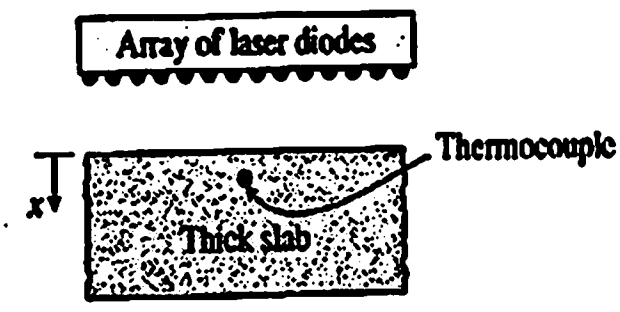


FIGURE P 18-78

Step 1 of 3

(a).

Calculate the energy pulse directed on the surface of slab per unit surface area by using the relation,

$$T(x,t)-T_i=\frac{e_s}{k\sqrt{\frac{\pi t}{\alpha}}}\exp\left(-\frac{x^2}{4\alpha t}\right) \quad \text{..... (1)}$$

Rearrange the equation as shown.

$$T(x,t)-T_i=\frac{e_s}{k\sqrt{\frac{\pi t}{\alpha}}}\exp\left(-\frac{x^2}{4\alpha t}\right)$$

$$e_s=\frac{k\sqrt{\frac{\pi t}{\alpha}}\left[T(x,t)-T_i\right]}{\exp\left(-\frac{x^2}{4\alpha t}\right)} \quad \text{..... (2)}$$

Here, thermal conductivity of the slab is k , thermal diffusivity of the slab is α , time taken by the slab to reach temperature $T(x,t)$ is t , depth to which thermocouple is inserted is x .

Step 2 of 3

Calculate the energy pulse directed on the surface of slab per unit surface area as shown.

Substitute $63.9\ \text{W/m}\cdot\text{K}$ for k , $30\ \text{s}$ for t , $18.8\times10^{-6}\ \text{m}^2/\text{s}$ for α , $130\ ^\circ\text{C}$ for $T(x,t)$, $20\ ^\circ\text{C}$ for T_i , and $0.025\ \text{m}$ for x in equation (2).

$$e_s=\frac{(63.9)\sqrt{\frac{\pi\times30}{18.8\times10^{-6}}}\left[130-20\right]}{\exp\left(-\frac{(0.025)^2}{4\times18.8\times10^{-6}\times30}\right)}$$

$$=\frac{7029\times2.24\times10^{-3}}{\exp(-0.277)}$$

$$=2.076\times10^7\ \text{J/m}^2$$

Therefore, the energy pulse directed on the surface of slab per unit surface area is $\boxed{2.076\times10^7\ \text{J/m}^2}$.

Step 3 of 3

(b).

Calculate the thermocouple reading after $60\ \text{s}$ using the relation in equation (1) as shown.

$$T(x,t)-T_i=\frac{e_s}{k\sqrt{\frac{\pi t}{\alpha}}}\exp\left(-\frac{x^2}{4\alpha t}\right)$$

Substitute $2.076\times10^7\ \text{J/m}^2$ for e_s , $63.9\ \text{W/m}\cdot\text{K}$ for k , $60\ \text{s}$ for t , $18.8\times10^{-6}\ \text{m}^2/\text{s}$ for α , $20\ ^\circ\text{C}$ for T_i , and $0.025\ \text{m}$ for x in equation (1).

$$T-20=\frac{2.076\times10^7}{(63.9)\sqrt{\frac{\pi\times60}{18.8\times10^{-6}}}}\exp\left(-\frac{(0.025)^2}{4\times18.8\times10^{-6}\times60}\right)$$

$$=20+10.26\exp(-0.1385)$$

$$=20+89.33$$

$$=109.33\ ^\circ\text{C}$$

Therefore, the thermocouple reading after $60\ \text{s}$ is $\boxed{109.33\ ^\circ\text{C}}$.

Question: 18.079P

What is the product solution method? How is it used to determine the transient temperature distribution in a two-dimensional system?

Step 1 of 1

The product solution enables us to determine the dimensionless temperature of two- or three-dimensional heat transfer problems as the product of dimensionless temperatures of one-dimensional heat transfer problems. The dimensionless temperature for a two-dimensional problem is determined by determining the dimensionless temperatures in both directions, and taking their product.

Question: 18.080P

How is the product solution used to determine the variation of temperature with time and position in three-dimensional systems?

Step 1 of 1

The dimensionless temperature for a three-dimensional heat transfer is determined by determining the dimensionless temperatures of one-dimensional geometries whose intersection is the three dimensional geometry, and taking their product.

Question: 18.081P

A short cylinder initially at a uniform temperature Ti is subjected to convection from all of its surfaces to a medium at temperature $T\infty$. Explain how you can determine the temperature of the midpoint of the cylinder at a specified time t.

Step 1 of 1

This short cylinder is physically formed by the intersection of a long cylinder and a plane wall. The dimensionless temperatures at the center of plane wall and at the center of the cylinder are determined first. Their product yields the dimensionless temperature at the center of the short cylinder.

Question: 18.082P

Consider a short cylinder whose top and bottom surfaces are insulated. The cylinder is initially at a uniform temperature *Ti* and is subjected to convection from its side surface to a medium at temperature T_∞ with a heat transfer coefficient of *h*. Is the heat transfer in this short cylinder one-or two-dimensional? Explain.

Step 1 of 1

The heat transfer in this short cylinder is one-dimensional since there is no heat transfer in the axial direction. The temperature will vary in the radial direction only.

Question: 18.083P pg 1

Consider a cubic block whose sides are 5 cm long and a cylindrical block whose height and diameter are also 5 cm. Both blocks are initially at 20°C and are made of granite (k = 2.5 W/m·K and α = 1.15 × 10-6 m2 /s). Now both blocks are exposed to hot gases at 500°C in a furnace on all of their surfaces with a heat transfer coefficient of 40 W/m2 K. Determine the center temperature of each geometry after 10, 20. and 60 min.

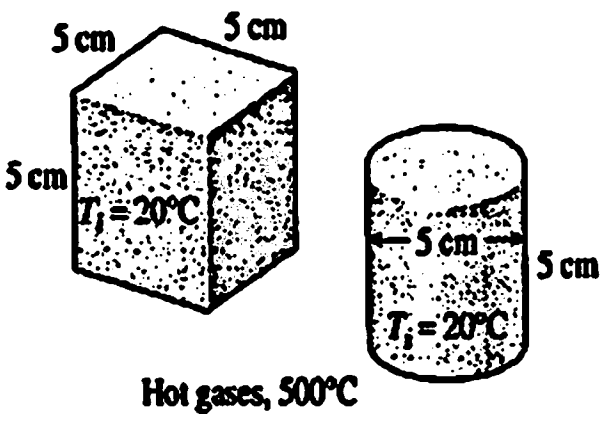


FIGURE P 18-83

Step 1 of 11

Given:

Length of the cubic block $L = 5$ cm

Height of the cylindrical block $H = 5$ cm

Diameter of the cylindrical block $D = 5$ cm

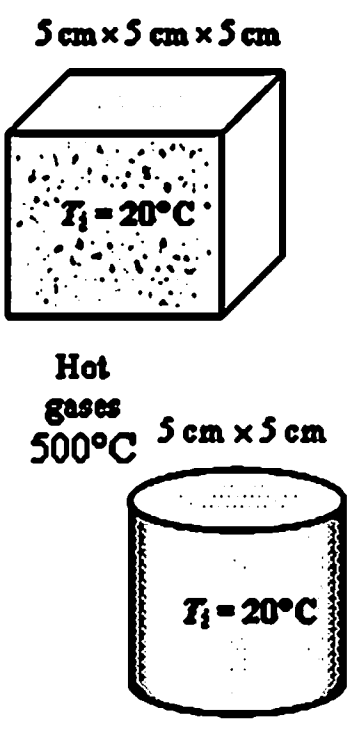
Initial temperature of the blocks $T_i = 20°C$

Thermal Conductivity, $k = 2.5\ W/m \cdot °C$

Thermal Diffusivity, $\alpha = 1.15 \times 10^{-6}\ m^2/s$

Heat transfer coefficient $h = 40\ \frac{W}{m^2 \cdot °C}$

Step 2 of 11



Step 3 of 11

Calculation after 10 minutes: The Biot number, the corresponding constants, and the Fourier number are

$$Bi = \frac{hL}{k}$$

$$Bi = \frac{(40\ W/m^2.°C)(0.025\ m)}{(2.5\ W/m.°C)}$$

$$Bi = 0.400$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 0.5932$ and $A_1 = 1.0580$

Step 4 of 11

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15 \times 10^{-6}\ m^2/s)(10\ min \times 60\ s/min)}{(0.025\ m)^2}$$

$$\tau = 1.104 > 0.2$$

To determine the center temperature, the product solution can be written as

$$\theta(0,0,0,t)_{block} = [\theta(0,t)_{wall}]^3$$

$$\frac{T(0,0,0,t) - T_\infty}{T_i - T_\infty} = \left(A_1 e^{-\lambda_1^2 \tau}\right)^3$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{(1.0580) e^{-(0.5932)^2 (1.104)}\right\}^3$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = 0.369$$

$$\boxed{T(0,0,0,t) = 323°C}$$

Step 5 of 11

After 20 minutes

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15 \times 10^{-6}\ m^2/s)(20\ min \times 60\ s/min)}{(0.025\ m)^2}$$

$$\tau = 2.208 > 0.2$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = \left\{(1.0580) e^{-(0.5932)^2 (2.208)}\right\}^3$$

$$\frac{T(0,0,0,t) - 500}{20 - 500} = 0.115$$

$$\boxed{T(0,0,0,t) = 445°C}$$

Step 6 of 11

After 60 minutes

$$\frac{\alpha t}{}$$

Step 6 of 11

After 60 minutes

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15\times10^{-6}\ \text{m}^2/\text{s})(60\ \text{min}\times 60\ \text{s/min})}{(0.025\ \text{m})^2}$$

$$\tau = 6.624 > 0.2$$

$$\frac{T(0,0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(6.624)}\right\}^3$$

Step 7 of 11

$$\frac{T(0,0,0,t)-500}{20-500} = 0.00109$$

$$\boxed{T(0,0,0,t)=500^\circ C}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

Step 8 of 11

Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 2.5$ cm and a plane wall of thickness $2L = 5$ cm.

After 10 minutes: The Biot number and the corresponding constants for the long cylinder are

$$Bi = \frac{hr_o}{k}$$

$$Bi = \frac{(40\ \text{W/m}^2.^\circ\text{C})(0.025\ \text{m})}{(2.5\ \text{W/m}.^\circ\text{C})}$$

$$Bi = 0.400$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$$\lambda_1 = 0.8516 \quad \text{and} \quad A_1 = 1.0931$$

Step 9 of 11

To determine the center temperature, the product solution can be written as

$$\theta(0,0,t)_{brick} = [\theta(0,t)_{wall}][\theta(0,t)_{cyl}]$$

$$\frac{T(0,0,t)-T_\infty}{T_i-T_\infty} = \left(A_1 e^{-\lambda_1^2\tau}\right)_{wall}\left(A_1 e^{-\lambda_1^2\tau}\right)_{cyl}$$

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(1.104)}\right\}\left\{(1.0931)e^{-(0.8516)^2(1.104)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.352$$

$$\boxed{T(0,0,t)=331^\circ C}$$

Step 10 of 11

After 20 minutes

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(2.208)}\right\}\left\{(1.0931)e^{-(0.8516)^2(2.208)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.107$$

$$\boxed{T(0,0,t)=449^\circ C}$$

Step 11 of 11

After 60 minutes

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(6.624)}\right\}\left\{(1.0931)e^{-(0.8516)^2(6.624)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.00092$$

$$\boxed{T(0,0,t)=500^\circ C}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

Question:

Repeat Prob. 18-83 with the heat transfer coefficient at the top and the bottom surfaces of each block being doubled to 80 W/m2 ·K.

Step 1 of 11

Given:

Length of the cubic block $L = 5$ cm

Height of the cylindrical block $H = 5$ cm

Diameter of the cylindrical block $D = 5$ cm

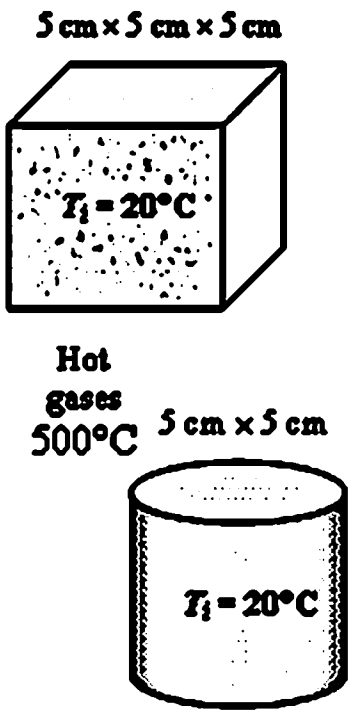
Initial temperature of the blocks $T_i = 20°C$

Thermal Conductivity, $k = 2.5$ W/m · °C

Thermal Diffusivity, $\alpha = 1.15\times10^{-6}$ m^2/s

Heat transfer coefficient $h = 40 \dfrac{\text{W}}{\text{m}^2 \cdot °\text{C}}$

Step 2 of 11



Step 3 of 11

Calculation after 10 minutes: The Biot number, the corresponding constants, and the Fourier number are

$$Bi = \frac{hL}{k}$$

$$Bi = \frac{(40\ \text{W/m}^2.°\text{C})(0.025\ \text{m})}{(2.5\ \text{W/m.°C})}$$

$$Bi = 0.400$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 0.5932$ and $A_1 = 1.0580$

Step 4 of 11

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15\times10^{-6}\ \text{m}^2/\text{s})(10\ \text{min}\times60\ \text{s/min})}{(0.025\ \text{m})^2}$$

$$\tau = 1.104 > 0.2$$

To determine the center temperature, the product solution can be written as

$$\theta(0,0,0,t)_{\text{block}} = [\theta(0,t)_{\text{wall}}]^2[\theta(0,t)_{\text{wall}}]$$

$$\frac{T(0,0,0,t)-T_\infty}{T_i-T_\infty} = \left(A_1 e^{-\lambda_1^2\tau}\right)^2\left(A_1 e^{-\lambda_1^2\tau}\right)$$

$$\frac{T(0,0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(1.104)}\right\}^2\left\{(1.1016)e^{-(0.7910)^2(1.104)}\right\}$$

$$\boxed{T(0,0,0,t) = 364°\text{C}}$$

Step 5 of 11

<u>After 20 minutes</u>

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15\times10^{-6}\ \text{m}^2/\text{s})(20\ \text{min}\times60\ \text{s/min})}{(0.025\ \text{m})^2}$$

$$\tau = 2.208 > 0.2$$

$$\frac{T(0,0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(2.208)}\right\}^2\left\{(1.1016)e^{-(0.7910)^2(2.208)}\right\}$$

$$\frac{T(0,0,0,t)-500}{20-500} = 0.0654$$

$$\boxed{T(0,0,0,t) = 469°\text{C}}$$

Step 6 of 11

After 60 minutes

$$\tau = \frac{\alpha t}{L^2}$$

$$\tau = \frac{(1.15\times10^{-6}\ \text{m}^2/\text{s})(60\ \text{min}\times60\ \text{s/min})}{(0.025\ \text{m})^2}$$

$$\tau = 6.624 > 0.2$$

Step 7 of 11

$$\frac{T(0,0,0,t)-500}{20-500} = \left\{(1.0580)e^{-(0.5932)^2(6.624)}\right\}^2\left\{(1.1016)e^{-(0.7910)^2(6.624)}\right\}$$

$$\frac{T(0,0,0,t)-500}{20-500} = 0.000186$$

$$\boxed{T(0,0,0,t)=500°\text{C}}$$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable

$$\frac{T(0,0,t)-500}{20-500}=0.000186$$

$\boxed{T(0,0,t)=500°C}$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable

Step 8 of 11

Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 2.5$ cm and a plane wall of thickness $2L = 5$ cm.

<u>After 10 minutes</u>: The Biot number and the corresponding constants for the long cylinder are

$$Bi = \frac{hr_o}{k}$$

$$Bi = \frac{(40\ \text{W/m}^2.°\text{C})(0.025\ \text{m})}{(2.5\ \text{W/m}.°\text{C})}$$

$$Bi = 0.400$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$$\lambda_1 = 0.8516 \quad \text{and} \quad A_1 = 1.0931$$

Step 9 of 11

To determine the center temperature, the product solution can be written as

$$\theta(0,0,t)_{block} = [\theta(0,t)_{wall}][\theta(0,t)_{cyl}]$$

$$\frac{T(0,0,t)-T_\infty}{T_i-T_\infty} = \left(A_1 e^{-\lambda_1^2 \tau}\right)_{wall}\left(A_1 e^{-\lambda_1^2 \tau}\right)_{cyl}$$

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.1061)e^{-(0.7910)^2(1.104)}\right\}\left\{(1.0931)e^{-(0.8516)^2(1.104)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.271$$

$\boxed{T(0,0,t)=370°C}$

Step 10 of 11

After 20 minutes

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.1016)e^{-(0.7910)^2(2.208)}\right\}\left\{(1.0931)e^{-(0.8516)^2(2.208)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.06094$$

$\boxed{T(0,0,t)=471°C}$

Step 11 of 11

After 60 minutes

$$\frac{T(0,0,t)-500}{20-500} = \left\{(1.1016)e^{-(0.7910)^2(6.624)}\right\}\left\{(1.0931)e^{-(0.8516)^2(6.624)}\right\}$$

$$\frac{T(0,0,t)-500}{20-500} = 0.0001568$$

$\boxed{T(0,0,t)=500°C}$

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable.

Question: 18.085P pg 1

A short brass cylinder (ρ = 8530 kg/m3 , cp = 0.389 kJ/kg-K, k = 110 W/m2·K, and α = 3.39 × 10–5 m2 /s) of diameter 4 cm and height 20 cm is initially at a uniform temperature of 150°C. The cylinder is now placed in atmospheric air at 20°C, where heat transfer takes place by convection with a heat transfer coefficient of 40 W/m2 -K. Calculate (*a*) the center temperature of the cylinder; (*b*) the center temperature of the top surface of the cylinder; and (*c*) the total heat transfer from the cylinder 15 min after the start of the cooling.

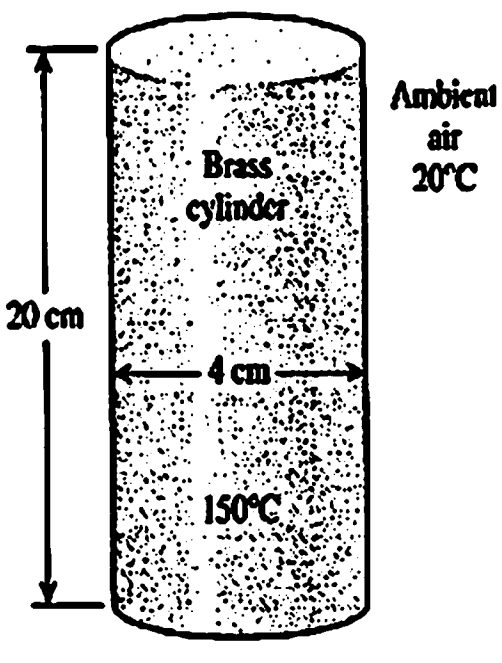


FIGURE P18–85

Step 1 of 8

Calculate Biot number,

$$Bi = \frac{hL_c}{k}$$

$$Bi = \frac{40 \times L_c}{110}$$

Now, characteristic length can be obtained by approximating a plane wall having thickness: $2L_c = 20$ cm

Putting this into Biot number expression, we get:

$$Bi = \frac{hL_c}{K}$$

$$= \frac{40 \times L_c}{110}$$

$$= \frac{40 \times \left(\frac{10}{100}\right)}{110}$$

$$Bi = 0.036$$

Step 2 of 8

For this Biot number, using table 4-2, we get: $\lambda = 0.187, A = 1.0059$

Again, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L^2}$$

$$= \frac{\left(3.39 \times 10^{-5}\right)(15 \times 60)}{\left(\frac{10}{100}\right)^2}$$

$$= 3.051$$

So, to determine the dimensionless temperature at the centre of approximated plane wall, we can use:

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = Ae^{-\lambda^2 \tau}$$

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = 1.0059 \times e^{-0.187^2 \times 3.051}$$

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = 0.904$$

Step 3 of 8

For long cylinder:

$$Bi = \frac{hL_c}{K}$$

$$= \frac{40 \times L_c}{110}$$

$$= \frac{40 \times \left(\frac{2}{100}\right)}{110}$$

$$Bi = 7.27 \times 10^{-3}$$

For this Biot number, using table 4-2, we get: $\lambda = 0.08, A = 1.0006$

And, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L^2}$$

$$= \frac{\left(3.39 \times 10^{-5}\right)(15 \times 60)}{\left(\frac{2}{100}\right)^2}$$

$$\tau = 76.275$$

Step 4 of 8

So, the dimensionless temperature is determined to be:

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = Ae^{-\lambda^2 \tau}$$

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = 1.0006 \times e^{-0.08^2 \times 76.275}$$

$$\left(\frac{T - T_\alpha}{T_i - T_\alpha}\right) = 0.614$$

Finally, temperature of the short cylinder at the centre is obtained as:

$$\frac{T - T_\alpha}{T_i - T_\alpha} = 0.904 \times 0.614$$

$$\Rightarrow \frac{T - T_\alpha}{T_i - T_\alpha} = 0.55$$

$$\Rightarrow \frac{T - 20}{(150 - 20)} = 0.55$$

$$\Rightarrow \frac{T-T_\alpha}{T_i-T_\alpha}=0.55$$

$$\Rightarrow \frac{T-20}{(150-20)}=0.55$$

$$\Rightarrow \boxed{T=91.5^\circ \text{ C}}$$

So the temperature of the short cylinder at centre is obtained to be $\boxed{91.5°C}$.

Step 5 of 8

For the determination of the centre temperature of the top surface of the cylinder, first the dimensionless temperature at the surface of the chosen approximated plane wall is to be obtained. This is:

$$\left(\frac{T-T_\alpha}{T_i-T_\alpha}\right)=Ae^{-\lambda^2\tau}\cos(\lambda)$$

$$=1.0059\times e^{-0.187^2\times 3.051}\times\cos(0.187)$$

$$=0.904\times\cos(0.187)$$

$$\left(\frac{T-T_\alpha}{T_i-T_\alpha}\right)=0.888$$

For long cylinder:

$$Bi=\frac{hL_c}{K}$$

$$=\frac{40\times L_c}{110}$$

$$=\frac{40\times\left(\frac{2}{100}\right)}{110}$$

$$Bi=7.27\times10^{-3}$$

Step 6 of 8

So, the dimensionless temperature is determined to be:

$$\left(\frac{T-T_\alpha}{T_i-T_\alpha}\right)=Ae^{-\lambda^2\tau}$$

$$=1.0006\times e^{-0.08^2\times 76.275}$$

$$\left(\frac{T-T_\alpha}{T_i-T_\alpha}\right)=0.614$$

Finally, temperature of the short cylinder at the top surface is obtained as:

$$\frac{T-T_\alpha}{T_i-T_\alpha}=0.888\times0.614$$

$$\frac{T-T_\alpha}{T_i-T_\alpha}=0.54$$

$$\frac{T-20}{(150-20)}=0.54$$

$$\boxed{T=90.2^\circ \text{ C}}$$

Step 7 of 8

So the temperature of the short cylinder at top surface is obtained 90.2° C.

Maximum heat transfer from the cylinder is obtained as: $Q=mC_p(150\text{-}20)$

Where mass (m) of cylinder is: $8530\times\pi\times0.02^2\times\frac{20}{100}=2.14$ Kg

Putting this data:

$$Q=mC_p(150\text{-}20)$$

$$=2.14\times0.389\times(150\text{-}20)$$

$$Q=108 \text{ KJ}$$

And, for dimensionless heat transfer we get for plane wall:

$$\left(\frac{Q}{108}\right)_W=1-0.904\times\frac{\sin(\lambda)}{\lambda}$$

$$\left(\frac{Q}{108}\right)_W=1-0.904\times\frac{\sin(0.187)}{0.187}$$

$$\left(\frac{Q}{108}\right)_W=0.1$$

Step 8 of 8

And for cylinder:

$$\left(\frac{Q}{108}\right)_C=1-2\times0.614\times\frac{J_1(\lambda)}{\lambda}$$

$$=1-2\times0.614\times\frac{J_1(0.187)}{0.187}$$

$$=1-2\times0.614\times\frac{0.093}{0.187}$$

$$\left(\frac{Q}{108}\right)_C=0.389$$

So total heat transfer for short cylinder is:

$$\left(\frac{Q}{108}\right)=\left(\frac{Q}{108}\right)_W+\left(\frac{Q}{108}\right)_C\times\left[1-\left(\frac{Q}{108}\right)_W\right]$$

$$\left(Q/108\right)=0.1+0.389\times[1-0.1]$$

$$\left(Q/108\right)=0.45$$

$$\boxed{Q=48.60 \text{ KJ}}$$

Finally, total heat transfer for short cylinder is obtained as $\boxed{48.60 \text{ KJ}}$

Question:

Reconsider Prob. 18-85. Using EES (or other) b & software, investigate the effect of the cooling time on the center temperature of the cylinder, the center temperature of the top surface of the cylinder, and the total heat transfer. Let the time vary from 5 min to 60 min. Plot the center temperature of the cylinder, the center temperature of the top surface, and the total heat transfer as a function of the time, and discuss the results.

Step 1 of 17

Given data:

Diameter of cylinder $D = 4$ cm

Height of cylinder $L = 20$ cm

Initial temperature of cylinder $T_i = 150°C$

Atmospheric temperature $T_\infty = 20°C$

The density of brass $\rho = 8530$ kg/m^3

The specific heat of brass are given to be $C_p = 0.389$ kJ/kg·°C,

The thermal conductivities of brass are given to be $k = 110$ W/m·°C,

The thermal diffusivity of brass are given to be $\alpha = 3.39\times10^{-5}$ m^2/s.

Step 2 of 17

For this problem:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{40\times L_c}{110}$$

Now, characteristic length can be obtained by approximating a plane wall having thickness: $2L_c = 20$ cm

Step 3 of 17

Putting this into Biot number expression, we get:

$$Bi = \frac{hL_c}{K}$$

$$= \frac{40\times L_c}{110}$$

$$= \frac{40\times\left(10/100\right)}{110}$$

$$= 0.036$$

Step 4 of 17

From the coefficient used in one term approximation solution of transient one dimensional heat conduction,

At Bi=0.036

$\lambda = 0.187, A = 1.0059$

Again, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L^2}$$

So, to determine the dimensionless temperature at the centre of approximated plane wall, we can use:

$$\frac{T - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2\tau}$$

$$\frac{T - T_\alpha}{T_i - T_\alpha} = 1.0059\times e^{-0.187^2\times\tau}$$

Step 5 of 17

For long cylinder:

$$Bi = \frac{hL_c}{K}$$

$$= \frac{40\times L_c}{110}$$

$$= \frac{40\times\left(2/100\right)}{110}$$

$$= 7.27\times10^{-3}$$

Step 6 of 17

From the coefficient used in one term approximation solution of transient one dimensional heat conduction,

At $Bi = 7.27\times10^{-3}$

$\lambda = 0.08, A = 1.0006$

And, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L^2}$$

So, the dimensionless temperature is determined to be:

$$\frac{T - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2\tau}$$

$$\frac{T - T_\alpha}{T_i - T_\alpha} = 1.0006\times e^{-0.08^2\times\tau}$$

Step 7 of 17

Finally, temperature of the short cylinder at the centre is obtained as:

$$\frac{T - T_\alpha}{T_i - T_\alpha} = 1.0059\times e^{-0.187^2\times\tau}\times 1.0006\times e^{-0.08^2\times\tau}$$

$$\frac{T - 20}{(150 - 20)} = 1.0059\times e^{-0.187^2\times\tau}\times 1.0006\times e^{-0.08^2\times\tau}$$

Step 8 of 17

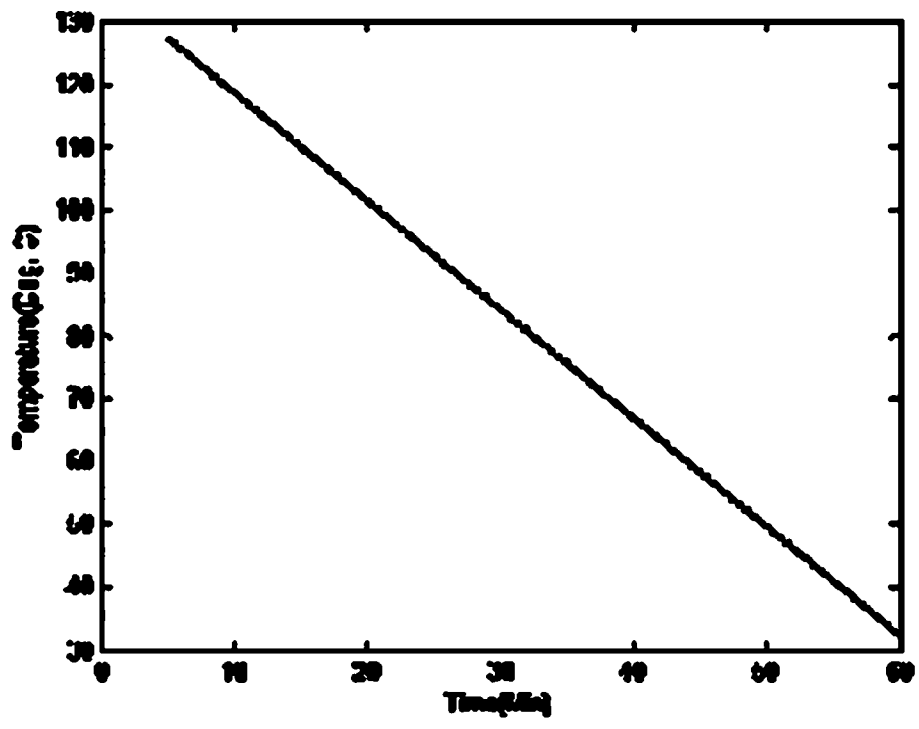
The MATLAB code for this plot is:

```
clear all
clc
x=linspace(5,60,2);
y=((3.39*(10^-5))*x*60)/(0.1*0.1);
z=(1.0059*exp(-0.187*0.187*y));
```

```
clear all
clc
x=linspace(5,60,2);
y=((3.39*(10^-5))*x*60)/(0.1*0.1);
z=(1.0059*exp(-0.187*0.187*y));
y1=((3.39*(10^-5))*x*60)/(0.02*0.02);
z1=(1.0006*exp(-0.08*0.08*y1));
zz=z.*z1.*(150-20)+20;
plot(x,zz);
```

Step 9 of 17

Below is shown the variation of the temperature of the short cylinder at centre versus time:



Step 10 of 17

For the determination of the centre temperature of the top surface of the cylinder, first the dimensionless temperature at the surface of the chosen approximated plane wall is to be obtained. This is:

$$\frac{T-T_\alpha}{T_i-T_\alpha}=Ae^{-\lambda^2\tau}\cos(\lambda)$$

$$\frac{T-T_\alpha}{T_i-T_\alpha}=1.0059\times e^{-0.187^2\times\tau}\times\cos(0.187)$$

For long cylinder:

$$Bi=\frac{hL_o}{K}$$

$$=\frac{40\times L_o}{110}$$

$$=\frac{40\times\left(2/100\right)}{110}$$

$$=7.27\times10^{-3}$$

Step 11 of 17

So, the dimensionless temperature is determined to be:

$$\frac{T-T_\alpha}{T_i-T_\alpha}=Ae^{-\lambda^2\tau}$$

$$\frac{T-T_\alpha}{T_i-T_\alpha}=1.0006\times e^{-0.08^2\times\tau}$$

Finally, temperature of the short cylinder at the top surface is obtained as:

$$\frac{T-T_\alpha}{T_i-T_\alpha}=1.0059\times e^{-0.187^2\times\tau}\times\cos(0.187)\times1.0006\times e^{-0.08^2\times\tau}$$

$$\frac{T-20}{(150-20)}=1.0059\times e^{-0.187^2\times\tau}\times\cos(0.187)\times1.0006\times e^{-0.08^2\times\tau}$$

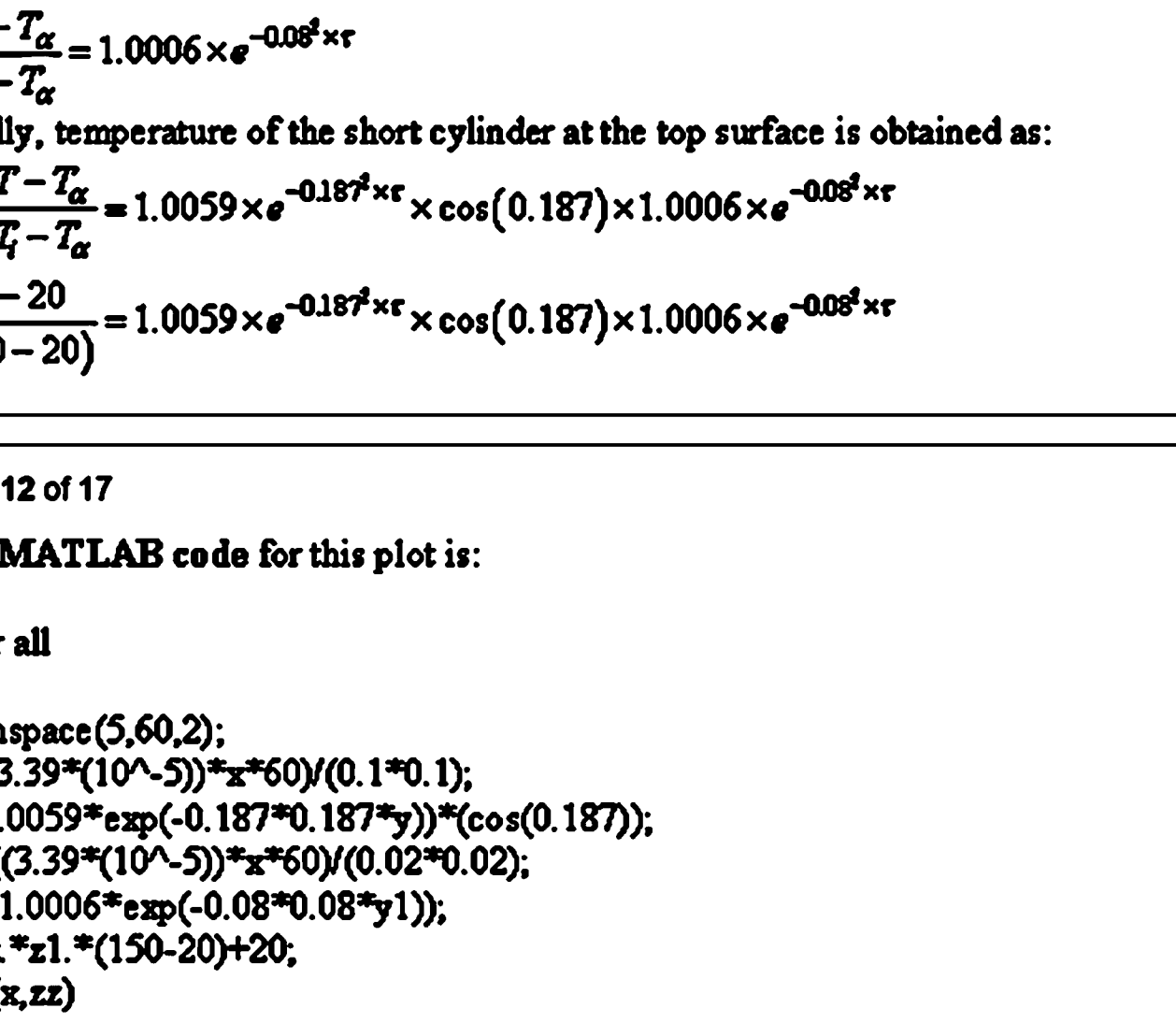
Step 12 of 17

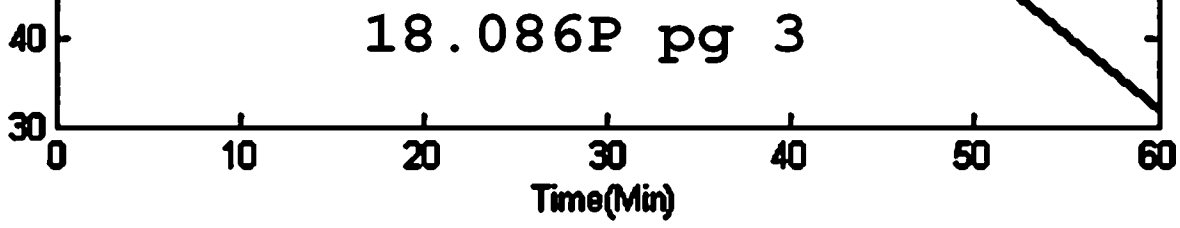
The MATLAB code for this plot is:

```
clear all
clc
x=linspace(5,60,2);
y=((3.39*(10^-5))*x*60)/(0.1*0.1);
z=(1.0059*exp(-0.187*0.187*y))*(cos(0.187));
y1=((3.39*(10^-5))*x*60)/(0.02*0.02);
z1=(1.0006*exp(-0.08*0.08*y1));
zz=z.*z1.*(150-20)+20;
plot(x,zz)
```

Step 13 of 17

Below is shown the variation of the temperature of the short cylinder at top surface versus time:





Step 14 of 17

Maximum heat transfer from the cylinder is obtained as: $Q = mC_p(150\text{-}20)$

Where mass (m) of cylinder is: $8530 \times \pi \times 0.02^2 \times \frac{20}{100} = 2.14 \text{ Kg}$

Putting this data:

$$Q = mC_p(150\text{-}20)$$
$$= 2.14 \times 0.389 \times (150\text{-}20)$$
$$= 108 \text{ KJ}$$

Step 15 of 17

And, for dimensionless heat transfer we get for plane wall:

$$\left(Q/108\right)_w = 1 - 1.0059 \times e^{-0.187^2 \times \tau} \times \frac{\sin(0.187)}{0.187}$$

And for cylinder:

$$\left(Q/108\right)_c = 1 - 2 \times 1.0006 \times e^{-0.08^2 \times \tau} \times \frac{J_1(0.08)}{0.08}$$

So total heat transfer for short cylinder is:

$$\left(Q/108\right) = \left(Q/108\right)_w + \left(Q/108\right)_c \times \left[1 - \left(Q/108\right)_w\right]$$

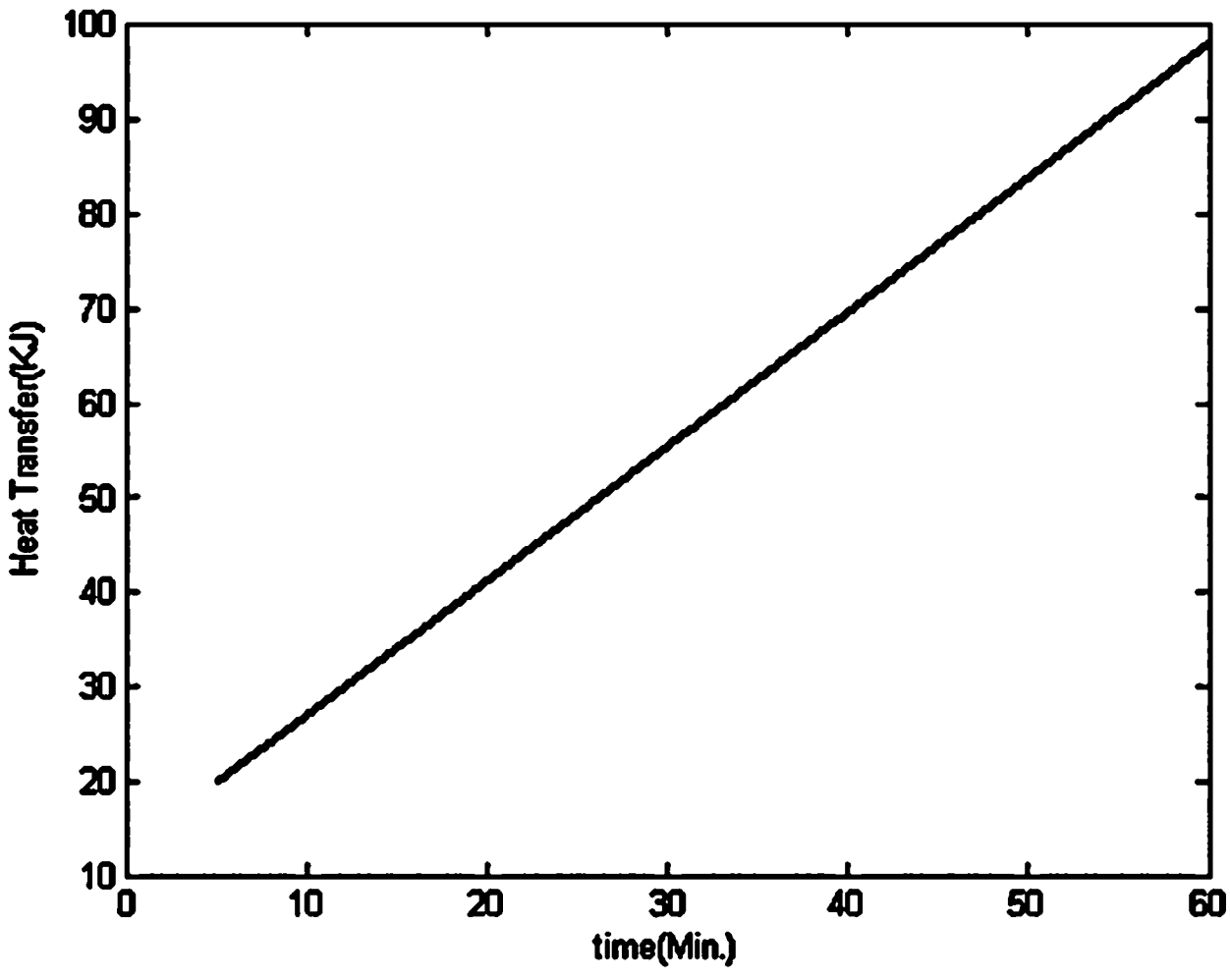
Step 16 of 17

The MATLAB code for this plot is:

```
clear all
clc
x=linspace(5,60,2);
y=((3.39*(10^-5))*x*60)/(0.1*0.1);
z=1-(((1.0059/0.187)*exp(-0.187*0.187*y))*(sin(0.187)));
y1=((3.39*(10^-5))*x*60)/(0.02*0.02);
z1=1-((2*1.0006*exp(-0.08*0.08*y1))*(0.093/0.187));
zz=108*(z+z1.*(1-z));
plot(x,zz)
```

Step 17 of 17

Below is shown the variation of the heat transfer of the short cylinder versus time:



Question: 18.087P

A semi-infinite aluminum cylinder (k = 237 W/m·K, α = 9.71 × 10–5 m2 /s) of diameter D = 15 cm is initially at a uniform temperature of Ti = 115°C. The cylinder is now placed in water at 10°C, where heat transfer lakes place by convection with a heat transfer coefficient of h = 140 W/m2 ·K. Determine the temperature at the center of the cylinder 5 cm from the end surface 8 min after the start of cooling.

Step 1 of 6

Given:

Diameter of cylinder $D = 15\text{cm}$

Initial temperature $T_i = 115^0\text{C}$

Temperature of water $T_\infty = 10^0\text{C}$

Convection heat transfer coefficient $h = 140\,\text{W/m}^2.^0\text{C}$

Thermal conductivity of aluminum $k = 237\,\text{W/m.°C}$

Thermal diffusivity of aluminum $\alpha = 9.71\times10^{-5}\text{m}^2/s$

Step 2 of 6

The semi-infinite cylinder can physically be formed by the intersection of a long cylinder of radius $r_o = D/2 = 7.5$ cm and a semi-infinite medium.

The dimensionless temperature 5 cm from the surface of a semi-infinite medium is first determined from

$$\frac{T(x,t)-T_i}{T_\infty-T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{hx}{k}+\frac{h^2\alpha t}{k^2}\right)\left[erfc\left(\frac{x}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)\right]$$

$$= erfc\left(\frac{0.05}{2\sqrt{(9.71\times10^{-5})(8\times60)}}\right) - \exp\left(\frac{(140)(0.05)}{237}+\frac{(140)^2(9.71\times10^{-5})(8\times60)}{(237)^2}\right)$$

$$\times\left[erfc\left(\frac{0.05}{2\sqrt{(9.71\times10^{-5})(8\times60)}}+\frac{(140)\sqrt{(9.71\times10^{-5})(8\times60)}}{237}\right)\right]$$

$$= erfc(0.1158) - \exp(0.0458)erfc(0.2433)$$

$$\frac{T(x,t)-T_i}{T_\infty-T_i} = 0.8699 - (1.0468)(0.7308)$$

$$\frac{T(x,t)-T_i}{T_\infty-T_i} = 0.1049$$

$$\theta_{semi-inf} = \frac{T(x,t)-T_\infty+T_\infty-T_i}{T_i-T_\infty} = 0.1049$$

$$\theta_{semi-inf} = \frac{T(x,t)-T_\infty}{T_i-T_\infty} - \frac{T_i-T_\infty}{T_i-T_\infty} = 0.1049$$

$$\frac{T(x,t)-T_\infty}{T_i-T_\infty} - 1 = 0.1049$$

$$\frac{T(x,t)-T_\infty}{T_i-T_\infty} = 1.1049$$

Step 3 of 6

The Biot number is calculated for the long cylinder to be

$$Bi = \frac{hr_o}{k}$$

$$= \frac{(140)\times(0.075)}{(237)}$$

$$Bi = 0.0443$$

The constants λ_1 and A_1 corresponding to this Biot number from Table transient heat conduction in cylinders

$\lambda_1 = 0.2948$ and

$A_1 = 1.0110$

Step 4 of 6

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2}$$

$$= \frac{(9.71\times10^{-5})(8\times60)}{(0.075)^2}$$

$$\tau = 8.286 > 0.2$$

Therefore, the one-term approximate solution (or the transient temperature charts) is applicable.

Step 5 of 6

Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{o,cyl} = \frac{T_0-T_\infty}{T_i-T_\infty}$$

$$= A_1 e^{-\lambda_1^2\tau}$$

$$= (1.0110)e^{-(0.2948)^2(8.286)}$$

$$\theta_{o,cyl} = 0.4921$$

Step 6 of 6

The center temperature of the semi-infinite cylinder then becomes

$$\left[\frac{T(x,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{semi-infinite}\\\text{cylinder}}} = \theta_{semi-inf}(x,t)\times\theta_{o,cyl}$$

$$= 1.1049\times0.4921$$

$$\left[\frac{T(x,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{semi-infinite}\\\text{cylinder}}} = 0.5437$$

$$\left[\frac{T(x,0,t)-10}{115-10}\right]_{\substack{\text{semi-infinite}\\\text{cylinder}}} = 0.5437$$

$$\boxed{T(x,0,t) = 67.0885^0\text{C}}$$

Question: 18.088P pg 1

A hot dog can be considered to be a cylinder 5 in long and 0.8 in in diameter whose properties are ρ = 61.2 lbm/ft3 , *cp* = 0.93 Btu/lbm-°F, *k* = 0.44 Btu/h·ft·°F, and α = 0.0077 ft2 /h. A hot dog initially at 40°F is dropped into boiling water at 212°F. If the heat transfer coefficient at the surface of the hot dog is estimated to be 120 Btu/h·ft2 ·°F, determine the center temperature of the hot dog after 5, 10, and 15 min by treating the hot dog as (*a*) a finite cylinder and (*b*) an infinitely long cylinder.

Step 1 of 10

Given:

Length of cylinder $L = 5$ in

Diameter of cylinder $D = 0.8$ in

The thermal conductivity of the hot dog $k = 0.44 \frac{\text{Btu}}{\text{h ft °F}}$,

The density of the hot dog $\rho = 61.2 \frac{\text{lbm}}{\text{ft}^3}$

The specific heat of hotdog $C_p = 0.93 \frac{\text{Btu}}{\text{lbm.°F}}$

The thermal diffusivity of hot dog $\alpha = 0.0077\ \text{ft}^2/\text{h}$

Step 2 of 10

Hot dog is dropped into boiling water.

The center temperature of the hot dog is do be determined by treating hot dog as a finite cylinder and also as an infinitely long cylinder.

When treating hot dog as a finite cylinder, heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial *x*- and the radial *r*-directions.

When treating hot dog as an infinitely long cylinder, heat conduction is one-dimensional in the radial *r*- direction.

The thermal properties of the hot dog are constant.

This hot dog can physically be formed by the intersection of a long cylinder of radius r_o $D/2 = (0.4/12)$ ft and a plane wall of thickness $2L = (5/12)$ ft. The distance *x* is measured from the midplane.

Step 3 of 10

<u>After 5 minutes</u>

First the Biot number is calculated for the plane wall to be

$$Bi = \frac{hL}{k}$$

$$Bi = \frac{(120\ \text{Btu/h.ft}^2\text{.°F})(2.5/12\ \text{ft})}{(0.44\ \text{Btu/h.ft.°F})}$$

$$Bi = 56.8$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 1.5421$ and $A_1 = 1.2728$

Step 4 of 10

The Fourier number $\tau = \frac{\alpha t}{L^2}$

$$\tau = \frac{(0.0077\ \text{ft}^2/\text{h})(5/60\ \text{h})}{(2.5/12\ \text{ft})^2}$$

$\tau = 0.015 < 0.2$ (Be cautious!)

Then the dimensionless temperature at the center of the plane wall is determined from

$$\theta_{o,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$\theta_{o,wall} = A_1 e^{-\lambda_1^2 \tau}$$

$$\theta_{o,wall} = (1.2728) e^{-(1.5421)^2 (0.015)}$$

$$\theta_{o,wall} \cong 1$$

We repeat the same calculations for the long cylinder,

$$Bi = \frac{hr_o}{k}$$

$$Bi = \frac{(120\ \text{Btu/h.ft}^2\text{.°F})(0.4/12\ \text{ft})}{(0.44\ \text{Btu/h.ft.°F})}$$

$$Bi = 9.1$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 2.1589$ and $A_1 = 1.5618$

Step 5 of 10

$$\tau = \frac{\alpha t}{r_o^2}$$

$$\tau = \frac{(0.0077\ \text{ft}^2/\text{h})(5/60\ \text{h})}{(0.4/12\ \text{ft})^2}$$

$$\tau = 0.578 > 0.2$$

$$\theta_{o,cyl} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$\theta_{o,cyl} = A_1 e^{-\lambda_1^2 \tau}$$

$$\theta_{o,cyl} = (1.5618) e^{-(2.1589)^2 (0.578)}$$

$$\theta_{o,cyl} = 0.106$$

Then the center temperature of the short cylinder becomes

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty}\right]_{\substack{\text{short} \\ \text{cylinder}}} = \theta_{o,wall} \times \theta_{o,cyl}$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty}\right]_{\substack{\text{short} \\ \text{cylinder}}} = 1 \times 0.106$$

$$\left[\frac{T(0,0,t) - T_\infty}{T_i - T_\infty}\right]_{\substack{\text{short} \\ \text{cylinder}}} = 0.106$$

$$\frac{T(0,0,t) - 212}{40 - 212} = 0.106$$

$$\boxed{T(0,0,t) = 194\text{°F}}$$

Step 6 of 10

$$\frac{T(0,0,t)-212}{40-212}=0.100$$

$\boxed{T(0,0,t)=194°F}$

Step 6 of 10

After 10 minutes

$$\tau=\frac{\alpha t}{L^2}$$

$$\tau=\frac{(0.0077\ \text{ft}^2/\text{h})(10/60\ \text{h})}{(2.5/12\ \text{ft})^2}$$

$$\tau=0.03<0.2 \quad \text{(Be cautious!)}$$

$$\theta_{o,wall}=\frac{T_0-T_\infty}{T_i-T_\infty}$$

$$\theta_{o,wall}=A_1e^{-\lambda_1^2\tau}$$

$$\theta_{o,wall}=(1.2728)e^{-(1.5421)^2(0.03)}$$

$$\theta_{o,wall}\cong 1$$

Step 7 of 10

$$\tau=\frac{\alpha t}{r_o^2}$$

$$\tau=\frac{(0.0077\ \text{ft}^2/\text{h})(10/60\ \text{h})}{(0.4/12\ \text{ft})^2}$$

$$\tau=1.156>0.2$$

$$\theta_{o,cyl}=\frac{T_0-T_\infty}{T_i-T_\infty}$$

$$\theta_{o,cyl}=A_1e^{-\lambda_1^2\tau}$$

$$\theta_{o,cyl}=(1.5618)e^{-(2.1589)^2(1.156)}$$

$$\theta_{o,cyl}=0.007$$

$$\left[\frac{T(0,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{short}\\\text{cylinder}}}=\theta_{o,wall}\times\theta_{o,cyl}$$

$$\left[\frac{T(0,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{short}\\\text{cylinder}}}=1\times 0.007$$

$$\left[\frac{T(0,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{short}\\\text{cylinder}}}=0.007$$

$$\frac{T(0,0,t)-212}{40-212}=0.007$$

$\boxed{T(0,0,t)=211°F}$

Step 8 of 10

After 15 minutes

$$\tau=\frac{\alpha t}{L^2}$$

$$\tau=\frac{(0.0077\ \text{ft}^2/\text{h})(15/60\ \text{h})}{(2.5/12\ \text{ft})^2}$$

$$\tau=0.045<0.2 \quad \text{(Be cautious!)}$$

$$\theta_{o,wall}=\frac{T_0-T_\infty}{T_i-T_\infty}$$

$$\theta_{o,wall}=A_1e^{-\lambda_1^2\tau}$$

$$\theta_{o,wall}=(1.2728)e^{-(1.5421)^2(0.045)}$$

$$\theta_{o,wall}\cong 1$$

Step 9 of 10

$$\tau=\frac{\alpha t}{r_o^2}$$

$$\tau=\frac{(0.0077\ \text{ft}^2/\text{h})(15/60\ \text{h})}{(0.4/12\ \text{ft})^2}$$

$$\tau=1.734>0.2$$

$$\theta_{o,cyl}=\frac{T_0-T_\infty}{T_i-T_\infty}$$

$$\theta_{o,cyl}=A_1e^{-\lambda_1^2\tau}$$

$$\theta_{o,cyl}=(1.5618)e^{-(2.1589)^2(1.734)}$$

$$\theta_{o,cyl}=0.0005$$

$$\left[\frac{T(0,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{short}\\\text{cylinder}}}=\theta_{o,wall}\times\theta_{o,cyl}$$

$$\left[\frac{T(0,0,t)-T_\infty}{T_i-T_\infty}\right]_{\substack{\text{short}\\\text{cylinder}}}=1\times 0.0005=0.0005$$

$$\frac{T(0,0,t)-212}{40-212}=0.0005$$

$\boxed{T(0,0,t)=212\ °F}$

Step 10 of 10

b) Treating the hot dog as an infinitely long cylinder will not change the results obtained in the part (a) since dimensionless temperatures for the plane wall is 1 for all cases.

Question: 18.089P

A 2-cm-high cylindrical ice block (k = 2.22 W/m·K and α = 0.124 × 10–7 m2 /s) is placed on a table on its base of diameter 2 cm in a room at 24°C. The heat transfer coefficient on the exposed surfaces of the ice block is 13 W/m2 ·K, and heat transfer from the base of the ice block to the table is negligible. If the ice block is not to start melting at any point for at least 3 h, determine what the initial temperature of the ice block should be.

Step 1 of 4

Given:

Height of cylinder $L = 2$ cm

Thermal conductivity of the ice $k = 2.22\ \text{W/m}\cdot{}^\circ\text{C}$

Thermal diffusivity of the ice $\alpha = 0.124\times10^{-7}\ \text{m}^2/\text{s}$

Base diameter of table $D = 2$ cm

Room temperature $T_\infty = 24^\circ\text{C}$

Heat transfer coefficient $h = 13\ \dfrac{\text{W}}{\text{m}^2\cdot{}^\circ\text{C}}$

Step 2 of 4

A cylindrical ice block is placed on a table. This cylindrical ice block can be treated as a short cylinder that can physically be formed by the intersection of a long cylinder of diameter $D = 2$ cm and an infinite plane wall of thickness $2L = 4$ cm. We measure x from the bottom surface of the block since this surface represents the adiabatic center surface of the plane wall of thickness $2L = 4$ cm. The melting starts at the outer surfaces of the top surface when the temperature drops below 0°C at this location. The Biot numbers, the corresponding constants, and the Fourier numbers are

$$Bi_{wall} = \frac{hL}{k}$$

$$Bi_{wall} = \frac{(13\ \text{W/m}^2.^\circ\text{C})(0.02\ \text{m})}{(2.22\ \text{W/m.}^\circ\text{C})}$$

$$Bi_{wall} = 0.1171$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$$\lambda_1 = 0.3318 \quad \text{and} \quad A_1 = 1.0187$$

Step 3 of 4

$$Bi_{cyl} = \frac{hr_o}{k}$$

$$Bi_{cyl} = \frac{(13\ \text{W/m}^2.^\circ\text{C})(0.01\ \text{m})}{(2.22\ \text{W/m.}^\circ\text{C})}$$

$$Bi_{cyl} = 0.05856$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$$\lambda_1 = 0.3407 \quad \text{and} \quad A_1 = 1.0146$$

$$\tau_{wall} = \frac{\alpha t}{L^2}$$

$$\tau_{wall} = \frac{(0.124\times10^{-7}\ \text{m}^2/\text{s})(2\ \text{h}\times3600\ \text{s/h})}{(0.02\ \text{m})^2}$$

$$\tau_{wall} = 0.2232 > 0.2$$

$$\tau_{cyl} = \frac{\alpha t}{r_o^2}$$

$$\tau_{cyl} = \frac{(0.124\times10^{-7}\ \text{m}^2/\text{s})(2\ \text{h}\times3600\ \text{s/h})}{(0.01\ \text{m})^2}$$

$$\tau_{cyl} = 0.8928 > 0.2$$

Step 4 of 4

Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable. The product solution for this problem can be written as

$$\theta(L, r_o, t)_{block} = \theta(L, t)_{wall}\,\theta(r_o, t)_{cyl}$$

$$\frac{0-20}{T_i-20} = \left[A_1 e^{-\lambda_1^2\tau}\cos(\lambda_1 L/L)\right]\left[A_1 e^{-\lambda_1^2\tau} J_0(\lambda_1 r_o/r_o)\right]$$

$$\frac{0-20}{T_i-20} = \left[(1.0187)e^{-(0.3318)^2(0.2232)}\cos(0.3318)\right]\left[(1.0146)e^{-(0.3407)^2(0.8928)}(0.9707)\right]$$

which gives $\boxed{T_i = -4^\circ\text{C}}$

Therefore, the ice will not start melting for at least 2 hours if its initial temperature is -4°C or below.

Question: 18.090P pg 1

A 30-cm-long cylindrical aluminum block (ρ = 2702 kg/m3 , cp = 0.896 kJ/kg·K. k = 236 W/m·K, and α = 9.75 × 10–5 m2 /s), 15 cm in diameter, is initially at a uniform temperature of 20°C. The block is to be heated in a furnace at 1200o C until its center temperature rises to 300°C. If the heat transfer coefficient on all surfaces of the block is 80 W/m2 ·K. determine how long the block should be kept in the furnace. Also, determine the amount of heat transfer from the aluminum block if it is allowed to cool in the room until its temperature drops to 20°C throughout.

Step 1 of 9

Given data:

Length of the cylindrical aluminum block, $2L = 30$ cm

Diameter of the cylindrical block, $D = 15$ cm

The density of the block, $\rho = 2702\ \text{kg/m}^3$

Thermal Conductivity of the block, $k = 236\ \text{W/m·K}$

Thermal Diffusivity, $\alpha = 9.75\times10^{-5}\ \text{m}^2/\text{s}$

Specific heat $c_p = 0.896\ \text{kJ/kg-K}$

Heat transfer coefficient, $h = 80\ \text{W/m}^2\text{·K}$

Initial temperature of the block $T_i = 20\ ^\circ\text{C}$

Furnace temperature, $T_\infty = 1200^\circ\text{C}$

Centre temperature, $T = 300^\circ\text{C}$

Step 2 of 9

This cylindrical block can physically be formed by the intersection of a long cylinder of radius

$r_o = D/2 = 7.5\text{cm}$

Calculation of Biot number, for the plain wall

And a plane wall of thickness

$2L = 30$ cm

$L = 15$ cm

The Biot number and the corresponding constants for the plane wall along the length is

$$\text{Bi} = \frac{hL}{k}$$

$$\text{Bi} = \frac{(80)(0.15)}{236}$$

$$\text{Bi} = 0.0508$$

Step 3 of 9

From the table transient one dimensional heat conduction in cylinders, the constants corresponding to this Biot number are determined as follow ,
By interpolating the values of at $\text{Bi} = 0.04$ **and** $\text{Bi} = 0.06$ **we get,**

$$\lambda_1 = \lambda_{0.0508} = \lambda_{0.04} + (\lambda_{0.06} - \lambda_{0.04})\left(\frac{0.0508-0.04}{0.06-0.04}\right)$$
$$= 0.1987 + [(0.2425-0.1987)\times 0.54]$$
$$= 0.1987 + 0.023652$$
$$= 0.222$$

$$A_1 = A_{0.0508} = A_{0.04} + (A_{0.06} - A_{0.04})\left(\frac{0.0508-0.04}{0.06-0.04}\right)$$
$$= 1.0066 + [(1.0098-1.0066)\times 0.54]$$
$$= 1.0066 + 0.00172$$
$$= 1.00832$$

So, the dimensionless temperature for approximated plane wall is obtained as:

$$\theta_{0,wall} = \frac{T-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}$$

$$\frac{T-T_\infty}{T_i-T_\infty} = 1.00832\times e^{-0.222^2\times\tau}$$

Step 4 of 9

The Biot number and the corresponding constants for the long cylinder along the radial direction is

$$\text{Bi} = \frac{hr_0}{k}$$

$$\text{Bi} = \frac{(80\ \text{W/m}^2.^\circ\text{C})(0.075\ \text{m})}{236\ \text{W/m.}^\circ\text{C}}$$

$$\text{Bi} = 0.0254$$

From the table transient one dimensional heat conduction in cylinders, the constants corresponding to this Biot number are determined as follow ,
By interpolating the values of at $\text{Bi} = 0.02$ **and** $\text{Bi} = 0.04$ **we get,**

$$\lambda_1 = \lambda_{0.0254} = \lambda_{0.02} + (\lambda_{0.04} - \lambda_{0.02})\left(\frac{0.0254-0.02}{0.04-0.02}\right)$$
$$= 0.1995 + [(0.2814-0.1995)\times 0.27]$$
$$= 0.1995 + 0.022113$$
$$= 0.222$$

$$A_1 = A_{0.0254} = A_{0.02} + (A_{0.04} - A_{0.02})\left(\frac{0.0254-0.02}{0.04-0.02}\right)$$
$$= 1.005 + [(1.0099-1.005)\times 0.27]$$
$$= 1.005 + 0.001323$$
$$= 1.006323$$

Step 5 of 9

So, the dimensionless temperature is determined to be:

$$\theta_{0,cyl} = \frac{T-T_\infty}{T_i-T_\infty} = Ae^{-\lambda^2\tau}$$

$$\frac{T-T_\infty}{T_i-T_\infty} = 1.0063\times e^{-0.222^2\times\tau}$$

Finally, combined non-dimensional temperature is:

$$\frac{T-T_\infty}{T_i-T_\infty}=1.0063\times e^{-0.222^2\times\tau}$$ 18.090P pg 2

Finally, combined non-dimensional temperature is:

$$\frac{T-T_\infty}{T_i-T_\infty}=\left[1.0084\times e^{-0.222^2\times\tau}\right]\times\left[1.0063\times e^{-0.222^2\times\tau}\right]$$

$$\frac{300-1200}{20-1200}=\left[1.0084\times e^{-0.222^2\times\tau}\right]\times\left[1.0041\times e^{-0.156^2\times\tau}\right]$$

$$0.763=\left[1.0147\times e^{-0.0985\times\tau}\right]$$

$$e^{-0.0985\times\tau}=0.752$$

$$0.0985\tau=0.285$$

$$\tau=2.893$$

Step 6 of 9

Now, Fourier number is obtained to be:

$$\tau=\frac{\alpha t}{L^2}$$

$$2.89=\frac{\left(9.75\times10^{-5}\right)t}{\left(\frac{15}{100}\right)^2}$$

$$t=666.923\text{ sec}$$

$$\boxed{t=11.11\text{ min.}}$$

So the required time for heating is 11.11 min.

Now, Maximum heat transfer from the block is obtained as:

$$Q=mc_p(20-1200)$$

Where mass of cylinder is:

$$m=\rho\times V$$

$$=\rho\times\pi r^2L$$

$$=2702\times\pi\times0.075^2\times\frac{30}{100}$$

$$m=14.32\text{ kg}$$

Step 7 of 9

Substituting known values in heat transfer

$$Q=mc_p(1200-20)$$

$$Q=14.32\times0.896\times(1200-20)$$

$$Q=15140.24\text{kJ}$$

Using the one term solutions the fractional heat transfer in plane wall is

$$\left(\frac{Q}{Q_{max}}\right)_{wall}=1-\theta_{0,wall}\frac{\sin\lambda_1}{\lambda_1}$$

$$\left(\frac{Q}{15140.24}\right)_W=1-\left[\left(1.00832\times e^{-0.222^2\times3.843}\right)\times\frac{\sin(0.222)}{0.222}\right]$$

$$\left(\frac{Q}{15140.24}\right)_W=1-0.01456$$

$$\left(\frac{Q}{15140.24}\right)_W=0.9854$$

Step 8 of 9

And for cylinder:

$$\left(\frac{Q}{Q_{max}}\right)_{cyl}=1-2\theta_{0,cyl}\frac{J_1(\lambda_1)}{\lambda_1}$$

$$\left(\frac{Q}{15140.24}\right)_{cyl}=1-\left(2\times1.0063\times e^{-0.222^2\times\tau}\times\frac{J_1(0.222)}{0.222}\right)$$

From first order Bessel functions table, by interpolating the values of $J_1(\lambda_1)$ at $\lambda_1=0.2$ and $\lambda_1=0.3$ we get the value at $\lambda_1=0.222$

$$J_1=J_{0.222}=J_{0.2}+(J_{0.3}-J_{0.2})\left(\frac{0.222-0.2}{0.3-0.2}\right)$$

$$=0.0995+\left[(0.1483-0.0995)\times0.22\right]$$

$$=0.0995+0.010736$$

$$=0.11$$

Step 9 of 9

Now, the fractional heat transfer in cylinder is,

$$\left(\frac{Q}{15140.24}\right)_{cyl}=1-2\times1.0063\times e^{-0.222^2\times2.893}\times\frac{0.11}{0.222}$$

$$\left(\frac{Q}{15140.24}\right)_{cyl}=0.135$$

So total heat transfer for block is:

$$\left(\frac{Q}{15140.24}\right)=\left(\frac{Q}{15140.24}\right)_{wall}+\left(\frac{Q}{15140.24}\right)_{cyl}\times\left[1-\left(\frac{Q}{15140.24}\right)_{wall}\right]$$

$$\left(\frac{Q}{15140.24}\right)=0.9847+0.135\times\left[1-0.9847\right]$$

$$\boxed{Q=14939.86\text{kJ}}$$

Finally, total heat transfer for block is obtained as, 14939.86 kJ

Question: 18.091P pg 1

Repeal Prob. 18-90 for the case where the aluminum block is inserted into the furnace on a low-conductivity material so that the heat transfer to or from the bottom surface of the block is negligible.

Step 1 of 10

Given data:

Length of the cylindrical aluminum block $L=30$ cm

Diameter of the cylindrical block, $D=15$ cm

The density of the block, $\rho=2702\ \text{kg/m}^3$

Thermal Conductivity of the block, $k=236\ \text{W/m}\cdot\text{K}$

Thermal Diffusivity, $\alpha=9.75\times10^{-5}\ \text{m}^2/\text{s}$

Specific heat $c_p=0.896\ \text{kJ/kg-K}$

Heat transfer coefficient, $h=80\ \text{W/m}^2\cdot\text{K}$

Initial temperature of the block $T_i=20\ ^\circ\text{C}$

Furnace temperature, $T_\infty=1200^\circ\text{C}$

Centre temperature, $T=300^\circ\text{C}$

Step 2 of 10

For this problem Biot number

$$\text{Bi}=\frac{hL_c}{k}$$

$$\text{Bi}=\frac{80\times L_c}{236}$$

Now, characteristic length can be obtained by approximating a plane wall having semi thickness: $L_c=30$ cm

Putting this into Biot number expression, we get:

$$\text{Bi}=\frac{hL_c}{K}$$

$$\text{Bi}=\frac{80\times L_c}{236}$$

$$\text{Bi}=\frac{80\times\left(\frac{30}{100}\right)}{236}$$

$$\text{Bi}=0.101$$

From the table transient one dimensional heat conduction in cylinders, the constants corresponding to this Biot number are determined as follow ,

$\lambda=0.3111, A=1.0161$

So, the dimensionless temperature for approximated plane wall is obtained as:

$$\frac{T-T_\infty}{T_i-T_\infty}=Ae^{-\lambda^2\tau}$$

$$\frac{T-T_\infty}{T_i-T_\infty}=1.0161\times e^{-0.3111^2\times\tau}$$

Step 3 of 10

Biot number for long cylinder:

$$\text{Bi}=\frac{hL_c}{K}$$

$$\text{Bi}=\frac{80\times L_c}{236}$$

$$\text{Bi}=\frac{80\times\left(\frac{7.5}{100}\right)}{236}$$

$$\text{Bi}=0.0254$$

Step 4 of 10

From the table transient one dimensional heat conduction in cylinders, the constants corresponding to this Biot number are determined as follow ,

By interpolating the values of at $\text{Bi}=0.02$ and $\text{Bi}=0.04$ we get,

$$\lambda_1=\lambda_{0.0254}=\lambda_{0.02}+(\lambda_{0.04}-\lambda_{0.02})\left(\frac{0.0254-0.02}{0.04-0.02}\right)$$

$$=0.1995+\left[(0.2814-0.1995)\times0.27\right]$$

$$=0.1995+0.022113$$

$$=0.222$$

$$A_1=A_{0.0254}=A_{0.02}+(A_{0.04}-A_{0.02})\left(\frac{0.0254-0.02}{0.04-0.02}\right)$$

$$=1.005+\left[(1.0099-1.005)\times0.27\right]$$

$$=1.005+0.001323$$

$$=1.006323$$

So, the dimensionless temperature is determined to be:

$$\theta_{0,cyl}=\frac{T-T_\alpha}{T_i-T_\alpha}=Ae^{-\lambda^2\tau}$$

$$\frac{T-T_\alpha}{T_i-T_\alpha}=1.0063\times e^{-0.222^2\times\tau}$$

Step 5 of 10

Finally, combined non-dimensional temperature is:

$$\frac{T-T_\alpha}{T_i-T_\alpha}=\left[1.0161\times e^{-0.3111^2\times\tau}\right]\times\left[1.0063\times e^{-0.222^2\times\tau}\right]$$

$$\frac{300-1200}{20-1200}=\left[1.0161\times e^{-0.3111^2\times\tau}\right]\times\left[1.0063\times e^{-0.222^2\times\tau}\right]$$

$$0.763=\left[1.0225\times e^{-0.146\times\tau}\right]$$

$$e^{-0.146\times\tau}=0.75$$

$$\tau=1.97$$

Step 6 of 10

Fourier number is obtained to be:

$$\tau=\frac{\alpha t}{L^2}$$

$$=\frac{\left(0.75\times10^{-5}\right)t}{}$$

Step 6 of 10

Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L^2}$$

$$1.97 = \frac{\left(9.75\times10^{-5}\right)t}{\left(\frac{30}{100}\right)^2}$$

$$t = 1818.46 \text{ sec}$$

$$\boxed{t = 30.30 \text{ min.}}$$

So the required time for heating is 30.30 min.

Step 7 of 10

Now, Maximum heat transfer from the block is obtained as:

$$Q = mc_p(20-1200)$$

Where mass of cylinder is:

$$m = \rho \times V$$

$$= \rho \times \pi r^2 L$$

$$= 2702 \times \pi \times 0.075^2 \times \frac{30}{100}$$

$$m = 14.32 \text{ kg}$$

Step 8 of 10

Substituting known values in heat transfer

$$Q = mc_p(1200-20)$$

$$Q = 14.32\times0.896\times(1200-20)$$

$$Q = 15140.24 \text{kJ}$$

Using the one term solutions the fractional heat transfer in plane wall is

$$\left(\frac{Q}{15140.24}\right)_W = 1-1.0161\times e^{-0.3111^2\times1.97}\times\frac{\sin(0.3111)}{0.3111}$$

$$\left(\frac{Q}{15140.24}\right)_W = 1-0.0146$$

$$\left(\frac{Q}{15140.24}\right)_W = 0.985$$

Step 9 of 10

And for cylinder:

$$\left(\frac{Q}{Q_{max}}\right)_{cyl} = 1-2\theta_{0,cyl}\frac{J_1(\lambda_1)}{\lambda_1}$$

$$\left(\frac{Q}{15140.24}\right)_{cyl} = 1-2\times1.0063\times e^{-0.222^2\times\tau}\times\frac{J_1(0.222)}{0.222}$$

From first order Bessel functions table, by interpolating the values of $J_1(\lambda_1)$ at $\lambda_1 = 0.2$ and $\lambda_1 = 0.3$ we get the value at $\lambda_1 = 0.222$ as,

$$J_1 = J_{0.222} = J_{0.2} + (J_{0.3} - J_{0.2})\left(\frac{0.222-0.2}{0.3-0.2}\right)$$

$$= 0.0995 + \left[(0.1483-0.0995)\times0.22\right]$$

$$= 0.0995 + 0.010736$$

$$= 0.11$$

Then the fractional heat transfer in cylinder is

$$\left(\frac{Q}{15140.24}\right)_{cyl} = 1-2\times1.0063\times e^{-0.222^2\times30.30}\times\frac{0.11}{0.222}$$

$$\left(\frac{Q}{15140.24}\right)_{cyl} = 0.775$$

Step 10 of 10

So total heat transfer for block is:

$$\left(\frac{Q}{15140.24}\right) = \left(\frac{Q}{15140.24}\right)_W + \left(\frac{Q}{15140.24}\right)_C \times \left[1-\left(\frac{Q}{15140.24}\right)_W\right]$$

$$\left(\frac{Q}{15140}\right) = 0.999 + 0.775\times[1-0.999]$$

$$\boxed{Q = 15136.83 \text{kJ}}$$

Finally, total heat transfer for block is obtained as 15136.83kJ

Question: 18.092P pg 1



Reconsider Prob. 18-90. Using EES (or other) MM software, investigate the effect of the final center temperature of the block on the heating time and the amount of heat transfer. Let the final center temperature vary from 50°C to 1000°C. Plot the time and the heat transfer as a function of the final center temperature, and discuss the results.

Step 1 of 9

Given:
Length of the cylindrical aluminum block $L = 30$ cm
Diameter of the cylindrical block $D = 15$ cm
The density of the block $\rho = 2702\ kg/m^3$
Thermal Conductivity, $k = 236\ W/m \cdot °C$
Thermal Diffusivity, $\alpha = 9.75 \times 10^{-5}\ m^2/s$
Specific heat $C_p = 0.896\ kJ/kg \cdot K$
Heat transfer coefficient $h = 80\ W/m^2 \cdot °C$

Step 2 of 9

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
2*L=0.30 "[m]"
2*r_o=0.15 "[m]"
T_i=20 "[C]"
T_infinity=1200 "[C]"
"T_o_o=300 [C], parameter to be varied"
h=80 "[W/m^2-C]"
```

Step 3 of 9

```
"PROPERTIES"
k=236 "[W/m-C]"
rho=2702 "[kg/m^3]"
C_p=0.896 "[kJ/kg-C]"
alpha=9.75E-5 "[m^2/s]"
```

Step 4 of 9

```
"ANALYSIS"
"This short cylinder can physically be formed by the intersection of a long cylinder of
radius r_o and a plane wall of thickness 2L"
"For plane wall"
Bi_w=(h*L)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1_w=0.1439 "w stands for wall"
A_1_w=1.0035
tau_w=(alpha*time)/L^2
theta_o_w=A_1_w*exp(-lambda_1_w^2*tau_w) "theta_o_w=(T_o_w-T_infinity)/(T_i-
T_infinity)"
```

Step 5 of 9

```
"For long cylinder"
Bi_c=(h*r_o)/k "c stands for cylinder"
"From biot number Table corresponding to this Bi number, we read"
lambda_1_c=0.1762
A_1_c=1.0040
tau_c=(alpha*time)/r_o^2
theta_o_c=A_1_c*exp(-lambda_1_c^2*tau_c) "theta_o_c=(T_o_c-T_infinity)/(T_i-
T_infinity)"
(T_o_o-T_infinity)/(T_i-T_infinity)=theta_o_w*theta_o_c "center temperature of
cylinder"
V=pi*r_o^2*(2*L)
```

Step 6 of 9

```
m=rho*V
Q_max=m*C_p*(T_infinity-T_i)
Q_w=1-theta_o_w*Sin(lambda_1_w)/lambda_1_w "Q_w=(Q/Q_max)_w"
Q_c=1-2*theta_o_c*J_1/lambda_1_c "Q_c=(Q/Q_max)_c"
J_1=0.0876 "From Table at lambda_1_c"
Q/Q_max=Q_w+Q_c*(1-Q_w) "total heat transfer"
```

Step 7 of 9

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 20
Select variables in equations as $T_{o,o}$ and time press ADD and then press ok
A table will appear put values of $T_{o,o}$ as 50,100,150....1000 in run 1 to 20
Repeat procedure for $T_{o,o}$ and Q
Then from Calculate -> Solve table

Solution appear

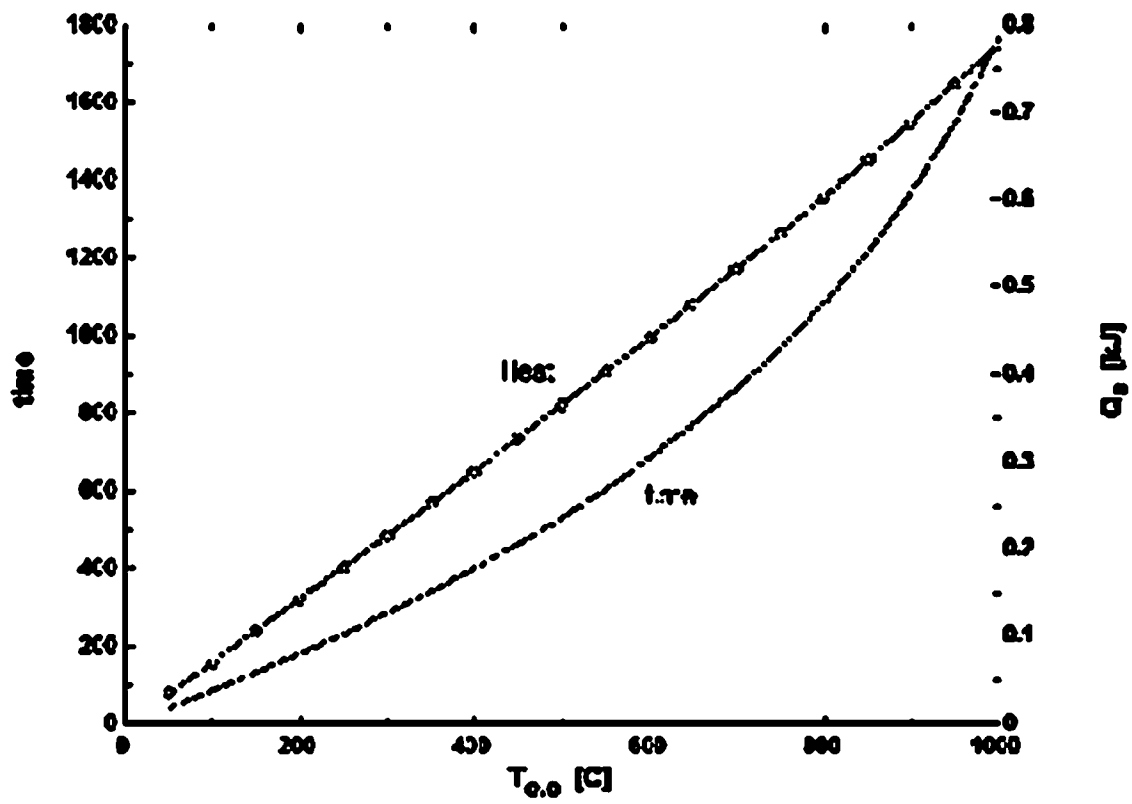
Solution appear

Step 8 of 9

▷ 1..20	1 $T_{o,o}$ [C]	2 time	3 Q_c [kJ]
Run 1	50	42.43	0.036
Run 2	100	86.33	0.07138
Run 3	150	132.3	0.107
Run 4	200	180.4	0.1429
Run 5	250	231.1	0.1791
Run 6	300	284.5	0.2156
Run 7	350	340.9	0.2525
Run 8	400	400.8	0.2896
Run 9	450	464.5	0.3272
Run 10	500	532.6	0.3651
Run 11	550	605.8	0.4035
Run 12	600	684.9	0.4423
Run 13	650	770.8	0.4817
Run 14	700	864.9	0.5216
Run 15	750	968.9	0.5622
Run 16	800	1085	0.6035
Run 17	850	1217	0.6456
Run 18	900	1369	0.6887
Run 19	950	1549	0.733
Run 20	1000	1770	0.7787

Step 9 of 9

Then from Plot -> New plot window -> X-Y Plot
Select X-axis Vel and Y-axis T_s
Repeat procedure for $T_{o,o}$ and Q
Press OK
Graph will be plotted.



Question: 18.093P

Spherical glass beads coming out of a kiln are allowed to cool in a room temperature of 30°C. A glass bead with a diameter of 10 mm and an initial temperature of 400°C is allowed to cool for 3 minutes. If the convection heat transfer coefficient is 28 W/m2 ·K. determine the temperature at the center of the glass bead using (*a*) Table 18-2 and (*b*) the Heisler chart (Figure 18-17). The glass bead has properties of ρ = 2800 kg/m3 , *cp* = 750 J/kg·K, and *k* = 0.7 W/m·K.

Step 1 of 7

Calculate the Biot number for the heating process of glass by using the relation,

$$\mathrm{Bi}=\frac{hL_c}{k} \quad \text{..... (1)}$$

Here, convective heat transfer coefficient is h , characteristic length of the glass is L_c , and conduction coefficient of the glass is k .

Calculate the characteristic length of the glass by using the relation,

$$L_c = r_o$$

Here, radius of the glass is r_o .

Substitute $0.005\ \mathrm{m}$ for r_o .

$$L_c = 0.005\ \mathrm{m}$$

Step 2 of 7

Calculate the Biot number for this process as shown.

Substitute $0.005\ \mathrm{m}$ for L_c , $28\ \mathrm{W/m^2 \cdot K}$ for h , and $0.7\ \mathrm{W/m \cdot K}$ for k in equation (1).

$$\mathrm{Bi}=\frac{28\times 0.005}{0.7}$$
$$=0.2$$

Write the formula for thermal diffusivity as shown.

$$\alpha=\frac{k}{\rho C_p}$$

Step 3 of 7

Calculate the Fourier number for this process by using the formula,

$$\tau=\frac{\alpha t}{r_o^2}$$

Here, diffusivity of glass is α , and time for which the glass is heated is t .

Substitute $\frac{k}{\rho C_p}$ for α .

$$\tau=\frac{\left(\frac{k}{\rho C_p}\right)t}{r_o^2} \quad \text{..... (2)}$$
$$=\frac{kt}{(\rho C_p)r_o^2}$$

Step 4 of 7

Calculate the Fourier number for this process as shown.

Substitute $0.7\ \mathrm{W/m \cdot K}$ for k , $(3\times 60)\ \mathrm{s}$ for t , $2800\ \mathrm{kg/m^3}$ for ρ , $750\ \mathrm{J/kg \cdot K}$ for C_p and $0.005\ \mathrm{m}$ for r_o in equation (2).

$$\tau=\frac{0.7\times 3\times 60}{(2800\times 750)(0.005)^2}$$
$$=\frac{126}{52.5}$$
$$=2.4$$

Step 5 of 7

(a).

Calculate the dimensionless temperature of a spherical glass bead at its centre by using the relation,

$$\theta_{0,sph}=A_1 e^{-\lambda_1^2 \tau} \quad \text{..... (3)}$$

Obtain the values of λ_1 and A_1 for $\mathrm{Bi}=0.2$ from table 4.2 as shown.

$$\lambda_1=0.7593$$
$$A_1=1.0592$$

Substitute 2.4 for τ , 1.0592 for A_1 , and 0.7593 for λ_1 in equation (2).

$$\theta_{0,sph}=(1.0592)e^{-\left[(0.7593)^2\times 2.4\right]}$$
$$=0.2655$$

Step 6 of 7

Calculate the temperature at the centre of the glass bead by using the relation,

$$\frac{T_0-T_\infty}{T_i-T_\infty}=\theta_{0,cyl}$$

Here, the temperature at the centre of the glass is T_0 , surroundings temperature is T_∞ , and initial temperature of the glass is T_i .

Substitute 0.2655 for $\theta_{0,cyl}$, $30\ ^\circ\mathrm{C}$ for T_∞ , and $400\ ^\circ\mathrm{C}$ for T_i .

$$\frac{T_0-30}{400-30}=0.2655$$
$$T_0-30=370\times 0.2655$$
$$T_0=30+98.23$$
$$=128.23\ ^\circ\mathrm{C}$$

Therefore, the temperature at the centre of the glass bead is $\boxed{128.23\ ^\circ\mathrm{C}}$.

Step 7 of 7

(b).

Calculate the value $\frac{1}{\mathrm{Bi}}$ as shown.

$$\frac{1}{\mathrm{Bi}}=\frac{1}{0.2}$$
$$=5$$

Obtain the value of dimensionless temperature at the centre, θ_0 corresponding to values $\frac{1}{\mathrm{Bi}}=5$ and $\tau=2.4$ from figure 4-17 as shown.

$$\theta_0=0.27$$

Calculate the temperature at the centre of the glass bead as shown.

$$\frac{T_0-T_\infty}{T_i-T_\infty}=\theta_{0,sph}$$

Substitute 0.27 for $\theta_{0,cyl}$, $30\ ^\circ\mathrm{C}$ for T_∞ , and $400\ ^\circ\mathrm{C}$ for T_i .

$$\frac{T_0-30}{400-30}=0.27$$
$$T_0-30=370\times 0.27$$
$$T_0=30+99.9$$
$$\approx 130\ ^\circ\mathrm{C}$$

Therefore, the temperature at the centre of the glass bead is $\boxed{130\ ^\circ\mathrm{C}}$.

Question: 18.094P

A large iron slab (ρ = 7870 kg/m3 , Cp = 447 J/kg·K. and k = 80.2 W/m·K) was initially heated to a uniform temperature of 150°C and then placed on concrete floor (ρ = 1600 kg/m3 , cp = 840 J/kg·K, and k = 0.79 W/m·K). concrete floor was initially at a uniform temperature of 30°C. Determine (*a*) The surface temperature between the iron slab and concrete floor and (*b*) The temperature of the concrete floor at the depth of 25 mm, if the surface temperature remains constant after 15 minutes.

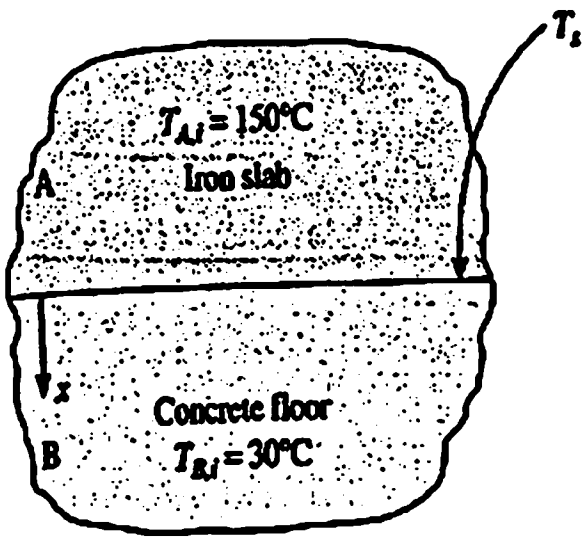


FIGURE P18–94

Step 1 of 2

(a) The surface temperature between the contact of two semi-infinite solids of different initial temperatures,

$$T_s = \frac{\sqrt{(k\rho c_p)_A}\,T_{A,i} + \sqrt{(k\rho c_p)_B}\,T_{B,i}}{\sqrt{(k\rho c_p)_A} + \sqrt{(k\rho c_p)_B}}$$

Substituting the values, $\rho_A = 7870\,\text{kg/m}^3$, $c_{A,p} = 447\,\text{J/kg·K}$, $k_A = 80.2\,\text{W/m·K}$, $T_{A,i} = 150°\text{C}$, $\rho_B = 1600\,\text{kg/m}^3$, $c_{B,p} = 840\,\text{J/kg·K}$, $k_B = 0.79\,\text{W/m·K}$ and $T_{B,i} = 30°\text{C}$, we get

$$T_s = \frac{\sqrt{(80.2\times7870\times447)}\times(150+273) + \sqrt{(0.79\times1600\times840)}\times(30+273)}{\sqrt{(80.2\times7870\times447)} + \sqrt{(0.79\times1600\times840)}}$$

$$= \frac{7105075.207 + 312216.469}{16796.86 + 1030.41}$$

$$\boxed{T_s = 416.06\,\text{K}}$$

Therefore, the surface temperature between the contacts of two semi-infinite solids of different initial temperatures is 416.06K

Step 2 of 2

(b) The temperature of the concrete floor at the depth of 25 mm

Thermal diffusivity of the concrete slab,

$$\alpha_B = \left(\frac{k}{\rho c_p}\right)_B$$

Substitute $0.79\,\text{W/m·K}$ for k_B, $1600\,\text{kg/m}^3$ for ρ_B and $840\,\text{J/kg·K}$ for $c_{B,p}$

$$\alpha_B = \left(\frac{k}{\rho c_p}\right)_B$$

$$= \left(\frac{0.79}{1600\times840}\right)$$

$$= 5.87\times10^{-7}\,\text{m}^2/\text{sec}$$

Substituting the values in below equation,

$$\frac{T(x,t) - T_i}{T_s - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

$$\frac{T(x,t) - (30+273)}{416.06 - 303} = \text{erfc}\left(\frac{25\times10^{-3}}{2\sqrt{(5.87\times10^{-7})(15\times60)}}\right)$$

$$\frac{T(x,t) - 303}{113.06} = \text{erfc}(0.5434)$$

By solving the above equation,

$$\boxed{T(x,t) = 352.995\,\text{K}}$$

Therefore, the temperature of the concrete floor at the depth of 25 mm is 352.995K

Question: 18.095P

In desert climates, rainfall is not a common occurrence since the rain droplets formed in the upper layer of the atmosphere often evaporate before they reach the ground. Consider a raindrop that is initially at a temperature of 5°C and has a diameter of 5 mm. Determine how long it will lake for the diameter of the raindrop to reduce to 3 mm as it falls through ambient air at 18°C with a heat transfer coefficient of 400 W/m2 ·K. The water temperature can be assumed to remain constant and uniform at 5°C at all times.

Step 1 of 5

Given data:

Initially temperature of the raindrop, $T_i = 5°C$

Initially Diameter of rain drop, $D_i = 5\,mm = 0.005\,m$

Final diameter of the rain drop, $D_i = 3mm = 0.003m$

Temperature of the air, $T_\infty = 18°C$

Step 2 of 5

From the properties of the various fluids

For water

Density, $\rho = 1000\,kg/m^3$

Latent heat vaporization, $h_{fg} = 2490\,kJ/kg$

Initially mass of the rain drop is

$$m_i = \rho V_i$$

$$= \rho\left(\frac{4}{3}\pi r_i^3\right)$$

$$= 1000\left(\frac{4}{3}\pi\left(\frac{0.005}{2}\right)^3\right)$$

$$= 0.0000654\,kg$$

Final mass of the rain drop is

$$m_f = \rho V_f$$

$$= \rho\left(\frac{4}{3}\pi r_f^3\right)$$

$$= 1000\left(\frac{4}{3}\pi\left(\frac{0.003}{2}\right)^3\right)$$

$$= 0.0000141\,kg$$

Step 3 of 5

Change in mass of the rain drop is

$$m = m_i - m_f$$

$$= 0.0000654 - 0.0000141$$

$$= 0.0000513\,kg$$

Heat transfer required for evaporation is

$$Q = mh_{fg}$$

$$= 0.0000513 \times 2490$$

$$= 0.1277\,kJ$$

Step 4 of 5

Average surface area of the rain drop is

$$A_s = \frac{4\pi\left(r_i^2 + r_f^2\right)}{2}$$

$$= \frac{4\pi\left(\left(\frac{0.005}{2}\right)^2 + \left(\frac{0.003}{2}\right)^2\right)}{2}$$

$$= \frac{4\pi\left(\left(2.25\times10^{-6}\right) + \left(6.25\times10^{-6}\right)\right)}{2}$$

$$= 5.340\times10^{-5}\,m^2$$

Rate of heat transfer due to convention is

$$Q_c = hA(T_i - T_\infty)$$

$$= 400 \times 5.341\times10^{-5}(18-5)$$

$$= 0.277\,J/s$$

Step 5 of 5

Then, time required to reduce the size of the rain drop is

$$\dot{Q} = \frac{Q}{\Delta t}$$

$$\Delta t = \frac{Q}{\dot{Q}}$$

$$= \frac{127.77\,J}{0.277\,J/s}$$

$$= 461s$$

$$= 7.68\,min$$

Therefore, time required to reduce the size of the rain drop is $\boxed{\Delta t = 7.68\,min}$.

Question: 18.096P

The water main in the cities must be placed at sufficient depth below the earth's surface to avoid freezing during extended periods of subfreezing temperatures. Determine the minimum depth at which the water main must be placed at a location where the soil is initially at 15°C and the earth's surface temperature under the worst conditions is expected to remain at -10°C for a period of 75 days. Take the properties of soil at that location to be k = 0.7 W/m·K and α = 1.4 × 10–5 m2 /s.

Step 1 of 2

Given:

Initial temperature of soil $T_i = 15^\circ C$

Expected surface temperature under worst conditions $T_s = -10^\circ C$

Thermal Conductivity of the soil, $k = 0.7\ W/m \cdot {}^\circ C$

Thermal Diffusivity of the soil, $\alpha = 1.4 \times 10^{-5}\ m^2/s$

Step 2 of 2

The water pipes are buried in the ground to prevent freezing. The temperature in the soil is affected by the thermal conditions at one surface only, and thus the soil can be considered to be a semi-infinite medium with a specified surface temperature of -10°C.

The thermal properties of the soil are constant.

The depth at which the temperature drops to 0°C in 75 days is determined using the analytical solution,

$$\frac{T(x,t)-T_i}{T_s-T_i} = erfc\left(\frac{x}{2\sqrt{\alpha t}}\right)$$

Substituting,

$$\frac{0-15}{-10-15} = erfc\left(\frac{x}{2\sqrt{(1.4\times10^{-5}\ m^2/s)(75\ day\times24\ h/day\times3600\ s/h)}}\right)$$

$$\boxed{x=7.05\ m}$$

Therefore, the pipes must be buried at a depth of at least 7.05 m.

Question: 18.097P

A hot dog can be considered to be a 12-cm-long cylinder whose diameter is 2 cm and whose properties are ρ = 980 kg/m3 , cp = 3.9 kJ/kg·K, k = 0.76 W/m·K, and α = 2 × 10–7 m2 /s. A hot dog initially at 5°C is dropped into boiling water at 100°C. The heat transfer coefficient at the surface of the hot dog is estimated to be 600 W/m·K. If the hot dog is considered cooked when its center temperature reaches 80°C, determine how long it will take to cook it in the boiling water.

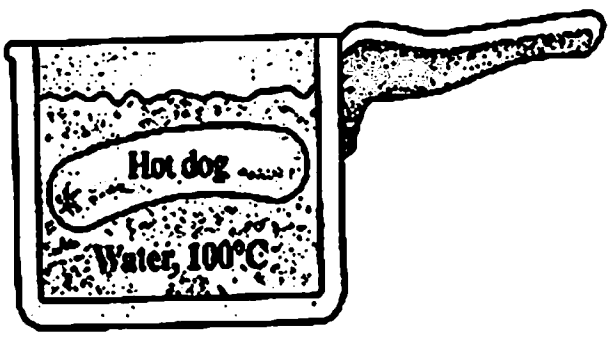


FIGURE P18–97

Step 1 of 4

Given:

Length of the cylindrical hot dog $L = 12$ cm

Diameter of the cylindrical hot dog $d = 2$ cm

Center temperature of the hot dog $T = 80°\,C$

Thermal Conductivity of the hot dog, $k = 0.76\ W/m \cdot °C$

Density of the hot dog, $\rho = 980\ \frac{kg}{m^3}$

Specific Heat, $Cp = 3.9\ \frac{kJ}{kg.\ C}$

Thermal Diffusivity of the hot dog, $\alpha = 2 \times 10^{-7}\ m^2/s$

Heat Transfer Coefficient, $h = 600\ \frac{W}{m^2 \cdot °C}$

Step 2 of 4

This hot dog can physically be formed by the intersection of an infinite plane wall of thickness $2L = 12$ cm, and a long cylinder of radius $r_o = D/2 = 1$ cm.

The Biot numbers and corresponding constants are first determined to be

$$Bi = \frac{hL}{k}$$

$$Bi = \frac{(600\ W/m^2.°C)(0.06\ m)}{(0.76\ W/m.°C)}$$

$$Bi = 47.37$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 1.5381$ and $A_1 = 1.2726$

Length of the cylindrical hot dog $L = 12$ cm

Diameter of the cylindrical hot dog $d = 2$ cm

Center temperature of the hot dog $T = 80°\,C$

Thermal Conductivity of the hot dog, $k = 0.76\ W/m \cdot °C$

Density of the hot dog, $\rho = 980\ \frac{kg}{m^3}$

Specific Heat, $Cp = 3.9\ \frac{kJ}{kg.\ C}$

Thermal Diffusivity of the hot dog, $\alpha = 2 \times 10^{-7}\ m^2/s$

Heat Transfer Coefficient, $h = 600\ \frac{W}{m^2 \cdot °C}$

Step 2 of 4

This hot dog can physically be formed by the intersection of an infinite plane wall of thickness $2L = 12$ cm, and a long cylinder of radius $r_o = D/2 = 1$ cm.

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$$Bi = 47.37$$

The constants λ_1 and A_1 corresponding to this Biot number are, from table 11-2

From interpolation method

$\lambda_1 = 1.5381$ and $A_1 = 1.2726$

Step 4 of 4

Therefore, it will take about 4.1 min for the hot dog to cook. Note that

$$\tau_{cyl} = \frac{\alpha t}{r_o^2}$$

$$\tau_{cyl} = \frac{(2 \times 10^{-7}\ m^2/s)(244\ s)}{(0.01\ m)^2}$$

$$\tau_{cyl} = 0.49 > 0.2$$

Thus the assumption $\tau > 0.2$ for the applicability of the one-term approximate solution is verified.

Question: 18.098P

A long roll of 2-m-wide and 0.5-cm-thick 1-Mn manganese steel plate coming off a furnace at 820°C is to be quenched in an oil bath (cp = 2.0 kJ/kg·K) at 185°C. The metal sheet is moving at a steady velocity of 20 m/min, and the oil bath is 9 m long. Taking the convection heat transfer coefficient on both sides of the plate to be 860 W/m2 ·K, determine the temperature of the sheet metal when it leaves the oil bath. Also, determine the required rate of heat removal from the oil to keep its temperature constant at 45°C.

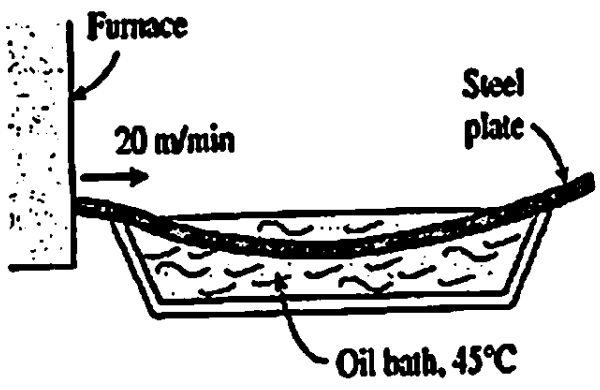


FIGURE P18–98

Question:

A long roll of 2-m-wide and 0.5-cm-thick 1-Mn manganese steel plate coming off a furnace at 820°C is to be quenched in an oil bath (cp = 2.0 kJ/kg·K) at 185°C. The metal sheet is moving at a steady velocity of 20 m/min, and the oil bath is 9 m long. Taking the convection heat transfer coefficient on both sides of the plate to be 860 W/m2 ·K, determine the temperature of the sheet metal when it leaves the oil bath. Also, determine the required rate of heat removal from the oil to keep its temperature constant at 45°C.

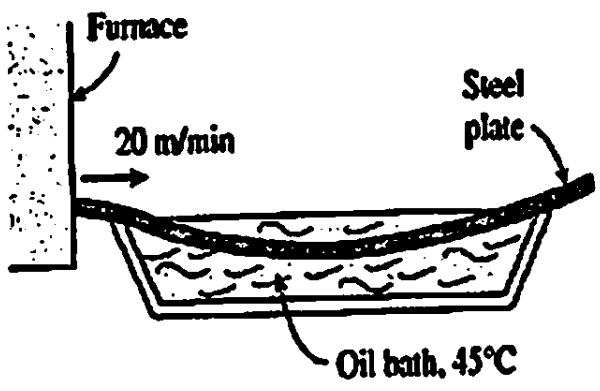


FIGURE P18–98

Step 1 of 6

Then, Biot number becomes

$$\text{Bi} = \frac{hL_c}{k}$$

$$= \frac{(860\ \text{W/m}^2.^\circ\text{C})(0.0025\ \text{m})}{60.5\ \text{W/m}.^\circ\text{C}}$$

$$= 0.036 < 0.1$$

Since Bi < 0.1, the lumped system analysis is applicable. Therefore,

$$b = \frac{hA_s}{\rho C_p V}$$

$$= \frac{h}{\rho C_p L_c}$$

$$= \frac{860\ \text{W/m}^2.^\circ\text{C}}{(7854\ \text{kg/m}^3)(434\ \text{J/kg.}^\circ\text{C})(0.0025\ \text{m})}$$

$$= 0.10092\ s^{-1}$$

Step 4 of 6

Time is

$$(t) = \frac{\text{length}}{\text{velocity}}$$

$$= \frac{9\ \text{m}}{20\ \text{m/min}}$$

$$= 0.45\ \text{min}$$

Then the temperature of the sheet metal when it leaves the oil bath is determined to be

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{T(t) - 45}{820 - 45} = e^{-(0.10092\ s^{-1})(27\ s)}$$

$$T(t) = 95.80^\circ\text{C}$$

Therefore, the temperature of the sheet metal when it leaves the oil bath is $T(t)$=95.80°C.

Step 5 of 6

The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho \dot{V}$$

$$= \rho w t V$$

$$= (7854\ \text{kg/m}^3)(2\ \text{m})(0.005\ \text{m})(20\ \text{m/min})$$

$$= 1570.8\ \text{kg/min}$$

$$= 26.173\ \text{kg/s}$$

Step 6 of 6

Then the rate of heat transfer from the sheet metal to the oil bath and thus the rate at which heat needs to be removed from the oil in order to keep its temperature constant at 45°C becomes

$$\dot{Q} = \dot{m} C_p [T(t) - T_\infty]$$

$$= (26.173\ \text{kg/s})(434\ \text{J/kg.}^\circ\text{C})(95.80 - 45)^\circ\text{C}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{T(t) - 45}{820 - 45} = e^{-(0.10092\ s^{-1})(27\ s)}$$

$$T(t) = 95.80^\circ\text{C}$$

Therefore, the temperature of the sheet metal when it leaves the oil bath is $T(t)$=95.80°C.

Step 5 of 6

Question: 18.099P

In *Betty Crocker's Cookbook*, it is stated that it takes 5 h to roast a 14-lb stuffed turkey initially at 40°F in an oven maintained at 325°F. It is recommended that a meat thermometer be used to monitor the cooking, and the turkey is considered done when the thermometer inserted deep into the thickest part of the breast or thigh without touching the bone registers 185°F. The turkey can be treated as a homogeneous spherical object with the properties ρ = lbm/ft3 , *cp* = 0.98 Btu/lbm·°F. *k* = 0.26 Btu/h·ft·°F, and *α* = 0.0035 ft2 /h. Assuming the tip of the thermometer is at one-third radial distance from the center of the turkey, determine (*a*) the average heat transfer coefficient at the surface of the turkey; (*b*) the temperature of the skin of the turkey when it is done; and (*c*) the total amount of heat transferred to the turkey in the oven. Will the reading of the thermometer be more or less than 185°F 5 min after the turkey is taken out of the oven?

Step 1 of 6

Given:

Weight of stuffed turkey $W = 14$ lb

Initial Temperature of the turkey $T_i = 40°F$

Oven Temperature $T_\infty = 325°F$

Thermal Conductivity of the turkey, k = 0.26 Btu/h.ft. °F

Density of the turkey, $\rho = 75 \frac{\text{lbm}}{\text{ft}^3}$

Specific Heat of the turkey, Cp =0.98 Btu/lbm.°F

Thermal Diffusivity of the turkey, $\alpha = 0.0035 \frac{\text{ft}^2}{\text{h}}$

Step 2 of 6

A stuffed turkey is cooked in an oven. The turkey is a homogeneous spherical object. Heat conduction in the turkey is one-dimensional because of symmetry about the midpoint. The thermal properties of the turkey are constant. Assuming the turkey to be spherical in shape, its radius is determined to be

The mass $m = \rho V$

The volume $V = \frac{m}{\rho}$

$$V = \frac{14 \text{ lbm}}{75 \text{ lbm/ft}^3}$$

$$V = 0.1867 \text{ ft}^3$$

$$V = \frac{4}{3}\pi r_o^3$$

The radius $r_o = \sqrt[3]{\frac{3V}{4\pi}}$

$$r_o = \sqrt[3]{\frac{3(0.1867 \text{ ft}^3)}{4\pi}}$$

$$r_o = 0.3545 \text{ ft}$$

The Fourier number is $\tau = \frac{\alpha t}{r_o^2}$

$$\tau = \frac{(3.5\times10^{-3} \text{ ft}^2/\text{h})(5 \text{ h})}{(0.3545 \text{ ft})^2}$$

$$\tau = 0.1392$$

Step 3 of 6

Which is close to 0.2 but a little below it. Therefore, assuming the one-term approximate solution for transient heat conduction to be applicable, the one-term solution formulation at one-third the radius from the center of the turkey can be expressed as

$$\theta(x,t)_{sph} = \frac{T(x,t) - T_\infty}{T_i - T_\infty}$$

$$\theta(x,t)_{sph} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o}$$

$$\frac{185 - 325}{40 - 325} = 0.491$$

$$0.491 = A_1 e^{-\lambda_1^2 (0.14)} \frac{\sin(0.333\lambda_1)}{0.333\lambda_1}$$

By trial and error, it is determined from Biot number table that the equation above is satisfied when $Bi = 20$ corresponding to $\lambda_1 = 2.9857$ and $A_1 = 1.9781$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k}$$

$$h = \frac{kBi}{r_o}$$

$$h = \frac{(0.26 \text{ Btu/h.ft.°F})(20)}{(0.3545 \text{ ft})}$$

$$\boxed{h = 14.7 \text{Btu/h.ft.°F}}$$

Step 4 of 6

(*b*) The temperature at the surface of the turkey is

$$\frac{T(r_o,t) - 325}{40 - 325} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\frac{T(r_o,t) - 325}{40 - 325} = (1.9781) e^{-(2.9857)^2 (0.14)} \frac{\sin(2.9857)}{2.9857}$$

$$\frac{T(r_o,t) - 325}{40 - 325} = 0.02953$$

$$\boxed{T(r_o,t) = 317 \text{ °F}}$$

Step 5 of 6

(*c*) The maximum possible heat transfer is

$$Q_{max} = mC_p(T_\infty - T_i)$$

$$Q_{max} = (14 \text{ lbm})(0.98 \text{ Btu/lbm.°F})(325 - 40)°F$$

$$Q_{max} = 3910 \text{ Btu}$$

Step 6 of 6

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{max}} = 1 - 3\theta_{o,sph} \frac{\sin(\lambda_1) - \lambda_1 \cos(\lambda_1)}{\lambda_1^3}$$

$$\frac{Q}{Q_{max}} = 1 - 3(0.491) \frac{\sin(2.9857) - (2.9857)\cos(2.9857)}{(2.9857)^3}$$

$$\frac{Q}{Q_{max}} = 0.828$$

$$Q = 0.828 Q_{max}$$

$$Q = (0.828)(3910 \text{ Btu})$$

$$\boxed{Q = 3238 \text{ Btu}}$$

Question: 18.100P

During a fire, the trunks of some dry oak trees (k = 0.17 W/m·K and α = 1.28 × 10–7 m2 /s) that are initially at a uniform temperature of 30°C are exposed to hot gases at 600°C for a period of 5 h, with a heat transfer coefficient of 65 W/m2 ·K on the surface. The ignition temperature of the trees is 410°C Treating the trunks of the trees as long cylindrical rods of diameter 20 cm, determine if these dry trees will ignite as the fire sweeps through them.

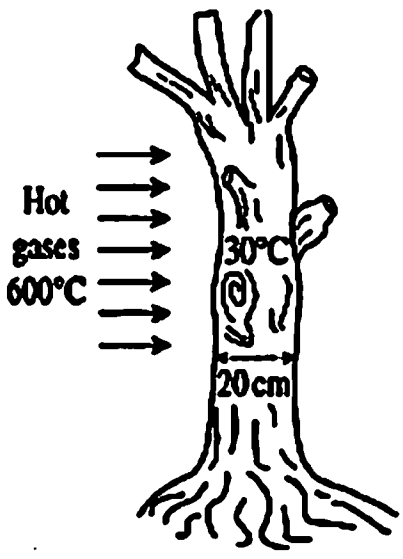


FIGURE P18–100

Step 1 of 5

Given data:

Thermal conductivity is $k = 0.17\,\text{W/m}\cdot\text{K}$

Thermal diffusivity is $\alpha = 1.28\times10^{-7}\,\text{m}^2/\text{s}$

Initially temperature is $T_i = 30\,^\circ\text{C}$

Time $t = 5\text{h}$

Diameter of the cylinder is $D = 20\,\text{cm}$

Heat transfer coefficient between man and ambiance is $h = 65\ \text{W/m}^2\cdot\text{K}$

Step 2 of 5

Characteristic length is

$$r_0 = \frac{r}{2}$$
$$= \frac{20}{2}$$
$$= 10\,\text{cm}$$
$$= 0.1\text{m}$$

Step 3 of 5

Then, biot number becomes

$$Bi = \frac{hL_c}{k}$$
$$= \frac{65\times0.1}{0.17}$$
$$= 38.2$$

Step 4 of 5

From the coefficient used in the one term approximation solution for transient heat conduction for cylinder

At Bi=38.2

We get: $\lambda = 2.34, A = 1.599$

Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$
$$= \frac{1.28\times10^{-7}\times(5\times3600\text{sec})}{0.1^2}$$
$$= 0.2304$$

Step 5 of 5

From the zeroth and first order Bessel functions of the first kind

At $J_o(2.34) = 0.033$

So, the dimensionless temperature after 5 hours of time is obtained as:

$$\frac{T-T_\alpha}{T_i-T_\alpha} = Ae^{-\lambda^2\tau}J_0\left(\frac{\lambda r}{r}\right)$$
$$\frac{T-T_\alpha}{T_i-T_\alpha} = 1.599\times e^{-2.34^2\times0.2304}\times J_0(2.34)$$
$$\frac{T-600}{30-600} = 1.599\times e^{-2.34^2\times0.2304}\times J_0(2.34)$$
$$\frac{T-600}{30-600} = 1.599\times e^{-2.34^2\times0.2304}\times 0.033$$
$$T = 591.481^\circ\text{C}$$

As this temperature is more than 410° C, the trees will ignite.

Question: 18.101P

Consider two 2-cm-thick large steel plates (k = 43 W/m·K and α = 1.17 × 10–5 m2 /s) that were put on top of each other while wet and left outside during a cold winter night at –10°C. The next day, a worker needs one of the plates, but the plates are stuck together because the freezing of the water between the two plates has bonded them together. In an effort to melt the ice between the plates and separate them, the worker takes a large hair dryer and blows hot air at 80°C all over the exposed surface of the plate on the top. The convection heat transfer coefficient at the top surface is estimated to be 40 W/m2 ·K. Determine how long the worker must keep blowing hot air before the two plates separate.

Step 1 of 3

The characteristic length of the plate,

$$L_c = L$$

Substitute 0.02 m for L,

$$L_c = L$$
$$= 0.02 \text{ m}$$

The Biot number,

$$Bi = \frac{hL_c}{k}$$

Substitute 40 $\text{W/m}^2\cdot^\circ\text{C}$ for h and 43 $\text{W/m}\cdot^\circ\text{C}$ for k

$$Bi = \frac{hL_c}{k}$$
$$= \frac{(40 \text{ W/m}^2.^\circ\text{C})(0.02)}{(43 \text{ W/m.}^\circ\text{C})}$$
$$= 0.019$$
$$Bi = 0.019 < 0.1$$

Since, the lumped system analysis is applicable. 9704044408

Step 2 of 3

The lumped system analysis,

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \qquad \text{..... (1)}$$

Here,

$$b = \frac{hA_s}{\rho C_p V}$$

Thermal diffusivity is $\alpha = \frac{k}{\rho c_p}$

Re-arranging the above relation we get

$$\rho c_p = \frac{k}{\alpha}$$

Hence,

$$b = \frac{h}{\rho C_p L_c}$$
$$= \frac{h}{\left(\frac{k}{\alpha}\right)L_c}$$

Substitute 40 $\text{W/m}^2\cdot^\circ\text{C}$ for h, 43 $\text{W/m}\cdot^\circ\text{C}$ for k and 1.17×10^{-5} m^2/s for α

$$b = \frac{h}{\left(\frac{k}{\alpha}\right)L_c}$$
$$= \frac{40 \text{ W/m}^2\cdot^\circ\text{C}}{\left(\frac{43}{1.17\times10^{-5}}\right)0.02}$$
$$= \frac{40 \text{ W/m}^2.^\circ\text{C}}{3.675\times10^6\times0.02 \text{ m}}$$
$$= 0.000544 \text{ s}^{-1}$$

Step 3 of 3

Now from equation (1),

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

Substitute 0 for $T(t)$, –10°C for T_i and 80°C for T_∞

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$
$$\frac{0-80}{-10-80} = e^{-(0.000544)x}$$
$$t = 216.513 \text{sec}$$

Therefore, time required for keeping the hot blow air before the two plates separate is

$$\boxed{t = 216.513 \text{sec}}$$

Question: 18.102P pg 1

Consider a plate of thickness 1 in, a long cylinder of diameter 1 in, and a sphere of diameter 1 in. all initially at 400°F and all made of bronze (k = 15.0 Btu/h·ft·°F and α = 0.333 ft2 /h). Now all three of these geometries are exposed to cool air at 75°F on all of their surfaces, with a heat transfer coefficient of 7 Btu/h·ft2 ·°F. Determine the center temperature of each geometry after 10 and 20 min. Explain why the center temperature of the sphere is always the lowest.

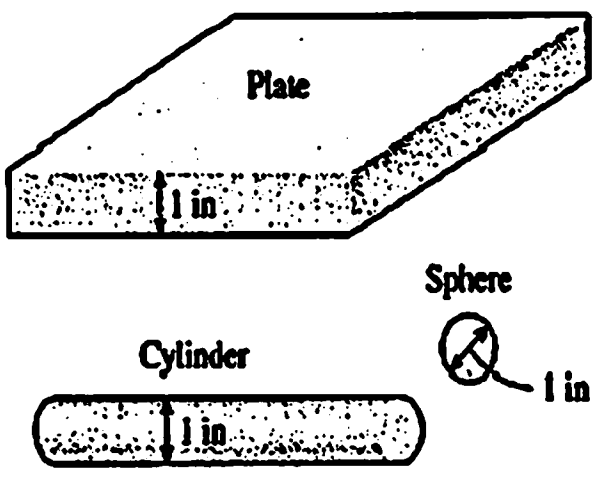


FIGURE P18–102E

Step 1 of 18

Given data:

Thickness of the plate = 1 in

Diameter of the cylindrical = 1 in

Initial Temperature = 400°F

Cool air temperature = 75°F

Thermal Conductivity, k = 15 Btu/h-ft·°F

Thermal Diffusivity, α = 0.333 ft^2 / hr

Heat transfer coefficient h = 7 Btu/h-ft^2 ·°F

Step 2 of 18

Calculation for 10 min. of time:

For the plate geometry:

Characteristic length of the plate is

$$L_c = \frac{L}{2}$$
$$= \frac{1}{2}$$
$$= 0.5\text{in}$$

Step 3 of 18

Then Biot number becomes

$$Bi = \frac{hL_c}{k}$$
$$= \frac{7 \times \left(\frac{0.5\text{in}}{12\text{ft}}\right)}{15}$$
$$= 0.019$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 4 of 18

Then, Fourier number becomes

$$\tau = \frac{\alpha t}{L_c^2}$$
$$= \frac{0.333 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$
$$= 32$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 5 of 18

From the one dimension term approximation solution of transient heat conduction for plane wall

At $Bi = 0.019$,

$\lambda = 0.141, A = 1.003$

Step 6 of 18

Then the center temperature of the plate becomes

$$\theta_{0,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$
$$A_1 e^{-\lambda_1^2 \tau} = (1.0033) e^{-(0.1410)^2 (31.97)}$$
$$A_1 e^{-\lambda_1^2 \tau} = 0.531$$
$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = A e^{-\lambda^2 \tau}$$
$$\frac{T(t) - 75}{400 - 75} = 1.003 e^{-0.141^2 \times 32}$$
$$T(t) = 248^\circ\text{F}$$

So the centre temperature of the flat plate after 10 min. is $\boxed{T(t) = 248^\circ\text{F}}$.

Step 7 of 18

For Cylinder:

From the one dimension term approximation solution of transient heat conduction for cylinder

At $Bi = 0.019$,

$\lambda_1 = 0.1995$ And $A_1 = 1.0050$

Then the center temperature of the plate becomes

$$\theta_{0,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$
$$A_1 e^{-\lambda_1^2 \tau} = (1.0050) e^{-(0.1995)^2 (31.97)}$$
$$A_1 e^{-\lambda_1^2 \tau} = 0.282$$

$$\theta_{0,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$A_1 e^{-\lambda_1^2 \tau} = (1.0050) e^{-(0.1995)^2 (31.97)}$$

$$A_1 e^{-\lambda_1^2 \tau} = 0.282$$

$$\frac{T_0 - 75}{400 - 75} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{T_0 - 75}{400 - 75} = 0.282$$

$$\boxed{T_0 = 167^\circ \text{F}}$$

So the centre temperature of the cylinder after 10 min. is $\boxed{T(t) = 166^\circ \text{F}}$.

Step 8 of 18

For Sphere:

From the one dimension term approximation solution of transient heat conduction for sphere

At $Bi = 0.019$,

Corresponding to this Biot number,

$\lambda_1 = 0.2445$ And $A_1 = 1.0060$

Step 9 of 18

Then the center temperature of the plate becomes

$$\theta_{0,wall} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$A_1 e^{-\lambda_1^2 \tau} = (1.0060) e^{-(0.2445)^2 (31.97)}$$

$$A_1 e^{-\lambda_1^2 \tau} = 0.149$$

$$\frac{T_0 - 75}{400 - 75} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{T_0 - 75}{400 - 75} = 0.149$$

$$\boxed{T_0 = 123^\circ \text{F}}$$

Therefore center temperature of the sphere is $\boxed{T(t) = 123^\circ \text{F}}$

Step 10 of 18

Calculation for 20 min of time:

For the plate geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12\text{ft}}\right)}{15}$$

$$= 0.019$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 11 of 18

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.333 \times \left(\frac{20}{60\text{min}}\right)}{\left(\frac{0.5}{12\text{ft}}\right)^2}$$

$$= 64$$

Since Fourier number greater than the 0.2.So one term approximation applicable.

Step 12 of 18

From the one dimension term approximation solution of transient heat conduction for plate

At $Bi = 0.019$,

We get: $\lambda = 0.141, A = 1.003$

Putting this into the expression for flat plate:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = A e^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.003 e^{-0.141^2 \times 64}$$

$$\boxed{T(t) = 166^\circ \text{F}}$$

So the centre temperature of the flat plate after 20 min. is $\boxed{T(t) = 166^\circ \text{F}}$.

Step 13 of 18

For the cylinder geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12\text{ft}}\right)}{15}$$

$$= 0.019$$

Step 14 of 18

From the one dimension term approximation solution of transient heat conduction for cylinder

At $Bi = 0.019$,

We get: $\lambda = 0.1995, A = 1.005$

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.333 \times \left(\frac{20}{60}\right)}{}$$

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.333 \times \left(\frac{20}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$= 64$$

Since Fourier number greater than the 0.2.So one term approximation applicable.

Step 15 of 18

Putting this into the expression for cylinder:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\Rightarrow \frac{T(t) - 75}{400 - 75} = 1.005e^{-0.1995^2 \times 64}$$

$$\Rightarrow T(t) = \boxed{100^\circ \text{F}}$$

So the centre temperature of the cylinder after 20 min. is $\boxed{T(t) = 100^\circ \text{F}}$.

Step 16 of 18

For the sphere geometry:

Biot number is

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12}\right)}{15}$$

$$= 0.019$$

Step 17 of 18

From the one dimension term approximation solution of transient heat conduction for sphere

We get: $\lambda = 0.245, A = 1.006$

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.333 \times \left(\frac{20}{60\,\text{min}}\right)}{\left(\frac{0.5\text{in}}{12\text{ft}}\right)^2}$$

$$= 64$$

Step 18 of 18

Putting this into the expression for sphere:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.006e^{-0245^2 \times 64}$$

$$\boxed{T(t) = 82^\circ \text{F}}$$

So the centre temperature of the sphere after 20 min. is 82° F.

Sphere being have largest surface area among the given geometries, its heat dissipation rate is higher. So the temperature of the sphere is always the lowest.

Question:

Repeat Prob. 18-102E for cast iron geometries (k = 29 Btu/h·ft·°F and α = 0.61 ft2 /h).

Step 1 of 20

Given data:

Thickness of the plate $= 1$ in

Diameter of the cylindrical $= 1$ in

Initial Temperature $= 400°F$

Cool air temperature $= 75°F$

Thermal Conductivity, $k = 29$ Btu/h-ft·° F

Thermal Diffusivity, $\alpha = 0.61 ft^2 / hr$

Heat transfer coefficient $h = 7$ Btu/h-ft^2 ·° F

Step 2 of 20

For 10 min. of time:

For the plate geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12ft}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 3 of 20

So, for 10 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.61 \times \left(\frac{10}{60\,min}\right)}{\left(\frac{0.5}{12ft}\right)^2}$$

$$= 59$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 4 of 20

From the one dimension term approximation solution of transient heat conduction for plane wall

At $Bi = 0.01$,

We get: $\lambda = 0.099, A = 1.0017$

Putting this into the expression for flat plate:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.0017e^{-0.099^2 \times 59}$$

$$\boxed{T(t) = 258^\circ F}$$

So the centre temperature of the flat plate after 10 min. is $\boxed{T(t) = 258^\circ F}$.

Step 5 of 20

For the cylinder geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 6 of 20

So, for 10 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.61 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$= 59$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 7 of 20

From the one dimension term approximation solution of transient heat conduction for cylinder

At $Bi = 0.01$,

We get: $\lambda = 0.141, A = 1.0025$

Putting this into the expression for cylinder:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.0025e^{-0.141^2 \times 59}$$

$$\boxed{T(t) = 176^\circ F}$$

So the centre temperature of the cylinder after 10 min. is $\boxed{T(t) = 176^\circ F}$.

Step 8 of 20

For the sphere geometry:

$$Bi = \frac{hL_c}{k}$$

$$7 \times \left(\frac{0.5}{12ft}\right)$$

For the sphere geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12\text{ft}}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 9 of 20

So, for 10 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.61 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$= 59$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 10 of 20

From the one dimension term approximation solution of transient heat conduction for sphere

We get: $\lambda = 0.173, A = 1.003$

So, for 10 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.61 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$= 59$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 11 of 20

Putting this into the expression for sphere:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.003e^{-0.173^2 \times 59}$$

$$T(t) = 131^\circ \text{F}$$

So the centre temperature of the sphere after 10 min. is $\boxed{T(t) = 131^\circ \text{F}}$.

Step 12 of 20

For 20 min. of time:

For the plate geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 13 of 20

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.61 \times \left(\frac{20}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$= 118$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 14 of 20

From the one dimension term approximation solution of transient heat conduction for plane wall

We get: $\lambda = 0.099, A = 1.0017$

Putting this into the expression for flat plate:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.0017e^{-0.099^2 \times 118}$$

$$T(t) = 177^\circ \text{F}$$

So the centre temperature of the flat plate after 20 min. is $\boxed{T(t) = 177^\circ \text{F}}$.

Step 15 of 20

And, for the cylinder geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 16 of 20

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 16 of 20

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$\Rightarrow \tau = \frac{0.61 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$\Rightarrow \tau = 118$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 17 of 20

From the one dimension term approximation solution of transient heat conduction for cylinder

We get: $\lambda = 0.141, A = 1.0025$

Putting this into the expression for cylinder:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.0025e^{-0.141^2 \times 118}$$

$$T(t) = 106^\circ \text{F}$$

So the centre temperature of the cylinder after 20 min. is $\boxed{T(t) = 106^\circ \text{F}}$.

Step 18 of 20

for the sphere geometry:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{7 \times \left(\frac{0.5}{12}\right)}{29}$$

$$= 0.01$$

Since Biot number less than 0.1, therefore lumped system analysis applicable.

Step 19 of 20

So, for 20 min. of time, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$\Rightarrow \tau = \frac{0.61 \times \left(\frac{10}{60}\right)}{\left(\frac{0.5}{12}\right)^2}$$

$$\Rightarrow \tau = 118$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 20 of 20

From the one dimension term approximation solution of transient heat conduction for sphere

$\lambda = 0.173, A = 1.003$

Putting this into the expression for sphere:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 75}{400 - 75} = 1.003e^{-0.173^2 \times 118}$$

$$T(t) = 84^\circ \text{F}$$

So the centre temperature of the sphere after 20 min. is $\boxed{T(t) = 84^\circ \text{F}}$.

Question: 18.104P

Reconsider Prob. 18-102E. Using EES (or other) software, plot the center temperature of each geometry as a function of the cooling time as the time varies from 5 min to 60 min. and discuss the results.

Step 1 of 9

Given data:

Thickness of the plate, $t = 1$ in.

Diameter of the cylindrical, $d = 1$ in.

Initial Temperature, $T_i = 400°F$

Cool air temperature, $T_{air} = 75°F$

Thermal Conductivity, $k = 15$ Btu/h-ft·°F

Thermal Diffusivity, $\alpha = 0.333\ ft^2/hr$

Heat transfer coefficient $h = 7$ Btu/h-ft^2·°F

Step 2 of 9

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
2*L=1/12 "[ft]"
2*r_o_c=1/12 "[ft], c stands for cylinder"
2*r_o_s=1/12 "[ft], s stands for sphere"
T_i=400 "[F]"
T_infinity=75 "[F]"
h=7 "[Btu/h-ft^2-F]"
"time=5 [min], parameter to be varied"
```

Step 3 of 9

```
"PROPERTIES"
k=15 "[Btu/h-ft-F]"
alpha=0.333*Convert(ft^2/h, ft^2/min) "[ft^2/min]"

"ANALYSIS"
"For plane wall"
Bi_w=(h*L)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1_w=0.1410
A_1_w=1.0033
tau_w=(alpha*time)/L^2
(T_o_w-T_infinity)/(T_i-T_infinity)=A_1_w*exp(-lambda_1_w^2*tau_w)
```

Step 4 of 9

```
"For long cylinder"
Bi_c=(h*r_o_c)/k
"From Table for biot number corresponding to this Bi number, we read"
lambda_1_c=0.1995
A_1_c=1.0050
tau_c=(alpha*time)/r_o_c^2
(T_o_c-T_infinity)/(T_i-T_infinity)=A_1_c*exp(-lambda_1_c^2*tau_c)
```

Step 5 of 9

```
"For sphere"
Bi_s=(h*r_o_s)/k
"From biot number Table corresponding to this Bi number, we read"
lambda_1_s=0.2445
A_1_s=1.0060
tau_s=(alpha*time)/r_o_s^2
(T_o_s-T_infinity)/(T_i-T_infinity)=A_1_s*exp(-lambda_1_s^2*tau_s)
```

Step 6 of 9

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 12
Select variables in equations as time and $T_{o,w}$ press ADD and then press ok
A table will appear put values of time as 5,10,15....60 in run 1 to 12
Repeat procedure for time and $T_{o,c}$ also time and $T_{o,s}$
Then from Calculate -> Solve table

Step 7 of 9

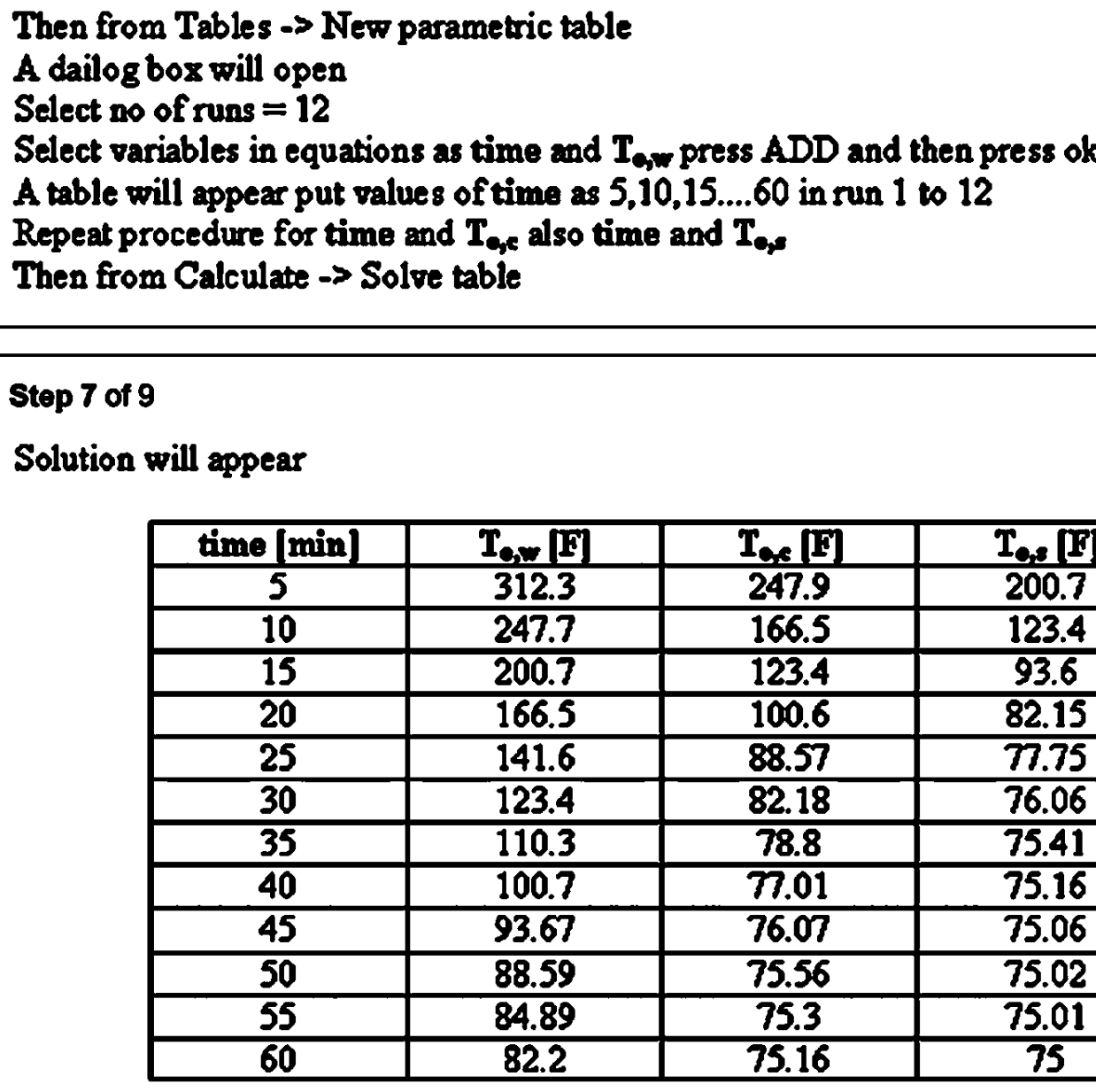
Solution will appear

time [min]	$T_{o,w}$ [F]	$T_{o,c}$ [F]	$T_{o,s}$ [F]
5	312.3	247.9	200.7
10	247.7	166.5	123.4
15	200.7	123.4	93.6
20	166.5	100.6	82.15
25	141.6	88.57	77.75
30	123.4	82.18	76.06
35	110.3	78.8	75.41
40	100.7	77.01	75.16
45	93.67	76.07	75.06
50	88.59	75.56	75.02
55	84.89	75.3	75.01
60	82.2	75.16	75

Step 8 of 9

Then from Plot -> New plot window -> X-Y Plot
Select X-axis time and Y-axis $T_{o,w}$
Repeat procedure for time and $T_{o,c}$ also time and $T_{o,s}$
Press OK
Graph will be plotted.

Step 9 of 9



Question: 18.105P

Engine valves (k = 48 W/m·K, cp = 440 J/kg·K, and ρ = 7840 kg/m3) are heated to 800°C in the heat treatment section of a valve manufacturing facility. The valves are then quenched in a large oil bath at an average temperature of 50°C. The heat transfer coefficient in the oil bath is 800 W/m2 ·K. The valves have a cylindrical stem with a diameter of 8 mm and a length of 10 cm. The valve head and the stem may be assumed to be of equal surface area, and the volume of the valve head can be taken to be 80 percent of the volume of stem. Determine how long it will take for the valve temperature to drop to (*a*) 400°C, (*b*) 200°C, and (*c*) 51 °C, and (*d*) the maximum heat transfer from a single valve.

Step 1 of 5

Given: Diameter of the watermelon = 20 cm

Initial Temperature of the watermelon T_i = 35°C

Lake temperature = 15°C

Center Temperature of the watermelon = 20°C

Time period =280min

$t = 4\text{h}\,40\text{min}$

$t = 16800\,\text{s}$

Thermal Conductivity, $k = 0.618\ \text{W/m°C}$

Thermal Diffusivity, $\alpha = 0.15\times10^{-6}\ \frac{\text{m}^2}{\text{s}}$

Density, $\rho = 995\ \frac{\text{kg}}{\text{m3}}$

Specific Heat, $C_p = 4.18\ \frac{\text{kJ}}{\text{kg°C}}$

Step 2 of 5

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2}$$

$$= \frac{(0.15\times10^{-6})(16800)}{(0.10)^2}$$

$$\tau = 0.252$$

$$\tau > 0.2$$

Step 3 of 5

Then the one-term solution can be written in the form

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

$$\frac{20-15}{35-15} = A_1 e^{-\lambda_1^2 (0.252)}$$

$$0.25 = A_1 e^{-\lambda_1^2 (0.252)}$$

It is determined from Biot number table by trial and error that this equation is satisfied when $Bi = 10$, which corresponds to $\lambda_1 = 2.8363$ and $A_1 = 1.9249$.

Step 4 of 5

Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k}$$

$$h = \frac{kBi}{r_o}$$

$$= \frac{(0.618)(10)}{(0.10)}$$

$$h = 61.8\ \frac{\text{W}}{\text{m}^2\,°\text{C}}$$

Step 5 of 5

The temperature at the surface of the watermelon:

For sphere,

$$\theta(r_o,t)_{sph} = \frac{T(r_o,t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

$$\theta(r_o,t)_{sph} = (1.9249)\times e^{-(2.8363)^2(0.252)} \times \frac{\sin(2.8363\ \text{rad})}{2.8363}$$

$$\frac{T(r_o,t) - 15}{35-15} = 0.0269$$

$$\boxed{T(r_o,t) = 15.5\ °\text{C}}$$

Question: 18.106P

Consider the engine block of a car made of cast iron (k = 52 W/m·K and α = 1.7 × 10–5 m2 /s). The engine can be considered to be a rectangular block whose sides are 80 cm, 40 cm, and 40 cm. The engine is at a temperature of 150°C when it is turned off. The engine is then exposed to atmospheric air at 17°C with a heat transfer coefficient of 6 W/m2 -K. Determine (*a*) the center temperature of the top surface whose sides are 80 cm and 40 cm and (*b*) the corner temperature after 45 min of cooling.

Step 1 of 12

Given:

Dimensions of engine = 80cmX 40cmX 40cm

Engine temperature = 140^0C

Atmospheric temperature = 17^0C

Step 2 of 12

The engine block of a car is allowed to cool in atmospheric air. Heat conduction in the block is three-dimensional, and thus the temperature varies in all three directions. The thermal properties of the block are constant.

Step 3 of 12

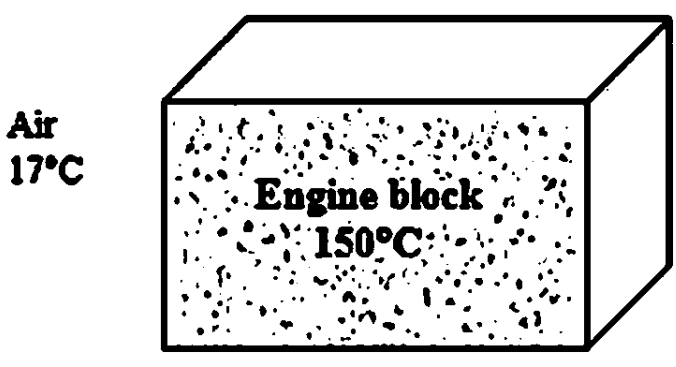
The thermal properties of cast iron are

k = 52 W/m.°C

$\alpha = 1.7\times10^{-5}$ m^2/s.

This rectangular block can physically be formed by by the intersection of two infinite plane walls of thickness 2L = 40 cm (call planes A and B) and an infinite plane wall of thickness 2L = 80 cm (call plane C). We measure x from the center of the block.

Step 4 of 12



Step 5 of 12

A) The Biot number is calculated for each of the plane wall to be

$$Bi_A = Bi_B = \frac{hL}{k} = \frac{(6\ \text{W/m}^2.^\circ\text{C})(0.2\ \text{m})}{(52\ \text{W/m}.^\circ\text{C})} = 0.0231$$

$$Bi_C = \frac{hL}{k} = \frac{(6\ \text{W/m}^2.^\circ\text{C})(0.4\ \text{m})}{(52\ \text{W/m}.^\circ\text{C})} = 0.0462$$

Step 6 of 12

The constants λ_1 and A_1 corresponding to these Biot numbers are,

$\lambda_{1(A,B)} = 0.150$ and $A_{1(A,B)} = 1.0038$

$\lambda_{1(C)} = 0.212$ and $A_{1(C)} = 1.0076$

The Fourier numbers are

$$\tau_{A,B} = \frac{\alpha t}{L^2} = \frac{(1.70\times10^{-5}\ \text{m}^2/\text{s})(45\ \text{min}\times60\ \text{s/min})}{(0.2\ \text{m})^2} = 1.1475 > 0.2$$

$$\tau_C = \frac{\alpha t}{L^2} = \frac{(1.70\times10^{-5}\ \text{m}^2/\text{s})(45\ \text{min}\times60\ \text{s/min})}{(0.4\ \text{m})^2} = 0.2869 > 0.2$$

Step 7 of 12

The center of the top surface of the block (whose sides are 80 cm and 40 cm) is at the center of the plane wall with 2L = 80 cm, at the center of the plane wall with 2L = 40 cm, and at the surface of the plane wall with 2L = 40 cm. The dimensionless temperatures are

$$\theta_{0,\text{wall(A)}} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}$$

$$= (1.0038)e^{-(0.150)^2(1.1475)}$$

$$= 0.9782$$

Step 8 of 12

$$\theta(L,t)_{\text{wall(B)}} = \frac{T(x,t) - T_\infty}{T_i - T_\infty}$$

$$= (1.0038)e^{-(0.150)^2(1.1475)}\cos(0.150)$$

$$= 0.9672$$

Step 9 of 12

$$\theta_{0,\text{wall(C)}} = \frac{T_0 - T_\infty}{T_i - T_\infty}$$

$$= (1.0076)e^{-(0.212)^2(0.2869)}$$

$$= 0.9947$$

Step 10 of 12

Then the center temperature of the top surface of the cylinder becomes

$$\left[\frac{T(L,0,0,t) - T_\infty}{T_i - T_\infty}\right]_{\substack{\text{short}\\ \text{cylinder}}} = \theta(L,t)_{\text{wall(B)}} \times \theta_{0,\text{wall(A)}} \times \theta_{0,\text{wall(C)}}$$

$$= 0.9672\times0.9782\times0.9947$$

$$= 0.9411$$

$$\frac{T(L,0,0,t) - 17}{150 - 17} = 0.9411$$

$$T(L,0,0,t) = 142.2^\circ\text{C}$$

$$\boxed{T(L,0,0,t) = 142.2^\circ\text{C}}$$

Step 11 of 12

(b) The corner of the block is at the surface of each plane wall. The dimensionless temperature for the surface of the plane walls with 2L = 40 cm is determined in part (a). The dimensionless temperature for the surface of the plane wall with 2L = 80 cm is determined from

$$\theta(L,t)_{\text{wall(C)}} = \frac{T(x,t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau}\cos(\lambda_1 L / L)$$

$$= (1.0076)e^{-(0.212)^2(0.2869)}\cos(0.212)$$

$$= 0.9724$$

Step 12 of 12

Then the corner temperature of the block becomes

$$\left[\frac{T(L,L,L,t) - T_\infty}{T_i - T_\infty}\right]_{\substack{\text{short}\\ \text{cylinder}}} = \theta(L,t)_{\text{wall,C}} \times \theta(L,t)_{\text{wall,B}} \times \theta(L,t)_{\text{wall,A}}$$

$$= 0.9724\times0.9672\times0.9672$$

$$= 0.9097$$

$$\frac{T(L,L,L,t) - 17}{150 - 17} = 0.9097$$

$$T(L,L,L,t) = 138.0^\circ\text{C}$$

$$\boxed{T(L,L,L,t) = 138.0^\circ\text{C}}$$

Question: 18.107P pg 1

8-cm-thick large food slabs tightly wrapped by thin paper are to be cooled in a refrigeration room maintained at 0°C. The heat transfer coefficient on the box surfaces is 25 W/m2 ·K and the boxes are to be kept in the refrigeration room for a period of 6 h. If the initial temperature of the boxes is 30°C determine the center temperature of the boxes if the boxes contain (*a*) margarine (k = 0.233 W/m·K and α = 0.11 × 10–6 m2 /s), (*b*) white cake (k = 0.082 W/m2·K and α = 0.10 × 10–6 m2 /s), and (*c*) chocolate cake (k = 0.106 W/m·K and α = 0.12 × 10–6 m2 /s).

Step 1 of 12

Given data:

Thickness of the food slabs $t = 8\text{ cm} = 0.08\text{ m}$

Initial Temperature of the watermelon $T_i = 30^\circ\text{C}$

Refrigeration room temperature $T_\infty = 0^\circ\text{C}$

Heat transfer coefficient $h = 25\text{ W/m}^2\,^\circ\text{C}$

Time period, $t = 6\text{ h}$

$= 6 \times 3600$

$= 21600\text{ s}$

Step 2 of 12

For margarine:

Thermal Conductivity, $k = 0.233\text{ W/m}\cdot\text{K}$

Thermal Diffusivity, $\alpha = 0.11\times10^{-6}\text{ m}^2/\text{s}$

For white cake:

Thermal Conductivity, $k = 0.082\text{ W/m}\cdot\text{K}$

Thermal Diffusivity, $\alpha = 0.10\times10^{-6}\text{ m}^2/\text{s}$

For chocolate cake:

Thermal Conductivity, $k = 0.106\text{ W/m}\cdot\text{K}$

Thermal Diffusivity, $\alpha = 0.12\times10^{-6}\text{ m}^2/\text{s}$

Step 3 of 12

Characteristic length of the slab is

$$L_c = \frac{L}{2}$$

$$= \frac{0.08}{2}$$

$$= 0.04\text{ m}$$

Step 4 of 12

For Margarine:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{25\times(0.04)}{0.233}$$

$$= 4.29$$

Since Biot number greater than 0.1, therefore lumped system analysis not applicable.

Step 5 of 12

So, for margarine, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.11\times10^{-6}\times(6\times3600\,\text{sec})}{(0.04)^2}$$

$$= 1.485$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 6 of 12

From the one dimension term approximation solution of transient heat conduction for plane wall

We get: $\lambda = 1.2792, A = 1.231$

Putting this into the expression:

$$\frac{T(t)-T_\alpha}{T_i-T_\alpha} = Ae^{-\lambda^2\tau}$$

$$\frac{T(t)-0}{30-0} = 1.231e^{-1.2792^2\times1.485}$$

$$T(t) = 3.25^\circ\text{C}$$

So the centre temperature of margarine after 6 hr is $\boxed{T(t) = 3.25^\circ\text{C}}$.

Step 7 of 12

For white cake:

Biot number is

$$Bi = \frac{hL_c}{k}$$

$$= \frac{25\times(0.04)}{0.082}$$

$$= 12.2$$

Since Biot number greater than 0.1, therefore lumped system analysis not applicable.

Step 8 of 12

So, for white cake, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.10\times10^{-6}\times(6\times3600SEC)}{(0.04)^2}$$

$$= 1.35$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 9 of 12

From the one dimension term approximation solution of transient heat conduction for plane wall

Step 9 of 12

From the one dimension term approximation solution of transient heat conduction for plane wall

We get: $\lambda = 1.4311, A = 1.2631$

Putting this into the expression:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 0}{30 - 0} = 1.2631e^{-1.4311^2 \times 1.35}$$

$$T(t) = 2.38^\circ \text{C}$$

So the centre temperature of white cake after 6 hr is $\boxed{T(t) = 2.38^\circ \text{C}}$.

Step 10 of 12

For chocolate cake:

$$Bi = \frac{hL_c}{k}$$

$$= \frac{25 \times (0.04)}{0.106}$$

$$= 9.43$$

Since Biot number greater than 0.1, therefore lumped system analysis not applicable.

Step 11 of 12

So, for chocolate cake, Fourier number is obtained to be:

$$\tau = \frac{\alpha t}{L_c^2}$$

$$= \frac{0.12 \times 10^{-6} \times (6 \times 3600 SEC)}{(0.04)^2}$$

$$= 2.7$$

Since Fourier number greater than the 0.2. .So one term approximation is applicable.

Step 12 of 12

From the one dimension term approximation solution of transient heat conduction for plane wall

$\lambda = 1.416, A = 1.261$

Putting this into the expression:

$$\frac{T(t) - T_\alpha}{T_i - T_\alpha} = Ae^{-\lambda^2 \tau}$$

$$\frac{T(t) - 0}{30 - 0} = 1.261e^{-1.416^2 \times 2.7}$$

$$T(t) = 0.2^\circ \text{C}$$

So the centre temperature of chocolate cake after 6 hr is $\boxed{T(t) = 0.2^\circ \text{C}}$.

Question: 18.108P

Long aluminum wires of diameter 3 mm (ρ = 2702 kg/m3 , cp = 0.896 kJ/kg·K, k = 236 W/m·K, and α = 9.75 × 10–5 m2 /s) are extruded at a temperature of 350°C and exposed to atmospheric air at 30°C with a heat transfer coefficient of 35 W/m2 ·K. (*a*) Determine how long it will take for the wire temperature to drop to 50°C. (*b*) If the wire is extruded at a velocity of 10 m/min, determine how far the

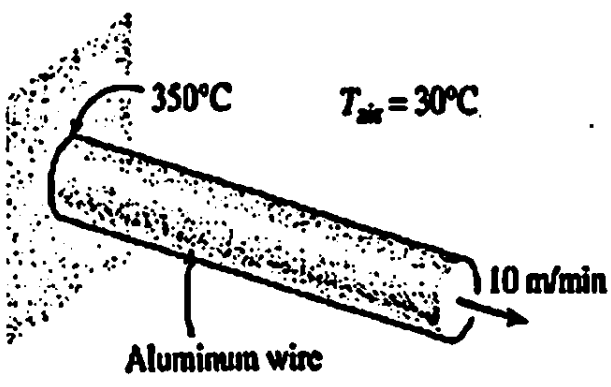


FIGURE P18–108

wire travels after extrusion by the time its temperature drops to50°C. What change in the cooling process would you propose to shorten this distance? (*c*) Assuming the aluminum wire leaves the extrusion room at 50°C, determine the rate of heat transfer from the wire to the extrusion room.

Step 1 of 7

Given:

Diameter of aluminum D= 3mm

Thermal conductivity $k = 236\ W/m\cdot^\circ C$

Density $\rho = 2702\ kg/m^3$

specific heat $C_p = 0.896\ kJ/kg\cdot^\circ C$

thermal diffusivity $\alpha = 9.75\ 10^{-5} m^2/s$

Extruded temperature $T_i = 350^\circ C$

Atmospheric temperature $T_\infty = 30^\circ C$

Step 2 of 7

Long aluminum wires are extruded and exposed to atmospheric air. Heat conduction in the wires is one-dimensional in the radial direction. The thermal properties of the aluminum are constant.

Step 3 of 7

(a) The characteristic length of the wire and the Biot number are

$$L_c = \frac{V}{A_s}$$

$$= \frac{\pi r_o^2 L}{2\pi r_o L}$$

$$= \frac{r_o}{2}$$

$$= \frac{0.0015\ m}{2}$$

$$= 0.00075\ m$$

$$Bi = \frac{hL_c}{k}$$

$$= \frac{(35\ W/m^2.^\circ C)(0.00075\ m)}{236\ W/m.^\circ C}$$

$$= 0.00011 < 0.1$$

Since Bi < 0.1, the lumped system analysis is applicable.

Step 4 of 7

Then,

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$b = \frac{hA_s}{\rho C_p V}$$

$$= \frac{h}{\rho C_p L_c}$$

$$= \frac{35\ W/m^2.^\circ C}{(2702\ kg/m^3)(896\ J/kg.^\circ C)(0.00075\ m)}$$

$$= 0.0193\ s^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{50 - 30}{350 - 30} = e^{-(0.0193\ s^{-1})t}$$

$t = 144\ s$

Time required to drop temperature is $\boxed{t = 144\ s}$

Step 5 of 7

(b) The wire travels a distance of

$$\text{velocity} = \frac{\text{length}}{\text{time}}$$

$$\text{Length} = (10/60\ m/s)(144\ s)$$

$$= 24\ m$$

length of the wire travel is $\boxed{L = 24\ m}$

This distance can be reduced by cooling the wire in a water or oil bath.

Step 6 of 7

(C) The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V}$$

$$= \rho(\pi r_0^2 / 4)V$$

$$= (2702\ kg/m^3)\pi(0.0015\ m)^2(10\ m/min)$$

$$= 0.191\ kg/min$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m} C_p [T(t) - T_\infty]$$

$$= (0.191\ kg/min)(0.896\ kJ/kg.^\circ C)(350 - 50)^\circ C$$

$$= 51.3\ kJ/min$$

Step 7 of 7

$$= 856\ W$$

The rate of heat transfer from the wire is $\boxed{856\ W}$

Question: 18.109P

Repeat Prob. 18-108 for a copper wire (ρ = 8950 kg/m3 , cp = 0.383 kJ/kg·K, k = 386 W/m·K, and α = 1.13 × 10–4 m2 /s).

Step 1 of 7

Given:

Diameter of aluminum D= 3mm

Thermal conductivity $k = 386\ W/m\cdot{}^{\circ}C$

Density $\rho = 8950\ kg/m^3$

Specific heat $C_p = 0.383\ kJ/kg\cdot{}^{\circ}C$

Thermal diffusivity $\alpha = 1.13\times 10^{-4} m^2/s$

Extruded temperature $T_i = 350^{\circ}C$

Atmospheric temperature $T_\infty = 30^{\circ}C$

Step 2 of 7

Long copper wires are extruded and exposed to atmospheric air.

Heat conduction in the wires is one-dimensional in the radial direction.The thermal properties of the copper are constant.

Step 3 of 7

Biot number is

$$L_c = \frac{V}{A_s}$$

$$= \frac{\pi r_o^2 L}{2\pi r_o L}$$

$$= \frac{r_o}{2}$$

$$= \frac{0.0015\ m}{2}$$

$$= 0.00075\ m$$

$$Bi = \frac{hL_c}{k}$$

$$= \frac{(35\ W/m^2.{}^{\circ}C)(0.00075\ m)}{386\ W/m.{}^{\circ}C}$$

$$= 0.000068 < 0.1$$

Since Bi < 0.1 the lumped system analysis is applicable.

Step 4 of 7

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$b = \frac{hA_s}{\rho C_p V}$$

$$= \frac{h}{\rho C_p L_c}$$

$$= \frac{35\ W/m^2.{}^{\circ}C}{(8950\ kg/m^3)(383\ J/kg.{}^{\circ}C)(0.00075\ m)}$$

$$= 0.0136\ s^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$\frac{50-30}{350-30} = e^{-(0.0136\ s^{-1})t}$$

$$t = 204\ s$$

Time required to drop temperature is $\boxed{t=204\ s}$

Step 5 of 7

(*b*) The wire travels a distance of

$$\text{velocity} = \frac{\text{length}}{\text{time}}$$

$$\text{length} = \left(\frac{10\ m/min}{60\ s/min}\right)(204\ s)$$

$$= 34\ m$$

Length of the wire travel is $\boxed{L=34\ m}$

This distance can be reduced by cooling the wire in a water or oil bath.

Step 6 of 7

(*c*) The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{V}$$

$$= \rho(\pi r_0^2 / 4)V$$

$$= (8950\ kg/m^3)\pi(0.0015\ m)^2(10\ m/min)$$

$$= 0.633\ kg/min$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}C_p[T(t)-T_\infty]$$

$$\dot{Q} = (0.633\ kg/min)(0.383\ kJ/kg.{}^{\circ}C)(350-50)^{\circ}C$$

Step 7 of 7

$$\dot{Q} = 72.7\ kJ/min$$

$$\dot{Q} = 1212\ W$$

Rate of heat transfer from the wire is $\boxed{Q=1212\ W}$

Question: 18.110P

A 40-cm-thick brick wall (k = 0.72 W/m·K, and α = 1.6 × 10–6 m2 /s) is heated to an average temperature of 18°C by the heating system and the solar radiation incident on it during the day. During the night, the outer surface of the wall is exposed to cold air at –3°C with an average heat transfer coefficient of 20 W/m2 ·K, determine the wall temperatures at distances 15, 30, and 40 cm from the outer surface for a period of 2 h.

Step 1 of 3

Given:

Dimensions of brick wall $= 10\text{m}(\text{long}) \times 3\text{m}(\text{high}) \times 0.3\text{m}(\text{thick})$

Temperature of inner part of house $T_i = 5^\circ\text{C}$

Outer surface temperature $T_s = 15^\circ\text{C}$

Thermal conductivity of brick $k = 0.72\ \text{W/m} \cdot {}^\circ\text{C}$

Thermal diffusivity $\alpha = 0.45\ 10^{-4}\ \text{m}^2/\text{s}$.

Step 2 of 3

A brick house made of brick that was initially cold is exposed to warm atmospheric air at the outer surfaces.

The exact analytical solution to this problem is

$$\frac{T(x,t)-T_i}{T_s-T_i} = erfc\left(\frac{x}{\sqrt{\alpha t}}\right)$$

Step 3 of 3

Substituting,

$$\frac{5.1-5}{15-5} = erfc\left(\frac{0.3\,\text{m}}{2\sqrt{(0.45\times10^{-6}\,\text{m}^2/\text{s})t}}\right)$$

$0.01 = \text{erfc}(1.8215)$

The time is determined to be

$$\left(\frac{0.3\,\text{m}}{2\sqrt{(0.45\times10^{-6}\,\text{m}^2/\text{s})t}}\right) = 1.8215$$

$t = 15{,}070\ \text{s}$

$= 251\ \text{min}$

Time taken to increase the temperature is $\boxed{251\ \text{min}}$

Question: 18.111P

Large steel plates 1.0-cm in thickness are quenched from 600°C to 100°C by submerging them in an oil reservoir held at 30°C. The average heat transfer coefficient for both faces of steel plates is 400 W/m2 ·K. Average steel properties are k = 45 W/m·K. ρ = 7800 kg/m3 , and cp = 470 J/kg·K. Calculate the quench time for steel plates.

Step 1 of 3

Given:

Thickness of steel plate $t = 1.0$ cm

Temperature of oil reservoir $= 30\ ^\circ C$

Heat transfer coefficient $h = 400\ W/m^2 \cdot K$

Thermal conductivity of steel $k = 45\ W/m \cdot K$

Density of steel $\rho = 7800\ kg/m^3$

Specific heat of steel $c_p = 470\ J/kg \cdot K$

Step 2 of 3

The thermal properties of the rod are constant. If the heat transfer coefficient is constant and uniform over the entire surface. Then the characteristic length and the Biot number for the copper rod are

$$L_c = \frac{V}{A_s}$$

$$= \frac{L}{2}$$

$$= \frac{0.01m}{2}$$

$$= 0.005m$$

$$Bi = \frac{hL_c}{k}$$

$$= \frac{(400 W/m^2.^\circ C)(0.005m)}{(45\ W/m.^\circ C)}$$

$$= 0.0044 < 0.1$$

Step 3 of 3

The lumped system analysis is applicable since Bi < 0.1. Then the temperature of the rod becomes

$$b = \frac{hA}{\rho C_p V}$$

$$= \frac{h}{\rho C_p L_c}$$

$$= \frac{400\ W/m^2.^\circ C}{(7800\ kg/m^3)(470\ J/kg.^\circ C)(0.005m)}$$

$$= 0.02182\ s^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$\frac{100 - 30}{600 - 30} = e^{-(0.02182 s^{-1})t}$$

$t = 96s$

$t = 1.6min$

$\boxed{time = 1.6min}$

Question: 18.112P

Aluminium wires, 4 mm in diameter, are produced by extrusion. The wires leave the extruder at an average temperature of 350°C and at a linear rate of 10 m/min. Before leaving the extrusion room, the wires are cooled to an average temperature of 50°C by transferring heat to the surrounding air at 25°C with a heat transfer coefficient of 50 W/m·K. Calculate the necessary length of the wire cooling section in the extrusion room.

Step 1 of 4

Characteristic length and Biot number for long cylinder can be calculated as follows

$$L_c = \frac{V}{A_s}$$

$$= \frac{(\pi D^2/4)L}{\pi DL}$$

$$L_c = \frac{D}{4}$$

Substitute 0.004 m for D,

$$L_c = \frac{D}{4}$$

$$= \frac{0.004\text{m}}{4}$$

$$= 0.001\text{m}$$

Step 2 of 4

From the equation of Biot number is given as

$$Bi = \frac{hL_c}{k}$$

Substitute 50 $\text{W/m}^2\cdot\text{K}$ for h and 237 $\text{W/m}\cdot^\circ\text{C}$ for k

$$Bi = \frac{hL_c}{k}$$

$$= \frac{(50)(0.001)}{(237)}$$

$$= 0.000210$$

Since Biot number is less than 0.1, therefore lumped system analysis is applicable.

Step 3 of 4

From the equation of lumped system analysis, we have

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

Where,

$$b = \frac{hA}{\rho c_p V}$$

$$= \frac{h}{\rho c_p L_c}$$

Substitute 50 $\text{W/m}^2\cdot\text{K}$ for h, 2702 kg/m^3 for ρ and 903 $\text{kJ/kg}\cdot\text{K}$ for c_p

$$b = \frac{h}{\rho c_p L_c}$$

$$= \frac{50}{(2702)(903)(0.001)}$$

$$= 0.0204\ \text{s}^{-1}$$

Step 4 of 4

Then, substituting the known values, we get

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

Substitute 50°C for $T(t)$, 350°C for T_i and 25°C for T_∞

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt}$$

$$\frac{50-25}{350-25} = e^{-(0.0204)t}$$

$$t = 126\text{s}$$

Length if wire necessary for cooling section will be calculated as

$$L = \frac{t}{V}$$

$$= \frac{126\text{sec} \times \frac{1\text{min}}{60\text{sec}}}{10\text{m/min}}$$

$$\boxed{L = 0.21\text{m}}$$

Therefore, length of the steel wire is 0.21m.

Question: 18.113P

Two metal rods are being heated in an oven with uniform ambient temperature of 1000°C and convection heat transfer coefficient of 25 W/m2 ·K. Rod A is made of aluminum (ρ = 2702 kg/m3 cp = 903 J/kg·K, and k = 237 W/m·K) and rod B is made of stainless steel (ρ = 8238 kg/m3 , cp 468 J/kg·K, and k = 13.4 W/m·K). Both rods have diameter of 25 mm and length of 1 m. If the initial temperature of both rods is 15°C, determine the average temperatures of both rods after 5 minutes elapsed time.

Step 1 of 4

The characteristic length of the cylinder ,

$$L_c = \frac{D}{4}$$

Substitute 0.025 m for D,

$$L_c = \frac{D}{4}$$
$$= \frac{0.025}{4}$$
$$= 0.00625 \text{ m}$$

The temperature of the rod after 't' mins of elapsed time is given by

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

Where,

$$b = \frac{h}{\rho c_p L_c}$$

Step 2 of 4

For rod A:

$$b = \frac{h}{\rho_A c_{p,A} L_c}$$

Substitute 25 W/m^2 ·K for h, 2702 kg/m^3 for ρ and 903 J/kg·K for c_p

$$b = \frac{h}{\rho_A c_{p,A} L_c}$$
$$= \frac{25}{(2702)(903)(0.00625)}$$
$$= 1.639 \times 10^{-3}$$

The temperature of the rod A after 5mins of elapsed time is given by

$$\frac{T(t)_A - T_\infty}{T_i - T_\infty} = e^{-bt}$$

Substitute 15°C for T_i and 1000°C for T_∞

$$\frac{T(t)_A - T_\infty}{T_i - T_\infty} = e^{-bt}$$
$$\frac{T(t)_A - 1000}{15 - 1000} = e^{-1.639 \times 10^{-3}(5 \times 60)}$$
$$T(t)_A - 1000 = -602.412$$
$$T(t)_A = 397.59°C$$

Therefore, the temperature of the rod A after 5mins of elapsed time is $T(t)_A = 397.59°C$

Step 3 of 4

For rod B:

$$b = \frac{h}{\rho_B c_{p,B} L_c}$$

Substitute 25 W/m^2 ·K for h, 8238 kg/m^3 for ρ and 468 J/kg·K for c_p

$$b = \frac{h}{\rho_B c_{p,B} L_c}$$
$$= \frac{25}{(8238)(468)(0.00625)}$$
$$= 1.0375 \times 10^{-3}$$

The temperature of the rod B after 5mins of elapsed time is given by

$$\frac{T(t)_B - T_\infty}{T_i - T_\infty} = e^{-bt}$$
$$\frac{T(t)_B - 1000}{15 - 1000} = e^{-1.0375 \times 10^{-3}(5 \times 60)}$$
$$T(t)_B - 1000 = -721.54$$
$$T(t)_B = 278.46°C$$

Therefore, the temperature of the rod B after 5mins of elapsed time is $T(t)_B = 278.46°C$

Step 4 of 4

The average temperature of both the rods after 5mins of elapsed time is given by

$$T_{avg} = \frac{T(t)_A + T(t)_B}{2}$$
$$= \frac{397.59 + 278.46}{2}$$
$$\boxed{T_{avg} = 338.025°C}$$

Therefore, the average temperature of both the rods after 5mins of elapsed time is 338.025°C

Question: 18.114P

In an event of a volcano eruption, lava at 1200°C is found flowing on the ground. The ground was initially at 15°C and the lava flow has a convection heat transfer coefficient of 3500 W/m2 ·K. Determine the ground surface (*a*) temperature and (*b*) heat flux after 2 s of lava flow. Assume the ground has properties of dry soil.

Step 1 of 4

(a).

Obtain the values for properties of ground from Appendix table A-8 as shown.

$$k = 1.0 \text{ W/m} \cdot \text{K}$$

$$\rho = 1500 \text{ kg/m}^3$$

$$C_p = 1900 \text{ J/kg} \cdot \text{K}$$

Calculate the thermal diffusivity of ground by using the relation,

$$\alpha = \frac{k}{\rho C_p}$$

Here, thermal conductivity of the ground is k , density of the ground material is ρ , and specific heat of the material is C_p .

Substitute $1.0 \text{ W/m} \cdot \text{K}$ for k , 1500 kg/m^3 for ρ , and $1900 \text{ J/kg} \cdot \text{K}$ for ρ .

$$\alpha = \frac{1}{1500 \times 1900}$$
$$= 3.509 \times 10^{-7} \text{ m}^2/\text{s}$$

Step 2 of 4

Calculate the surface temperature of the ground using the relation,

$$\frac{T_s - T_i}{T_\infty - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right) - \exp\left(\frac{h^2 \alpha t}{k^2}\right)\text{erfc}\left(\frac{h\sqrt{\alpha t}}{k}\right) \quad \text{..... (1)}$$

Here, initial temperature is T_i , surface temperature is T_s , surrounding temperature is T_∞ , depth from the ground surface is x , convection heat transfer of lava flow is h , and time required for heat flux transfer is t .

Calculate the value of $\frac{h^2 \alpha t}{k^2}$ as shown.

Substitute $3500 \text{ W/m}^2 \cdot \text{K}$ for h , $3.509 \times 10^{-7} \text{ m}^2/\text{s}$ for α , 2 s for t , $1.0 \text{ W/m} \cdot \text{K}$ for k

$$\frac{h^2 \alpha t}{k^2} = \frac{3500^2 \times 3.509 \times 10^{-7} \times 2}{1^2}$$
$$= 8.597$$

Calculate the value of $\frac{h\sqrt{\alpha t}}{k}$ as shown.

Substitute $3500 \text{ W/m}^2 \cdot \text{K}$ for h , $3.509 \times 10^{-7} \text{ m}^2/\text{s}$ for α , 2 s for t , $1.0 \text{ W/m} \cdot \text{K}$ for k

$$\frac{h\sqrt{\alpha t}}{k} = \frac{3500 \times \sqrt{3.509 \times 10^{-7} \times 2}}{1}$$
$$= 2.932$$

Step 3 of 4

Calculate the surface temperature of the ground 2 s after the eruption of lava as shown.

Substitute 1200 °C for T_∞ , 15 °C for T_i , 0 for x , $3.509 \times 10^{-7} \text{ m}^2/\text{s}$ for α , 2 s for t , 8.597 for $\frac{h^2 \alpha t}{k^2}$, and 2.932 for $\frac{h\sqrt{\alpha t}}{k}$ in equation (1).

$$\frac{T_s - 15}{1200 - 15} = \text{erfc}\left(\frac{(0)}{2\sqrt{3.509 \times 10^{-7} \times 2}}\right) - \exp(8.597)\text{erfc}(2.932)$$

$$T_s - 15 = (1200 - 15)\left[1 - (5415.39)\left(3.37 \times 10^{-5}\right)\right]$$

$$T_s = 15 + 1185(1 - 0.182)$$
$$= 983.74 \text{ °C}$$

Therefore, the surface temperature of the ground 2 s after the eruption of lava is $\boxed{983.74 \text{ °C}}$.

Step 4 of 4

(b).

Calculate the heat flux of the lava flow 2 s after eruption using the relation,

$$\dot{q}_s = h(T_\infty - T_s)$$

Substitute 1200 °C for T_∞ , 983.74 °C for T_s , and $3500 \text{ W/m}^2 \cdot \text{K}$ for h .

$$\dot{q}_s = 3500(1200 - 983.74)$$
$$= 759610 \text{ W/m}^2$$
$$= 759.61 \text{ kW/m}^2$$

Therefore, the heat flux of the lava flow 2 s after eruption is $\boxed{759.61 \text{ kW/m}^2}$.

Question: 18.115P

A large heated steel block (ρ = 7832 kg/m3 cp = 434 J/kg·K. k = 63.9 W/m·K. and α = 18.8 × 10–6 m2 /s) is allowed to cool in a room at 25°C. The steel block has an initial temperature of 450°C and the convection heal transfer coefficient is 25 W/m2 ·K. Assuming that the steel block can be treated as a quarter-infinite medium, determine the e temperature at the edge of the steel block after 10 minutes of cooling.

Step 1 of 4

Multidimensional solutions expressed as products of one-dimensional solutions for bodies that are initially at a uniform temperature T_i and expose to convection from all surfaces to a medium at T_∞ table for Quater-infinite medium the one term approximation is expressed as

$$\theta(x,y,t)=\theta_{\text{semi-inf}}(x,t)\theta_{\text{semi-inf}}(y,t) \quad \text{..... (1)}$$

Convection on the surface at the edge of the steel block along the x-axis is given by,

$$1-\frac{T(x,t)-T_i}{T_\infty-T_i}=1-\operatorname{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)-\exp\left(\frac{hx}{k}+\frac{h^2\alpha t}{k^2}\right)\operatorname{erfc}\left(\frac{x}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)$$

$$\frac{T_\infty-T_i-T(x,t)+T_i}{T_\infty-T_i}=1-\left[\begin{array}{l}\operatorname{erfc}\left(\dfrac{0}{2\sqrt{(18.8\times10^{-6}\times600)}}\right)\\ -\exp\left(\dfrac{25\times0}{63.9}+\dfrac{(25^2\times18.8\times10^{-6}\times600)}{63.9^2}\right)\operatorname{erfc}\left(\dfrac{0}{2\sqrt{(18.8\times10^{-6}\times600)}}+\dfrac{\left(25\sqrt{18.8\times10^{-6}\times600}\right)}{63.9}\right)\end{array}\right]$$

$$\frac{T(x,t)-T_\infty}{T_i-T_\infty}=1-\left[\operatorname{erfc}(0)-\exp(0+1.727\times10^{-3})\operatorname{erfc}(0+0.0415)\right]$$

$$=1-[1-1.00173\times0.9549]$$

$$=0.9565$$

Step 2 of 4

Specified surface temperature at the edge of the steel block along the y-axis is given by,

$$1-\frac{T(y,t)-T_i}{T_\infty-T_i}=1-\operatorname{erfc}\left(\frac{y}{2\sqrt{\alpha t}}\right)-\exp\left(\frac{hy}{k}+\frac{h^2\alpha t}{k^2}\right)\operatorname{erfc}\left(\frac{y}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)$$

Substitute 25 $\text{W/m}^2\cdot\text{K}$ for h, $18.8\times10^{-6}\ \text{m}^2/\text{s}$ for α and 600 sec for t,

$$1-\frac{T(y,t)-T_i}{T_\infty-T_i}=1-\operatorname{erfc}\left(\frac{y}{2\sqrt{\alpha t}}\right)-\exp\left(\frac{hy}{k}+\frac{h^2\alpha t}{k^2}\right)\operatorname{erfc}\left(\frac{y}{2\sqrt{\alpha t}}+\frac{h\sqrt{\alpha t}}{k}\right)$$

$$\frac{T_\infty-T_i-T(y,t)+T_i}{T_\infty-T_i}=1-\left[\begin{array}{l}\operatorname{erfc}\left(\dfrac{0}{2\sqrt{(18.8\times10^{-6}\times600)}}\right)\\ -\exp\left(\dfrac{25\times0}{63.9}+\dfrac{(25^2\times18.8\times10^{-6}\times600)}{63.9^2}\right)\operatorname{erfc}\left(\dfrac{0}{2\sqrt{(18.8\times10^{-6}\times600)}}+\dfrac{\left(25\sqrt{18.8\times10^{-6}\times600}\right)}{63.9}\right)\end{array}\right]$$

$$\frac{T(x,t)-T_\infty}{T_i-T_\infty}=1-\left[\operatorname{erfc}(0)-\exp(0+1.727\times10^{-3})\operatorname{erfc}(0+0.0415)\right]$$

$$=1-[1-1.00173\times0.9549]$$

$$=0.9565$$

Step 3 of 4

The one dimensional solution for the semi infinite solid along the x-axis is given by,

$$\theta_{\text{semi-infinite}}(x,t)=\left(\frac{T(x,t)-T_\infty}{T_i-T_\infty}\right)$$

$$=0.9565$$

The one dimensional solution for the semi infinite solid along the y-axis is given by,

$$\theta_{\text{semi-infinite}}(y,t)=\left(\frac{T(y,t)-T_\infty}{T_i-T_\infty}\right)$$

$$=0.9565$$

Step 4 of 4

Thus the transient temperature distribution for the steel block is given by

$$\left(\frac{T(x,y,t)-T_\infty}{T_i-T_\infty}\right)_{\text{steelblock}}=\theta_{\text{semi-infinite}}(x,t)\theta_{\text{semi-infinite}}(y,t)$$

$$=0.9565\times0.9565$$

$$\frac{T(x,y,t)-T_\infty}{T_i-T_\infty}=0.9149$$

$$T(x,y,t)=0.9149(T_i-T_\infty)+T_\infty$$

$$=0.9149(450-25)+25$$

$$\boxed{T=413.87^\circ\text{C}}$$

The temperature at the edge of the steel block after 10 minutes of cooling is 413.87°C

CHAPTER 19
SOLUTIONS

Question: 19.001P

Define incompressible flow and incompressible fluid. Must the flow of a compressible fluid necessarily be treated as compressible?

Step 1 of 1

Incompressible flow is type of flow in which density of the fluid remains constant. Incompressible fluids are those fluids whose density is independent of pressure.

It is not necessary that flow of compressible fluid is compressible as during the flow of compressible fluid density may remain constant.

Question: 19.002P

What is forced convection? How does it differ from natural convection? Is convection caused by winds forced or natural convection?

Step 1 of 1

Forced or natural convection can be distinguished only based on how the motion of fluid is initiated.

When the fluid motion is initiated by an external means such as a pump or a fan to cause the fluid to flow over a surface or in a tube, it is called forced convection.

If any fluid motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid then it is called as natural convection.

Convection by winds will be natural convection for the earth, whereas these will be forced convection for bodies subjected to the winds as it makes no difference for the body whether the air motion is caused by a fan or by the winds.

Question: 19.003P

What is external forced convection? How does it differ from internal forced convection? Can a heat transfer system involve both internal and external convection at the same time? Give an example.

Step 1 of 1

When the fluid is forced to flow over a surface it is external forced convection. When the fluid is forced to flow in a tube it is Internal forced convection. It is possible to have both Internal and external convection simultaneously as is case of pipe transporting a fluid in a windy area.

Question: 19.004P

In which mode of heat transfer is the convection heat transfer coefficient usually higher, natural convection or forced convection? Why?

Step 1 of 1

As coefficient of heat transfer depends on the fluid velocity.

Coefficient of convective heat transfer will be higher in case of forced convection as it involves high velocity of fluid.

Question: 19.005P

Consider a hot baked potato. Will the potato cool faster or slower when we blow the warm air coming from our lungs on it instead of letting it cool naturally in the cooler air in the room? Explain.

Step 1 of 1

As motion of fluid causes heat transfer to raise so despite smaller temperature difference potato will cool faster by blowing warm air to it.

Question: 19.006P

What is the physical significance of the Nusselt number? How is it defined?

Step 1 of 2

Nusselt number represents the ratio of heat flux by fluid when heat is transferred through fluid layer by convection when fluid involves some motion to heat flux by fluid when heat is transferred through fluid layer by conduction when fluid is motionless. Nusselt number is defines as

$$Nu = \frac{hL}{k}$$

Where L is the characteristic length of the surface

k is the thermal conductivity of the fluid

h is coefficient of heat transfer by convection

Step 2 of 2

A Nusselt number close to unity, namely convection and conduction of similar magnitude, is characteristic of "slug flow" or laminar flow. A larger Nusselt number corresponds to more active convection, with turbulent flow typically in the 100-1000 range

magnitude, is characteristic of "slug flow" or laminar flow. A larger Nusselt number corresponds to more active convection, with turbulent flow typically in the 100-1000 range

Question: 19.007P

When is heat transfer through a fluid conduction and when is it convection? For what case is the rate of heat transfer higher? How does the convection heat transfer coefficient differ from the thermal conductivity of a fluid?

Step 1 of 1

In the absence heat transfer through a fluid is conduction and in the presence of bulk fluid motion it is convection.

Heat transfer rate will be higher in convection due to presence of fluid motion. Coefficient of heat transfer by convection depends on fluid properties as well as fluid motion whereas thermal conductivity does not depend on the flow it is a material property. And also it is depends on the temperature of the metal or material.

Question: 19.008P

During air cooling of potatoes, the heat transfer coefficient for combined convection, radiation, and evaporation is determined experimentally to be as shown:

Air Velocity, m/s	Heat Transfer Coefficient, W/m2·K
0.66	14.0
1.00	19.1
1.36	20.2
1.73	24.4

Consider an 8-cm-diameter potato initially at 20°C with a thermal conductivity of 0.49 W/m•K. Potatoes are cooled by refrigerated air at 5°C at a velocity of 1 m/s. Determine the initial rate of heat transfer from a potato, and the initial value of the temperature gradient in the potato at the surface.

Step 1 of 5

Given: -

Diameter of potato = 8 cm

Initial temperature = 20

Thermal conductivity = 0.49 W/m.°C

Final temperature of potato = 5°C

Velocity = 1 m/s

Step 2 of 5

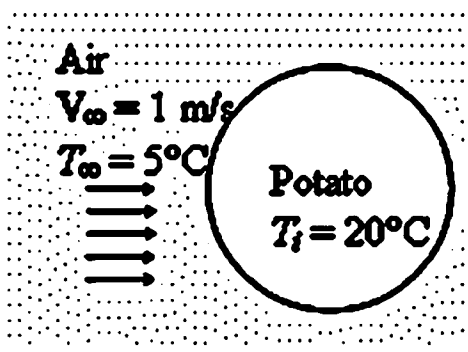
Heat transfer area

$$A_s = \pi D^2$$

$$= \pi(0.08\ \text{m})^2$$

$$= 0.020096\ \text{m}^2$$

Step 3 of 5



Step 4 of 5

Heat transfer rate can be calculated as

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$\dot{Q} = (19.1\ \text{W/m}^2.\text{°C})(0.020096\ \text{m}^2)(20-5)\text{°C}$$

$$\dot{Q} = 5.8\ \text{W}$$

Step 5 of 5

Initial value of temperature gradient can be calculated as follows

$$\dot{q}_{conv} = \dot{q}_{cond}$$

$$= -k\left(\frac{\partial T}{\partial r}\right)_{r=R} = h(T_s - T_\infty)$$

$$\left.\frac{\partial T}{\partial r}\right|_{r=R} = -\frac{h(T_s - T_\infty)}{k}$$

$$= -\frac{(19.1\ \text{W/m}^2.\text{°C})(20-5)\text{°C}}{(0.49\ \text{W/m.°C})}$$

$$= -585\ \text{°C/m}$$

Question: 19.009P

An average man has a body surface area of 1.8 m2 and a skin temperature of 33°C. The convection heat transfer coefficient for a clothed person walking in still air is expressed as h =8.6V0.53 for 0.5 < V < 2 m/s, where V is the walking velocity in m/s. Assuming the average surface temperature of the clothed person to be 30°C, determine the rate of heat loss from an average man walking in still air at 7°C by convection at a walking velocity of (*a*) 0.5 m/s. (*b*) 1.0 m/s, (*c*) 1.5 m/s, and (*d*) 2.0 m/s.

Step 1 of 9

Given data:

Surface area of body, $A_s = 1.8\,\text{m}^2$

Skin temperature, $T_{skin} = 33°\text{C}$

Average surface temperature of clothed person, $T_s = 30°\text{C}$

The temperature of still air, $T_\infty = 7°\text{C}$

Coefficient of heat transfer for convection, $h = 8.6V^{0.53}$

Step 2 of 9

(*a*)

The walking velocity of average man, $V = 0.5\,\text{m/s}$

Coefficient of heat transfer for convection,

$$h = 8.6V^{0.53}$$
$$= 8.6\times(0.5)^{0.53}$$
$$= 5.956\ \text{W/m}^2\cdot°\text{C}$$

Coefficient of heat transfer for convection is $\boxed{h = 5.956\ \text{W/m}^2\cdot°\text{C}}$

Step 3 of 9

By using the formula for rate of heat loss, we get

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 5.956\times1.8\times(30-7)$$
$$= 246.578\,\text{W}$$

Rate of heat loss from average man is $\boxed{\dot{Q} = 246.578\,\text{W}}$

Step 4 of 9

(*b*)

The walking velocity of average man, $V = 1.0\,\text{m/s}$

Coefficient of heat transfer for convection,

$$h = 8.6V^{0.53}$$
$$= 8.6\times(1.0)^{0.53}$$
$$= 8.6\ \text{W/m}^2\cdot°\text{C}$$

Coefficient of heat transfer for convection is $\boxed{h = 8.6\ \text{W/m}^2\cdot°\text{C}}$

Step 5 of 9

By using the formula for rate of heat loss, we get

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 8.6\times1.8\times(30-7)$$
$$= 356.04\,\text{W}$$

Rate of heat loss from average man is $\boxed{\dot{Q} = 356.04\,\text{W}}$

Step 6 of 9

(*c*)

The walking velocity of average man, $V = 1.5\,\text{m/s}$

Coefficient of heat transfer for convection,

$$h = 8.6V^{0.53}$$
$$= 8.6\times(1.5)^{0.53}$$
$$= 10.661\ \text{W/m}^2\cdot°\text{C}$$

Coefficient of heat transfer for convection is $\boxed{h = 10.661\ \text{W/m}^2\cdot°\text{C}}$

Step 7 of 9

By using the formula for rate of heat loss, we get

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 10.661\times1.8\times(30-7)$$
$$= 441.365\,\text{W}$$

Rate of heat loss from average man is $\boxed{\dot{Q} = 441.365\,\text{W}}$

Step 8 of 9

(*d*)

The walking velocity of average man, $V = 2.0\,\text{m/s}$

Coefficient of heat transfer for convection,

$$h = 8.6V^{0.53}$$
$$= 8.6\times(2.0)^{0.53}$$
$$= 12.417\ \text{W/m}^2\cdot°\text{C}$$

Coefficient of heat transfer for convection is $\boxed{h = 12.417\ \text{W/m}^2\cdot°\text{C}}$

Step 9 of 9

By using the formula for rate of heat loss, we get

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 12.417\times1.8\times(30-7)$$
$$= 514.063\,\text{W}$$

Rate of heat loss from average man is $\boxed{\dot{Q} = 514.063\,\text{W}}$

Question: 19.010P

The upper surface of a 50-cm-thick solid plate (k= 237 W/m·K) is being cooled by water with temperature of 20°C. The upper and lower surfaces of the solid plate maintained at constant temperatures of 60°C and 120°C, respectively. Determine the water convection heat transfer coefficient and the water temperature gradient at the upper plate surface.

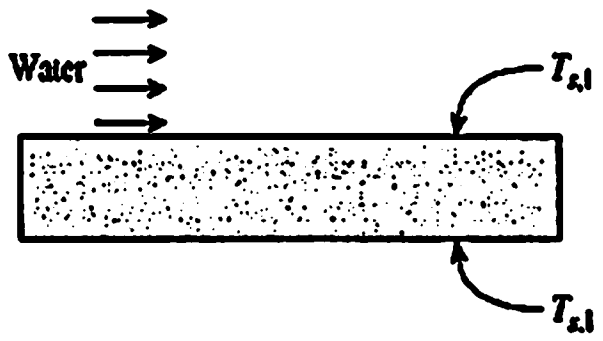


FIGURE P19-10

Step 1 of 3

Calculate the amount of heat transfer through a solid plate by using the Fourier conduction equation

$$\dot{q} = k\frac{dT}{dy}$$
$$= k\frac{(T_L - T_s)}{dy}$$

Here,

Thermal conductivity of the solid plate is k

Temperature gradient is $\frac{dT}{dy}$

Thickness of the solid plate is dy

Lower temperature of the solid plate is T_L

Upper or surface temperature of the solid plate is T_s

Amount of heat flux is $\dot{q}$

Substitute $237\,\text{W/m}\cdot\text{K}$ **for** k. $120°\text{C}$ **for** T_L, $60°\text{C}$ **for** T_s 0.5m **for** dy

$$\dot{q} = k\frac{dT}{dy}$$
$$= k\frac{(T_L - T_s)}{dy}$$
$$= 237\frac{(120-60)}{0.5}$$
$$= 28440\,\text{W/m}^2$$

Step 2 of 3

Calculate the mean film temperature of the water is

$$T_f = \frac{20+60}{2}$$
$$= 40°\text{C}$$

From the properties saturated water table

At $T_f = 40°\text{C}$

Thermal conductivity of water is $k_w = 0.631\,\text{W/m}\cdot\text{K}$

Calculate the heat transfer coefficient by using the energy balance equation

$$\dot{q} = h(T_s - T_\infty)$$

Here,

Ambient temperature of the water is T_∞

Heat transfer coefficient is h

Substitute $28440\,\text{W/m}^2$ **for** $\dot{q}$, $60°\text{C}$ **for** T_s, $20°\text{C}$ **for** T_∞ **in above equation**

$$\dot{q} = h(T_s - T_\infty)$$
$$28440 = h(60-20)$$
$$h = 711\,\text{W/m}^2\cdot\text{K}$$

Hence, heat transfer coiffing of water is $\boxed{711\,\text{W/m}^2\cdot\text{K}}$

Step 3 of 3

Calculate the temperature gradient between the water and surface of the solid plate is

$$\dot{q} = -k_w\frac{dT}{dy}$$

Substitute $28440\,\text{W/m}^2$ **for** $\dot{q}$, $0.631\,\text{W/m}\cdot\text{K}$ **for** k_w, 0.5m **for** dy

$$\dot{q} = -k_w\frac{dT}{dy}$$
$$28440 = -0.631\frac{dT}{dy}$$
$$\frac{dT}{dy} = -45071.31°\text{C/m}$$

Hence, temperature gradient is $\boxed{-45071.31°\text{C/m}}$

Question: 19.011P

Consider airflow over a plate surface maintained at a temperature of 220°C. The temperature profile of the airflow is given as

$$T(y) = T_\infty - (T_\infty - T_s)\exp\left(-\frac{V}{\alpha_{fluid}}y\right)$$

The airflow at 1 atm has a free stream velocity and temperature of 0.08 m/s and 20°C, respectively. Determine the heat flux on the plate surface and the convection heat transfer coefficient of the airflow.

Step 1 of 4

Calculate the mean film temperature of air.

$$T_f = \frac{T_s + T_\infty}{2}$$

Here, T_s is the surface temperature of the plate and T_∞ is the temperature of free stream.

Substitute $220°C$ for T_s and $20°C$ for T_∞.

$$T_f = \frac{220+20}{2}$$
$$= 120°C$$

Obtain the properties of air from Table A-15 (Properties of air at 1 atm pressure) in the text book, at $T_f = 120°C$. find the value of thermal diffusivity (α_{fluid}) and thermal conductivity (k) of air.

Thermal diffusivity is $\alpha_{fluid} = 3.565\times10^{-5}\ m^2/s$.

Thermal conductivity is $k = 0.03235\ W/m\cdot K$.

Step 2 of 4

Calculate the temperature gradient for the given profile.

$$T(y) = T_\infty - (T_\infty - T_s)\exp\left(\frac{-V}{\alpha_{fluid}}y\right)$$

Here, T_∞ is the ambient temperature, V is the free stream velocity of air, and y is the temperature displacement thickness.

Differentiate the expression with respect to y.

$$T(y) = T_\infty - (T_\infty - T_s)\exp\left(\frac{-V}{\alpha_{fluid}}y\right)$$
$$\frac{dT}{dy} = T_\infty - (T_\infty - T_s)\frac{-V}{\alpha_{fluid}}\exp\left(\frac{-V}{\alpha_{fluid}}y\right)$$

Substitute 0 for y to calculate the temperature gradient at the surface of the plate.

$$\frac{dT}{dy} = T_\infty - (T_\infty - T_s)\frac{-V}{\alpha_{fluid}}\exp\left(\frac{-V}{\alpha_{fluid}}0\right) \quad \text{..... (1)}$$
$$= T_\infty - (T_\infty - T_s)\frac{-V}{\alpha_{fluid}}$$

Step 3 of 4

Substitute $20°C$ for T_∞, $220°C$ for T_s, $0.08\ m/s$ for V and $3.565\times10^{-5}\ m^2/s$ for α in equation (1).

$$\frac{dT}{dy} = T_\infty - (T_\infty - T_s)\frac{-V}{\alpha_{fluid}}$$
$$= 20 - (20-220)\frac{-0.08}{3.565\times10^{-5}}$$
$$= -448787.8541°C/m$$

Calculate the heat flux on the plate surface.

$$\dot{q} = -k\frac{dT}{dy}$$

Substitute $0.03235\ W/m\cdot K$ for k, $448787.8541°C/m$ for $\frac{dT}{dy}$.

$$\dot{q} = -(0.03235)(-448787.8541)$$
$$= 1.45\times10^4\ W/m^2$$

Therefore, the heat flux on the plate surface is $\boxed{1.45\times10^4\ W/m^2}$.

Step 4 of 4

Calculate the heat transfer coefficient.

$$\dot{q} = h(T_s - T_\infty)$$

Here, h is the heat transfer coefficient.

Substitute $1.45\times10^4\ W/m^2$ for $\dot{q}$, $20°C$ for T_∞, and $220°C$ for T_s.

$$1.45\times10^4 = h(220-20)$$
$$h = 72.6\ W/m^2\cdot K$$

Therefore, the heat transfer coefficient is $\boxed{72.6\ W/m^2\cdot K}$.

Question: 19.012P

The convection heat transfer coefficient for a clothed person standing in moving air is expressed as h =14.8V0.69 for 0.15 < V < 1.5 m/s, where V is the air velocity. For a person with a body surface area of 1.7 m2 and an average surface temperature of 29°C, determine the rate of heat loss from the person in windy air at 10°C by convection for air velocities of (*a*) 0.5 m/s, (*b*) 1.0 m/s, and (*c*) 1.5 m/s.

Step 1 of 4

Given: -

Body surface area = 1.7m

Average surface temperature = 29°C

Step 2 of 4

Coefficient of heat transfer for convection and heat loss rate for different velocities are

$(a)\ h = 14.8V^{0.53}$

$h = 14.8(0.5\ \text{m/s})^{0.69}$

$h = 9.174\ \text{W/m}^2.^\circ\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty)$

$\dot{Q} = (9.174\ \text{W/m}^2.^\circ\text{C})(1.7\ \text{m}^2)(29-10)^\circ\text{C}$

$\dot{Q} = 296.3\ \text{W}$

Heat loss from the person is $\boxed{\dot{Q} = 296.3\ \text{W}}$

Step 3 of 4

$(b)\ h = 14.8V^{0.53}$

$h = 14.8(1.0\ \text{m/s})^{0.69}$

$h = 14.8\ \text{W/m}^2.^\circ\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty)$

$\dot{Q} = (14.8\ \text{W/m}^2.^\circ\text{C})(1.7\ \text{m}^2)(29-10)^\circ\text{C}$

$\dot{Q} = 478.0\ \text{W}$

Heat loss from the person is $\boxed{\dot{Q} = 478.0\ \text{W}}$

Step 4 of 4

$(c)\ h = 14.8V^{0.53}$

$h = 14.8(1.5\ \text{m/s})^{0.69}$

$h = 19.58\ \text{W/m}^2.^\circ\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty)$

$\dot{Q} = (19.58\ \text{W/m}^2.^\circ\text{C})(1.7\ \text{m}^2)(29-10)^\circ\text{C}$

$\dot{Q} = 632.4\ \text{W}$

Heat loss from the person is $\boxed{\dot{Q} = 632.4\ \text{W}}$

Question: 19.013P

During air cooling of oranges, grapefruit, and tangelos, the heat transfer coefficient for combined convection, radiation, and evaporation for air velocities of $0.11 < V < 0.33$ m/s is determined experimentally and is expressed as $h = 5.05\,k_{air}\,Re^{1/3}/D$, where the diameter D is the characteristic length. Oranges arc cooled by refrigerated air at 3°C and 1 atm at a velocity of 0.3 m/s. Determine (*a*) the initial rate of heat transfer from a 7-cm-diameter orange initially at 15°C with a thermal conductivity of 0.70 W/m·K, (*b*) the value of the initial temperature gradient inside the orange at the surface, and (*c*) the value of the Nusselt number.

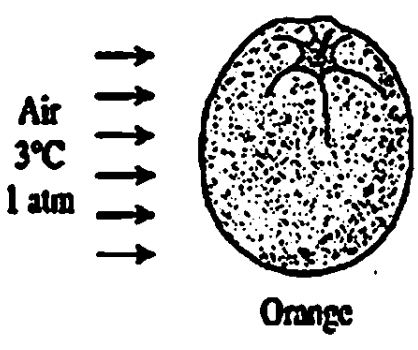


FIGURE P19-13

Step 1 of 6

Calculate the mean film temperature of the air is

$$T_f = \frac{T_\infty + T}{2}$$

Here,

Ambient air temperature is T_∞

Temperature of the orange is T

Substitute $15°C$ for T, $3°C$ for T_∞

$$T_f = \frac{15+3}{2}$$
$$= 9°C$$

From the properties of air

At $T_f = 9°C$

Use interpolation between temperature ranges of $5°C$ to $10°C$

Thermal conductivity of air is

$$k_{air} = 0.02401 + \frac{(9-5)(0.02439-0.02401)}{(10-5)}$$
$$= 0.024314\,W/m\cdot K$$

Kinematic viscosity is

$$\nu = 1.382\times10^{-5} + \frac{(9-5)(1.426-1.382)10^{-5}}{(10-5)}$$
$$= 1.41\times10^{-5}\,m^2/s$$

Step 2 of 6

Calculate the area of the orange is

$$A = \pi D^2$$

Here,

Area of the orange is A

Diameter of the orange is D

Substitute $0.07\,m$ for D in above expression

$$A = \pi D^2$$
$$= \pi(0.07)^2$$
$$= 0.01539\,m^2$$

Step 3 of 6

Calculate the Reynolds number

$$Re = \frac{VD}{\nu}$$

Here,

Reynolds number is Re

Velocity of the air is V

Substitute $0.3\,m/s$ for V, $1.41\times10^{-5}\,m^2/s$ for ν $0.07\,m$ for D,

$$Re = \frac{VD}{\nu}$$
$$= \frac{0.3\times0.07}{1.41\times10^{-5}}$$
$$= 1489.361$$

Calculate the heat transfer coefficient is

$$h = \frac{5.05\times k_{air}\times Re^{1/3}}{D}$$

Here,

Heat transfer coefficient is h

Substitute $0.024314\,W/m\cdot K$ for k_{air}. $0.07\,m$ for D, 1489.361 for Re

$$h = \frac{5.05\times k_{air}\times Re^{1/3}}{D}$$
$$= \frac{5.05\times0.024314\times(1489.361)^{1/3}}{0.07}$$
$$= 19.54\,W/m^2\cdot K$$

Step 4 of 6

a)

Calculate the initial rate of heat transfer is

$$Q = Ah(T_s - T_\infty)$$

Substitute $0.01539\,m^2$ for A. $19.54\,W/m^2\cdot K$ for h. $15°C$ for T, $3°C$ for T_∞

$$Q = Ah(T_s - T_\infty)$$
$$= 0.01539\times19.54(15-3)$$
$$= 3.61W$$

Hence, the initial rate of heat transfer is $\boxed{3.61\,W}$

Step 5 of 6

b)

Calculate the temperature gradient at surface of orange by using the energy balance equation

$$Q = -kA\frac{dT}{dy} = hA(T_s - T_\infty)$$

$$-k\frac{dT}{dy} = h(T_s - T_\infty)$$

Here,

Thermal conductivity of orange is k

Substitute $0.70\,W/m\cdot K$ for k, $19.54\,W/m^2\cdot K$ for h. $15°C$ for T, $3°C$ for T_∞

$$Q = -kA\frac{dT}{dy} = hA(T_s - T_\infty)$$

$$-k\frac{dT}{dy} = h(T_s - T_\infty)$$

Question: 19.014P

What does the friction coefficient represent in flow over a flat plate? How is it related to the drag force acting on the plate?

Step 1 of 1

Resistance to fluid flow over a flat plate is represented by friction coefficient. Friction coefficient is proportional to the drag force acting on the plate. The drag coefficient is equivalent to the mean friction coefficient for a flat surface.

Question: 19.015P

In an experiment, the local heat transfer over a flat plate was correlated in the form of local Nusselt number as expressed by the following correlation

$$Nu_x = 0.035 Re_x^{0.8} Pr^{1/3}$$

Determine the ratio of the average convection heat transfer coefficient *(h)* over the entire plate length to the local convection heat transfer coefficient *(hx)* at $x = L$.

Step 1 of 5

Given data:

Nusselt number is expressed by the following correlation,

$$Nu_x = 0.035 Re_x^{0.8} Pr^{1/3}$$

Step 2 of 5

As given the local heat transfer over flat plate (in form of Nusselt number)

$$Nu_x = 0.035 Re_x^{0.8} Pr^{1/3}$$

As we also know that heat transfer coefficient can be expressed by using the relation

$$Nu = \frac{hx}{k}$$

$$h = \frac{Nu \cdot k}{x}$$

Then the local heat transfer coefficient,

$$h_x = Nu_x \frac{k}{x}$$

$$= 0.035 Re_x^{0.8} Pr^{1/3} . \frac{k}{x}$$

Step 3 of 5

Let

Velocity of the fluid be V

Kinematic viscosity of the fluid be ν

Reynolds number we can write it as,

$$Re_x = \frac{Vx}{\nu}$$

Local heat transfer coefficient,

$$h_x = 0.035 \left(\frac{V \times x}{\nu}\right)^{0.8} Pr^{1/3} k \cdot x^{-1}$$

$$= 0.035 \left(\frac{V}{\nu}\right)^{0.8} Pr^{1/3} . k x^{0.8-1}$$

$$= Ax^{-0.2}$$

Where A is a constant and $A = 0.035 \left(\frac{V}{\nu}\right)^{0.8} Pr^{1/3} . k$

Therefore at $x = L$ the local convection heat transfer coefficient can be re written as,

$$h_{x=L} = AL^{-0.2}$$

Step 4 of 5

The average convection heat transfer coefficient over entire plate length is,

$$h = \frac{1}{L} \int_0^L h_x dx$$

$$= \frac{1}{L} \int_0^L AL^{-0.2} dx$$

$$= \frac{A}{L} \int_0^L L^{-0.2}$$

$$= \frac{A}{L} \frac{L^{-0.2+1}}{(-0.2+1)}$$

$$= \frac{A}{0.8L} L^{0.8}$$

$$= 1.25 AL^{-0.2}$$

Step 5 of 5

The ratio of the average heat transfer coefficient over the entire plate length to the local convection heat transfer coefficient is,

$$R = \frac{h}{h_L}$$

$$= \frac{1.25 AL^{-0.2}}{AL^{-0.2}}$$

$$= 1.25$$

Therefore, the ratio of the average heat transfer coefficient over the entire plate length to the local convection heat transfer coefficient is $\boxed{R = 1.25}$

Question: 19.016P

A 5-m long strip of sheet metal is being transported on a conveyor at a velocity of 5 m/s, while the coating on the upper surface is being cured by infrared lamps. The coating on the upper surface of the metal strip has an absorptivity of 0.6 and an emissivity of 0.7, while the surrounding ambient air temperature is 25°C. Radiation heat transfer occurs only on the upper surface, while convection heat transfer occurs on both upper and lower surfaces of the sheet metal. If the infrared lamps supply a constant heat flux of 5000 W/m2, determine the surface temperature of the sheet metal. Evaluate the properties of air at 80°C.

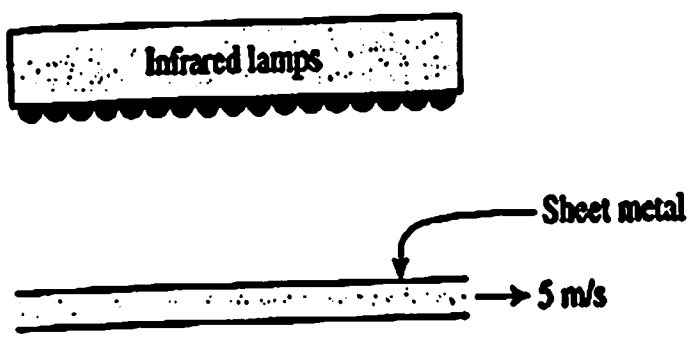


FIGURE P19-16

Step 1 of 4

Given data:

Length of sheet metal strip $L = 5\text{m}$

Velocity of conveyor $V = 5\text{m/s}$

Absorptivity of metal strip $\alpha = 0.6$

Emissivity of metal strip $\varepsilon = 0.7$

Surrounding ambient temperature,

$T_\infty = 25°\text{C}$

$= 25+273\,\text{K}$

$= 298\,\text{K}$

Heat flux supplied by infrared lamps, $q_s = 5000\,\text{W/m}^2$

Step 2 of 4

From properties of air table at temperature at $T = 80°\text{C}$, we have

Thermal conductivity $k = 0.02953\,\text{W/m}\cdot\text{K}$

Kinematic viscosity $\nu = 2.097\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number $\text{Pr} = 0.7154$

Now calculating Reynolds number

$$\text{Re}_L = \frac{VL}{\nu}$$

$$= \frac{5\times5}{2.097\times10^{-5}}$$

$$= 1.192\times10^{6}$$

Since Reynolds number is in between $5\times10^5 < \text{Re}_L < 10^7$ therefore the flow is combined laminar and turbulent flow

Step 3 of 4

Nusselt number for this combined flow can be expressed as,

[Combined flow, $0.6 \le \text{Pr} \le 60,\ 5\times10^5 \le \text{Re}_L \le 10^7$]

$$\text{Nu} = \frac{hL}{k}$$

$$= \left(0.037\text{Re}_L^{0.8} - 871\right)\text{Pr}^{1/3}$$

$$= \left(0.037\left(1.192\times10^6\right)^{0.8} - 871\right)(0.7154)^{1/3}$$

$$= 1815.721\times0.8943$$

$$= 1623.79$$

And Heat transfer coefficient,

$$h = \frac{k\cdot\text{Nu}}{L}$$

$$= \frac{1623.79\times0.02953}{5}$$

$$= 9.5901\,\text{W/m}^2\cdot\text{K}$$

Step 4 of 4

Now from the steady state condition and the energy balance equation,

$$Q_{in} - Q_{out} = 0$$

$$Q_{absorbed} - Q_{radiation} - Q_{convection} = 0$$

As radiation is only from upper surface and convection from both sides,

$$Q_{absorbed} = Q_{radiation} + 2Q_{convection}$$

$$\alpha\cdot q_s\cdot A_s = A_s\varepsilon\sigma\left(T_s^4 - T_\infty^4\right) + 2\cdot h\cdot A_s\left(T_s - T_\infty\right)$$

$$\alpha\cdot q_s = \varepsilon\sigma\left(T_s^4 - T_\infty^4\right) + 2\cdot h\left(T_s - T_\infty\right)$$

$$0.6\times5000 = 0.7\times5.67\times10^{-8}\left((T_s+273)^4 - (298)^4\right) + 2\times9.593\times(T_s - 25)$$

$$3000 = 3.969\times10^{-8}\left((T_s+273)^4 - (298)^4\right) + 19.186(T_s - 25)$$

Solving the above equation, we get $T_s = 138.412°\text{C}$

Therefore the surface temperature of the sheet metal is $\boxed{T_s = 138.412°\text{C}}$

Question:

The local atmospheric pressure in Denver, Colorado (elevation 1610 m), is 83.4 kPa. Air at this pressure and at 30°C flows with a velocity of 6 m/s over a 2.5-m × 8-m flat plate whose temperature is 120°C. Determine the rate of heat transfer from the plate if the air flows parallel to the (*a*) 8-m-long side and (*b*) the 2.5-m side.

Step 1 of 10

Given: -
Pressure of air = 83.4 kPa
Temperature of air = 30°C
Air flows at velocity = 6m/s
Size of flat plate = 2.5m × 8m
Temperature of flat plate = 120

Step 2 of 10

Pressure in atm units is

$$P = (83.4 \text{ kPa})\frac{1 \text{ atm}}{101.325 \text{ kPa}} = 0.823 \text{ atm}$$

Properties of air at 0.823 atm and average film temperature (120 + 30)/2 =75°C is taken from properties of air table

$k = 0.02917$ W/m.°C

$$\upsilon = \frac{\upsilon_{@1atm}}{P_{atm}}$$

$$\frac{(2.046\times10^{-5} \text{ m}^2/\text{s})}{0.823}$$

$$=2.486\times10^{-5} \text{ m}^2/\text{s}$$

$\text{Pr} = 0.7166$

Step 3 of 10

(a) Reynolds number when air flows parallel to 8 m side

$$\text{Re}_L = \frac{VL}{\upsilon}$$

$$= \frac{(6 \text{ m/s})(8 \text{ m})}{2.486\times10^{-5} \text{ m}^2/\text{s}}$$

$$=1.931\times10^6$$

Step 4 of 10

This is greater than value of critical Reynolds number. Therefore combined laminar and turbulent flow will be there. So heat transfer rate using nusselt number is calculated as follows

$$Nu = \frac{hL}{k}$$

$$= (0.037\text{Re}_L^{0.8} - 871)\text{Pr}^{1/3}$$

$$= [0.037(1.931\times10^6)^{0.8} - 871](0.7166)^{1/3}$$

$$= 2757$$

$$h = \frac{k}{L}Nu$$

$$= \frac{0.02917 \text{ W/m.°C}}{8 \text{ m}}(2757)$$

$$= 10.05 \text{ W/m}^2\text{.°C}$$

$$A_s = wL$$

$$= (2.5 \text{ m})(8 \text{ m})$$

$$=20 \text{ m}^2$$

Step 5 of 10

$$\dot{Q} = hA_s(T_\infty - T_s)$$

$$= (10.05 \text{ W/m}^2\text{.°C})(20 \text{ m}^2)(120\text{-}30)\text{°C}$$

$$= 18{,}096 \text{ W}$$

Rate of heat transfer is $\boxed{18.10 \text{ kW}}$

Step 6 of 10

(b) Reynolds number when air flows parallel to 2.5 m side

$$\text{Re}_L = \frac{V_\infty L}{\upsilon}$$

$$= \frac{(6 \text{ m/s})(2.5 \text{ m})}{2.486\times10^{-5} \text{ m}^2/\text{s}}$$

$$= 6.034\times10^5$$

Step 7 of 10

This is greater than value of critical Reynolds number. Therefore combined laminar and turbulent flow will be there. So heat transfer rate using nusselt number is calculated as follows

$$Nu = \frac{hL}{k}$$

$$= (0.037\text{Re}_L^{0.8} - 871)\text{Pr}^{1/3}$$

$$= [0.037(6.034\times10^5)^{0.8} - 871](0.7166)^{1/3}$$

$$= 615.1$$

$$h = \frac{k}{L}Nu$$

$$= \frac{0.029717 \text{ W/m.°C}}{2.5 \text{ m}}(615.1)$$

$$= 7.177 \text{ W/m}^2\text{.°C}$$

$$A_s = wL$$

$$= (8 \text{ m})(2.5 \text{ m})$$

$$=20 \text{ m}^2$$

Step 8 of 10

$$\dot{Q} = hA_s(T_\infty - T_s)$$

$$= (7.177 \text{ W/m}^2\text{.°C})(20 \text{ m}^2)(120\text{-}30)\text{°C}$$

$$= 12{,}919 \text{ W}$$

Step 9 of 10

$= (7.177\ W/m^2.°C)(20\ m^2)(12-30)°C$ 109017P pg 2

$= 12{,}919\ W$

Step 9 of 10

Rate of heat transfer is $\boxed{12.92\ kW}$

Step 10 of 10

Question: 19.018P pg 1

During a cold winter day, wind at 42 km/h is blowing parallel to a 6-m-high and 10-m-long wall of a house. If the air outside is at 5°C and the surface temperature of the wall is 12°C, determine the rate of heat loss from that wall by convection. What would your answer be if the wind velocity was doubled?

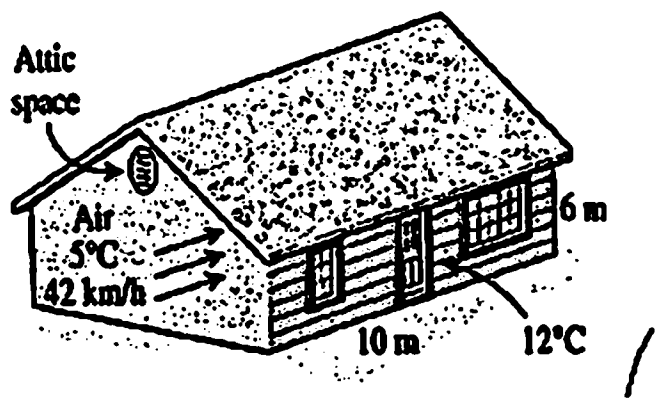


FIGURE P19-18

Step 1 of 9

Given data:

Height of wall, $H = 6\,\text{m}$

Length of wall, $L = 10\,\text{m}$

Velocity of wind flow,

$V = 42\,\text{km/h}$

$= 42 \times 1000/3600\ \text{m/s}$

$= 11.67\,\text{m/s}$

Temperature of air flow, $T_\infty = 5°\text{C}$

Surface temperature of the wall, $T_s = 12°\text{C}$

Step 2 of 9

By using the formula for film temperature, we get

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{12+5}{2}$$

$$= 8.5°\text{C}$$

From table properties of air, by using interpolation method the values between $5°\text{C} - 10°\text{C}$

Thermal conductivity,

$$k = 0.02401 + \frac{0.02439 - 0.02401}{(10-5)}(8.5-5)$$

$$k = 0.024276\,\text{W/m}\cdot\text{K}$$

Kinematic viscosity,

$$\nu = 1.382\times10^{-5} + \frac{(1.426-1.382)10^{-5}}{(10-5)(8.5-5)}$$

$$\nu = 1.4128\times10^{-5}\,\text{m}^2/\text{s}$$

Step 3 of 9

From table properties of air, by using interpolation method the values between $5°\text{C} - 10°\text{C}$

Prandtl number,

$$\text{Pr} = 0.7350 + \frac{0.7336 - 0.7350}{(10-5)}(8.5-5)$$

$$\text{Pr} = 0.73402$$

The Reynolds number of the flow,

$$\text{Re}_L = \frac{VL}{\nu}$$

$$= \frac{11.67\times10}{1.4128\times10^{-5}}$$

$$= 8.26\times10^6$$

Since Reynolds number is in between $5\times10^5 < \text{Re}_L < 10^7$ therefore the flow is combined laminar and turbulent flow

Step 4 of 9

Nusselt number for this combined flow can be expressed as,

[Combined flow, $0.6 \le \text{Pr} \le 60,\ 5\times10^5 \le \text{Re}_L \le 10^7$]

$$\text{Nu} = \frac{hL}{k}$$

$$= \left(0.037\text{Re}_L^{0.8} - 871\right)\text{Pr}^{1/3}$$

$$= \left[0.037\left(8.26\times10^6\right)^{0.8} - 871\right](0.73402)^{1/3}$$

$$= 10617.37$$

Step 5 of 9

Then heat transfer coefficient,

$$\frac{hL}{k} = 10617.37$$

$$h = \frac{10617.37\times k}{L}$$

$$= \frac{10617.37\times0.024276}{10}$$

$$= 25.77\ \text{W/m}^2\cdot\text{K}$$

Step 6 of 9

Heat transfer surface area,

$$A_s = h\times L$$

$$= 10\times6$$

$$= 60\,\text{m}^2$$

Rate of heat loss by convection is,

$$Q = hA_s(T_s - T_\infty)$$

$$= 25.77\times60\times(12-5)$$

$$= 10823.4\,\text{W}$$

$$= 10.8\,\text{kW}$$

$$Q = hA_s(T_s - T_\infty)$$
$$= 25.77 \times 60 \times (12-5)$$
$$= 10823.4\,\text{W}$$
$$= 10.8\,\text{kW}$$

Therefore, rate of heat loss by convection is, $\boxed{Q = 10.8\,\text{kW}}$

Step 7 of 9

If the wind velocity is doubled

$$V = 2V$$
$$= 84\,\text{km/h}$$
$$= 84 \times \frac{1000}{3600}$$
$$= 23.33\,\text{m/s}$$

Reynolds number,

$$\text{Re}_L = \frac{VL}{\nu}$$
$$= \frac{23.33 \times 10}{1.4128 \times 10^{-5}}$$
$$= 1.65 \times 10^7$$

Step 8 of 9

Assuming Nusselt number is,

$$Nu = \frac{hL}{k}$$
$$= \left(0.037\,\text{Re}_L^{0.8} - 871\right)\text{Pr}^{1/3}$$
$$= \left[0.037\left(1.65 \times 10^7\right)^{0.8} - 871\right](0.73402)^{1/3}$$
$$= 19048.96$$

Heat transfer coefficient,

$$h_{2V} = \frac{\text{Nu} \cdot k}{L}$$
$$= \frac{19048.96 \times 0.024276}{10}$$
$$= 46.243\,\text{W/m}^2 \cdot \text{K}$$

Step 9 of 9

Then the rate of heat loss from the wall by convection is,

$$Q_{2V} = h_{2V} A_s (T_s - T_\infty)$$
$$= 46.243 \times 60 \times (12-5)$$
$$= 19422.06\,\text{W}$$
$$= 19 \cdot 422\,\text{kW}$$

Therefore, rate of heat loss by convection when velocity doubled is, $\boxed{Q_{2V} = 19 \cdot 422\,\text{kW}}$

Question: 19.019P pg 1

Reconsider Prob. 19-18. Using EES (or other) software, investigate the effects of wind velocity and outside air temperature on the rate of heat loss from the wall by convection. Let the wind velocity vary from 10 km/h to 80 km/h and the outside air temperature from 0°C to 10°C. Plot the rate of heat loss as a function of the wind velocity and of the outside temperature, and discuss the results.

Step 1 of 8

The given problem can be solved using EES software by following the below steps.

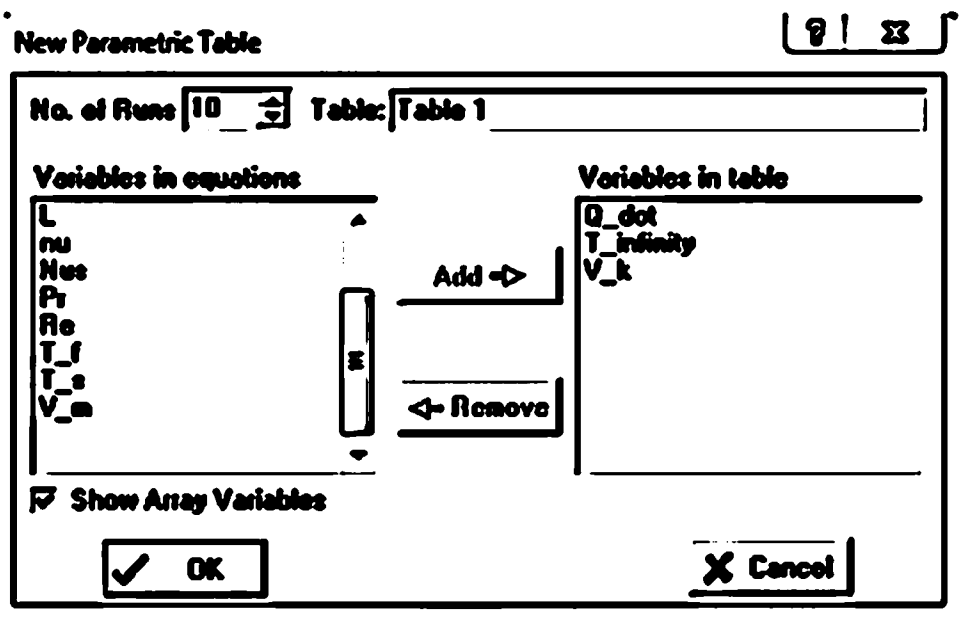
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
H_1=6[m]                                        "Height of the wall"
L=10[m]                                         "Length of the wall"
"V_k is varied from 10[km/hr] to 80[km/hr]"     "Velocity of air"
V_m=V_k*convert(km/hr,m/s)
"T_infinity is varied from 0[C] to 10[C]"       "Temperature of air"
T_s=12[C]                                       "Temperature of wall"

T_f=(T_s+T_infinity)/2
k=Conductivity(Air,T=T_f)                       "Properties of air at T_f temperature"
nu=Viscosity(Air,T=T_f)/Density(Air,T=T_f,P=1[atm]*convert(atm,kPa))
Pr=Prandtl(Air,T=T_f)
Re=V_m*L/nu                                     "Reynolds number"
Nus=(0.037*Re^0.8-871)*Pr^(1/3)
Nus=h*L/k
A_s=H_1*L
Q_dot=h*A_s*(T_s-T_infinity)*convert(W,kW)
```

Step 2 of 8

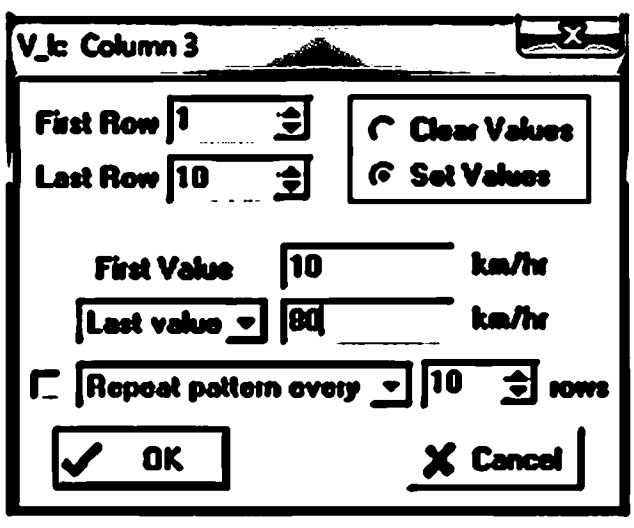
Now to solve the equation for a range of wind velocities and outside air temperatures, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 10.
Select variables in equations as "Q_dot", "T_infinity", and "V_k" and press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 8

First the wind velocity is varied at constant outside air temperature of $T_\infty = 5°C$

To enter the values in the "V_k" column, click on the drop tab in "V_k" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 8

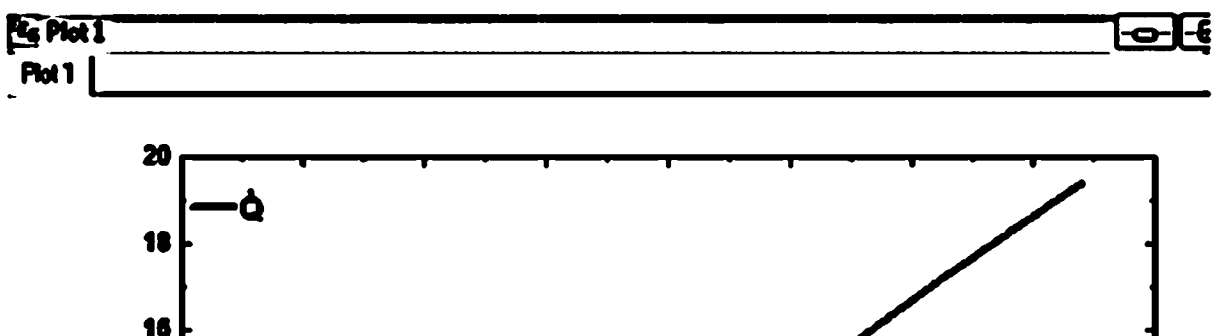
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

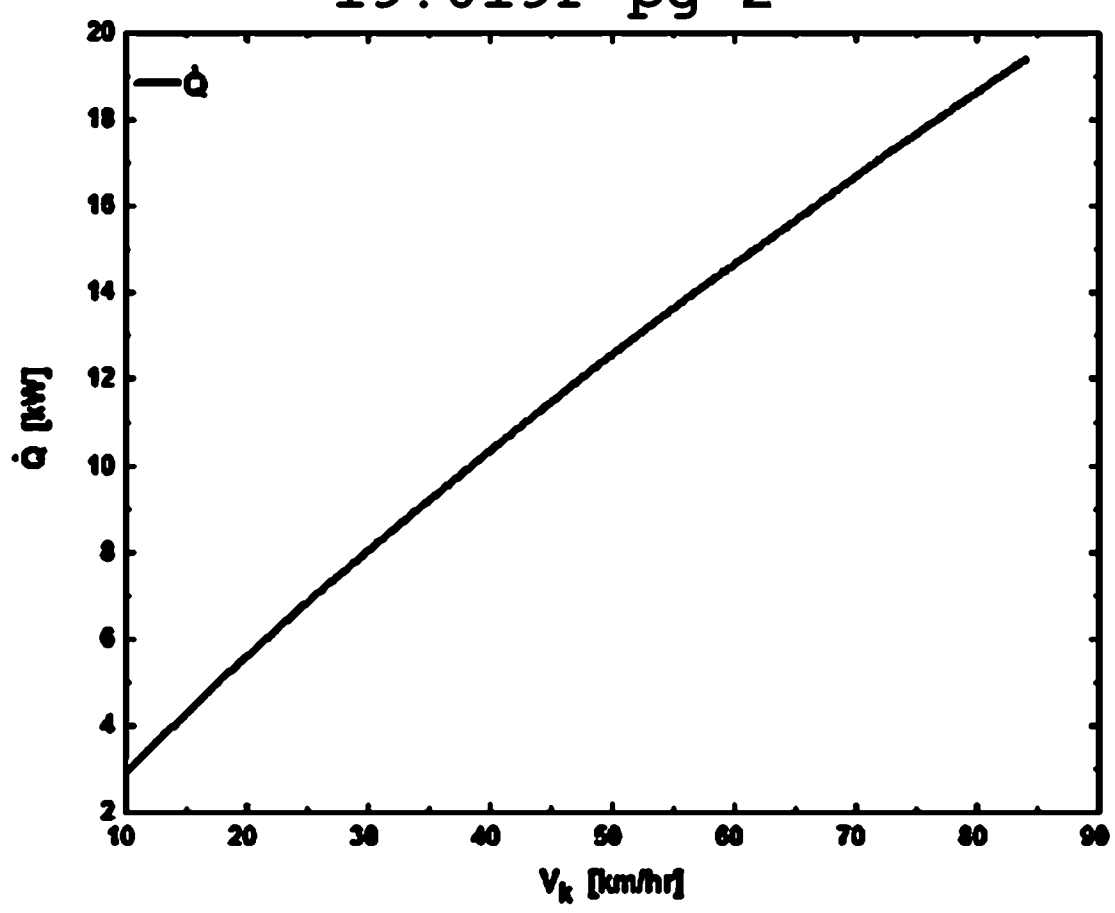
Table 1

$\blacktriangleright_{1..10}$	1 $\dot{Q}$ [kW]	2 T_∞ [C]	3 V_k [km/hr]
Run 1	2.885	5	10
Run 2	5.039	5	17.78
Run 3	7.006	5	25.56
Run 4	8.855	5	33.33
Run 5	10.62	5	41.11
Run 6	12.32	5	48.89
Run 7	13.96	5	56.67
Run 8	15.56	5	64.44
Run 9	17.12	5	72.22
Run 10	19.42	5	84

Step 5 of 8

Inorder to plot the rate of heat loss as a function of wind velocity, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the rate of heat loss as a function of wind velocity:



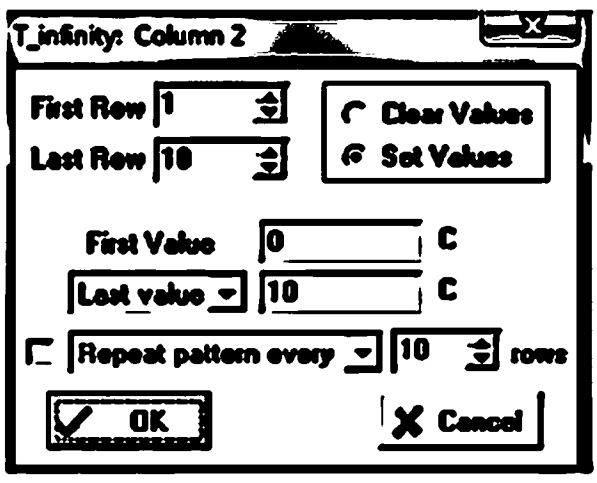


From the above plot we can interpret that as the wind velocity increases, the rate of heat loss also increases.

Step 6 of 8

Now the outside temperature of the air is varied at constant wind velocity of V_k = 42 km/hr

To enter the values in the "T_infinity" column, click on the drop tab in "T_infinity" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 7 of 8

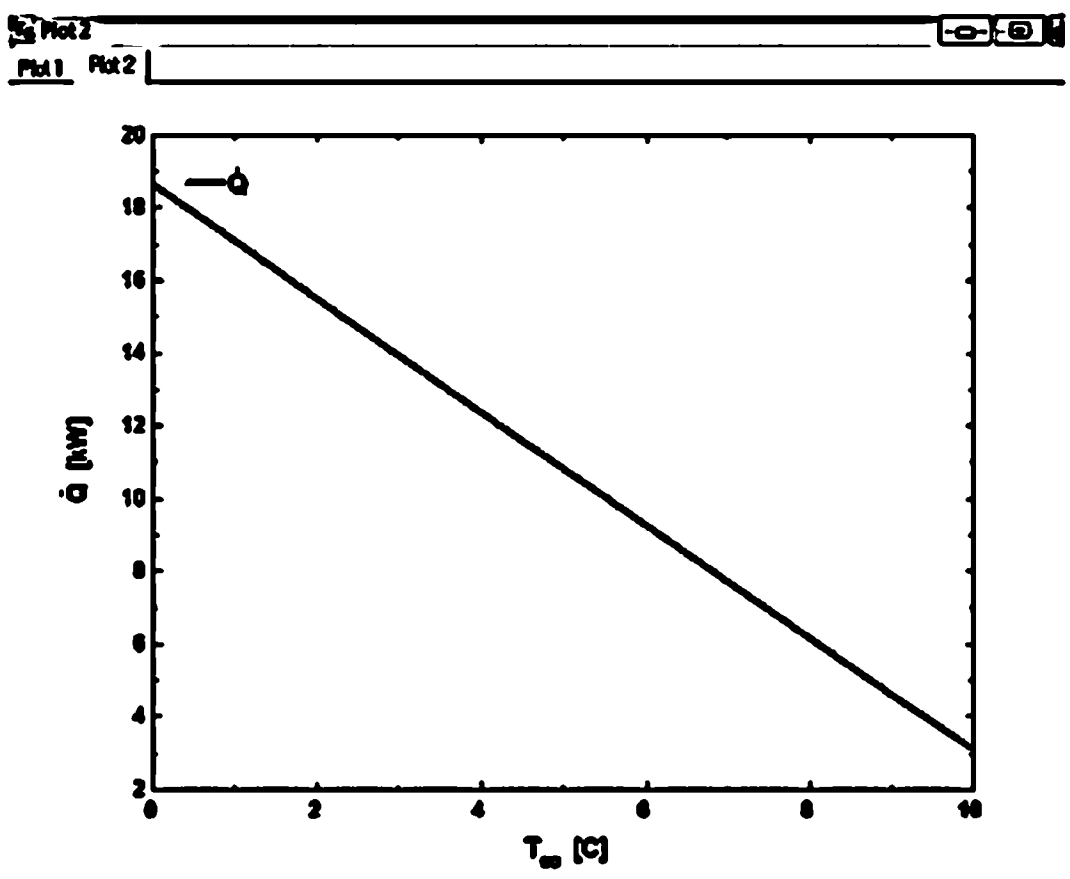
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1_10	1 Q̇ [kW]	2 T∞ [C]	3 V_k [km/hr]
Run 1	18.65	0	42
Run 2	16.9	1.111	42
Run 3	15.16	2.222	42
Run 4	13.42	3.333	42
Run 5	11.68	4.444	42
Run 6	9.95	5.556	42
Run 7	8.224	6.667	42
Run 8	6.502	7.778	42
Run 9	4.785	8.889	42
Run 10	3.072	10	42

Step 8 of 8

Inorder to plot the rate of heat loss as a function of outside air temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the rate of heat loss as a function of outside air temperature:



From the above plot we can interpret that as the temperature of the outside air increases, the rate of heat loss decreases.

Question: 19.020P

Water at 43.3°C flows over a large plate at a velocity of 30.0 cm/s. The plate is 1.0 m long (in the flow direction), and its surface is maintained at a uniform temperature of 10.0°C. Calculate the steady rate of heat transfer per unit width of the plate.

Step 1 of 5

Given: - Temperature of water $T_\infty = 43.3°C$

Velocity of air $V = 30.0$ cm/s

Length of plate $L = 1.0$ m

Temperature of surface $T_w = 10°C$

Step 2 of 5

Properties of water at 1 atm and average film temperature $\frac{(10+43.3)}{2} = 26.65^0C$ is taken from properties of saturated water table

Density, $\rho = 996.6 kg/m^3$

Thermal conductivity $k = 0.610$ W/m°C

Dynamic viscosity, $\mu = 0.854 \times 10^{-3}$ kg/ms

Prandtl number $Pr = 5.85$

Step 3 of 5

Reynolds number when air flows parallel to 1 m side

$$Re_L = \frac{VL\rho}{\mu}$$

$$= \frac{[0.3 m/s](1 m)(996.6 kg/m^3)}{0.854 \times 10^{-3} kg/m \cdot s}$$

$$Re_L = 3.501 \times 10^5$$

Since $Re_L < 5 \times 10^5$, the flow is laminar.

Step 4 of 5

For flow over a flat plate,
Nusselt number will be calculated as

$$Nu = \frac{hL}{k}$$

$$= 0.664 Re_L^{1/2} Pr^{1/3} = 0.664(3.501 \times 10^5)^{1/2}(5.85)^{1/3}$$

$$Nu = 707.9$$

$$h = \frac{k}{L} Nu$$

$$= \frac{0.610 \text{ W/m °C}}{1 m}(707.9)$$

$$h = 431.8 \text{ W/m}^2 \text{ °C}$$

Step 5 of 5

Steady rate of heat transfer per unit width will be

$$\dot{Q} = hA_s(T_\infty - T_s)$$

$$= (431.8 \text{ W/m}^2 \text{ °C})(1 m^2)(43.3\text{-}10)°C$$

$$\boxed{\dot{Q} = 14379.93 \text{ W}}$$

Question: 19.021P

Warm air is blown over the inner surface of an automobile windshield to defrost ice accumulated on the outer surface of the windshield. Consider an automobile windshield (kW= 0.8 Btu/h·ft·R) with an overall height of 20 in. and thickness of 0.2 in. The outside air (1 atm) ambient temperature is 8°F and the average airflow velocity over the outer windshield surface is 50 mph, while the ambient temperature inside the automobile is 77°F. Determine the value of the convection heat transfer coefficient, for the warm air blowing over the inner surface of the windshield, necessary to cause the accumulated ice to begin melting. Assume the windshield surface can be treated as a flat plate surface.

We don't have the solution to this problem yet.

Get help from a Chegg subject expert.

Ask an expert

Question: 19.022P

Hot carbon dioxide exhaust gas at 1 atm is being cooled by flat plates. The gas at 220°C flows in parallel over the upper and lower surfaces of a 1.5 m-long flat plate at a velocity of 3 m/s. If the flat plate surface temperature is maintained at 80°C, determine (*a*) the local convection heat transfer coefficient at 1 m from the leading edge, (*b*) the average convection heat transfer coefficient over the entire plate, and (*c*) the total heat flux transfer to the plate.

Step 1 of 7

Given data:

Length of flat plate, $L = 1.5\text{m}$

Velocity of flow, $V = 3\text{m/s}$

Temperatures of hot carbon dioxide exhaust gas, $T_\infty = 220°\text{C}$

Temperature of flat surface, $T_s = 80°\text{C}$

Step 2 of 7

By using the formula for film temperature, we get

$$T_f = \left(\frac{T_s + T_\infty}{2}\right)$$

$$= \frac{(220+80)}{2}$$

$$= 150°\text{C}$$

From the table properties of carbon dioxide, at film temperature $T_f = 150°\text{C}$

Thermal conductivity, $k = 0.02652\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.627\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7445$

Step 3 of 7

(a)

First of all we will calculate the Reynolds number at $x = 1\text{m}$

$$\text{Re}_x = \frac{Vx}{\nu}$$

$$= \frac{3\times1}{\left(1.627\times10^{-5}\right)}$$

$$\text{Re}_x = 184388.45$$

Since $\text{Re}_x < 5\times10^5$ therefore the flow is laminar

Nusselt number for laminar flow can be expressed as,

$$\text{Nu}_x = 0.332\text{Re}_x^{0.5}\text{Pr}^{1/3}$$

$$= 0.332(184388.45)^{0.5}(0.7445)^{1/3}$$

$$= 129.209$$

Step 4 of 7

Now, local heat transfer coefficient,

$$\text{Nu}_x = \frac{h_x \cdot x}{k}$$

$$h_x = \frac{\text{Nu}_x \cdot k}{x}$$

$$= \frac{129.209\times0.02652}{1}$$

$$= 3.427\ \text{W/m}^2\cdot\text{K}$$

Therefore, the local convection coefficient is $\boxed{h_x = 3.427\ \text{W/m}^2\cdot\text{K}}$

Step 5 of 7

(b)

Reynolds number for entire length of the plate is

$$\text{Re}_L = \frac{VL}{\nu}$$

$$= \frac{3\times1.5}{1.627\times10^{-5}}$$

$$= 2.76582\times10^5$$

Since $\text{Re}_L < 5\times10^5$ therefore the flow is laminar

So, the average Nusselt number relation for heat transfer coefficient.

$$\text{Nu}_L = 0.664\text{Re}_L^{0.5}\text{Pr}^{1/3}$$

$$= 0.664\left(2.76582\times10^5\right)^{0.5}(0.7445)^{1/3}$$

$$= 316.496$$

Step 6 of 7

Therefore average heat transfer coefficient,

$$\text{Nu}_L = \frac{hL}{k}$$

$$h = \frac{\text{Nu}_L \cdot k}{L}$$

$$= \frac{316.496\times0.02652}{1.5}$$

$$= 5.595\ \text{W/m}^2\cdot\text{K}$$

Therefore, average heat transfer coefficient is $\boxed{h = 5.595\ \text{W/m}^2\cdot\text{K}}$

Step 7 of 7

(c)

Now, the total heat flux to the flat plate surface for both upper and lower portion can be expressed as,

$$\dot{q}_s = 2\times h\left(T_\infty - T_s\right)$$

$$= 2\times5.595(220-80)$$

$$= 1566.6\ \text{W/m}^2$$

The total heat flux to the flat plate surface for both upper and lower portion is,

$\boxed{\dot{q}_s = 1566.6\ \text{W/m}^2}$

Question: 19.023P

Hot engine oil tit 150°C is flowing in parallel over a flat plate at a velocity of 2 m/s. Surface temperature of the 0.5-m long flat plate is constant at 50°C. Determine (*a*) the local convection heat transfer coefficient at 0.2 m from the leading edge and the average convection heat transfer coefficient, and (*b*) repeat part (*a*) using the Churchill and Ozoe (1973) relation.

Step 1 of 8

Given data:

Temperature of hot engine oil, $T_s = 150°C$

Velocity of oil flow, $V = 2 m/s$

Length of flat plate, $L = 0.5m$

Temperature of flat plate, $T_\infty = 50°C$

Step 2 of 8

By using the formula for film temperature, we get

$$T_f = (T_s + T_\infty)/2$$
$$= (50+150)/2$$
$$= 100°C$$

From table properties of engine oil at film temperature, $T_f = 100°C$

Kinematic viscosity $\nu = 2.046\times10^{-5} m^2/s$

Thermal conductivity $k = 0.1367 W/m\cdot K$

Prandtl number $Pr = 279.1$

Step 3 of 8

(a)

Now, we will calculate Reynolds number, Re at $x = 0.2m$

$$Re_x = \frac{Vx}{\nu}$$
$$= \frac{2\times0.2}{2.046\times10^{-5}}$$
$$Re_x = 19550.34$$

Since $Re_x < 5\times10^5$ therefore the flow is laminar

Nusselt number relation for local convection heat transfer coefficient is:

$$Nu_x = 0.332 Re_x^{0.5} Pr^{1/3}$$
$$= 0.332(19550.34)^{0.5}(279.1)^{1/3}$$
$$Nu_x = 303.4$$

Step 4 of 8

Local heat transfer coefficient is,

$$Nu_x = \frac{h_x \cdot x}{k}$$
$$h_x = \frac{Nu_x \cdot k}{x}$$
$$= \frac{303.4\times0.1307}{0.2}$$
$$= 207.35 W/m^2\cdot K$$

Therefore local heat transfer coefficient, $\boxed{h_x = 207.35 W/m^2\cdot K}$

Step 5 of 8

Now for Average heat transfer coefficient:

$$Re_L = \frac{VL}{\nu}$$
$$= \frac{2\times0.5}{2.046\times10^{-5}}$$
$$Re_L = 48875.9$$

Since $Re_L < 5\times10^5$ therefore the flow is laminar

Nusselt number relation for average convection heat transfer coefficient is:

$$Nu_L = \frac{hL}{k}$$
$$= 0.664 Re_L^{0.5} Pr^{1/3}$$
$$= 0.664\times(48875.9)^{0.5}(279.1)^{1/3}$$
$$Nu_L = 959.33$$

Step 6 of 8

Average convection heat transfer coefficient is,

$$h = \frac{Nu_L \cdot k}{L}$$
$$= \frac{959.33\times0.1367}{0.5}$$
$$= 262.28 W/m^2\cdot K$$

Therefore Average convection heat transfer coefficient, $\boxed{h = 262.28 W/m^2\cdot K}$

Step 7 of 8

(b)

Now by using Churchill and Ozoe relation

Taking Nusselt number relation for local convection heat transfer coefficient:

At $x = 0.2m$ from loading edge

$$Nu_x = \frac{0.3387 Pr^{1/3} Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$

$$\frac{h_x \cdot x}{k} = \frac{0.3387 Pr^{1/3} Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$

$$h_x = \frac{0.3387 Pr^{1/3} Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}} \times \frac{k}{x}$$

$$= \frac{0.3387(279.1)^{1/3}(19550.34)^{0.5}}{\left[1+(0.0468/279.1)^{2/3}\right]^{0.25}} \times \frac{0.1367}{0.2}$$

$$= \frac{309.489}{1.00075} \times \frac{0.1367}{0.2}$$

$$= 211.377 W/m^2\cdot K$$

Therefore local heat transfer coefficient is $\boxed{h_x = 211.377 W/m^2\cdot K}$

Step 8 of 8

The average convection heat transfer coefficient for the entire plate will be

$$h = \frac{1}{L}\int_0^L h_x dm$$

$$= \frac{1}{L}\int_0^L \frac{0.3387 Pr^{1/3} Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}} \cdot \frac{k}{x}$$

$$= \frac{0.3387 Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\cdot k}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}\cdot L}\int_0^L \frac{x^{1/2}}{x} \qquad \left[\text{since } Re_x = \frac{Vx}{\nu}\right]$$

$$= \frac{0.3387 Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\cdot k}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}\cdot L}\left[\frac{x^{-1/2+1}}{-1/2+1}\right]_0^L$$

$$= \frac{0.3387 Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\cdot k}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}\cdot L} 2L^{1/2}$$

$$= \frac{0.3387(279.1)^{1/3}\times\left(\frac{2}{2.046\times10^{-5}}\right)^{1/2}\times0.1367\times2\times(0.5)^{1/2}}{\left[1+(0.0468/279.1)^{2/3}\right]^{1/4}\times0.5}$$

$$= \frac{2.2134\times312.652\times0.1933}{0.5}$$

$$= 267.536 W/m^2\cdot K$$

Therefore, by using the Churchill and Ozoe relation the average convection heat transfer coefficient is $\boxed{h = 267.536 W/m^2\cdot K}$

Question: 19.024P

Mercury at 25°C flows over a 3-m-long and 2-m-wide flat plate maintained at 75°C with a velocity of 0.8 m/s. Determine the rate of heat transfer from the entire plate.

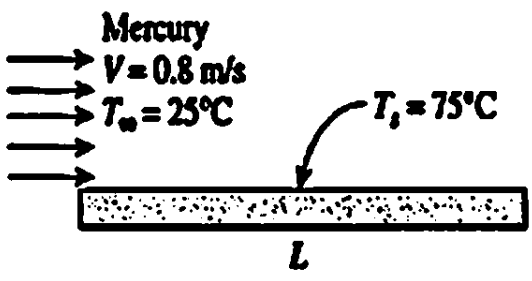


FIGURE P19-24

Step 1 of 6

Find the average film temperature by using the expression.

$$T_f = \frac{T_s + T_\infty}{2}$$

Here, T_s is temperature of the plate and T_∞ is the mercury temperature.

Substitute $75°C$ for T_s and $25°C$ for T_∞

$$T_f = \frac{75+25}{2}$$
$$= 50°C$$

Obtain the properties of mercury at $T_f = 50°C$ from properties of liquid metals.

Thermal conductivity, $k = 8.83632\ W/m°C$.

Prandtl number $Pr = 0.0223$

Kinematic viscosity, $\nu = 1.056\times10^{-7}\ m^2/s$

Step 2 of 6

Write the expression for the local Nusselt number for liquid metals.

$$Nu_x = \frac{h_x L}{k} = 0.565(Re_x\ Pr)^{1/2}$$

$$h_x = \frac{k}{L}0.565(Re_x\ Pr)^{1/2}$$

Here, h_x is the local heat transfer coefficient, L is the length of the plate.

Calculate the average heat transfer coefficient for the entire surface of the plate by using the equation.

$$h = \frac{1}{L}\int_0^L h_x dx$$

$$h = \frac{1}{L}\frac{k}{L}0.565(Pr)^{1/2}\left(\frac{V}{\nu}\right)^{1/2} 2L^{3/2}$$

$$\frac{hL}{k} = 1.13(Pr)^{1/2}\left(\frac{VL}{\nu}\right)^{1/2}$$

$$Nu = 1.13(Re_L\ Pr)^{1/2}$$

Step 3 of 6

Find the Reynolds number by using the equation.

$$Re_L = \frac{VL}{\nu}$$

Here, L is length of the plate, V is velocity of mercury.

Substitute 3 m for L and 0.8 m/s for V.

$$Re_L = \frac{0.8\times3}{1.056\times10^{-7}}$$
$$= 2.272\times10^7$$

The flow is turbulent, since Reynold's number $Re_L > 10^7$.

Step 4 of 6

Find the Nusselt number by using the above expression.

$$Nu = 1.13(Re_L\ Pr)^{1/2}$$
$$= 1.13[(2.272\times10^7)(0.0223)]^{1/2}$$
$$= 804.33$$

Step 5 of 6

Find the heat transfer coefficient by using the equation.

$$Nu = \frac{hL}{k}$$

$$804.33 = \frac{h\times3}{8.83632}$$

$$h = 2369.11\ W/m^2\cdot K$$

Find the surface area of the plate by using the equation.

$$A_s = wL$$

Here, w is width of the plate.

Substitute 2 m for w .

$$A_s = 2\times3$$
$$= 6\ m^2$$

Step 6 of 6

Determine the rate of heat transfer by using the equation.

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 2369.11\times6\times(75-25)$$
$$= 710733\ W$$
$$= 710.7\ kW$$

Therefore, rate of heat transfer from the entire plate is $\boxed{710.7\ kW}$.

Question:

Parallel plates form a solar collector that covers a roof, as shown in the figure. The plates are maintained at 15°C, while ambient air at 10°C flows over the roof with V= 4 m/s. Determine the rate of convective heat loss from (*a*) the first plate and (*b*) the third plate.

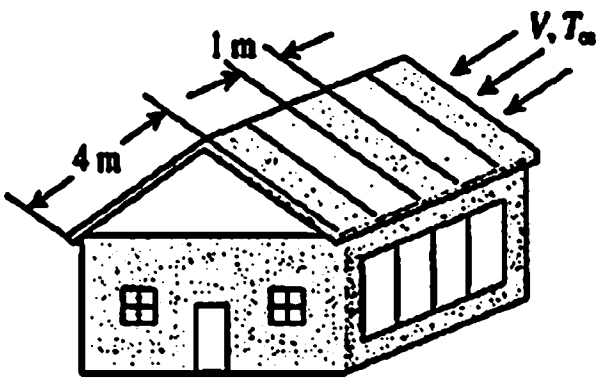
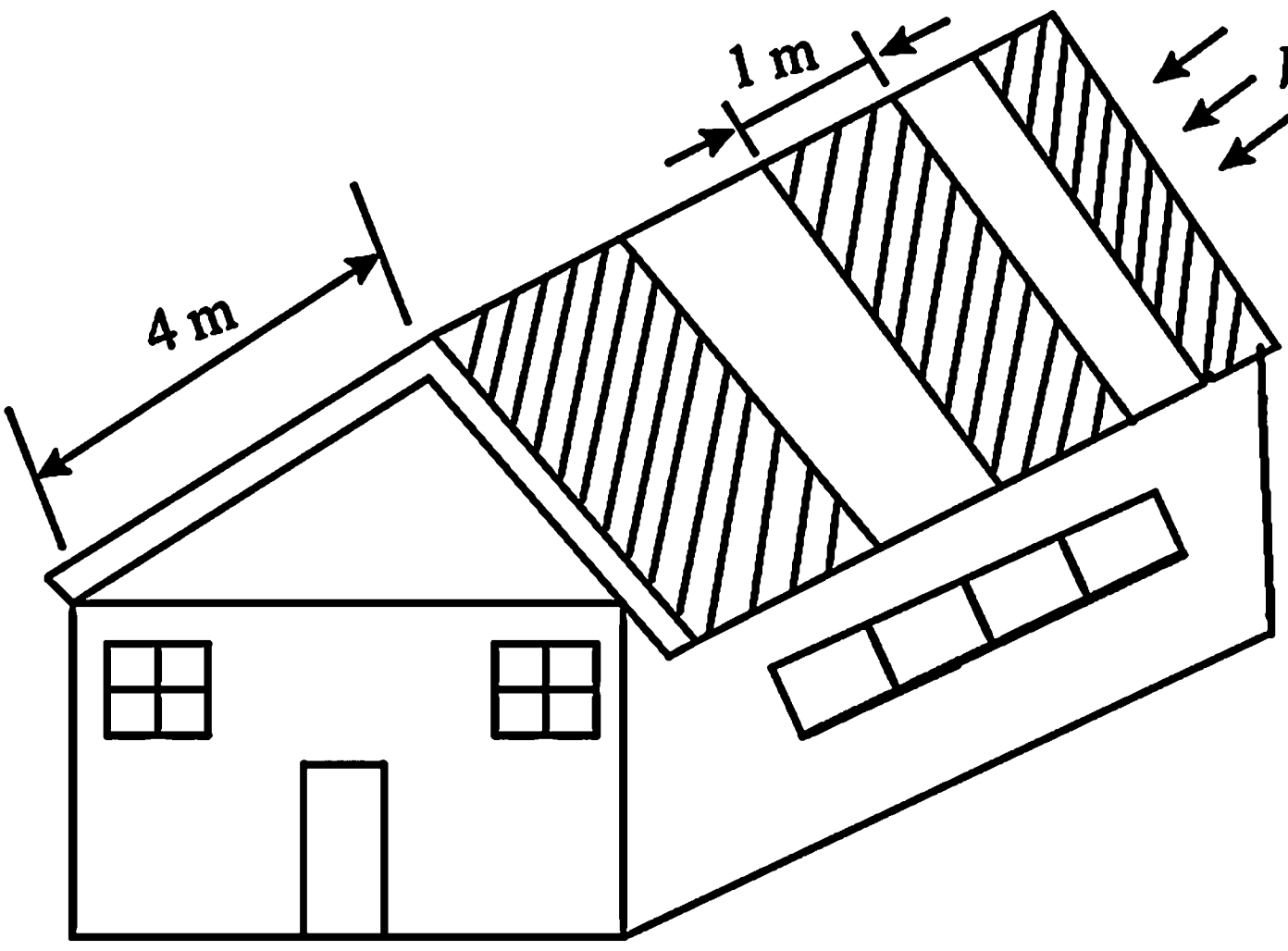


FIGURE P19-25

Step 1 of 9

Parallel plates form a solar collector that covers a roof is as follows.



Step 2 of 9

Consider Steady state operations exist and Radiation is ignored.

Consider the Temperature of the plates, $T_s = 15°C$.

Consider the Ambient air temperature, $T_\infty = 10°C$

Take the properties of air at film temperature, $\left(T_f\right)$.

The film temperature is given by equation as follows,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

Substitute the corresponding values in the above equation.

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(15+10)}{2}$$
$$= 12.5°C$$

Find the properties of air at 1 atm pressure and film temperature.

The Kinematic viscosity at film temperature is mean value of the kinematic viscosities of temperature between $10°C$ and $15°C$. The kinematic viscosity is as follows.

$$\nu = \frac{(1.470+1.426)\times 10^{-5}}{2}$$
$$= 1.448\times 10^{-5}\ m^2/s$$

Step 3 of 9

The Thermal conductivity at film temperature is mean value of the Thermal conductivities of temperature between $10°C$ and $15°C$. The Thermal conductivity is as follows.

$$k = \frac{0.02439 + 0.02476}{2}$$
$$= 0.024575\ W/m\cdot K$$

The Prandtl number at film temperature is mean value of the Prandtl numbers of temperature between $10°C$ and $15°C$. The Prandtl number is as follows.

$$Pr = \frac{0.7323+0.7336}{2}$$
$$= 0.73295$$

Step 4 of 9

Calculate the critical length first for the $\mathrm{Re}_{critical} = 5\times 10^5$.

The Reynolds number is given by the equation as follows.

$$\mathrm{Re} = \frac{Vx}{\nu}$$

Here, V is the velocity of the flow, x is the distance from the leading edge to point of interest, and ν is the kinematic viscosity of the flow.

The critical Reynolds number is given by the equation as follows.

$$\mathrm{Re}_{critical} = \frac{Vx_{cr}}{\nu}$$
$$x_{cr} = \frac{\mathrm{Re}_{critical}\times \nu}{V}$$
$$= \frac{5\times 10^5 \times 1.448\times 10^{-5}}{4}$$

$$x_{cr} = \frac{\mathrm{Re}_{critical} \times \nu}{V}$$

$$= \frac{5\times10^{5} \times 1.448\times10^{-5}}{4}$$

$$x_{cr} = 1.81$$

(a)

Calculating the Reynolds number for the first plate.

$$\mathrm{Re}_1 = \frac{VL_1}{\nu}$$

Substitute $1\ \mathrm{m}$ for L_1 and other corresponding values in the above equation.

$$\mathrm{Re}_1 = \frac{VL_1}{\nu}$$

$$= \frac{4\times1}{1.448\times10^{-5}}$$

$$= 2.76\times10^{5}$$

Nusselt number is given by the expression as follows.

$$\mathrm{Nu} = 0.664\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{1/3}$$

Here, Nu is the Nusselt number.

Calculating the Nusselt number for the first plate.

$$\mathrm{Nu}_1 = 0.664\,\mathrm{Re}_1^{0.5}\,\mathrm{Pr}^{1/3}$$

$$= 0.664\left(2.76\times10^{5}\right)^{0.5}(0.73295)^{1/3}$$

$$= 314.519$$

Step 5 of 9

The Nusselt number is also given by the expression as follows.

$$\mathrm{Nu} = \frac{hL}{k}$$

Here, h is the coefficient of heat transfer and L is the length of the plate.

The Nusselt number for the first plate is,

$$\mathrm{Nu}_1 = \frac{h_1 L}{k}$$

Substitute the corresponding values in the above equation.

$$\mathrm{Nu}_1 = \frac{h_1 L}{k}$$

$$314.519 = \frac{h_1 \times 1}{0.024575}$$

$$h_1 = 7.73\ \mathrm{W/m^2{\cdot}^{\circ}C}$$

Surface Area of the plate is, $A = lb$

Here, l is the length of the plate and b is the width of the plate.

Substitute $1\mathrm{m}^2$ for l and $4\ \mathrm{m}^2$ for b in the above equation of area.

$$A_{s_1} = lb$$

$$= 1\times4$$

$$= 4\mathrm{m}^2$$

The convection heat transfer is as follows.

$$Q = h_1 A_{s_1}\left(T_s - T_\infty\right)$$

Substitute the corresponding values of plate 1 in the above equation.

$$Q = h_1 A_{s_1}\left(T_s - T_\infty\right)$$

$$= 7.73\times4(15-10)$$

$$= 154.6\ \mathrm{W}$$

Therefore the convection heat transfer from the first plate is $\boxed{154.65\ \mathrm{W}}$

Step 6 of 9

(b)

Calculating the Reynolds number for the second plate.

$$\mathrm{Re}_2 = \frac{VL_2}{\nu}$$

Substitute $2\ \mathrm{m}$ for L_2 and other corresponding values in the above equation.

$$\mathrm{Re}_2 = \frac{VL_2}{\nu}$$

$$= \frac{4\times2}{1.448\times10^{-5}}$$

$$= 5.52\times10^{5}$$

Because the Reynolds number is greater than 5×10^{5} the Nusselt number is,

$$\mathrm{Nu} = \left(0.037\,\mathrm{Re}^{0.8} - 871\right)\mathrm{Pr}^{1/3}$$

Substitute the corresponding values of plate 2 in the above equation.

$$\mathrm{Nu}_2 = \left(0.037\,\mathrm{Re}^{0.8} - 871\right)\mathrm{Pr}^{1/3}$$

$$= \left(0.037\left(5.52\times10^{5}\right)^{0.8} - 871\right)(0.73295)^{1/3}$$

$$= 523.2$$

Step 7 of 9

The Nusselt number for the second plate is,

$=323.2$

Step 7 of 9

The Nusselt number for the second plate is,

$$\mathrm{Nu}_2 = \frac{h_2 \cdot L_2}{k}$$

Substitute the corresponding values in the above equation.

$$h_2 = \frac{\mathrm{Nu}_2 \cdot k}{L_2}$$
$$= \frac{444.79 \times 0.024575}{2}$$
$$= 5.465\ \mathrm{W/m^2{}^\circ C}$$

Calculating the Reynolds number for third plate.

$$\mathrm{Re}_3 = \frac{VL_3}{\nu}$$

Substitute $3\ \mathrm{m}$ for L_3 and other corresponding values in the above equation.

$$\mathrm{Re}_3 = \frac{VL_3}{\nu}$$
$$= \frac{4 \times 3}{\left(1.448 \times 10^{-5}\right)}$$
$$= 8.287 \times 10^5$$

Because the Reynolds number is greater than 5×10^5 the Nusselt number is,

$$\mathrm{Nu} = \left(0.037\,\mathrm{Re}^{0.8} - 871\right)\mathrm{Pr}^{1/3}$$

Substitute the corresponding values of plate 3 in the above equation.

$$\mathrm{Nu}_2 = \left(0.037\,\mathrm{Re}^{0.8} - 871\right)\mathrm{Pr}^{1/3}$$
$$= \left(0.037\left(8.287 \times 10^5\right)^{0.8} - 871\right)(0.73295)^{1/3}$$
$$= 1025.8$$

Step 8 of 9

The Nusselt number for the third plate is,

$$\mathrm{Nu}_3 = \frac{h_3 \cdot L_3}{k}$$

Substitute the corresponding values in the above equation.

$$\mathrm{Nu}_3 = \frac{h_3 \cdot L_3}{k}$$
$$1025.8 = \frac{h_3 \times 3}{0.024575}$$
$$h_3 = 8.4\ \mathrm{W/m^2{}^\circ C}$$

Surface Area of the plate is, $A = lb$

Here, l is the length of the plate and b is the width of the plate.

Substitute $1\mathrm{m}^2$ for l and $4\ \mathrm{m}^2$ for b in the above equation of area.

$$A_{s_3} = lb$$
$$= 1 \times 4$$
$$= 4\mathrm{m}^2$$

Step 9 of 9

The convection heat transfer for plate 3 is as follows.

$$Q = h_3 A_{s_3}\left(T_s - T_\infty\right)$$

Substitute the corresponding values of plate 3 in the above equation.

$$Q = h_3 A_{s_3}\left(T_s - T_\infty\right)$$
$$= 8.4 \times 4 \times (15 - 10)$$
$$= 168\ \mathrm{W}$$

Therefore heat loss from the third plate is, $\boxed{168\ \mathrm{W}}$.

Question: 19.026P

Consider a hot automotive engine, which can be approximated as a 0.5-m-high, 0.40-m-wide, and 0.8-m-long rectangular block. The bottom surface of the block is at a temperature of 100°C and has an emissivity of 0.95. The ambient air is at 20°C, and the road surface is at 25°C. Determine the rate of heat transfer from the bottom surface of the engine block by convection and radiation as the car travels at a velocity of 80 km/h. Assume the flow to be turbulent over the entire surface because of the constant agitation of the engine block.

Step 1 of 8

Given: - Size of engine = 0.8m × 0.40m × 0.5m

Temperature of bottom surface of block T_s = 100°C

Emissivity of surface $\varepsilon = 0.95$

Road surface temperature $T_{sur} = 25°C$

Ambient air temperature $T_\infty = 20°C$

Step 2 of 8

Properties of air at 1 atm and average film temperature $\frac{(100+20)}{2} = 60°C$ is taken from properties of air table

Thermal conductivity, $k = 0.02808$ W/m°C

Kinematic viscosity, $\upsilon = 1.896\times10^{-5}$ m^2/s

Prandtl number, Pr = 0.7202

Step 3 of 8

Reynolds number is calculated as follows

$$\mathrm{Re}_L = \frac{VL}{\upsilon}$$

$$= \frac{\left[(80\times\frac{1000}{3600})\ \text{m/s}\right](0.8\ \text{m})}{1.896\times10^{-5}\ \text{m}^2/\text{s}}$$

$$\mathrm{Re}_L = 9.376\times10^5$$

Since $5\times10^5 < \mathrm{Re}_L < 10^7$, the flow is laminar turbulent.

But turbulent flow will be assumed here due to agitation of engine block.

Step 4 of 8

So Nusselt number is calculated as follows

$$Nu = \frac{hL}{k}$$

$$= 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{1/3}$$

$$= 0.037(9.376\times10^5)^{0.8}(0.7202)^{1/3}$$

$$Nu = 1988$$

Step 5 of 8

Coefficient of heat transfer :

$$h = \frac{k}{L}Nu$$

$$= \frac{0.02808\ \text{W/m°C}}{0.8\ \text{m}}(1988)$$

$$h = 69.78\ \text{W/m}^2\text{°C}$$

Step 6 of 8

Heat transfer surface area,

$$A_s = wL$$

$$= (0.8\ \text{m})(0.4\ \text{m})$$

$$A_s = 0.32\ \text{m}^2$$

Rate of heat transfer:

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty)$$

$$= (69.78\ \text{W/m}^2\text{°C})(0.32\ \text{m}^2)(100\text{-}20)\text{°C}$$

$$\dot{Q}_{conv} = 1786.37\ \text{W}$$

Step 7 of 8

The radiation heat transfer from the same surface is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{sur}^4)$$

$$= (0.95)(0.32\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{K}^4)\left[(100+273\ \text{K})^4 - (25+273\ \text{K})^4\right]$$

$$\dot{Q}_{rad} = 197.7\ \text{W}$$

Step 8 of 8

Then the total rate of heat transfer from that surface becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

$$= (1786.37+197.7)\ \text{W}$$

$$\boxed{\dot{Q}_{total} = 1984\ \text{W}}$$

Question: 19.027P

The top surface of the passenger car of a train moving at a velocity of 95 km/h is 2.8 m wide and 8 m long. The top surface is absorbing solar radiation at a rate of 380 W/m2, and the temperature of the ambient air is 30°C. Assuming the roof of the car to be perfectly insulated and the radiation heat exchange with the surroundings to be small relative to convection, determine the equilibrium temperature of the top surface of the car.

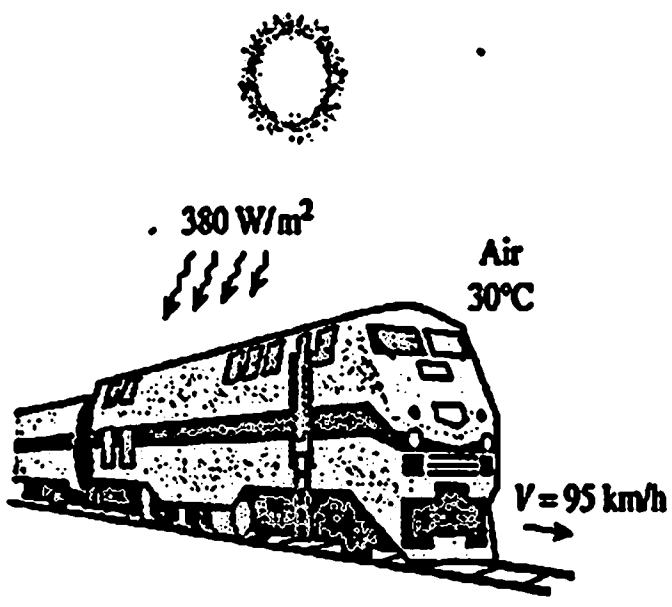


FIGURE P19-27

Step 1 of 4

Given data:

Velocity of the passenger car,

$V = 95\,\text{km/h}$

$$= \frac{95\times1000}{3600}$$

$$= 26.39\ \text{m/s}$$

Length of the surface, $L = 8\,\text{m}$

Width of the surface, $w = 2.8\,\text{m}$

Rate of solar radiation absorbed on the top surface, $\dot{q}_r = 380\,\text{W/m}^2$

Ambient air temperature, $T_\infty = 30°\text{C}$

Step 2 of 4

From the table of air properties, at ambient temperature $T_\infty = 30°\text{C}$

Thermal conductivity $k = 0.02588\,\text{w/m}°\text{C}$

Kinematic viscosity $\nu = 1.608\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number $\text{Pr} = 0.7282$

The convection heat transfer rate from the top surface of the car to the air must be equal to the solar radiation absorbed by the same surface in order to reach steady state conditions

$$\dot{q}_r = \dot{q}_s = h\left(T_s - T_\infty\right) \quad \ldots\ldots (1)$$

Calculating Reynolds number to get flow regime

$$\text{Re} = \frac{VL}{\nu}$$

$$= \frac{26.39\times8}{1.608\times10^{-5}}$$

$$= 1.312\times10^{7}$$

Reynolds number is greater than 5×10^5 then the flow is combined laminar and turbulent flow

Step 3 of 4

Then the Nusselt number relationship for average heat transfer coefficient can be calculated as,

$$\text{Nu} = \left(0.037\text{Re}_L^{0.8} - 871\right)\text{Pr}^{1/3}$$

$$= \left(0.037\left(1.312\times10^7\right)^{0.8} - 871\right)\times(0.7282)^{1/3}$$

$$= 17433.117\times0.8996$$

$$= 15682.83$$

Step 4 of 4

Then the heat transfer coefficient is,

$$\text{Nu} = \frac{hL}{k}$$

$$15682.83 = \frac{h\times8}{0.02588}$$

$$h = 50.733\,\text{W/m}^2\cdot\text{K}$$

Substituting the values T_s, T_∞ and h in equation (1), we get

$$\dot{q}_s = h\left(T_s - T_\infty\right)$$

$$380 = 50.733\left(T_s - 30\right)$$

$$T_s = 37.49°\text{C}$$

Therefore, equilibrium temperature of the top surface of the car $\boxed{T_s = 37.49°\text{C}}$

Question: 19.028P pg 1

Reconsider Prob. 19-27. Using EES (or other) software, investigate the effects of the train velocity and the rate of absorption of solar radiation on the equilibrium temperature of the top surface of the car. Let the train velocity vary from 10 km/h to 120 km/h and the rate of solar absorption from 100 W/m2 to 500 W/m2. Plot the equilibrium temperature as functions of train velocity and solar radiation absorption rate, and discuss the results.

Step 1 of 10

Case 1: Varying air velocity

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

Equations Window

```
" Given data"
"V=95 [km/h]"convert(km/h,m/s)"
L=8 [m]
w=2.8 [m]
q_dot_r=380 [W/m^2]
T_infinity=30 [C]
P_1=1 [atm]"convert(atm,kPa)

rho_1=Density(Air,T=T_infinity,P=P_1)
k_1=Conductivity(Air,T=T_infinity)
mu_1=Viscosity(Air,T=T_infinity)
Pr_1=Prandtl(Air,T=T_infinity)

"Reynolds number "
Re=(rho_1*V*L)/mu_1

"Heat transfer rate from the top surface of the car to the air must be equal to the solar radiation absorbed"

q_dot_r=q_dot_s

q_dot_s=h*(T_s-T_infinity)

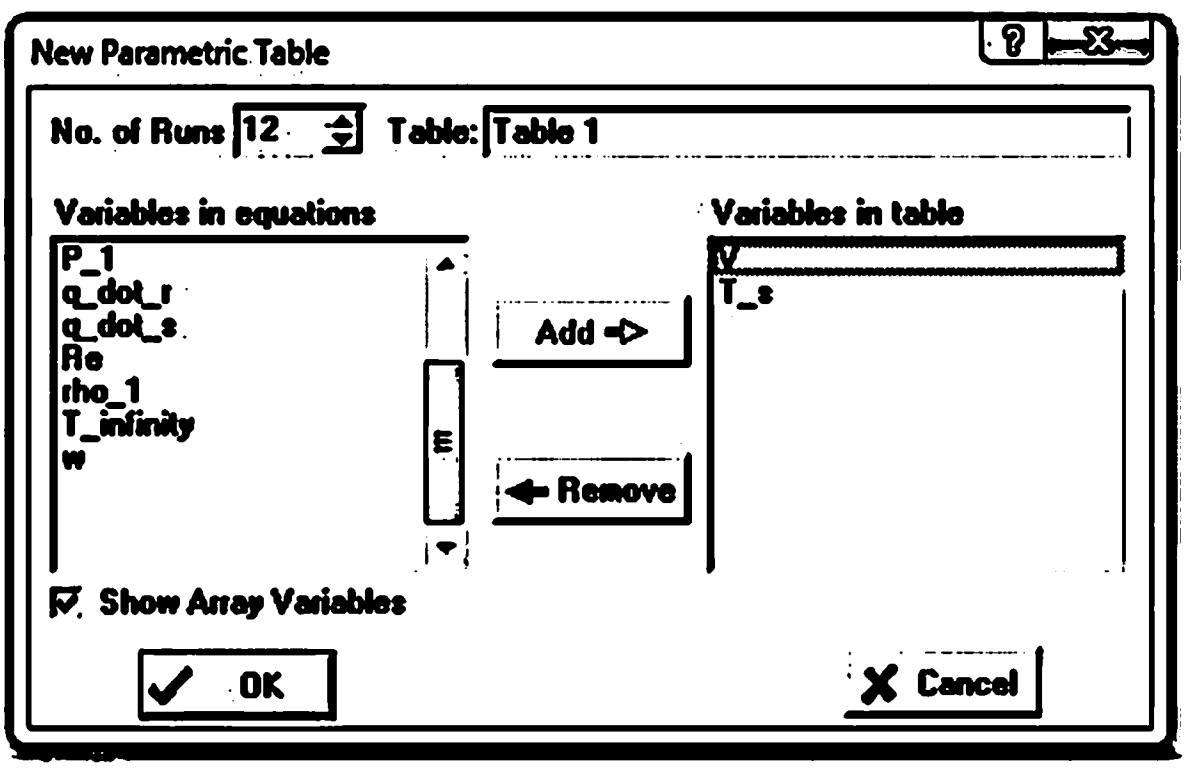

"Nusselt number"

Nu=((0.037*Re^0.8)-871)*Pr_1^(1/3)


"Heat transfer coefficient"
k_1=h*L/Nu
```

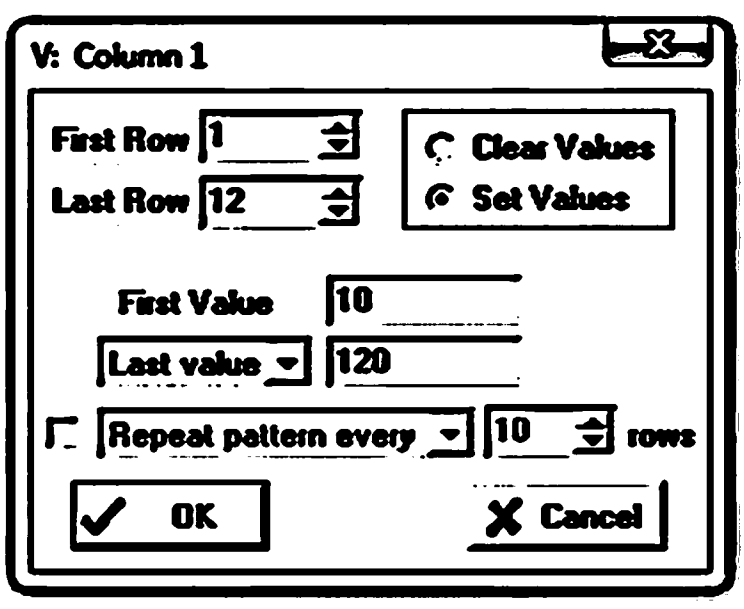
Step 2 of 10

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 3 of 10

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

Table 1

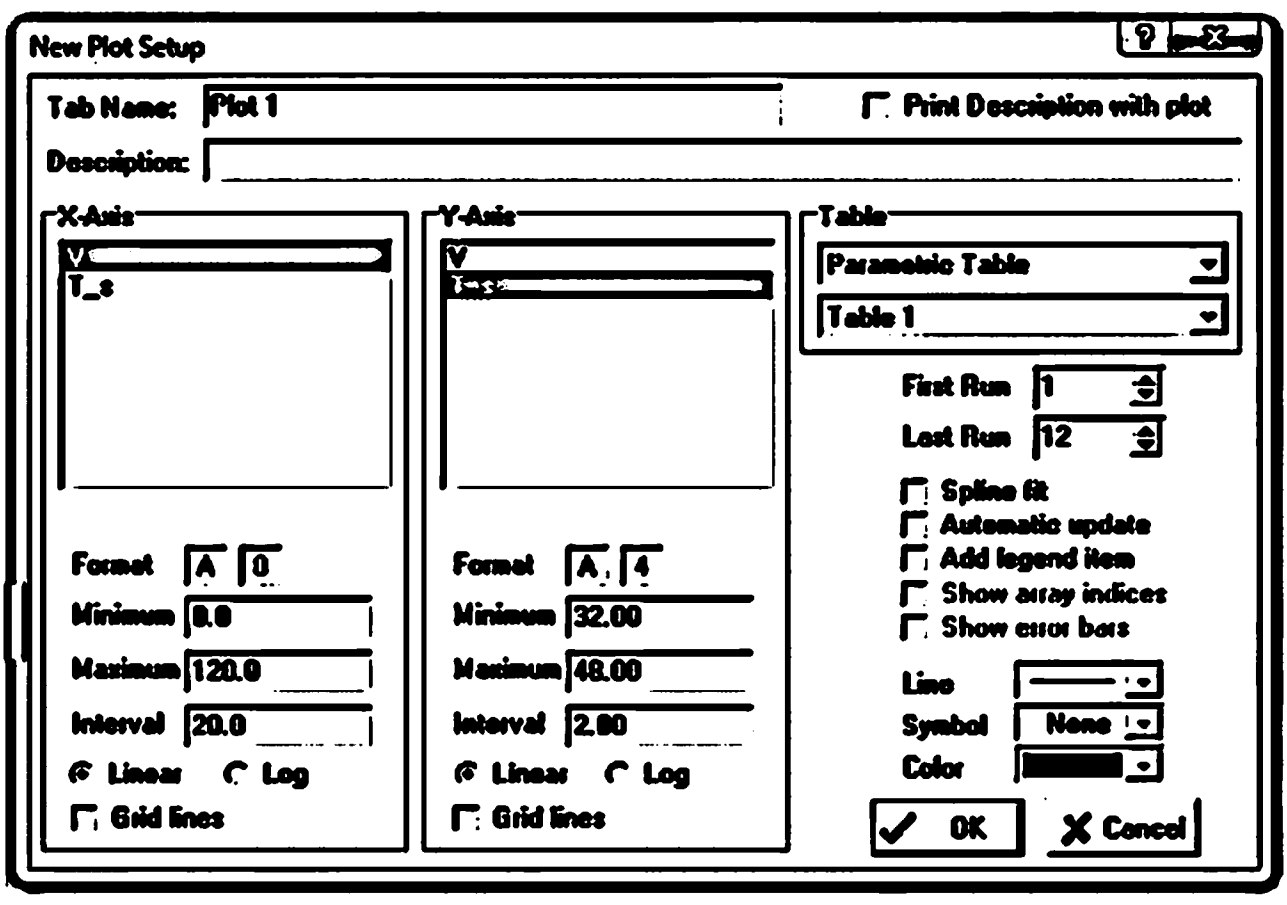
1..12	V	T_s
Run 1	10	47.29
Run 2	20	39.46
Run 3	30	36.73
Run 4	40	35.29
Run 5	50	34.4
Run 6	60	33.79
Run 7	70	33.34
Run 8	80	33
Run 9	90	32.72
Run 10	100	32.5
Run 11	110	32.31
Run 12	120	32.15

Step 4 of 10

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,

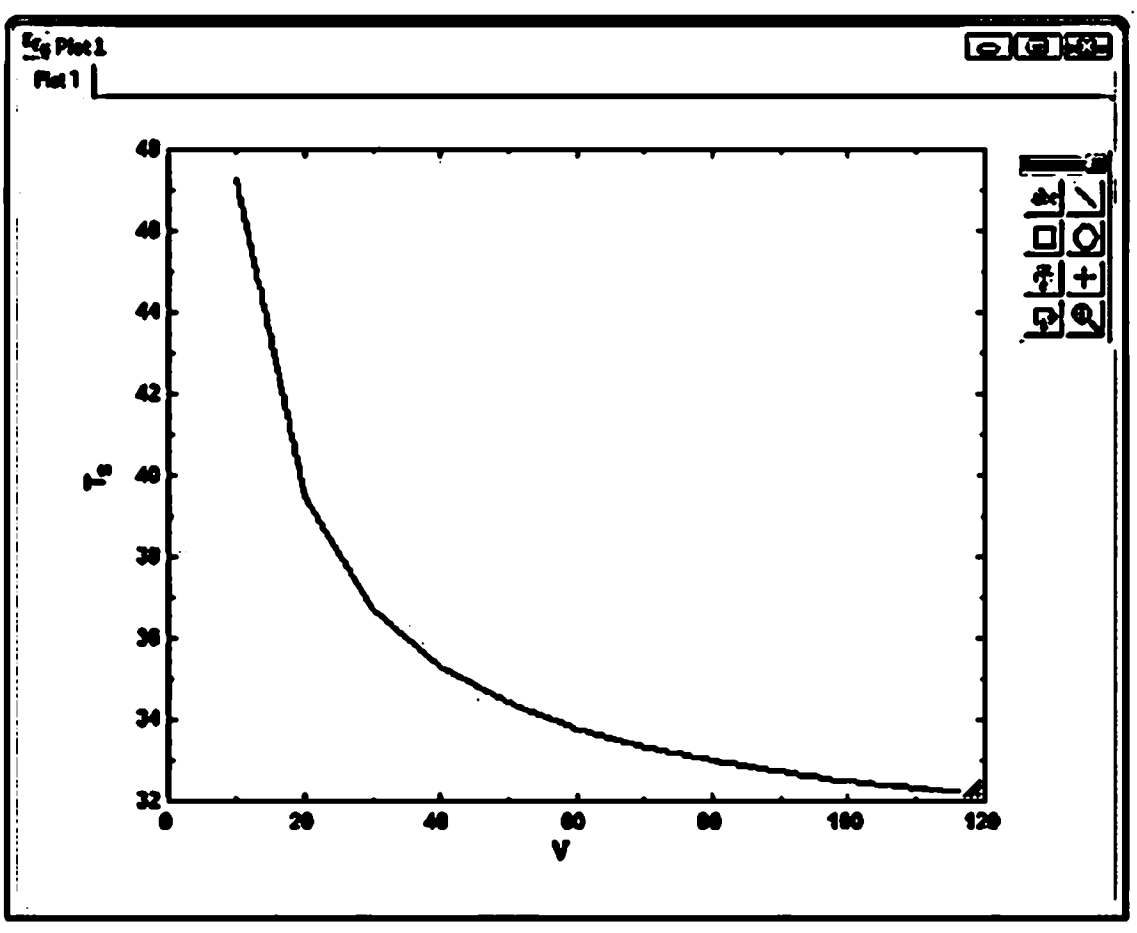
Step 4 of 10

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



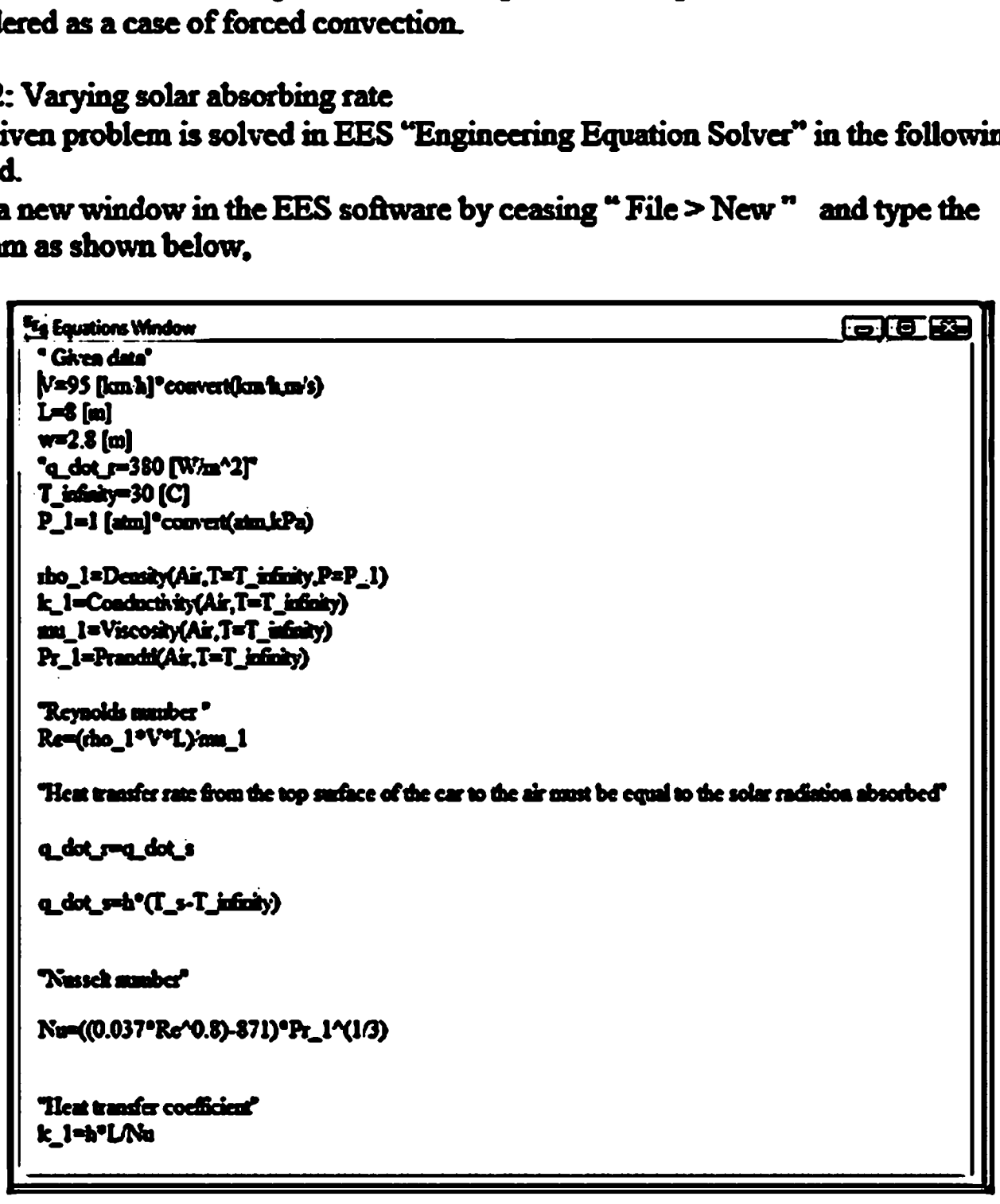
Step 5 of 10

By clicking on OK the graph will be generated as shown,



With the increase in velocity of the car the top surface temperature decreases.This can be considered as a case of forced convection.

Case 2: Varying solar absorbing rate
This given problem is solved in EES "Engineering Equation Solver" in the following method.
Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

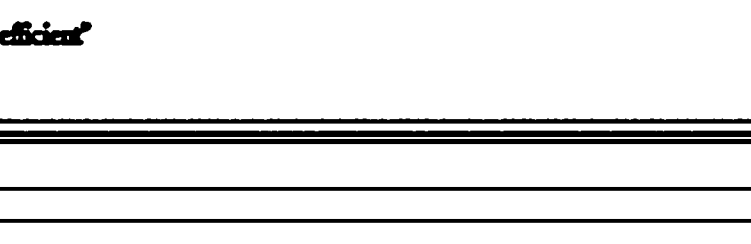


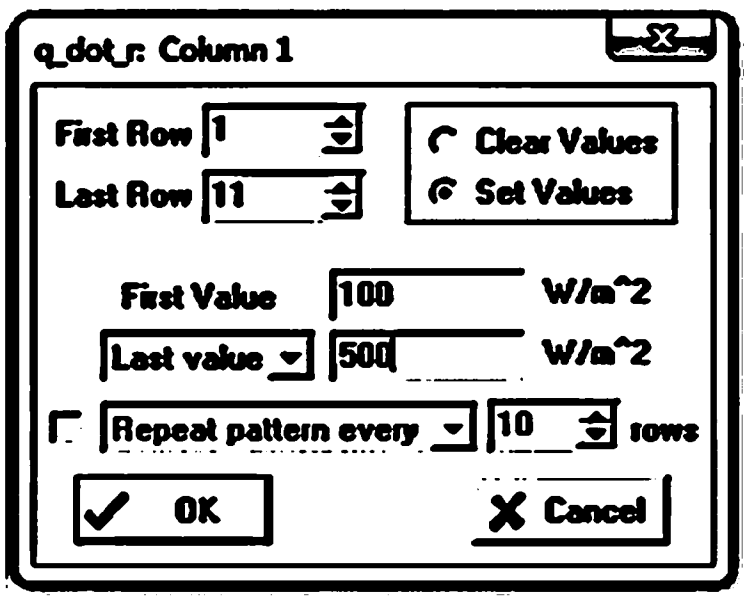
Step 6 of 10

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,

Step 7 of 10

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,





Step 8 of 10

Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

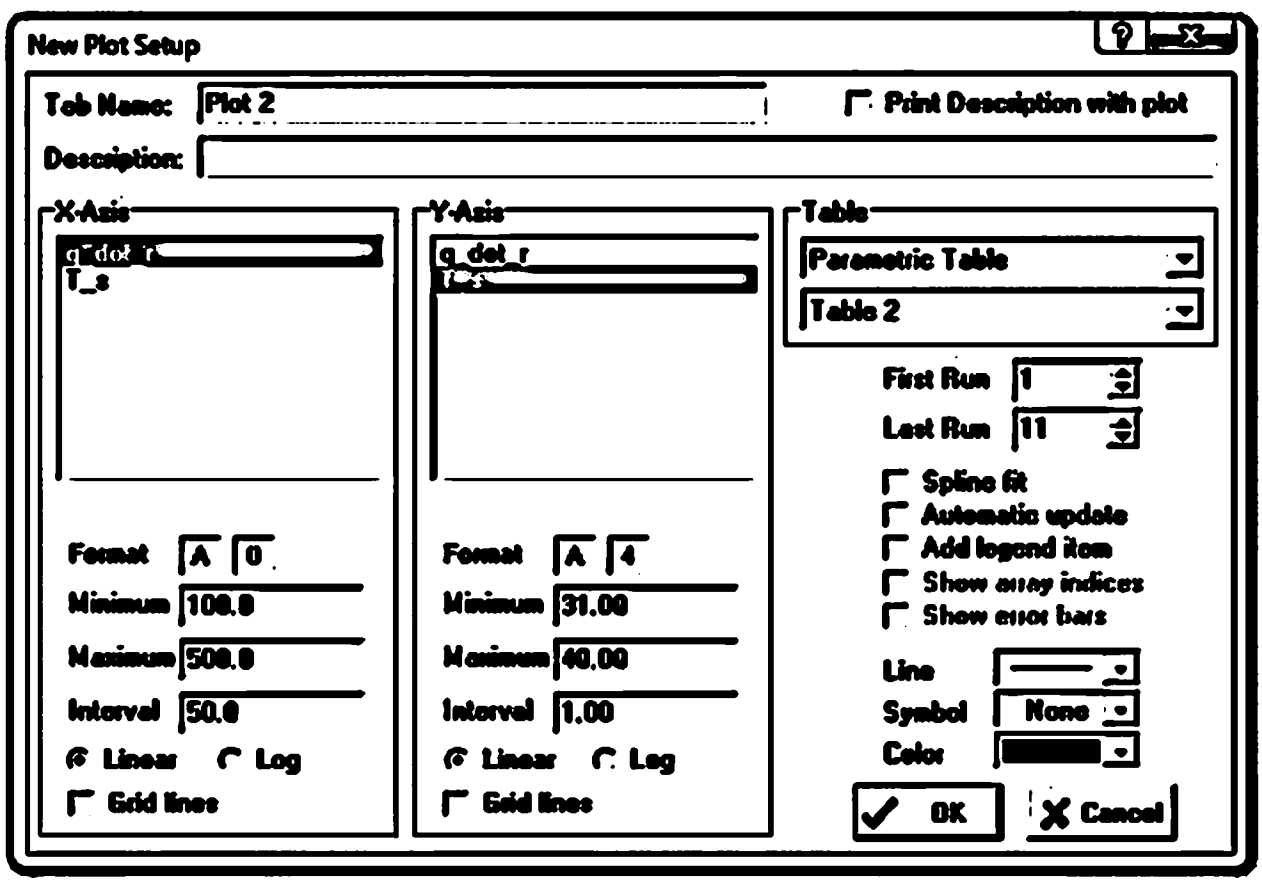
Parametric Table

Table 1 | Table 2

1..11	$\dot{q}_r$ [W/m2]	T_s
Run 1	100	31.97
Run 2	140	32.76
Run 3	180	33.55
Run 4	220	34.34
Run 5	260	35.12
Run 6	300	35.91
Run 7	340	36.7
Run 8	380	37.49
Run 9	420	38.28
Run 10	460	39.06
Run 11	500	39.85

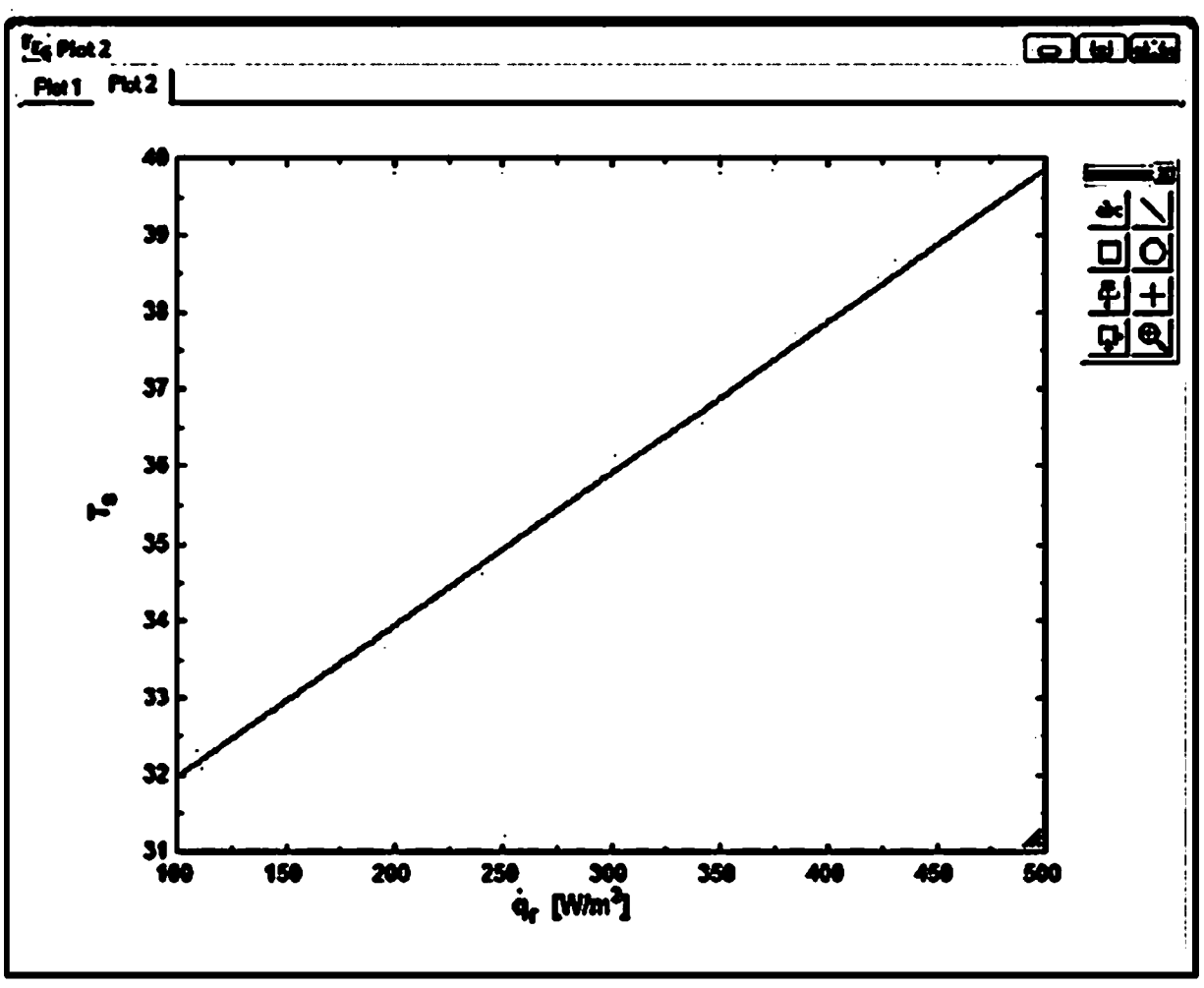
Step 9 of 10

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 10 of 10

By clicking on OK the graph will be generated as shown,



From the graph we can see that with the increase in the solar absorption rate the top surface temperature of the car increases linearly.

Question: 19.029P

A 15-cm × 15-cm circuit board dissipating 20 W of power uniformly is cooled by air, which approaches the circuit board at 20°C with a velocity of 6 m/s. Disregarding any heat transfer from the back surface of the board, determine the surface temperature of the electronic components (*a*) at the leading edge and (*b*) at the end of the board. Assume the flow to be turbulent since the electronic components are expected to act as turbulators.

Step 1 of 6

Obtain the Properties of air at 1 atm. and an assumed film temperature of $35°C$ from the table A-22 "ideal gas properties of air at 1 atm. Pressure".

Thermal conductivity, $k = 0.0265\ \text{W/m°C}$

Kinematic viscosity, $\nu = 1.655\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7268$

Step 2 of 6

(a)

Write the relation for Nusselt number.

$$\text{Nu}_x = \frac{h_x x}{k}$$

$$h_x = \frac{k_x}{x}\text{Nu}_x$$

Here, Nu_x is the local Nusselt number and x is the distance from the leading edge.

Substitute 0 for x as the heat transfer at the leading edge of the board is determined.

$$\begin{aligned} h_x &= \frac{k_x}{x}\text{Nu}_x \\ &= \frac{k_x}{0}\times\text{Nu}_x \\ &= \infty \end{aligned}$$

Step 3 of 6

Calculate the surface temperature at the end of the board.

$$\dot{Q} = h_x A_s (T_s - T_\infty)$$

$$T_s = \frac{\dot{Q}}{h_x A_s} + T_\infty$$

Here, $\dot{Q}$ is the rate of heat dissipation, T_s is the surface temperature and T_∞ is the air temperature.

Substitute ∞ for h_x and $20°C$ for T_∞.

$$\begin{aligned} T_s &= \frac{\dot{Q}}{h_x A_s} + T_\infty \\ &= \frac{20}{\infty\times A_s} + 20 \\ &= 0 + 20 \\ &= 20°C \end{aligned}$$

As the Coefficient of heat transfer by convection at leading edge tends to infinity, the temperature of surface at the leading edge would be approaching temperature of air that is $\boxed{20°C}$.

Step 4 of 6

(b)

Determine the Reynolds number.

$$\text{Re}_x = \frac{Vx}{\upsilon}$$

Here, V is the velocity of air and x is the distance from the leading edge.

Substitute $6\ \text{m/s}$ for V, $1.655\times10^{-5}\ \text{m}^2/\text{s}$ for ν, and 0.15 m for x.

$$\begin{aligned} \text{Re} &= \frac{(6\ \text{m/s})(0.15\ \text{m})}{1.655\times10^{-5}\ \text{m}^2/\text{s}} \\ &= 5.438\times10^4 \end{aligned}$$

Since $\text{Re} < 5\times10^5$, the flow is laminar. But turbulent flow is assumed as electronic component are expected to act as turbulators.

Step 5 of 6

Calculate the Nusselt number assuming a uniform heat flux using the relation,

$$\text{Nu}_x = 0.0308\,\text{Re}_x^{0.8}\,\text{Pr}^{1/3}$$

Substitute 5.438×10^4 for Re and 0.7268 for Pr.

$$\begin{aligned} \text{Nu}_x &= 0.0308(5.438\times10^4)^{0.8}(0.7268)^{1/3} \\ &= 170.10 \end{aligned}$$

Step 6 of 6

Calculate the heat transfer coefficient using the relation,

$$\text{Nu}_x = \frac{h_x x}{k}$$

$$h_x = \frac{k_x}{x}\text{Nu}_x$$

Substitute 170.10 for Nu_x, $0.0265\ \text{W/m°C}$ for k and 0.15 m for x as the heat transfer is to be determined at the end of the borad.

$$h_x = \frac{0.0265\ \text{W/m°C}}{0.15\ \text{m}}(170.10)$$

$$h_x = 30.051\ \text{W/m}^2\cdot°\text{C}$$

Calculate the surface area of the board.

$$\begin{aligned} A_s &= 0.15\times0.15 \\ &= 0.0225\ \text{m}^2 \end{aligned}$$

Here, A_s is the surface area.

Calculate the surface temperature at the end of the board.

$$\dot{Q} = h_x A_s (T_s - T_\infty)$$

Here, T_s is the surface temperature and T_∞ is the air temperature.

Substitute $20°C$ for T_∞, $0.0225\ \text{m}^2$ for A_s, and $30.051\ \text{W/m}^2\cdot°\text{C}$ for h_x

$$20 = 30.051\times0.0225(T_s - 20)$$

$$20 = 0.676(T_s - 20)$$

$$T_s = 49.58°C$$

Therefore, the surface temperature at the end of the board is $\boxed{49.58°C}$.

Question: 19.030P

Consider a refrigeration truck traveling at 70 mph at a location where the air temperature is 80°F. The refrigerated compartment of the truck can be considered to be a 9-ft-wide, 7-ft-high, and 20-ft-long rectangular box. The refrigeration system of the truck can provide 3 tons of refrigeration (i.e., it can remove heat at a rate of 600 Btu/min). The outer surface of the truck is coated with a low-emissivity material, and thus radiation heat transfer is very small. Determine the average temperature of the outer surface of the refrigeration compartment of the truck if the refrigeration system is observed to be operating at half the capacity. Assume the air flow over the entire outer surface to be turbulent and the heat transfer coefficient at the front and rear surfaces to be equal to that on side surfaces.

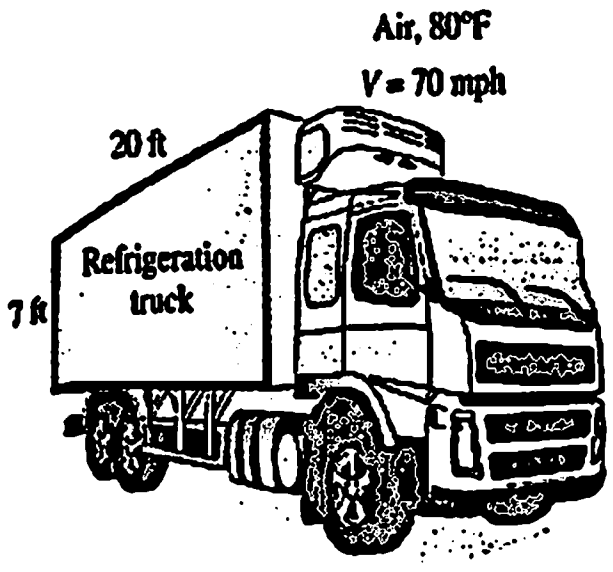


FIGURE P19-30E

Step 1 of 5

Given data:

Temperature of air $T_\infty = 80°F$

Truck velocity,

$V = 70\,\text{mph}$

$$= \frac{70 \times 5280}{3600}$$

$$= 102.67\,\text{ft/s}$$

Width of compartment, $w = 9\text{ft}$

Height of compartment, $H = 7\text{ft}$

Length of compartment, $L = 20\text{ft}$

Capacity of Refrigeration system, $Q_r = 3\text{tons}$

We know one ton of refrigeration is

$1\text{ton} = 200\,\text{Btu/min}$

Then

$Q_r = 3 \times 200$

$= 600\,\text{Btu/min}$

$= 36000\,\text{Btu/hour}$

Step 2 of 5

Loading solution. Please wait ...

Step 3 of 5

Then from equation (1), we get Nusselt number

$$\text{Nu} = 0.037\,\text{Re}_L^{0.8}\,\text{Pr}^{1/3}$$

$$= 0.037\left(1.21 \times 10^7\right)^{0.8}(0.7290)^{1/3}$$

$$\text{Nu} = 15440.89$$

Then the heat transfer coefficient will be

$$\text{Nu} = \frac{hL}{k}$$

$$15440.89 = \frac{h \times 20}{0.01481}$$

$$h = 11.434\,\text{Btu/h} \cdot \text{ft}^2 \cdot \text{R}$$

Step 4 of 5

Since it is given that the system is observed to be operated at half of its capacity

Therefore

$$\dot{Q} = \frac{Q_r}{2}$$

$$= \frac{36000}{2}$$

$$= 18000\,\text{Btu/h}$$

Heat transfer surface area is,

$$A_s = 2\left[(20 \times 9) + (20 \times 7) + (9 \times 7)\right]$$

$$= 766\,\text{ft}^2$$

Step 5 of 5

We know the newton's law of cooling,

$$\dot{Q} = hA_s\left(T_\infty - T_s\right)$$

$$18000 = 11.434 \times 766\left(80 - T_s\right)$$

$$2.055 = \left(80 - T_s\right)$$

$$T_s = 77.945°F$$

Therefore, the average temperature of the outer surface of the refrigeration compartment of the truck is $\boxed{T_s = 77.945°F}$

Question: 19.031P

Liquid mercury at 250°C is flowing in parallel over a flat plate at a velocity of 0.3 m/s. Surface temperature of the 0.1-m-long flat plate is constant at 50°C. Determine (*a*) the local convection heat transfer coefficient at 5 cm from the leading edge and (*b*) the average convection heat transfer coefficient over the entire plate.

Step 1 of 6

Given data:

Temperature of liquid mercury, $T_\infty = 250°C$

Velocity of flow, $V = 0.3\ m/s$

Length of the plate, $L = 0.1\ m$

Temperature of the flat plate, $T_s = 50°C$

Distance from the leading edge, $x = 5\ cm$

Step 2 of 6

The film temperature of the liquid mercury is determined as,

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(50+250)}{2}$$
$$= 150°C$$

From the table, properties of liquid mercury at 1 atm pressure and at $T = 150°C$

Thermal conductivity, $k = 10.07780\ W/m \cdot K$

Kinematic viscosity, $\nu = 8.514\times10^{-8}\ m^2/s$

Prandtl number, $Pr = 0.0152$

Step 3 of 6

(a)

The Reynolds number for the flow is determined from the equation given by

$$Re_x = \frac{Vx}{\nu}$$
$$= \frac{0.3\times\left(5\times10^{-2}\right)}{8.514\times10^{-8}}$$
$$= 176180.41$$

Since the obtained Reynolds number is less than 5×10^5 i.e. $Re < 5\times10^5$, the flow is said to be laminar.

Step 4 of 6

The average Nusselt number for parallel flow over flat plate is determined from the Churchill & Ozoe relation given by

$$Nu_x = \frac{0.3387Pr^{1/3}Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$
$$= \frac{0.3387\times(0.0152)^{1/3}\times(176180.41)^{1/2}}{\left[1+\left(\frac{0.0468}{0.0152}\right)^{2/3}\right]^{1/4}}$$
$$= 26.51$$

The local convective heat transfer coefficient at 5 cm from the leading edge is determined from the equation given by,

$$Nu_x = \frac{h_x x}{k}$$
$$h_x = \frac{Nu_x \times k}{x}$$
$$= \frac{26.51\times10.07780}{\left(5\times10^{-2}\right)}$$
$$= 5342.24\ W/m^2 \cdot K$$

The local convective heat transfer coefficient at 5 cm from the leading edge is

$$\boxed{h_x = 5342.24\ W/m^2 \cdot K}$$

Step 5 of 6

(b)

The Average convection heat transfer coefficient over entire plate can be calculated by integrating h_x over the entire length,

$$h = \frac{1}{L}\int_0^L h_x dx \qquad \ldots\ldots (1)$$

Local heat transfer coefficient,

$$h_x = \frac{0.3387Pr^{1/3}Re_x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}\times\frac{k}{x}$$
$$= \frac{0.3387Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\times x^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}\times\frac{k}{x}$$

As other values will remain constant during integrating

Therefore $h_x = Ax^{-1/2}$

Where

$$A = \frac{0.3387Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\times k}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$

Step 6 of 6

Substituting the value of h_x in equation (1), we get

$$h = \frac{1}{L}\int_0^L Ax^{-1/2}dx$$
$$= \frac{A}{L}\int_0^L x^{-1/2}dx$$
$$= \frac{A}{L}\left[2x^{1/2}\right]_0^L$$
$$= \frac{2A}{L^{1/2}}$$

The average convective heat transfer coefficient over the entire plate is determined as,

$$h = \frac{2}{L^{1/2}}\times\frac{0.3387Pr^{1/3}\left(\frac{V}{\nu}\right)^{1/2}\times k}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$
$$= \frac{2}{(0.1)^{1/2}}\times\frac{0.3387\times(0.0152)^{1/3}\times\left(\frac{0.3}{8.514\times10^{-8}}\right)^{1/2}\times10.07780}{\left[1+\left(\frac{0.0468}{0.0152}\right)^{2/3}\right]^{1/4}}$$
$$= 7555\ W/m^2 \cdot K$$

Therefore, the average convective heat transfer coefficient over the entire plate is

$$\boxed{h = 7555\ W/m^2 \cdot K}$$

Question: 19.032P

Liquid mercury at 250°C is flowing with a velocity of 3 m/s in parallel over a 0.1 -m long flat plate where there is an unheated starting length of 5cm. The heated section of the flat Plate is maintained at a constant temperature of 50°C. Determine (*a*) the local convection heat transfer coefficient at the trailing edge, (*b*) the average convection heat transfer coefficient for the heated section, and (*c*) the rate of heat transfer per unit width for the heated section.

Step 1 of 7

Given data:

Temperature of liquid Mercury, $T_\infty = 250°C$

Velocity of flow, $V = 0.3\ m/s$

Length of the flat plate, $L = 0.1\ m$

Temperature of the flat plate, $T_s = 50°C$

Length of unheated portion, $\xi = 5\ cm$

Step 2 of 7

The film temperature of the liquid mercury is determined as,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(50+250)}{2}$$

$$= 150°C$$

From the table, properties of liquid mercury at 1 atm pressure and at $T = 150°C$

Thermal conductivity, $k = 10.07780\ W/m \cdot K$

Kinematic viscosity, $\nu = 8.514 \times 10^{-6}\ m^2/s$

Prandtl number, $Pr = 0.0152$

Step 3 of 7

(a)

The Reynolds number for the flow at the trailing edge, $x = L$ is determined from the equation given by,

$$Re_{x=L} = \frac{Vx}{\nu}$$

$$= \frac{0.3 \times 0.1}{8.514 \times 10^{-6}}$$

$$= 352360.82$$

Since the obtained Reynolds number is less than 5×10^5 i.e. $Re < 5 \times 10^5$, the flow is said to be laminar.

The average Nusselt number for parallel flow over flat plate is determined from the Churchill & Ozoe relation given by

$$Nu_x = \frac{Nu_{x(\xi=0)}}{\left[1-(\xi/x)^{3/4}\right]^{1/3}} \qquad \ldots\ldots (1)$$

Step 4 of 7

$$Nu_{x(\xi=0)} = \frac{0.3387 Pr^{1/3} Re_{x=L}^{1/2}}{\left[1+(0.0468/Pr)^{2/3}\right]^{1/4}}$$

$$= \frac{0.3387 \times (0.0152)^{1/3} \times (352360.82)^{1/2}}{\left[1+\left(\frac{0.0468}{0.0152}\right)^{2/3}\right]^{1/4}}$$

$$= 37.48$$

Substituting the above value $Nu_{x(\xi=0)}$ in equation (1), we get

$$Nu_x = \frac{37.48}{\left[1-\left(\frac{5\times10^{-2}}{0.1}\right)^{3/4}\right]^{1/3}}$$

$$= 50.65$$

Step 5 of 7

The local convective heat transfer coefficient at the trailing edge is determined from the equation given by

$$Nu_x = \frac{h_x L}{k}$$

$$h_x = \frac{Nu_x \times k}{L}$$

$$= \frac{50.65 \times 10.07780}{0.1}$$

$$= 5104.05\ W/m^2 \cdot K$$

Therefore, the local convective heat transfer coefficient at the trailing edge is

$$\boxed{h_x = 5104.05\ W/m^2 \cdot K}$$

Step 6 of 7

(b)

The average convection heat transfer coefficient for the heated section is determined from the equation given by

$$h = \frac{2\left[1-(\xi/x)^{3/4}\right]}{1-\xi/L} \times h_{x=L}$$

$$= \frac{2 \times \left(1-\left(\frac{5\times10^{-2}}{0.1}\right)^{3/4}\right)}{1-\frac{5\times10^{-2}}{0.1}} \times 5104.05$$

$$= 8276.65\ W/m^2 \cdot K$$

Therefore, the average convection heat transfer coefficient for the heated section is

$$\boxed{h = 8276.65\ W/m^2 \cdot K}$$

Step 7 of 7

(c)

The rate of heat transfer per unit width for the heated section is determined from the equation given by,

$$\dot{Q} = hA_s(T_\infty - T_s)$$

$$= h \times \left((L-\xi) \times w\right) \times (T_\infty - T_s)$$

$$= 8276.65 \times \left(\left(0.1-5\times10^{-2}\right) \times 1\right) \times (250-50)$$

$$= 82766.5\ W/m$$

Therefore, the rate of heat transfer per unit width for the heated section is

$$\boxed{\dot{Q} = 82766.5\ W/m}$$

Question: 19.033P

Air at 60°F flows over a 10-ft-long flat plate at 7 ft/s Determine the local heat transfer coefficient at intervals of 1ft and plot the results against the distance from the leading edge.

Step 1 of 6

Given data:

Temperature of air, $T = 60°C$

Velocity of air, $V = 7\,ft/s$

Length of flat plate, $L = 10\,ft$

Step 2 of 6

Properties of air at 1 atm and average film temperature 60°F is taken from properties of air table

$k = 0.01433\ Btu/h \cdot ft \cdot °F$

$\upsilon = 0.1588 \times 10^{-3}\ ft^2/s$

$Pr = 0.7321$

Reynolds number for 1 ft is calculated as

$$Re_L = \frac{VL}{\upsilon}$$
$$= \frac{(7\ ft/s)(1\ ft)}{0.1588 \times 10^{-3}\ ft^2/s}$$
$$= 4.408 \times 10^4$$

This is less than critical value of Reynolds number. So, the flow is laminar.

Step 3 of 6

The local Nusselt number is given by

$$Nu_x = \frac{hx}{k}$$
$$= 0.332 Re_x^{0.5} Pr^{1/3}$$
$$= 0.332\left(4.408 \times 10^4\right)^{0.5} (0.7321)^{1/3}$$
$$= 62.82$$

Step 4 of 6

The local heat transfer and friction coefficients are given by

$$h_x = \frac{k}{x} Nu$$
$$= \frac{0.01433\ Btu/h \cdot ft \cdot °F}{1\,ft}(62.82)$$
$$= 0.9002\ Btu/h \cdot ft^2 \cdot °F$$

$$C_{f,x} = \frac{0.664}{Re^{0.5}}$$
$$= \frac{0.664}{\left(4.407 \times 10^4\right)^{0.5}}$$
$$= 0.00316$$

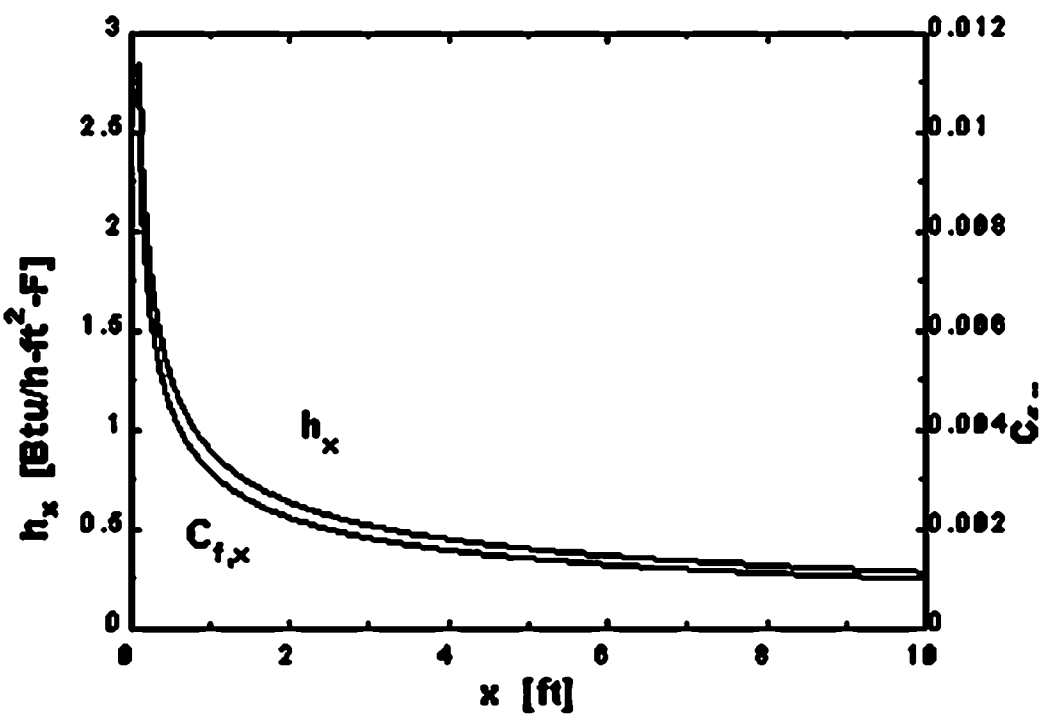
Step 5 of 6

We repeat calculations for all 1-ft intervals. The results are

x	h_x	$C_{f,x}$
1	0.9005	0.003162
2	0.6367	0.002236
3	0.5199	0.001826
4	0.4502	0.001581
5	0.4027	0.001414
6	0.3676	0.001291
7	0.3404	0.001195
8	0.3184	0.001118
9	0.3002	0.001054
10	0.2848	0.001

Step 6 of 6

Graph:



Question:

Reconsider Prob. 19-33E. Using EES (or other) software, evaluate the local heat transfer coefficieot along the plate at intervals of 0.1 ft, and plot them against the distance from the fading edge.

Step 1 of 2

4749-19-34P

AID: 530 | 28/05/2012

Given data:

Temperature of the air, $T_{air} = 60°F$

Length of the flat plate, $L = 10 ft$

Velocity of the air, $V_{air} = 7 ft/s$

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.

EES Limited Academic Version: C:\EES (LIMITED ACADEMIC)\HELLO.TXT - [Equations Window]

File Edit Search Options Calculate Tables Plots Windows Help Thermodynamics

```
"GIVEN"
T_air=60 [F]
"x=10 [ft]"
Vel=7 [ft/s]
"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_air)
Pr=Prandtl(Fluid$, T=T_air)
rho=Density(Fluid$, T=T_air, P=14.7)
mu=Viscosity(Fluid$, T=T_air)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho
"ANALYSIS"
Re_x=(Vel*x)/nu
"Reynolds number is calculated to be smaller than the critical Re number. The flow is laminar."
Nusselt_x=0.332*Re_x^0.5*Pr^(1/3)
h_x=k/x*Nusselt_x
C_f_x=0.664/Re_x^0.5
```

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed:

EES Limited Academic Version: C:\EES (LIMITED ACADEMIC)\HE

File Edit Search Options Calculate Tables Plots W

Table 1 | Table 2 | Table 3 | Table 4 | Table 5 | Table 6 | Table 7 | Tab

1..11	$C_{f,x}$	h_x	x [ft]
Run 1	0.0001512	367	0.1
Run 2	0.0001097	266.3	0.19
Run 3	0.00009035	219.3	0.28
Run 4	0.00007859	190.8	0.37
Run 5	0.00007049	171.1	0.46
Run 6	0.00006446	156.5	0.55
Run 7	0.00005976	145.1	0.64
Run 8	0.00005595	135.8	0.73
Run 9	0.00005279	128.2	0.82
Run 10	0.00005012	121.7	0.91
Run 11	0.00004781	116.1	1

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File Edit Search Options Calculate Tables Plots Wi

Table 1 | Table 2 | Table 3 | Table 4 | Table 5 | Table 6 | Table 7 | Tabl

1..11	$C_{f,x}$	h_x	x [ft]
Run 1	0.00004781	116.1	1
Run 2	0.00004558	110.7	1.1
Run 3	0.00004364	105.9	1.2
Run 4	0.00004193	101.8	1.3
Run 5	0.0000404	98.09	1.4
Run 6	0.00003903	94.76	1.5
Run 7	0.0000378	91.75	1.6
Run 8	0.00003667	89.01	1.7
Run 9	0.00003563	86.5	1.8
Run 10	0.00003468	84.2	1.9
Run 11	0.0000338	82.07	2

Step 2 of 2

From the above table, we say that varies of local heat transfer coefficient along the plate intervals is small, hence we jump the distance of the plate from 2 ft to 9 ft.

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File Edit Search Options Calculate Tables Plots W

Table 1 | Table 2 | Table 3 | Table 4 | Table 5 | Table 6 | Table 7 | Tal

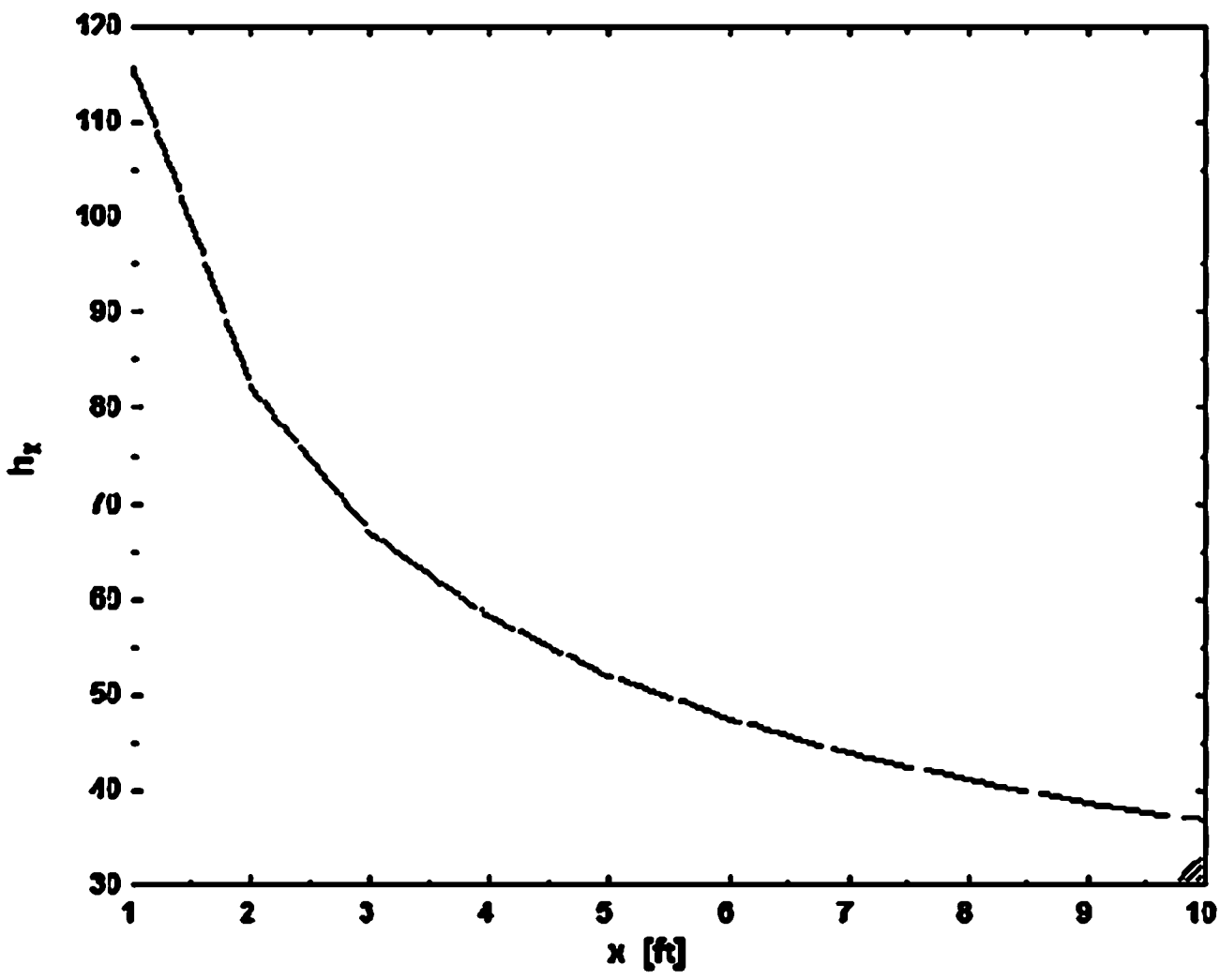
1..11	$C_{f,x}$	h_x	x [ft]
Run 1	0.00001594	38.69	9
Run 2	0.00001585	38.47	9.1
Run 3	0.00001576	38.26	9.2
Run 4	0.00001568	38.06	9.3
Run 5	0.00001559	37.85	9.4
Run 6	0.00001551	37.65	9.5

19 034P pg 2

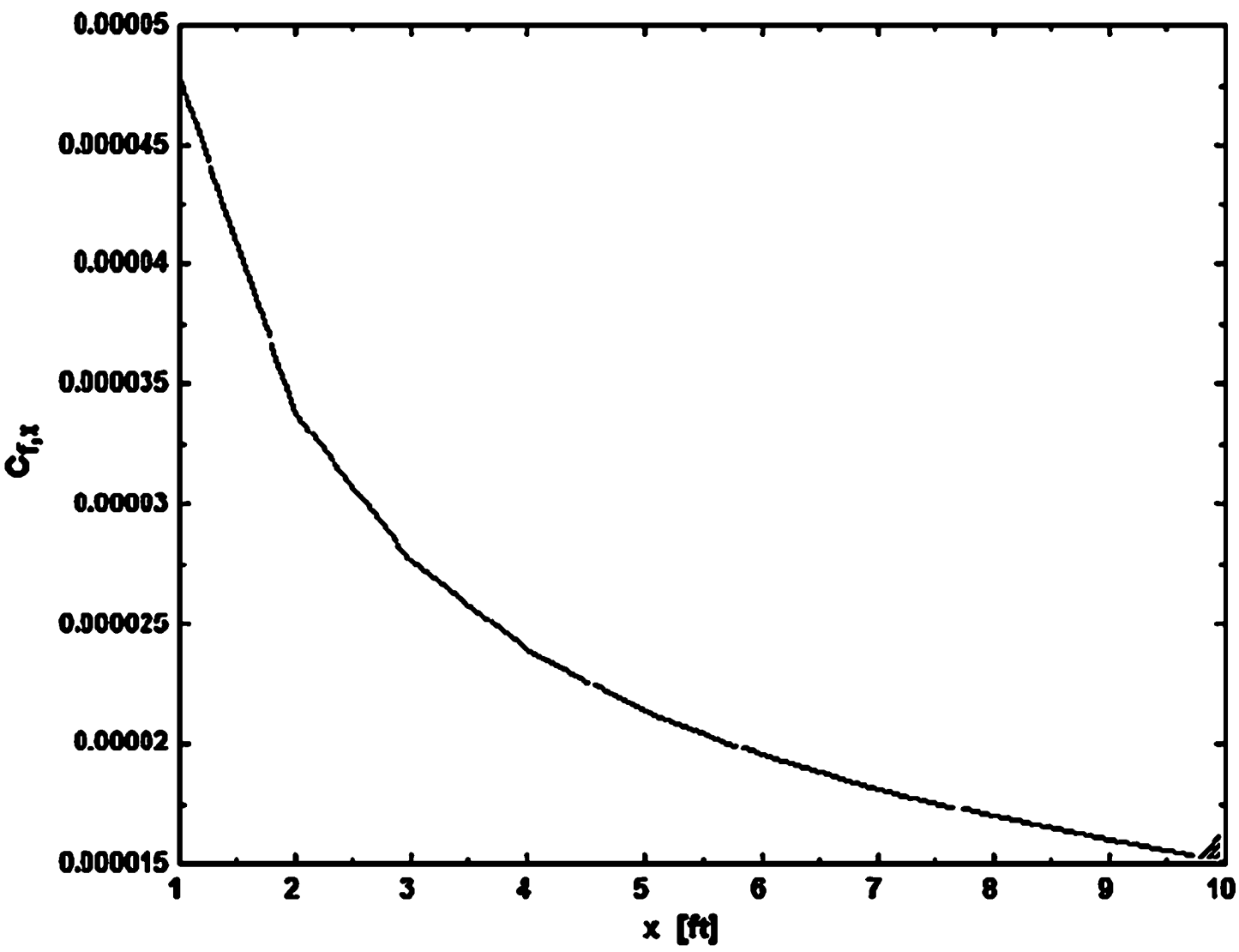
Run 2	0.00001585	38.47	9.1
Run 3	0.00001576	38.26	9.2
Run 4	0.00001568	38.06	9.3
Run 5	0.00001559	37.85	9.4
Run 6	0.00001551	37.65	9.5
Run 7	0.00001543	37.46	9.6
Run 8	0.00001535	37.26	9.7
Run 9	0.00001527	37.07	9.8
Run 10	0.00001519	36.89	9.9
Run 11	0.00001512	36.7	10

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

By plotting the graph between heat transfer coefficient and distance:



By plotting the graph between the local heat transfer coefficient and distance of the plate:



Question: 19.035P

An array of power transistors, dissipating 5 W of power each, are to be cooled by mounting them on a 25-cm × 25-cm square aluminum plate and blowing air at 35°C over the plate with a fan at a velocity of 4 m/s. The average temperature of the plate is not to exceed 65°C. Assuming the heat transfer from the back side of me Plate to be negligible and disregarding radiation, determine the number of transistors that can be placed on this plate.

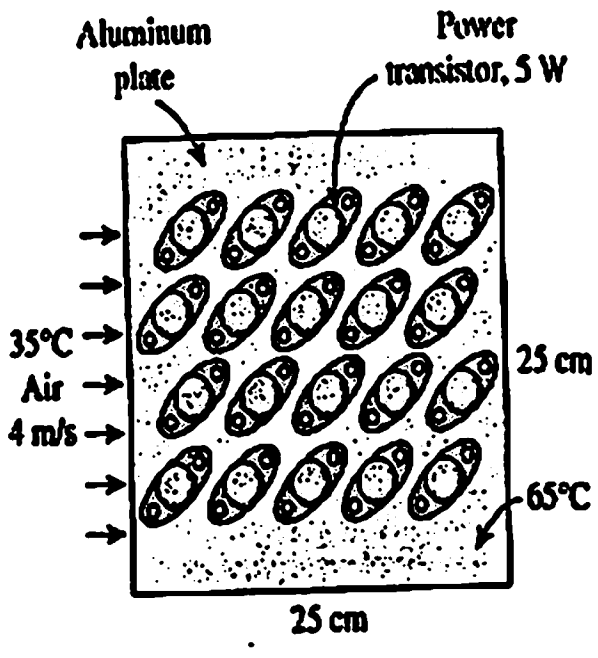


FIGURE P1 9-35

Step 1 of 3

Calculate the film temperature of the air

$$T_f = \frac{(T_s + T_\infty)}{2}$$

Substitute $35°C$ for T_∞ , and $65°C$ for T_s

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(65+35)}{2}$$
$$= 50°C$$

From the table, properties of air at $1\ \text{atm}$ pressure and at $T = 50°C$

Thermal conductivity, $k = 0.02735\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Step 2 of 3

Determine the Reynolds number

$$\text{Re} = \frac{VL}{\nu}$$

Substitute $4\ \text{m/s}$ for V , 0.25 m for L, and $1.798\times10^{-5}\ \text{m}^2/\text{s}$ for ν

$$\text{Re} = \frac{VL}{\nu}$$
$$= \frac{4\times\left(25\times10^{-2}\right)}{1.798\times10^{-5}}$$
$$= 55617.35$$

Since the obtained Reynolds number is less than 5×10^5 i.e. $\text{Re} < 5\times10^5$, the flow is said to be laminar. So determine the average Nusselt number for laminar flow substituting 55617.35 for Re and 0.7228 for Pr

$$\text{Nu} = 0.664\text{Re}^{0.5}\text{Pr}^{1/3}$$
$$= 0.664\times(55617.35)^{0.5}\times(0.7228)^{1/3}$$
$$= 140.53$$

Step 3 of 3

Calculate the heat transfer coefficient using the Nusselt number. Substitute 0.25 m for L and $0.02735\ \text{W/m}\cdot\text{K}$ for k

$$h = \frac{\text{Nu}\times k}{L}$$
$$= \frac{140.53\times0.02735}{0.25}$$
$$= 15.37\ \text{W/m}^2\cdot\text{K}$$

Determine the number of transistors that can be placed on the plate by using the heat transfer rates.

$$N = \frac{\dot{Q}}{P}$$
$$= \frac{hA(T_s - T_\infty)}{P}$$

Substitute $5\ \text{W}$ for P , $\left(0.25\times0.25\ \text{m}^2\right)$ for A , $35°C$ for T_∞ , $15.37\ \text{W/m}^2\cdot\text{K}$ for h , and $65°C$ for T_s in the above equation

$$N = \frac{hA(T_s - T_\infty)}{P}$$
$$= \frac{15.37(0.25\times0.25)(65-35)}{5}$$
$$= 5.76$$

The number of transistors that can be placed on the on the plate is $\boxed{N = 5}$

Question: 19.036P

Repeat Prob. 19-35 for a location at an elevation of 1610 m where the atmospheric pressure is 83.4 kPa.

Step 1 of 6

Given data:

Power dissipation by each transistor, $\dot{Q} = 5\,\text{W}$

Temperature of air, $T_\infty = 35°\text{C}$

Velocity of air, $V = 4\,\text{m/s}$

Size of square plate,

$L = 25\ \text{cm}$

$= 0.25\text{m}$

Plate average temperature, $T_s = 65°\text{C}$

Location elevation, $z = 1610\,\text{m}$

Atmospheric pressure,

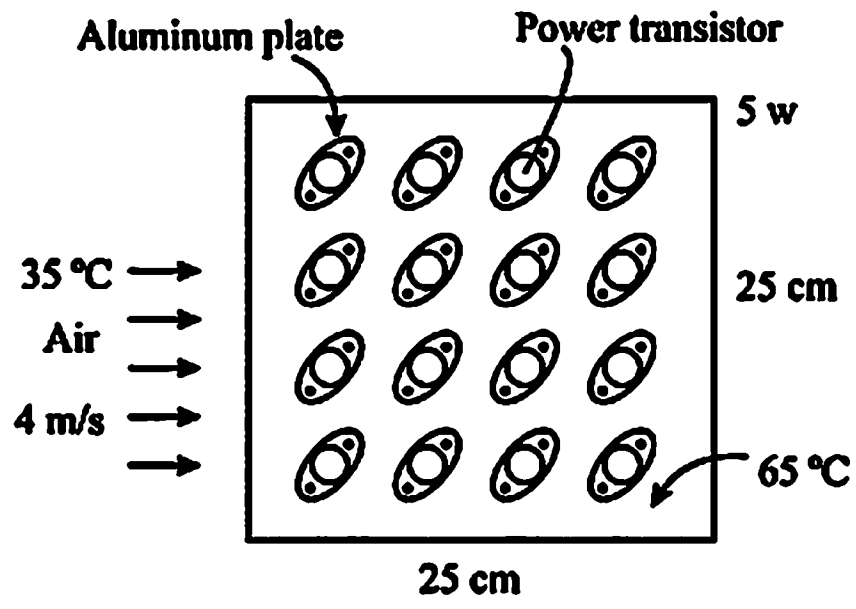
$p_{atm} = 83.4\,\text{kPa}$

$= (83.4)/101.325$

$= 0.823\ \text{atm}$

Step 2 of 6

An array of power transistors are placed on square aluminum plate is as shown below:



Step 3 of 6

Assuming no heat transfer from back side of the plate and there is no radiation heat transfer

By using the formula for film temperature, we get

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{65+35}{2}$$

$$= 50°\text{C}$$

At 1atm pressure and film temperature $T_f = 50°\text{C}$ from properties of air table

Thermal conductivity, $k = 0.02735\,\text{W/m}\cdot°\text{C}$

Kinematic viscosity, $\nu_{1\,atm} = 1.798\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

There is no change in properties of Prandtl number and thermal conductivity when pressure changes but kinematic viscosity of air at 0.823 atm is

$\nu = 1.798\times10^{-5}/0.823$

$= 2.184\times10^{-5}\,\text{m}^2/\text{s}$

Step 4 of 6

Loading solution. Please wait ...

Step 5 of 6

Nusselt number can be written as

$$\text{Nu} = \frac{hL}{k}$$

$$102.697 = \frac{h\times0.25}{0.02735}$$

$h = 11.235\,\text{W/m}^2\cdot\text{K}$

Heat transfer surface Area of the square plate

$A_s = L^2$

$= (0.25)^2$

$= 0.0625\,\text{m}^2$

Step 6 of 6

By Newton's law of cooling, the total heat transfer from the plate will be,

$\dot{Q}_t = hA_s(T_s - T_\infty)$

$= 11.235\times0.0625\times(65-35)$

$= 21.0656\,\text{W}$

Therefore Number of transistors that can be placed over plate can be

$$N = \frac{\text{Total heat transfer, } \dot{Q}_{total}}{\text{heat transfer by one transistor, } \dot{Q}}$$

$$= \frac{21.0656}{5}$$

$= 4.213$

$= 5$

Therefore the number of transistors that can be placed on this plate is $\boxed{N = 5}$

Question: 19.037P

Air at 25°C and 1 atm is flowing over a long flat plate with a velocity of 8 m/s. Determine the distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location.

Step 1 of 2

Obtain properties of air at temperature $25°C$ from table A-15, "Properties of air at 1 atm pressure" in the text book.

Density, $\rho = 1.184\ kg/m^3$ and

Kinematic viscosity, $\nu = 1.562\times10^{-5}\ m^2/s$

Find the distance from the leading edge of the plate where the flow becomes turbulent by using the following equation:

$$Re_{cr} = \frac{V_\infty x}{\nu}$$

Here, V_∞ is fluid velocity, Re_{cr} is Reynolds number where the flow is turbulent, and x is distance from the leading edge of the plate where the flow becomes turbulent.

Substitute 5×10^5 for Re_{cr}, $1.562\times10^{-5}\ m^2/s$ for ν, and 8 m/s for V_∞

$$5\times10^5 = \frac{8\times x}{1.562\times10^{-5}}$$
$$x = 0.976\,m$$

Therefore, the distance from the leading edge of the plate is $\boxed{0.976\,m}$

Step 2 of 2

Find the thickness of boundary layer by using the following equation:

$$\delta_x = \frac{5x}{Re_x^{1/2}}$$

Substitute 0.976 m for x and 5×10^5 for Re_x

$$\delta_{cr} = \frac{5\times0.976}{(5\times10^5)^{1/2}}$$
$$= 0.0069\,m$$
$$= 0.69\,cm$$

Therefore, thickness of the boundary layer where the flow becomes turbulent is $\boxed{0.69\ cm}$

Question: 19.038P

Repeat Prob. 19-37 for water.

Step 1 of 4

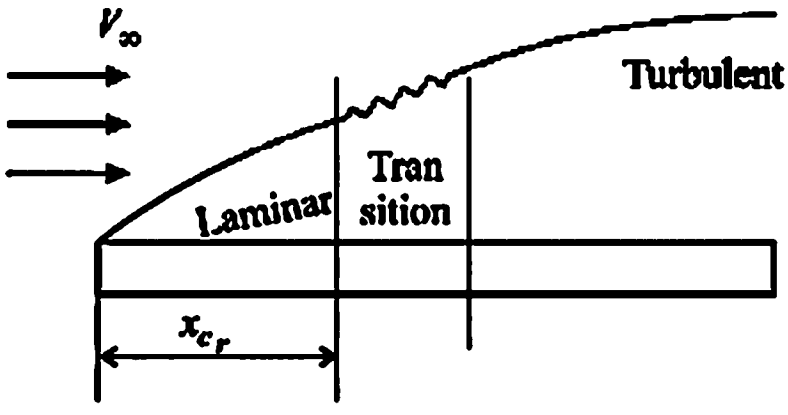
Given: - Temperature of water = 25°C

Velocity of water $V_\infty = 8$ m/s

Step 2 of 4

Properties of water at 1 atm and temperature 25°C is taken from properties of saturated water table

$\rho = 997\ \text{kg/m}^3$ and $\mu = 0.891\times10^{-3}\ \text{kg/ms}$



Step 3 of 4

Taking value of Reynolds number as 5×10^5 distance from the leading edge of the plate where the flow becomes turbulent is calculated as follows

$$\text{Re}_{cr} = \frac{\rho V_\infty x_{cr}}{\mu}$$

$$x_{cr} = \frac{\mu \text{Re}_{cr}}{\rho V_\infty}$$

$$= \frac{(0.891\times10^{-3}\ \text{kg/m}\cdot\text{s})(5\times10^5)}{(997\ \text{kg/m}^3)(8\ \text{m/s})}$$

$$= 0.056\ \text{m}$$

$$\boxed{x_{cr} = 5.6\ \text{cm}}$$

Step 4 of 4

Thickness of boundary layer is

$$\delta_{cr} = \frac{5x}{\text{Re}_x^{1/2}}$$

$$\delta_{cr} = \frac{5x_{cr}}{\text{Re}_{cr}^{1/2}}$$

$$= \frac{5(0.056\ \text{m})}{(5\times10^5)^{1/2}}$$

$$= 0.00040\ \text{m}$$

$$\boxed{\delta_{cr} = 0.4\ \text{mm}}$$

Question: 19.039P

Solar radiation is incident on the glass cover of a solar collector at a rate of 700 W/m2. The glass transmits 88 percent of the incident radiation and has an emissivity of 0.90. The entire hot water needs of a family in summer can be met by two collectors 1.2 m high and 1 m wide. The two collectors are attached to each other on one side so that they appear like a single collector 1.2 m × 2 m in size. The temperature of the glass cover is measured to be 35°C on a day when the surrounding air temperature is 25°C and the wind is blowing al 30 km/h. The effective sky temperature for radiation exchange between the glass cover and the open sky is -40°C. Water enters the tubes attached to the absorber plate at a rate of 1 kg/min. Assuming the back surface of the absorber plate to be heavily insulated and the only heat loss to occur through the glass cover, determine (*a*) the total rate of heat loss from the collector, (*b*) the collector efficiency, which is the ratio of the amount of heat transferred to the water to the solar energy incident on the collector, and (*c*) the temperature rise of water as it flows through the collector.

Tsky=-40°C

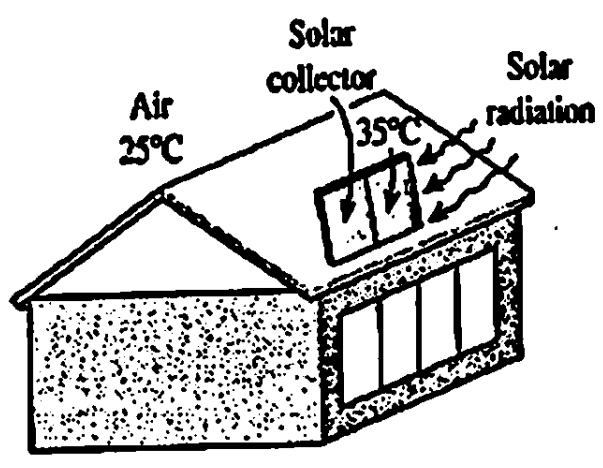


FIGURE P19-39

Step 1 of 10

Given: - Rate of incident solar radiation $= 700\ W/m^2$
Emissivity of glass $= 0.90$
Radiation transmitted by glass $= 88\%$
Dimension of collector $= 1.2\ m \times 2\ m$
Temperature of glass $= 35°C$
Surrounding temperature $= 25°C$
Velocity of wind $= 30\ km/h$

Step 2 of 10

Properties of air at 1 atm and average film temperature $\frac{(35+25)}{2} = 30°C$ is taken from properties of air table

$k = 0.02588\ W/m°C$
$\upsilon = 1.608 \times 10^{-5}\ m^2/s$
$Pr = 0.7282$

Step 3 of 10

(*a*) Reynolds number is calculated assuming that wind flows across 2 m surface

$$Re_L = \frac{VL}{\upsilon}$$
$$= \frac{(30 \times 1000/3600)m/s(2\ m)}{1.608 \times 10^{-5}\ m^2/s}$$
$$Re_L = 1.036 \times 10^6$$

Since $5 \times 0^5 \le Re_L \le 10^7$
Combined laminar and turbulent flow will be there.

Step 4 of 10

Nusslt number is calculated as follows:

$$Nu = \frac{hL}{k}$$
$$= (0.037 Re^{0.8} - 871) Pr^{1/3}$$
$$= [0.037(1.036 \times 10^6)^{0.8} - 871](0.7282)^{1/3}$$
$$Nu = 1376.98$$
$$h = \frac{k}{L} Nu$$
$$= \frac{0.02588\ W/m°C}{2\ m}(1376.98)$$
$$h = 17.82\ W/m^2\ °C$$

Step 5 of 10

Rate of heat loss by convection from collector is

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s)$$
$$= (17.82\ W/m^2\ °C)(2 \times 1.2\ m^2)(35\text{-}25)°C$$
$$\dot{Q}_{conv} = 427.68\ W$$

Step 6 of 10

Heat loss rate by radiation from collector is

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s^4 - T_{sur}^4)$$
$$= (0.90)(2 \times 1.2\ m^2)(5.67 \times 10^{-8}\ W/m^{2\circ}C)\left[(35+273\ K)^4 - (-40+273\ K)^4\right]$$
$$\dot{Q}_{rad} = 741.2\ W$$

Step 7 of 10

And $\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$
$= 427.68 + 741.2$
$\boxed{\dot{Q}_{total} = 1169.88 W}$

Step 8 of 10

(*b*) Net heat transfer rate is

$$\dot{Q}_{net} = \dot{Q}_{in} - \dot{Q}_{out}$$
$$= \alpha A I - \dot{Q}_{out}$$
$$= (0.88)(2 \times 1.2\ m^2)(700\ W/m^2) - 1169.88\ W$$
$$\dot{Q}_{net} = 1478.4 - 1169.88$$
$$\dot{Q}_{net} = 308.52\ W$$

Step 9 of 10

$$\eta_{collector} = \frac{\dot{Q}_{net}}{\dot{Q}_{in}}$$
$$= \frac{308.52\ W}{1478.4\ W}$$
$$\boxed{\eta_{collector} = 20.9\%}$$

Step 10 of 10

(*c*) Temperature change in water flowing through collector

$$\dot{Q}_{net} = \dot{m} C_p \Delta T$$
$$\Delta T = \frac{\dot{Q}_{net}}{\dot{m} C_p}$$
$$= \frac{308.52\ W}{(1/60\ kg/s)(4180\ J/kg°C)}$$
$$\boxed{\Delta T = 4.43°C}$$

Question: 19.040P

A 15-mm × 15-mm silicon chip is mounted such that the edges are flush in a substrate. The chip dissipates 1.4 W of power uniformly, while air at 20°C (1 atm) with a velocity of 25 m/s is used to cool the upper surface of the chip. If the substrate provides an unheated starting length of 15 mm, determine the surface temperature at the trailing edge of the chip. Evaluate the air properties at 50°C.

Step 1 of 4

Consider the properties of air from the table, "properties of air at $1\ \text{atm}$ pressure" recorded at temperature, $T = 50°\text{C}$

Thermal conductivity, $k = 0.02735\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Determine the characteristic length, *L*.

$$L = x + \xi$$

Here, *x* is the length of the chip and ξ is the unheated stating length.

Substitute $15\ \text{mm}$ for *x* and $15\ \text{mm}$ for ξ.

$$L = 15 + 15$$
$$L = 30\ \text{mm}$$

Determine the Reynolds number for the flow at the trailing edge $(x = L + \xi)$.

$$\text{Re} = \frac{VL}{\nu}$$

Here, Re_x is the Reynolds number and L is the characteristic length.

Substitute $25\ \text{m/s}$ for *V*, $30\ \text{mm}$ for L, and $1.798\times10^{-5}\ \text{m}^2/\text{s}$ for ν.

$$\text{Re} = \frac{25\times\left(\left(15\times10^{-3}\right)+\left(15\times10^{-3}\right)\right)}{1.798\times10^{-5}}$$
$$= 41713.014$$

Since the obtained Reynolds number is less than 5×10^5 i.e. $\text{Re} < 5\times10^5$, the flow is laminar.

Step 2 of 4

Calculate the local Nusselt number, $\text{Nu}_{x(\xi=0)}$ for laminar flow with uniform heat flux.

$$\text{Nu}_{x(\xi=0)} = 0.453\text{Re}^{0.5}\text{Pr}^{1/3}$$

Substitute 41713.014 for Re and 0.7228 for Pr.

$$\text{Nu}_{x(\xi=0)} = 0.453(41713.014)^{0.5}(0.7228)^{1/3}$$
$$\text{Nu}_{x(\xi=0)} = 83.03$$

Calculate the Nusselt number for the silicon chip whose edges are flush in substrate.

$$\text{Nu}_x = \frac{\text{Nu}_{x(\xi=0)}}{\left[1-(\xi/x)^{3/4}\right]^{1/3}}$$

Substitute 83.03 for $\text{Nu}_{x(\xi=0)}$, $15\ \text{mm}$ for *x*, and $15\ \text{mm}$ for ξ.

$$\text{Nu}_x = \frac{83.03}{\left[1-\left(\frac{15\times10^{-3}}{0.03}\right)^{3/4}\right]^{1/3}}$$
$$= 112.19$$

Step 3 of 4

Calculate the local heat transfer coefficient.

$$\text{Nu}_x = \frac{h_x x}{k}$$

Rewrite the equation.

$$h_x = \frac{\text{Nu}_x \times k}{x}$$

Substitute 112.19 for Nu_x, $15\ \text{mm}$ for *x*, and $0.02735\ \text{W/m}\cdot\text{K}$ for *k*.

$$h_x = \frac{112.19\times0.02735}{0.03}$$
$$= 102.28\ \text{W/m}^2\cdot\text{K}$$

Step 4 of 4

Consider Newton's law of cooling and calculate the surface temperature.

$$\dot{Q} = h_x A_s (T_s - T_\infty)$$
$$= h_x \times x^2 \times (T_s - T_\infty)$$

Rewrite and solve for T_s.

$$T_s = T_\infty + \frac{\dot{Q}}{h_x x^2}$$

Here, $\dot{Q}$ is the rate of heat transfer and T_∞ is ambient temperature.

Substitute $20°\text{C}$ for T_∞, $1.4\ \text{W}$ for $\dot{Q}$, $102.28\ \text{W/m}^2\cdot\text{K}$ for h_x, and $15\ \text{mm}$ for *x*.

$$T_s = 20 + \frac{1.4}{102.28\times\left(15\times10^{-3}\right)^2}$$
$$= 80.8°\text{C}$$

Thus, the surface temperature at the trailing edge of the chip is $\boxed{T_s = 80.8°\text{C}}$

Question: 19.041P

Air at 1 atm and 20°C is flowing over the top surface of a 0.5-m long thin flat plate. The air stream velocity is 50 m/s and the plate is maintained at a constant surface temperature of 180°C. Determine (*a*) the average convection heat transfer coefficient, and (*b*) repeat part (*a*) using the modified Reynolds analogy.



FIGURE P19-41

Step 1 of 5

Given data:

Pressure of the air, $p = 1$ atm

Temperature of the air, $T_\infty = 20°C$

Length of the plate, $L = 0.5$ m

Velocity of air stream, $V = 50$ m/s

Surface temperature of the plate, $T_s = 180°C$

Step 2 of 5

The film temperature of air,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(180 + 20)}{2}$$

$$= 100°C$$

From the table, properties of air at 1 atm pressure

At $T = 100°C$

Thermal conductivity, $k = 0.03095$ W/m·K

Kinematic viscosity, $\nu = 2.306 \times 10^{-5}$ m^2/s

Prandtl number, $Pr = 0.7111$

Step 3 of 5

(a)

The Reynolds number for the flow of the given system,

$$Re_L = \frac{VL}{\nu}$$

$$= \frac{50 \times 0.5}{2.306 \times 10^{-5}}$$

$$= 1084128.36$$

Since the obtained Reynolds number is greater than 5×10^5 ($Re > 5 \times 10^5$), the flow is said to be combined laminar and turbulent.

The average friction coefficient over the entire plate for the combined laminar and turbulent flow,

$$C_f = \frac{0.074}{Re_L^{1/5}} - \frac{1742}{Re_L}$$

$$= \frac{0.074}{(1084128.36)^{1/5}} - \frac{1742}{1084128.36}$$

$$= 2.987 \times 10^{-3}$$

Therefore, the average friction coefficient, $\boxed{C_f = 2.987 \times 10^{-3}}$

Step 4 of 5

(b)

The Nusselt number for the combined flow,

$$Nu_L = \left(0.037 Re_L^{0.8} - 871\right) Pr^{1/3}$$

$$= \left(0.037 \times (1084128.36)^{0.8} - 871\right) \times (0.7111)^{1/3}$$

$$= 1445.4$$

The average convection heat transfer,

$$Nu_L = \frac{hL}{k}$$

$$h = \frac{Nu_L \times k}{L}$$

$$= \frac{1445.4 \times 0.03095}{0.5}$$

$$= 89.47 \text{ W/m}^2 \cdot \text{K}$$

Therefore, the average convection heat transfer coefficient, $\boxed{h = 89.47 \text{ W/m}^2 \cdot \text{K}}$

Step 5 of 5

(c)

The Nusselt number using the modified Reynolds analogy,

$$Nu_L = C_f \frac{Re}{2} Pr^{1/3}$$

$$= 2.987 \times 10^{-3} \times \frac{1084128.36}{2} \times (0.7111)^{1/3}$$

$$= 1445.2$$

The average convection heat transfer coefficient,

$$Nu_L = \frac{hL}{k}$$

$$h = \frac{Nu_L \times k}{L}$$

$$= \frac{1445.2 \times 0.03095}{0.5}$$

$$= 89.46 \text{ W/m}^2 \cdot \text{K}$$

Therefore, the average convection heat transfer coefficient by modified Reynolds analogy, $\boxed{h = 89.46 \text{ W/m}^2 \cdot \text{K}}$

Question: 19.042P

Consider laminar flow of air across a hot circular cylinder. At what point on the cylinder will the heat transfer be highest? What would your answer be if the flow were turbulent?

Step 1 of 1

For laminar flow, the heat transfer coefficient will be the highest at the stagnation point which corresponds to $\theta = 0^\circ$. This answer changes to $\theta = 90^\circ$ to $\theta = 120^\circ$ for turbulent flow.

Question: 19.043P

A heated long cylindrical rod is placed in a cross flow of air at 20°C (1 atm) with velocity of 10 m/s. The rod has a diameter of 5 mm and its surface has an emissivity of 0.95. If the surrounding temperature is 20°C and the heat flux dissipated from the rod is 16000 W/m2, determine the surface temperature of the rod. Evaluate the air properties al 70°C.

Step 1 of 4

Given data:

Temperature of air, $T_\infty = 20°C$

Velocity of air, $V = 10$ m/s

Diameter of the rod, $D = 5$ mm

Surface emissivity of the rod, $\varepsilon = 0.95$

Temperature of the surroundings, $T_{sur} = 20°C$

Heat flux dissipated from the rod, $\dot{q}_s = 16000$ W/m^2

Step 2 of 4

From the table, properties of air at 1 atm pressure

At $T = 70°C$

Thermal conductivity, $k = 0.02881$ W/m·K

Kinematic viscosity, $\nu = 1.995\times10^{-5}$ m^2/s

Prandtl number, Pr = 0.7177

The Reynolds number for the flow,

$$\mathrm{Re} = \frac{VD}{\nu}$$

$$= \frac{10\times\left(5\times10^{-3}\right)}{1.995\times10^{-5}}$$

$$= 2506.26$$

Since the obtained Reynolds number is less than 2×10^5 ($\mathrm{Re} < 2\times10^5$), the flow is said to be laminar.

Step 3 of 4

The average Nusselt number for cross flow over cylinder,

Churchill & Bernstein relation,

$$\mathrm{Nu}_{cyl} = 0.3 + \frac{0.62\mathrm{R}_e^{1/2}\mathrm{P}_r^{1/3}}{\left[1+\left(\frac{0.4}{\mathrm{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(2506.26)^{0.5}(0.7177)^{1/3}}{\left[1+\left(\frac{0.4}{0.7177}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{2506.26}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 25.735$$

The convective heat transfer coefficient of air,

$$\mathrm{Nu}_{cyl} = \frac{hD}{k}$$

$$h = \frac{\mathrm{Nu}_{cyl}\times k}{D}$$

$$= \frac{25.735\times0.02881}{\left(5\times10^{-3}\right)}$$

$$= 148.28\ \mathrm{W/m^2\cdot K}$$

Step 4 of 4

The surface temperature of the rod,

$$\dot{q}_s = \dot{q}_{conv} + \dot{q}_{rad}$$

$$16000 = \left[h\left(T_s - T_\infty\right)\right] + \left[\varepsilon\sigma\left(T_s^4 - T_{sur}^4\right)\right]$$

$$16000 = \left[148.28\left(T_s - 20\right)\right] + \left[0.95\times5.67\times10^{-8}\left(\left(T_s + 273\right)^4 - 293^4\right)\right]$$

$$T_s = 121.8°C$$

Therefore the surface temperature of the rod, $\boxed{T_s = 121.8°C}$

Question: 19.044P

A stainless steel ball (ρ = 8055 kg/m3, *cp* = 480 J/kg·K) of diameter *D* =15 cm is removed from the oven at a uniform temperature of 350°C. The ball is then subjected to the flow of air at 1 atm pressure and 30°C with a velocity of 6 m/s. The surface temperature of the ball eventually drops to 250°C. Determine the average convection heat transfer coefficient during this cooling process and estimate how long this process has taken.

Step 1 of 6

Take the properties of air at 1 atm and average film temperature $\frac{(350+250)}{2}=300°C$ from the table A-21, "Ideal gas properties of air" in the textbook.

Thermal conductivity, $k=0.04418\ \text{W/m}\cdot°\text{C}$

Kinematic viscosity, $\upsilon=4.765\times10^{-5}\ \text{m}^2/\text{s}$

Density, $\rho=0.6158\text{kg/m}^3$

Specific heat, $C_p=1044\text{J/kg}\cdot\text{K}$

Dynamic viscosity, $\mu_{@30°C}=1.872\times10^{-5}\ \text{kg/m}\cdot\text{s}$

Dynamic viscosity, $\mu_{s,@300\,°C}=2.934\times10^{-5}\ \text{kg/m}\cdot\text{s}$

Prandtl number, $\text{Pr}=0.6935$

Step 2 of 6

Find the Reynolds number by using the equation as follows:

$$\text{Re}=\frac{VD}{\upsilon}$$

Here, *V* is velocity of air flow, and *D* is diameter of the steel ball.

Substitute 6 m/s for *V* and 0.15 m for *D*.

$$\text{Re}=\frac{6\times0.15}{4.765\times10^{-5}}$$
$$=1.888\times10^{4}$$

Find the Nusselt number by using the equation as follows:

$$\text{Nu}=2+\left[0.4\,\text{Re}^{0.5}+0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Substitute 1.888×10^{4} for Re, 0.6935 for Pr, 1.872×10^{-5} for μ_∞, and 2.934×10^{-5} for μ_s.

$$\text{Nu}=2+\left[0.4(1.888\times10^{4})^{0.5}+0.06(1.888\times10^{4})^{2/3}\right](0.6935)^{0.4}\left(\frac{1.872\times10^{-5}}{2.934\times10^{-5}}\right)^{1/4}$$
$$=2+[54.96+42.514]0.772$$
$$=77.249$$

Step 3 of 6

Find the heat transfer coefficient by using Nusselt's number value as follows:

$$\text{Nu}=\frac{hD}{k}$$

$$h=\frac{k}{D}\text{Nu}$$
$$=\frac{0.04418}{0.15}\times77.249$$
$$=22.752\ \text{W/m}^2\cdot\text{K}$$

Therefore, heat transfer coefficient during the cooling process is $\boxed{22.752\ \text{W/m}^2\cdot\text{K}}$

Find the average convection heat transfer rate by using the equation as follows:

$$\dot{Q}_{conv}=hA_s(T_s-T_\infty)$$

Here, A_s is the surface area of the steel ball.

Calculate the surface area of the shell as follows:

$$A_s=\pi D^2$$

Substitute 0.15 m for *D*.

$$A_s=\pi(0.15)^2$$
$$=0.07069\ \text{m}^2$$

Substitute $0.07069\ \text{m}^2$ for A_s, 350°C for T_s and 30°C for T_∞.

$$\dot{Q}_{conv}=hA_s(T_s-T_\infty)$$
$$=22.752\times(0.07069)(350\text{-}30)$$
$$=514.668\ \text{W}$$

Step 4 of 6

Find the total heat transfer rate as follows:

$$Q_{total}=mC_{p,ball}(T_1-T_2)$$

Here, m is mass of the ball, T_1 is initial temperature and T_2 is final temperature, and $C_{p,ball}$ is specific heat of steel ball.

Find the mass of steel ball by using the equation.

$$m=\rho V$$
$$=\rho\times\frac{\pi D^3}{6}$$

Substitute 0.15 m for *D*.

$$m=(8055)\frac{\pi(0.15)^3}{6}$$
$$=14.234\,\text{kg}$$

Step 5 of 6

Now find the total heat transfer by substituting 350°C for T_1, 250°C for T_2 and 480 J/kg for $C_{p,ball}$ in the equation,

$$Q_{total}=14.234\times480\times(350-250)$$
$$=683232\,\text{J}$$

Calculate the time of cooling process as follows:

$$\Delta t=\frac{Q_{total}}{\dot{Q}_{conv}}$$
$$=\frac{683232}{514.668}$$
$$=1327.519\,\text{s}$$
$$=22.125\,\text{min}$$

Therefore, the total time taken during the cooling process is $\boxed{22.125\,\text{min}}$.

Step 6 of 6

Question: 19.045P

Reconsider Prob. 19-44. Using EES (or other) software, investigate the effect of air velocity on the average convection heat transfer coefficient and the cooling time. Let the air velocity vary from 1 m/s to 10 m/s. Plot the heat transfer coefficient and the cooling time as a function of air velocity, and discuss the results.

Step 1 of 4

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 4

Equation window:

```
"Known"
D=0.15 "[m]"
T_1=350 "[C]"
T_2=250 "[C]"
T_infinity=30 "[C]"
P=101.3 "[kPa]"
"Vel=6 [m/s], parameter to be varied"
rho_ball=8055 "[kg/m^3]"
C_p_ball=480 "[J/kg-C]"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=P)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
nu=mu_infinity/rho
mu_s=Viscosity(Fluid$, T=T_s_ave)
T_s_ave=1/2*(T_1+T_2)

"ANALYSIS"
Re=(Vel*D)/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=k/D*Nusselt
A=pi*D^2
Q_dot_ave=h*A*(T_s_ave-T_infinity)
Q_total=m_ball*C_p_ball*(T_1-T_2)
m_ball=rho_ball*V_ball
V_ball=(pi*D^3)/6
time=Q_total/Q_dot_ave*Convert(s, min)
```

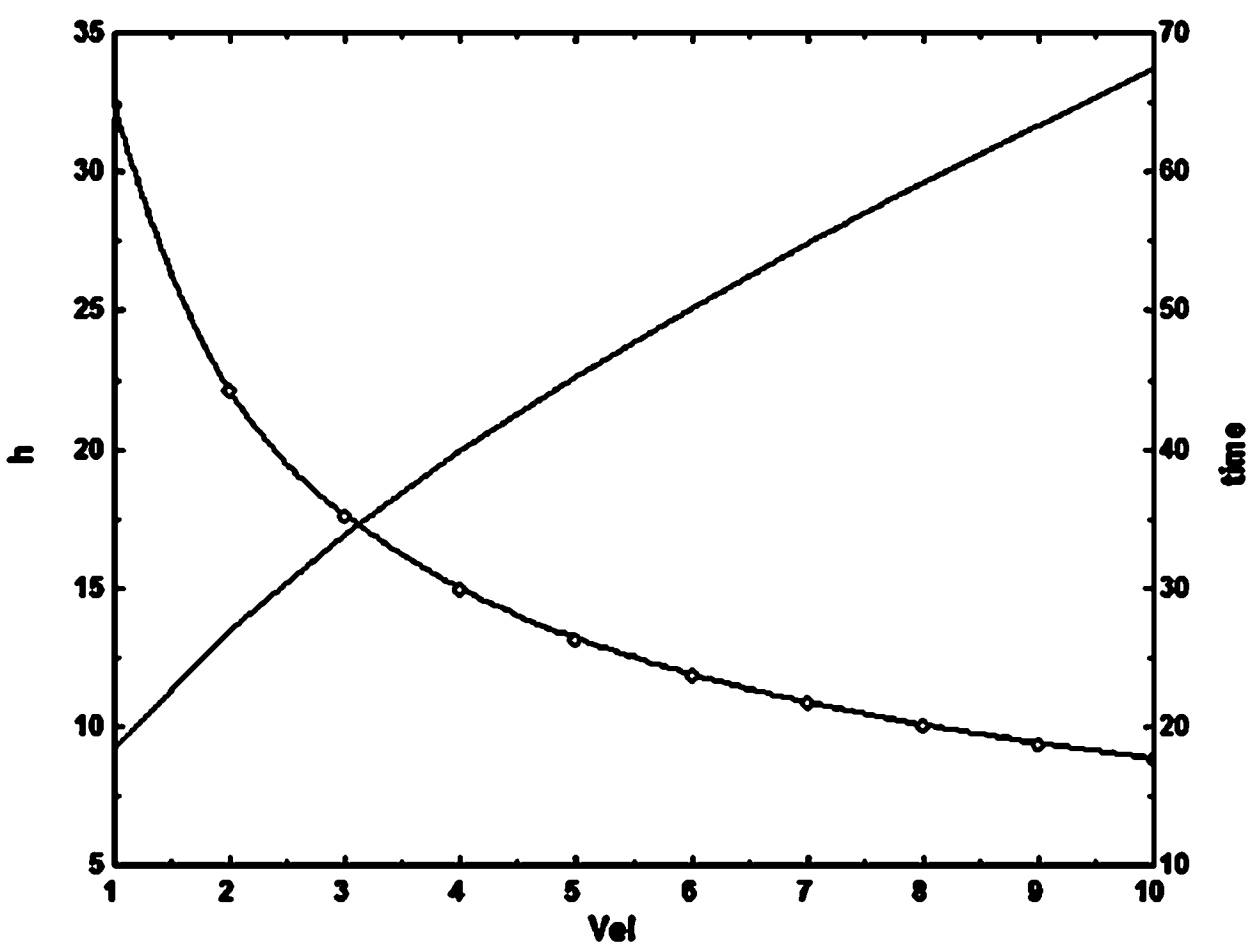
Step 3 of 4

Table of heat transfer coefficient and time as a function of velocity:

1..10	1 Vel	2 time	3 h
Run 1	1	64.87	9.198
Run 2	2	44.23	13.49
Run 3	3	35.23	16.93
Run 4	4	29.94	19.93
Run 5	5	26.37	22.62
Run 6	6	23.77	25.11
Run 7	7	21.76	27.42
Run 8	8	20.15	29.61
Run 9	9	18.83	31.69
Run 10	10	17.72	33.68

Step 4 of 4

Graph between heat transfer coefficient and time as a function of velocity:



Question: 19.046P

A person extends his uncovered arms into the windy air outside at 54°F and 30 mph in order to feel nature closely. Initially, the skin temperature of the arm is 84°F. Treating the arm as a 2-ft-long and 3-in-diameter cylinder, determine the rate of heat loss from the arm.

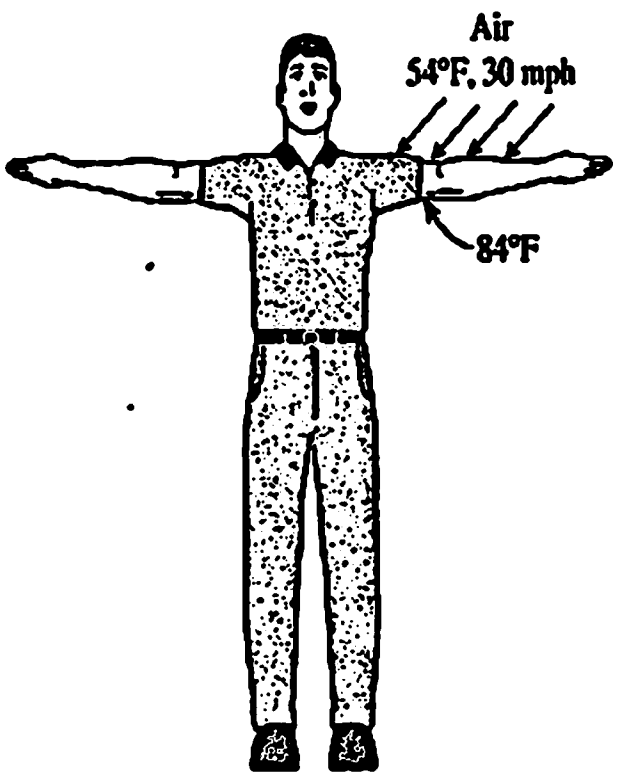


FIGURE P19-46E

Step 1 of 4

Given data:

Temperature of air, $T_\infty = 54°F$

Velocity of air,

$$V = 30\,\text{mph} = \frac{30\times 5280}{3600} = 44\ \text{ft/s}$$

Skin temperature of the arm, $T_s = 84°F$

Diameter of arm, $D = 3\text{in} = 0.25\text{ft}$

Length of the arm, $L = 2\text{ft}$

Step 2 of 4

The film temperature,

$$T_f = \frac{(T_s + T_\infty)}{2} = \frac{(84+54)}{2} = 69°F$$

From the table, properties of air at 1 atm pressure

At $T_f = 69°F$

By linear interpolation

Thermal conductivity,

$$k = 0.01433 + \frac{0.01457 - 0.01433}{(70-60)}(69-60) = 0.014546\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$$

Kinematic viscosity,

$$\nu = 1.588\times 10^{-4} + \frac{(1.643-1.588)\times 10^{-4}}{(70-60)}(69-60) = 0.73075\ \text{ft}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = 0.7321 + \frac{0.7306 - 0.7321}{(70-60)}(69-60) = 0.73075$$

Step 3 of 4

The Reynolds number for the flow,

$$\text{Re} = \frac{VD}{\nu} = \frac{44\times 0.25}{1.6375\times 10^{-4}} = 67175.57$$

Since the obtained Reynolds number is less than 2×10^5 i.e. $\text{Re} < 2\times 10^5$, the flow is said to be laminar.

The Nusselt number,

$$\text{Nu}_{cyl} = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+\left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(67175.57)^{0.5}(0.73075)^{1/3}}{\left[1+\left(\frac{0.4}{0.73075}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{67175.57}{282000}\right)^{5/8}\right]^{4/5} = 167.73$$

Step 4 of 4

The convective heat transfer coefficient,

$$\text{Nu}_{cyl} = \frac{hD}{k}$$

$$h = \frac{\text{Nu}_{cyl}\times k}{D} = \frac{167.73\times 0.014546}{0.25} = 9.76\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

The rate of heat loss from the arm,

$$\dot{Q} = hA_s(T_s - T_\infty) = h\times(\pi DL)\times(T_s - T_\infty)$$

$$= 9.76\times(\pi\times 0.25\times 2)\times(84-54) = 459.93\ \text{Btu/h}$$

Therefore the rate of heat loss from the arm, $\boxed{\dot{Q} = 459.93\ \text{Btu/h}}$

Question: 19.047P pg 1

Reconsider Prob. 19-46E. Using EES (or other) software, investigate the effects of air temperature and wind velocity on the rate of heat loss from the arm. Let the air temperature vary from 20°F to 80°F and the wind velocity from 10 mph to 40 mph. Plot the rate of heat loss as a function of air temperature and of wind velocity, and discuss the results.

Step 1 of 8

The given problem can be solved using EES software by following the below steps.

1. **Open EES software**
2. **Create a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in dailog box**

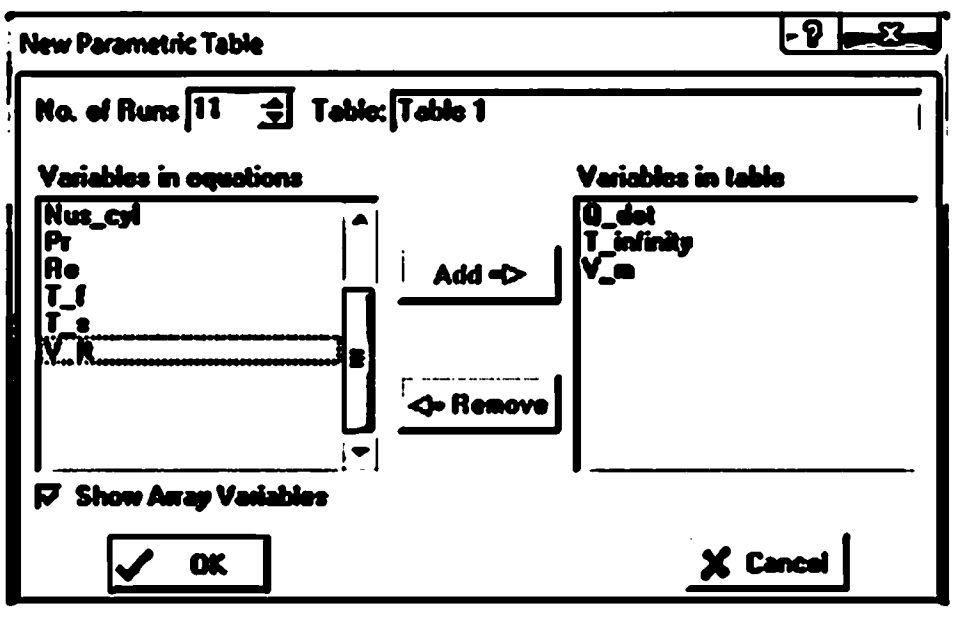
```
"T_infinity is varied from 20[F] to 80[F]"          "Temperature of air"
"V_m=10[mph] to 40[mph]"                            "Velocity of air"
V_ft=V_m*convert(mile/hr,ft/sec)
T_s=84[F]                                           "Temperature of skin"
D=3[in]*convert(in,ft)                              "Diameter of arm"
L=2[ft]                                             "Length of arm"

T_f=(T_s+T_infinity)/2                              "Film temperature"
k=Conductivity(Air,T=T_f)                           "Properties of air at T_f temperature"
nu=Viscosity(Air,T=T_f)/Density(Air,T=T_f,P=1[atm]*convert(atm,psi))*convert(ft^2/hr,ft^2/s)
Pr=Prandtl(Air,T=T_f)
Re=V_ft*D/nu                                        "Reynolds number"
Nus_cyl=0.3+0.62*Re^0.5*Pr^(1/3)/(1+(0.4/Pr)^(2/3))^(1/4)*(1+(re/282000)^(5/8))^(4/5)
Nus_cyl=h*D/k
A_s=pi*D*L
Q_dot=h*A_s*(T_s-T_infinity)
```

Step 2 of 8

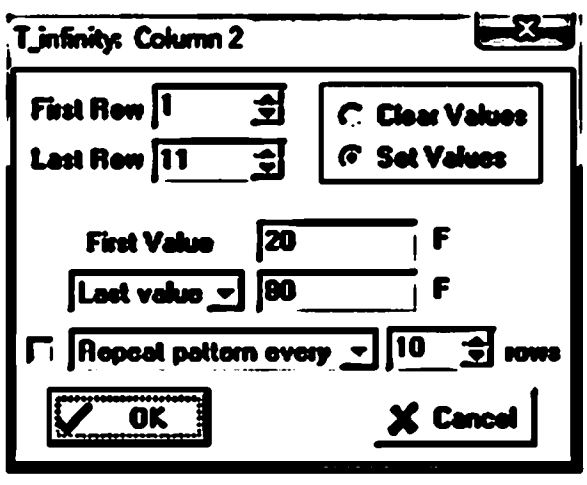
The units are in English system. So, click on "unit system" in "options menu" and change to English system.

Now to solve the equation for a range of air temperatures and range of wind velocities, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "Q_dot", "T_infinity", and "V_m" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 8

First the temperature of air is varied at constant overall constant velocity of air, V = 30 mph
To enter the values in the "T_infinity" column, click on the drop tab in "T_infinity" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 8

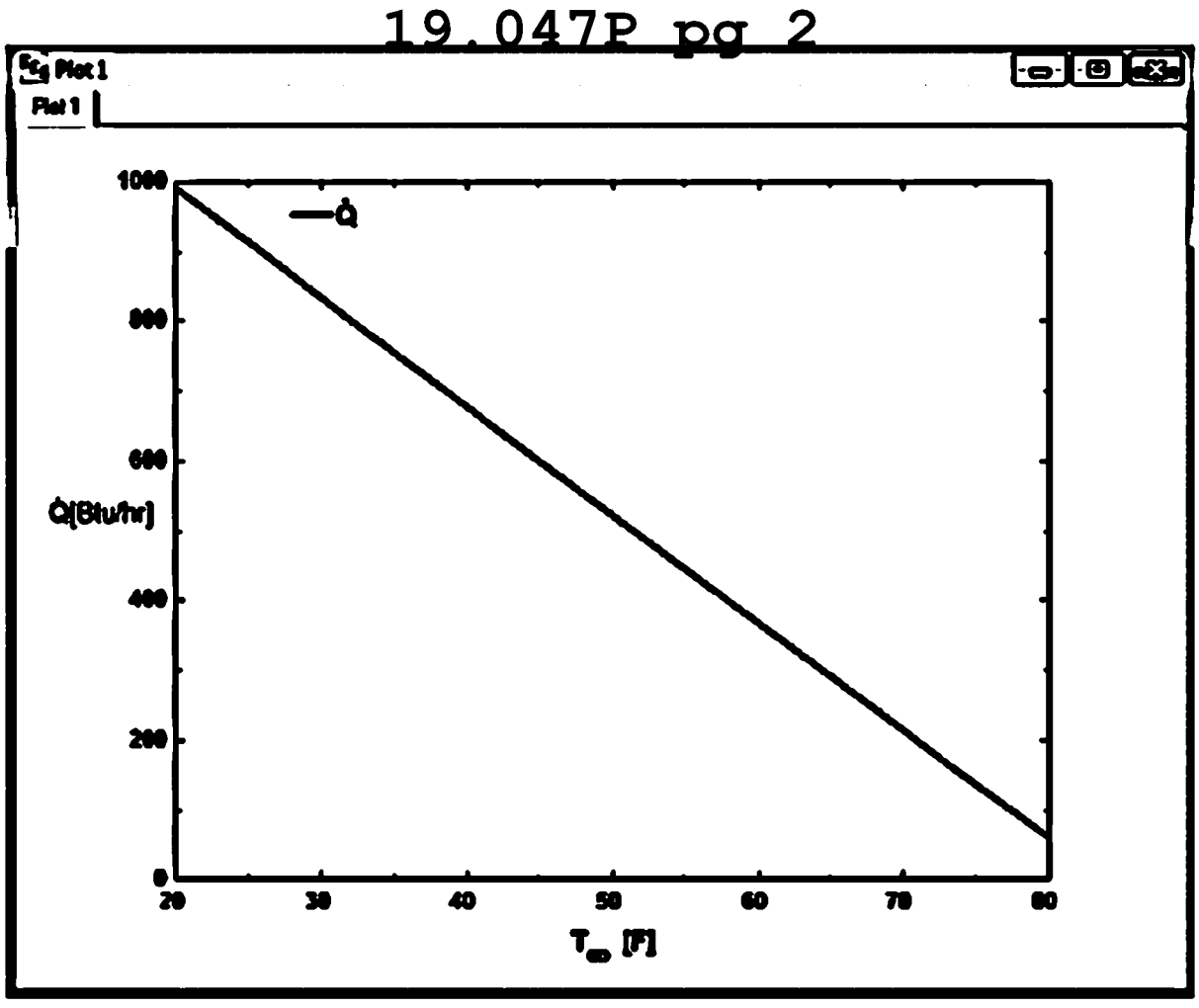
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	1 $\dot{Q}$	2 T_∞ [F]	3 V_m
Run 1	990.4	20	30
Run 2	896	26	30
Run 3	801.8	32	30
Run 4	708.1	38	30
Run 5	614.6	44	30
Run 6	521.5	50	30
Run 7	428.8	56	30
Run 8	336.3	62	30
Run 9	244.2	68	30
Run 10	152.4	74	30
Run 11	60.85	80	30

Step 5 of 8

Inorder to plot the rate of heat loss as function of temperature of air, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot showing the rate of heat loss as function of temperature of air is shown:

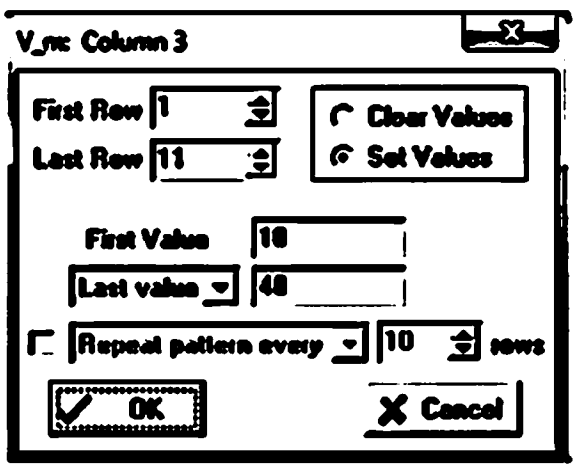


From the above plot we can interpret that as the temperature of air increases, the rate of heat loss decreases.

Step 6 of 8

Now the velocity of air is varied at constant temperature of air, $T_\infty = 54°F$

To enter the values in the "V_m" column, click on the drop tab in "V_m" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 7 of 8

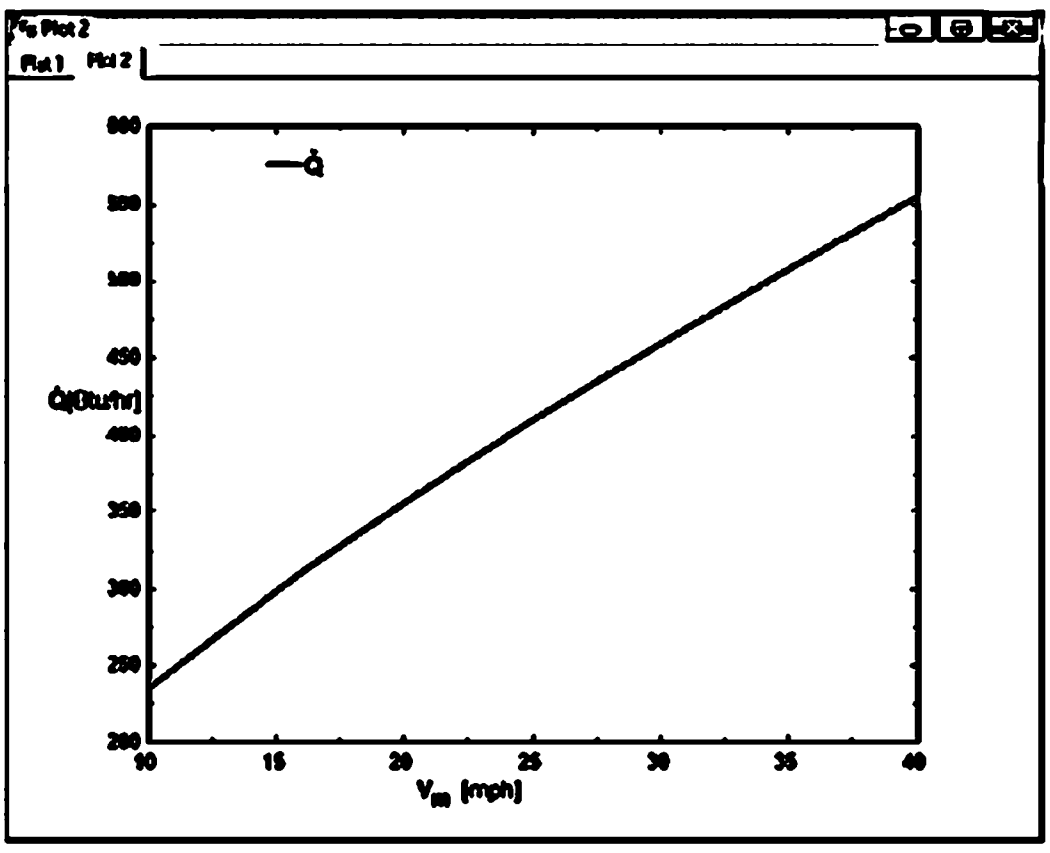
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	1 Q̇	2 T_∞ [F]	3 V_m
Rm 1	234.8	54	10
Rm 2	274	54	13
Rm 3	310.4	54	16
Rm 4	344.8	54	19
Rm 5	377.6	54	22
Rm 6	409.2	54	25
Rm 7	439.8	54	28
Rm 8	469.5	54	31
Rm 9	498.5	54	34
Rm 10	526.9	54	37
Rm 11	554.7	54	40

Step 8 of 8

Inorder to the rate of heat loss as function of velocity of air, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot showing the rate of heat loss as function of velocity of air is shown:



From the above plot we can interpret that as the velocity of air increases, the rate of heat loss also increases.

Question: 19.048P

An average person generates heat at a rate of 84 W while resting. Assuming one-quarter of this heat is lost from the head and disregarding radiation, determine the average surface temperature of the head when it is not covered and is subjected to winds at 10°C and 25 km/h. The head can be approximated as a 30-cm-diameter sphere.

Step 1 of 5

From the properties of air table,

At pressure $1\,\text{atm}$ and average film temperature $10°\text{C}$.

The thermal conductivity is $k = 0.02439\ \text{W/m.°C}$

The kinematic viscosity is $\upsilon = 1.426\times10^{-5}\ \text{m}^2/\text{s}$

The dynamic viscosity is $\mu_\infty = 1.778\times10^{-5}\ \text{kg/m.s}$

At $15°\text{C}$

The dynamic viscosity is $\mu_{s,@15°\text{C}} = 1.802\times10^{-5}\ \text{kg/m.s}$

The Prandtl number is $\text{Pr} = 0.7336$

Step 2 of 5

Calculate the Reynolds number,

$$\text{Re} = \frac{VD}{\upsilon}$$

Here,

The velocity of the wind is V

The diameter of the sphere is D

Substitute $6.94\,\text{m/s}$ for V, $0.3\,\text{m}$ for D and $1.426\times10^{-5}\ \text{m}^2/\text{s}$ for υ

$$\begin{aligned}\text{Re} &= \frac{VD}{\upsilon}\\ &= \frac{(6.94\,\text{m/s})(0.3\,\text{m})}{1.426\times10^{-5}\ \text{m}^2/\text{s}}\\ &= 1.461\times10^5\end{aligned}$$

Step 3 of 5

Calculate the Nusselt number,

$$Nu = 2+\left[0.4\,\text{Re}^{0.5}+0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Substitute 1.461 for Re, 0.7336 for Pr, $1.778\times10^{-5}\ \text{kg/m}\cdot\text{s}$ for μ_∞ and

$1.802\times10^{-5}\ \text{kg/m.s}$ for $\mu_{s,@15°\text{C}}$

$$\begin{aligned}Nu &= 2+\left[0.4\,\text{Re}^{0.5}+0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}\\ &= 2+\left[0.4(1.461\times10^5)^{0.5}+0.06(1.461\times10^4)^{2/3}\right](0.7336)^{0.4}\left(\frac{1.778\times10^{-5}}{1.802\times10^{-5}}\right)^{1/4}\\ &= 283.2\end{aligned}$$

Calculate the Coefficient for heat transfer,

$$h = \left(\frac{k}{D}\right)Nu$$

Substitute $0.02439\ \text{W/m}\cdot°\text{C}$ for k, $0.3\,\text{m}$ for D and 283.2 for Nu

$$\begin{aligned}h &= \left(\frac{k}{D}\right)Nu\\ &= \left(\frac{0.02439\ \text{W/m.°C}}{0.3\,\text{m}}\right)283.2\\ &= 23.02\ \text{W/m}^2\cdot°\text{C}\end{aligned}$$

Step 4 of 5

Calculate the Surface temperature of the head,

$$T_s = T_\infty + \frac{\dot{Q}}{hA_s} \quad \text{..... (1)}$$

Here,

The heat generated by the average person is $\dot{Q}$

The surface area of the head is A_s

Find the surface area of the head,

$$A_s = \pi D^2$$

Substitute $0.3\,\text{m}$ for D,

$$\begin{aligned}A_s &= \pi D^2\\ &= \pi(0.3\,\text{m})^2\\ &= 0.2827\ \text{m}^2\end{aligned}$$

Step 5 of 5

Substitute $23.02\ \text{W/m}^2\cdot°\text{C}$ for h, $0.2827\ \text{m}^2$ for A_s and $84\,\text{W}$ for $\dot{Q}$ in equation (1)

$$\begin{aligned}T_s &= T_\infty + \frac{\dot{Q}}{hA_s}\\ &= 10\ °\text{C}+\left(\frac{(84/4)}{(23.02\ \text{W/m}^2.°\text{C})(0.2827\ \text{m}^2)}\right)\\ &= 13.2\ °\text{C}\end{aligned}$$

Therefore, the average surface temperature of the person is $\boxed{13.2\ °\text{C}}$

Question: 19.049P

A long 12-cm-diameter steam pipe whose external surface temperature is 90°C passes through some open area that is not protected against the winds. Determine the rate of heat loss from the pipe per unit of its length when the air is at 1 atm pressure and 7°C and the wind is blowing across the pipe at a velocity of 65 km/h.

Step 1 of 4

Given data:

Diameter of the pipe, $D = 12\,\text{cm}$

Surface temperature of the pipe, $T_s = 90°\text{C}$

Temperature of the air, $T_\infty = 7°\text{C}$

Pressure of the air, $p = 1\,\text{atm}$

Wind flow velocity,

$$V = 65\,\text{km/h} = \frac{65\times 1000}{3600}$$

$$V = 18.06\,\text{m/s}$$

Step 2 of 4

The film temperature,

$$T_f = \frac{(T_s + T_\infty)}{2} = \frac{(90+7)}{2} = 48.5°\text{C}$$

From the table, properties of air at 1 atm pressure

At $T_f = 48.5°\text{C}$

By linear interpolation

Thermal conductivity,

$$k = 0.02699 + \frac{0.02735 - 0.02699}{50-45}(48.5-45) = 0.027242\,\text{W/m}\cdot\text{K}$$

Kinematic viscosity,

$$\nu = 1.750\times 10^{-5} + \frac{(1.798-1.750)10^{-5}}{(50-45)}(48.5-45) = 1.7836\times 10^{-5}\,\text{m}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = 0.7241 + \frac{0.7228-0.7241}{(50-45)}(48.5-45) = 0.72319$$

Step 3 of 4

The Reynolds number for the flow,

$$\text{Re} = \frac{VD}{\nu} = \frac{18.06\times\left(12\times 10^{-2}\right)}{1.7836\times 10^{-5}} = 121507.06$$

Since the obtained Reynolds number is less than 2×10^5 i.e. $\text{Re} < 2\times 10^5$, the flow is said to be laminar.

The Nusselt number,

$$\text{Nu}_{cyl} = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+\left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(121507.06)^{0.5}(0.72319)^{1/3}}{\left[1+\left(\frac{0.4}{0.72319}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{121507.06}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 247.56$$

Step 4 of 4

The convective heat transfer coefficient,

$$\text{Nu}_{cyl} = \frac{hD}{k}$$

$$h = \frac{\text{Nu}_{cyl}\times k}{D} = \frac{247.56\times 0.027242}{\left(12\times 10^{-2}\right)} = 56.2\,\text{W/m}^2\cdot\text{K}$$

The rate of heat loss from the pipe per unit length,

$$\dot{Q} = hA_s(T_s - T_\infty) = h\times(\pi DL)\times(T_s - T_\infty)$$

$$= 56.2\times\left(\pi\times 12\times 10^{-2}\times 1\right)\times(90-7) = 1758.5\,\text{W/m}$$

Therefore the rate of heat loss from the pipe per unit length, $\boxed{\dot{Q} = 1758.5\,\text{W/m}}$

Question: 19.050P

In a geothermal power plant, the used geothermal water at 80°C enters a 15-cm-diameter and 400-m-long uninsulated pipe at a rate of 8.5 kg/s and leaves at 70°C before being reinjected back to the ground. Windy air at 15°C flows normal to the pipe. Disregarding radiation, determine the average wind velocity in km/h.

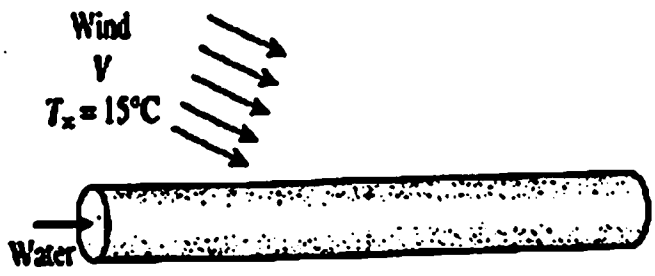


FIGURE P19-50

Step 1 of 4

Calculate the film temperature.

$$T_f = \frac{\frac{(70+80)}{2}+15}{2}$$
$$= 45°C$$

Obtain the properties of air at 1 atm and average film temperature $45°C$ from properties of air table.

Thermal conductivity is, $k = 0.02699\ W/m°C$

Kinematic viscosity is, $\upsilon = 1.75\times10^{-5}\ m^2/s$

Prandtl number is, $Pr = 0.7241$

Obtain the Specific heat of water at average temperature of water $75°C$ is $C_p = 4193\ J/kg°C$

Step 2 of 4

Energy change from inlet to outlet gives rate of heat transfer from pipe

$$\dot{Q} = \dot{m}C_p(\Delta T)$$
$$= (8.5)(4193\times(80\text{-}70))$$
$$\dot{Q} = 356{,}405\ W$$

Calculate the coefficient of heat transfer by using the following relation:

$$h = \frac{\dot{Q}}{A_s(T_s - T_\infty)}$$
$$= \frac{356{,}150W}{\pi(0.15\ m)(400m)(75-15)°C}$$
$$h = 31.51 W/m^2\,°C$$

Step 3 of 4

Calculate the Nusselt number by using the following relation:

$$Nu = \frac{hD}{k}$$
$$= \frac{(31.51W/m^2\,°C)(0.15m)}{0.02699W/m°C}$$
$$Nu = 175.13$$

Calculate the Reynolds number from Nusselt number equation.

$$Nu = 0.3 + \frac{0.62\,Re^{0.5}\,Pr^{1/3}}{\left[1+\left(\frac{0.4}{Pr}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{Re}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$175.13 = 0.3 + \frac{0.62(Re)^{0.5}\times(0.7241)^{1/3}}{\left[1+\left(\frac{0.4}{0.7241}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{Re}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$175.13 = 0.3 + 0.4895(Re)^{0.5}\left(\frac{2549.21+Re^{5/8}}{2549.21}\right)^{4/5}$$

$$Re = 71{,}900$$

Step 4 of 4

Calculate the velocity of wind.

$$Re = \frac{VD}{\upsilon}$$
$$71{,}900 = \frac{V(0.15m)}{1.726\times10^{-5}\ m^2/s}$$
$$V = 8.273\ m/s\times\frac{18}{5}$$
$$V = 29.783\ kmph$$

Therefore the wind velocity is, $\boxed{V = 29.783\ kmph}$.

Question: 19.051P

A 5-mm-diameter electrical transmission line carries an electric current of 50 A and has a resistance of 0.002 ohm per meter length. Determine the surface temperature of the wire during a windy day when the air temperature is 10°C and the wind is blowing across the transmission line at 50 km/h.

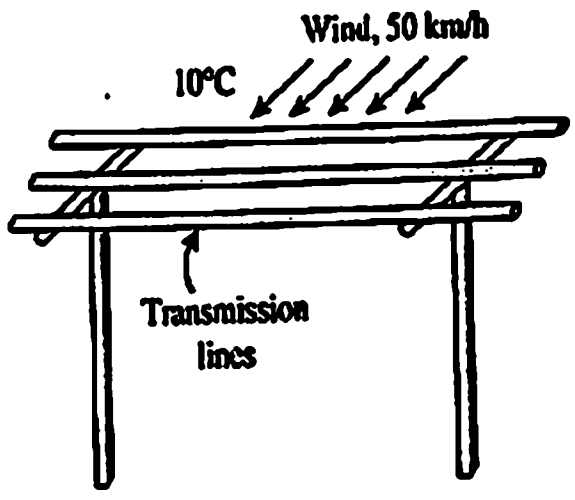


FIGURE P19-51

Step 1 of 5

Given data:

Diameter of transmission line, $D = 5\,\text{mm}$

Electrical current, $I = 50\,\text{A}$

Resistance of the transmission line per meter length, $R = 0.002\ \Omega/\text{m}$

Temperature of the air, $T_\infty = 10°\text{C}$

Air flow velocity,

$$V = 50\ \text{km/h} = \frac{50\times1000}{3600} = 13.89\ \text{m/s}$$

Step 2 of 5

The film temperature of air is assumed as $T_f = 10°\text{C}$

From the table, properties of air at 1 atm

At $T_f = 10°\text{C}$

Thermal conductivity, $k = 0.02439\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.426\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7336$

The Reynolds number for the flow,

$$\text{Re} = \frac{VD}{\nu} = \frac{13.89\times\left(5\times10^{-3}\right)}{1.426\times10^{-5}} = 4870.26$$

Since the obtained Reynolds number is less than 2×10^5 ($\text{Re} < 2\times10^5$), the flow is said to be laminar.

Step 3 of 5

The Nusselt number,

$$\text{Nu}_{cyl} = 0.3+\frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+\left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3+\frac{0.62(4870.26)^{0.5}(0.7336)^{1/3}}{\left[1+\left(\frac{0.4}{0.7336}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4870.26}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 36.8$$

Step 4 of 5

The convective heat transfer coefficient,

$$\text{Nu}_{cyl} = \frac{hD}{k}$$

$$h = \frac{\text{Nu}_{cyl}\times k}{D} = \frac{36.8\times0.02439}{\left(5\times10^{-3}\right)} = 179.5\ \text{W/m}^2\cdot\text{K}$$

The rate of heat generation in the transmission line per unit length,

$$\dot{Q} = I^2R = 50^2\times0.002 = 5\ \text{W/m}$$

Step 5 of 5

The surface temperature of the wire,

$$\dot{Q} = hA_s\left(T_s - T_\infty\right)$$

$$T_s = T_\infty + \frac{\dot{Q}}{hA_s} = T_\infty + \frac{\dot{Q}}{h\times(\pi DL)} = 10+\frac{5}{179.5\times\left(\pi\times5\times10^{-3}\times1\right)} = 11.8°\text{C}$$

Therefore the surface temperature of the wire, $\boxed{T_s = 11.8°\text{C}}$

Question: 19.052P

Reconsider Prob. 19-51. Using EES (or other) software, investigate the effect of the wind velocity on the surface temperature of the wire. Let the wind velocity vary from 10 km/h to 80 km/h. Plot the surface temperature as a function of wind velocity, and discuss the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

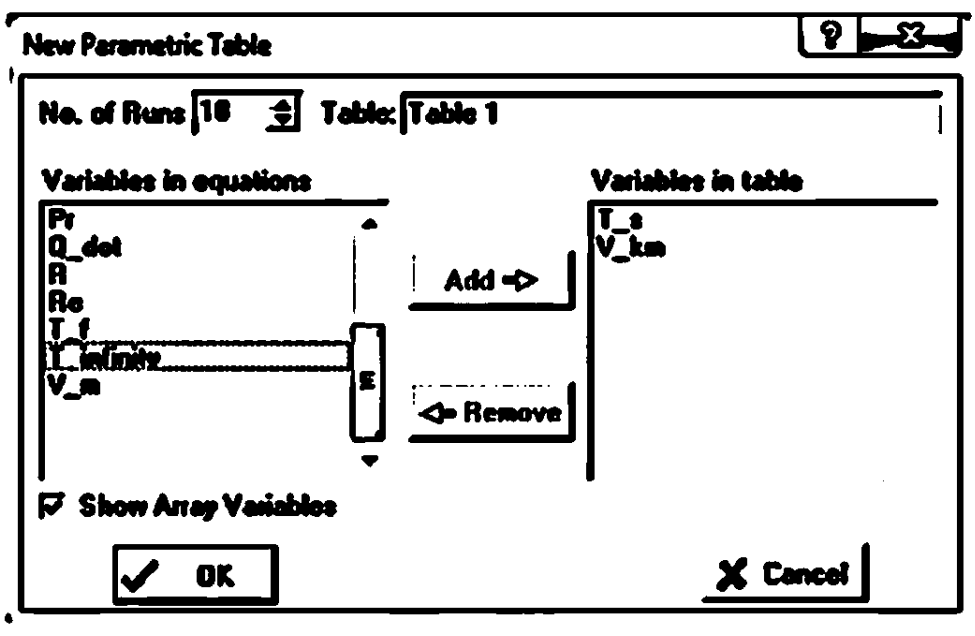
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
D=5[mm]*convert(mm,m)                          "Diameter of wire"
I=50[Amp]                                      "Electric Current"
R=0.002[ohm]                                   "Resistance of wire"
T_infinity=10[C]                               "Temperature of air"
"V_km is varied from 10[km/hr] to 80[km/hr]"   "Velocity of air"
L=1[m]

T_f=(T_s+T_infinity)/2
V_m=V_km*convert(km/hr,m/s)
k=Conductivity(Air,T=T_f)                      "Properties of air at T_f temperature"
nu=Viscosity(Air,T=T_f)/Density(Air,T=T_f,P=1[atm]*convert(atm,kPa))
Pr=Prandtl(Air,T=T_f)
Re=V_m*D/nu                                    "Reynolds number"
Nus_cyl=0.3+0.62*Re^0.5*Pr^(1/3)/(1+(0.4/Pr)^(2/3))^(1/4)*(1+(re/282000)^(5/8))^(4/5)
Nus_cyl=h*D/k
Q_dot=I^2*R
A_s=pi*D*L
Q_dot=h*A_s*(T_s-T_infinity)
```

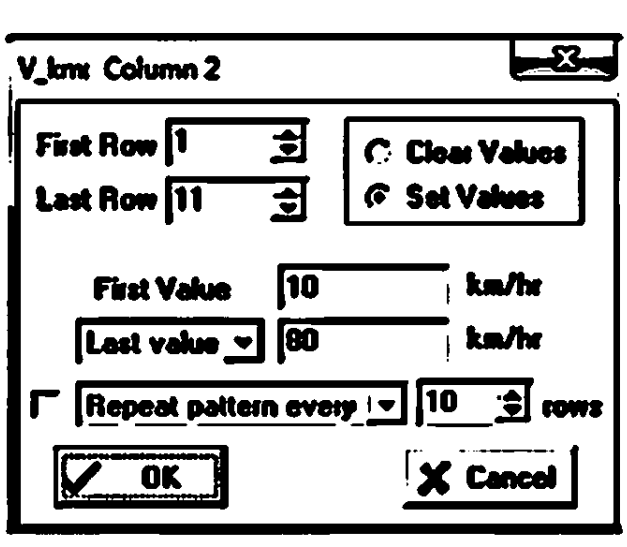
Step 2 of 5

Now to solve the equation for a range of velocity of air, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "T_s" and "V_km" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

The velocity of the air is varied from 10 km/hr to 80 km/hr
To enter the values in the "V_km" column, click on the drop tab in "V_km" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

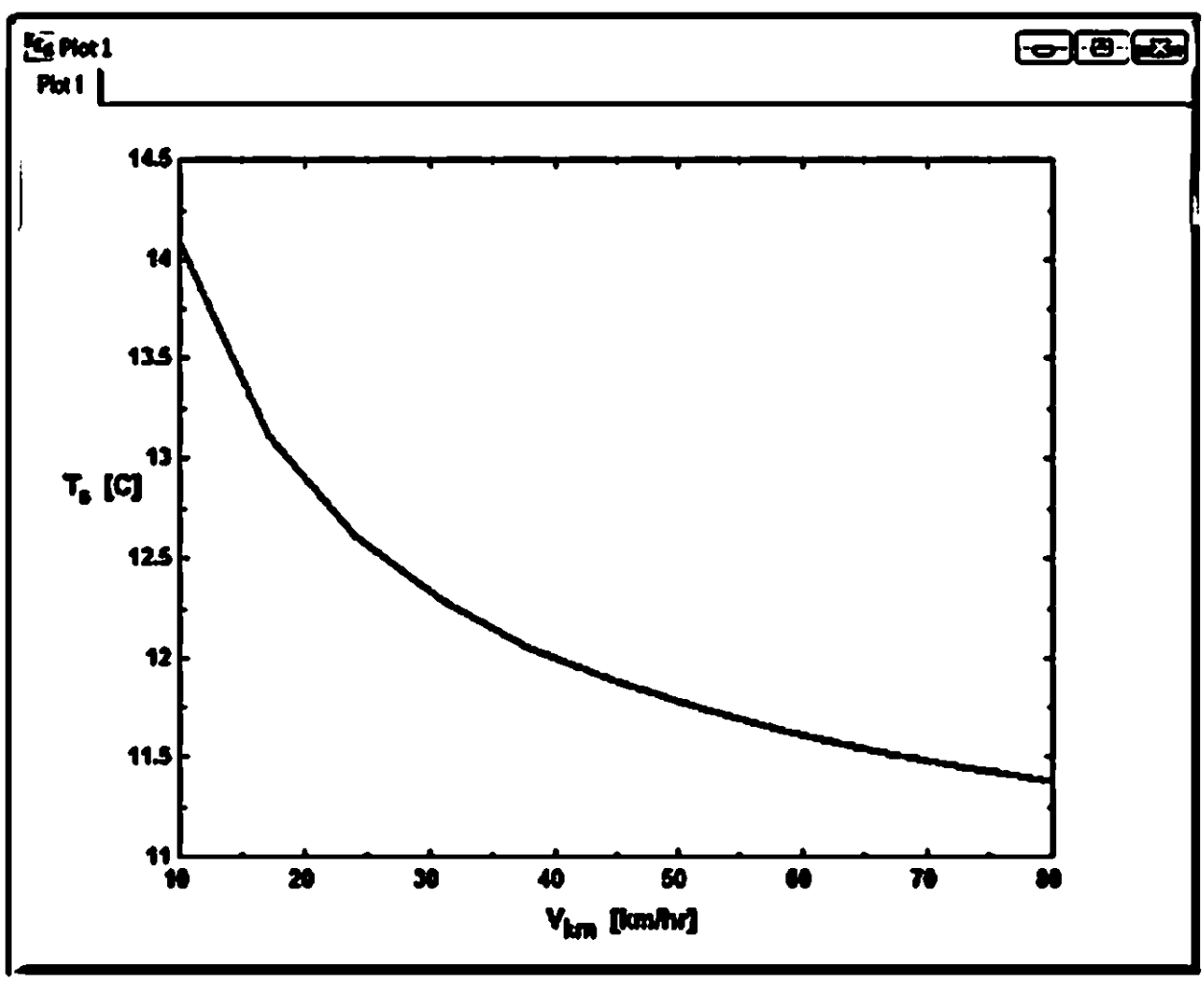
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	T_s [C]	V_{km} [km/hr]
Run 1	14.08	10
Run 2	13.12	17
Run 3	12.61	24
Run 4	12.28	31
Run 5	12.05	38
Run 6	11.88	45
Run 7	11.74	52
Run 8	11.63	59
Run 9	11.53	66
Run 10	11.45	73
Run 11	11.38	80

Step 5 of 5

To plot surface temperature of transmission line as function of velocity of air, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot surface temperature of transmission line as function of velocity of air is shown:



From the above plot we can interpret that as velocity of air increases the temperature of transmission line decreases.

Question: 19.053P

A heating system is to be designed to keep the wings of an aircraft cruising al a velocity of 900 km/h above freezing temperatures during flight at 12,200-m altitude where the standard atmospheric conditions are -55.4T and 18.8 kPa. Approximating the wing as a cylinder of elliptical cross section whose minor axis is 50 cm and disregarding radiation, determine the average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area.

Step 1 of 4

Take the freezing temperature is $T_s = 0°C$

Now calculate the average film temperature by using the expression,

$$T_{avg} = \frac{T_s + T_\infty}{2}$$

Here, T_∞ is atmospheric temperature

Substitute $-55.4°C$ for T_∞

$$T_{avg} = \frac{0-55.4}{2}$$
$$= -27.7°C$$

Obtain the properties of air at 1 atm and film temperature $T_{avg} = -27.7°C$ from "properties of air table"

Thermal conductivity

$$k = 0.02134 + (0.02211 - 0.02134)\left(\frac{-27.7-(-30)}{-20-(-30)}\right)$$
$$= 0.02152\ W/m°C$$

Kinematic viscosity

$$v = 1.087\times10^{-5} + (1.169\times10^{-5} - 1.087\times10^{-5})\left(\frac{-27.7-(-30)}{-20-(-30)}\right)$$
$$= 1.1058\times10^{-5}\ m^2/s$$

Prandtl number

$$Pr = 0.7425 + (0.7408 - 0.7425)\left(\frac{-27.7-(-30)}{-20-(-30)}\right)$$
$$= 0.7422$$

Step 2 of 4

Kinematic viscosity will be affected by change in atmospheric pressure. Convert atmospheric pressure in atm unit

$$P = (18.8\ kPa)\frac{1\ atm}{101.325\ kPa}$$
$$= 0.1855\ atm$$

Kinematic viscosity depends on pressure. Find the viscosity at this pressure by using the expression,

$$v = \frac{v_{@1atm}}{0.1855}$$
$$= \frac{(1.1058\times10^{-5})}{0.1855}$$
$$= 5.961\times10^{-5}\ m^2/s$$

Step 3 of 4

Find the Reynolds number by using the equation,

$$Re = \frac{VD}{v}$$

Here, V is velocity of air, D is diameter of the cylinder

Substitute $\left(900\ km/h = 900\times\frac{1000}{3600}\ m/s\right)$ for V and 0.5 m for D

$$Re = \frac{\left[900\times\frac{1000}{3600}\right](0.5)}{5.961\times10^{-5}}$$
$$= 2.096\times10^{6}$$

Assume the Nusselt number relation by using the below equation and find the Nusselt number,

$$Nu = 0.248\,Re^{0.612}\,Pr^{1/3}$$
$$= 0.248\times(2.096\times10^{6})^{0.612}\times0.7422^{1/3}$$
$$= 1659.54$$

Step 4 of 4

Now find the heat transfer coefficient for wing surface

$$Nu = \frac{hD}{k}$$

$$h = \frac{k}{D}Nu$$

$$h = \frac{0.02152}{0.5}\times1659.54$$
$$= 71.42\ W/m^2\cdot K$$

Now find the heat transfer rate per unit surface area by using the equation,

$$\dot{q} = h(T_s - T_\infty)$$
$$= 71.42\times(0-(-55.4))$$
$$= 3956.668\ W/m^2$$

Therefore, heat transfer per unit area is $\boxed{3956.668\ W/m^2}$

Question: 19.054P

A long aluminum wire of diameter 3 mm is extruded at a temperature of 280°C. The wire is subjected to cross air flow at 20°C at a velocity of 6 m/s. Determine the rate of heat transfer from the wire to the air per meter length when it is first exposed to the air.

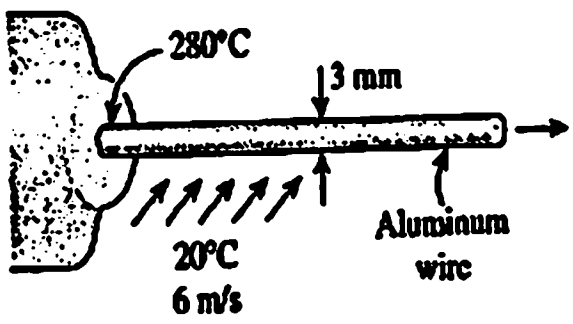


FIGURE P19-54

Step 1 of 4

Given data:

Diameter of aluminum wire, $D = 3\text{mm}$

Temperature of aluminum wire, $T_s = 280°\text{C}$

Temperature of air, $T_\infty = 20°\text{C}$

Velocity of air flow, $V = 6\ \text{m/s}$

Step 2 of 4

The film temperature,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(280+20)}{2}$$

$$= 150°\text{C}$$

From the table, properties of air at 1 atm pressure

At $T_f = 150°\text{C}$

Thermal conductivity,

$$k = \frac{0.03374+0.03511}{2}$$

$$= 0.034425\ \text{W/m}\cdot\text{K}$$

Kinematic viscosity,

$$\nu = \frac{(2.745+2.975)\times10^{-5}}{2}$$

$$= 2.86\times10^{-5}\ \text{m}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = \frac{0.7041+0.7014}{2}$$

$$= 0.70275$$

Step 3 of 4

The Reynolds number for the flow,

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{6\times\left(3\times10^{-3}\right)}{2.86\times10^{-5}}$$

$$= 629.37$$

Since the obtained Reynolds number is less than 2×10^5 ($\text{Re} < 2\times10^5$), the flow is said to be laminar.

The Nusselt number is determined from the relation given by

$$\text{Nu}_{cyl} = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+\left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(629.37)^{0.5}(0.70275)^{1/3}}{\left[1+\left(\frac{0.4}{0.70275}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{629.37}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 12.65$$

Step 4 of 4

The convective heat transfer coefficient,

$$\text{Nu}_{cyl} = \frac{hD}{k}$$

Rewrite and solve for h:

$$h = \frac{\text{Nu}_{cyl}\times k}{D}$$

$$= \frac{12.65\times0.034425}{\left(3\times10^{-3}\right)}$$

$$= 145.16\ \text{W/m}^2\cdot\text{K}$$

The rate of heat transfer from the wire to the air per meter length is determined from the equation given by

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= h\times(\pi DL)\times(T_s - T_\infty)$$

$$= 145.16\times\left(\pi\times3\times10^{-3}\times1\right)\times(280-20)$$

$$= 355.7\ \text{W/m}$$

Therefore the rate of heat transfer from the wire to the air per meter length,

$$\boxed{\dot{Q} = 355.7\ \text{W/m}}$$

Question: 19.055P

An incandescent lightbulb is an inexpensive but 1 highly inefficient device that converts electrical energy into light. It converts about 10 percent of the electrical energy it consumes into light while converting the remaining 90 percent into heat. (A fluorescent lightbulb will give the same amount of light while consuming only one-fourth of the electrical energy, and it will last 10 times longer than an incandescent lightbulb.) The glass bulb of the lamp heats up very quickly as a result of absorbing all that heat and dissipating it to the surroundings by convection and radiation.

Consider a 10-cm-diameter 100-W lightbulb cooled by a fan that blows air at 30°C to the bulb at a velocity of 2 m/s. The surrounding surfaces are also at 30°C, and the emissivity of the glass is 0.9. Assuming 10 percent of the energy passes through the glass bulb as light with negligible absorption and the rest of the energy is absorbed and dissipated by the bulb itself, determine the equilibrium temperature of the glass bulb,

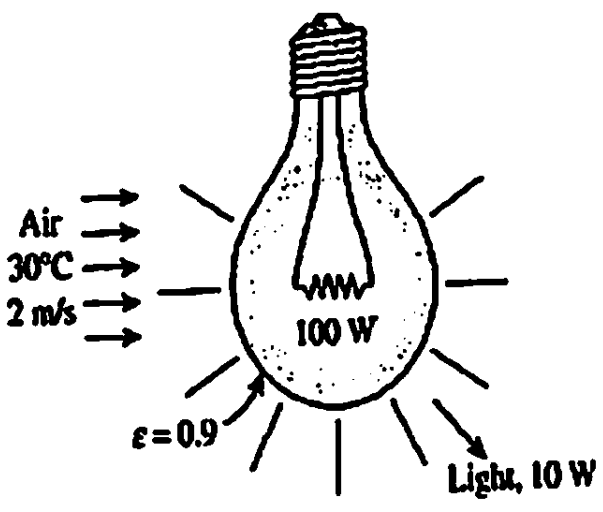


FIGURE P19-55

Step 1 of 5

Obtain the Properties of air at 1 atm. and 30C from the table A-22 "ideal gas properties of air at 1 atm. Pressure".

Thermal conductivity, $k = 0.02588\ \text{W/m.°C}$

Kinematic viscosity, $\upsilon = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Dynamic viscosity at ambient air temperature, $\mu_\infty = 1.872\times10^{-5}\ \text{kg/m.s}$

Prandtil number, $\text{Pr} = 0.7282$

Step 2 of 5

Calculate the Reynolds number.

$$\text{Re} = \frac{VD}{\upsilon}$$

Here, D is the diameter of the bulb.

Substitute $2\ \text{m/s}$ for V, $1.608\times10^{-5}\ \text{m}^2/\text{s}$ for v and 0.1 m for D.

$$\text{Re} = \frac{(2\ \text{m/s})(0.1\ \text{m})}{1.608\times10^{-5}\ \text{m}^2/\text{s}}$$
$$= 1.244\times10^4$$

Step 3 of 5

Calculate the Nusselt number using the relation,

$$Nu = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Here, μ_s is the dynamic viscosity of air at the surface temperature of the bulb.

As the surface temperature of the bulb is not known, assume a surface temperature of $100°\text{C}$ to proceed with the calculations.

Obtain the Properties of air at 1 atm. and 30C from the table A-22 "ideal gas properties of air at 1 atm. Pressure".

Dynamic viscosity at surface temperature, $\mu_{s,\text{at }100°\text{C}} = 2.181\times10^{-5}\ \text{kg/m}\cdot\text{s}$

Substitute 1.244×10^4 for Re, 0.7282 for Pr, $1.872\times10^{-5}\ \text{kg/m}\cdot\text{s}$ for μ_∞ and $2.181\times10^{-5}\ \text{kg/m}\cdot\text{s}$ for $\mu_{s,\text{at }100°\text{C}}$.

$$Nu = 2 + \left[0.4(1.244\times10^4)^{0.5} + 0.06(1.244\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.181\times10^{-5}}\right)^{1/4}$$

$$Nu = 67.14$$

Step 4 of 5

Calculate the heat transfer coefficient using the relation,

$$Nu = \frac{hD}{k}$$
$$h = \frac{k}{D}Nu$$

Substitute $0.02588\ \text{W/m.°C}$ for k, 0.1 m for D and 67.14 for Nu.

$$h = \frac{0.02588\ \text{W/m.°C}}{0.1\ \text{m}}(67.14)$$
$$= 17.37\ \text{W/m}^2\cdot\text{°C}$$

Calculate the amount of heat generated from the bulb.

$$\dot{Q}_{lost} = 0.9\times100\ \text{W}$$
$$= 90\ \text{W}$$

Calculate the surface area of the bulb.

$$A_s = \pi D^2$$

Substitute 0.1 m for D.

$$A_s = \pi D^2$$
$$= \pi(0.1\ \text{m})^2$$
$$= 0.0314\ \text{m}^2$$

Step 5 of 5

Calculate the surface temperature by applying heat balance to the bulb.

$$\dot{Q}_{lost} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$\dot{Q}_{lost} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$

Here, T_s is surface temperature of bulb, T_∞ is ambient temperature, ε is the emissivity and σ is the Stefan-Boltzmann constant.

Substitute 90 W for $\dot{Q}_{lost}$, $17.37\ \text{W/m}^2\cdot\text{°C}$ for h, 0.9 for ε and $5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ for σ, and $30°\text{C}$ for T_s.

$$90 = (17.37\times0.0314)\left[T_s - (30+273)\right] + \left(0.9\times0.0314\times5.67\times10^{-8}\right)\left[T_s^4 - (30+273)^4\right]$$
$$90 = (0.54542)\left[T_s - 303\right] + \left(1.6023\times10^{-9}\right)\left[T_s^4 - (303)^4\right]$$
$$T_s = 409.9\ \text{K}$$
$$T_s = 409.9 - 273$$
$$T_s = 136.9°\text{C}$$

Therefore, the equilibrium temperature of the glass bulb is $\boxed{T_s = 136.9°\text{C}}$.

Question: 19.056P

A 12-ft-long, 1.5-kW electrical resistance wire is made of 0.1-in-diameter stainless steel (k = 8.7 Btu/h-ft-°F). The resistance wire operates in an environment at 85°F. Determine the surface temperature of the wire if it is cooled by a fan blowing air at a velocity of 20 ft/s.

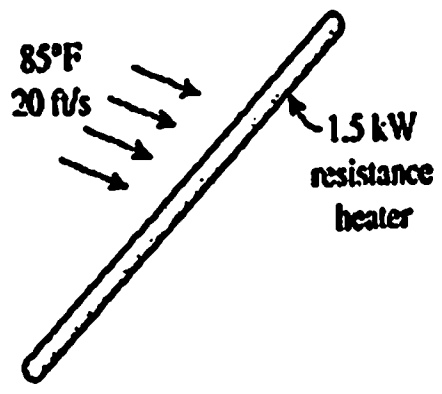


FIGURE P19-56

Step 1 of 2

Given

Length of wire (L) = 12 ft

Diameter of wire (D) = 0.1 in

$= 8.33\times10^{-3}$ ft

Electric resistance of wire = 1.5 kW

Thermal conductivity of stainless steel (k) = 8.7 Btu/h.ft.°F

Temperature of air = 85°F

Velocity of air (V) = 20 ft/s

Step 2 of 2

Properties of air from ideal gas properties of air table at 1 atm and 200°F are

$k = 0.01761$ Btu/h.ft.°F

$\upsilon = 0.2406\times10^{-3}$ ft^2/s

Pr = 0.7124

Reynolds number, $\mathrm{Re} = \dfrac{VD}{\upsilon}$

$$\mathrm{Re} = \frac{(20\text{ ft/s})(0.1/12\text{ ft})}{0.2406\times10^{-3}\text{ ft}^2/\text{s}}$$

$$\mathrm{Re} = 692.8$$

Nusselt number, $Nu = \dfrac{hD}{k}$

$$Nu = 0.3+\frac{0.62\mathrm{Re}^{0.5}\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$Nu = 0.3+\frac{0.62(692.8)^{0.5}(0.7124)^{1/3}}{\left[1+(0.4/0.7124)^{2/3}\right]^{1/4}}\left[1+\left(\frac{692.8}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$Nu = 13.34$$

Coefficient of heat transfer, $h = \dfrac{k}{D}Nu$

$$h = 28.19\text{ Btu/h.ft}^2\text{.°F}$$

$$h = \frac{0.01761\text{ Btu/h.ft.°F}}{(0.1/12\text{ ft})}(13.34)$$

Outer surface temperature of wire is calculated using $T_s = T_\infty + \dfrac{\dot{Q}}{hA}$

Where $A_s = \pi DL$

$$= \pi\times8.33\times10^{-3}\times12$$

$$= 0.3142\text{ ft}^2$$

$$T_s = T_\infty + \frac{\dot{Q}}{hA}$$

$$T_s = 85°\text{F} + \frac{(1500\times3.41214)\text{ Btu/h}}{(28.19\text{ Btu/h.ft}^2\text{.°F})(0.3142\text{ ft}^2)}$$

$$T_s = 662.9°\text{F}$$

Hence surface temperature of the wire is, $\boxed{T_s = 662.9°\text{F}}$

Question: 19.057P

The components of an electronic system arc located in a 1.5-m-long horizontal duct whose cross section is 20 cm × 20 cm. The components in the duct are not allowed to come into direct contact with cooling air, and thus are cooled by air at 30°C flowing over the duct with a velocity of 200 m/min. If the surface temperature of the duct is not to exceed 65°C, determine the total power rating of the electronic devices that can be mounted into the duct.

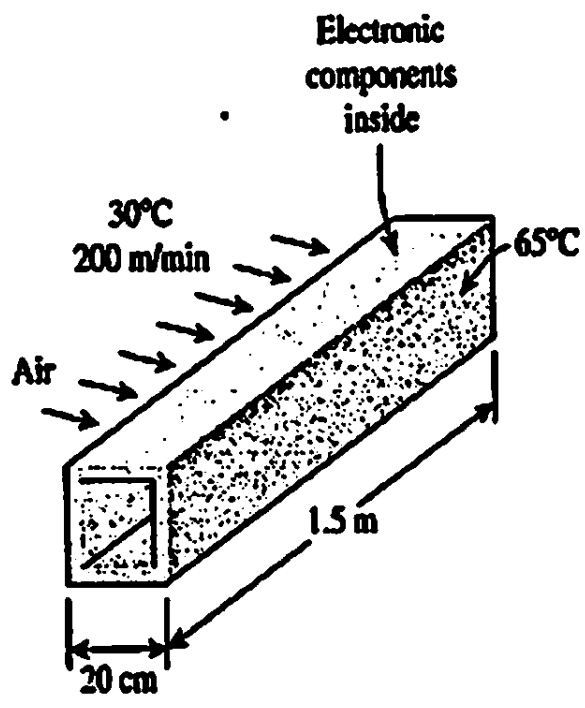


FIGURE P19-57

Step 1 of 9

Calculate the average film mean temperature by using the equation:

$$T_{avg} = \frac{T_\infty + T_s}{2}$$

Here, T_{avg} is the average film mean temperature, T_∞ is the temperature of air and T_s is the surface temperature of the duct.

Substitute $30°C$ for T_∞ and $65°C$ for T_s .

$$T_{avg} = \frac{30+65}{2}$$
$$= 47.5°C$$

Step 2 of 9

Obtain the properties of air at a temperature of $47.5°C$, from the "Properties of air at 1 atm pressure" table.

$$k = 0.02717 \text{ W/m.°C}$$

$$\upsilon = 1.774 \times 10^{-5} \text{ m}^2/\text{s}$$

$$\text{Pr} = 0.7235$$

Here, k is the thermal conductivity, υ is the kinematic viscosity and Pr is the Prandtl number.

Step 3 of 9

Calculate the Reynolds number for the given flow by using the equation:

$$\text{Re} = \frac{VD}{\upsilon}$$

Here, Re is the Reynolds number for the given flow, V is the velocity of flow and D is the length of the side of the duct.

Substitute 60 m/min for V , 0.2 m for D and $1.774 \times 10^{-5} \text{ m}^2/\text{s}$.

$$\text{Re} = \frac{200 \times 0.2}{60 \times 1.774 \times 10^{-5}}$$
$$= 3.578 \times 10^4$$

Step 4 of 9

Here, the Reynold's number is in between 5000 and 10000.So, obtain the equation for the Nusselt number for the flow of air over a square cross sectioned duct from the "Emperical correlations for the average Nusselt number for the forced convection over circular and noncircular cylinders in cross flow" table.

$$Nu = 0.102 \text{Re}^{0.675} \text{Pr}^{1/3}$$

Here, Nu is the Nusselt number for the flow of air.

Step 5 of 9

Calculate the Nusselt number for the flow by using the equation:

$$Nu = 0.102 \text{Re}^{0.675} \text{Pr}^{1/3}$$

Substitute $\left(3.758 \times 10^4\right)$ for Re and 0.7235 for Pr .

$$Nu = 0.102 \times (3.758 \times 10^4)^{0.675} \times (0.7235)^{1/3}$$
$$= 112.2$$

Step 6 of 9

Calculate the heat transfer coefficient by using the equation:

$$h = \frac{k}{D} Nu$$

Here, h is the heat transfer coefficient.

Substitute $0.0217 \text{ W/m} \cdot °C$ for k , 0.2 m for D and 112.2 for Nu .

$$h = \frac{0.02717 \times 112.2}{0.2}$$
$$= 15.24 \text{ W/m}^2 \cdot \text{C}$$

Step 7 of 9

Calculate the surface area of the duct by using the equation:

$$A_s = 4DL$$

Here, A_s is the surface area of the duct and L is the length of the duct.

Substitute 0.2 m for D and 1.5 m for L .

$$A_s = (4 \times 0.2 \times 1.5)$$
$$= 1.2 \text{ m}^2$$

Step 8 of 9

Calculate the rate of heat transfer by using the equation:

$$\dot{Q} = hA_s(T_s - T_\infty)$$

Here, $\dot{Q}$ is the rate of heat transfer.

Substitute $15.24 \text{ W/m}^2 \cdot °C$ for h , 1.2 m^2 for A_s , $65°C$ for T_s and $30°C$ for T_∞ .

$$\dot{Q} = 15.24 \times 1.2 \times (65 - 30)$$
$$= 640.08 \text{ W}$$
$$\simeq 640 \text{ W}$$

Therefore, the total power rating of the electronic device is $\boxed{\dot{Q} = 640 \text{ W}}$.

Step 9 of 9

Question: 19.058P

Repeat Prob. 19-57 for a location at 3000-m altitude where the atmospheric pressure is 70.12 kPa.

Step 1 of 5

Length of duct $L = 1.5\,\text{m}$

Cross section of duct $A_c = 20\,\text{cm} \times 20\,\text{cm}$

Temperature of air $T = 30°\text{C}$

Velocity of air flowing over the duct $V = 200\,\text{m/min}$

Surface temperature of duct $T_s = 65°\text{C}$

Atmospheric pressure at location 3000 m altitude is $p_{3000\text{m}} = 70.12\,\text{kPa}$

Step 2 of 5

Average film temperature

$$T_{avg} = \frac{T_s + T}{2}$$
$$= \frac{65+30}{2}$$
$$= 47.5°\text{C}$$

Properties of air from ideal gas properties of air table at 1atm and $T_{avg} = 47.5°\text{C}$

$$k = 0.02699 + (0.02735 - 0.02699)\left(\frac{47.5-45}{50-45}\right)$$
$$= 0.02717\ \text{W/m.°C}$$
$$\nu = 1.750\times10^{-5} + (1.798\times10^{-5} - 1.750\times10^{-5})\left(\frac{47.5-45}{50-45}\right)$$
$$= 1.774\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7241 + (0.7228 - 0.7241)\left(\frac{47.5-45}{50-45}\right)$$
$$= 0.72345$$

There is a change in kinematic viscosity of air if atmospheric pressure changes.

Therefore, kinematic viscosity of air at $p_{3000\text{m}} = 70.12\,\text{kPa}$

$$\nu_{@70.12\,\text{kPa}} = \left(\frac{101.325}{70.12}\right)\times1.774\times10^{-5}$$
$$= 2.5634\times10^{-5}\ \text{m}^2/\text{s}$$

Step 3 of 5

Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{\frac{200}{60}\times0.2}{2.5634\times10^{-5}}$$
$$= 2.6007\times10^{4}$$

Nusselt number can be calculated by using the equation,

$$\text{Nu} = \frac{hD}{k}$$
$$= 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3}$$
$$= 0.102\left(2.6007\times10^{4}\right)^{0.675}(0.72345)^{1/3}$$
$$= 87.482$$

Step 4 of 5

Heat transfer coefficient can be calculated as,

$$\text{Nu} = \frac{hD}{k}$$
$$h = \frac{k}{D}\text{Nu}$$
$$= \frac{0.02717}{0.2}\times87.482$$
$$= 11.884\,\text{W/m}^2\cdot°\text{C}$$

Step 5 of 5

Rate of heat transfer

$$\dot{Q} = hA_s(T_s - T_\infty)$$

Where surface area is

$$A_s = 4sL$$
$$= 4\times0.2\times1.5$$
$$= 1.2\,\text{m}^2$$

Hence,

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= 11.884\times1.2\times(65-30)$$
$$= 499.128\,\text{W}$$

The total power rating of the electronic device is $\boxed{\dot{Q} = 499.128\,\text{W}}$

Question: 19.059P pg 1

A 0.4-W cylindrical electronic component with diameter 0.3 cm and length 1.8 cm and mounted on a circuit board is cooled by air flowing across it at a velocity of 240 m/min. If the air temperature is 35°C, determine the surface temperature of the component.

Step 1 of 4

Given
Diameter of electronic component (D) = 0.3 cm
Length of electronic component (L) = 1.8 cm
Velocity of air (V) = 240 m/min
Temperature of air = 35°C

Step 2 of 4

Properties of air from ideal gas properties of air table at 1 atm and 50°C are

$k = 0.02735$ W/m.°C

$\upsilon = 1.798\times10^{-5}$ m^2/s

Pr = 0.7228

Reynolds number, $\mathrm{Re} = \dfrac{VD}{\upsilon}$

$$\mathrm{Re} = \frac{(240/60\ \text{m/s})(0.003\ \text{m})}{1.798\times10^{-5}\ \text{m}^2/\text{s}}$$

$$\mathrm{Re} = 667.4$$

Nusselt number, $Nu = \dfrac{hD}{k}$

$$Nu = 0.3+\frac{0.62\mathrm{Re}^{0.5}\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$Nu = 0.3+\frac{0.62(667.4)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{667.4}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$Nu = 13.17$$

Step 3 of 4

Convective heat transfer coefficient is $h = \dfrac{k}{D}Nu$

$$h = \frac{0.02735}{0.003}(13.17)$$

$$h = 120.0\ \text{W/m}^2.°\text{C}$$

Temperature of surface is calculated using, $T_s = T_\infty + \dfrac{\dot{Q}}{hA}$

Where, $A_s = \pi DL$

$$A_s = \pi\times0.003\times0.018$$

$$A_s = 0.0001696\ \text{m}^2$$

Hence, $T_s = 35+\dfrac{0.4}{120.0\times0.0001696}$

$$T_s = 54.6°\text{C}$$

Surface temperature of the component is $\boxed{T_s = 54.6°\text{C}}$

Step 4 of 4

Question: 19.060P

Consider a 50-cm-diameter and 95-cm-long hot water tank. The tank is placed on the roof of a house. The water inside the tank is heated to 80°C by a flat-plate solar collector during the day. The tank is then exposed to windy air at 18°C with an average velocity of 40 km/h during the night. Estimate the temperature of the tank after a 45-min period. Assume the tank surface to be at the same temperature as the water inside, and the heat transfer coefficient on the top and bottom surfaces to be the same as that on the side surface.

Step 1 of 5

Given
Diameter of water tank (D) = 50 cm
Length of tank (L) = 90 cm
Temperature of water = 80°C
Temperature of air = 18°C
Velocity of air (V) = 40 km/h

Step 2 of 5

Properties of water at 80°C from properties of water table

$\rho = 971.8\ \text{kg/m}^3$

$C_p = 4.197\ \text{kJ/kg.°C}$

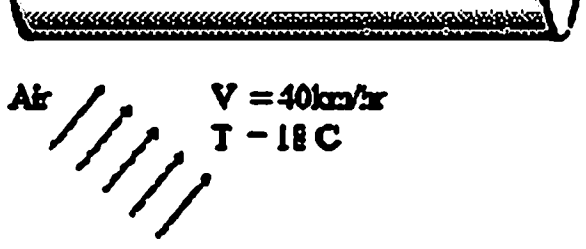
Properties of air from properties of air table at 1 atm and 50°C,

$k = 0.02735\ \text{W/m.°C}$

$\upsilon = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

$\text{Pr} = 0.7228$

Step 3 of 5



Step 4 of 5

Reynolds number, $\text{Re} = \dfrac{VD}{\upsilon}$

$$\text{Re} = \frac{\left(\frac{40\times1000}{3600}\ \text{m/s}\right)(0.50\ \text{m})}{1.798\times10^{-5}\ \text{m}^2/\text{s}}$$

$$\text{Re} = 309{,}015$$

Nusselt number, $\text{Nu} = 0.3+\dfrac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\dfrac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$

$$\text{Nu} = 0.3+\frac{0.62(309{,}015)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{309{,}015}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$\text{Nu} = 484.9$$

Coefficient of heat transfer, $h = \dfrac{k}{D}\text{Nu}$

$$h = \frac{0.02735\ \text{W/m.°C}}{0.50\ \text{m}}(484.9)$$

$$h = 26.53\ \text{W/m}^2\text{.°C}$$

Step 5 of 5

Mass of water in the tank is $m = \rho V$

$$m = \rho\pi\frac{D^2}{4}L$$

$$m = (971.8\ \text{kg/m}^3)\pi\frac{(0.50\ \text{m})^2}{4}(0.95\ \text{m})$$

$$m = 181.27\ \text{kg}$$

Heat rejected by water is $Q = mC_p(T_2 - T_1)$

$$Q = (181.27\,\text{kg})(4.197\ \text{kJ/kg°C})(80 - T_2)\text{°C}$$

$$Q = 760.798\times(80 - T_2)\ \text{kJ}$$

Average heat transfer rate is $\dot{Q} = \dfrac{Q}{\Delta t}$

$$\dot{Q} = \frac{760.798\times(80-T_2)}{45\times60}\ \text{kJ/s}$$

$$\dot{Q} = 0.28178\times(80-T_2)\ \text{kW}$$

Surface area, $A_s = \pi DL + 2\pi\dfrac{D^2}{4}$

$$A_s = \pi(0.5)(0.95) + 2\pi(0.5)^2/4$$

$$A_s = 1.885\ \text{m}^2$$

Heat transfer rate, $\dot{Q} = hA_s(T_s - T_\infty)$

$$\dot{Q} = (26.53\ \text{W/m}^2\text{.°C})(1.885\ \text{m}^2)\left(\frac{80+T_2}{2} - 18\right)\text{°C}$$

$$\dot{Q} = 0.05\times\left(\frac{80+T_2}{2} - 18\right)\ \text{kW}$$

By equating both the values of $\dot{Q}$, we get

$$0.28178\times(80 - T_2) = 0.05\times\left(\frac{80+T_2}{2} - 18\right)$$

$$22.5424 - (0.28178\times T_2) = 2 + (0.025\times T_2) - 0.9$$

$$21.4424 = 0.30678\times T_2$$

$$T_2 = 69.89\ ^0\text{C}$$

Final temperature of tank, $\boxed{T_2 = 69.89\ ^0\text{C}}$

Question: 19.061P pg 1

Reconsider Prob. 19-60. Using EES (or other) software, plot the temperature of the tank as a function of the cooling time as the time varies from 30 min to 5 h, and discuss the results.

Step 1 of 7

To solve this question using EES software open EES software
Then FILE->New
A dailogbox will open type the following in dailog box

```
"GIVEN"
D=0.50 "[m]"
L=0.95 "[m]"
T_w1=80 "[C]"
T_infinity=18 "[C]"
Vel=40 "[km/h]"
"time=45 [min], parameter to be varied"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_w_ave+T_infinity)
rho_w=Density(water, T=T_w_ave, P=101.3)
C_p_w=CP(Water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C)
T_w_ave=1/2*(T_w1+T_w2)
```

Step 2 of 7

```
"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*D)/nu
Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8)
)^(4/5)
h=k/D*Nusselt
A=pi*D*L+2*pi*D^2/4
Q_dot=h*A*(T_w_ave-T_infinity)
m_w=rho_w*V_w
V_w=pi*D^2/4*L
Q=m_w*C_p_w*(T_w1-T_w2)
Q_dot=Q/(time*Convert(min, s))
```

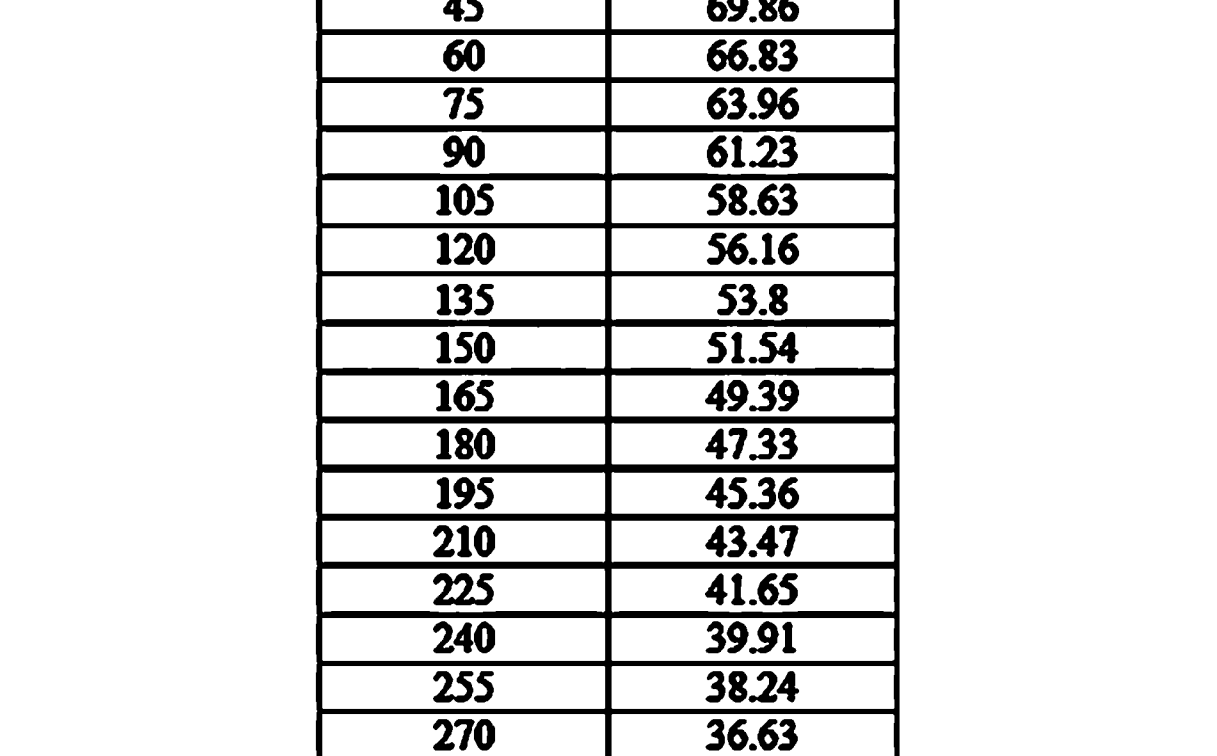
Step 3 of 7

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 19
Select variables in equations as time and T_{w2} press ADD and then press ok
A table will appear put values of time as 30,45,60....300 in run 1to 19
Then from Calculate -> Solve table
Solution will appear

time [min]	T_{w2} [C]
30	73.06
45	69.86
60	66.83
75	63.96
90	61.23
105	58.63
120	56.16
135	53.8
150	51.54
165	49.39
180	47.33
195	45.36
210	43.47
225	41.65
240	39.91
255	38.24
270	36.63
285	35.09
300	33.6

Step 4 of 7

Then from Plot -> New plot window -> X-Y Plot
Select X-axis time and Y-axis T_{w2}
Press OK
Graph will be plotted.



Step 5 of 7

Now change the code in the dialog box as

```
"GIVEN"
```

Step 5 of 7

Now change the code in the dialog box as

```
"GIVEN"
Vel=70 "[km/h], parameter to be varied"
w=2.8 "[m]"
L=8 "[m]"
"q_dot_rad=200 [W/m^2], parameter to be varied"
T_infinity=30 "[C]"
"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)
```

Step 6 of 7

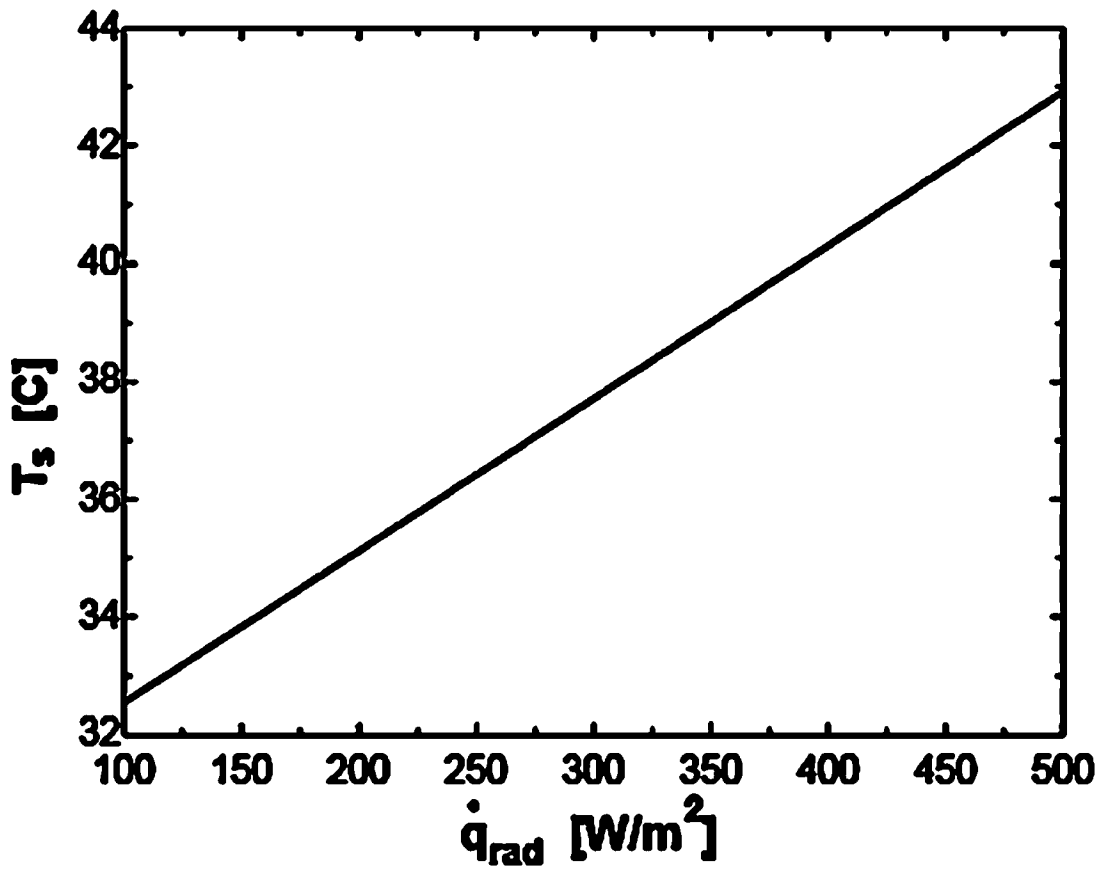
```
"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*L)/nu
"Reynolds number is greater than the critical Reynolds number. We use
combined laminar and turbulent flow relation for Nusselt number"
Nusselt=(0.037*Re^0.8-871)*Pr^(1/3)
h=k/L*Nusselt
q_dot_conv=h*(T_s-T_infinity)
q_dot_conv=q_dot_rad
```

Then from Tables -> New parametric table
A dailog box will open
Select no of runs = 17
Select variables in equations as Q_{rad} and T_s press ADD and then press ok
A table will appear put values of Q_{rad} as 100,125,150....500 in run 1to 17
Then from Calculate -> Solve table
Solution will appear

Q_{rad} [W/m^2]	T_s [C]
100	32.56
125	33.2
150	33.84
175	34.48
200	35.13
225	35.77
250	36.42
275	37.07
300	37.71
325	38.36
350	39.01
375	39.66
400	40.31
425	40.97
450	41.62
475	42.27
500	42.93

Step 7 of 7

Then from Plot -> New plot window -> X-Y Plot
Select X-axis Q_{rad} and Y-axis T_s
Press OK
Graph will be plotted.



Question: 19.062P

A 2.2-m-diameter spherical tank of negligible thickness contains iced water at 0°C. Air at 25°C flows over the tank with a velocity of 6 m/s. Determine the rate of heat transfer to the tank and the rate at which ice melts. The heat of fusion of water at 0°C is 333.7 U/kg.

Step 1 of 5

Given data:

Diameter of spherical tank, $D = 2.2$ m

Temperature of the air, $T_\infty = 25°C$

Air flow velocity, $V = 6$ m/s

Temperature of the surface of the tank, $T_s = 0°C$

Heat of fusion of water, $h_{fg} = 337.7$ kJ/kg

Step 2 of 5

Loading solution. Please wait ...

Step 3 of 5

The Nusselt number,

$$Nu_{sph} = 2+\left[0.4R_e^{1/2}+0.06R_e^{2/3}\right]P_r^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2+\left[0.4(845256.9)^{0.5}+0.06(845256.9)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4}$$

$$= 810.5$$

Step 4 of 5

The convective heat transfer,

$$Nu_{sph} = \frac{hD}{k}$$

$$h = \frac{Nu_{sph}\times k}{D}$$

$$= \frac{810.5\times0.02551}{2.2}$$

$$= 9.4\ W/m^2\cdot K$$

The rate of heat transfer to the tank,

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= 9.4\times\left(\pi\times2.2^2\right)\times(25-0)$$

$$= 3572.54\ W$$

Therefore the rate of heat transfer to the tank, $\boxed{\dot{Q} = 3572.54\ W}$

Step 5 of 5

The rate at which ice melts,

$$\dot{m} = \frac{\dot{Q}}{h_{fg}}$$

$$= \frac{3572.54}{333.7\times10^3}$$

$$= 0.0107\ kg/s$$

Therefore the rate at which ice melts, $\boxed{\dot{m} = 0.0107\ kg/s}$

Question: 19.063P

During a plant visit, it was noticed that a 12-m-long section of a 12-cm-diameter steam pipe is completely exposed to the ambient air. The temperature measurements indicate that the average temperature of the outer surface of the steam pipe is 75°C when the ambient temperature is 5°C. There are also light winds in the area at 25 km/h. The emissivity of the outer surface of the pipe is 0.8, and the average temperature of the surfaces surrounding the pipe, including the sky, is estimated to be 0°C. Determine the amount of heat lost from the steam during a 10-h-long work day.

Steam is supplied by a gas-fired steam generator that has an efficiency of 80 percent, and the plant pays $1.05/therm of natural gas. If the pipe is insulated and 90 percent of the heat loss is saved, determine the amount of money this facility will save a year as a result of insulating the steam pipes. Assume the plant operates every day of the year for 10 h. State your assumptions.

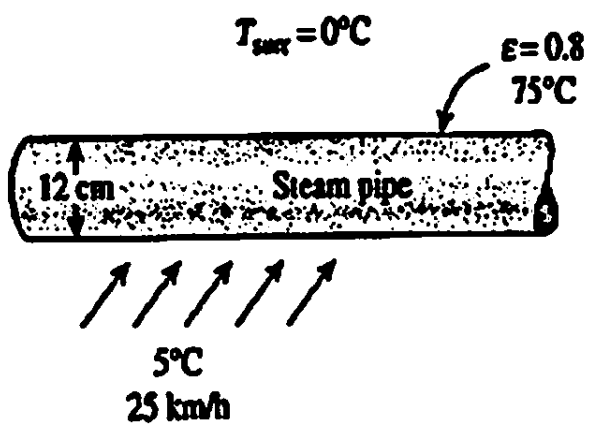


FIGURE P19-63

Step 1 of 8

Given data:

Diameter of the steam pipe, $D=12$ cm

Length of the steam pipe, $L=12$ m

Outer surface temperature of the pipe, $T_s=75°C$

Ambient air temperature, $T_\infty=5°C$

Velocity of air, $V=25$ km/h $=6.94$ m/s

Emissivity of the pipe, $\varepsilon=0.8$

Temperature of the surroundings, $T_{surr}=0°C$

Efficiency of the steam generator, $\eta_g=0.8$

Step 2 of 8

The film temperature of the air is determined as

$$T_f=\frac{(T_s+T_\infty)}{2}$$
$$=\frac{(75+5)}{2}$$
$$=40°C$$

From the table, properties of air at 1 atm pressure and $T=40°C$ we get

Thermal conductivity, $k=0.02662$ W/m·K

Kinematic viscosity, $\nu=1.702\times10^{-5}$ m^2/s

Prandtl number, $\Pr=0.7255$

The Reynolds number for the flow is determined from the equation given by

$$\mathrm{Re}=\frac{VD}{\nu}$$
$$=\frac{6.94\times\left(12\times10^{-2}\right)}{1.702\times10^{-5}}$$
$$=48930.67$$

Since the obtained Reynolds number is less than 2×10^5, the flow is said to be laminar.

Step 3 of 8

The average Nusselt number for cross flow over cylinder is determined from the Churchill & Bernstein relation given by

$$\mathrm{Nu}_{cyl}=0.3+\frac{0.62\mathrm{R}_e^{1/2}\mathrm{P}_r^{1/3}}{\left[1+\left(\frac{0.4}{\mathrm{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$=0.3+\frac{0.62(48930.67)^{0.5}(0.7255)^{1/3}}{\left[1+\left(\frac{0.4}{0.7255}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{48930.67}{282000}\right)^{5/8}\right]^{4/5}$$

$$=136.82$$

Step 4 of 8

The convective heat transfer coefficient of air is determined from the equation given by

$$\mathrm{Nu}_{cyl}=\frac{hD}{k}$$

Rewrite and solve for h:

$$h=\frac{\mathrm{Nu}_{cyl}\times k}{D}$$
$$=\frac{136.82\times0.02662}{\left(12\times10^{-2}\right)}$$
$$=30.35\ \mathrm{W/m^2\cdot K}$$

Step 5 of 8

The rate of heat loss from the pipe is determined from the equation given by

$$\dot{Q}=\dot{Q}_{conv}+\dot{Q}_{rad}$$
$$=hA_s(T_s-T_\infty)+\varepsilon A_s\sigma\left(T_s^4-T_{surr}^4\right)$$
$$=\left[h\times(\pi DL)\times(T_s-T_\infty)\right]+\left[\varepsilon\times(\pi DL)\times\sigma\times\left(T_s^4-T_{surr}^4\right)\right]$$
$$=\left[30.35\times\left(\pi\times12\times10^{-2}\times12\right)\times(75-5)\right]$$
$$+\left[0.8\times\left(\pi\times12\times10^{-2}\times12\right)\times5.67\times10^{-8}\times\left((75+273)^4-(0+273)^4\right)\right]$$
$$=11480.748\ \mathrm{W}$$

Step 6 of 8

The amount of heat lost from the steam is determined as

$$Q=\dot{Q}\times10\ \mathrm{hr}$$
$$=11480.748\times10\times3600$$
$$=413306928\ \mathrm{J}$$
$$=4.13\times10^5\ \mathrm{kJ}$$

The amount of heat lost from the steam during a 10-h-long work day is $\boxed{Q=4.13\times10^5\ \mathrm{kJ}}$

Step 7 of 8

The amount of heat lost per year is determined as

$$Q_y=4.13\times10^5\times365$$
$$=1.51\times10^8\ \mathrm{kJ/year}$$

The amount of heat lost per year considering the generator efficiency is determined as

$$Q_g=\frac{Q_y}{\eta_g}$$
$$=\frac{1.51\times10^8}{0.8}$$
$$=1.89\times10^8\ \mathrm{kJ/year}$$
$$=\frac{1.8857\times10^8}{105500}$$
$$=1787.31\ \mathrm{therms/year}$$

Step 8 of 8

Given that if the pipe is insulated 90% of heat lost is saved

The amount of heat lost saved is determined as

$$Q_{saved}=0.9\times Q_g$$
$$=0.9\times1787.31$$
$$=1608.58\ \mathrm{therms/year}$$

The amount of money saved by insulating the steam pipe is determined as

$$\$_{saved}=Q_{saved}\times1.05$$
$$=1608.58\times1.05$$
$$=1689\$$$

The amount of money saved per year by insulating the steam pipe is $\boxed{\$_{saved}=1689\$}$

Question: 19.064P

Reconsider Prob. 19-63. There seems to be some uncertainty about the average temperature of the surfaces surrounding the pipe used in radiation calculations, and you are asked to determine if it makes any significant difference in overall heat transfer. Repeat the calculations for average surrounding and surface temperatures of -20°C and 25°C, respectively, and determine the change in the values obtained.

Step 1 of 7

Given:

Diameter of steam pipe, $D = 12\,\text{cm} = 0.12\,\text{m}$

Length of steam pipe, $L = 12\,\text{m}$

Outer surface temperature of pipe, $T_s = 75°\text{C}$

Ambient air temperature, $T_\infty = 5°\text{C}$

Velocity of air,

$$V = 25\,\text{km/h}$$

$$= 25 \times \frac{1000}{3600}$$

$$= 6.944\,\text{m/s}$$

Emissivity of the surface of the pipe, $\varepsilon = 0.8$

Surrounding temperature, $T_{sur} = 0°\text{C}$

Step 2 of 7

Film temperature of air is

$$T_f = (T_s + T_\infty)/2$$

$$= (75+5)/2$$

$$= 40°\text{C}$$

Let us take the air properties at film temperature, T_f

From the table, properties of air at 1 atm pressure and $T = 40°\text{C}$

Thermal conductivity, $k = 0.02662\,\text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.702 \times 10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7255$

Calculating Reynolds number to know the flow conditions,

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{6.944 \times 0.12}{1.702 \times 10^{-5}}$$

$$= 48958.87$$

This is less than $\text{Re} < \text{Re}_{\text{critical}} = 5 \times 10^5$, therefore the flow is laminar flow.

Step 3 of 7

The average Nusselt number for cross flow over cylinder is determined from the Churchill & Bernstein relation given by

$$Nu = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(48958.87)^{0.5}(0.7255)^{1/3}}{\left[1+\left(\frac{0.4}{0.7255}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{48958.87}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{123.268}{1.1371} \times (1.2598)$$

$$= 136.87$$

Step 4 of 7

The convective heat transfer coefficient of air is determined from the equation given by

$$\text{Nu} = \frac{hD}{k}$$

Rewrite and solve for h:

$$h = \frac{\text{Nu} \times k}{D}$$

$$= \frac{136.87 \times 0.02662}{\left(12 \times 10^{-2}\right)}$$

$$= 30.368\ \text{W/m}^2\cdot\text{K}$$

Step 5 of 7

Therefore rate of heat loss by convection and radiation will be

$$\dot{Q}_t = R_{conv} + Q_{rad}$$

$$= hA_s\left(T_s - T_\infty\right) + \varepsilon A_s \sigma\left(T_s^4 - T_\infty^4\right)$$

$$= h\pi DL\left(T_s - T_\infty\right) + \varepsilon\pi DL\sigma\left(T_s^4 - T_\infty^4\right) \quad \ldots\ldots (1)$$

$$= 30.368 \times \pi \times 0.12 \times 12(75-5) + 0.8 \times \pi \times 0.12 \times 12 \times 5.67 \times 10^{-8} \times \left((75+273)^4 - (273+5)^4\right)$$

$$= 9516.71 + 1783.91$$

$$= 11300.62\,\text{W}$$

Step 6 of 7

Now as given the new average surrounding temperature $T_\infty = -20°\text{C}$ used in radiation calculations

From equation (1) calculating $\dot{Q}_t$ by putting this new value of surrounding temperature in radiation calculation

$$\dot{Q}_{t1} = h\pi DL\left(T_s - T_\infty\right) + \varepsilon\pi DL\sigma\left(T_s^4 - T_\infty^4\right)$$

$$= 30.368 \times \pi \times 0.12 \times 12(75-5) + 0.8 \times \pi \times 0.12 \times 12 \times 5.67 \times 10^{-8} \times \left((75+273)^4 - (-20+273)^4\right)$$

$$= 9616.71 + 2168.804$$

$$= 11785.514\,\text{W}$$

Therefore the change in amount of heat transfer when the new average surrounding temperature $T_\infty = -20°\text{C}$ used in radiation calculation is

$$\dot{Q}_{t1} - \dot{Q}_t = 11785.514 - 11300.62$$

$$= 484.894\,\text{W}$$

Therefore, the change in amount of heat transfer when the new average surrounding temperature $T_\infty = -20°\text{C}$ used is $\boxed{\dot{Q}_{t1} - \dot{Q}_t = 484.894\,\text{W}}$

Step 7 of 7

Now as given the new average surface temperature $T_s = 25°\text{C}$ used in radiation calculations

From equation (1) calculating $\dot{Q}_t$ by putting this new value of surface temperature in radiation calculation

$$\dot{Q}_{t2} = h\pi DL\left(T_s - T_\infty\right) + \varepsilon\pi DL\sigma\left(T_s^4 - T_\infty^4\right)$$

$$= 30.368 \times \pi \times 0.12 \times 12(75-5) + 0.8 \times \pi \times 0.12 \times 12 \times 5.67 \times 10^{-8} \times \left((25+273)^4 - (5+273)^4\right)$$

$$= 9616.711 + 392.623$$

$$= 10009.334\,\text{W}$$

Here we can see that the heat transfer is decreases when the surface temperature $T_s = 25°\text{C}$ used in radiation calculation

The change in heat transfer is given by

$$\dot{Q}_t - \dot{Q}_{t2} = 11300.62 - 10009.334$$

$$= 1291.286\,\text{W}$$

Therefore, the change in heat transfer is $\boxed{\dot{Q}_t - \dot{Q}_{t2} = 1291.286\,\text{W}}$

Question: 19.065P

A 10-cm-diameters, 30-cm-high cylindrical bottle contains cold water at 3°C. The bottle is placed in windy air at 27°C. The water temperature is measured to be 11°C after 45 min of cooling. Disregarding radiation effects and heat transfer from the top and bottom surfaces, estimate the average wind velocity.

Step 1 of 6

Given
Diameter of bottle = 10 cm
Height of bottle = 30 cm
Temperature of cold water = 3°C
Temperature of air = 27°C
Temperature of water after 45 min = 11°C

Step 2 of 6

Average film temperature of water, $T = \frac{3+11}{2}$

$$T = 7^0\text{C}$$

Properties of water from properties of water table at 7°C are

$\rho = 999.8\ \text{kg/m}^3$

$C_p = 4200\ \text{J/kg.°C}$

Average film temperature of air, $T = \frac{7+27}{2}$

$$T = 17^0\text{C}$$

Properties of air from properties of air table at 1 atm and 17°C are

$k = 0.02491\ \text{W/m.°C}$

$\upsilon = 1.489\times10^{-5}\ \text{m}^2/\text{s}$

$\text{Pr} = 0.7317$

Step 3 of 6

Mass of water in the bottle, $m = \rho V$

$$m = \rho\pi\frac{D^2}{4}L$$

$$m = (999.8\ \text{kg/m}^3)\pi\frac{(0.10\ \text{m})^2}{4}(0.30\ \text{m})$$

$$m = 2.356\ \text{kg}$$

Heat added to water, $Q = mC_p(T_2 - T_1)$

$Q = (2.356\ \text{kg})(4200\ \text{J/kg.°C})(11\text{-}3)\text{°C}$

$Q = 79{,}162\ \text{J}$

Heat transfer rate, $\dot{Q} = \frac{Q}{\Delta t}$

$$\dot{Q} = \frac{79{,}162\ \text{J}}{45\times60\ \text{s}}$$

$$\dot{Q} = 29.32\ \text{W}$$

Step 4 of 6

Surface area of the cylinder, $A_s = \pi DL$

$$A_s = \pi(0.10\ \text{m})(0.30\ \text{m})$$

$$A_s = 0.09425\ \text{m}^2$$

Heat transfer rate of conduction, $\dot{Q}_{conv} = hA_s(T_s - T_\infty)$

Equating the heat transfer rates,

$29.32\ \text{W} = h\times(0.09425\ \text{m}^2)(27\text{-}7)\text{°C}$

$h = 15.55\ \text{W/m}^2\text{.°C}$

Step 5 of 6

Nusselt number, $Nu = \frac{hD}{k}$

$$Nu = \frac{(15.55\ \text{W/m}^2\text{.°C})(0.10\ \text{m})}{0.02491\ \text{W/m.°C}}$$

$$Nu = 62.42$$

Reynolds number is calculate using the relation,

$$Nu = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$62.42 = 0.3 + \frac{0.62\text{Re}^{0.5}(0.7317)^{1/3}}{\left[1+(0.4/0.7317)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$\text{Re} = 12{,}856$

Step 6 of 6

Velocity of air is calculated using the relation,

$$\text{Re} = \frac{VD}{\upsilon}$$

$$12{,}856 = \frac{V(0.10\ \text{m})}{1.489\times10^{-5}\ \text{m}^2/\text{s}}$$

$V = 1.91\ \text{m/s}$

Average wind velocity is $\boxed{V = 1.91\ \text{m/s}}$

Question: 19.066P

A 0.2 m ×0.2 m street sign surface has an absorptivity of 0.6 and an emissivity of 0.7, while the street sign is subjected to a cross flow wind at 20°C with a velocity of 1 m/s. Solar radiation is incident on the street sign at a rate of 1100 W/m2, and the surrounding temperature is 20°C. Determine the surface temperature of the street sign. Evaluate the air properties at 30°C.

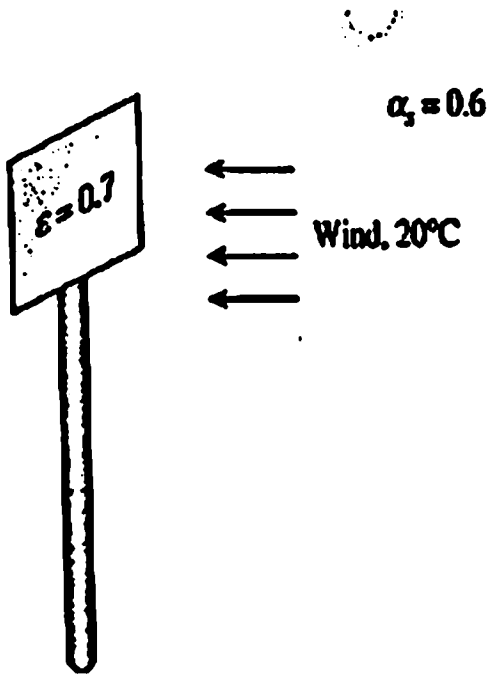


FIGURE P19-66

Step 1 of 5

Given data:

Size of street sign surface, $L \times w = 0.2\text{m} \times 0.2\text{m}$

Absorptivity of the surface, $\alpha = 0.6$

Emissivity of the surface, $\varepsilon = 0.7$

Wind flow velocity, $V = 1\ \text{m/s}$

Temperature of the air, $T_\infty = 20°\text{C}$

Incident solar radiation, $\dot{q}_s = 1100\ \text{W/m}^2$

Temperature of the surroundings, $T_{sur} = 20°\text{C}$

Step 2 of 5

From the table, properties of air at 1 atm pressure and $T = 30°\text{C}$

Thermal conductivity, $k = 0.02588\ \text{W/m} \cdot \text{K}$

Kinematic viscosity, $\nu = 1.608 \times 10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

The Reynolds number for the flow is determined from the equation given by

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{1 \times 0.2}{1.608 \times 10^{-5}}$$
$$= 12437.81$$

Step 3 of 5

From the table, empirical correlations for the average Nusselt number for forced convection over circular and noncircular cylinders in cross flow

For vertical plate Reynolds number ranging $4000 - 15000$

$$\text{Nu} = 0.228\text{Re}^{0.731}\text{Pr}^{1/3}$$

The Nusselt number from the given equation is determined as

$$\text{Nu} = 0.228 \times (12437.81)^{0.731} \times (0.7282)^{1/3}$$
$$= 201.96$$

Step 4 of 5

The convective heat transfer coefficient of air is determined from the equation given by

$$\text{Nu} = \frac{hD}{k}$$

Rewrite and solve for h:

$$h = \frac{\text{Nu} \times k}{D}$$
$$= \frac{201.96 \times 0.02588}{0.2}$$
$$= 26.13\ \text{W/m}^2 \cdot \text{K}$$

Step 5 of 5

The surface temperature of the street sign is determined by applying the energy balance given by

$$\alpha \times \dot{q}_s = \dot{q}_{conv} + \dot{q}_{rad}$$
$$\alpha \times \dot{q}_s = \left[h(T_s - T_\infty)\right] + \left[\varepsilon\sigma\left(T_s^4 - T_{sur}^4\right)\right]$$
$$0.6 \times 1100 = \left[26.13 \times (T_s - 20)\right] + \left[0.7 \times 5.67 \times 10^{-8}\left((T_s + 273)^4 - (20 + 273)^4\right)\right]$$
$$T_s = 41.6°\text{C}$$

Hence, the surface temperature of the street sign is $\boxed{T_s = 41.6°\text{C}}$

Question: 19.067P

What is the physical significance of the number of transfer units NTU = *hAs/mcp*? What do small and large NTU values tell about a heat transfer system?

Step 1 of 1

NTU (i.e. number of transfer units) is one of the parameters for the measure of effectiveness of the heat transfer system.
Larger NTU value indicates that the limit has reached for heat transfer whereas smaller value of NTU indicates more opportunities for heat transfer.

Question: 19.068P

Consider the flow of mercury (a liquid metal) in a tube. How will the hydrodynamic and thermal entry lengths compare if the flow is laminar? How would they compare if the How were turbulent?

Step 1 of 1

Considering the flow of mercury in a tube:

In laminar flow of mercury, thermal entry length is smaller than the hydrodynamic entry length.
For turbulent flow of mercury, thermal entry length is equal to the hydrodynamic entry length.

Question: 19.069P

What do the average velocity Vavg and the mean temperature *Tm* represent in flow through circular tubes of constant diameter?

Step 1 of 1

Average velocity represents the velocity which satisfies the conservation of mass in incompressible flow of fluid in circular tube of constant diameter.

Average temperature or mean temperature represents the temperature which satisfies the conservation of energy in incompressible flow of fluid in a circular tube of constant diameter.

Question: 19.070P

What does the logarithmic mean temperature difference represent for flow in a tube whose surface temperature is constant? Why do we use the logarithmic mean temperature instead of the arithmetic mean temperature?

Step 1 of 1

Logarithmic Mean Temperature Difference represents average temperature difference between the fluid and the surface. It reflects the exponential decay of local temperature difference.

$$\Delta T_{ln} = \frac{\Delta T_e - \Delta T_i}{\ln\left(\frac{\Delta T_e}{\Delta T_i}\right)}$$

here $\Delta T_e = T_s - T_e$

$\Delta T_i = T_s - T_i$

T_s = Surface temperature

T_i = Inlet temperature

T_e = Exit temperature

If we use arithmetic mean temperature difference instead of LMTD then if there is a larger difference between ΔT_e and ΔT_i, error increases to the undesirable level.

Question: 19.071P

How is the thermal entry length defined for flow in a tube? In what region is the flow in a tube fully developed?

Step 1 of 2

When a fluid enters a circular tube with different surface temperature, convection heat transfer takes place between the surface and fluid particles and thermal boundary layer develops.

Step 2 of 2

The thermal boundary layer develops from the surface and reaches the tube center; this region is called the thermal entrance region.

The length of tube in thermal entrance region is called the thermal entry length, L_t .

Fully developed region occurs after the thermal entry length and the dimensionless temperature profile, $\frac{(T_s - T)}{(T_s - T_m)}$ remains constant.

Question: 19.072P

Consider laminar forced convection in a circular tube. Will the heat flux be higher near the inlet of the tube or near the exit? Why?

Step 1 of 2

For a laminar flow the thickness of the thermal boundary layer near the entry of the circular tube is zero and gradually increases up to the fully developed region.

Step 2 of 2

The heat transfer coefficient depends on boundary layer thickness hence heat transfer coefficient is more at the tube inlet where thickness of boundary layer is small.

The heat flux is directly proportional to the heat transfer coefficient, so the heat flux is higher near the tube inlet.

Question: 19.073P

Consider turbulent forced convection in a circular tube. Will the heat flux be higher near the inlet of the tube or near the exit? Why?

Step 1 of 1

The heat flux will be always higher at the inlet of the tube. This is mainly due to decrease in the heat transfer coefficient with increase in the thickness of the boundary layer.

The heat transfer coefficient will be highest at the tube inlet where the thickness of the thermal boundary layer is zero. The heat transfer coefficient decreases gradually to the fully developed value.

Question: 19.074P

Consider fluid flow in a tube whose surface temperature remains constant. What is the appropriate temperature difference for use in Newton's law of cooling with an average heat transfer coefficient?

Step 1 of 1

The appropriate temperature difference for use in Newton's law of cooling with an average heat transfer coefficient is arithmetic mean temperature difference (ΔT_{am}).

$$\Delta T_{am} = T_s - T_b$$

Where, T_s is surface temperature of the tube

T_b is the bulk mean temperature

$$T_b = \frac{T_i + T_e}{2}$$

T_i is mean fluid temperature at the inlet of the tube

T_e is mean fluid temperature at the exit of the tube

Therefore the arithmetic mean temperature difference is

$$\Delta T_{am} = T_s - \left(\frac{T_i + T_e}{2}\right)$$

Question: 19.075P

Air enters an 18-cm-diameter 12-m-long underwater duct at 50°C and 1 atm at a mean velocity of 7 m/s, and is cooled by the water outside. If the average heat transfer coefficient is 65 W/m2-K and the tube temperature is nearly equal to the water temperature of 10°C, determine the exit temperature of air and the rate of heat transfer.

Step 1 of 7

Given:

Diameter of duct, $D = 18 \text{ cm} = 0.18 \text{ m}$

Length of duct, $L = 12 \text{ m}$

Temperature of air, $T_i = 50°C$

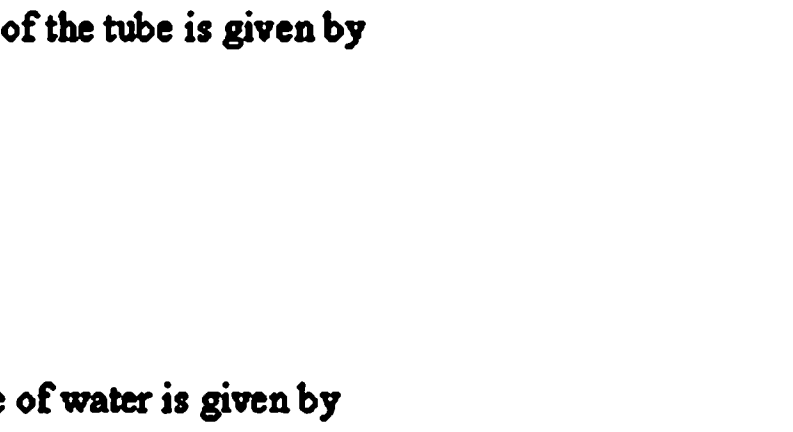
Mean velocity of air, $V_{avg} = 7 \text{ m/s}$

Tube temperature, $T_s = 10°C$

Average heat transfer co-efficient is $h = 65 \text{ W/m}^2 \cdot \text{K}$

Step 2 of 7

Schematic diagram for pipe flow:



Step 3 of 7

As we know, the rate of heat transfer can be expressed by

$\dot{Q} = hA_s \Delta T_{ln}$

where, A_s – Surface area of heat transfer.

h – Heat transfer coefficient (average)

ΔT_{ln} – Log mean temperature difference

Step 4 of 7

Mean film temperature of the air is

$$T_f = \frac{50+10}{2}$$

$$= 30°C$$

From the properties of the air at $T_f = 30°C$

Density, $\rho = 1.164 \text{ kg/m}^3$

Specific heat, $c_p = 1007 \text{ J/kg} \cdot \text{k}$

Step 5 of 7

Cross sectional area of the tube is given by

$$A_c = \frac{\pi}{4} D^2$$

$$= \frac{\pi}{4}(0.18)^2$$

$$= 0.0254 \text{ m}^2$$

Then, mass flow rate of water is given by

$$\dot{m} = \rho A_c V_{avg}$$

$$= (1.164)(0.0254)(7)$$

$$= 0.207 \text{ kg/s}$$

Therefore surface area of heat transfer,

$$A_s = \pi D L$$

$$= \pi \times (0.18) \times 12$$

$$= 6.786 \text{ m}^2$$

Step 6 of 7

We know exit temperature can be expressed by the relationship i.e. for mean fluid temperature at the tube exit.

$$T_e = T_s - (T_s - T_i) exp(-hA_s / \dot{m} c_p)$$

$$= 10 - (10 - 50) \exp\left(\frac{-65 \times 6.786}{0.207 \times 1007}\right)$$

$$= 14.82°C$$

Therefore, the exit temperature is $\boxed{T_e = 14.82°C}$

Step 7 of 7

Logarithmic mean temperature deference, T_{ln}

$$\Delta T_{ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$

$$= \frac{14.82 - 50}{\ln\left(\frac{10 - 14.82}{10 - 50}\right)}$$

$$= 16.625°C$$

Rate of heat transfer,

$$\dot{Q} = hA_s \Delta T_{ln}$$

$$= 65 \times 6.786 \times 16.625$$

$$= 7333.07 \text{ W}$$

Therefore, rate of heat transfer is $\boxed{\dot{Q} = 7333.07 \text{ W}}$

Question:

Cooling water available at 10°C is used to condense steam at 30°C in the condenser of a power plant at arate of 0.15 kg/s by circulating the cooling water through a bank of 15-m-long 1.2-cm-internal-diameter thin copper tubes. Water enters the tubes at a mean velocity of 4 m/s and leaves at al temperature of 24°C. The tubes are nearly isothermal at 30° C Determine the average heat transfer coefficient between the water, the tubes, and the number of tubes needed to achieve the indicated heat transfer rate in the condenser.

Step 1 of 11

Calculate the bulk mean temperature of water to evaluate the properties of water.

$$T_b = \frac{T_i + T_e}{2}$$

Here, T_b is the bulk mean temperature of water, T_i is the inlet temperature and T_e is the exit temperature.

Substitute $24°C$ for T_e and $10°C$ for T_i.

$$\begin{aligned} T_b &= \frac{T_i + T_e}{2} \\ &= \frac{10+24}{2} \\ &= 17°C \end{aligned}$$

Step 2 of 11

Obtain the properties of water at the bulk mean temperature $17°C$ from the "Properties of saturated water" table.

Density of the water, $\rho = 998.7\,kg/m^3$

Viscosity of the water, $\nu = 1.1233\times10^{-6}\,m^2/s$

Prandtl number of the water, $Pr = 7.99$

Thermal conductivity of the water, $k = 0.590\ W/m\cdot K$

Specific heat of the water, $C_p = 4.183\ kJ/kg\cdot K$

Step 3 of 11

Calculate the cross sectional area of the copper tube by using the equation:

$$A = \frac{\pi}{4}d^2$$

Here, A is the cross sectional area of the copper tube and d is the internal diameter of the copper tube.

Substitute $0.012\,m$ for d.

$$\begin{aligned} A &= \frac{\pi}{4}\times0.012^2 \\ &= 1.1309\times10^{-4}\,m^2 \end{aligned}$$

Step 4 of 11

Calculate the mass flow rate of water passing through the tubes by using the equation:

$$\dot{m} = \rho A V$$

Here, $\dot{m}$ is the mass flow rate of water passing through the tubes and V is the flow velocity.

Substitute $998.7\,kg/m^3$ for ρ, $\left(1.1309\times10^{-4}\right)m^2$ for A and $4\,m/s$ for V

$$\begin{aligned} \dot{m} &= 998.7\times1.1309\times10^{-4}\times4 \\ &= 0.4518\,kg/s \end{aligned}$$

Step 5 of 11

Calculate the heat transfer through one tube.

$$\dot{Q} = \dot{m}c_p\left(T_e - T_i\right)$$

Here, $\dot{Q}$ is the heat transfer through one tube, T_e is the exit temperature and T_i is the inlet temperature.

Substitute $0.4518\,kg/s$ for $\dot{m}$, $4.183\,kJ/kg\cdot K$ for c_p, $24°C$ for T_e and $10°C$ for T_i.

$$\begin{aligned} \dot{Q} &= 0.4518\times4.183\times(24-10) \\ &= 26.46\,kW \end{aligned}$$

Step 6 of 11

Calculate the log-mean temperature difference by using the equation:

$$\Delta T_{ln} = \frac{\Delta T_e - \Delta T_i}{\ln\left(\Delta T_e / \Delta T_i\right)}$$

Substitute $\left(T_s - T_e\right)$ for ΔT_e and $\left(T_s - T_i\right)$ for ΔT_i.

$$\Delta T_{ln} = \frac{\left(T_s - T_e\right)-\left(T_s - T_i\right)}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$

Substitute $30°C$ for T_s, $24°C$ for T_e and $10°C$ for T_i.

$$\begin{aligned} \Delta T_{ln} &= \frac{(30-24)-(30-10)}{\ln\left(\frac{30-24}{30-10}\right)} \\ &= \frac{6-20}{\ln\left(\frac{6}{20}\right)} \\ &= 11.63°C \end{aligned}$$

Step 7 of 11

$= 11.63°C$

Step 7 of 11

Calculate the surface area of each copper tube by using the equation:

$$A_s = \pi dL$$

Here, A_s is the surface area of each copper tube and L is the length of each copper tube.

Substitute $0.012\,\text{m}$ for d and $5\,\text{m}$ for L.

$$\begin{aligned} A_s &= \pi \times 0.012 \times 5 \\ &= 0.1885\,\text{m}^2 \end{aligned}$$

Step 8 of 11

Calculate the average heat transfer coefficient by using the equation:

$$\dot{Q} = hA_s\Delta T_{lm}$$

$$\Rightarrow h = \frac{\dot{Q}}{A_s\Delta T_{lm}}$$

Here, $\dot{Q}$ is the heat transfer through one tube and h is the average heat transfer coefficient.

Substitute $26.46\,\text{kW}$ for $\dot{Q}$, $0.1885\,\text{m}^2$ for A_s and $11.63°C$ for ΔT_{lm}.

$$\begin{aligned} h &= \frac{26.46}{0.1885 \times 11.63} \\ &= 12.1\,\text{W/m}^2 \cdot °C \end{aligned}$$

Step 9 of 11

Obtain the enthalpy of vaporization of water at $30°C$ from the "Properties of saturated water" table.

$$h_{fg} = 2431\,\text{kJ/kg}$$

Here, h_{fg} is the enthalpy of vaporization of water at $30°C$.

Step 10 of 11

Calculate the total heat transfer rate by using the equation:

$$\dot{Q}_{total} = \dot{m}_{cond} h_{fg}$$

Here, $\dot{Q}_{total}$ is the total heat transfer rate and $\dot{m}_{cond}$ is the mass flow rate of the condensate.

Substitute $0.15\,\text{kg/s}$ for $\dot{m}_{cond}$ and $2431\,\text{kJ/kg}$ for h_{fg}.

$$\begin{aligned} \dot{Q}_{total} &= 0.15 \times 2431 \\ &= 364.65\,\text{kW} \end{aligned}$$

Step 11 of 11

Calculate the number of tubes needed to achieve the indicated heat transfer rate in the condenser by using the equation:

$$n = \frac{\dot{Q}_{total}}{\dot{Q}}$$

Here, n is the number of tubes needed to achieve the indicated heat transfer rate in the condenser.

Substitute $364.65\,\text{kW}$ for $\dot{Q}_{total}$ and $26.46\,\text{kW}$ for $\dot{Q}$.

$$\begin{aligned} n &= \frac{364.65}{26.46} \\ &= 13.78 \\ &\simeq 14 \end{aligned}$$

Therefore, the number of tubes needed to achieve the indicated heat transfer rate in the condenser is $\boxed{14}$.

Question: 19.077P

Repeat Prob. 19-76 for steam condensing at a rate of 0.60 kg/s.

Step 1 of 4

Calculate the Mass flow rate of water

$$\dot{m}_w = \rho A V_{mean}$$
$$= \rho \frac{\pi}{4} D^2 V_{mean}$$

Here,

Density of the water is ρ

Diameter of the tube is D

Mean velocity of flow, V_{mean}

Substitute $4\ \text{m/s}$ for V_{mean}, $0.012\ \text{m}$ for D, $998.7\ \text{kg/m}^3$ for ρ

$$\dot{m}_w = \rho A V_{mean}$$
$$= \rho \frac{\pi}{4} D^2 V_{mean}$$
$$= 998.7 \times \frac{\pi}{4} \times 0.012^2 \times 4$$
$$= 0.4518\ \text{kg/sec}$$

Step 2 of 4

Calculate the rate of heat transfer from one tube is

$$\dot{Q} = \dot{m}_w c_p (T_e - T_i)$$

Substitute $4.183\ \text{kJ/kg} \cdot \text{K}$ for c_p, $0.4518\ \text{kg/sec}$ for $\dot{m}_w$

$$\dot{Q} = \dot{m}_w c_p (T_e - T_i)$$
$$= 0.4518 \times 4.183 \times (24 - 10)$$
$$= 26.458\ \text{kW}$$

Step 3 of 4

Calculate the Surface area of the tube

$$A_s = \pi D L$$

Here,

Length of the tube is L

Substitute $5\ \text{m}$ for L, $0.012\ \text{m}$ for D

$$A_s = \pi D L$$
$$= \pi \times 0.012 \times 5$$
$$= 0.1884\ \text{m}^2$$

Now, log-mean temperature difference is to be determined

$$\Delta T_{lm} = \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e / \Delta T_i)}$$

Where $\Delta T_e = T_s - T_e$

$$= 30 - 24$$
$$= 6°\text{C}$$

$$\Delta T_i = T_s - T_i$$
$$= 30 - 10$$
$$= 20°\text{C}$$

Substitute $6°\text{C}$ for ΔT_e, $20°\text{C}$ for ΔT_i

$$\Delta T_{lm} = \frac{6 - 20}{\ln\left(\frac{6}{20}\right)}$$

$$\Delta T_{lm} = 11.62°\text{C}$$

Step 4 of 4

Calculate the total heat transfer required or indicated heat transfer rate are:

$$Q_1 = h A_s \Delta T_{lm}$$

Substitute $13152.08\ \text{W/m}^2 \cdot \text{K}$ for h, $284.6°\text{C}$ for ΔT_{lm}

$$= 13152.08 \times 3.14 \times 0.012 \times 5 \times 284.62$$
$$= 141.05\ \text{kW}$$

Calculate the Number of tube required be n

$$n = \frac{\dot{Q}_1}{\dot{Q}}$$

Substitute $141.05\ \text{kW}$ for Q_1, $8.788\ \text{kW}$ for Q

$$n = \frac{\dot{Q}_1}{\dot{Q}}$$
$$= \frac{141.05}{8.788}$$
$$= 16.05$$

Hence, the number of tubes is required for heat transfer is **17**

Question: 19.078P

Combustion gases passing through a 5-cm-internal-diameter circular tube are used to vaporize waste water at atmospheric pressure. Hot gases enter the tube at 115 kPa 1 and 250°C at a mean velocity of 5 m/s, and leave at 150°C. If the average heat transfer coefficient is 120 W/m2·K and the j inner surface temperature of the tube is 110°C, determine (*a*) the tube length and (*b*) the rate of evaporation of water.

Step 1 of 8

Given data:

Diameter of tube, $D = 5\text{ cm} = 0.05\text{ m}$

Hot gases inlet temperature, $T_i = 250°C$

Mean velocity, $V_{avg} = 5\text{ m/s}$

Exit temperature, $T_e = 150°C$

Average heat transfer coefficient, $h = 120\text{ W/m}^2\cdot\text{K}$

Inner surface temperature of tube, $T_s = 110°C$

Hot gases entry pressure, $P = 115\text{ kPa}$

Step 2 of 8

Mean film temperature of gas is given by

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{(250+150)}{2}$$
$$= 200°C$$

From the table of 'properties of air at 1 atm pressure' at $T_{avg} = 200°C$ we get

Specific heat, $c_p = 1023\text{ J/kg}\cdot\text{K}$

Constant, $R = 0.287\text{ kJ/kg}\cdot\text{K}$

Step 3 of 8

Density of the air,

$$\rho = \frac{P}{RT}$$
$$= \frac{115}{0.287\times(250+273)}$$
$$= 0.766\text{kg/m}^3$$

Step 4 of 8

Mass flow rate,

$$\dot{m} = \rho A_C \cdot V_{avg}$$
$$= 0.766\times\frac{\pi D^2}{4}\times 5$$
$$= \frac{0.766\times\pi\times(0.05)^2\times 5}{4}$$
$$= 7.52\times10^{-3}\text{ kg/s}$$

Therefore rate of heat transfer,

$$\dot{Q} = \dot{m}c_p(T_i - T_e)$$
$$= 7.52\times10^{-3}\times1023\times(250-150)$$
$$= 769.32\text{W}$$

Step 5 of 8

Therefore logarithmic mean temperature,

$$\Delta T_{ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$
$$= \frac{150-250}{\ln\left[\frac{110-150}{110-250}\right]}$$
$$= 79.82°C$$

Step 6 of 8

The amount of heat transfer is determined by the relation

$$Q = hA_s\Delta T_{ln}$$

Re-arranging the above relation for area we get

$$A_s = \frac{Q}{h\Delta T_{ln}}$$
$$= \frac{769.32}{120\times79.82}$$
$$= 0.0803\text{m}^2$$

Step 7 of 8

a)

We know surface area A_s can also be expressed by

$$A_s = \pi DL$$

Therefore

$$L = \frac{A_s}{\pi D}$$
$$= \frac{0.0803}{\pi\times(0.05)}$$
$$= 0.511\text{m}$$

Therefore length of the tube is $\boxed{L = 51.13\text{ cm}}$

Step 8 of 8

b)

Rate of evaporation, $\dot{m}_w$ can be determined from i.e for water

$$\dot{Q} = \dot{m}_w$$

As heat transfer rate, $\dot{Q}$ to water can also be expressed as:

$\dot{Q}$ = Rate of evaporation of water × Enthalpy of vaporization

$$\dot{Q} = \dot{m}_w \times h_{fg}$$

Take value of h_{fg} from saturated water properties table.

At $T = 100°C$

$h_{fg} = 2257\text{ kJ/kg}$

Rate of evaporation of water is given by

$$\dot{m}_w = \frac{\dot{Q}}{h_{fg}}$$
$$= \frac{769.32}{2257\times1000}$$

Therefore, rate of evaporation of water is $\boxed{\dot{m}_w = 3.409\times10^{-4}\text{ kg/s}}$

Question: 19.079P

Repeat Prob. 19-78 for a heat transfer coefficient of 40 W/m2·K.

Step 1 of 8

Given data:

Diameter of tube, $D = 5\text{ cm} = 0.05\text{ m}$

Hot gases inlet temperature, $T_i = 250°\text{C}$

Mean velocity, $V_{avg} = 5\text{ m/s}$

Exit temperature, $T_e = 150°\text{C}$

Average heat transfer coefficient, $h = 40\text{ W/m}^2\text{-K}$

Inner surface temperature of tube, $T_s = 110°\text{C}$

Hot gases entry pressure, $P = 115\text{ kPa}$

Step 2 of 8

Calculate the mean film temperature of gas,

$$T_{avg} = \frac{T_i + T_e}{2}$$

Where,

T_i is hot gases inlet temperature,

T_e is the exit temperature.

Substitute $T_i = 250°\text{C}$ and $T_e = 150°\text{C}$ in the above equation,

$$T_{avg} = \frac{(250+150)}{2}$$

$$= 200°\text{C}$$

From the table of 'properties of air at 1 atm pressure'

At $T_{avg} = 200°\text{C}$

Specific heat, $c_p = 1023\text{ J/kg·K}$

Constant, $R = 0.287\text{ kJ/kg·K}$

Step 3 of 8

Calculate the density of the air,

$$\rho = \frac{P}{RT_i}$$

Where,

P is the hot gases entry pressure,

Substitute $P = 115\text{ kPa}$, $R = 0.287\text{ kJ/kg·K}$ and $T_i = 250°\text{C}$ in the above equation,

$$\rho = \frac{115}{0.287\times(250+273)}$$

$$= 0.766\text{kg/m}^3$$

Step 4 of 8

Calculate the mass flow rate,

$$\dot{m} = \rho A_C \cdot V_{avg}$$

Where,

A_C is the area of the cross-section of the tube,

V_{avg} is the mean velocity.

$$\dot{m} = 0.766\times\frac{\pi D^2}{4}\times 5$$

$$= \frac{0.766\times\pi\times(0.05)^2\times 5}{4}$$

$$= 7.52\times10^{-3}\text{ kg/s}$$

Calculate the rate of heat transfer,

$$\dot{Q} = \dot{m}c_p(T_i - T_e)$$

$$= 7.52\times10^{-3}\times1023\times(250-150)$$

$$= 769.32\text{W}$$

Step 5 of 8

Calculate the logarithmic mean temperature,

$$\Delta T_{ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$

T_s is the inner surface temperature of tube,

Substitute $T_s = 110°\text{C}$ and the respective values in the above equation,

$$\Delta T_{ln} = \frac{150-250}{\ln\left[\frac{110-150}{110-250}\right]}$$

$$= 79.82°\text{C}$$

Step 6 of 8

Calculate the amount of heat transfer,

$$Q = hA_s\Delta T_{ln}$$

Re-arranging the above relation for area we get

$$A_s = \frac{Q}{h\Delta T_{ln}}$$

h is the average heat transfer coefficient,

Substitute $h = 40\text{ W/m}^2\text{-K}$ and the respective values in the above equation,

$$A_s = \frac{769.32}{40\times79.82}$$

$$= 0.2409\text{ m}^2$$

Step 7 of 8

(a)

calculate the surface area A_s,

$$A_s = \pi DL$$

$$L = \frac{A_s}{\pi D}$$

$$= \frac{0.2409}{\pi\times(0.05)}$$

$$= 1.534\text{m}$$

Therefore length of the tube is $\boxed{153.4\text{ cm}}$

Step 8 of 8

b)

Rate of evaporation, $\dot{m}_w$ can be determined from i.e for water

$$\dot{Q} = \dot{m}_w$$

As heat transfer rate, $\dot{Q}$ to water can also be expressed as:

$\dot{Q}$ = Rate of evaporation of water × Enthalpy of vaporization

$$\dot{Q} = \dot{m}_w \times h_{fg}$$

Take value of h_{fg} from saturated water properties table.

At $T = 100°\text{C}$

$h_{fg} = 2257\text{ kJ/kg}$

Rate of evaporation of water is given by

$$\dot{m}_w = \frac{\dot{Q}}{h_{fg}}$$

$$= \frac{769.32}{2257\times1000}$$

Therefore, rate of evaporation of water is $\boxed{\dot{m}_w = 3.409\times10^{-4}\text{ kg/s}}$

Question: 19.080P

Determine the convection heat transfer coefficient for the flow of (*a*) air and (*b*) water at a velocity of 5 m/s in an 8-cm-diameter and 10-m-long tube when the tube is subjected to uniform heat flux from all surfaces. Use fluid properties at 25°C.

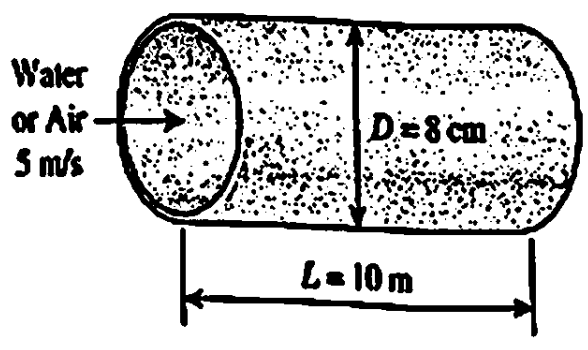


FIGURE P19-80

Step 1 of 8

Given data:

Diameter of pipe, $D = 8\,\text{cm} = 0.08\,\text{m}$

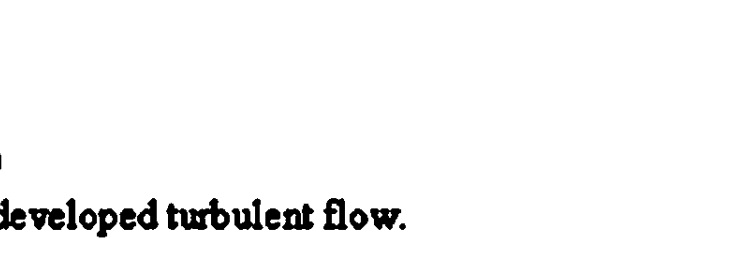
Length of pipe, $L = 10\,\text{m}$

Mean velocity of fluid flow, $V = 5\,\text{m/s}$

Fluid temperature, $T_f = 25°\text{C}$

Step 2 of 8

Schematic Diagram of pipe flow:



Step 3 of 8

(a)

For Air,

From the table properties of air at 1 atm pressure at $T = 25°\text{C}$

Thermal conductivity, $k = 0.02551\ \text{w/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.562\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7296$

Reynolds number is

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{5\times0.08}{1.562\times10^{-5}}$$
$$= 25608.19$$

As $\text{Re} > 10{,}000$. Therefore it is a turbulent flow.

Step 4 of 8

Now, entry length L_h, Turbulent

$$L_h = 10\text{D}$$
$$= 10\times0.08$$
$$= 0.8\ \text{m}$$

As $L_h < L$ (given)

Therefore it's a fully developed turbulent flow.

Step 5 of 8

Then, Nusselt number is given by

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$
$$= 0.023(25608.19)^{0.8}\,\text{Pr}^{0.4}$$
$$= 68.18$$

Also, N_u can be expressed as:

$$Nu = \frac{hD}{k}$$

Therefore heat transfer coefficient,

$$h = \frac{k}{D}\times Nu$$
$$= \frac{0.02551}{0.08}\times68.18$$
$$= 21.74$$

Therefore, heat transfer coefficient is $\boxed{h_{air} = 21.74\ \text{W/m}^2\cdot\text{K}}$

Step 6 of 8

(b)

For water,

From the properties of saturated water at $T = 25°\text{C}$

Thermal conductivity, $k = 0.607\ \text{W/m}\cdot\text{K}$

Density, $\rho = 997\ \text{kg/m}^3$

Dynamic viscosity, $\mu = 0.891\times10^{-3}\ \text{kg/m}\cdot\text{s}$

Prandtl number, $\text{Pr} = 6.14$

Step 7 of 8

Kinematic viscosity is given by

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.891\times10^{-3}}{997}$$
$$= 8.937\times10^{-7}\,\text{m}^2/\text{s}$$

Reynolds number is given by

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{5\times0.08}{8.937\times10^{-7}}$$
$$= 447577.49$$

Step 8 of 8

Definitely the flow is fully developed turbulent flow,

$$Nu = 0.23\text{Re}^{0.8}\text{Pr}^{0.4}.$$
$$= 0.023(447577.49)^{0.8}(6.14)^{0.4}$$
$$= 1576.485$$

Heat transfer coefficient for water,

$$h_{water} = \frac{k\cdot Nu}{D}$$
$$= \frac{0.607\times1576.485}{0.08}$$
$$= 11961.58\ \text{W/m}^2\cdot\text{K}$$

Therefore, heat transfer coefficient for water $\boxed{h_{water} = 11961.58\ \text{W/m}^2\cdot\text{K}}$

Question: 19.081P pg 1

An 8-m long, uninsulated square duct of cross section 0.2 m × 0.2 m and relative roughness 10-3 passes through the attic space of a house. Hot air enters the duct at 1 atm and 80°C at a volume flow rate of 0.15 m3/s. The duel surface is nearly isothermal at 60°C. Determine the rate of heat loss from the duct to the attic space and the pressure difference between the inlet and outlet sections of the duct.

Step 1 of 9

Given data:

The length of the square duct, $L = 8\ \text{m}$

Cross section of square duct, $a \times a = 0.2\text{m} \times 0.2\text{m}$

Relative roughness of the duct, $\frac{\varepsilon}{D} = 10^{-3}$

Temperature of air enters the duct, $T_i = 80°\text{C}$

Volume flow rate of air at inlet, $\dot{V} = \dot{V}_i = 0.15\text{m}^3/s$

Surface temperature of the duct, $T_s = 60°\text{C}$

Step 2 of 9

Properties of air at 80°C and 1atm are

$\rho = 0.9994\ \text{kg/m}^3$

$C_p = 1008\ \text{J/kg}\cdot\text{K}$

$k = 0.02953\ \text{W/m}\cdot\text{K}$

$\text{Pr} = 0.7154$

$\nu = 2.097\times10^{-5}\ \text{m}^2/s$

Step 3 of 9

Characteristic length of square duct D_h is given by

$$D_h = \frac{4A_c}{p}$$

Where A_c is the Cross-sectional area of square duct and p is the wetted perimeter

Now

$$D_h = \frac{4a^2}{4a}$$
$$= a$$
$$= 0.2\ \text{m}$$

Average velocity of air

$$V_{avg} = \frac{\dot{V}}{A_c}$$
$$= \frac{0.15}{(0.2)^2}$$
$$= 3.75\ \text{m/s}$$

Step 4 of 9

Reynolds number

$$\text{Re} = \frac{V_{avg}D_h}{\nu}$$
$$= \frac{3.75\times0.2}{2.097\times10^{-5}}$$
$$= 35765.379$$

Here Reynolds number is greater than 10,000. Hence the flow is turbulent and entry lengths in this case is roughly 10 times the hydraulic diameter

$$L_h \approx L_t \approx 10D$$
$$= 10\times0.2$$
$$= 2\text{m}$$

As this length is quite small when compared to the total length of tube, we can assume fully developed flow for the entire tube length.

Step 5 of 9

Nusselt number,

$$\text{Nu} = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.3}$$

$$\frac{hD_h}{k} = 0.023(35765.379)^{0.8}(0.7154)^{0.3}$$

$$\frac{hD_h}{k} = 91.38$$

Then, heat transfer coefficient

$$h = \frac{k}{D_h}\times91.38$$
$$= \frac{0.02953\times(91.38)}{0.2}$$
$$= 13.49\ \text{W/m}^2\cdot\text{K}$$

Step 6 of 9

Surface area of square duct

$$A_s = 4aL$$
$$= 4\times(0.2)\times(8)$$
$$= 6.4\ \text{m}^2$$

Mass flow rate,

$$\dot{m} = \rho\dot{V}$$
$$= (0.9994)\times(0.15)$$
$$= 0.150\ \text{kg/s}$$

Step 7 of 9

Exit temperature of air,

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}C_p}\right)$$

Step 7 of 9

Exit temperature of air,

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}C_p}\right)$$

$$= 60 - (60-80)\exp\left(-\frac{13.5\times 6.4}{0.15\times 1008}\right)$$

$$T_e = 71.3^0\text{C}$$

Logarithmic mean temperature difference,

$$\Delta T_{lm} = \frac{T_i - T_e}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$

$$= \frac{80-71.3}{\ln\left(\frac{60-71.3}{60-80}\right)}$$

$$= -15.2^\circ\text{C}$$

Step 8 of 9

Heat transfer,

$$Q = hA_s\Delta T_{lm}$$

$$= (13.5)(6.4)(-15.2)$$

$$= -1313.28\ \text{W}$$

Therefore air will lose heat at the rate of $\boxed{1313.28\ \text{W}}$ as it flows through the duct in the attic.

Step 9 of 9

The friction factor can be determined by using the equation

$$\frac{1}{\sqrt{f}} = -1.8\log\left(\frac{6.9}{\text{Re}} + \left(\frac{\varepsilon/D}{3.7}\right)^{1.11}\right)$$

$$= -1.8\log\left(\frac{6.9}{35765.379} + \left(\frac{10^{-3}}{3.7}\right)^{1.11}\right)$$

$$f = 0.024$$

Pressure difference between inlet and outlet sections of the duct can be calculated as

$$\Delta P_f = \frac{fL\rho V_{avg}^2}{D_h\times 2}$$

$$= \frac{0.024\times 8\times 0.9994\times (3.75)^2}{0.2\times 2}$$

$$= 6.74595\ \text{N/m}^2$$

Pressure difference, $\boxed{\Delta P_f = 6.74595\ \text{N/m}^2}$

Question: 19.082P

A 12-m long and 12-mm inner-diameter pipe made of commercial steel is used to heat a liquid in an industrial process. The liquid enters the pipe with *Ti*= 25°C, *V* = 0.8 m/s. A uniform heat flux is maintained by anelectric resistance heater wrapped around the outer surface of the pipe, so that the fluid exits at75°C. Assuming fully developed flow and taking the average fluid properties to be ρ =1000 kg/m3, $c\rho$ =4000 J/kg·K, μ= 2 x 10-3 kg/m·s. *k* =0.48 W/m·K, and Pr = 10, determine:

(a) The required surface heat flux *qs* produced by the heater

(b) The surface temperature at the exit, *Ts*

Step 1 of 6

Given data:

Length of pipe, $L = 12$ m

Diameter of pipe, $D = 12$ mm

Liquid entry temperature, $T_i = 25°C$

Velocity, $V = 0.8 m/s$

Exit temperature of fluid, $T_e = 75°C$

Density, $\rho = 1000 kg/m^3$

Specific heat, $C_p = 4000$ J/kg·K

Dynamic viscosity, $\mu = 2 \times 10^{-3}$ kg/m·s

Thermal conductivity, $k = 0.48$ W/m·K

Prandtl number, $Pr = 10$

Step 2 of 6

(a) The required surface heat flux, $\dot{q}_s$

Mass flow rate of the liquid

$$\dot{m} = \rho A V$$

$$= 1000 \times \pi \frac{(0.012)^2}{4} \times 0.8$$

$$= 0.09048 kg/s$$

Rate of heat transfer

$$\dot{Q} = \dot{m} C_p (T_e - T_i)$$

$$= 0.09048 \times 4000 \times (75 - 25)$$

$$= 18095.6 W$$

Step 3 of 6

Therefore surface heat flux produced by the heater

$$\dot{q}_s = \frac{\dot{Q}}{A_s}$$

$$= \frac{18095.6}{\pi D L}$$

$$= \frac{18095.6}{\pi \times 0.012 \times 12}$$

$$= 40000 W/m^2$$

The required surface heat flux is given by $\boxed{\dot{q}_s = 40,000 W/m^2}$

Step 4 of 6

(b) Surface temperature at exit, T_s

Let us calculate the Reynolds number to see the flow regime

$$R_e = \frac{\rho V D}{\mu}$$

$$= \frac{1000 \times 0.8 \times 0.012}{2 \times 10^{-3}}$$

$$= 4800$$

Since R_e calculated is more than 2300 but less than 10,000, the flow is a transitional flow as given, assuming it as fully developed turbulent flow, the Nusselt number can be

$$Nu = 0.023 Re^{0.8} Pr^{0.4}$$

$$= 0.023 (4800)^{0.8} (10)^{0.4}$$

$$= 50.90$$

Step 5 of 6

Heat transfer coefficient using the Nusselt number as

$$Nu = \frac{hD}{k}$$

$$h = \frac{k}{D} \times Nu$$

$$= \frac{0.48}{0.012} \times (50.90)$$

$$= 2036 W/m^2 \cdot K$$

Step 6 of 6

The surface heat flux can be given by the relation,

$$\dot{q}_s = h(T_s - T_e)$$

$$T_s = \frac{\dot{q}_s}{h} + T_e$$

$$= \frac{40,000}{2036} + 75$$

$$= 94.65°C$$

Hence the surface temperature at the exit is $\boxed{T_s = 94.65°C}$

Question: 19.083P

Determine the hydrodynamic and thermal entry lengths for water, engine oil, and liquid mercury flowing through a 2.5-cm-diameter smooth tube with mass flow rate of 0.01 kg/s and temperature of 100°C.

Step 1 of 7

Given data:

Diameter of smooth tube, $D = 2.5\ \text{cm}$

Mass flow rate of the fluid, $\dot{m} = 0.01\ \text{kg/s}$

Temperature of fluid, $T_f = 100°\text{C}$

Step 2 of 7

If the flowing fluid is water:

Taking properties of water at $T = 100°\text{C}$

Dynamic viscosity $\mu = 0.282\times10^{-3}\ \text{kg/m}\cdot\text{s}$

Prandtl number $\text{Pr} = 1.75$

Now calculating Reynolds number, Re to know about flow region

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.01}{\pi\times0.025\times0.282\times10^{-3}}$$

$$= 1806.013$$

Since it is less than 2300, therefore it is a laminar flow.

Step 3 of 7

Hydrodynamic length for laminar flow

$$L_h \approx 0.05\text{Re}\times\text{D}$$

$$\approx 0.05\times1806.013\times0.025$$

$$\approx 2.26\,\text{m}$$

Hence the Hydrodynamic length for water is $\boxed{L_h \approx 2.26\,\text{m}}$

Thermal entry length for laminar flow

$$L_t \approx 0.05\text{Re}\times\text{Pr}\times\text{D}$$

$$\approx 0.05\times1806.013\times1.75\times0.025$$

$$\approx 3.95\,\text{m}$$

Hence the thermal entry length for water is $\boxed{L_t \approx 3.95\,\text{m}}$

Step 4 of 7

If the flowing fluid is Engine oil:

Taking properties of Engine oil at 100°C

Dynamic viscosity $\mu = 0.01718\ \text{kg/m}\cdot\text{s}$

Prandtl number $\text{Pr} = 279.1$

Now calculating Reynolds number, Re to know about flow region

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.01}{\pi\times0.025\times0.01718}$$

$$= 29.645$$

Since it is less than 2300, therefore it is a laminar flow

Step 5 of 7

Hydrodynamic length for laminar flow

$$L_h \approx 0.05\text{Re}\times\text{D}$$

$$\approx 0.05\times29.645\times0.025$$

$$\approx 0.0371\,\text{m}$$

Hence the Hydrodynamic length for engine oil is $\boxed{L_h \approx 0.0371\,\text{m}}$

Thermal entry length for laminar flow

$$L_t \approx 0.05\text{Re}\times\text{Pr}\times\text{D}$$

$$\approx 0.05\times29.645\times279.1\times0.025$$

$$\approx 10.34\,\text{m}$$

Hence the thermal entry length for engine oil is $\boxed{L_t \approx 10.34\,\text{m}}$

Step 6 of 7

If the flowing fluid is Mercury:

Taking properties of liquid mercury at 100°C

Dynamic viscosity $\mu = 1.245\times10^{-3}\ \text{kg/m}\cdot\text{s}$

Prandtl number $\text{Pr} = 0.018$

Now calculating Reynolds number, Re to know about flow region

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.01}{\pi\times0.025\times1.245\times10^{-3}}$$

$$= 409.07$$

Since is less than 2300, therefore flow is laminar

Step 7 of 7

Hydrodynamic length for laminar flow

$$L_h \approx 0.05\text{Re}\times\text{D}$$

$$\approx 0.05\times409.07\times0.025$$

$$\approx 0.5113\,\text{m}$$

Hence the Hydrodynamic length for mercury is $\boxed{L_h \approx 0.5113\,\text{m}}$

Thermal entry length for laminar flow

$$L_t \approx 0.05\text{Re}\times\text{Pr}\times D$$

$$\approx 0.05\times409.07\times0.0180\times0.025$$

$$\approx 9.204\times10^{-3}\,\text{m}$$

Hence the thermal entry length for mercury is $\boxed{L_t \approx 9.204\times10^{-3}\,\text{m}}$

Question: 19.084P

Water is to be heated from 10°C to 80°C as it flows through a 2-cm-internal-diameter, 13-m-long tube. The tube is equipped with an electric resistance heater, which provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated, so that in steady operation all the heat generated in the heater is transferred to the water in the tube. If the system is to provide hot water at a rate of 5 L/min, determine the power rating of the resistance heater. Also, estimate the inner surface temperature of the pipe at the exit.

Step 1 of 8

Convert water flow rate in the tube from L/min to m^3/s by using conversion factor,

$$\dot{V} = 5\,L/min$$

$$= \frac{5\,L \times 10^{-3}\,\frac{m^3}{L}}{1\,min \times 60\,\frac{s}{min}}$$

$$= 8.33 \times 10^{-5}\,m^3/s$$

Step 2 of 8

Calculate the average temperature of water by using the equation,

$$T_{avg} = \frac{T_i + T_e}{2}$$

Here, T_i is water inlet temperature and T_e is Water outlet or exit temperature

Substitute 10°C for T_i and 80°C for T_e

$$T_{avg} = \frac{10+80}{2}$$

$$= 45°C$$

Take properties of water at average temperature $T_{avg} = 45°C$ from properties of water table.

Density $\rho = 990.1\,kg/m^3$

Specific heat $c_p = 4180\,J/kg\cdot K$

Thermal conductivity $k = 0.637\,W/m\cdot °C$

Step 3 of 8

Calculate mass flow rate of the water

$$\dot{m} = \rho\dot{V}$$

$$= 990.1 \times 8.33 \times 10^{-5}$$

$$= 0.0825\,kg/s$$

Power rating of the resistance heater is equal to the heat transfer rate to the water

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

$$= 0.0825 \times 4180(80-10)$$

$$= 24132.28\,W$$

Therefore, the power rating of the resistance heater $\dot{Q}$ is $\boxed{24132.28\,W}$

Step 4 of 8

Calculate average velocity of the water in the tube

$$V_{avg} = \frac{\dot{V}}{A_c}$$

Here, A_c is cross sectional area of the tube of diameter D

Substitute $\frac{\pi D^2}{4}$ for A_c

$$V_{avg} = \frac{\dot{V}}{\frac{\pi D^2}{4}}$$

$$= \frac{4\dot{V}}{\pi D^2}$$

Substitute 0.02 m for D

$$V_{avg} = \frac{4 \times 8.33 \times 10^{-5}}{\pi \times (0.02)^2}$$

$$= 0.265\,m/s$$

Step 5 of 8

Take properties from saturated water property table at 45°C

Dynamic Viscosity $\mu = 0.596 \times 10^{-3}\,kg/m\cdot s$

Prandtl number $Pr = 3.91$

Kinematic viscosity of water is

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{0.596 \times 10^{-3}}{990.1}$$

$$= 6.02 \times 10^{-7}\,m^2/s$$

+

Step 6 of 8

Calculate the Reynolds number of flow region

$$Re = \frac{V_{avg}D}{\nu}$$

$$= \frac{0.265 \times 0.02}{6.02 \times 10^{-7}}$$

$$= 8804.58$$

Hydraulic and thermal entry lengths are equal in this flow region so that

$$L_h = L_t$$

$$= 10D$$

$$= 10 \times 0.02$$

$$= 0.2\,m$$

Step 7 of 8

The entry length is very less than the given length $L = 13\,m$ so that the flow is a fully developed turbulent flow

Calculate the Nusselt number by using the equation,

$$Nu = 0.023\,Re^{0.8}Pr^{0.4}$$

$$= 0.023 \times (8804.58)^{0.8} \times (3.91)^{0.4}$$

$$= 56.8$$

Find the heat transfer coefficient by using the equation,

$$h = \frac{k}{D}Nu$$

$$= \frac{0.637}{0.02} \times 56.80$$

$$= 1809.15\,W/m^2\cdot °C$$

Step 8 of 8

Calculate heat transferred surface area of the tube is

$$A_s = \pi DL$$

$$= \pi \times 0.02 \times 13$$

$$= 0.8168\,m^2$$

Calculate inner surface temperature, T_s by using heat transfer equation,

$$\dot{Q} = hA_s(T_s - T_e)$$

$$24132.28 = 1809.15 \times 0.8168 \times (T_s - 80)$$

$$T_s = \frac{24132.28}{1809.15 \times 0.8168} + 80$$

$$= 96.33°C$$

Hence the inner surface temperature, T_s at the exit is $\boxed{96.33°C}$

Question: 19.085P

Hot air at atmospheric pressure and 75°C enters a 10-m-long uninsulated square duct of cross section 0.15 m × 0.15 m that passes through the attic of a house at a rate of 0.2 m3/s. The duct is observed to be nearly isothermal at 70°C. Determine the exit temperature of the air and the rate of heat loss from the duct to the air space in the attic.

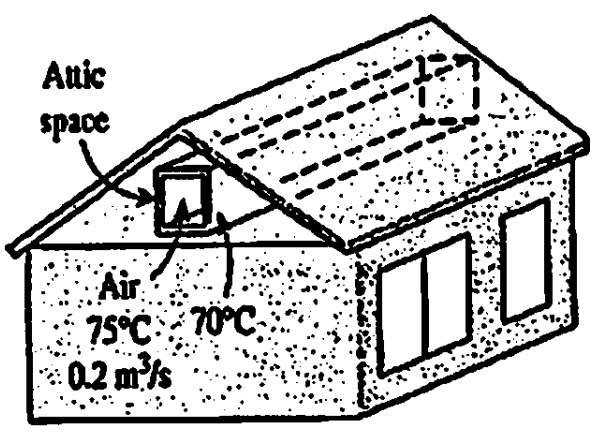


FIGURE PI 9-85

Step 1 of 9

Given data:

Length of square duct $L = 10\text{m}$

Dimensions of duct $H \times W = 0.15 \times 0.15\text{m}$

Inlet temperature of Air $T_i = 75°\text{C}$

Duct temperature $T_s = 70°\text{C}$

Volume flow rate $\dot{V} = 0.2\text{m}^3/s$

Step 2 of 9

The expression for determining the exit temperature of hot air is given by

$$T_e = T_s - (T_s - T_i)\exp\left(-hA/\dot{m}C_p\right)$$

Taking properties of hot air at 75°C the properties of air are

Thermal conductivity of air, $k = 0.02917\,\text{W/m}\cdot°\text{C}$

Density of air, $\rho = 1.0137\,\text{kg/m}^3$

Kinematic viscosity of air, $\nu = 2.046\times10^{-5}\,\text{m}^2/s$

Specific heat of air, $c_p = 1007.5\,\text{J/kg}\cdot\text{K}$

Prandtl number $\text{Pr} = 0.71655$

Step 3 of 9

The hydraulic diameter for the square pipe is

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4\times0.15\times0.15}{4\times0.15}$$
$$= 0.15\text{m}$$

Average velocity of the flow is

$$V_{avg} = \frac{\dot{V}}{A_c}$$
$$= \frac{0.20}{(0.15)^2}$$
$$= 8.889\,\text{m/s}$$

Step 4 of 9

Reynolds number for the flow is given by

$$\text{Re} = \frac{V_{avg}\cdot D_h}{\nu}$$
$$= \frac{8.889\times0.15}{2.046\times10^{-5}}$$
$$= 65167.8$$

Since the Reynolds number is greater than 10,000 , the flow is turbulent

The hydraulic entry length is

$$L_h \approx 10D_h$$
$$= 10\times0.15$$
$$= 1.5\text{m}$$

Step 5 of 9

Hydraulic entry length is much shorter than the given duct length so the flow is a fully developed turbulent flow. Now the Nusselt number is given by

$$N_u = 0.023\text{Re}^{0.8}\text{Pr}^{0.3}$$
$$= 0.023\times(65167.8)^{0.8}\times(0.71655)^{0.3}$$
$$= 147.75$$

Heat transfer coefficient is given by the expression

$$h = \frac{k\cdot N_u}{D_h}$$
$$= \frac{0.02917\times147.75}{0.15}$$
$$= 28.73\,\text{W/m}^2\cdot°\text{C}$$

Step 6 of 9

Surface area of the duct is

$$A_s = 4aL$$
$$= 4\times0.15\times10$$
$$= 6\text{m}^2$$

Mass flow rate of the air is

$$\dot{m} = \rho\dot{V}$$
$$= 1.0137\times0.20$$
$$= 0.20274\,\text{kg/s}$$

Step 7 of 9

Exit temperature of hot air, which can be determined by the relation as

$$T_e = T_s - (T_s - T_i)\exp\left(-hA/\dot{m}C_p\right)$$
$$= 70-(70-75)\exp\left(\frac{-28.73\times6}{0.20274\times1007.5}\right)$$
$$= 72.15°\text{C}$$

Hence Exit temperature of hot air is given by $\boxed{T_e = 72.15°\text{C}}$

Step 8 of 9

Logarithmic mean temperature ΔT_{lm} for the given system is given by

$$\Delta T_{lm} = \frac{T_e - T_i}{\ln\left[\frac{T_s - T_e}{T_s - T_i}\right]}$$
$$= \frac{72.15-75}{\ln\left[\frac{70-72.15}{70-75}\right]}$$
$$= \frac{-2.85}{-0.843}$$
$$= 3.3768°\text{C}$$

Step 9 of 9

Rate of heat loss from the duct to the air space in the attic is given by

$$\dot{Q} = hA_s\Delta T_{lm}$$
$$= 28.73\times6\times3.3768$$
$$= 581.954\,\text{W}$$
$$= 582\,\text{W}$$

Hence the rate of heat loss from the duct is $\boxed{\dot{Q} = 582\,\text{W}}$

Question: 19.086P

Reconsider Prob. 19-85. Using EES (or other) software, investigate the effect of the volume flow rate of air on the exit temperature of air and the rate of heat loss. Let the flow rate vary from 0.05 m3/s to 0.15 m3/s. Plot the exit temperature and the rate of heat loss as a function of flow rate, and discuss the results.

Step 1 of 4

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Crease a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"Given"
L=10[m]                 "Length of the duct"
a=0.15[m]               "Side of the cross-section"
T_i=75[C]               "Inlet temperature"
T_s=70[C]               "Surface temperature"
"V_dot=0.2[m^3/s]"      "Volumetric flow rate"
P_1=101.3[kPa]          "assume atmospheric pressure"

"Properties"
k=Conductivity(Air,T=T_i)              "thermal conductivity"
rho=Density(Air,T=T_i,P=P_1)           "Density"
mu=Viscosity(Air,T=T_i)                "kinematic viscosity"
cp=SpecHeat(Air,T=T_i)*1000            "Specific Heat in J/kg-K"
Pr=Prandtl(Air,T=T_i)                  "Prandtl number"
```

Step 2 of 4

```
"Analysis"
D_h=4*A_c/p                            "hydraulic diameter"
A_c=a^2                                "cross-sectional area"
p=4*a                                  "perimeter"
V_avg=V_dot/A_c                        "Average velocity"
Re= V_avg*D_h* rho/ mu                 "Reynolds number"
L_h=10*D_h                             "Charecteristic length"
N_u=0.023*(Re^0.8)*(Pr^0.3)            "Nusselt number"
h=k*N_u/D_h                            "Heat transfer coefficient"
A_s=4*a*L                              "Surface area"
m_dot=rho*V_dot                        "Mass flow rate"
T_e=T_s-((T_s-T_i)*(exp(-h*A_s/(m_dot*cp))))
                                       "Exit temperature"
Delta_lm=(T_e-T_i)/(ln((T_s-T_e)/(T_s-T_i)))
                                       "log mean temperature difference"
Q_dot=h*A_s*Delta_lm                   "Heat loss"
```

Step 3 of 4

Now to solve the equation for a range of volume flow rate values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 11

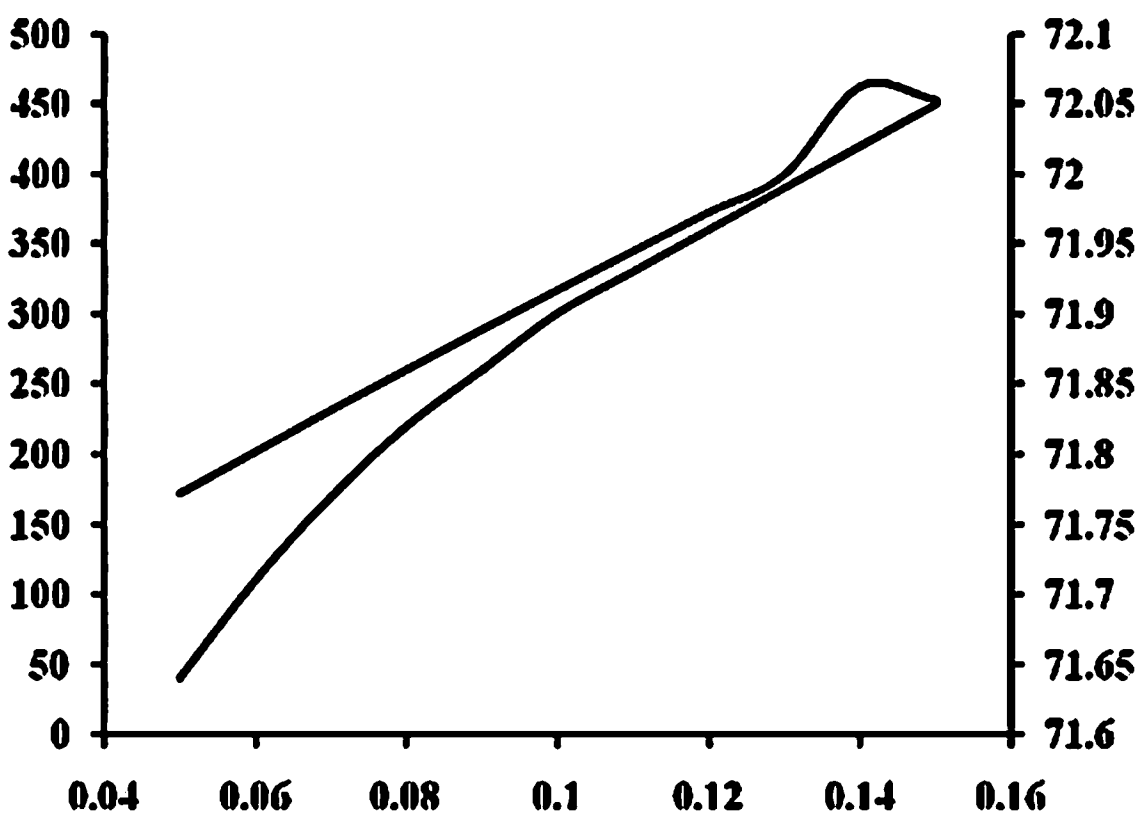
Select variables in equations as $\dot{V}$, T_e and $\dot{Q}$ press 'Add' and then press 'ok'

A table will appear and enter the values of volume flow rate '$\dot{V}$' as 0.05,0.06....0.15 in runs 1 to 11. Then click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

1..11	1 $\dot{V}$ [m3/s]	2 $\dot{Q}$ [W]	3 T_e [C]
Run 1	0.05	171.5	71.64
Run 2	0.06	201.7	71.71
Run 3	0.07	231.3	71.77
Run 4	0.08	260.3	71.82
Run 5	0.09	288.9	71.86
Run 6	0.1	317	71.9
Run 7	0.11	344.8	71.93
Run 8	0.12	372.2	71.96
Run 9	0.13	399.3	71.99
Run 10	0.14	426.2	72.02
Run 11	0.15	452.7	72.05

Step 4 of 4

Inorder to find the variation of exit temperature and heat loss with volumetric flow rate, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as $\dot{V}$ and Y-axis as $\dot{Q}$ and T_e. Press OK
The Graph will be plotted as shown below.



This graph represents that by increasing the volumetric flow rate, the temperature and heat loss increases.

Question: 19.087P

Consider a fluid with a Prandtl number of 7 flowing through a smooth circular tube. Using the Colburn, Petukhov, and Gnielinski equations, determine the Nusselt numbers for Reynolds numbers at 3500, 104, and 5 × 105.Compare and discuss the results.

Step 1 of 8

Given data:
A fluid flowing through a smooth circular tube
Fluid Prandtl number $\mathrm{Pr} = 7$
The Reynolds numbers are
$\mathrm{Re}_1 = 3500$
$\mathrm{Re}_2 = 10^4$
$\mathrm{Re}_3 = 5\times10^5$

Step 2 of 8

The Colburn equation is given as

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{1/3} \quad \ldots\ldots (1)$$

The Petukhov equation is given as

$$Nu = \frac{(f/8)\mathrm{Re}\cdot\mathrm{Pr}}{1.07+12.7(f/8)^{0.5}\left(\mathrm{Pr}^{2/3}-1\right)} \quad \ldots\ldots (2)$$

Where $f = (0.790\ln\mathrm{Re}-1.64)^{-2}$ ..... (3)

The Gnielinski equation is given as

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{0.5}\left(\mathrm{Pr}^{2/3}-1\right)} \quad \ldots\ldots (4)$$

Step 3 of 8

For $\mathrm{Re}_1 = 3500$ the equation (1) becomes,

$$Nu = 0.023(3500)^{0.8}(7)^{1/3}$$
$$= 30.108$$

Hence the Nusselt number as per Colburn equation is $\boxed{Nu = 30.108}$

For $\mathrm{Re}_1 = 3500$ the equation (3) becomes,

$$f = (0.79\ln 3500 - 1.64)^{-2}$$
$$= 0.04328$$

Step 4 of 8

Thus the Nusselt number according to equation (2) is given by

$$Nu = \frac{(0.04328/8)\times3500\times7}{1.07+12.7(0.04328/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{132.545}{3.554}$$
$$= 37.29$$

Hence the Nusselt number as per Petukhov equation is $\boxed{Nu = 37.29}$

For $\mathrm{Re}_1 = 3500$ the equation (4) becomes,

$$Nu = \frac{(0.04328/8)(3500-1000)\times7}{1+12.7(0.04328/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{94.675}{3.484}$$
$$= 27.174$$

Hence the Nusselt number as per Gnielinski equation is $\boxed{Nu = 27.174}$

Step 5 of 8

For $\mathrm{Re}_2 = 10^4$ the equation (1) becomes,

$$Nu = 0.023\left(10^4\right)^{0.8}(7)^{1/3}$$
$$= 69.73$$

Hence the Nusselt number as per Colburn equation is $\boxed{Nu = 69.37}$

For $\mathrm{Re}_2 = 10^4$ the equation (3) becomes,

$$f = \left[0.79\ln\left(10^4\right)-1.64\right]^{-2}$$
$$= 0.03148$$

Step 6 of 8

Thus the Nusselt number according to equation (2) is given by

$$Nu = \frac{(0.03148/8)\times10^4\times7}{1.07+12.7(0.03148/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{275.45}{3.1886}$$
$$= 86.48$$

Hence the Nusselt number as per Petukhov equation is $\boxed{Nu = 86.48}$

For $\mathrm{Re}_2 = 10^4$ the equation (4) becomes,

$$Nu = \frac{(0.03148/8)\left(10^4-1000\right)\times7}{1+12.7(0.03148/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{247.905}{3.1186}$$
$$= 79.49$$

Hence the Nusselt number as per Gnielinski equation is $\boxed{Nu = 79.49}$

Step 7 of 8

For $\mathrm{Re}_3 = 5\times10^5$ the equation (1) becomes,

$$Nu = 0.023\left(5\times10^5\right)^{0.8}(7)^{1/3}$$
$$= 1594$$

Hence the Nusselt number as per Colburn equation is $\boxed{Nu = 1594}$

For $\mathrm{Re}_3 = 5\times10^5$ the equation (3) becomes,

$$f = \left[0.79\ln\left(5\times10^5\right)-1.64\right]^{-2}$$
$$= 0.01313$$

Step 8 of 8

Thus the Nusselt number according to equation (2) is given by

$$Nu = \frac{(0.01313/8)\times\left(5\times10^5\right)\times7}{1.07+12.7(0.01313/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{5744.375}{2.438}$$
$$= 2356$$

Hence the Nusselt number as per Petukhov equation is $\boxed{Nu = 2356}$

For $\mathrm{Re}_3 = 5\times10^5$ the equation (4) becomes,

$$Nu = \frac{(0.01313/8)\left(5\times10^5-1000\right)\times7}{1+12.7(0.01313/8)^{0.5}\left(7^{2/3}-1\right)}$$
$$= \frac{5732.88}{2.368}$$
$$= 2421$$

Hence the Nusselt number as per Gnielinski equation is $\boxed{Nu = 2421}$

Question: 19.088P

A concentric annulus tube has inner and outer diameters of 25 mm and 100 mm, respectively. Liquid water flows at a mass flow rate of 0.05 kg/s through the annulus with the inlet and outlet mean temperatures of 20°C and 80°C, respectively. The inner tube wall is maintained with a constant surface temperature of 120°C, while the outer tube surface is insulated. Determine the length of the concentric annulus tube.

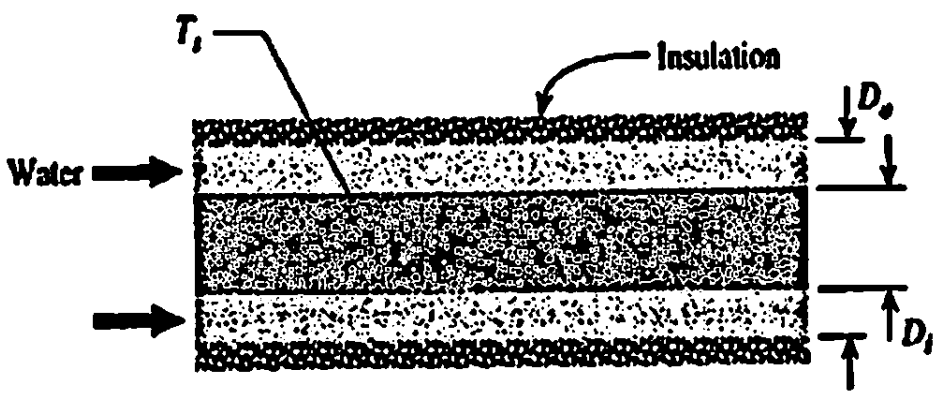


FIGURE P19-88

Step 1 of 5

Calculate the average temperature of water

$$T_{avg} = \frac{T_e + T_i}{2}$$

Here, T_e is the outer temperature and T_i is the initially temperature.

Substitute $20°C$ for T_e and $80°C$ for T_i.

$$\begin{aligned} T_{avg} &= \frac{T_e + T_i}{2} \\ &= \frac{20+80}{2} \\ &= 50°C \end{aligned}$$

Step 2 of 5

From the properties of water, the average temperature is $50°C$.

Thermal conductional, $k = 0.644\ W/m \cdot °C$

Dynamic viscosity, $\mu = 0.547 \times 10^{-3}\ kg/m \cdot s$

Calculate the Reynolds number.

$$R_e = \frac{4\dot{m}}{\pi(D_o - D_i)\mu}$$

Here, $\dot{m}$ is the mass flow rate, μ is the dynamic viscosity, D_o is the outer diameter of the pipe and D_i is the inner diameter of the pipe

Substitute $0.05\ m/s$ for $\dot{m}$, $0.1\ m$ for D_o and $0.025\ m$ for D_i

$$\begin{aligned} R_e &= \frac{4\dot{m}}{\pi(D_o - D_i)\mu} \\ &= \frac{4 \times 0.05}{\pi(0.1 - 0.025)\left(0.547 \times 10^{-3}\right)} \\ &= 931 \end{aligned}$$

Step 3 of 5

Reynolds number is less than $2,300$, so flow is laminar and assuming it is a fully developed flow.

Write the ratio of diameters.

$$\frac{D_i}{D_o} = 0.25$$

Take Nusselt number Nu_i, from key and Perkins table at the ratio of 0.25.

$$Nu_i = 7.37$$

Calculate the heat transfer coefficient

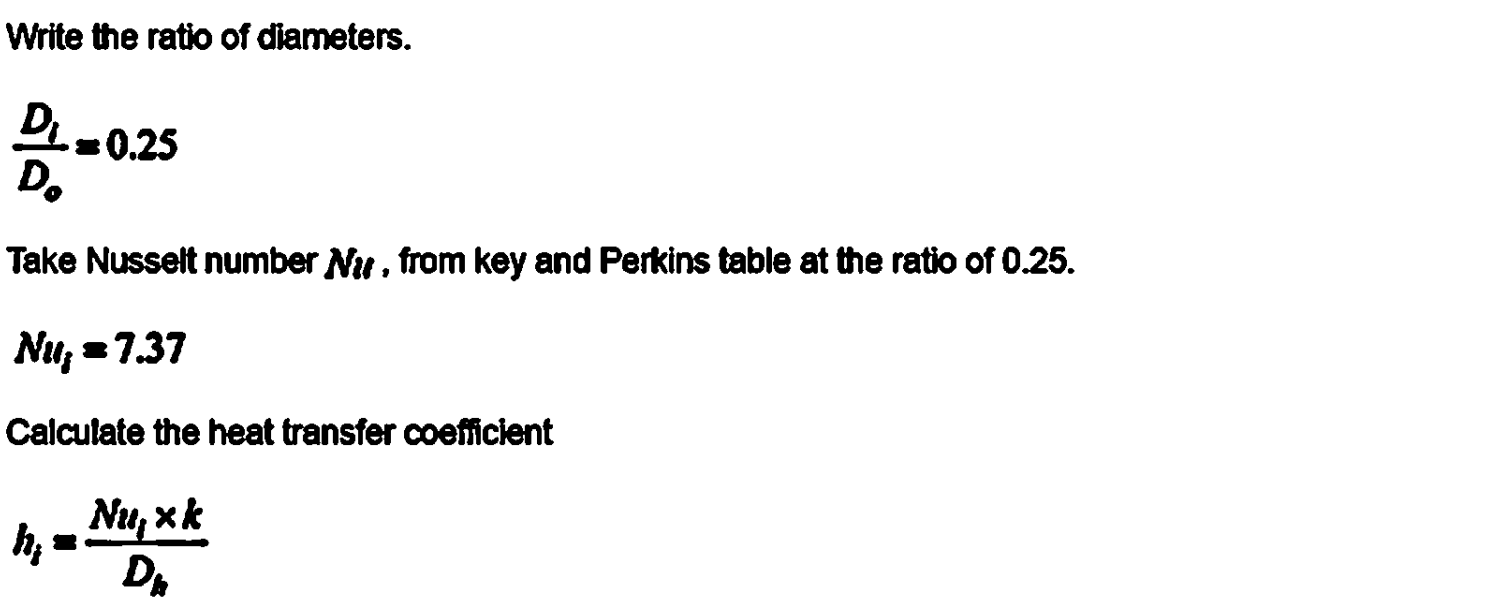
$$h_i = \frac{Nu_i \times k}{D_h}$$

Substitute 7.37 for Nu_i, $0.644\ W/m \cdot K$ for k

$$\begin{aligned} h_i &= \frac{Nu_i \times k}{D_h} \\ &= \frac{7.37 \times 0.644}{0.1 - 0.025} \\ &= \frac{7.37 \times 0.644}{0.075} \\ &= 63.28\ W/m^2 \cdot K \end{aligned}$$

Step 4 of 5

The given figure is shown as follows:



Step 5 of 5

Calculate the Logarithmic mean temperature of the annulus tube.

$$\begin{aligned} \Delta T_{lm} &= \frac{T_i - T_e}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \\ &= \frac{20-80}{\ln\left(\frac{120-80}{120-20}\right)} \\ &= \frac{-60}{\ln(0.4)} \\ &= 65.48°C \end{aligned}$$

Calculate the heat transfer in the system

$$\dot{Q} = h A_S \Delta T_m$$

Here, A_s is the surface area of the tube.

Substitute $65.48°C$ for ΔT_m

$$\begin{aligned} \dot{Q} &= h A_S \Delta T_m \\ &= 63.28 \times (\pi D_i L) \times 65.48 \\ &= 63.28 \times \pi \times 0.025L \times 65.48 \\ &= 325.44L \end{aligned}$$

The heat transfer is equal to the heat gained by water. Hence, obtain as follows: $\dot{Q} = \dot{m} C_p (T_e - T_i)$

$$\begin{aligned} 325.44L &= 0.05 \times 4181 \times (80-20) \\ L &= \frac{0.05 \times 4181 \times 60}{325.44} \\ &= 38.54\ m \end{aligned}$$

Hence, the length of the concrete annulus tube is $\boxed{38.54\ m}$.

Question: 19.089P pg 1

The hot water needs of a household are to be met by heating water at 55°F to 180°F by a parabolic solar collector at a rate of 5 lbm/s. Water flows through a 1.25-in-diame-ter thin aluminum tube whose outer surface is blackanodized in order to maximize its solar absorption ability. The center-line of the tube coincides with the focal line of the collector, and a glass sleeve is placed outside the tube to minimize the heat losses. If solar energy is transferred to water at a net rate of 350 Btu/h per ft length of the tube, determine the required length of the parabolic collector to meet the hot water requirements of this house. Also, determine the surface temperature of the tube at the exit.

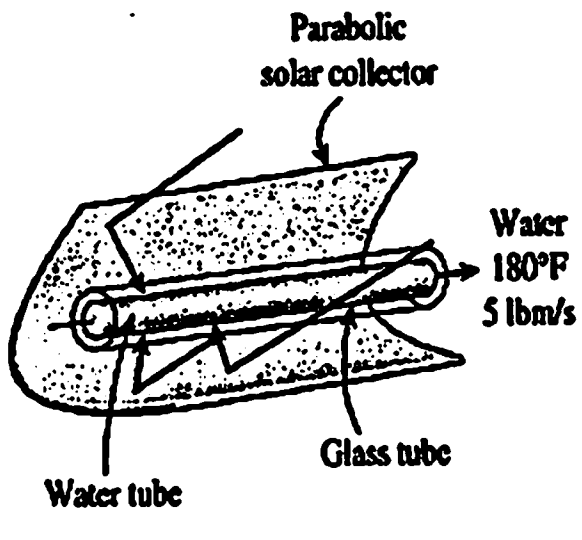


FIGURE P19-89E

Step 1 of 8

Given data:

Inlet water temperature $T_i = 55°F$

Outlet water temperature $T_e = 180°F$

Mass flow rate $\dot{m} = 5 lbm/s$

Diameter of aluminum tube $D = 1.25 in$

Solar energy transferred $\dot{Q} = 350 Btu/h \cdot ft$

Step 2 of 8

Average temperature of the water,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= (180 + 55)/2$$
$$= 117.5°F$$

Properties of water at an average temperature,

Density $\rho = 61.75 lbm/ft^3$

Thermal conductivity $k = 0.370 Btu/ft \cdot °F$

Specific heat $C_p = 0.999 Btu/lbm \cdot °F$

Prandtl number $Pr = 3.75$

Step 3 of 8

Kinematic viscosity is

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{3.84 \times 10^{-4}}{61.75}$$
$$= 6.219 \times 10^{-6} m^2/s$$

Now, the total rate of heat transfer can be expressed as

$$\dot{Q}_t = \dot{m} C_p (T_e - T_i)$$
$$= 5 \times 0.999 (180 - 55)$$
$$= 624.375 Btu/s$$
$$= 2.25 \times 10^6 Btu/h$$

Step 4 of 8

In steady condition, the heat transferred to water is equal to the net rate per feet length so that

$$\dot{Q} = 350 Btu/h \cdot ft$$

The expression for length is

$$L = \frac{\dot{Q}_t}{\dot{Q}}$$
$$= \frac{2.25 \times 10^6}{350}$$
$$= 6422.14 ft$$

Required length of the parabolic collector is $\boxed{L = 6422.14 ft}$

Step 5 of 8

Surface temperature of the tube at the exit:

Average velocity of the water is

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$
$$= \frac{\dot{m} \times 4}{\rho \times \pi D^2}$$
$$= \frac{5 \times 4}{61.75 \times \pi \times (0.1042)^2}$$
$$= 9.495 ft/s$$

Step 6 of 8

The hydraulic diameter is

$$D_h = D$$
$$= 0.1042$$

Reynolds number of the flow in the tube

$$Re = \frac{V_{avg} \cdot D_h}{\nu}$$
$$= \frac{9.495 \times 0.1042}{6.219 \times 10^{-6}}$$

Reynolds number of the flow in the tube

$$\mathrm{Re} = \frac{V_{avg} \cdot D_h}{\nu}$$
$$= \frac{9.495 \times 0.1042}{6.219 \times 10^{-6}}$$
$$= 159089.4$$

Since it is much larger than 10,000 , it's a fully turbulent flow so entry length can be

$$L_h \approx L_t$$
$$\approx 10\mathrm{D}$$
$$\approx 10 \times 0.1042$$
$$\approx 1.042\,\mathrm{ft}$$

Step 7 of 8

The above entry length is much shorter than the calculated length, therefore fully developed turbulent flow

Nusselt number is

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$$
$$= 0.023 \times 159089.4^{0.8} \times 3.75^{0.4}$$
$$= 565.78$$

Therefore heat transfer coefficient can be

$$h = \frac{k \cdot Nu}{D_h}$$
$$= \frac{0.370 \times 565.8}{0.1042}$$
$$= 2009\,\mathrm{Btu/h \cdot ft^2 \cdot {}^\circ F}$$

Step 8 of 8

Total Heat flux on the tube is

$$\dot{q}_s = \frac{\dot{Q}_t}{A_s}$$
$$= \frac{2.25 \times 10^6}{\pi(0.1042) \times 6422}$$
$$= 1070.3\,\mathrm{Btu/h \cdot ft^2}$$

The heat flux on the tube can be written as follows

$$\dot{q}_s = h(T_s - T_e)$$

$$T_s = \frac{\dot{q}_s}{h} + T_e$$
$$= \frac{1070.3}{2009} + 180$$
$$= 180.53^\circ\mathrm{C}$$

Hence the surface temperature of the at the exit is given by $\boxed{T_s = 180.53^\circ\mathrm{C}}$

Question: 19.090P

Determine the average velocity, hydrodynamic and thermal entry lengths for water, engine oil, and liquid mercury flowing through a standard 2-in Schedule 40 pipe with mass flow rate of 0.1 lbm/s and temperature of 100°F.

Step 1 of 8

Given data:

Mass flow rate, $\dot{m} = 0.1\,\text{lbm/s}$

Inlet temperature, $T_i = 100°\text{F}$

Pipe considered is 'Standard 2-in schedule 40 pipe'

Step 2 of 8

The diameter for nominal size of given pipe is $D = 2.067$ in

Area of pipe cross section,

$$A_c = \frac{\pi D^2}{4}$$

$$= \frac{\pi(2.067/12)^2}{4}$$

$$= 0.0233\,\text{ft}^2$$

From the properties table of water at 100°F,

Density, $\rho = 62\,\text{lbm/ft}^3$

Dynamic viscosity, $\mu = 4.578\times10^{-4}\,\text{lbm/ft-s}$

Prandtl number, $\text{Pr} = 4.54$

Step 3 of 8

Average velocity of flow,

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$

$$= \frac{0.1}{62\times0.0233}$$

$$= 0.0692\,\text{ft/s}$$

Therefore the average velocity of flow is $\boxed{V_{avg} = 0.0692\,\text{ft/s}}$

Reynolds number,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.1}{\pi\times0.17225\times4.578\times10^{-4}}$$

$$= 1614.64$$

Step 4 of 8

As the Reynolds number is less than 2300, the flow is laminar flow. Hence

$$L_h = 0.05\times\text{Re}\times D$$

$$= 0.05\times1614.64\times0.17225$$

$$= 13.9\,\text{ft}$$

Hence the hydrodynamic entry length is $\boxed{L_h = 13.9\,\text{ft}}$

Now the thermal entry length,

$$L_t = \text{Pr}\times L_h$$

$$= 4.54\times13.90$$

$$= 63.13\,\text{ft}$$

Hence the thermal entry length is $\boxed{L_t = 63.13\,\text{ft}}$

Step 5 of 8

From the properties table of Engine oil at 100°F,

Density, $\rho = 54.77\,\text{lbm/ft}^3$

Dynamic viscosity, $\mu = 163\times10^{-3}\,\text{lbm/ft-s}$

Prandtl number, $\text{Pr} = 3275$

Average velocity of flow,

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$

$$= \frac{0.1}{54.77\times0.0233}$$

$$= 0.07836\,\text{ft/s}$$

Therefore the average velocity of flow is $\boxed{V_{avg} = 0.07836\,\text{ft/s}}$

Step 6 of 8

Reynolds number is determined by

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.1}{\pi\times0.17225\times163\times10^{-3}}$$

$$= 4.535$$

As the Reynolds number is less than 2300, the flow is laminar flow. Hence

$$L_h = 0.05\times\text{Re}\times\text{D}$$

$$= 0.05\times4.535\times0.17225$$

$$= 0.0391\,\text{ft}$$

Hence the hydrodynamic entry length is $\boxed{L_h = 0.0391\,\text{ft}}$

Now the thermal entry length,

$$L_t = L_h\times\text{Pr}$$

$$= 0.0391\times3275$$

$$= 127.91\,\text{ft}$$

Hence the thermal entry length is $\boxed{L_t = 127.91\,\text{ft}}$

Step 7 of 8

From the properties table of liquid mercury at 100°F,

Density, $\rho = 842.9\,\text{lbm/ft}^3$

Dynamic viscosity, $\mu = 9.919\times10^{-4}\,\text{lbm/ft.s}$

Prandtl number, $\text{Pr} = 0.02363$

Average velocity of flow,

$$V_{avg} = \frac{\dot{m}}{\rho A_C}$$

$$= \frac{0.1}{842.9\times0.0233}$$

$$= 5.09\times10^{-3}\,\text{ft/s}$$

Therefore the average velocity of flow is $\boxed{V_{avg} = 5.09\times10^{-3}\,\text{ft/s}}$

Step 8 of 8

Reynolds number,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.1}{\pi\times0.17225\times9.919\times10^{-4}}$$

$$= 745.22$$

As the Reynolds number is less than 2300, the flow is laminar flow. Hence

$$L_h = 0.05\times\text{Re}\times D$$

$$= 0.05\times745.22\times0.17225$$

$$= 6.418\,\text{ft}$$

Hence the hydrodynamic entry length is $\boxed{L_h = 6.418\,\text{ft}}$

Now the thermal entry length,

$$L_t = \text{Pr}\times L_h$$

$$= 6.418\times0.02363$$

$$= 0.152\,\text{ft}$$

Hence the thermal entry length is $\boxed{L_t = 0.152\,\text{ft}}$

Question: 19.091P

In a manufacturing plant that produces cosmetic products, glycerin is being heated by flowing through a 25-mm-diamcter and 10-m long tube. With a mass flow rate of 0.5 kg/s, the flow of glycerin enters the tube at 25°C. The tube surface is maintained at a constant surface temperature of 140°C. Determine the outlet mean temperature and the total rate of heat transfer for the tube. Evaluate the properties for glycerin at 30°C.

Step 1 of 6

Given data:

Diameter of the tube, $D = 0.025$ m

Length of the tube, $L = 10$ m

Mass flow rate, $\dot{m} = 0.5$ kg/s

Glycerin inlet temperature, $T_i = 25°C$

Tube surface temperature, $T_s = 140°C$

Step 2 of 6

The expression for the outlet temperature T_o is

$$T_o = T_s - (T_s - T_i)\exp\left[-\frac{hA_s}{\dot{m}c_p}\right]$$

where h - Heat transfer coefficient

A_s - Heat transfer surface area

$\dot{m}$ - Mass flow rate

c_p - Specific heat at constant pressure

Step 3 of 6

Properties of glycerin at 30°C:

Density, $\rho = 1258$ kg/m^3

Specific heat at constant pressure, $c_p = 2447$ J/kg·K

Thermal conductivity, $k = 0.2860$ W/m·K

Dynamic viscosity, $\mu = 0.6582$ kg/m·s

Prandtl number, $\Pr = 5631$

Now Reynolds number can be calculated using the equation,

$$\begin{aligned}\mathrm{Re} &= \frac{\rho VD}{\mu}\\ &= \frac{\rho \dot{v} D}{A\mu}\\ &= \frac{4\dot{m}}{\pi D\mu}\\ &= \frac{4\times 0.5}{\pi\times 0.025\times 0.6582}\\ &= 38.69\end{aligned}$$

Since the value is less than 2300, the flow is laminar.

Step 4 of 6

Now, the hydrodynamic entry length is given by,

$$\begin{aligned}L_h &= 0.05\mathrm{Re}.D\\ &= 0.05\times 38.69\times 0.025\\ &= 0.04836 \text{ m}\end{aligned}$$

Thermal entry length is given by,

$$\begin{aligned}L_t &= L_h \times \Pr\\ &= 0.04836\times 5631\\ &= 272.32 \text{ m}\end{aligned}$$

As the thermal entry length is greater than the given tube length, the flow is not thermally developed but developing. Therefore, the average Nusselt number for the thermal entrance region can be determined using the following equation.

$$\begin{aligned}\mathrm{Nu} &= 3.66 + \frac{0.065(D/L)\mathrm{Re}\cdot\Pr}{1+0.04\left[(D/L)\cdot\mathrm{Re}\cdot\Pr\right]^{2/3}}\\ &= 3.66 + \frac{0.065(0.025/10)(38.69)(5631)}{1+0.04\left[(0.025/10)(38.69)(5631)\right]^{2/3}}\\ &= 3.66 + \frac{35.4}{1+2.6677}\\ &= 13.312\end{aligned}$$

Step 5 of 6

The convective heat transfer coefficient,

$$\begin{aligned}h &= \frac{k\cdot\mathrm{Nu}}{D}\\ &= \frac{0.2860\times 13.312}{0.025}\\ &= 152.29 \text{ W/m}^2\cdot\text{K}\end{aligned}$$

Heat transfer surface area,

$$\begin{aligned}A_s &= \pi DL\\ &= \pi\times 0.025\times 10\\ &= 0.7854 \text{ m}^2\end{aligned}$$

The mean temperature at outlet,

$$\begin{aligned}T_o &= 140 - (140-25)\times\exp\left[-\frac{152.29\times 0.7854}{0.5\times 2447}\right]\\ &= 140 - (115\times 0.907)\\ &= 35.7°C\end{aligned}$$

Hence the mean temperature at outlet of the tube, $\boxed{T_o = 35.7°C}$

Step 6 of 6

The total rate of heat transfer for the tube,

$$\begin{aligned}\dot{Q} &= \dot{m}c_p(T_o - T_i)\\ &= 0.5\times 2447\times(35.71-25)\\ &= 0.5\times 2447\times 10.7\\ &= 13091.45 \text{ W}\\ &= 13.1 \text{ kW}\end{aligned}$$

Therefore, the total rate of heat transfer for the tube, $\boxed{\dot{Q} = 13.1 \text{ kW}}$

Question: 19.092P

Liquid glycerin is flowing through a 25-mm-diametcr and 10-m-long tube. The liquid glycerin enters the tube at 20°C with a mass flow rate of 0.5 kg/s. If the outlet mean temperature is 40°C and the tube surface temperature is constant, determine the surface temperature of the tube.

Step 1 of 6

Calculate the bulk mean temperature.

$$T_b = \frac{T_i + T_e}{2}$$

Here, T_i is the inlet temperature, and T_e is the outlet mean temperature.

Substitute $20°C$ for T_i and $40°C$ for T_e.

$$T_b = \frac{20+40}{2}$$
$$= 30°C$$

Obtain the properties of glycerin at $30°C$ from the table A-13, "Properties of liquids" in the textbook.

$$\rho = 1258\,kg/m^3$$
$$c_p = 2447\,J/kg \cdot K$$
$$k = 0.286\ W/m \cdot K$$
$$\mu = 0.6582\ kg/m \cdot s$$
$$Pr = 5631$$

Step 2 of 6

Calculate the Reynolds number.

$$Re = \frac{4\dot{m}}{\pi D \mu}$$

Here, $\dot{m}$ is the mass flow rate, and D is the diameter of the tube.

Substitute $0.5\ kg/s$ for $\dot{m}$, 25 mm for *D*, and $0.6582\ kg/m \cdot s$ for μ.

$$Re = \frac{4 \times 0.5}{\pi \times 25 \times 10^{-3} \times 0.6582}$$
$$= 38.68$$

As the Reynolds number is less than 2300, the flow is a laminar flow.

Step 3 of 6

Calculate the hydrodynamic entry length.

$$L_h = 0.05 \times Re \times D$$

Substitute 38.68 for Re and 0.025 m for *D*.

$$L_h = 0.05 \times 38.68 \times 0.025$$
$$= 0.04835\ m$$

The hydrodynamic entry length is shorter then given length, L. So the flow is hydrodynamically developed.

Now thermal entry length,

$$L_t = L_h \times Pr$$

Substitute 0.04835 m for L_h and 5631 for Pr.

$$L_t = 0.04835 \times 5631$$
$$= 272.25\,m$$

Step 4 of 6

As the thermal entry length is greater than the given length, the flow is not thermally developed but developing.

Calculate the average Nusselt's number.

$$Nu = 3.66 + \frac{0.065(D/L)Re \cdot Pr}{1 + 0.04\left[(D/L) \cdot Re \cdot Pr\right]^{2/3}}$$

Substitute 0.025 m for *D*, 10 m or *L*, 38.68 for Re, and 5631 for Pr.

$$Nu = 3.66 + \frac{0.065(0.025/10)(38.69)(5631)}{1 + 0.04\left[(0.025/10)(38.69)(5631)\right]^{2/3}}$$
$$= 13.312$$

Calculate the heat transfer coefficient.

$$h = \frac{k \times Nu}{D}$$

Substitute 13.312 for Nu, $0.286\ W/m \cdot K$ for *k*, and 0.025 m for *D*.

$$h = \frac{0.2860 \times 13.312}{0.025}$$
$$= 152.29\,W/m^2\,K$$

Step 5 of 6

Calculate the heat transfer surface area.

$$A_s = \pi DL$$

Substitute 0.025 m for *D* and 10 m or *L*.

$$A_s = \pi \times 0.025 \times 10$$
$$= 0.7854\ m^2$$

Step 6 of 6

Calculate the exit temperature relation as follows:

$$T_e = T_s - (T_s - T_e)\exp\left[-\frac{hA_s}{\dot{m}c_p}\right]$$

Substitute $40°C$ for T_e, $152.29\ W/m^2 \cdot K$ for *h*, $0.7854\ m^2$ for A_s, $0.5\ kg/s$ for $\dot{m}$, and $2447\,J/kg \cdot K$ for c_p.

$$40 = T_s - (T_s - 25)\exp\left[-\frac{152.29 \times 0.7854}{0.5 \times 2447}\right]$$
$$T_s = 40 + 0.907(T_s - 25)$$
$$T_s = 40 + 0.907T_s - 22.675$$
$$= 186.3°C$$

Hence, the surface temperature of the tube is $\boxed{T_s = 234.75°C}$

Question: 19.093P

A 15-cm ×20-cm printed circuit board whose components are not allowed to come into direct contact with air for reliability reasons is to be cooled by passing cool air through a 20- cm-long channel of rectangular cross section 0.2 cm × 14 cm drilled into the board. The heat generated by the electronic components is conducted across the thin layer of the board to the channel, where it is removed by air that enters the channel at 15°C. The heat flux at the top surface of the channel can be considered to be uniform, and heat transfer through other surfaces is negligible. If the velocity of the air at the inlet of the channel is not to exceed 4 m/s and the surface temperature of the channel is to remain under 50°C, determine the maximum total power of the electronic components that can safely be mounted on this circuit board.

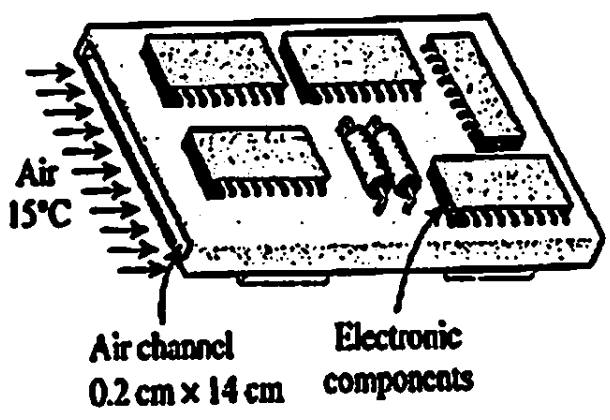


FIGURE P19-93

Step 1 of 6

Calculate the Hydraulic diameter of the channel cross section,

$$D_h = \frac{4A_c}{p}$$

Here, A_c is channel cross section and p is perimeter of the cross section

Substitute $0.002\,\text{m} \times 0.14\,\text{m}$ for A_c and $0.002\,\text{m} + 0.14\,\text{m}$ for p

$$D_h = \frac{4\times(0.002\times0.14)}{2(0.002+0.14)}$$

$$= 3.94\times10^{-3}\,\text{m}$$

Assume average temperature of air is at $30\,^\circ\text{C}$ and take properties of air at average temperature,

Density, $\rho = 1.164\ \text{kg/m}^3$

Thermal conductivity, $k = 0.02588\ \text{W/m}\cdot\text{K}$

Specific heat, $c_p = 1007\ \text{J/kg}\cdot\text{K}$

Kinematic viscosity, $v = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

Step 2 of 6

The Reynolds number for flow is

$$\text{Re} = \frac{V_{avg} D_h}{v}$$

$$= \frac{4\times3.94\times10^{-3}}{1.608\times10^{-5}}$$

$$= 980$$

Since the Reynolds number, is less than 2300, the flow is laminar flow.

Calculate the hydrodynamic entry length by using the equation,

$$L_h = 0.05\times\text{Re}\times D_h$$

$$= 0.05\times980\times3.94\times10^{-3}$$

$$= 0.193\ \text{m}$$

This is less than the length of PCB; $L = 20\,\text{cm} = 0.2\,\text{m}$ hence the flow is fully developed laminar flow.

Step 3 of 6

Calculate the ratio of the channel cross section,

$$\frac{a}{b} = \frac{0.14}{0.002}$$

$$= 70$$

From the table of Nusselt number for different cross-sections, for $a/b = 70$ the Nusselt number is,

$$Nu = 8.24$$

Find the heat transfer coefficient,

$$h = \frac{k\times Nu}{D_h}$$

$$= \frac{0.02588\times8.24}{3.94\times10^{-3}}$$

$$= 54.12\ \text{W/m}^2\,^\circ\text{C}$$

Calculate the mass flow rate by using the equation,

$$\dot{m} = \rho V A_c$$

Here, Velocity of inlet air, V

Substitute $4\,\text{m/s}$ for V

$$\dot{m} = 1.164\times4\times(0.002\times0.14)$$

$$= 1.304\times10^{-3}\ \text{kg/s}$$

Step 4 of 6

Calculate the rate of heat transfer due to convection by using the equation,

$$\dot{Q} = hA_s(T_s - T_e)$$

Here, surface temperature of channel is T_s

Substitute $50\,^\circ\text{C}$ for T_s

$$\dot{Q} = (54.12)(0.14\times0.2)(50 - T_e)$$

$$= 1.515(50 - T_e)$$

Calculate the rate of heat transfer by using another equation,

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

Here, Inlet temperature of air, T_i

Substitute $15\,^\circ\text{C}$ for T_i

$$\dot{Q} = 1.304\times10^{-3}\times1007\times(T_e - 15)$$

$$= 1.313(T_e - 15)$$

Step 5 of 6

By equating the heat transfer rates,

$$1.515(50 - T_e) = 1.313(T_e - 15)$$

$$75.75 - 1.515T_e = 1.313T_e - 19.695$$

$$2.828T_e = 95.445$$

$$T_e = 33.75\,^\circ\text{C}$$

Step 6 of 6

Calculate heat transfer rate by using the equation,

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

$$= 1.304\times10^{-3}\times1007(33.75 - 15)$$

$$= 24.62\ \text{W}$$

Therefore, the total power, $\dot{Q}$ can be mounted on PCB is $\boxed{24.62\ \text{W}}$

Question: 19.094P

Repeat Prob. 19-93 by replacing air with helium, which has six times the thermal conductivity of air.

Step 1 of 6

Given data:

Length of PCB, $L = 20\text{cm} = 0.2\text{m}$

Width of PCB, $W = 15\text{cm} = 0.15\text{m}$

Channel cross section dimension is $= 0.002\text{m} \times 0.14\text{m}$

Inlet temperature of helium, $T_i = 15^\circ\text{C}$

Velocity of inlet helium, $V = 4\text{m/s}$

Surface temperature of channel, $T_s = 50^\circ\text{C}$

Step 2 of 6

Hydraulic diameter

$$D_h = \frac{4A_c}{p}$$

$$= \frac{4\times(0.002\times0.14)}{2(0.002+0.14)}$$

$$= 3.94\times10^{-3}\text{m}$$

From properties table of air at average temperature of 30°C,

Thermal conductivity of air, $k = 0.02588\ \text{W/m-K}$

Specific heat, $c_p = 1007\ \text{J/kg-K}$

Kinematic viscosity, $\nu = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

As Thermal conductivity of Helium is 6 times that of the air, $k = 0.15528\ \text{W/mK}$

Step 3 of 6

The Reynolds number for flow is

$$\text{Re} = \frac{V_{avg} D_h}{\nu}$$

$$= \frac{4\times3.94\times10^{-3}}{1.608\times10^{-5}}$$

$$= 980$$

Since the Reynolds number, is less than 2300, the flow is laminar flow.

Hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D_h$$

$$= 0.05\times980\times3.94\times10^{-3}$$

$$= 0.193\,\text{m}$$

This is less than the length of PCB hence the flow is fully developed laminar flow.

Step 4 of 6

From the table of Nusselt number for different cross-sections, for $a/v = 70$ the Nusselt number is,

$$Nu = 8.24$$

Now heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D_h}$$

$$= \frac{0.15528\times8.24}{3.94\times10^{-3}}$$

$$= 324.72\ \text{W/m}^2\text{°C}$$

The mass flow rate,

$$\dot{m} = \rho V A_c$$

$$= 1.164\times4\times2.8\times10^{-4}$$

$$= 1.304\times10^{-3}\ \text{kg/s}$$

Step 5 of 6

The rate of heat transfer due to convection is,

$$\dot{Q} = hA_s(T_s - T_e)$$

$$= (324.72)(0.14\times0.2)(50 - T_e)$$

$$= 9.09(50 - T_e)$$

The rate of heat transfer is also given by,

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

$$= 1.304\times10^{-3}\times1007\times(T_e - 15)$$

$$= 1.313(T_e - 15)$$

Step 6 of 6

By equating the heat transfer rates,

$$9.09(50 - T_e) = 1.313(T_e - 15)$$

$$454.5 - 9.09T_e = 1.313T_e - 19.695$$

$$10.403T_e = 474.195$$

$$T_e = 45.58^0$$

Now heat transfer rate is

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

$$= 1.304\times10^{-3}\times1007(45.58 - 15)$$

$$= 40.155\ \text{W}$$

Therefore the total power that can be mounted on PCB is $\boxed{\dot{Q} = 40.155\ \text{W}}$

Question:

Reconsider Prob. 19-93. Using EES (or other) software, investigate the effects of air velocity at the inlet of the channel and the maximum surface temperature on the maximum total power dissipation of electronic components. Let the air velocity vary from 1 m/s to 20 m/s and the surface temperature from 30°C to 90°C. Plot the power dissipation as functions of air velocity and surface temperature, and discuss the results.

Step 1 of 10

Given data:

Length of the *PCB*, $L = 20\ \text{cm} = 0.2\ \text{m}$

Width of *PCB*, $w = 15\ \text{cm} = 0.15\ \text{m}$

Cross-section of the channel, $0.002\ \text{m} \times 0.14\ \text{m}$

Inlet temperature of air, $T_i = 15°\text{C}$

Velocity of inlet air, $V = 4\ \text{m/s}$

Surface temperature of the channel, $T_s = 50°\text{C}$

Step 2 of 10

Case 1: Varying air velocity

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

Equations Window

```
" Given data"
"V=4 [m/s]"                          " Variable in case 1"
L=20 [cm]
w=15 [cm]
w_ac=0.002 [m]
L_ac=0.14 [m]
T_i=15 [C]
T_s=50 [C]                           "Varible in case 2"
P_1=1 [atm]*convert(atm,kPa)

T_a=(T_s+T_i)/2                      "Average temprature"
A_c=w_ac*L_ac                        "Cross sectional area of air channel"
p=2*(w_ac+L_ac)                      "Perimeter of air channel"
rho=Density(Air,T=T_a,P=P_1)
k=Conductivity(Air,T=T_a)
cp=Cp(Air,T=T_a)
mu_1=Viscosity(Air,T=T_a)
D_h=(4*A_c)/p                        "Hydraulic diameter"

Re=(rho*V*D_h)/mu_1                  "Reynolds number "

L_h=0.05*Re*D_h                      "Hydrodynamic entry length"

"From the table of 'Nusselt number for diffrent cross- sections' for a/v=70 is "

Nus=8.24

k=h*D_h/Nus                          "Heat transfer coefficient"

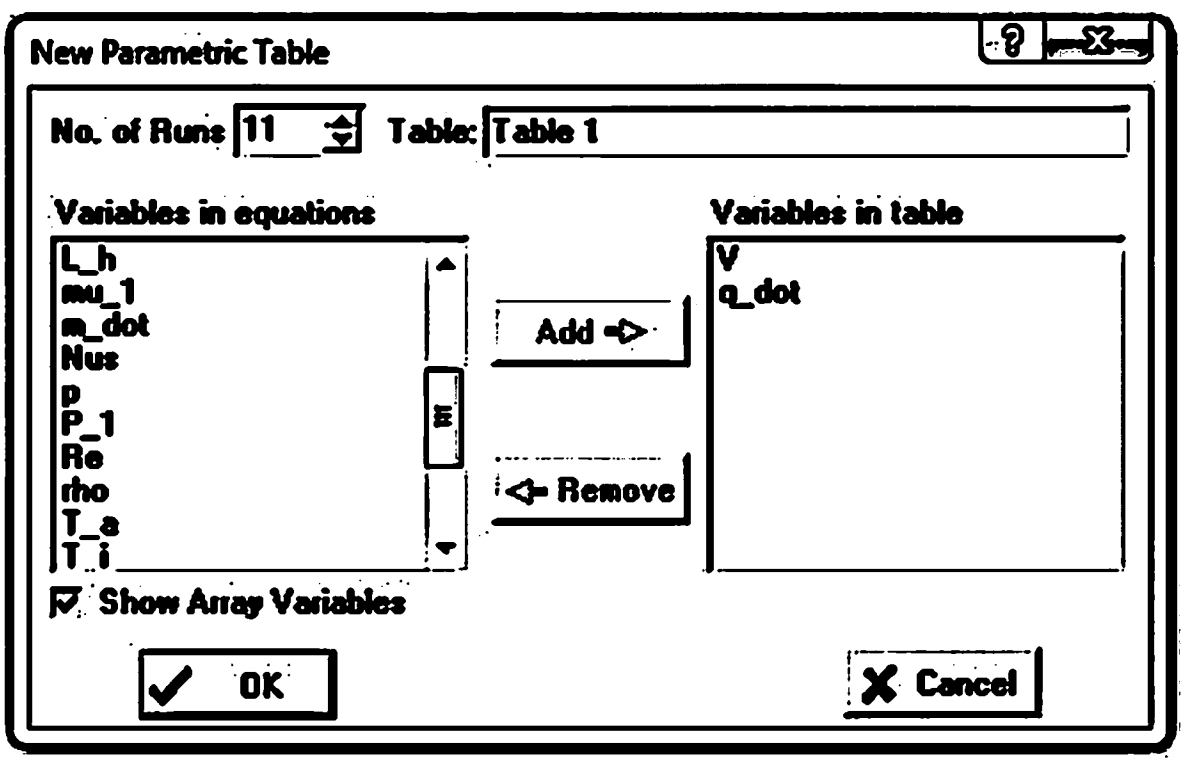
m_dot=rho*v*A_c                      " Mass flow rate"

q_dot=h*A_c*(T_s-T_o)                "Rate of heat transfer"

q_dot=m_dot*cp*(T_o-T_i)             " Heat transfer rate due to convection"
```

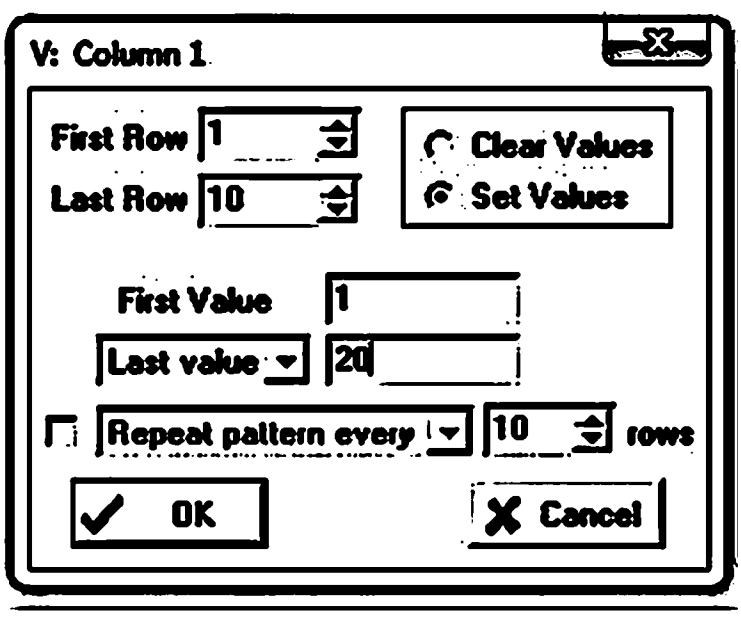
Step 3 of 10

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,

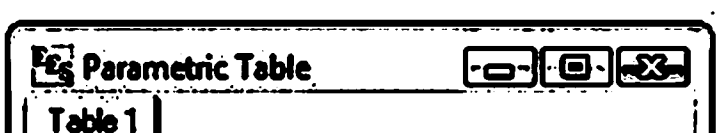


Step 4 of 10

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

19.095P pg 2

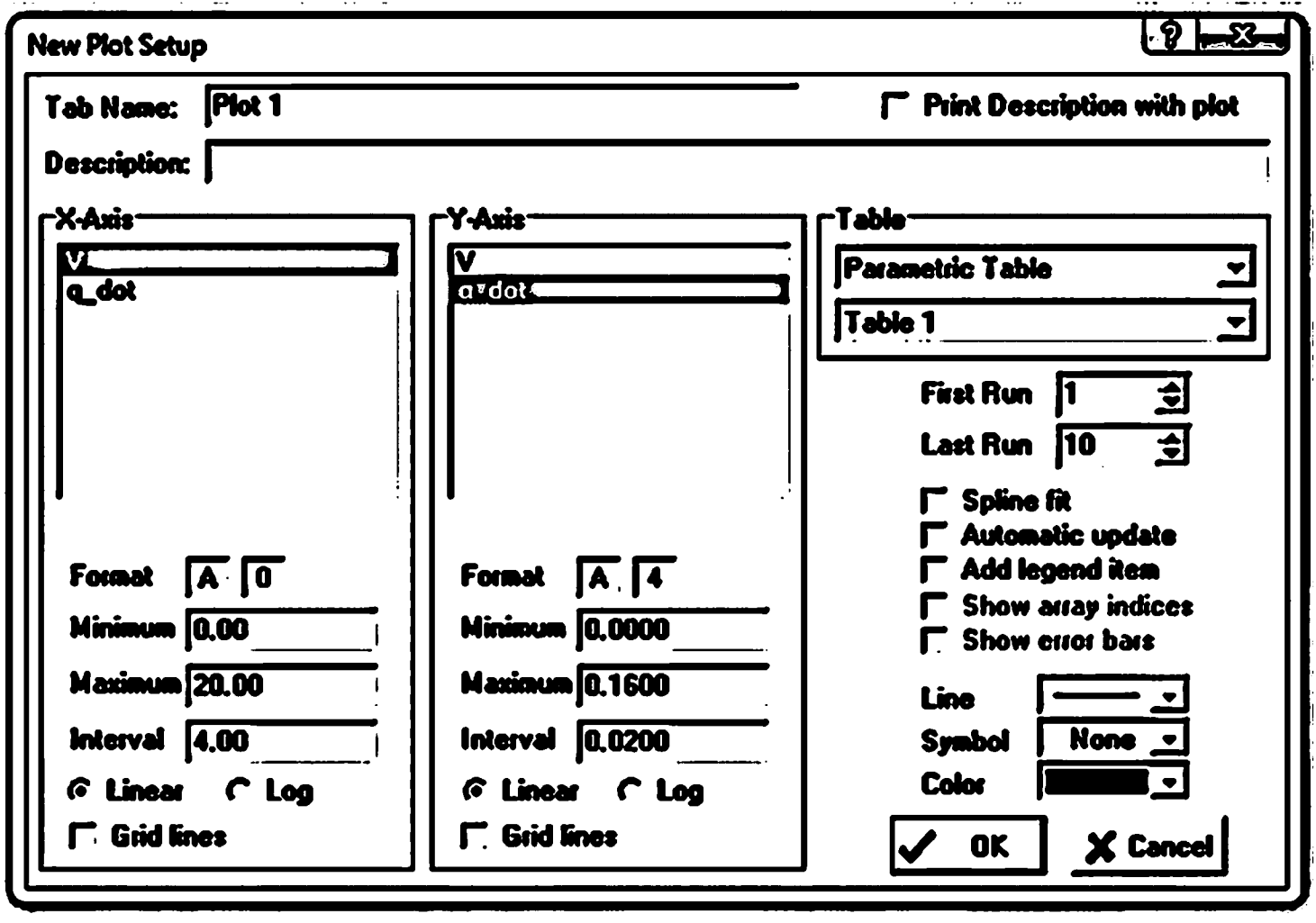
Parametric Table

Table 1

Δ 1..10	1 $\dot{V}$ [m/s]	2 $\dot{q}$ [kW]
Run 1	1	0.01114
Run 2	3.111	0.03319
Run 3	5.222	0.05345
Run 4	7.333	0.07214
Run 5	9.444	0.08943
Run 6	11.56	0.1055
Run 7	13.67	0.1204
Run 8	15.78	0.1343
Run 9	17.89	0.1473
Run 10	20	0.1595

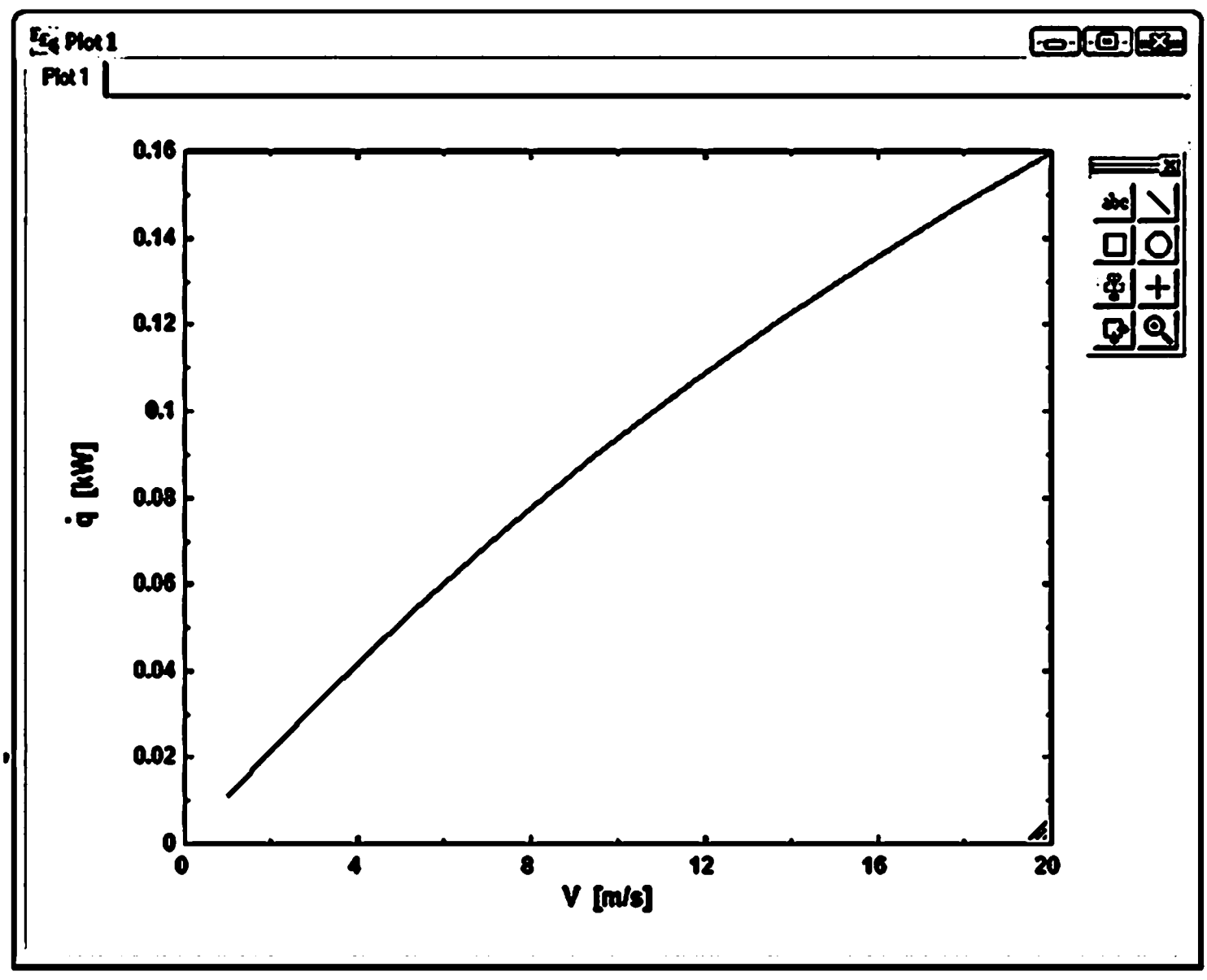
Step 5 of 10

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 6 of 10

By clicking on OK the graph will be generated as shown,



With the increase in inlet air velocity the maximum power dissipation increases with a decreasing slope.

Case 2: Varying surface temperature

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

Equations Window

```
" Given data"
V=4 [m/s]                          " Variable in case 1"
L=20 [cm]
w=15 [cm]
w_ac=0.002 [m]
L_ac=0.14 [m]
T_i=15 [C]
"T_s=50 [C]"                       "Varible in case 2"
P_1=1 [atm]*convert(atm,kPa)

T_a=(T_s+T_i)/2                    "Average temprature"
A_c=w_ac*L_ac                      "Cross sectional area of air channel"
p=2*(w_ac+L_ac)                    "Perimeter of air channel"
rho=Density(Air,T=T_a,P=P_1)
k=Conductivity(Air,T=T_a)
cp=Cp(Air,T=T_a)
mu_1=Viscosity(Air,T=T_a)

D_h=(4*A_c)/p                      "Hydraulic diameter"

Re=(rho*V*D_h)/mu_1                "Reynolds number "

L_h=0.05*Re*D_h                    "Hydrodynamic entry length"

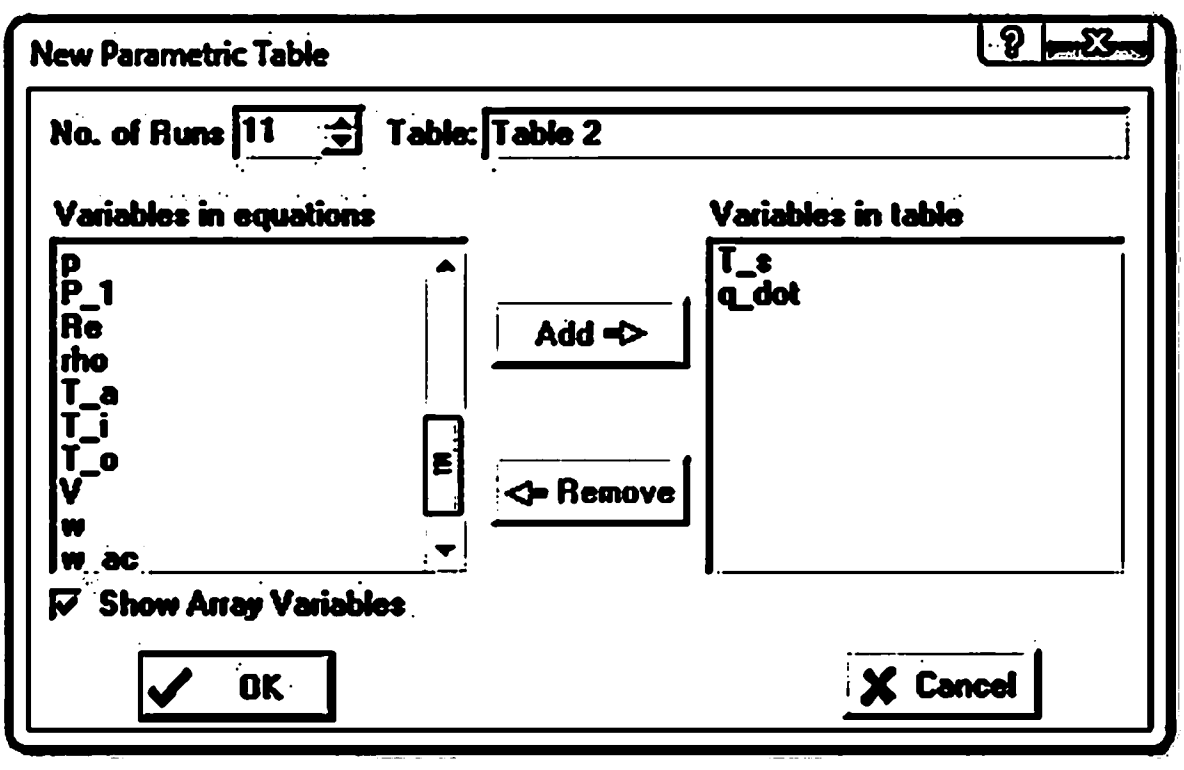
"From the table of 'Nusselt number for diffrent cross- sections' for a/v=70 is "

Nus=8.24
```

```
L_h=0.05*Re*D_h                    "Hydrodynamic entry length"
"From the table of 'Nusselt number for diffrent cross- sections' for a/v=70 is "
Nus=8.24
k=h*D_h/Nus                        "Heat transfer coefficient"
m_dot=rho*v*A_c                    " Mass flow rate"
q_dot=h*A_c*(T_s-T_o)              "Rate of heat transfer"
q_dot=m_dot*cp*(T_o-T_i)           " Heat transfer rate due to convection"
```

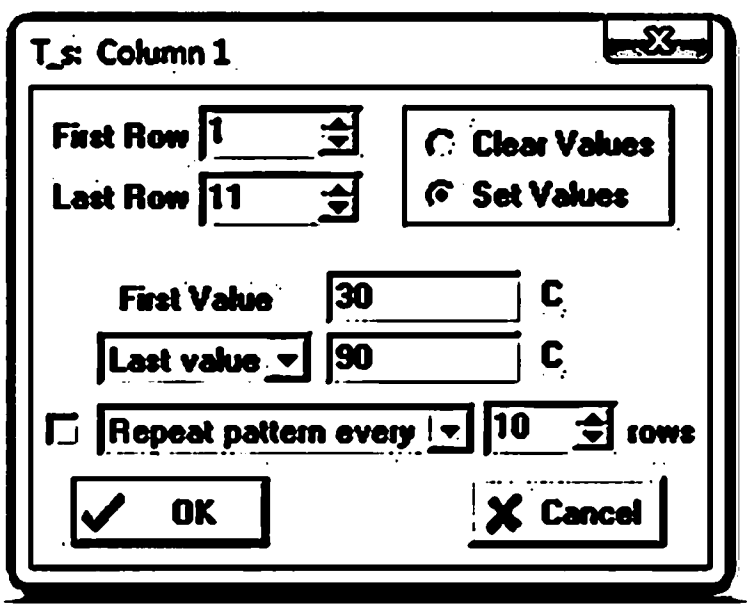
Step 7 of 10

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 8 of 10

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

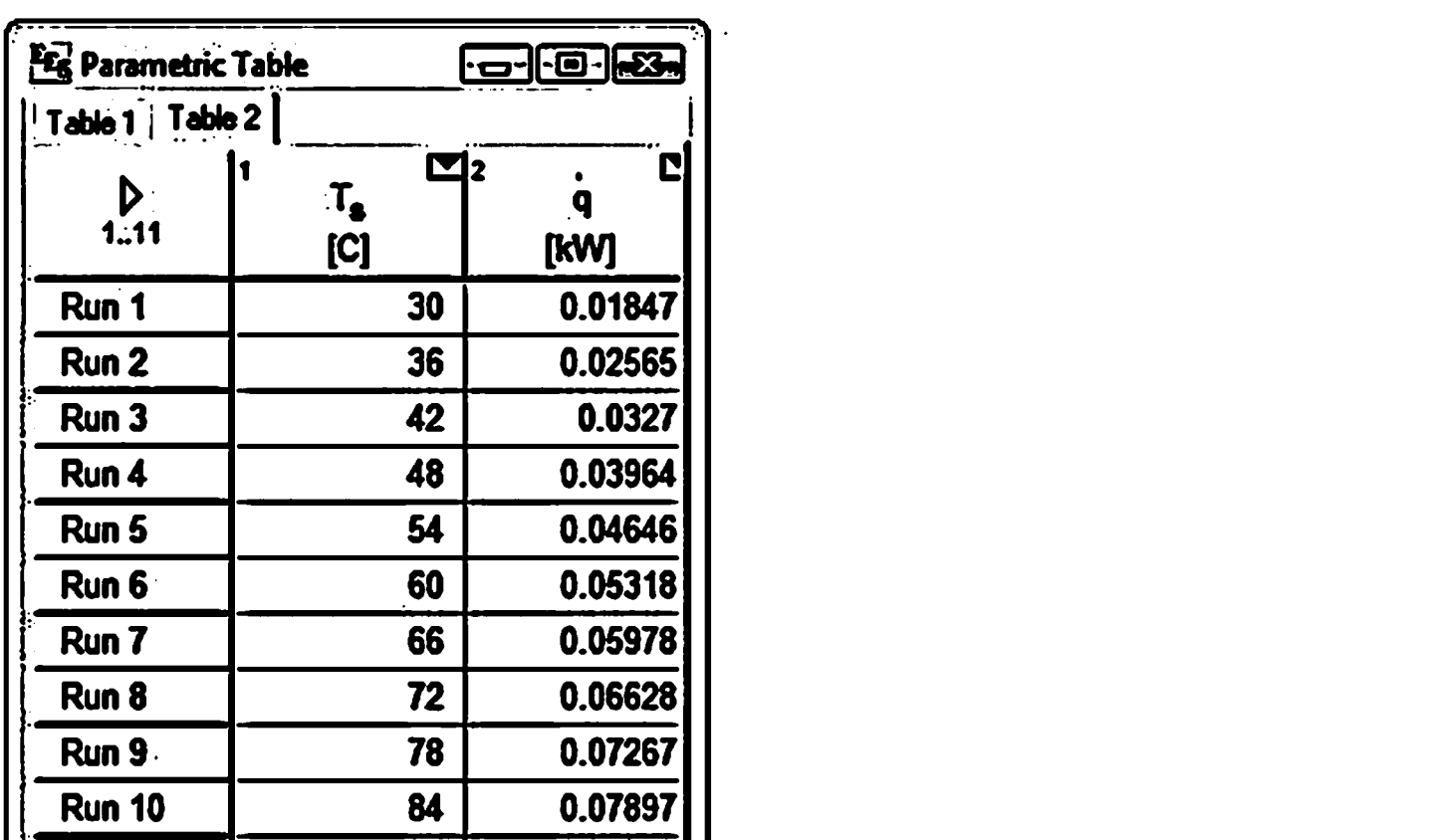
Parametric Table

Table 1 | Table 2

1..11	T_s [C]	$\dot{q}$ [kW]
Run 1	30	0.01847
Run 2	36	0.02565
Run 3	42	0.0327
Run 4	48	0.03964
Run 5	54	0.04646
Run 6	60	0.05318
Run 7	66	0.05978
Run 8	72	0.06628
Run 9	78	0.07267
Run 10	84	0.07897
Run 11	90	0.08516

Step 9 of 10

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,

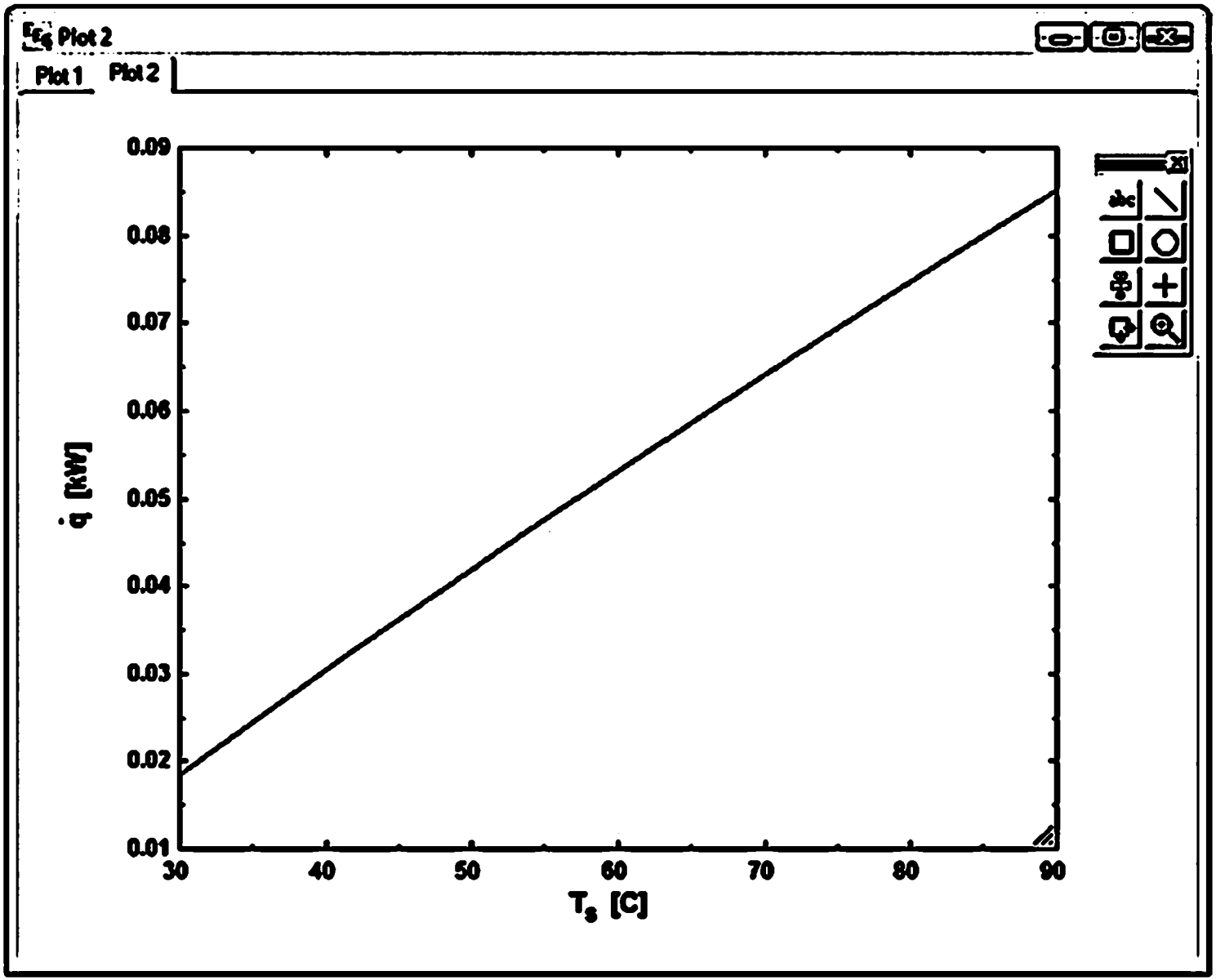


Step 10 of 10

By clicking on OK the graph will be generated as shown,

Step 10 of 10

By clicking on OK the graph will be generated as shown,



With the increase in surface temperature the maximum power dissipation increases with a decreasing slope which is much slower rate comparatively to increase in air velocity.

Question: 19.096P

Hot air at 60°C leaving the furnace of a house enters a 12-m-long section of a sheet metal duct of rectangular cross section 20 cm ×20 cm at an average velocity of 4 m/s. The thermal resistance of the duct is negligible, and the outer surface of the duct, whose emissivity is 0.3, is exposed to the cold air at 10°C in the basement, with a convection heat transfer coefficient of 10 W/m2·K. Taking the walls of the basement to be at 10°C also, determine (*a*) the temperature at which the hot air will leave the basement and (*b*) the rate of heat loss from the hot air in the duct to the basement.

Step 1 of 5

Calculate the characteristic length,

$$L_h = \frac{4A_c}{p}$$

Here,

The cross-section area of the duct is A_c

The perimeter of the duct

$$\begin{aligned} L_h &= \frac{4A_c}{p} \\ &= \frac{4(0.2\times 0.2)}{4\times 0.2} \\ &= 0.2\,\text{m} \end{aligned}$$

Step 2 of 5

Calculate the Reynolds's number,

$$\text{Re} = \frac{\rho V_{avg} D}{\mu}$$

Here,

The density of air is ρ

The average velocity of the air is V_{avg}

$$\begin{aligned} \text{Re} &= \frac{\rho V_{avg} D}{\mu} \\ &= \frac{4\times 0.2}{1.896\times 10^{-5}} \\ &= 42194.09 \end{aligned}$$

Since $\text{Re} > 10000$, the flow is turbulent flow.

Entry lengths, $L_h \approx L_t \approx 10D$

$$\begin{aligned} L_h &= 10D \\ &= 10\times 0.2 \\ &= 2\,\text{m} \end{aligned}$$

Step 3 of 5

Since, the entry length is shorter than the total length of the duct.

Therefore, the flow is fully turbulent.

Calculate the nusselt number,

$$\begin{aligned} Nu &= 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} \\ &= 0.023\times 42194.03^{0.8}\times 0.7202^{0.3} \\ &= 104.5116 \end{aligned}$$

Calculate the heat transfer coefficient by using Nusselt number,

$$Nu = \frac{hD_h}{k}$$

$$\begin{aligned} h &= \frac{Nu\times k}{D_h} \\ &= \frac{104.5116\times 0.02808}{0.2} \\ &= 14.67\,\text{W/m}^2\cdot\text{K} \end{aligned}$$

Step 4 of 5

Calculate the surface area of the tube,

$$A_s = 4aL$$

Here,

The length of the sheet metal is *L*

$$\begin{aligned} A_s &= 4aL \\ &= 4\times 0.2\times 12 \\ &= 9.6\ \text{m}^2 \end{aligned}$$

Calculate the mass flow rate of the air,

$$\begin{aligned} \dot{m} &= \rho A_c V \\ &= 1.059\times(0.2\times 0.2)4 \\ &= 0.16944\ \text{kg/sec} \end{aligned}$$

Step 5 of 5

a) Find the exit temperature of the air by using the equation,

$$\begin{aligned} T_e &= T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \\ &= 10 - (10-60)\exp\left(-\frac{14.67\times 9.6}{0.16944\times 1007}\right) \\ &= 31.9°\text{C} \end{aligned}$$

Therefore, the exit temperature of the air is $\boxed{31.9°\text{C}}$

b) Calculate the logarithmic mean temperature difference,

$$\begin{aligned} \Delta T_{lm} &= \frac{60-31.90}{\ln\left(\frac{10-31.90}{10-60}\right)} \\ &= -34.03°\text{C} \end{aligned}$$

Calculate the heat loss from the air,

$$\dot{Q} = hA_s\Delta T_{lm}$$

Here,

The Convection heat coefficient is *h*

The Surface area of the tube is A_s

The logarithmic mean temperature difference is ΔT_{lm}

Now substitute the known values in the above expression,

$$\begin{aligned} \dot{Q} &= hA_s\Delta T_{lm} \\ &= 14.67\times 9.6\times(-34.038) \\ &= -4793.70\,\text{W} \end{aligned}$$

Therefore, the heat loss from the duct, $\boxed{-4793.70\,\text{W}}$

Question: 19.097P pg 1

Reconsider Prob. 19-96. Using EES (or other) software, investigate the effect of air velocity and the surface emissivity on the exit temperature of air and the rate of heat loss. Let the air velocity vary from 1 m/s to 10 m/s and the emissivity from 0.1 to 1.0. Plot the exit temperature and the rate of heat loss function the air velocity and emissivity, and discuss the results.

Step 1 of 9

Case 1: Varying air velocity

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

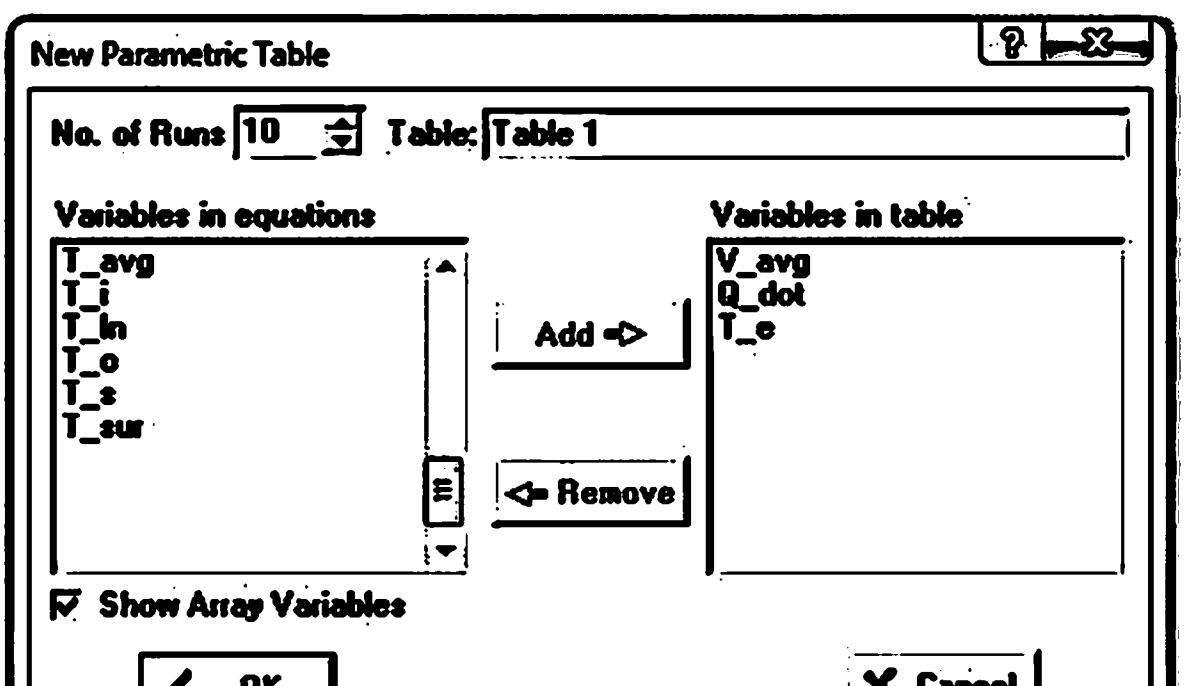
Equations Window

```
"Given data"
L=12 [m]
s=0.20 [m]
"V_avg=4 [m/s]"                    "Variable in case 1"
epsilon=0.3                        "Variable in case 2"
T_i=60 [C]
T_o=10 [C]
h_o=10 [W/m^2-C]
T_sur=10 [C]

"Properties of the air at inlet temprature "
Fluid$='air'
Cp=CP(Fluid$, T=T_avg)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid$, T=T_avg)
Pr=Prandtl(Fluid$, T=T_avg)
rho=Density(Fluid$, T=T_avg, P=101.3)
mu=Viscosity(Fluid$, T=T_avg)
```

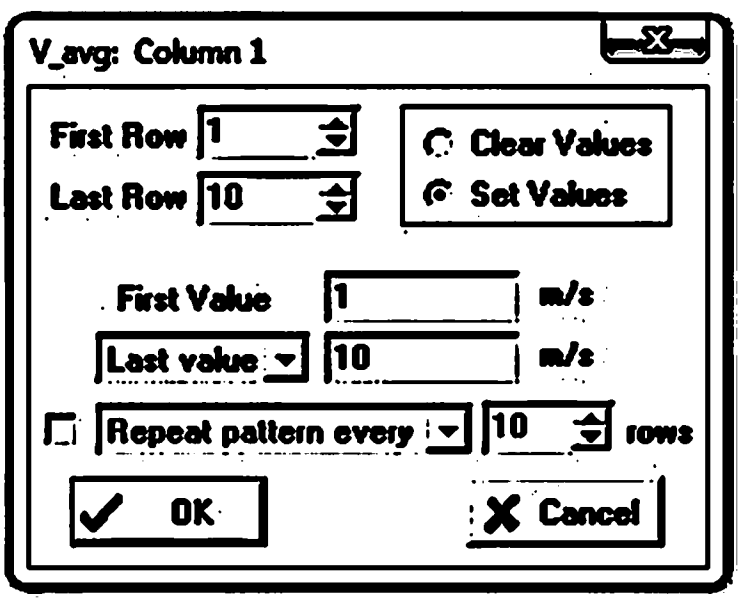
Step 2 of 9

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 3 of 9

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

Table 1

1..10	1 V_{avg} [m/s]	2 $\dot{Q}$	3 T_e
Run 1	1	1150	33.84
Run 2	2	1809	39.41
Run 3	3	2272	42.77
Run 4	4	2621	45.09
Run 5	5	2897	46.82
Run 6	6	3122	48.16
Run 7	7	3309	49.24
Run 8	8	3468	50.14
Run 9	9	3605	50.88
Run 10	10	3725	51.52

Run 9	19	097P 3605 pg	2 50.88
Run 10	10	3725	51.52

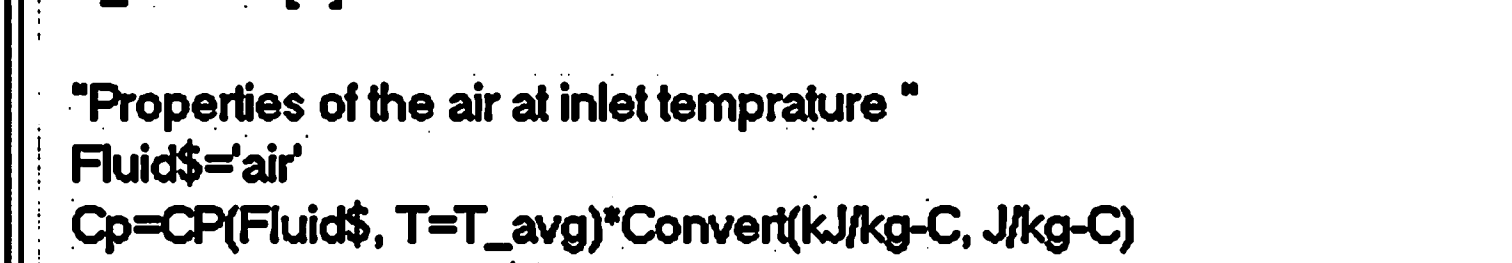
Step 4 of 9

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 5 of 9

By clicking on OK the graph will be generated as shown,



With the increase in inlet air velocity power dissipation decreases along with exist temperature as shown.

Case 2: Varying surface temperature

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

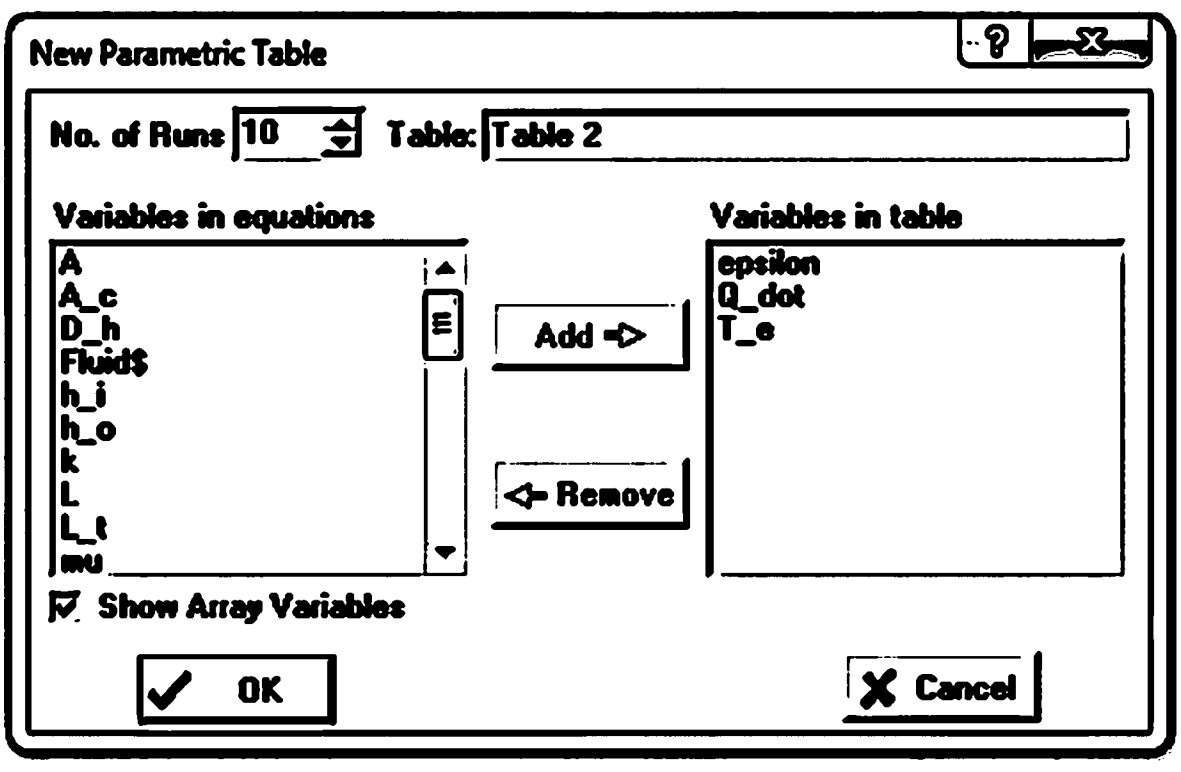
Equations Window

```
"Given data"
L=12 [m]
s=0.20 [m]
V_avg=4 [m/s]                          "Variable in case 1"
"epsilon=0.3"                          "Variable in case 2"
T_i=60 [C]
T_o=10 [C]
h_o=10 [W/m^2-C]
T_sur=10 [C]

"Properties of the air at inlet temprature "
Fluid$='air'
Cp=CP(Fluid$, T=T_avg)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid$, T=T_avg)
Pr=Prandtl(Fluid$, T=T_avg)
rho=Density(Fluid$, T=T_avg, P=101.3)
mu=Viscosity(Fluid$, T=T_avg)
```

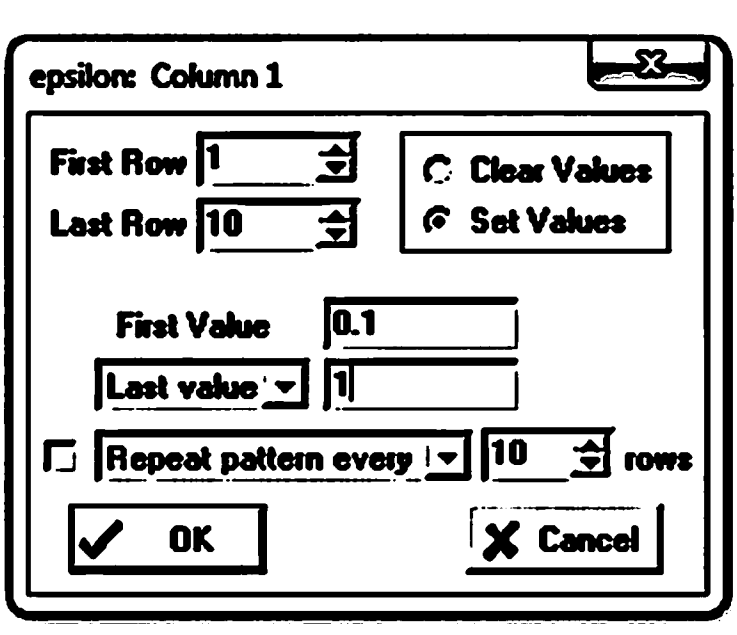
Step 6 of 9

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 7 of 9

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

Table 1 | Table 2

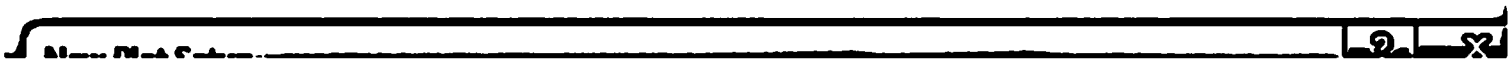
1..10	ε	$\dot{Q}$	T_e
Run 1	0.1	2494	45.81
Run 2	0.2	2560	45.44
Run 3	0.3	2621	45.09
Run 4	0.4	2679	44.76
Run 5	0.5	2734	44.45
Run 6	0.6	2786	44.15

Run 2	0.2	2660	45.44
Run 3	0.3	2621	45.09
Run 4	0.4	2679	44.76
Run 5	0.5	2734	44.45
Run 6	0.6	2786	44.15
Run 7	0.7	2835	43.87
Run 8	0.8	2882	43.6
Run 9	0.9	2927	43.35
Run 10	1	2969	43.11

19.097P pg 3

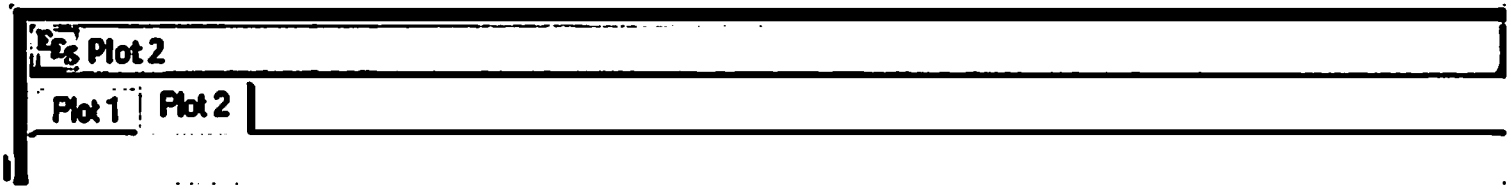
Step 8 of 9

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 9 of 9

By clicking on OK the graph will be generated as shown,



With the increase in the values of epsilon the exit temperature decreases with heat lost to air increase.

Question: 19.098P

Liquid water flows at a mass flow rate of 0.7 kg/s through the concentric annulus tube with the inlet and outlet mean temperature of 20°C and 80°C, respectively. The concentric annulus tube has inner and outer diameters of 10 mm and 100 mm, respectively. the inner tube wall is maintained with a constant surface temperature of 120°C, while the outer tube surface is insulated. Determine the length of the concentric annulus tube.

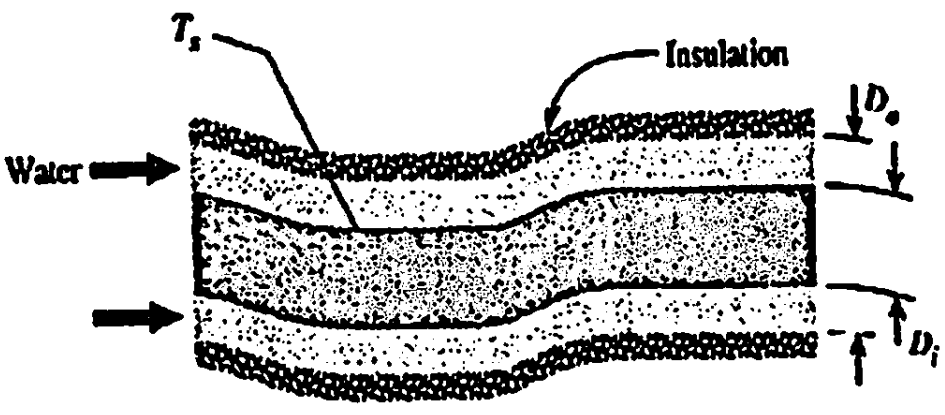


FIGURE P19–98

Step 1 of 5

Calculate the average temperature of water.

$$T_{avg} = \frac{T_e + T_i}{2}$$

Here, T_i is the inlet mean temperature and T_e is the outlet mean temperature of water.

Substitute $20°C$ for T_i and $80°C$ for T_e .

$$T_{avg} = \frac{20+80}{2}$$
$$= 50°C$$

Step 2 of 5

Obtain the following properties from Table A-9 "Properties of Saturated water" at average temperature of $50°C$.

Thermal conductivity, $k = 0.644\ \mathrm{W/m°C}$

Dynamic viscosity, $\mu = 0.547\times10^{-3}\ \mathrm{kg/m\cdot s}$

Specific heat, $c_p = 4181\ \mathrm{J/kg\cdot K}$

Obtain a relation for average velocity of flow.

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$
$$= \frac{\dot{m}}{\rho\frac{\pi\left(D_0^2 - D_i^2\right)}{4}}$$
$$= \frac{4\dot{m}}{\rho\pi\left(D_0^2 - D_i^2\right)}$$

Here, D_0 is the outer diameter of the annular tube, D_i is the inner diameter of the annular tube, and $\dot{m}$ is the mass flow rate.

Step 3 of 5

Calculate the Reynolds number using the following relation.

$$\mathrm{Re} = \frac{\rho V_{avg}(D_o - D_i)}{\mu}$$
$$= \frac{\rho 4\dot{m}(D_o - D_i)}{\mu\rho\pi\left(D_0^2 - D_i^2\right)}$$
$$= \frac{4\dot{m}}{\mu\pi(D_o + D_i)}$$

Substitute $0.01\mathrm{m}$ for D_i , $0.1\mathrm{m}$ for D_o , $0.547\times10^{-3}\ \mathrm{kg/m\cdot s}$ for μ , and $0.7\mathrm{kg/s}$ for $\dot{m}$.

$$\mathrm{Re} = \frac{4\times0.7}{\pi\times0.547\times10^{-3}(0.1+0.01)}$$
$$= 14812.49$$

As the Reynolds number is more than 10000, the flow through the annulus is turbulent.

Step 4 of 5

Use Dittus-Boelter relation for Nusselt number:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$$
$$= 0.023\times(14812.49)^{0.8}\times3.55^{0.4}$$
$$= 82.85$$

Calculate the convection heat transfer coefficient,

$$h_i = \frac{Nu\times k}{(D_o - D_i)}$$

Substitute 82.85 for Nu , $0.644\ \mathrm{W/m°C}$ for k , $0.01\mathrm{m}$ for D_i , and $0.1\mathrm{m}$ for D_o .

$$h_i = \frac{82.85\times0.644}{(0.1-0.01)}$$
$$= 592.83\ \mathrm{W/m^2°C}$$

Step 5 of 5

Calculate the length of the concentric annulus tube using the following relation:

$$L = -\frac{\dot{m}c_p}{\pi D_i h_i}\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)$$

Here, T_s is the surface temperature of inner tube, c_p is the specific heat of water.

Substitute $0.7\mathrm{kg/s}$ for $\dot{m}$, $4181\ \mathrm{J/kg\cdot K}$ for c_p , $0.01\mathrm{m}$ for D_i , $592.83\ \mathrm{W/m^2°C}$ for h_i , $120°C$ for T_s , $20°C$ for T_i , and $80°C$ for T_e .

$$L = -\frac{0.7\times4181}{\pi(0.01)(592.83)}\ln\left(\frac{120-80}{120-20}\right)$$
$$= -(-144\ \mathrm{m})$$
$$= 144\mathrm{m}$$

Therefore, the length of the concentric annulus tube is $\boxed{144\ \mathrm{m}}$.

Question: 19.099P

Consider a 10-m-long smooth rectangular tube, with a=50 mm and b = 25 mm, that is maintained at a constant surface temperature. Liquid water enters the tube at 20° with a mass flow rate of 0.01 kg/s. Determine the tube surface temperature necessary to heat the water to the desired outlet temperature of 80°C.

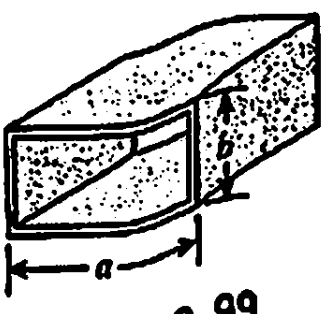


FIGURE P19–99

Step 1 of 6

Given:

Length of rectangular tube, $L = 10\,\text{m}$

Width, $a = 0.05\text{m}$

Height, $b = 0.025\text{m}$

Inlet flow temperature, $T_i = 20°\text{C}$

Mass flow rate, $\dot{m} = 0.01\,\text{kg/s}$

Outlet flow temperature, $T_e = 80°\text{C}$

Step 2 of 6

Average temperature,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{20+80}{2}$$
$$= 50°\text{C}$$

Hydraulic diameter of rectangular tube,

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4(a\times b)}{2(a+b)}$$
$$= \frac{4(0.05\times 0.025)}{2(0.05+0.025)}$$
$$= 0.0333\text{m}$$

Step 3 of 6

Average velocity,

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$
$$= \frac{0.01}{988.1\times(0.05\times 0.025)}$$
$$= 8.096\times 10^{-3}\,\text{m/s}$$

Properties of water at 1atm & average temperature,

Density, $\rho = 988.1\,\text{kg/m}^3$

Specific heat, $c_p = 4181\,\text{J/kg-K}$

Thermal conductivity, $k = 0.644\,\text{W/m-K}$

Kinematic viscosity, $\mu = 0.547\times 10^{-3}\,\text{kg/m-s}$

Prandtl number, $\text{Pr} = 3.55$

Step 4 of 6

Reynolds number,

$$\text{Re} = \frac{\rho V_{avg} D_h}{\mu}$$
$$= \frac{988.1\times 8.096\times 10^{-3}\times 0.0333}{0.547\times 10^{-3}}$$
$$= 487.02$$

As the obtained Reynolds number is less than the 2300, the flow is laminar.

Hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D_h$$
$$= 0.05\times 487.02\times 0.0333$$
$$= 0.811\,\text{m}$$

Thermal entry length,

$$L_t = \text{Pr}\times L_h$$
$$= 3.55\times 0.811$$
$$= 2.879\text{m}$$

Step 5 of 6

As both the entry lengths are shorter than the total tube length, the flow is a fully developed laminar flow. Now a/b value is

$$a/b = \frac{0.05}{0.025}$$
$$= 2$$

From the table of 'Nusselt numbers for different cross sections', the Nusselt number for $a/b = 2$ is

$$Nu = 3.39$$

Heat transfer coefficient,

$$h = \frac{k\times Nu}{D_h}$$
$$= \frac{0.644\times 3.39}{0.0333}$$
$$= 65.56\,\text{W/m}^2\text{K}$$

Step 6 of 6

The surface area,

$$A_s = 2(a+b)L$$
$$= 2(0.05+0.025)10$$
$$= 1.5\,\text{m}^2$$

The tube exit temperature,

$$T_e = T_s - (T_s - T_i)\exp\left[-\frac{hA_s}{\dot{m}C_p}\right]$$
$$80 = T_s - (T_s - 20)\exp\left[-\frac{65.56\times 1.5}{0.01\times 4181}\right]$$
$$80 = T_s - (T_s - 20)\exp[-2.352]$$

By solving this, we get $T_s = 86.31°\text{C}$

Hence the surface temperature is $\boxed{T_s = 86.31°\text{C}}$

Question: 19.100P

The components of an electronic system dissipating 220 W are located in a l-m-long horizontal duct whose cross section is 16 cm ×16 cm. The components in the duct are cooled by forced air, which enters at 27°C at a rate of 0.65 m3/min. Assuming 85 percent of the heat generated inside is transferred to air flowing through the duet and the remaining 15 percent is lost through the outer surfaces of the duct, determine (*a*) the exit temperature of air and (*b*) the highest component surface temperature in the duct.

Step 1 of 4

Given data:

Width of horizontal duct, $W = 0.16\,\text{m}$

Height of horizontal duct, $H = 0.16\,\text{m}$

Length of duct, $L = 1\text{m}$

Air inlet temperature, $T_i = 27^\circ\text{C}$

Volume flow rate, $\dot{V} = 0.65\,\text{m}^3/\text{min} = 0.01083\,\text{m}^3/\text{s}$

Heat dissipation, $\dot{Q} = 220\ \text{W}$

Step 2 of 4

Assuming average temperature as 35°C, the properties of air are

Density, $\rho = 1.145\,\text{kg}/\text{m}^3$

Specific heat, $c_p = 1007\,\text{J}/\text{kg}\cdot\text{k}$

Thermal conductivity, $k = 1.655\times10^{-5}\,\text{m}^2/\text{s}$

Kinematic viscosity, $\nu = 1.655\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7268$

As 85% of heat is transferred to air, the heat transfer to air can be written as

$$\dot{Q}_{air} = \dot{m}c_p(T_e - T_i)$$

$$0.85\dot{Q} = \rho\dot{V}c_p(T_e - T_i)$$

$$0.85\times220 = 1.145\times0.01083\times1007\times(T_e - 27)$$

$$187 = 12.487(T_e - 27)$$

$$T_e - 27 = 14.9754$$

$$T_e = 41.98^\circ\text{C}$$

Therefore the exit temperature of air is $\boxed{T_e = 41.98^\circ\text{C}}$

Step 3 of 4

The average velocity of the flow,

$$V_{avg} = \frac{\dot{V}}{A_c}$$

$$= \frac{0.01083}{0.16\times0.16}$$

$$= 0.423\,\text{m}/\text{s}$$

Reynolds number is determined by

$$\text{Re} = \frac{V_{avg}}{\nu}\times D_h$$

$$= \frac{0.423}{1.655\times10^{-5}}\times\left(\frac{4\times0.16\times0.16}{4\times0.16}\right)$$

$$= 4089.88$$

As the value of Reynolds number lies in between 2300 to 10000, there is turbulence in the flow. So by considering the flow as fully developed turbulent flow,

Step 4 of 4

Therefore Nusselt number is determined by using the following equation

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$

$$= 0.023(4089.88)^{0.8}(0.7268)^{0.4}$$

$$= 15.69$$

Now heat transfer coefficient is determined by

$$h = \frac{Nu\times k}{D_h}$$

$$= \frac{15.69\times0.02625}{0.16}$$

$$= 2.574\ \text{W}/\text{m}^2\,^\circ\text{C}$$

Therefore heat transfer rate,

$$\dot{Q} = hA_s(T_s - T_e)$$

$$187 = 2.574\times(4\times0.16)(T_s - 41.98)$$

$$T_s - 41.98 = 113.515$$

$$T_s = 155.5^0\text{C}$$

Hence the highest component surface temperature is $\boxed{T_s = 155.5^\circ\text{C}}$

Question: 19.101P

Repeat Prob. 19-100 for a circular horizontal duct of 15-cm diameter.

Step 1 of 4

Given data:

Diameter of duct, $D = 0.15\text{m}$

Length of duct, $L = 1\text{m}$

Air inlet temperature, $T_i = 27^\circ\text{C}$

Volume flow rate, $\dot{V} = 0.65\ \text{m}^3/\text{min} = 0.01083\ \text{m}^3/\text{s}$

Heat dissipation, $\dot{Q} = 220\ \text{W}$

Step 2 of 4

Assuming average temperature as 35°C,

The properties of air are

Density, $\rho = 1.145\ \text{kg}/\text{m}^3$

Specific heat, $c_p = 1007\text{J/kg}-\text{k}$

Thermal conductivity, $k = 1.655\times10^{-5}\ \text{m}^2/\text{s}$

Kinematic viscosity, $\nu = 1.655\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7268$

As 85% of heat is transferred to air, the heat transfer to air can be written as

$$\dot{Q}_{air} = \dot{m}c_p(T_e - T_i)$$

$$0.85\dot{Q} = \rho\dot{V}c_p(T_e - T_i)$$

$$0.85\times220 = 1.145\times0.01083\times1007\times(T_e - 27)$$

$$187 = 12.487(T_e - 27)$$

$$T_e - 27 = 14.9754$$

$$T_e = 41.98^\circ\text{C}$$

Therefore the exit temperature of air is $\boxed{T_e = 41.98^\circ\text{C}}$

Step 3 of 4

The average velocity of the flow,

$$V_{avg} = \frac{\dot{V}}{A_c}$$

$$= \frac{0.01083\times4}{\pi\times(0.15)^2}$$

$$= 0.6129\ \text{m/s}$$

Thus the Reynolds number is determined by

$$\text{Re} = \frac{V_{avg}\times D_h}{\nu}$$

$$= \frac{0.6129\times0.15}{1.655\times10^{-5}}$$

$$= 5554.98$$

As the value of Reynolds number lies in between 2300 to 10000, there is turbulence in the flow. So by considering the flow as fully developed turbulent flow,

Step 4 of 4

Therefore Nusselt number,

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$

$$= 0.023(5554.98)^{0.8}(0.7268)^{0.4}$$

$$= 20.05$$

Now heat transfer coefficient,

$$h = \frac{Nu\times k}{D_h}$$

$$= \frac{20.05\times0.02625}{0.15}$$

$$= 3.51\ \text{W}/\text{m}^2\,^\circ\text{C}$$

Therefore heat transfer rate,

$$\dot{Q} = hA_s(T_s - T_e)$$

$$187 = 3.51\times\pi\times0.15\times1(T_s - 41.98)$$

$$T_s - 41.98 = 113.065$$

$$T_s = 155.04^\circ\text{C}$$

Hence the highest component surface temperature is $\boxed{T_s = 155.04^\circ\text{C}}$

Question: 19.102P pg 1

A computer cooled by a fan contains eight printed circuit boards (PCBs), each dissipating 12 W of power. The height of the PCBs is 12 cm and the length is 15 cm. The clearance between the tips of the components on the PCB and the back surface of the adjacent PCB is 0.3 cm. The cooling air is supplied by a 10-W fan mounted at the inlet. If the temperature rise of air as it flows through the case of the computer is not to exceed 10°C, determine (*a*) the flow rate of the air that the fan needs to deliver, (*b*) the fraction of the temperature rise of air that is due to the heat generated by the fan and its motor, and (*c*) the highest allowable inlet air temperature if the surface temperature of the components is not to exceed 70°C anywhere in the system. Use air properties at 25°C.

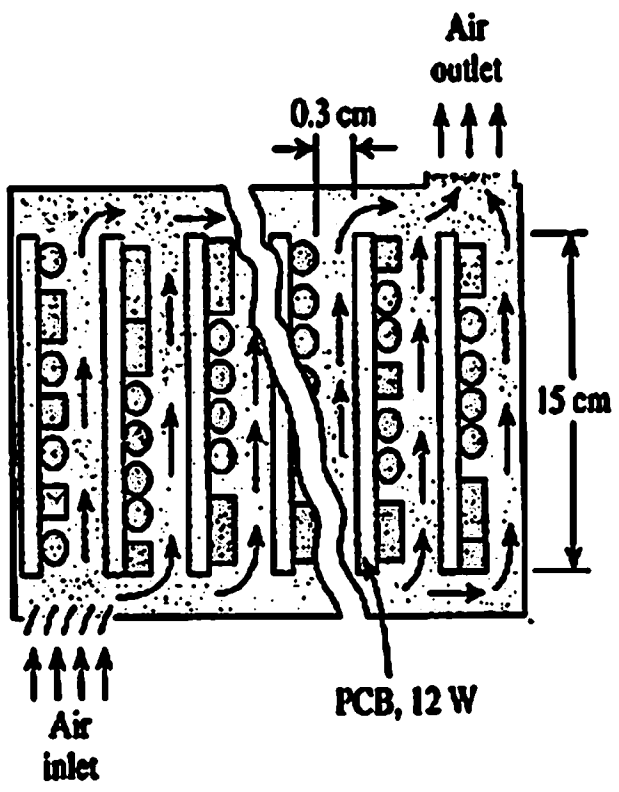


FIGURE P19–102

Step 1 of 5

Given:

Number of circuit boards, $N = 8$

Heat dissipated by each circuit board, $Q = 12\,\text{W}$

Height of *PCB*, $H = 0.12\,\text{m}$

Length of *PCB*, $L = 0.15\,\text{m}$

Clearance between the circuit boards, $w = 0.003\,\text{m}$

Fan power, $p = 10\,\text{W}$

Temperature rise allowed $(T_e - T_i) = 10°\text{C}$

Surface temperature of components, $T_s = 70°\text{C}$

Step 2 of 5

(*a*)

The properties of air at $25°\text{C}$

Therefore density, $\rho = 1.184\ \text{kg/m}^3$

Specific heat, $c_p = 1007\ \text{J/kg-k}$

Thermal conductivity, $k = 0.02551\ \text{W/m-k}$

Kinematic viscosity, $v = 1.562 \times 10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7296$

The heat transfer rate,

$$\dot{Q}_t = \dot{m} c_p (T_e - T_i)$$

$$\dot{m} = \frac{\dot{Q}_t}{c_p (T_e - T_i)}$$

$$= \frac{(N \times Q) + p}{c_p (T_e - T_i)}$$

$$= \frac{(8 \times 12) + 10}{1007 \times 10}$$

$$= 0.01053\ \text{kg/s}$$

Therefore the mass flow rate is $\boxed{\dot{m} = 0.01053\,\text{kg/s}}$

Step 3 of 5

(*b*)

The power used by the fan is

$$p = \dot{m} C_p (T_e - T_i)$$

$$T_e - T_i = \frac{p}{\dot{m} C_p}$$

$$= \frac{10}{0.01053 \times 1007}$$

$$= 0.943°\text{C}$$

The fraction of heat generated by the fan,

$$f = \frac{(T_e - T_i)_p}{T_e + T_i}$$

$$= \frac{0.943}{10}$$

$$= 0.0943$$

Therefore fraction of heat generated by the fan, $\boxed{f = 0.0943}$

Step 4 of 5

Average velocity,

Step 4 of 5

Average velocity,

$$V_{avg} = \frac{\dot{m}}{N\rho A_c}$$
$$= \frac{\dot{m}}{N \times \rho \times (w \times H)}$$
$$= \frac{0.01053}{8 \times 1.184(0.003 \times 0.12)}$$
$$= 3.08\,\text{m/s}$$

Hydraulic diameter,

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4(w \times H)}{2(w + H)}$$
$$= \frac{2(0.003 \times 0.12)}{(0.003 + 0.12)}$$
$$= 5.854 \times 10^{-3}\,\text{m}$$

Reynolds number,

$$\text{Re} = \frac{V_{avg} \times D_h}{\nu}$$
$$= \frac{3.08 \times 5.854 \times 10^{-3}}{1.562 \times 10^{-5}}$$
$$= 1154.24$$

As the Reynolds number is less than 2300, the flow is laminar.

Now the entry length,

$$L_h = 0.05 \times R_e \times D_h$$
$$= 0.3378\,\text{m}$$

This length is less than the total length, therefore the flow is fully developed laminar flow. Now the *a*/*b* ratio is

$$a/b = \frac{0.12}{0.003}$$
$$= 40$$

From the table of 'Nusselt Numbers for various cross-sections', for *a*/*b* ratio 40,

$$Nu = 8.24$$

Step 5 of 5

The heat transfer coefficient

$$h = \frac{k \times Nu}{D_h}$$
$$= \frac{0.02551 \times 8.24}{5.854 \times 10^{-3}}$$
$$= 35.91\,\text{W/m}^2\text{°C}$$

Since we know highest temperature will be at exit point, the heat transfer

$$Q_t = hA_s(T_s - T_e)$$
$$106 = 35.91 \times \left[8 \times 2(0.12 \times 0.15 + 0.003 \times 0.15)\right](70 - T_e)$$
$$106 = 35.91 \times 0.2952(70 - T_e)$$
$$T_e = 60°\text{C}$$

Therefore the exit temperature is $\boxed{T_e = 60°\text{C}}$

Inlet temperature,

$$T_i = T_e - 10$$
$$= 60 - 10$$
$$= 50°\text{C}$$

Therefore the inlet temperature is $\boxed{T_i = 50°\text{C}}$

Question: 19.103P

Consider a 25-mm diameter and 15-m long smooth tube that is used for heating fluids. The wall is heated electrically to provide a constant surface heat flux along the entire tube. Fluids enter the tube at 50°C and exit at 150°C. If the mass flow rate is maintained at 0.01 kg/s, determine the convection heat transfer coefficients at the tube outlet for water, engine oil, and liquid mercury.

Step 1 of 8

Given data:

Diameter of tube, $D = 0.025\text{m}$

Length of tube, $L = 15\text{m}$

Inlet temperature of fluid, $T_i = 50°\text{C}$

Outlet temperature of liquid $T_e = 150°\text{C}$

Mass flow rate, $\dot{m} = 0.01\text{kg/s}$

Step 2 of 8

Average temperature,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{50+150}{2}$$
$$= 100°\text{C}$$

The properties of water at average temperature,

Density, $\rho = 957.9\ \text{kg/m}^3$

Thermal conductivity, $k = 0.679\ \text{W/m}\cdot\text{K}$

Specific heat, $c_p = 4217\text{J/kg}\cdot\text{K}$

Prandtl Number, $\text{Pr} = 1.75$

Kinematic viscosity, $\mu = 0.282\times10^{-3}\ \text{kg/m}\cdot s$

Step 3 of 8

Reynolds number,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$
$$= \frac{4\times0.01}{\pi\times0.025\times0.282\times10^{-3}}$$
$$= 1806$$

As the Reynolds number is less than 2300. The flow is laminar.

The hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D$$
$$= 0.05\times1806\times0.025$$
$$= 2.257\text{m}$$

The thermal entry length,

$$L_t = \text{Pr}\times L_h$$
$$= 1.75\times2.257$$
$$= 3.951\text{m}$$

As the entry lengths are shorter, the flow is a fully developed laminar flow

Step 4 of 8

By taking Nusselt number value from the table 'Nusselt numbers for fully developed laminar flows', for circular tube cross section, $Nu = 4.36$

Heat transfer coefficient

$$h = \frac{k\cdot Nu}{D}$$
$$= \frac{0.679\times4.36}{0.025}$$
$$= 118.42\text{W/m}^2°\text{C}$$

Therefore, the heat transfer coefficient $\boxed{h = 118.42\text{W/m}^2°\text{C}}$

Step 5 of 8

The properties of engine oil at average temperature,

Thermal conductivity, $k = 0.1367\ \text{W/m}\cdot\text{K}$

Prandtl Number, $\text{Pr} = 279.1$

Kinematic viscosity, $\mu = 0.01718\ \text{kg/m}\cdot s$

Reynolds number,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$
$$= \frac{4\times0.01}{\pi\times0.025\times0.01718}$$
$$= 29.64$$

As the Reynolds number is less than 2300. The flow is laminar.

The hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D$$
$$= 0.05\times29.64\times0.025$$
$$= 0.037\text{m}$$

The thermal entry length,

$$L_t = \text{Pr}\times L_h$$
$$= 279.1\times0.037$$
$$= 10.34\text{ m}$$

As the entry lengths are shorter, the flow is a fully developed laminar flow

Step 6 of 8

By taking Nusselt number value from the table 'Nusselt numbers for fully developed laminar flows', for circular tube cross section, $Nu = 4.36$

Heat transfer coefficient

$$h = \frac{k\cdot Nu}{D}$$
$$= \frac{0.1367\times4.36}{0.025}$$
$$= 23.84\text{W/m}^2°\text{C}$$

Therefore, the heat transfer coefficient $\boxed{h = 23.84\text{W/m}^2°\text{C}}$

Step 7 of 8

The properties of engine oil at average temperature,

Thermal conductivity, $k = 9.46706\ \text{W/m}°\text{C}$

Prandtl Number, $\text{Pr} = 0.018$

Kinematic viscosity, $\mu = 1.245\times10^{-3}\ \text{kg/m}\cdot s$

Reynolds number is determined by

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$
$$= \frac{4\times0.01}{\pi\times0.025\times1.245\times10^{-3}}$$
$$= 409$$

As the Reynolds number is less than 2300. The flow is laminar.

The hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D$$
$$= 0.05\times409\times0.025$$
$$= 0.511\text{m}$$

As the entry length is shorter, the flow is a fully developed laminar flow

Step 8 of 8

By taking Nusselt number value from the table 'Nusselt numbers for fully developed laminar flows', for circular tube cross section, $Nu = 4.36$

Heat transfer coefficient

$$h = \frac{k\cdot Nu}{D}$$
$$= \frac{9.46706\times4.36}{0.025}$$
$$= 1651.05\text{W/m}^2°\text{C}$$

Therefore, the heat transfer coefficient $\boxed{h = 1651.05\text{W/m}^2°\text{C}}$

Hence liquid mercury has largest heat transfer coefficient out of all three given fluids.

Question:

Consider a 25-mm diameter and 15-m long smooth tube that is maintained at a constant surface temperature. Fluids enter the tube at 50°C with a mass flow rate of 0.01 kg/s. Determine the tube surface temperatures necessary to heat water, engine oil, and liquid mercury to the desired outlet temperature of 150°C.

Step 1 of 13

Given data:

Diameter of smooth tube, $D = 0.025\text{m}$

Length of tube, $L = 15\text{m}$

Inlet temperature of fluid, $T_i = 50°\text{C}$

Mass flow rate, $\dot{m} = 0.01\ \text{kg/s}$

Desired outlet temperature, $T_e = 150°\text{C}$

Step 2 of 13

Average temperature of water,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{50+150}{2}$$
$$= 100°\text{C}$$

The properties of water at average temperature,

Thermal conductivity, $k = 0.679\ \text{W/m}\cdot\text{K}$

Specific heat, $c_p = 4217\ \text{J/kg}\cdot\text{K}$

Prandtl number, $\text{Pr} = 1.75$

Kinematic viscosity, $\mu = 0.282\times10^{-3}\ \text{kg/m}\cdot s$

Step 3 of 13

Reynolds number,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$
$$= \frac{4\times0.01}{\pi\times0.025\times\left(0.282\times10^{-3}\right)}$$
$$= 1806$$

As the Reynolds number is less than 2300, the flow is laminar.

Now hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D$$
$$= 0.05\times1806\times0.025$$
$$= 2.257\text{m}$$

Thermal entry length,

$$L_t = \text{Pr}\times L_h$$
$$= 2.257\times1.75$$
$$= 3.951\text{m}$$

As both the values are less than the tube length, the flow is fully developed laminar flow.

Step 4 of 13

Taking Nusselt number value from the table of 'Nusselt numbers and friction factors for fully developed laminar flows', for Circular tube cross section at constant surface temperature, we get $Nu = 3.66$

Heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D}$$
$$= \frac{0.679\times3.66}{0.025}$$
$$= 99.41\ \text{W/m}^2°\text{C}$$

Also the surface area of tube is

$$A_s = \pi DL$$
$$= \pi\times0.025\times15$$
$$= 1.178\ \text{m}^2$$

Step 5 of 13

The exit temperature of the fluid can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left[-\frac{hA_s}{\dot{m}C_p}\right]$$
$$150 = T_s - (T_s - 50)\exp\left[-\frac{99.41\times1.178}{0.01\times4217}\right]$$
$$150 = T_s - (T_s - 50)\exp[-2.7769]$$
$$150 = T_s - 0.062T_s + 3.1$$
$$146.9 = 0.938T_s$$
$$T_s = 156.6°\text{C}$$

Hence the tube surface temperature is $\boxed{T_s = 156.63°\text{C}}$

Step 6 of 13

Average temperature of Engine oil,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{50+150}{2}$$
$$= 100°\text{C}$$

The properties of Engine oil at average temperature,

Thermal conductivity, $k = 0.1367\ \text{W/m}\cdot\text{K}$

Specific heat, $c_p = 2220\ \text{J/kg}\cdot\text{K}$

Prandtl number, $\text{Pr} = 279.1$

Kinematic viscosity, $\mu = 0.01718\ \text{kg/m}\cdot s$

Step 7 of 13

Reynolds number,

Kinematic viscosity, $\mu = 0.01718$ kg/m·s

Step 7 of 13

Reynolds number,

$$\mathrm{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times 0.01}{\pi\times 0.025\times 0.01718}$$

$$= 29.64$$

As the Reynolds number is less than 2300, the flow is laminar.

Now hydrodynamic entry length,

$$L_h = 0.05\times R_e \times D$$

$$= 0.05\times 29.64\times 0.025$$

$$= 0.037\,\mathrm{m}$$

Thermal entry length,

$$L_t = P_r \times L_h$$

$$= 279.1\times 0.037$$

$$= 10.34\,\mathrm{m}$$

As both the values are less than the tube length, the flow is fully developed laminar flow.

Step 8 of 13

Taking Nusselt number value from the table of 'Nusselt numbers and friction factors for fully developed laminar flows', for Circular tube cross section at constant surface temperature, we get $Nu = 3.66$

Heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D}$$

$$= \frac{3.66\times 0.1367}{0.025}$$

$$= 20\ \mathrm{W/m^2{}^\circ C}$$

Also the surface area of tube is

$$A_s = \pi DL$$

$$= \pi\times 0.025\times 15$$

$$= 1.178\,\mathrm{m^2}$$

Step 9 of 13

The exit temperature of the fluid can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left[-\frac{hA_s}{\dot{m}C_p}\right]$$

$$150 = T_s - (T_s - 50)\exp\left(\frac{-20\times 1.178}{0.01\times 2220}\right)$$

$$150 = T_s - (T_s - 50)\exp(-1.06126)$$

$$150 = T_s - 0.346T_s + 17.3$$

$$132.7 = 0.654T_s$$

$$T_s = 202.9^\circ\mathrm{C}$$

Hence the tube surface temperature is $\boxed{T_s = 202.9^\circ\mathrm{C}}$

Step 10 of 13

Average temperature of liquid mercury,

$$T_{avg} = \frac{T_i + T_e}{2}$$

$$= \frac{50+150}{2}$$

$$= 100^\circ\mathrm{C}$$

The properties of liquid mercury at average temperature,

Thermal conductivity, $k = 9.46706\ \mathrm{W/m\cdot K}$

Specific heat, $c_p = 137.1\ \mathrm{J/kg\cdot K}$

Prandtl number, $\mathrm{Pr} = 0.0180$

Kinematic viscosity, $\mu = 1.245\times 10^{-3}\ \mathrm{kg/m\cdot s}$

Step 11 of 13

Reynolds number,

$$\mathrm{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times 0.01}{\pi\times 0.025\times 1.245\times 10^{-3}}$$

$$= 409$$

As the Reynolds number is less than 2300, the flow is laminar.

Now hydrodynamic entry length,

$$L_h = 0.05\times \mathrm{Re}\times D$$

$$= 0.05\times 409\times 0.025$$

$$= 0.511\,\mathrm{m}$$

As the value is less than the tube length, the flow is fully developed laminar flow.

Step 12 of 13

Taking Nusselt number value from the table of 'Nusselt numbers and friction factors for fully developed laminar flows', for Circular tube cross section at constant surface temperature, we get $Nu = 3.66$

Heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D}$$

$$= \frac{3.66\times 9.46706}{0.025}$$

$$= 1385.98\ \mathrm{W/m^2{}^\circ C}$$

Also the surface area of tube is

$$A_s = \pi DL$$

$$= \pi\times 0.025\times 15$$

$= 1385.98\ \mathrm{W/m^2{}^\circ C}$

Also the surface area of tube is

$$A_s = \pi DL$$
$$= \pi \times 0.025 \times 15$$
$$= 1.178\ \mathrm{m^2}$$

Step 13 of 13

The exit temperature of the fluid can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left[-\frac{hA_s}{\dot{m}C_p}\right]$$

$$150 = T_s - (T_s - 50)\exp\left(-\frac{1385.98 \times 1.178}{0.01 \times 137.1}\right)$$

$$150 = T_s - (T_s - 50)\exp[-1190.87]$$

$$150 = T_s - 0$$

$$T_s = 150^\circ C$$

Therefore the tube surface temperature is $\boxed{T_s = 150^\circ C}$

Question: 19.105P

Air at 20°C (1 atm) enters into a 5-mm diameter and 10-cm long circular tube at an average velocity of 5 m/s. The tube wall is maintained at a constant surface temperature of 160°C. Determine the convection heat transfer coefficient and the outlet mean temperature. Evaluate the air properties at 50°C

Step 1 of 7

Given data:

Diameter of circular tube, $D = 0.005\text{m}$

Length of tube, $L = 0.1\text{m}$

Average velocity, $V_{avg} = 5\text{m/s}$

Tube wall surface temperature, $T_s = 160°\text{C}$

Air inlet temperature, $T_i = 20°\text{C}$

Step 2 of 7

The properties of air at 50°C,

Specific heat, $c_p = 1007\ \text{J/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.02735\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Reynolds number is determined by

$$\text{Re} = \frac{V_{avg}D}{\nu}$$
$$= \frac{5\times0.005}{1.798\times10^{-5}}$$
$$= 1390.4$$

As the Reynolds number is less than 2300, the flow is laminar.

Step 3 of 7

The hydrodynamic entry length,

$$L_h = 0.05\times\text{Re}\times D$$
$$= 0.05\times1390.4\times0.005$$
$$= 0.348\text{m}$$

The thermal entry length,

$$L_t = L_h\cdot\text{Pr}$$
$$= 0.348\times0.7228$$
$$= 0.25\ \text{m}$$

The hydrodynamic and thermal entry lengths are more than the total length. So the flow is not a fully developed laminar flow but a hydro-dynamically & thermally developing flow.

Step 4 of 7

From Sieder and Tate equation,

$$Nu = 1.86\left(\frac{\text{RePr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}$$

Where

μ_b – Dynamic viscosity taken at average temperature, 50°C

μ_s – Dynamic viscosity taken at tube surface temperature,

From properties tables, we get the values as

$\mu_b = 1.963\times10^{-5}\ \text{kg/m-s}$

$\mu_s = 2.420\times10^{-5}\ \text{kg/m-s}$

Hence the Nusselt number is

$$Nu = 1.86\left(\frac{1390.4\times0.7228\times0.005}{0.1}\right)^{1/3}\left(\frac{1.963\times10^{-5}}{2.420\times10^{-5}}\right)^{0.14}$$
$$= 1.86\left(\frac{5.0249}{0.1}\right)^{1/3}(0.811157)^{0.14}$$
$$= 6.665$$

Step 5 of 7

Therefore heat transfer coefficient,

$$h = \frac{Nu\cdot k}{D}$$
$$= \frac{6.665\times0.02735}{0.005}$$
$$= 36.46\ \text{W/m}^2°\text{C}$$

Hence the heat transfer coefficient is $\boxed{h = 36.46\ \text{W/m}^2°\text{C}}$

Step 6 of 7

Now heat transfer surface area,

$$A_s = \pi DL$$
$$= \pi\times0.005\times0.1$$
$$= 1.571\times10^{-3}\ \text{m}^2$$

Mass flow rate,

$$\dot{m} = \rho A_c V_{avg}$$
$$= 1.092\times\frac{\pi(0.005)^2}{4}\times5$$
$$= 1.072\times10^{-4}\ \text{kg/s}$$

Step 7 of 7

From the relation for exit temperature,

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}C_p}\right)$$
$$= 160 - (160-20)\exp\left(\frac{-36.46\times1.571\times10^{-3}}{1.072\times10^{-4}\times1007}\right)$$
$$= 160 - 140\exp\left(\frac{-0.057388}{0.10795}\right)$$
$$= 77.728°\text{C}$$

Therefore the outlet mean temperature is $\boxed{T_e = 77.728°\text{C}}$

Question: 19.106P

Air (1 atm) enters into a 5-cm diameter circular tube at 20°C with an average velocity of 5 m/s. The tube wall is maintained at a constant surface temperature of 160°C, and the outlet mean temperature is 80°C. Estimate the length of the tube.

Step 1 of 6

Given data:

Diameter of tube, $D = 0.05\text{m}$

Inlet air temperature, $T_i = 20°\text{C}$

Average velocity of air, $V_{avg} = 5\ \text{m/s}$

Tube wall constant temperature, $T_s = 160°\text{C}$

Outlet air temperature, $T_e = 80°\text{C}$

Step 2 of 6

Average temperature,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{80+20}{2}$$
$$= 50°\text{C}$$

The properties of air at average temperature,

Specific heat, $c_p = 1007\ \text{J/kg-K}$

Thermal conductivity, $k = 0.02735\ \text{W/m°C}$

Dynamic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Kinematic viscosity, $\mu = 1.963\times10^{-5}\ \text{kg/m}\cdot\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Density, $\rho = 1.092\ \text{kg/m}^3$

Step 3 of 6

Reynolds number,

$$\text{Re} = \frac{V_{avg}\times D}{\nu}$$
$$= \frac{5\times0.05}{1.798\times10^{-5}}$$
$$= 13904.34$$

As the Reynolds number is more than 10000, the flow is turbulent.

Now entry lengths,

$$L_h \approx L_t$$
$$= 10D$$
$$= 10\times0.05$$
$$= 0.5\text{m}$$

Step 4 of 6

By assuming that the entry length is shorter than the total flow length, we get the Nusselt number as

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$
$$= 0.023(13904.34)^{0.8}(0.7228)^{0.4}$$
$$= 41.67$$

Now heat transfer coefficient,

$$h = \frac{k\times Nu}{D}$$
$$= \frac{0.02735\times41.67}{0.05}$$
$$= 22.795\ \text{W/m}^2\text{°C}$$

Step 5 of 6

The surface area of the flow,

$$A_s = \pi DL$$
$$= \pi\times0.05\times L$$
$$= 0.15708\times L$$

Mass flow rate,

$$\dot{m} = \rho A_c V_{avg}$$
$$= \rho\left(\frac{\pi D^2}{4}\right)V_{avg}$$
$$= 1.092\times\frac{\pi(0.05)^2}{4}\times5$$
$$= 0.01072\ \text{kg/s}$$

Step 6 of 6

From the relation from energy balance on a control volume,

$$\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{hA_s}{\dot{m}C_p}$$
$$\ln\frac{160-80}{160-20} = -\frac{22.795\times0.15708\times L}{0.01072\times1007}$$
$$-0.5596 = -0.33169\times L$$
$$L = \frac{0.5596}{0.33169}$$
$$L = 1.687\ \text{m}$$

As the entry length (0.5m) is less than the total tube length (1.687m), our assumption is valid. Hence the length of the tube is $\boxed{L = 1.687\text{m}}$

Question: 19.107P

A 20 mm × 20 mm silicon chip is mounted such that the edges are flush in a substrate. The substrate provides an unheated starting length of 20 mm that acts as turbulator. Airflow at 25°C (1 atm) with a velocity of 25 m/s is used to cool the upper surface of the chip. If the maximum surface temperature of the chip cannot exceed 75°C. determine the maximum allowable power dissipation on the chip surface.

Step 1 of 4

Given data:

Dimensions of the silicon chip, $L \times w = 20 \text{ mm} \times 20 \text{ mm}$

Unheated starting length, $\xi = 20 \text{ mm}$

Air flow temperature, $T_\infty = 25°C$

Velocity of flow, $V = 25 \text{ m/s}$

Surface temperature of chip, $T_s = 75°C$

Step 2 of 4

Film temperature of air is determined as

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(75+25)}{2}$$
$$= 50°C$$

From the table, properties of air at 1 atm pressure

At $T = 50°C$

Thermal conductivity, $k = 0.02735 \text{ W/m} \cdot \text{K}$

Kinematic viscosity, $\nu = 1.798 \times 10^{-5} \text{ m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Here in this case, as uniform heat flux is there on chip surface, the maximum surface temperature occurs at trailing edge where heat transfer coefficient is minimum.

The Reynolds number for the flow at the trailing edge $(x = L + \xi)$ is determined from the equation given by

$$\text{Re}_x = \frac{Vx}{\nu}$$
$$= \frac{25 \times \left(\left(20 \times 10^{-3}\right) + \left(20 \times 10^{-3}\right)\right)}{1.798 \times 10^{-5}}$$
$$= 55617.35$$

Since the obtained Reynolds number is less than 5×10^5 i.e. $\text{Re} < 5 \times 10^5$, the flow is said to be laminar.

Step 3 of 4

The average Nusselt number for parallel flow over flat plate is determined from the Churchill & Ozoe relation given by

$$\text{Nu}_x = \frac{\text{Nu}_{x(\xi=0)}}{\left[1 - (\xi / x)^{3/4}\right]^{1/3}}$$
$$= \frac{0.453 \text{Re}^{0.5} \text{Pr}^{1/3}}{\left[1 - (\xi / x)^{3/4}\right]^{1/3}}$$
$$= \frac{0.453 \times (55617.35)^{0.5} \times (0.7228)^{1/3}}{\left[1 - \left(\frac{20 \times 10^{-3}}{0.04}\right)^{3/4}\right]^{1/3}}$$
$$= 129.54$$

The local convective heat transfer coefficient at the trailing edge is determined from the equation given by

$$\text{Nu}_x = \frac{h_x x}{k}$$

Rewrite and solve for h_x is

$$h_x = \frac{\text{Nu}_x \times k}{x}$$
$$= \frac{129.54 \times 0.02735}{0.04}$$
$$= 88.57 \text{ W/m}^2 \cdot \text{K}$$

Step 4 of 4

The maximum allowable power dissipation on the surface of the chip is determined from the equation given by

$$\dot{Q} = h_x A_s (T_s - T_\infty)$$
$$= h_x \times L^2 \times (T_s - T_\infty)$$
$$= 88.57 \times \left(20 \times 10^{-3}\right)^2 \times (75 - 25)$$
$$= 1.77 \text{ W}$$

Therefore, the maximum allowable power dissipation on the surface of the chip is $\boxed{\dot{Q} = 1.77 \text{ W}}$

Question: 19.108P

In the effort to increase the removal of heat from a hot surface at 120°C, a cylindrical pin fin (kf =237 W/m · K) with diameter of 5 mm is attached to the hot surface. Air at 20°C (1 atm) is flowing across the pin fin with a velocity of 10 m/s. Determine the maximum possible rate of heat transfer from the pin fin. Evaluate the air properties at 70°C.

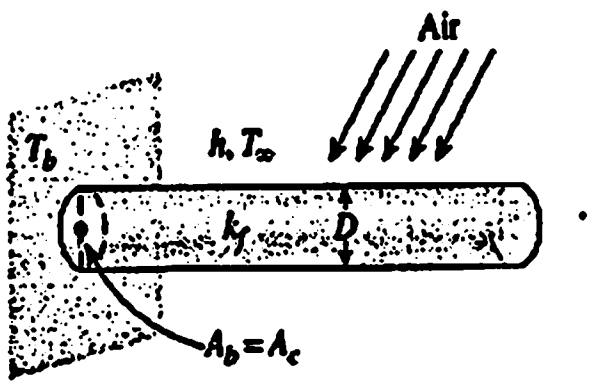


FIGURE P19–108

Step 1 of 5

Given data:

Temperature of hot surface $T_b = 120°C$

Thermal conductivity of fin $k_f = 237\ W/m \cdot K$

Diameter of fin $D = 5mm = 0.005m$

Outside air temperature $T_\infty = 20°C$

Velocity of air flow $V = 10\ m/s$

Step 2 of 5

Film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{120 + 20}{2}$$

$$T_f = 70°C$$

By taking properties of air at $T_f = 70°C$

From the properties of air table,

Thermal conductivity $k = 0.02881\ W/m \cdot K$

Density $\rho = 1.028\ kg/m^3$

Specific heat $c_p = 1007\ J/kg \cdot K$

Dynamic viscosity $\mu = 2.052 \times 10^{-5}\ N \cdot s/m^2$

Kinematic viscosity $\nu = 1.995 \times 10^{-5}\ m^2/s$

Prandtl number $Pr = 0.7177$

Step 3 of 5

Calculating the Reynolds number

$$Re_D = \frac{VD}{\nu}$$

$$= \frac{10 \times 0.005}{1.995 \times 10^{-5}}$$

$$Re_D = 2506.265$$

Now as we know Churchill and Bernstein relation for the Nusselt number calculation

$$Nu_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{\left[1 + (0.4/Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5}$$

$$\frac{h \times 0.005}{0.02881} = 0.3 + \frac{0.62(2506.3)^{1/2}(0.7177)^{1/3}}{\left[1 + (0.4/0.7177)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{2506.3}{282000}\right)^{5/8}\right]^{4/5}$$

$$h \times 0.1735 = 0.3 + \frac{27.79}{1.138} \times 1.0415$$

$$h = 148.319\ W/m^2 \cdot K$$

Step 4 of 5

As per the requirement of the statement we need to calculate the maximum rate of heat transfer from pin fin and we know maximum heat transfer occurs when the fin is infinitely long.

$$Q_{infinite} = \sqrt{hpk_f} A_c (T_b - T_\infty) \qquad \ldots\ldots (1)$$

Where perimeter of the surface

$$p = \pi D$$

$$= \pi \times 0.005$$

$$p = 0.0157m$$

Cross sectional area of the pin fin is

$$A_c = \pi D^2/4$$

$$= \frac{\pi}{4} \times 0.005^2$$

$$A_c = 1.96 \times 10^{-5}\ m^2$$

Step 5 of 5

Now substituting the values in the equation (1)

$$Q_{infinite} = \sqrt{hpk_f} A_c (T_b - T_\infty)$$

$$= \sqrt{148.28 \times 0.0157 \times 237 \times 1.96 \times 10^{-5}} (120 - 20)$$

$$Q_{infinite} = 10.399\ W$$

Therefore, the maximum possible rate of heat transfer from the pin fin is

$$\boxed{Q_{infinite} = 10.399\ W}$$

Question: 19.109P

The passenger compartment of a minivan traveling at 70 mph can be modeled as a 3.2-ft-high, 6-ft-wide, and 12-ft-long rectangular box whose walls have an insulating value of *R*-3 (i.e., a wall thickness-to-thermal conductivity ratio of 3 h·ft2·°F/Btu). The interior of a minivan is maintained at an average temperature of 70°F during a trip at night while the outside air temperature is 90°F.

The average heat transfer coefficient on the interior surfaces of the van is 1.2 Btu/h·ft2·°F. The air flow over the exterior surfaces can be assumed to be turbulent because of the intense vibrations involved, and the heat transfer coefficient on the front and back surfaces can be taken to be equal to that on the top surface. Disregarding any heat gain or loss by radiation, determine the rate of heat transfer from the ambient air to the van.

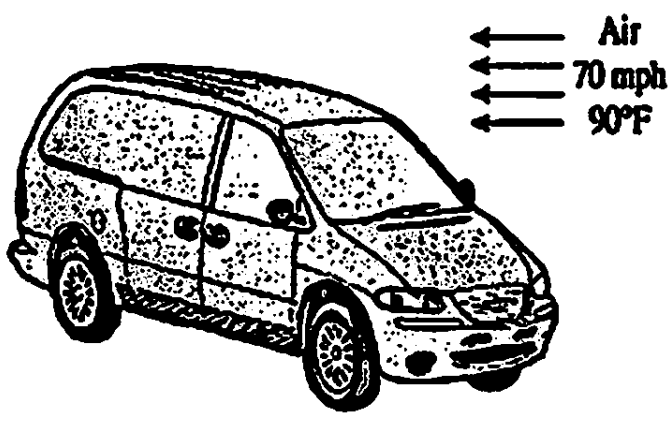


FIGURE P19–109E

Step 1 of 6

Average film temperature,

$$T_{avg} = \frac{70+90}{2}$$
$$= 80°F$$

From properties of air table at 1 atm and 80°F,
Thermal conductivity, $k = 0.01481$ Btu/h·ft·°F
Kinematic viscosity, $\upsilon = 0.1697\times10^{-3}$ ft^2/s
Prandtl number, Pr = 0.7290

Step 2 of 6

Reynolds number,

$$\mathrm{Re}_L = \frac{VL}{\upsilon}$$

Where
V is the velocity of the van,
L is the length of the van and
ν is the kinematic viscosity.
Substitute 70 mph for V, 12 ft for L and 0.1697×10^{-3} ft^2/s for ν in the above equation,

$$\mathrm{Re}_L = \frac{\left(70\,\mathrm{mph}\times\frac{5288\,\mathrm{ft/s}}{3600\,\mathrm{mph}}\right)(12\,\mathrm{ft})}{0.1697\times10^{-3}\ \mathrm{ft^2/s}}$$
$$= 7.27\times10^{6}$$

This value is greater than the value of critical Reynolds number. Therefore turbulent flow will exist.

Step 3 of 6

Calculate the Nusselt number for the given system,

$$\mathrm{Nu} = \frac{h_0 L}{k}$$
$$= 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{1/3}$$
$$= 0.037\left(7.27\times10^{6}\right)^{0.8}(0.7290)^{1/3}$$
$$= 10284.20$$

Calculate the hence heat transfer coefficient,

$$h_0 = \frac{k}{L}\mathrm{Nu}$$

Where,
k is the thermal coefficient.

Substitute $(0.01481\,\mathrm{Btu/ft\cdot°F})$ for k and the respective values in the above equation,

$$h_0 = \frac{0.01481}{12}\times10284.20$$
$$= 12.69\ \mathrm{Btu/h\cdot ft^2\cdot °F}$$

Step 4 of 6

Calculate the surface area of the van,

$$A_s = 2\left[(3.2\times6)+(6\times12)+(12\times3.2)\right]$$
$$= 259.20\,\mathrm{ft^2}$$

Calculate the thermal resistances for system,

$$R_i = \frac{1}{h_i A_s}$$

Where,
h_i is the average heat transfer co-efficient on the interior surfaces of the van.

Substitute $(1.2\,\mathrm{Btu/h\cdot ft^2\cdot °F})$ for h_i and the respective values in the above equation,

$$R_i = \frac{1}{1.2\times259.20}$$
$$= 0.00321\ \mathrm{h\cdot °F/Btu}$$

Step 5 of 6

Calculate the insulation thermal resistances for system,

$$R_{ins} = \frac{(R-3)_{value}}{A_s}$$
$$= \frac{3}{259.20}$$
$$= 0.01157\ \mathrm{h\cdot °F/Btu}$$
$$R_0 = \frac{1}{h_0 A_s}$$
$$= \frac{1}{(12.69)(259.20)}$$
$$= 0.00030\ \mathrm{h\cdot °F/Btu}$$

Step 6 of 6

Calculate the total resistance,

$$R_{total} = R_i + R_{ins} + R_0$$
$$= 0.00321+0.01157+0.00030$$
$$= 0.0150\ \mathrm{h\cdot °F/Btu}$$

Calculate the heat transfer rate,

$$\dot{Q} = \frac{T_{air} - T_{van}}{R_{total}}$$

Where,
T_{air} is the temperature of the air and
T_{van} is the temperature of the van.

Substitute 90°F for T_{air}, 70°F for T_{van} and the respective values in the above equation,

$$\dot{Q} = \frac{90-70}{0.0150}$$
$$= 1333.34\ \mathrm{Btu/h}$$

Therefore, heat transferred from ambient air to van is **1333.34 Btu/h**

Question: 19.110P

Consider a house that is maintained at a constant temperature of 22°C. One of the walls of the house has three single-pane glass windows that are 1.5 m high and 1.8 m long. The glass (k =0.78 W/m.K) is 0.5 cm thick, and the heat transfer coefficient on the inner surface of the glass is 8 W/m2·K. Now winds at 35 km/h start to blow parallel to the surface of this wall. If the air temperature outside is −2°C, determine the rate of heat loss through the windows of this wall. Assume radiation heat transfer to be negligible.

Step 1 of 6

Given

Spectral transmissivity of a glass cover can be expressed as

$\tau_1 = 0$ for $\lambda < 0.3\mu m$

$\tau_2 = 0.9$ for $0.3 \le \lambda \le 3\mu m$

$\tau_3 = 0$ for $\lambda > 3\mu m$

Rate of solar radiation, $I = 950\ W/m^2$

Temperature of absorber plate, $T_a = 340K$

Step 2 of 6

(a) The solar flux incident on the absorber plate at temperature is $T = 5800K$ is calculated as follows

$\lambda_1 T = 0.3 \times 5800$

$= 1740\ \mu mK$

$\lambda_2 T = 3 \times 5800$

$= 17400\ \mu mK$

From table, Black body radiation functions

The blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined as

$f_{\lambda_1} = 0.033454$

$f_{\lambda_2} = 0.978746$

Step 3 of 6

The absorptivity of the surface becomes

$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \tau_3 \left(1 - f_{\lambda_2}\right)$$

$$= 0.0 \times 0.033454 + 0.9(0.978746 - 0.033454) + 0.0(1 - 0.978746)$$

$$= 0 + 0.851 + 0$$

$$= 0.851$$

The rate of absorption of solar radiation is given by

$$E_{absorbed} = \alpha I$$

$$= 0.851 \times 950$$

$$= 808.45\ W/m^2$$

Rate of absorption of solar radiation, $\boxed{E_{absorbed} = 808.45\ W/m^2}$

Step 4 of 6

(b) The transmissivity of the glass cover for radiation emitted by the absorber plate temperature at $T_a = 340K$ is determined as follows

$\lambda_1 T_a = 0.3 \times 300$

$= 90\ \mu mK$

$\lambda_2 T_a = 3 \times 300$

$= 900\ \mu mK$

From table, Black body radiation functions

The blackbody radiation functions corresponding to $\lambda_1 T_a$ and $\lambda_2 T_a$ are determined as

$f_{\lambda_1} = 0.0$

$f_{\lambda_2} = 0.000169$

Step 5 of 6

Then, the absorptivity of the surface becomes

$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \tau_3 \left(1 - f_{\lambda_2}\right)$$

$$= 0.0 \times 0.0 + 0.9(0.000169 - 0.0) + 0.0(1 - 0.000169)$$

$$= 0 + 0.0001521 + 0$$

$$= 0.00015$$

The absorptivity of the surface, $\boxed{\tau(T) = 0.00015}$

Step 6 of 6

(c) The rate of heat transfer to the cooling water if the glass temperature is $T_a = 340K$ is determined as follows

Rate of heat transfer is given by

$$\dot{Q}_{water} = (\tau_{solar} - \tau_{room})I$$

$$= (0.851 - 0.00015)950$$

$$= 0.85085 \times 950$$

$$= 808.31\ W/m^2$$

Rate of heat transfer to the cooling water, $\boxed{\dot{Q}_{water} = 808.31\ W/m^2}$

Question: 19.111P

Consider a person who is trying to keep cool on a hot summer day by turning a fan on and exposing his body to air flow. The air temperature is 32°C, and the fan is blowing air at a velocity of 5 m/s. The surrounding surfaces are at 40°C. and the emissivity of the person can be taken to be 0.9. If the person is doing light work and generating sensible heat at a rate of 90 W, determine the average temperature of the outer surface (skin or clothing) of the person. The average human body can be treated as a 30-cm-diameter cylinder with an exposed surface area of 1.7 m2.

Step 1 of 2

Given

Temperature of wall, $T_w = 500\ K$

Temperature reading of thermo couple, $T_{th} = 850\ K$

Emissivity of the thermocouple junction, $\varepsilon = 0.6$

Coefficient of convection heat transfer, $h = 60\ W/m^2$

Step 2 of 2

Temperature of the air is calculated as

$$T_f = T_{th} + \frac{\varepsilon\sigma\left(T_{th}^4 - T_w^4\right)}{h}$$

$$= 850 + \frac{0.6\left(5.67\times10^{-8}\right)\left(850^4 - 500^4\right)}{60}$$

$$= 850 + 260.54$$

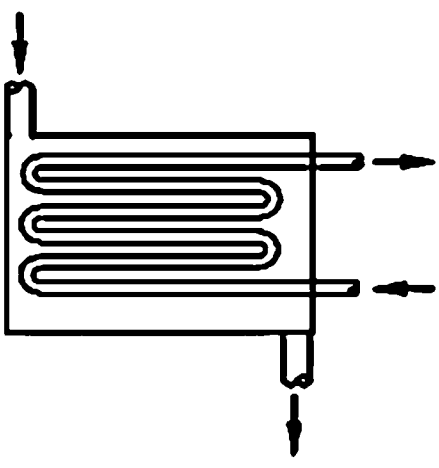
$$= 1110.54\,K$$

Temperature of the air, $\boxed{T_f = 1110.54\,K}$

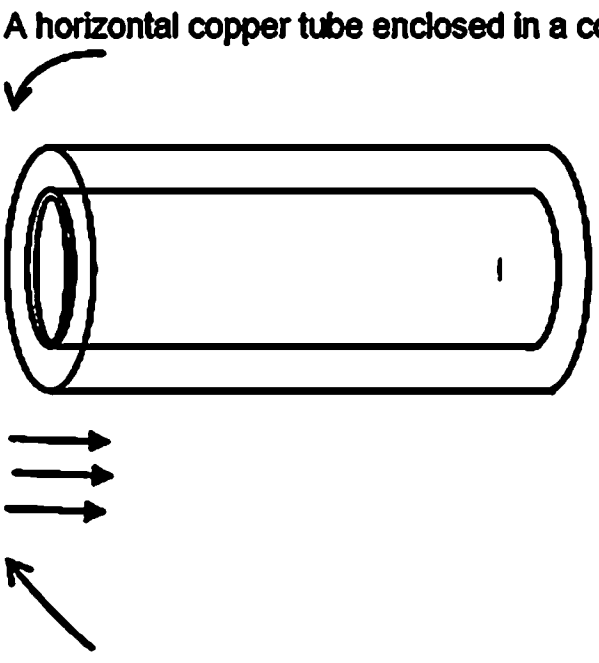
Question: 19.112P pg 1

A 0.5-cm-thick, 12-cm-high, and 18-cm-long circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. The board is impregnated with copper fillings and has an effective thermal conductivity of 16 W/m-K. All the heat generated in the chips is conducted across the circuit board and is dissipated from the back side of the board to the ambient air at 30°C, which is forced to flow over the surface by a fan at a free-stream velocity of 300 m/min. Determine the temperatures on the two sides of the circuit board.

Step 1 of 8



A horizontal copper tube enclosed in a concentric thin glass tube is as shown in figure,



Step 2 of 8

Calculate the average temperature

$$T_f = \frac{(T_1 + T_2)}{2}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Substitute $60°C$ for T_1, $40°C$ for T_2

$$\begin{aligned} T_{avg} &= \frac{T_1 + T_2}{2} \\ &= \frac{60+40}{2} \\ &= 50°C \end{aligned}$$

From table properties of air at 1atm and the average temperature $T_{avg} = 50°C$ the properties of air are

$$\text{Thermal conductivity}, k = 0.02735\ \text{W/m}\cdot\text{K}$$
$$\text{kinematic viscosity}, \nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Prandtle number}, \text{Pr} = 0.7228$$

Step 3 of 8

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$$1K = 1°C + 273$$

Find the Coefficient of volume expansion

$$\begin{aligned} \beta &= \frac{1}{T_{avg}} \\ &= \frac{1}{(50+273)} \\ &= 3.0959\times10^{-3}\ K^{-1} \end{aligned}$$

Calculate the characteristic length in this case is the distance between the two cylinders,

$$L_c = \frac{(D_2 - D_1)}{2}$$

Here,

Outer diameter of the solar collector, D_1

Diameter of the glass tube, D_2

Substitute $5\,\text{cm}$ for D_1, $12\,\text{cm}$ for D_2

$$\begin{aligned} L_c &= \frac{(D_2 - D_1)}{2} \\ &= \frac{(0.12-0.05)}{2} \\ &= 0.035\,\text{m} \end{aligned}$$

Step 4 of 8

Calculate the Rayleigh number

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}\text{Pr}$$

Here,

Acceleration due to gravity is g

Substitute $3.0959\,\text{K}^{-1}$ for β, $60°C$ for T_1, $40°C$ for T_2, $0.035\,\text{m}$ for L, $9.81\,\text{m/s}^2$ for g, $1.798\times10^{-5}\ \text{m}^2/\text{s}$ for ν

$$\frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}$$

Acceleration due to gravity is g

Substitute $3.0959\,K^{-1}$ for β, $60°C$ for T_1, $40°C$ for T_2, $0.035\,m$ for L, $9.81\,m/s^2$ for g, $1.798\times10^{-5}\ m^2/s$ for ν

$$Ra = \frac{g\beta(T_1-T_2)L_c^3}{\nu^2}Pr$$

$$= \frac{9.81(m/s^2)\times3.0959\times10^{-3}(K^{-1})\times(60-40)(K)\times(0.035)^3(m^3)}{(1.798\times10^{-5}\,m^2/s)^2}\times0.7228$$

$$= 58229$$

Calculate the effective thermal conductivity

$$F_{cyc} = \frac{\left[\ln\left(\frac{D_2}{D_1}\right)\right]^4}{L_c^3\left(D_1^{-3/5}+D_2^{-3/5}\right)^5}$$

$$= \frac{\left[\ln\left(\frac{0.12}{0.05}\right)\right]^4}{(0.035)^3\left[(0.05\,m)^{-3/5}+(0.12\,m)^{-3/5}\right]^5}$$

$$= \frac{0.5874}{3.5008}$$

$$= 0.1678$$

Step 5 of 8

Calculate the Further calculation,

$$k_{eff} = 0.386k\left(\frac{Pr}{0.861+Pr}\right)^{1/4}\left(F_{cyc}Ra_L\right)^{1/4}$$

Substitute 0.7228 for Pr, 0.1678 for F_{cyl}

$$k_{eff} = 0.386k\left(\frac{Pr}{0.861+Pr}\right)^{1/4}\left(F_{cyc}Ra_L\right)^{1/4}$$

$$= 0.386\times0.02735(W/m\cdot K)\times\left(\frac{0.7228}{0.861+0.7228}\right)^{1/4}\times(0.1678\times58229)^{1/4}$$

$$= 0.01055\times0.8219\times9.942$$

$$= 0.086207\ W/m\cdot K$$

Step 6 of 8

Calculate the rate of heat transfer between the cylinders by convection

$$\dot{Q}_{conv} = \frac{2\pi k_{eff}}{\ln\left(\frac{D_2}{D_1}\right)}(T_1-T_2)$$

$$= \frac{2\pi\times0.08620}{\ln\left(\frac{12}{5}\right)}(60-40)$$

$$= 12.374\,W$$

Step 7 of 8

Calculate the rate of heat transfer by radiation is

$$\dot{Q}_{rad} = \frac{\sigma A_1\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)}$$

Here,

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Emissivity of the absorber plate, ε_1

Emissivity of the glass surface, ε_2

Substitute, $333\,K$ for T_1, $313\,K$ for T_2, 0.85 for ε_1, $5.67\times10^{-8}\,W/m^2K^4$ for σ, 0.9 for ε_2

$$\dot{Q}_{rad} = \frac{\sigma A_1\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)}$$

$$= \frac{5.67\times10^{-8}(W/m^2\text{-}K^4)\times\pi\times0.05(m)\times1(m)\times(333^4-313^4)}{\frac{1}{0.85}+\frac{1-0.9}{0.9}\left(\frac{5}{12}\right)}$$

$$= \frac{24.033}{1.2227}$$

$$= 19.655\,W$$

Step 8 of 8

Calculate the total rate of heat loss from the glass cover becomes

$$\dot{Q}_{total} = \dot{Q}_{conv}+\dot{Q}_{rad}$$

$$= 12.374(W)+19.655(W)$$

$$= 32.029\,W$$

Therefore, the rate of heat loss from the collector by natural convection and radiation per meter length of the tube is $\boxed{32.029\,W}$

Question: 19.113P

A 3-m-internal-diameter spherical tank made of 1-cm-thick stainless steel (k= 15 W/m·K) is used to store iced water at 0°C. The tank is located outdoors at 30°C and is subjected to winds at 25 km/h. Assuming the entire steel tank to be at 0°C and thus its thermal resistance to be negligible, determine (*a*) the rate of heat transfer to the iced water in the tank and (*b*) the amount of ice at 0°C that melts during a 24-h period. The heat of fusion of water atl atmospheric pressure is *hif* = 333.7 kJ/kg. Disregard any heat transfer by radiation.

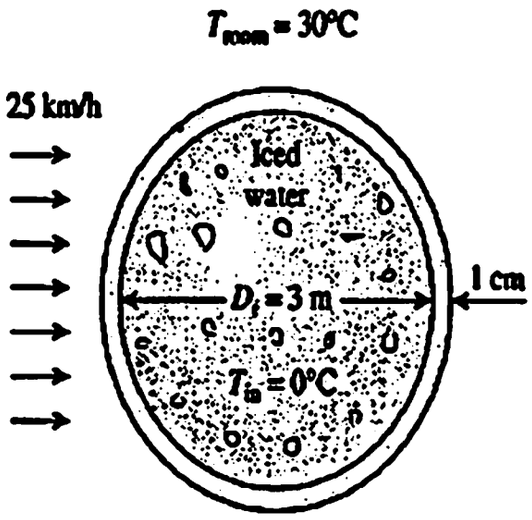


FIGURE P19–113

Step 1 of 2

Given

Height of the cube, $h = 8$ in

Length of the cube, $L = 12$ in

Width of the cube is, $w = 12$ in

Surface temperature of the box, $T_s = 130\ °F$

Emissivity of the outer surface of the box, $\varepsilon = 0.95$

Power dissipation from the box, $\dot{Q} = 90$ W

Step 2 of 2

Surface area of the electronic box,

$$A_s = 4\times(8\times12)+(12\times12)$$
$$= 528\text{in}^2$$
$$= 3.67\text{ft}^2$$

Radiation heat transfer is given by

$$\dot{Q}_{rad} = \varepsilon A_s \sigma\left(T_s^4 - T_{surr}^4\right)$$

$$\Rightarrow 90\times3.41 = 0.95\times3.67\times0.1714\times10^{-8}\left(590^4 - T_{surr}^4\right)$$

$$\Rightarrow 590^4 - T_{surr}^4 = 5.136\times10^{10}$$

$$\Rightarrow T_{surr}^4 = 6.98\times10^{10}$$

$$\Rightarrow T_{surr} = 514\text{R}$$
$$= 54°\text{F}$$

Highest surrounding temperature, $\boxed{T_{surr} = 54°\text{F}}$

Question: 19.114P

Repeat Prob. 19-113, assuming the inner surface of the tank to be at 0°C but by taking the thermal resistance of the tank and heat transfer by radiation into consideration. Assume the average surrounding surface temperature for radiation exchange to be 25°C and the outer surface of the tank to have an emissivity of 0.75.

Step 1 of 4

Assume outer surface of the inner tank to be surface 1 and inner surface of outer tank to be surface 2.

Find the inner diameter of the spherical tank,

$$D_1 = D_i + 2t$$

Here, D_i is inner diameter of the sphere and t is thickness of the wall.

Substitute 2 m for D_i and 0.005 m for t

$$\begin{aligned} D_1 &= 2 + 2\times 0.005 \\ &= 2.01\,\text{m} \end{aligned}$$

Find the inner diameter of the outer tank,

$$D_2 = D_1 + 2l_{air}$$

Here, l_{air} is thickness of air space between two walls.

Substitute 2.01 m for D_1 and 0.015 m for l_{air}

$$\begin{aligned} D_2 &= 2.01 + 2\times 0.015 \\ &= 2.04\,\text{m} \end{aligned}$$

Find the inner tank outer surface area of the sphere.

$$A_1 = \pi D_1^2$$

Substitute 2.01 m for D_1

$$\begin{aligned} A_1 &= \pi \times 2.01^2 \\ &= 12.69\,\text{m}^2 \end{aligned}$$

Step 2 of 4

(a)

Find the rate of heat transfer to the iced water in the spherical tank by using the following equation:

$$\dot{Q}_{12} = \frac{A_1\sigma\left(T_2^4 - T_1^4\right)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)^2}$$

Here, σ is Stefan Boltzmann's constant, $\varepsilon_1, \varepsilon_2$ are the emissivity's of the inner and outer surfaces, T_1 is the inner wall temperature, and T_2 is the outer wall temperature.

Substitute $12.69\,\text{m}^2$ for A_1, 0.15 for $\varepsilon_1, \varepsilon_2$, $5.67\times10^{-8}\,\text{W/m}^2.\text{K}^4$ for σ, 2.01 m for D_1, 2.04 m for D_2, $(20+273)\,\text{K}$ for T_2, and $(0+273)\,\text{K}$ for T_1.

$$\begin{aligned} \dot{Q}_{12} &= \frac{12.69\times5.67\times10^{-8}\left(293^4 - 273^4\right)}{\frac{1}{0.15} + \frac{1-0.15}{0.15}\left(\frac{2.01}{2.04}\right)^2} \\ &= \frac{1306.2788}{12.00728} \\ &= 108.79\,\text{W} \end{aligned}$$

Therefore, heat transfer rate to the iced water is $\boxed{108.79\,\text{W}}$.

Step 3 of 4

(b)

Find the amount of ice melts at $0°\text{C}$ during a 24-h period.

$$Q = \dot{Q}\Delta t$$

Here, t is the time period.

Substitute 108.79 W for $\dot{Q}$ and $(24\times3600)\,\text{s}$ for t.

$$\begin{aligned} Q &= 108.79\times(24\times3600) \\ &= 9399456\,\text{J} \\ &= 9399.456\,\text{kJ} \end{aligned}$$

Step 4 of 4

Find the amount of ice melts during the 24 hour period.

$$Q = mL_{ice}$$

Here, m is amount of ice and L_{ice} is latent heat of ice.

Substitute $9399.456\,\text{kJ}$ for Q and 334 kJ/kg for L_{ice}

$$\begin{aligned} 9399.456 &= m\times334 \\ m &= 28.14\,\text{kg} \end{aligned}$$

Therefore, the amount of ice melts in 24 h period is $\boxed{28.14\,\text{kg}}$.

Question: 19.115P

A transistor with a height of 0.32 in and a diameter of 0.22 in is mounted on a circuit board. The transistor is cooled by air flowing over it at a velocity of 600 ft/min. If the air temperature is 120°F and the transistor case temperature is not to exceed 180°F, determine the amount of power this transistor can dissipate safely.

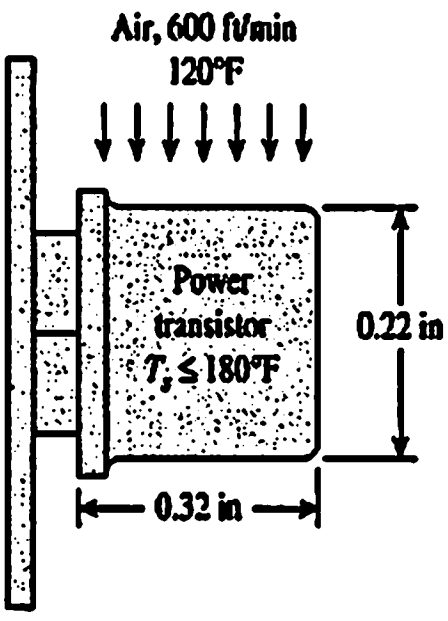


FIGURE P19–115E

Step 1 of 4

Given data:

Height of the transistor, $H = 0.32$ in.$=0.027$ft

Diameter of the transistor, $D = 0.22$ in.$=0.0183$ft

Air flow velocity, $V = 600$ ft/min

$= 10$ ft/s

Temperature of the air, $T_\infty = 120°F$

Temperature of transistor, $T_s = 180°F$

Step 2 of 4

The film temperature is determined as

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(180+120)}{2}$$

$$= 150°F$$

From the table, properties of air at 1 atm pressure

At $T_f = 150°F$

Thermal conductivity, $k = 0.01646$ Btu/h·ft·R

Kinematic viscosity, $\nu = 2.099\times10^{-4}$ ft^2/s

Prandtl number, $\Pr = 0.7188$

Step 3 of 4

The Reynolds number for the flow is determined as

$$\mathrm{Re} = \frac{VD}{\nu}$$

$$= \frac{10\times0.0183}{2.099\times10^{-4}}$$

$$= 871.84$$

The Nusselt number is determined from the relation given by

$$\mathrm{Nu}_{cyl} = 0.3 + \frac{0.62\mathrm{Re}^{0.5}\mathrm{Pr}^{1/3}}{\left[1+\left(\frac{0.4}{\mathrm{Pr}}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62\times(871.84)^{0.5}\times(0.7188)^{1/3}}{\left[1+\left(\frac{0.4}{0.7188}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{871.84}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 15.02$$

Step 4 of 4

The convective heat transfer coefficient is determined from the equation given by

$$\mathrm{Nu} = \frac{hD}{k}$$

Rewrite and solve for h:

$$h = \frac{\mathrm{Nu}_{cyl}\times k}{D}$$

$$= \frac{15.02\times0.01646}{0.0183}$$

$$= 13.51 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

The amount of power dissipated by the transistor is determined from the equation given by

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= h\times(\pi DH)\times(T_s - T_\infty)$$

$$= 13.51\times(\pi\times0.0183\times0.027)\times(180-120)$$

$$= 1.26 \text{ Btu/h}$$

Therefore, the amount of power dissipated by the transistor is $\boxed{\dot{Q} = 1.26 \text{ Btu/h}}$

Question: 19.116P pg 1

The roof of a house consists of a 15-cm-thick concrete slab (k =2 W/m·K) that is 15 m wide and 20 m long. The convection heat transfer coefficient on the inner surface of the roof is 5 W/m2·K. On a clear winter night, the ambient air is reported to be at 10°C, while the night sky temperature is 100 K. The house and the interior surfaces of the wall are maintained at a constant temperature of 20°C. The emissivity of both surfaces of the concrete roof is 0.9. Considering both radiation and convection heat transfer, determine the rate of heat transfer through the roof when wind at 60 km/h is blowing over the roof.

If the house is heated by a furnace burning natural gas with an efficiency of 85 percent, and the price of natural gas is $1.20/therm, determine the money lost through the roof that night during a 14-h period.

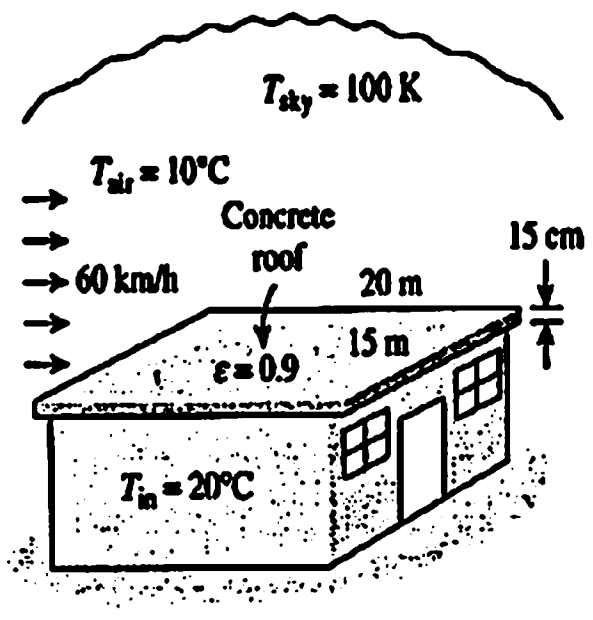


FIGURE P19–116

Step 1 of 6

Schematic diagram of solar system:

Step 2 of 6

Calculate the mean film temperature is

$$T_f = \frac{(T_1 + T_2)}{2}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Substitute $80°C$ for T_1, $32°C$ for T_2

$$T_f = \frac{(T_1 + T_2)}{2}$$
$$= \frac{(80+32)}{2}$$
$$= 56°C$$

From the table the properties of air, at 1 atm and the film temperature of are

$$\text{Thermal conductivity}, k = 0.0277\ \text{W/m}\cdot\text{K}$$
$$\text{kinematic viscosity}, \nu = 1.857\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Prandtle number}, \text{Pr} = 0.7212$$

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$$1K = 1°C + 273$$

Calculate the coefficient of volume expansion (β) is

$$\beta = \frac{1}{(56+273)}$$
$$= 0.003041\,K^{-1}$$

For $\theta = 20°$, we have horizontal rectangular enclosure. The characteristic length in this case is the distance between the two glasses $L_c = L = 0.03\,\text{m}$

Step 3 of 6

Calculate the Rayleigh number is

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2}\text{Pr}$$

Here,

Acceleration due to gravity is g

Substitute $0.003041\,K^{-1}$ for β, $80°C$ for T_1, $32°C$ for T_2 $0.03\,\text{m}$ for L, $9.81\,\text{m/s}^2$ for g, $1.857\times10^{-5}\ \text{m}^2/\text{s}$ for ν

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2}\text{Pr}$$
$$= \frac{9.81(\text{m}^2/\text{s})\times0.003041(\text{K}^{-1})\times(80\text{-}32)(\text{K})\times0.03^3(\text{m}^3)}{\left(1.857\times10^{-5}\ \text{m}^2/\text{s}\right)^2}\times0.7212$$
$$= 80857.878$$

Step 4 of 6

Calculate the Nusselt number correlation for the Natural convention is

$$\text{Nu} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\theta}\right]\times\left[1 - \frac{1708(\sin1.8\theta)^{1.6}}{Ra\cos\theta}\right] + \left[\frac{(Ra\cos\theta)^{1/3}}{18} - 1\right]$$

Here,

Tilted angle of solar collector is θ

Substitute $20°$ for θ, 80857.878 for Ra

$$\text{Nu} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\theta}\right]\times\left[1 - \frac{1708(\sin1.8\theta)^{1.6}}{Ra\cos\theta}\right] + \left[\frac{(Ra\cos\theta)^{1/3}}{18} - 1\right]$$
$$= 1 + 1.44\left[1 - \frac{1708}{}\right]\times\left[1 - \frac{1708(\sin1.8\times20)^{1.6}}{}\right.$$

$$Nu = 1+1.44\left[1-\frac{1708}{Ra\cos\theta}\right]\times\left[1-\frac{1708(\sin 1.8\theta)^{1.6}}{Ra\cos\theta}\right]+\left[\frac{(Ra\cos\theta)^{1/3}}{18}-1\right]$$

$$=1+1.44\left[1-\frac{1708}{80857.878\times\cos 20}\right]\times\left[1-\frac{1708(\sin 1.8\times 20)^{1.6}}{80857.878\times\cos 20}\right]$$

$$+\left[\frac{(80857.878\times\cos 20)^{1/3}}{18}-1\right]$$

$$=1+1.44\times 0.9775\times 0.9904+1.353$$

$$=3.747$$

Calculate the Surface is

$$A_s = H\times W$$

Here,

Height of the solar collector, H

Width of the solar collector, W

Substitute $1.5\,\text{m}$ for H , 3m for W

$$A_s = H\times W$$

$$=1.5(\text{m})\times 3(\text{m})$$

$$=4.5\,\text{m}^2$$

Since Nusselt number is

$$Nu = \frac{hL}{k}$$

$$h = \frac{k\,Nu}{L}$$

Here,

Heat transfer coefficient is h

Step 5 of 6

Calculate the rate of heat transfer due to natural convection

$$\dot{Q}_{conv} = hA_s(T_1-T_2)$$

$$=\frac{k\text{Nu}}{L}A_s(T_1-T_2)$$

Substitute $0.0277\ \text{W/m}\cdot\text{K}$ for k , 3.747 for Nu

$$\dot{Q}_{conv} = hA_s(T_1-T_2)$$

$$=\frac{kNu}{L}A_s(T_1-T_2)$$

$$=\frac{0.02779(\text{W/m}\cdot\text{K})\times 3.747}{0.03(\text{m})}\times 4.5(\text{m}^2)\times(80-32)$$

$$=750\,\text{W}$$

Therefore, the rate of heat transfer due to natural convection is $\boxed{750\,\text{W}}$

Step 6 of 6

Calculate the rate of heat transfer due to radiation

$$\dot{Q}_{rad} = \frac{\sigma A_2\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Emissivity of the absorber plate, ε_1

Emissivity of the glass surface, ε_2

Substitute $80°\text{C}$ for T_1 , $32°\text{C}$ for T_2 , 0.8 for ε_1 , 0.9 for ε_2

$$\dot{Q}_{rad} = \frac{\sigma A_2\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$

$$=\frac{5.67\times 10^{-8}(\text{W/m}^2\cdot\text{K}^4)\times 4.5(\text{m}^2)\times(353^4-305^4)}{\frac{1}{0.9}+\frac{1}{0.8}-1}$$

$$=1289\ \text{W}$$

Therefore, rate of heat transfer due to radiation is $\boxed{1289\,\text{W}}$

Question: 19.117P pg 1

Steam at 250°C flows in a stainless steel pipe (k =15 W/m•K) whose inner and outer diameters are 4 cm and 4.6 cm, respectively. The pipe is covered with 3.5-cm-thick glass wool insulation (k = 0.038 W/m•K) whose outer surface has an emissivity of 0.3. Heat is lost to the surrounding air and surfaces at 3°C by convection and radiation. Taking the heat transfer coefficient inside the pipe to be 80 W/m2•K, determine the rate of heat loss from the steam per unit length of the pipe when air is flowing across the pipe at 4 m/s.

Step 1 of 7

Outer diameter of the insulated pipe is determined as,

$$D_3 = D_2 + 2t$$

Here, D_2 is outer diameter of the pipe and t is thickness of insulation

Substitute 4.6 cm for D_2 and 3.5 cm for t

$$D_3 = 4.6 + (2 \times 3.5)$$
$$= 11.6 \text{ cm}$$

Assume the outside surface temperature of the pipe $T_s = 12°C$

Calculate the film temperature of air by using the equation,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

Here, T_∞ is temperature of the surrounding air

$$T_f = \frac{(12+3)}{2}$$
$$= 7.5°C$$

From the table, properties of air at 1 atm pressure and $T_f = 7.5°C$ by using interpolation

Thermal conductivity,

$$k = \frac{0.02439 + 0.02401}{2}$$
$$= 0.0242 \text{ W/m·K}$$

Kinematic viscosity,

$$\nu = \frac{(1.382 + 1.426) \times 10^{-5}}{2}$$
$$= 1.404 \times 10^{-5} \text{ m}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = \frac{0.7350 + 0.7336}{2}$$
$$= 0.7343$$

Step 2 of 7

Calculate the Reynolds number of the flow by using the equation,

$$\text{Re} = \frac{VD_t}{\nu}$$

Here, V is velocity of air

Substitute 4 m/s for V

$$\text{Re} = \frac{4 \times \left(11.6 \times 10^{-2}\right)}{1.404 \times 10^{-5}}$$
$$= 33048.43$$

Find the Nusselt number by using the equation,

$$\text{Nu}_{cyl} = 0.3 + \frac{0.62\text{Re}^{0.5}\text{Pr}^{1/3}}{\left[1 + \left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62 \times (33048.43)^{0.5} \times (0.7343)^{1/3}}{\left[1 + \left(\frac{0.4}{0.7343}\right)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{33048.43}{282000}\right)^{5/8}\right]^{4/5}$$

$$= 107.79$$

Step 3 of 7

Calculate the convective heat transfer coefficient by using the equation

$$\text{Nu} = \frac{hD_t}{k}$$

Rewrite and solve for h:

$$h = \frac{\text{Nu} \times k}{D_t}$$
$$= \frac{107.79 \times 0.0242}{0.116}$$
$$= 22.49 \text{ W/m}^2\text{·K}$$

Calculate heat transfer surface area per meter length of the pipe,

$$A_s = \pi D_t L$$
$$= \pi \times \left(11.6 \times 10^{-2}\right) \times 1$$
$$= 0.3644 \text{ m}^2$$

Step 4 of 7

The heat gained by the pipe from the steam per unit length is determined from the equation given by

$$\dot{Q}_{gain} = \frac{T_{in} - T_{out}}{R_{inside} + R_{pipe} + R_{ins}}$$
$$= \frac{T_{in} - T_s}{R_{inside} + R_{pipe} + R_{ins}}$$
$$= \frac{T_{in} - T_s}{\frac{1}{h_i A_i} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi k_{pipe} L} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{2\pi k_{ins} L}}$$

Here, T_{in} is temperature of the steam, T_s is surface temperature of the steam, h_i is heat transfer coefficient inside the pipe, k_{pipe} is thermal conductivity of the steel pipe, k_{ins} is thermal conductivity of the insulation, r_1 is inner radius of the pipe, r_2 is outer radius of the steel pipe and r_3 is outer diameter of the insulated pipe.

Step 5 of 7

Substitute 250°C for T_{in}, 80 W/m^2·°C for h_i, $\pi D_i L$ for A_i, 15 W/m·K for

Step 5 of 7

Substitute 250°C for T_{in}, 80 W/m^2·°C for h_i, $\pi D_1 L$ for A_i, 15 W/m·K for k_{pipe}, 0.038 W/m·K for k_{ins}, 0.02 m for r_1, 0.023 m for r_2, 0.058 m for r_3 and 1 m for L

$$\dot{Q}_{gain} = \frac{250 - T_s}{\frac{1}{\left(80 \times \pi \times \left(4 \times 10^{-2}\right) \times 1\right)} + \frac{\ln\left(\frac{0.023}{0.02}\right)}{(2\pi \times 15 \times 1)} + \frac{\ln\left(\frac{0.058}{0.023}\right)}{(2\pi \times 0.038 \times 1)}}$$

$$= \frac{250 - T_s}{0.0995 + \left(1.483 \times 10^{-3}\right) + 3.874}$$

$$\dot{Q}_{gain} = \frac{250 - T_s}{3.975} \qquad \text{..... (1)}$$

Step 6 of 7

Calculate the heat lost by the pipe to the surroundings per unit length of the pipe,

$$\dot{Q}_{lost} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

$$= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_\infty^4\right)$$

Here, ε is emissivity of insulation and σ is Stefan Boltzmann's constant.

Substitute, 0.3 for ε and 5.67×10^{-8} W/m^2·K^4 for σ

$$= 22.49 \times 0.3644(T_s - 3) + 0.3 \times 0.3644 \times 5.67 \times 10^{-8}$$

$$\times\left((T_s + 273)^4 - (3 + 273)^4\right)$$

$$\dot{Q}_{lost} = 8.199(T_s - 3) + 6.198 \times 10^{-9}\left[(T_s + 273)^4 - (276)^4\right] \qquad \text{..... (2)}$$

The heat lost by the pipe to the surroundings is nothing but the heat gained by the pipe from the steam.

So, calculate the surface temperature of the pipe, by equating the equations (1) and (2)

$$\dot{Q}_{gain} = \dot{Q}_{lost}$$

$$\frac{250 - T_s}{3.98} = 8.199(T_s - 3) + 6.198 \times 10^{-9}\left[(T_s + 273)^4 - (276)^4\right]$$

Solve the above equation,

$$T_s = 9.9°C$$

Since the obtained value is close to the assumed value. So, the assumption is correct.

Step 7 of 7

Substitute the value of T_s in equation (2) to obtain the heat loss from the steam per unit length of the pipe,

$$\dot{Q}_{lost} = 8.199(9.9 - 3) + 6.198 \times 10^{-9}\left[\left(282.9^4 - 276^4\right)\right]$$

$$= 60.325 \text{ W/m}$$

Therefore, the heat loss from the steam per unit length of the pipe $\dot{Q}_{lost}$ is $\boxed{60.325 \text{ W/m}}$

Question: 19.118P

Consider a house that is maintained at 24°C at all times. The walls of the house have *R*-3.38 insulation in SI units (i.e., an *L/k* value or a thermal resistance of 3.38 m2-C/W). During a cold winter night, the outside air temperature is 6°C and wind at 50 km/h is blowing parallel to a 4-m-high and 8-m-long wall of the house. If the heat transfer coefficient on the interior surface of the wall is 12 W/m2-K, determine the rate of heat loss from that wall of the house. Draw the thermal resistance network and disregard radiation heat transfer.

Step 1 of 5

Given data:

Temperature inside house, $T_{in} = 24°C$

Thermal resistance of the wall, $R'_{wall} = 3.38\ m^2 \cdot °C/W$

Outside air temperature, $T_{out} = 6°C$

Velocity of the wind, $V = 50\ km/h$

$= 13.89\ m/s$

Height of the wall of the house, $H = 4\ m$

Length of the wall of the house, $L = 8\ m$

Heat transfer coefficient on the interior surface of the wall, $h_i = 12\ W/m^2 \cdot K$

Step 2 of 5

Let us assume the film temperature of air as $T_f = 10°C$

From the table, properties of air at 1 atm pressure

At $T = 10°C$

Thermal conductivity, $k = 0.02439\ W/m \cdot K$

Kinematic viscosity, $\nu = 1.426 \times 10^{-5}\ m^2/s$

Prandtl number, $Pr = 0.7336$

The Reynolds number for the flow is determined from the equation given by

$$Re = \frac{VL}{\nu}$$

$$= \frac{13.89 \times 8}{1.426 \times 10^{-5}}$$

$$= 7791803$$

Since the obtained Reynolds number $Re > Re_{critical}$ i.e. 5×10^5 but less than 10^7, so the flow is a combined laminar and turbulent flow.

The Nusselt number for the combined flow is determined from the relationship given by

$$Nu = \left(0.037 Re^{0.8} - 871\right) Pr^{1/3}$$

$$= \left(0.037 \times (7791803)^{0.8} - 871\right) \times (0.7336)^{1/3}$$

$$= 10095.36$$

Step 3 of 5

The convective heat transfer coefficient of air is determined from the equation given by

$$Nu = \frac{hD}{k}$$

Rewrite and solve for h:

$$h = \frac{Nu \times k}{L}$$

$$= \frac{10095.36 \times 0.02439}{8}$$

$$= 30.78\ W/m^2 \cdot K$$

The surface area of the wall is determined as

$$A_s = H \times L$$

$$= 4 \times 8$$

$$= 32\ m^2$$

Step 4 of 5

The thermal resistance network is shown below:



Step 5 of 5

From the thermal resistance network the rate of heat loss from the wall is determined from the equation given by

$$\dot{Q} = \frac{T_{in} - T_{out}}{R_{in} + R_{wall} + R_{out}}$$

$$= \frac{T_{in} - T_{out}}{\left(\frac{1}{h_i A_s}\right) + \left(\frac{R'_{wall}}{A_s}\right) + \left(\frac{1}{hA_s}\right)}$$

$$= \frac{24 - 6}{\left(\frac{1}{12 \times 32}\right) + \left(\frac{3.38}{32}\right) + \left(\frac{1}{30.78 \times 32}\right)}$$

$$= 164.77\ W$$

Therefore, the rate of heat loss from the wall of the house is $\boxed{\dot{Q} = 164.77\ W}$

Question: 19.119P

An automotive engine can be approximated as a 0.4-m-high, 0.60-m-wide, and 0.7-m-long rectangular block. The bottom surface of the block is at a temperature of 75°C and has an emissivity of 0.92. The ambient air is at 5°C, and the road surface is at 10°C. Determine the rate of heat transfer from the bottom surface of the engine block by convection and radiation as the car travels at a velocity of 60 km/h. Assume the flow to be turbulent over the entire surface because of the constant agitation of the engine block. How will the heat transfer be affected when a 2-mm-thick gunk (k =3 W/m•K) has formed at the bottom surface as a result of the dirt and oil collected at that surface over time? Assume the metal temperature under the gunk still to be 75°C.

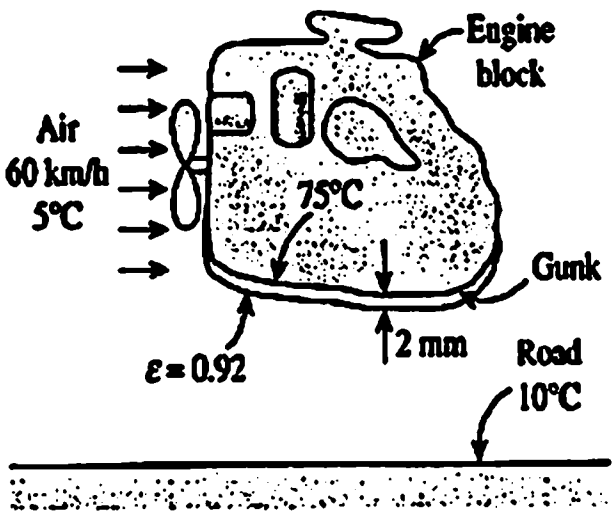


FIGURE P19–119

Step 1 of 4

Given
Height of engine block = 0.4 m
Width of engine block = 0.6 m
Length of engine block = 0.7 m
Temperature of bottom surface = 75°C
Emissivity (ε) = 0.92
Temperature of air = 5°C
Temperature of road surface = 10°C
Velocity of car = 60 km/h
Temperature of metal = 75°C
Thermal conductivity of gunk material = 3 W/m.°C

Step 2 of 4

Average film temperature $T_{avg} = \dfrac{75+5}{2}$

$$T_{avg} = 40^0\text{C}$$

From properties of air table at 1 atm and 40°C,

$k = 0.02662$ W/m.°C
$\upsilon = 1.702\times10^{-5}$ m^2/s
Pr = 0.7255

Reynolds number, $\text{Re}_L = \dfrac{VL}{\upsilon}$

$$\text{Re}_L = \frac{[(60\times1000/3600)\text{ m/s}](0.7\text{ m})}{1.702\times10^{-5}\text{ m}^2/\text{s}}$$

$$\text{Re}_L = 6.855\times10^5$$

This value is less than the critical Reynolds number value. But we assume turbulent flow due to the constant agitation of the engine block.

Nusselt number, $Nu = 0.037\text{Re}_L^{0.8}\,\text{Pr}^{1/3}$

$$Nu = 0.037(6.855\times10^5)^{0.8}(0.7255)^{1/3}$$

$$Nu = 1551$$

And $Nu = \dfrac{hL}{k}$

Heat transfer rate, $h = \dfrac{k}{L}Nu$

$$h = \frac{0.02662}{0.7}\times(1551)$$

$$h = 58.97\text{ W/m}^2.^\circ\text{C}$$

Step 3 of 4

Heat loss rate by convection, $\dot{Q}_{conv} = hA_s(T_\infty - T_s)$

$$\dot{Q}_{conv} = (58.97)\times(0.6\times0.7)\times(75-5)$$

$$\dot{Q}_{conv} = 1734\text{ W}$$

Heat loss rate by radiation, $\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$

$$\dot{Q}_{rad} = (0.92)(0.6\times0.7)\left(5.67\times10^{-8}\right)\left[(75+273)^4 - (10+273)^4\right]$$

$$\dot{Q}_{rad} = 181\text{ W}$$

Total heat loss rate is $\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$

$$\dot{Q}_{total} = 1734 + 181$$

$$\dot{Q}_{total} = 1915\text{ W}$$

The rate of heat transferred from the bottom surface is $\boxed{\dot{Q}_{total} = 1915\text{ W}}$

Step 4 of 4

Heat transfer rate with gunk is $\dot{Q} = \dfrac{T_\infty - T_s}{\dfrac{1}{hA_s} + \dfrac{L}{kA_s}}$

$$\dot{Q} = \frac{(75\text{-}5)^\circ\text{C}}{\dfrac{1}{58.97\times(0.6\times0.7)} + \dfrac{0.002}{3\times(0.6\times0.7)}}$$

$$\dot{Q} = 1668\text{ W}$$

Total heat loss rate with gunk is $\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$

$$\dot{Q}_{total} = 1668 + 181$$

$$\dot{Q}_{total} = 1849\text{ W}$$

The rate of heat transferred from the bottom surface with gunk is $\boxed{\dot{Q}_{total} = 1849\text{ W}}$

Question: 19.120P pg 1

The boiling temperature of nitrogen at atmospheric pressure at sea level (1 atm pressure) is −196°C. Therefore, nitrogen is commonly used in low-temperature scientific studies, since the temperature of liquid nitrogen in a tank open to the atmosphere will remain constant at -196°C until it is depleted. Any heat transfer to the tank will result in the evaporation of some liquid nitrogen, which has a heat of vaporization of 198 kJ/kg and a density of 810 kg/m3 at 1 atm.

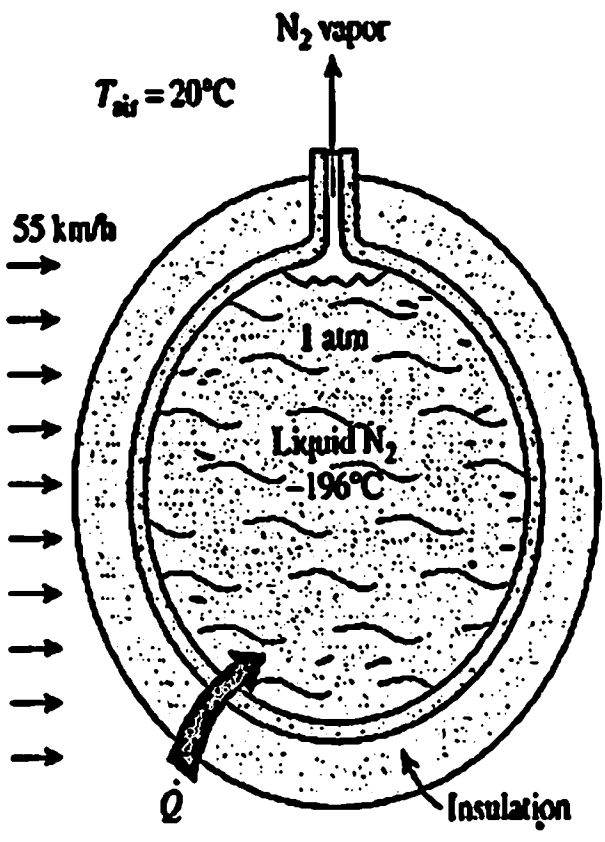


FIGURE P19–120

Consider a 4-m-diameter spherical tank that is initially filled with liquid nitrogen at 1 atm and – 196°C. The tank is exposed to 20° C ambient air and 55 km/h winds. The temperature of the thin-shelled spherical tank is observed to be almost the same as the temperature of the nitrogen inside. Disregarding any radiation heat exchange, determine the rate of evaporation of the liquid nitrogen in the tank as a result of heat transfer from the ambient air if the tank is (*a*) not insulated, (*b*) insulated with 5-cm-thick fiberglass insulation (k =0.035 W/m·K), and (*c*) insulated with 2-cm-thick superinsulation that has an effective thermal conductivity of 0.00005 W/m·K.

Step 1 of 14

Given data:

Boiling temperature of nitrogen at 1 atm T= -196°C

Heat of vaporization, h_f =198 kJ/kg

Density of nitrogen, ρ=810 kg/m^3

Diameter of spherical tank, D = 4 m

Temperature of air, T_s = 20°C

Velocity of wind, V=55 km/h

Thickness of insulation, t_1 = 5 cm

Thermal conductivity of insulation material, k_i = 0.035 W/m · °C

Thickness of super insulation, t_{supi} = 2cm

Thermal conductivity of super insulation material, k_{supi} = 0.00005 W/m · °C

Step 2 of 14

From properties of air table taking properties of air at 1 atm and free stream temperature of 20°C

Thermal conductivity, k = 0.02514 W/m.°C

Kinetic viscosity, $\upsilon = 1.516\times10^{-5}$ m^2/s

Dynamic viscosity, $\mu_\infty = 1.825\times10^{-5}$ kg/m · s

$\mu_{s@-196\,°C} = 5.023\times10^{-6}$ kg/m · s

Prandntl number, Pr = 0.7309

Step 3 of 14

(a)

When there is no insulation of the spherical tank then diameter $D = D_i = 4$ m

Reynolds number

$$\mathrm{Re} = \frac{VD}{\upsilon}$$

$$= \frac{[(55\times1000/3600)\ \mathrm{m/s}](4\ \mathrm{m})}{1.516\times10^{-5}\ \mathrm{m^2/s}}$$

$$= 4.031\times10^6$$

For flow over a sphere Nusselt number can be expressed as

$$\mathrm{Nu} = 2+\left[0.4\mathrm{Re}^{0.5}+0.06\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$=2+\left[0.4\times\left(4.031\times10^6\right)^{0.5}+0.06\left(4.031\times10^6\right)^{2/3}\right]\times0.7309^{0.4}\left(\frac{1.825\times10^{-5}}{5.023\times10^{-6}}\right)^{1/4}$$

$$=2+[803.094+1519.706]\times1.2179$$

$$\mathrm{Nu} = 2830.93$$

Then heat transfer coefficient is

$$\mathrm{Nu} = \frac{hD}{k}$$

$$h = \frac{k}{D}\mathrm{Nu}$$

$$= \frac{0.02514}{4}\times2830.93$$

$$h = 17.79\ \mathrm{W/m^2\cdot K}$$

Step 4 of 14

Heat transfer rate will be

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= h\times\pi D^2\times(T_s - T_\infty)$$

$$= 17.79\times[\pi\times4^2][(20-(-196)]$$

$$\dot{Q} = 193152.15\ \mathrm{W}$$

Step 5 of 14

Rate of evaporation of liquid nitrogen is determined as

$$\dot{Q} = \dot{m}h_f$$

$$\dot{m} = \frac{\dot{Q}}{h_f}$$

Step 5 of 14

Rate of evaporation of liquid nitrogen is determined as

$$\dot{Q} = \dot{m}h_f$$

$$\dot{m} = \frac{\dot{Q}}{h_f}$$

$$= \frac{193.152}{198}$$

$$\dot{m} = 0.9755 \text{kg/s}$$

Therefore rate of evaporation of liquid nitrogen when it is not insulated is

$$\boxed{\dot{m} = 0.9755 \text{kg/s}}$$

Step 6 of 14

(b)

The spherical tank is insulated with 5 cm thick fiberglass insulation

Then outer diameter of the sphere is

$$D_0 = D + 2 \times 0.05$$

$$= 4 + 0.1$$

$$= 4.1 \text{m}$$

As insulation is provided on the outer surface the surface temperature of the sphere changes, therefore taking Dynamic viscosity value,

$$\mu_{s,-100^\circ\text{C}} = 1.189 \times 10^{-5} \text{kg/m} \cdot s$$

Step 7 of 14

Reynolds number can be calculated as

$$\text{Re} = \frac{VD}{\upsilon}$$

$$= \frac{[(55 \times 1000/3600) \text{ m/s}](4.1 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}}$$

$$\text{Re} = 4.131 \times 10^6$$

For flow over a sphere Nusselt number can be expressed as

$$\text{Nu} = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4 \times \left(4.131 \times 10^6\right)^{0.5} + 0.06\left(4.131 \times 10^6\right)^{2/3}\right] \times 0.7309^{0.4}\left(\frac{1.825 \times 10^{-5}}{1.189 \times 10^{-5}}\right)^{1/4}$$

$$= 2 + [812.99 + 1544.737] \times 0.9818$$

$$\text{Nu} = 2316.81$$

Step 8 of 14

Heat transfer coefficient is

$$\text{Nu} = \frac{hD}{k}$$

$$h = \frac{k}{D}\text{Nu}$$

$$= \frac{0.02514}{4.1} \times 2316.81$$

$$h = 14.206 \text{W/m}^2 \cdot \text{K}$$

Step 9 of 14

Surface area

$$A_s = \pi D^2$$

$$= \pi(4.1 \text{ m})^2$$

$$= 52.81 \text{ m}^2$$

Rate of heat transfer

$$\dot{Q} = \frac{T_\infty - T_{s,tank}}{R_{insulation} + R_{conv}}$$

$$= \frac{T_\infty - T_{s,tank}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{hA_s}}$$

$$= \frac{[20 - (-196)]^\circ\text{C}}{\frac{(2.05 - 2) \text{ m}}{4\pi(0.035 \text{ W/m.}^\circ\text{C})(2.05 \text{ m})(2 \text{ m})} + \frac{1}{(14.206 \text{ W/m}^2.^\circ\text{C})(52.81 \text{ m}^2)}}$$

$$= \frac{216}{0.02772 + 0.00133}$$

$$\dot{Q} = 7435.45 \text{W}$$

Step 10 of 14

Rate of evaporation of liquid nitrogen is determined as

$$\dot{Q} = \dot{m}h_f$$

$$\dot{m} = \frac{\dot{Q}}{h_f}$$

$$= \frac{7.435}{198}$$

$$\dot{m} = 0.0375 \text{kg/s}$$

Rate of evaporation of liquid nitrogen is $\boxed{\dot{m} = 0.0375 \text{kg/s}}$

Step 11 of 14

(c)

The spherical tank is insulated with 2 cm thick super insulation

Taking dynamic viscosity value at the new value surface temperature 0°C to be

$$\mu = 1.729 \times 10^{-5} \text{ kg/m} \cdot s$$

Then outer diameter of the sphere is

$$D_0 = D + 2 \times 0.02$$

$$= 4 + 0.04$$

$$= 4.04 \text{m}$$

Reynolds number

$$\text{Re} = \frac{VD}{\upsilon}$$

$$=4+0.04$$
$$=4.04\,\text{m}$$

Reynolds number

$$\text{Re}=\frac{VD}{\upsilon}$$
$$=\frac{[(55\times1000/3600)\,\text{m/s}](4.04\,\text{m})}{1.516\times10^{-5}\,\text{m}^2/\text{s}}$$
$$\text{Re}=4.071\times10^{6}$$

For flow over a sphere Nusselt number can be expressed as

$$\text{Nu}=2+\left[0.4\text{Re}^{0.5}+0.06\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$=2+\left[0.4\times\left(4.071\times10^{6}\right)^{0.5}+0.06\left(4.071\times10^{6}\right)^{2/3}\right]\times0.7309^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4}$$
$$=2+[807.06+1529.743]\times0.894$$
$$\text{Nu}=2091.10$$

Step 12 of 14

Heat transfer coefficient

$$\text{Nu}=\frac{hD}{k}$$
$$h=\frac{k}{D}\text{Nu}$$
$$=\frac{0.02514}{4.04}\times2091.10$$
$$h=13.012\,\text{W/m}^2\cdot\text{K}$$

Step 13 of 14

Surface area

$$A_s=\pi D^2$$
$$=\pi(4.04\,\text{m})^2$$
$$=51.28\,\text{m}^2$$

Heat transfer rate for liquid nitrogen is

$$\dot{Q}=\frac{T_\infty-T_{s,\text{tank}}}{R_{\text{insulation}}+R_{\text{conv}}}$$
$$=\frac{T_\infty-T_{s,\text{tank}}}{\frac{r_2-r_1}{4\pi k r_1 r_2}+\frac{1}{hA_s}}$$
$$=\frac{[20-(-196)]^\circ\text{C}}{\frac{(2.02-2)\,\text{m}}{4\pi(0.00005\,\text{W/m.}^\circ\text{C})(2.02\,\text{m})(2\,\text{m})}+\frac{1}{(13.012\,\text{W/m}^2.^\circ\text{C})(51.28\,\text{m}^2)}}$$
$$=\frac{216}{7.878+0.00149}$$
$$\dot{Q}=27.412\,\text{W}$$

Step 14 of 14

Rate of evaporation of liquid nitrogen is determined as

$$\dot{Q}=\dot{m}h_f$$
$$\dot{m}=\frac{\dot{Q}}{h_f}$$
$$=\frac{0.027412}{198}$$
$$=1.38\times10^{-4}\,\text{kg/s}$$

Therefore, Rate of evaporation of liquid nitrogen when the sphere is insulated with super insulation is $\boxed{\dot{m}=1.38\times10^{-4}\,\text{kg/s}}$

Question:

Repeat Prob. 19-120 for liquid oxygen, which has a boiling temperature of −183°C, a heat of vaporization of 213 kJ/kg, and a density of 1140 kg/m3 at 1 atm pressure.

Step 1 of 13

Taking properties of air at 1 atm for free stream temperature of 20°C from properties of air table

Thermal conductivity, $k = 0.02514\ \text{W/m}\cdot{}^\circ\text{C}$

Kinetic viscosity, $\upsilon = 1.516\times10^{-5}\ \text{m}^2/\text{s}$

Dynamic viscosity, $\mu_\infty = 1.825\times10^{-5}\ \text{kg/m}\cdot\text{s}$

$\mu_{s,@-183^\circ C} = 6.127\times10^{-5}\ \text{kg/m-s}$

Prandntl number, $\text{Pr} = 0.7309$

Step 2 of 13

(a)

$D = D_i = 4\ \text{m}$ When no insulation is there, and

Calculate e the Reynolds number,

$$\text{Re} = \frac{VD}{\upsilon}$$

V is the velocity of the wind and

L is the diameter of the spherical tank.

Substitute 55 km/h for V, 4 m for D and $1.516\times10^{-5}\ \text{m}^2/\text{s}$ for ν in the above equation,

$$\text{Re} = \frac{\left(55\text{km/h}\times\frac{1000}{3600}\text{m/s}\right)(4\ \text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}}$$
$$= 4.03\times10^6$$

Step 3 of 13

Calculate the Nusselt number,

$$\text{Nu} = \frac{hD}{k}$$
$$= 2+\left[0.4\text{Re}^{0.5}+0.06\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Substitute the respective values in the above equation,

$$\text{Nu} = 2+\left[0.4\left(4.03\times10^6\right)^{0.5}+0.06\left(4.03\times10^6\right)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.05\times10^{-5}}\right)^{1/4}$$
$$= 2+[803+1519.46]1.0129$$
$$= 2354.4$$

Calculate the coefficient of convective heat transfer,

$$h = \frac{k}{D}\text{Nu}$$

Substitute the respective values in the above equation,

$$h = \frac{0.02514}{4}\times2354.4$$
$$= 14.79\ \text{W/m}^2\cdot{}^\circ\text{C}$$

Step 4 of 13

Calculate the heat transfer rate,

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= h(\pi D^2)(T_s - T_\infty)$$

Where,

T_s is the surface temperature and

T_∞ is the boiling temperature of the oxygen at 1 atm.

Substitute 20°C for T_s, -183°C for T_∞ and the respective values in the above equation,

$$\dot{Q} = (14.79)\left(\pi\times4^2\right)(20-(-183))$$
$$= 150915.58\ \text{W}$$

Step 5 of 13

Calculate the Rate of evaporation of liquid oxygen,

$$\dot{Q} = \dot{m}h_{fg}$$
$$\dot{m} = \frac{\dot{Q}}{h_{fg}}$$

Where,

h_{fg} is the heat of vaporization.

Substitute $(213\,\text{kJ/kg})$ for h_{fg} and the respective values in the above equation,

$$\dot{m} = \frac{142.4}{213}$$
$$= 0.6685\ \text{kg/s}$$

Therefore, Rate of evaporation of liquid oxygen $\boxed{\dot{m} = 0.6685\ \text{kg/s}}$

Step 6 of 13

(b)

After insulation diameter and surface temperature will change. So $D = D_0 = 4.1$ m and surface temperature will be evaluating dynamic viscosity a -100°C .

Calculate the Reynolds number,

$$\text{Re} = \frac{VD}{\upsilon}$$
$$= \frac{\left(55\text{km/h}\times\frac{1000}{3600}\text{m/s}\right)(4.1\ \text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}}$$
$$= 4.132\times10^6$$

Step 7 of 13

Calculate the Nusselt number,

$$\text{Nu} = \frac{hD}{k}$$

Step 7 of 13

Calculate the Nusselt number,

$$\mathrm{Nu}=\frac{hD}{k}$$

$$=2+\left[0.4\mathrm{Re}^{0.5}+0.06\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Substitute the respective values in the above equation,

$$\mathrm{Nu}=2+\left[0.4\left(4.132\times10^{6}\right)^{0.5}+0.06\left(3.005\times10^{6}\right)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.189\times10^{-5}}\right)^{1/4}$$

$$=2+[813.1+1544.97]0.982$$

$$=2317.7$$

Calculate the coefficient of convective heat transfer,

$$h=\frac{k}{D}\mathrm{Nu}$$

$$=\frac{0.02514}{4.1}\times2317.7$$

$$=14.21\ \mathrm{W/m^2\cdot{}^\circ C}$$

Step 8 of 13

Calculate the surface area of the spherical tank,

$$A_s=\pi D^2$$

$$=\pi(4.1\ \mathrm{m})^2$$

$$=52.81\ \mathrm{m^2}$$

Calculate the Heat transfer rate for liquid oxygen,

$$\dot{Q}=\frac{T_\infty-T_{s,\mathrm{tank}}}{R_{\mathrm{insulation}}+R_{\mathrm{conv}}}$$

$$=\frac{T_\infty-T_{s,\mathrm{tank}}}{\frac{r_2-r_1}{4\pi k r_1 r_2}+\frac{1}{hA_s}}$$

Where,

T_∞ is the temperature of the air,

$T_{s,\mathrm{tank}}$ is the surface temperature of the tank,

$R_{\mathrm{insulation}}$ is the co-efficient of thermal resistance of the insulation and

R_{conv} is the convective co-efficient of thermal resistance.

Substitute $20^\circ\mathrm{C}$ for T_∞, $(-183^\circ\mathrm{C})$ for $T_{s,\mathrm{Tank}}$, 2.05 m for r_1, 2 m for r_2 and the respective values in the above equation,

$$\dot{Q}=\frac{[20-(-183)]^\circ\mathrm{C}}{\frac{(2.05-2)\ \mathrm{m}}{4\pi(0.035\ \mathrm{W/m.^\circ C})(2.05\ \mathrm{m})(2\ \mathrm{m})}+\frac{1}{(14.21\ \mathrm{W/m^2.^\circ C})(52.81\ \mathrm{m^2})}}$$

$$=\frac{203}{0.0277+1.33\times10^{-3}}$$

$$=6986.186\ \mathrm{W}$$

Step 9 of 13

Calculate the Rate of evaporation of liquid oxygen,

$$\dot{Q}=\dot{m}h_{fg}$$

$$\dot{m}=\frac{\dot{Q}}{h_{fg}}$$

$$=\frac{6.986186}{213}$$

$$=0.0328\ \mathrm{kg/s}$$

Rate of evaporation of liquid oxygen is $\boxed{0.0328\mathrm{kg/s}}$

Step 10 of 13

(c)

Taking dynamic viscosity value at the new value surface temperature $0^\circ\mathrm{C}$ to be

$\mu=1.729\times10^{-5}\ \mathrm{kg/m.s}$.

Diameter will be $D=D_0=4.04$ m calculating Nusselt number

$$\mathrm{Re}=\frac{VD}{\upsilon}$$

$$=\frac{[(55\times1000/3600)\ \mathrm{m/s}](4.04\ \mathrm{m})}{1.516\times10^{-5}\ \mathrm{m^2/s}}$$

$$=4.07\times10^{6}$$

Step 11 of 13

Calculate the Nusselt number,

$$\mathrm{Nu}=\frac{hD}{k}$$

$$=2+\left[0.4\mathrm{Re}^{0.5}+0.06\mathrm{Re}^{2/3}\right]\mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

Substitute the respective values in the above equation,

$$\mathrm{Nu}=2+\left[0.4(4.07\times10^{6})^{0.5}+0.06(4.07\times10^{6})^{2/3}\right](0.713)^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4}$$

$$=2+[806.97+1529.5]0.885$$

$$=2069.78$$

Step 12 of 13

Calculate the Convection heat transfer coefficient,

$$h=\frac{k}{D}\mathrm{Nu}$$

$$=\frac{0.02514}{4.04}2069.78$$

$$=12.88\ \mathrm{W/m^2\cdot{}^\circ C}$$

Calculate the surface area of the spherical tank,

$$A_s=\pi D^2$$

$$=\pi(4.04\ \mathrm{m})^2$$

$$=51.28\ \mathrm{m^2}$$

Calculate the heat transfer rate for liquid nitrogen

Calculate the surface area of the spherical tank.

19.1211 pg 3

$$A_s = \pi D^2$$
$$= \pi(4.04\ \text{m})^2$$
$$= 51.28\ \text{m}^2$$

Calculate the heat transfer rate for liquid nitrogen,

$$\dot{Q} = \frac{T_\infty - T_{s,tank}}{R_{insulation} + R_{conv}}$$
$$= \frac{T_\infty - T_{s,tank}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{hA_s}}$$

Substitute the respective values in the above equation,

$$\dot{Q} = \frac{20-(-183)}{\frac{(2.02-2)}{4\pi(0.00005)(2.02\times 2)} + \frac{1}{(12.88)(51.28)}}$$
$$= \frac{20-(-183)}{7.87+1.514\times 10^{-3}}$$
$$= 25.759\ \text{W}$$

Step 13 of 13

Calculate the Rate of evaporation of liquid oxygen,

$$\dot{Q} = \dot{m} h_{fg}$$
$$\dot{m} = \frac{\dot{Q}}{h_{fg}}$$
$$= \frac{0.025759}{213}$$
$$= 1.21\times 10^{-4}\ \text{kg/s}$$

Therefore, Rate of evaporation of liquid oxygen is $\boxed{1.21\times 10^{-4}\ \text{kg/s}}$

Question: 19.122P

Four power transistors, each dissipating 10 W, are mounted on a thin vertical aluminum plate (k= 237 W/m·K) 22 cm × 22 cm in size. The heat generated by the transistors is to be dissipated by both surfaces of the plate to the surrounding air at 20°C, which is blown over the plate by a fan at a velocity of 5 m/s. The entire plate can be assumed to be nearly isothermal, and the exposed surface area of the transistor can be taken to be equal to its base area. Determine the temperature of the aluminum plate.

Step 1 of 5

Given data:

No. of transfers, $N = 4$

Power Dissipation by each transistor, $P = 10\,\text{W}$

Thermal conductivity of the plate is $k = 237\,\text{W/m}\cdot\text{K}$

Aluminum plate dimensions $(L \times B) = 22 \times 22\,\text{cm}$ or $0.22 \times 0.22\,\text{m}$

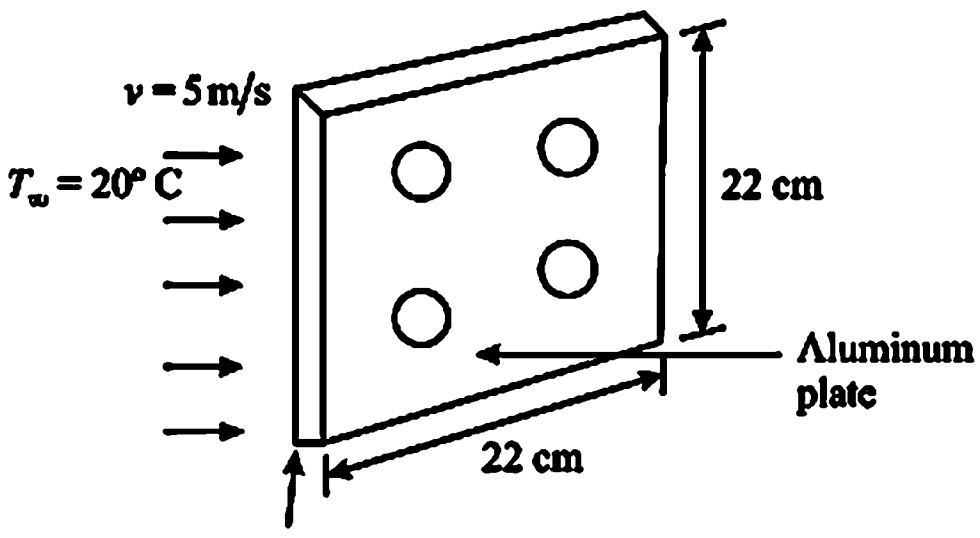
Ambient air temperature, $T_\infty = 20^\circ\text{C}$

Velocity of air flow, $V = 5\,\text{m/s}$

Plate is isothermal.

Step 2 of 5

Figure showing the aluminum plate,



Step 3 of 5

Let's take the properties of air at files temperature say 30°C (Assuming).

Kinematic viscosity, $\nu = 1.608 \times 10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

Reynolds number

$$\text{Re} = \frac{VL}{\nu}$$
$$= \frac{5 \times 0.22}{1.608 \times 10^{-5}}$$
$$= 68408$$

Since the value is less than $\text{Re}_{\text{critical}}$ that is 5×10^5 the flow is laminar here.

Step 4 of 5

Nusselt number calculation (based on Reynolds number and Prandtl number)

$$Nu = 0.664\,\text{Re}_L^{0.5}\text{Pr}^{1/3}$$
$$= 0.664 \times (68408)^{0.5}(0.7282)^{1/3}$$
$$= 156.25$$

Also we know that,

$$Nu = \frac{hL}{k}$$
$$h = \frac{k \cdot Nu}{L}$$
$$= 18.38\,\text{W/m}^2\,^\circ\text{C}$$

Step 5 of 5

Heat transfer surface area

$$A_s = (L \times B) \times \text{number of plates}$$
$$= (0.22)^2 \times 2$$
$$= 0.0968\,\text{m}^2$$

Total heat dissipation or transfer,

$$\dot{Q} = P \times N$$
$$= 40\,\text{W}$$

Heat transfer through convection is given by.

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$40 = 18.38 \times 0.0968(T_s - 20)$$
$$T_s = 42.48^\circ\text{C}$$

Therefore, Surface temperature of the aluminum plate is obtained as $\boxed{T_s = 42.5^\circ\text{C}}$

Question: 19.123P

It is well known that cold air feels much colder in windy weather than what the thermometer reading indicates because of the "chilling effect" of the wind. This effect is due to the increase in the convection heat transfer coefficient with increasing air velocities. The *equivalent temperature* in °F is given by (1993 *ASHRAE Handbook of Fundamentals,* Atlanta, GA, p. 8.15)

where *V* is the wind velocity in mph and *T*ambient air temperature in °F in calm air. which is takento a be air with light winds at speeds up to 4 mph. The constant 91.4°F in the above equation is the mean skin temperature of a resting person in a comfortable environment. Windy air at atemperature *Tambient* and velocity Vwill feel as cold as calmair at a temperature *Tequiv*. The equation above is valid for winds up to 43 mph. Winds at higher velocities produce little additional chilling effect. Determine the equivalent temperature of an environment at 10°F at wind speeds of 10, 20. 30, and 40 mph. Exposed flesh can freeze within one minute at i a temperature below −25°F in calm weather. Does a person need to be concerned about this possibility in any of the cases above?

$$T_{equiv} = 91.4 - (91.4 - T_{ambient})(0.475 - 0.0203V + 0.304\sqrt{V})$$



FIGURE P19–123E

Step 1 of 5

Given data:

Equivalent temperature, $T_{equivalent} = 91.4-(91.4-T_{ambinet})\left(0.475-0.0203V+0.304\sqrt{V}\right)$

Wind velocity is V in mph

Ambient air temperature, $T_{ambinet}$

Equation validity for winds up to 43mph

Ambient Temperature, $T_{ambient} = 10°F$

Equivalent temperature $T_{equivalent}$ for wind speed of 10, 20, 30, 40 mph.

Step 2 of 5

Equivalent temperature is given by

$$T_{equivalent} = 91.4-(91.4-T_{ambient})\left(0.475-0.0203V+0.304\sqrt{V}\right) \quad \ldots\ldots (1)$$

For $V = 10$mph

$$T_{equivalent} = 91.4-(91.4-10)\left(0.475-0.0203\times 10+0.304\sqrt{10}\right)$$

$$\boxed{T_{equivalent} = -8.99°F}$$

Step 3 of 5

For $V = 20$mph

$$T_{equivalent} = 91.4-(91.4-10)\left(0.475-0.0203\times 20+0.304\sqrt{20}\right)$$

$$\boxed{T_{equivalent} = -24.9°F}$$

Step 4 of 5

For $V = 30$mph

$$T_{equivalent} = 91.4-(91.4-10)\left(0.475-0.0203\times 30+0.304\sqrt{30}\right)$$

$$\boxed{T_{equivalent} = -33.2°F}$$

Step 5 of 5

For $V = 40$mph

$$T_{equivalent} = 91.4-(91.4-10)\left(0.475-0.0203\times 40+0.304\sqrt{40}\right)$$

$$\boxed{T_{equivalent} = -37.7°F}$$

In the second case the equivalent temperature is almost equal to given freezing temperature that is 25°C and in last two cases it is more than given. So for all these three cases person should be concerned about his possibility of freezing.

Question:

Reconsider Prob. 19-123E. Using EES (or other) software, plot the equivalent wind chill temperatures in °F as a function of wind velocity in the range of 5 mph to 50 mph for ambient temperatures of 20°F, 40°F and 60°F. Discuss the results.

Step 1 of 13

$$T_{equiv} = 91.4 - (91.4 - T_{ambient})\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

Step 2 of 13

Case: I

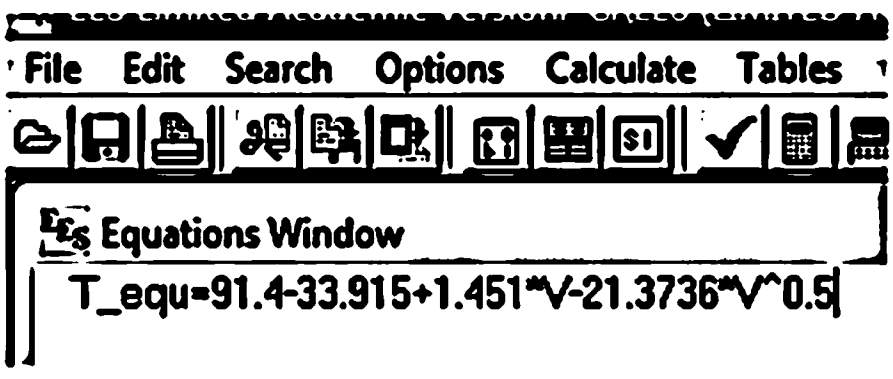
When the ambient temperature is $T_{ambient} = 20°F$

By substituting the known values in above expression, we get

$$\begin{aligned} T_{equiv} &= 91.4 - (91.4 - T_{ambient})\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \\ &= 91.4 - (91.4 - 20)\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \\ &= 91.4 - 71.4\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \\ &= 91.4 - 33.915 + 1.451V - 21.736\sqrt{V} \end{aligned}$$

Step 3 of 13

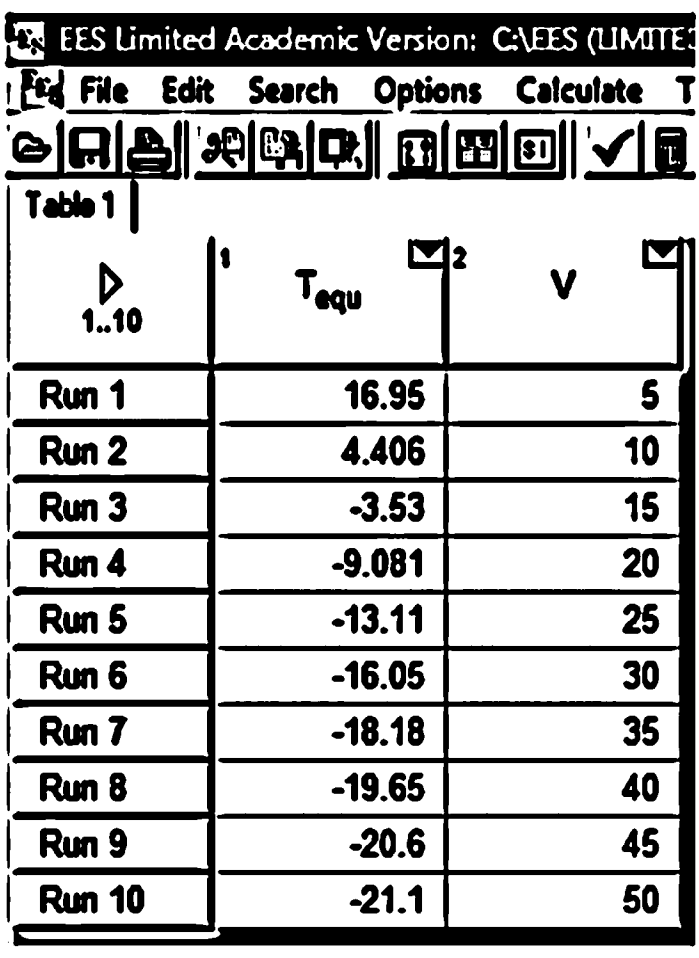
Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.



Step 4 of 13

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

<u>The following results are displayed:</u>

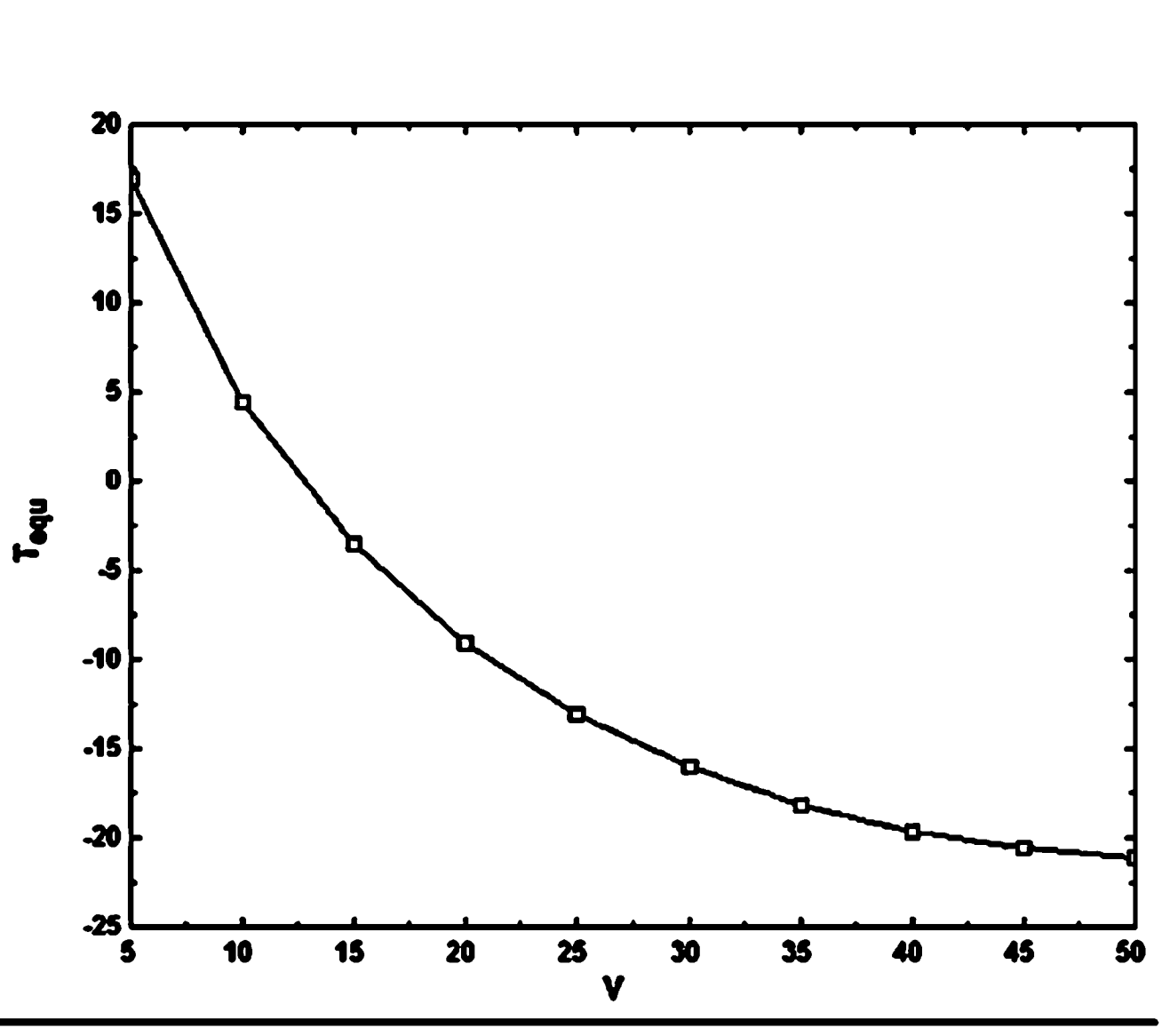


1..10	T_{equ} [1]	V [2]
Run 1	16.95	5
Run 2	4.406	10
Run 3	-3.53	15
Run 4	-9.081	20
Run 5	-13.11	25
Run 6	-16.05	30
Run 7	-18.18	35
Run 8	-19.65	40
Run 9	-20.6	45
Run 10	-21.1	50

Step 5 of 13

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

<u>By plotting the graph between temperature and velocity:</u>



Step 6 of 13

Case: 2

When the ambient temperature is $T_{ambient} = 40°F$

By substituting the known values in above expression, we get

$$\begin{aligned} T_{equiv} &= 91.4 - (91.4 - T_{ambient})\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \\ &= 91.4 - (91.4 - 40)\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \\ &= 91.4 - 51.4\left(0.475 - 0.0203V + 0.304\sqrt{V}\right) \end{aligned}$$

$$T_{equiv} = 91.4 - (91.4 - T_{ambient})\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

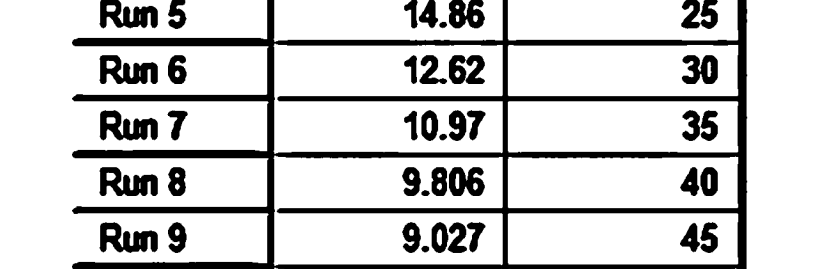
$$= 91.4 - (91.4 - 40)\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

$$= 91.4 - 51.4\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

$$= 91.4 - 24.415 + 1.045V - 15.65\sqrt{V}$$

Step 7 of 13

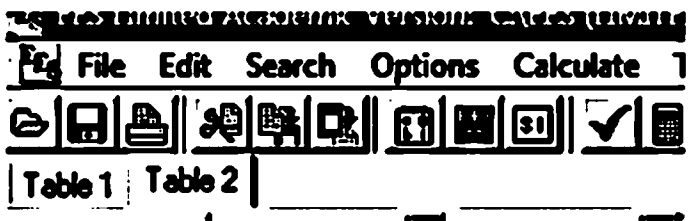
Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.



Step 8 of 13

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed:

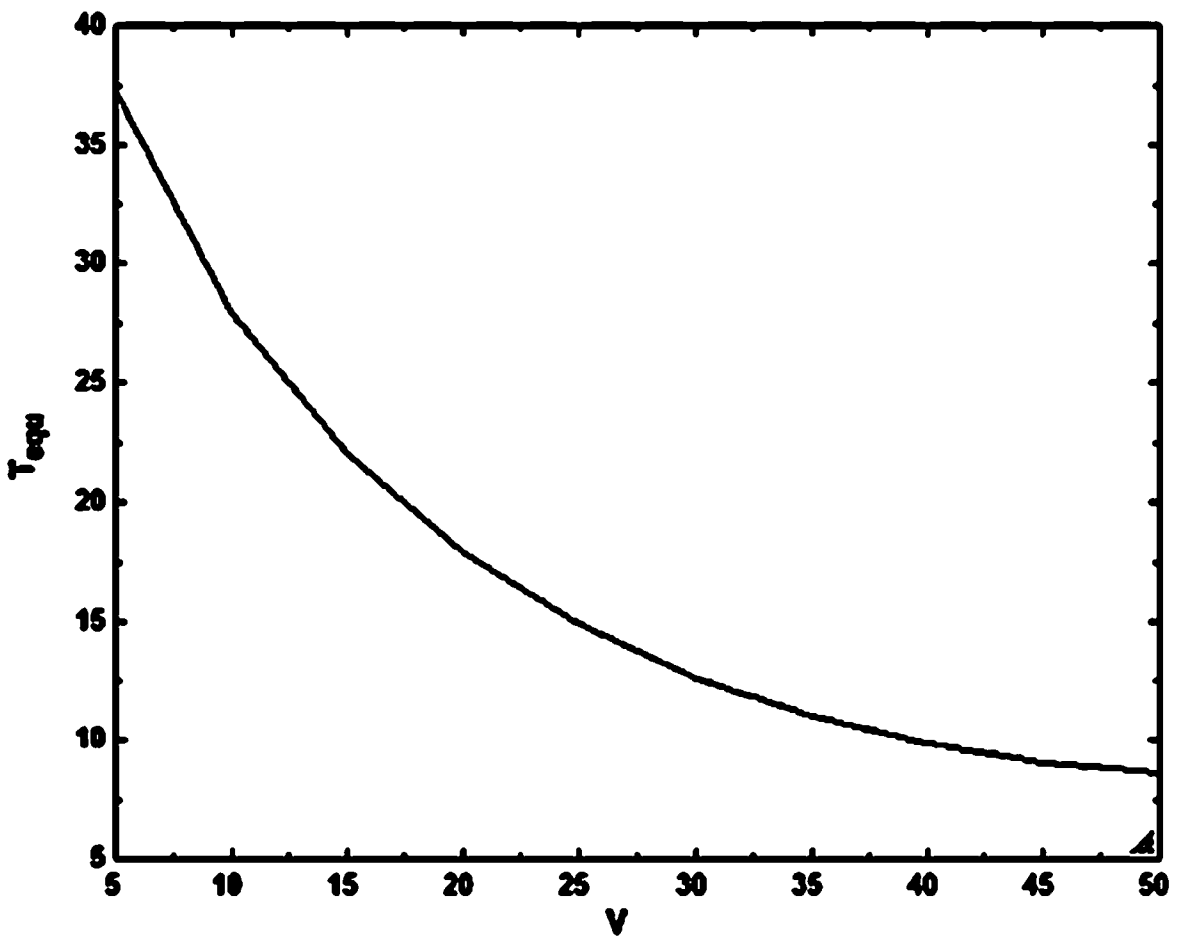


1..10	1 T_{equ}	2 V
Run 1	37.22	5
Run 2	27.95	10
Run 3	22.05	15
Run 4	17.9	20
Run 5	14.86	25
Run 6	12.62	30
Run 7	10.97	35
Run 8	9.806	40
Run 9	9.027	45
Run 10	8.573	50

Step 9 of 13

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

By plotting the graph between temperature and velocity:



Step 10 of 13

Case: 3

When the ambient temperature is $T_{ambient} = 60°F$

By substituting the known values in above expression, we get

$$T_{equiv} = 91.4 - (91.4 - T_{ambient})\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

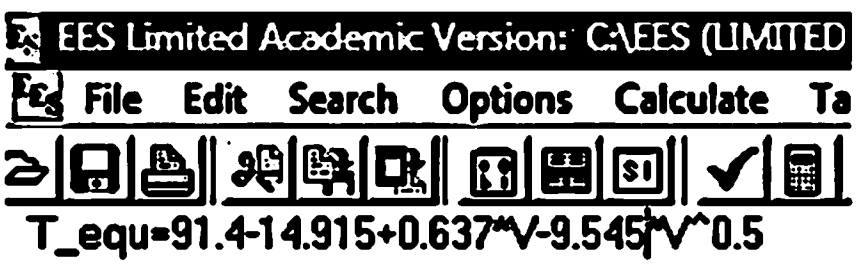
$$= 91.4 - (91.4 - 60)\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

$$= 91.4 - 31.4\left(0.475 - 0.0203V + 0.304\sqrt{V}\right)$$

$$= 91.4 - 14.915 + 0.637V - 9.545\sqrt{V}$$

Step 11 of 13

Open the EES software, then click file on menu bar and select the new file on menu bar. Then write the program in equation window.



Step 12 of 13

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed:

After writing the program in equation window select table on menu bar and click new parameter table. Now the dialog box will be appeared on the screen and then select the efficiency, head loss and work done by the pump. Finally click ADD, then click OK button.

The following results are displayed:

EES Limited Academic Version: C:\EES (EMU)

File Edit Search Options Calculate

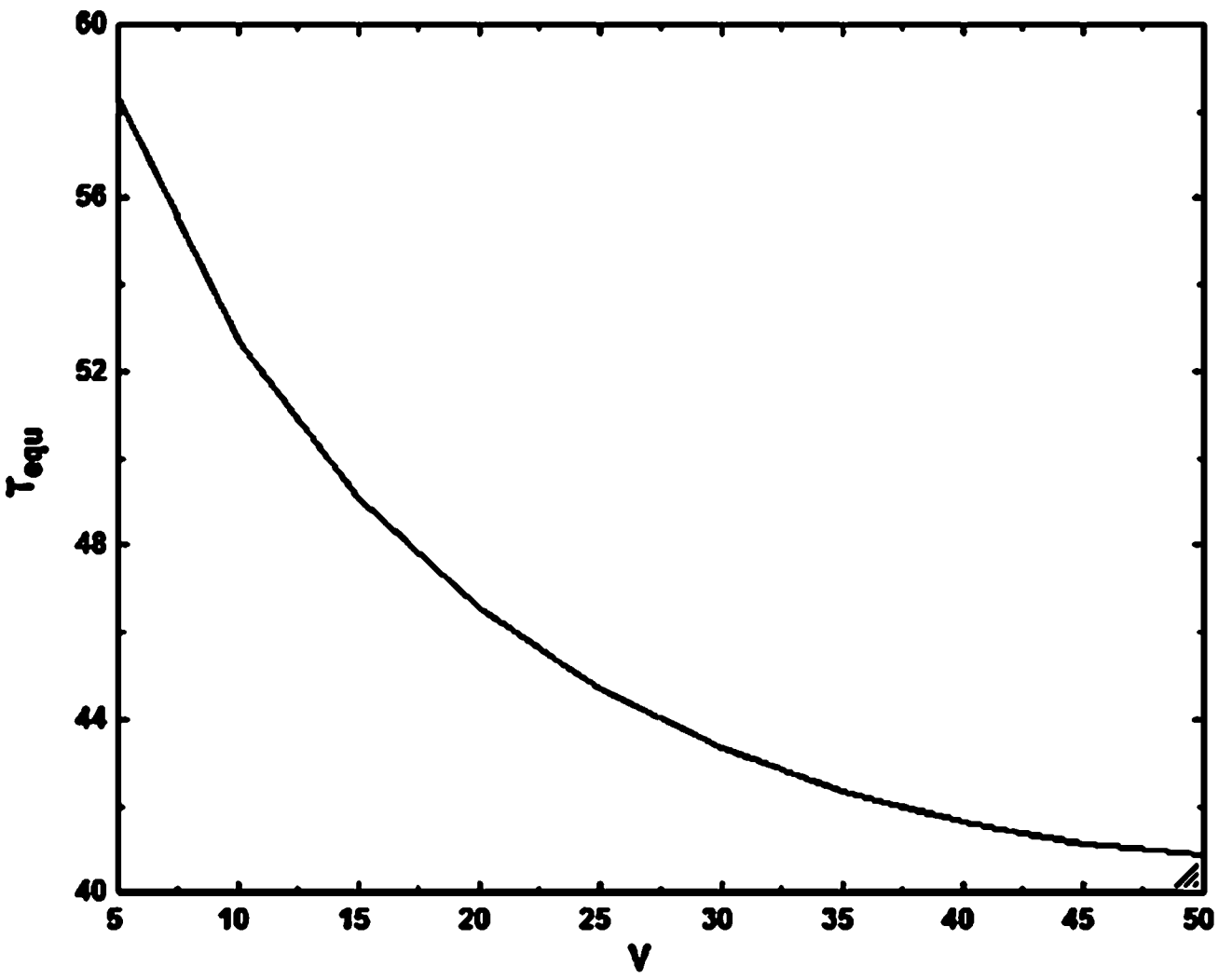
Table 1 | Table 2 | Table 3

1..10	1 T_{equ}	2 V
Run 1	58.33	5
Run 2	52.67	10
Run 3	49.07	15
Run 4	46.54	20
Run 5	44.69	25
Run 6	43.31	30
Run 7	42.31	35
Run 8	41.6	40
Run 9	41.12	45
Run 10	40.84	50

Step 13 of 13

Select the plot on menu bar, then click x-y plot .Dialog box will be appeared and select variables to draw graph, then click OK button.

By plotting the graph between temperature and velocity:



Question: 19.125P

To cool a storehouse in the summer without using a conventional air-conditioning system, the owner decided to hire an engineer to design an alternative system that would make use of the water in the nearby lake. The engineer decided to How air through a thin smooth 10-cm diameter copper tube that is submerged in the nearby lake. The water in the lake is typically maintained at a constant temperature of 15°C and a convection heat transfer coefficient of 1000 W/m2•K. If air (1 atm) enters the copper tube at amean temperature of 30°C with an average velocity of 2.5 m/s, determine the necessary copper tube length so that the outlet mean temperature of the air is 20°C.

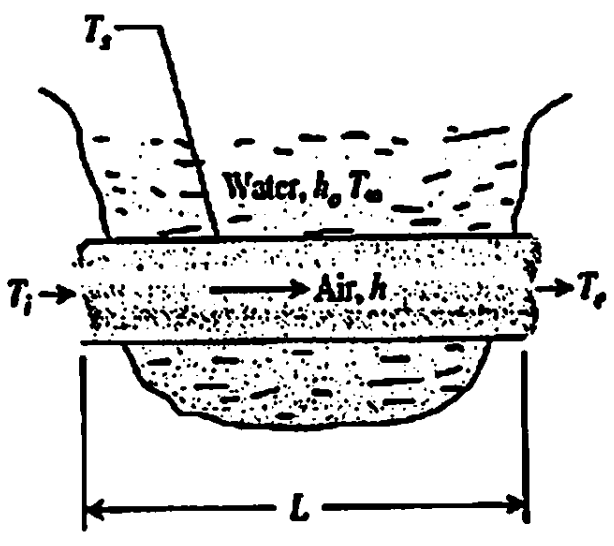


FIGURE P19–125

Step 1 of 8

Diameter of copper tube $D = 10\,\text{cm} = 0.1\text{m}$

The temperature of water, $T_\infty = 15°\text{C}$

Convection heat transfer coefficient, $h_0 = 1000\,\text{W/m}^2\cdot\text{K}$

Air inlet temperature, $T_i = 30\,°\text{C}$

Average velocity of air, $V_{avg} = 2.5\text{m/s}$

Outlet air temperature, $T_e = 20\,°\text{C}$

Step 2 of 8

Calculation of copper tube length as follows:

The average temperature is,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{30+20}{2}$$
$$= 25\,°\text{C}$$

From the properties of air at one atmospheric pressure table, at $T = 25\,°\text{C}$

Density, $\rho = 1.184\,\text{kg/m}^3$

Specific heat, $c_p = 1007\text{J/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.02551\text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.562\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7296$

Step 3 of 8

Now the Reynolds number,

$$\text{Re} = \frac{V_{avg}D}{\nu}$$
$$= \frac{2.5\times0.1}{1.562\times10^{-5}}$$
$$= 16005.121$$

This value is greater than 10,000 , so the flow is turbulent.

Step 4 of 8

By using the Dittus – Boelter equation, Nusselt number can be expressed as,

$$\text{Nu} = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$
$$= 0.023\times(16005.121)^{0.8}\times(0.7296)^{0.4}$$
$$= 46.81$$

We know that Nusselt number is also expressed as,

$$\text{Nu} = \frac{hD}{k}$$
$$h = \frac{\text{Nu}\times k}{D}$$
$$= \frac{46.81\times0.02551}{0.1}$$
$$h = 11.94\,\text{W/m}^2\cdot°\text{C}$$

Step 5 of 8

We can calculate the mass flow rate as, $\dot{m} = \rho A_c V_{avg}$

Cross-sectional area of tube is, $A_c = \frac{\pi D^2}{4}$

Therefore,

$$\dot{m} = \frac{\rho\pi D^2}{4}\times V_{avg}$$
$$= \frac{1.184\times\pi\times(0.1)^2\times2.5}{4}$$
$$= 0.0232477\,\text{kg/s}$$

Step 6 of 8

Therefore, heat transfer to lower temperature,

$$\dot{Q} = \dot{m}c_p(T_i - T_e)$$
$$= 0.0232477\times1007\times(30-20)$$
$$= 234.104\,\text{W}$$

Now calculating surface temperature T_s per 1 m length of tube by the equation,

$$\dot{Q} = h_o A_s (T_s - T_\infty)$$

Heat transfer surface area,

$$A_s = \pi DL$$
$$= \pi\times0.1\times1$$
$$= 0.314\,\text{m}^2$$

Step 7 of 8

Therefore, as steady condition is,

The heat given by air = Heat transferred is water.

$$\dot{Q} = h_o A_s (T_s - T_\infty)$$
$$234.104 = 1000\times0.314(T_s - 15)$$
$$(T_s - 15) = \frac{234.104}{314}$$
$$T_s - 15 = 0.74554$$
$$T_s = 15.74\,°\text{C}$$

Now, by substituting the known values in outlet temperature equation we get,

Step 8 of 8

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right)$$
$$= T_s - (T_s - T_i)\exp\left(-\frac{h\pi DL}{\dot{m}c_p}\right)$$
$$20 = 15.74 - (15.74 - 30)\exp\left(\frac{-11.94\times0.314\text{L}}{0.02324\times1007}\right)$$

The above equation can be simplify as follows,

Therefore, length of the pipe is $\boxed{L = 32.60\,\text{ft}}$.

Question: 19.126P

Liquid mercury is flowing at 0.6 kg/s through a 5-cm diameter tube with inlet and outlet mean temperatures of 100°C and 200°C, respectively. The tube surface temperature is maintained constant at 250°C. Determine the tube length using (*a*) the appropriate Nusselt number relation for liquid metals and (*b*) the Dittus-Boelter equation, (*c*) Compare the results of (*a*) and (*b*).

Step 1 of 7

Given data:

Mass flow rate of mercury, $\dot{m} = 0.6\ \text{kg/s}$

Diameter of the tube, $D = 0.05\ \text{m}$

Inlet mean temperature, $T_i = 100°\text{C}$

Outlet mean temperature, $T_e = 200°\text{C}$

Surface temperature of the tube, $T_s = 250°\text{C}$

Step 2 of 7

(*a*)

From properties table of mercury,

Kinematic viscosity, $\nu = 8.514\times10^{-8}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.0152$

Dynamic viscosity at 150°C, $\mu = 1.126\times10^{-3}\ \text{kg/m-s}$

Reynolds number for the given flow is

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$

$$= \frac{4\times0.6}{\pi\times0.05\times1.126\times10^{-3}}$$

$$= 13569.16$$

As the obtained Reynolds number is greater than 10,000 the flow is turbulent flow.

Step 3 of 7

Prandtl number at T_s (250°) is $\text{Pr}_s = 0.0119$

According to the relation recommended by Sleicher and Rouse for liquid metals with the Nusselt number is

$$Nu = 4.8 + 0.0156\text{Re}^{0.85}\text{Pr}_s^{0.93}$$

$$= 4.8 + 0.0156(13569.16)^{0.85}(0.0119)^{0.93}$$

$$= 5.624$$

Heat transfer coefficient for the liquid metal is

$$h = \frac{k\cdot Nu}{D}$$

$$= \frac{10.0778\times5.624}{0.05}$$

$$= 1133.6\ \text{W/m}^2\cdot°\text{C}$$

Step 4 of 7

Calculating length, through energy balance relation,

$$L = -\frac{\dot{m}c_p}{\pi Dh}\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)$$

$$= -\frac{0.6\times136.1}{\pi\times0.05\times1133.6}\ln\left(\frac{250-200}{250-100}\right)$$

$$= 0.5038\,\text{m}$$

Therefore the length of the tube is $\boxed{L = 0.5038\,\text{m}}$

Step 5 of 7

(*b*)

According to Dittus-Boelter equation,

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$

$$= 0.023(13569.16)^{0.8}(0.0152)^{0.4}$$

$$= 8.7198$$

Therefore heat transfer coefficient for the liquid metal is

$$h = \frac{Nu\cdot k}{D}$$

$$= \frac{8.7198\times10.0778}{0.05}$$

$$= 1757.53\ \text{W/m}^2\cdot°\text{C}$$

Step 6 of 7

Calculating length, through energy balance relation,

$$L = -\frac{\dot{m}c_p}{\pi Dh}\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)$$

$$= -\frac{0.6\times136.1}{\pi\times0.05\times1757.53}\ln\left(\frac{250-200}{250-100}\right)$$

$$= 0.325\,\text{m}$$

Therefore the length of the tube is $\boxed{L = 0.325\,\text{m}}$

Step 7 of 7

(*c*)

As we can see from the calculated values, the length of the tube required for satisfying Sleicher and Rouse relation must be larger than that of the tube required for satisfying Dittus-Boelter criteria.

Question: 19.127P

In the effort to find the best way to cool a smooth thin-walled copper tube, an engineer decided to flow air either through the tube or across the outer tube surface. The tube has a diameter of 5 cm, and the surface temperature is maintained constant. Determine (*a*) the convection heat transfer coefficient when air is flowing through its inside at 25 m/s with bulk mean temperature of 50°C and (*b*) the convection heat transfer coefficient when air is flowing across its outer surface at 25 m/s with film temperature of 50°C.

Step 1 of 4

Given data:

Diameter of tube, $D = 0.05\text{m}$

Bulk mean temperature, $T = 50^\circ\text{C}$

Velocity of air flow, $V_{avg} = 25\text{m/s}$

Step 2 of 4

From the properties table for air,

Specific heat, $c_p = 1007\ \text{J/kg}\cdot\text{k}$

Thermal conductivity, $k = 0.02735\ \text{W/m -k}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Reynolds number is expressed as

$$\text{Re} = \frac{V_{avg}D}{\nu}$$

$$= \frac{25\times0.05}{\left(1.798\times10^{-5}\right)}$$

$$= 69521.7$$

At the obtained Reynolds number is greater than 10,000 the flow is turbulent flow.

Step 3 of 4

(*a*) Air flowing inside:

Nusselt number is determined by

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$

$$= 0.023(69521.7)^{0.8}(0.7228)^{0.4}$$

$$= 151$$

Heat transfer coefficient when air is flowing through the tube is

$$h = \frac{k\cdot Nu}{D}$$

$$= \frac{0.02735\times151}{0.05}$$

$$= 82.61\ \text{W/m}^2\cdot^\circ\text{C}$$

Therefore the Heat transfer coefficient is $\boxed{h = 82.61\ \text{W/m}^2\cdot^\circ\text{C}}$

Step 4 of 4

(*b*) Air flowing outside:

When air is flowing across tube's outer surface, Nusselt number according to Zukauskas equation is

$$Nu = 0.027\text{Re}^{0.805}\text{Pr}^{1/3}$$

$$= 0.027(69521.7)^{0.805}(0.7228)^{1/3}$$

$$= 191.55$$

Heat transfer coefficient when air is flowing across its outer surface is

$$h = \frac{Nu\cdot k}{D}$$

$$= \frac{191.55\times0.02735}{0.05}$$

$$= 104.78\ \text{W/m}^2\cdot^\circ\text{C}$$

Therefore the Heat transfer coefficient is $\boxed{h = 104.78\ \text{W/m}^2\cdot^\circ\text{C}}$

Convection coefficient h is more when air is being flown across tube's outer surface.

Question: 19.128P

A silicon chip is cooled by passing water through microchannels etched in the back of the chip, as shown in Fig- P19-128. The channels are covered with a silicon cap. Consider a 12-mm × 12-mm square chip in which $N = 60$ rectangular microchannels, each of width $W = 50$ μm and height $H = 200$ μm have been etched. Water enters the microchannels at a temperature $Ti = 290$ K, and a total flow rate of 0.005 kg/s. The chip and cap are maintained at a uniform temperature of 350 K. Assuming that the flow in the channels is fully developed, all the heat generated by the circuits on the top surface of the chip is transferred to the water, and using circular tube correlations, determine:

(*a*) The water outlet temperature, *Te*

(*b*) The chip power dissipation, *We*

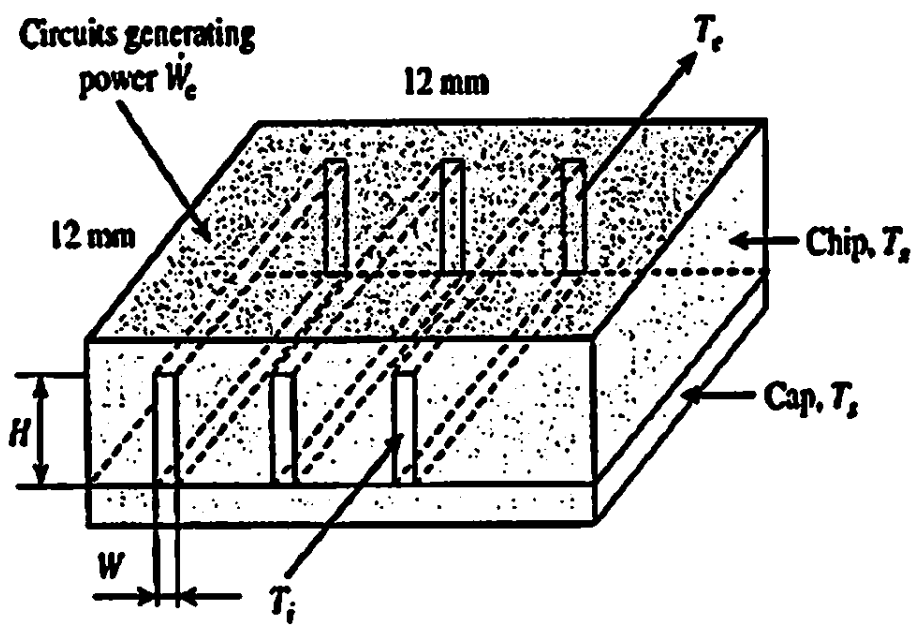


FIGURE P19–128

Step 1 of 6

Given data:

Length of chip, $L = 0.012\,\text{m}$

Width of chip, $w_c = 0.012\,\text{m}$

Number of micro-channels, $N = 60$

Height of micro channel, $H = 200\,\mu\text{m}$

Width of micro channel, $w = 50\,\mu\text{m}$

Inlet water temperature, $T_i = 290\,\text{K}$

Total flow rate, $\dot{m}_t = 0.005\,\text{kg/s}$

Uniform temperature of chip & cap, $T_s = 350\,\text{K}$

Step 2 of 6

The properties of water from properties table at assumed value of $T_{avg} = 30°\text{C}$,

Specific heat, $c_p = 4178\,\text{J/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.615\,\text{W/m}\cdot\text{K}$

Dynamic viscosity, $\mu = 0.798\times10^{-3}\,\text{kg/m}\cdot\text{s}$

Density, $\rho = 996\,\text{kg/m}^3$

Mass flow rate through single channel,

$$\dot{m} = \frac{\dot{m}_t}{N}$$
$$= \frac{0.005}{60}$$
$$= 8.33\times10^{-5}\,\text{kg/s}$$

Cross sectional area,

$$A_c = H \times w$$
$$= 200\times50$$
$$= 10{,}000\,\mu\text{m}^2$$
$$= 10^{-8}\,\text{m}^2$$

Step 3 of 6

Hydraulic diameter,

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4\times10000}{2(200+50)}$$
$$= 80\mu\text{m}$$

Reynolds number,

$$\text{Re} = \frac{\rho V D_h}{\mu}$$
$$= \frac{\rho(\dot{m}/\rho A_c)D_h}{\mu}$$
$$= \frac{\dot{m}D_h}{A_c\mu}$$
$$= \frac{\left(8.33\times10^{-5}\right)\times\left(80\times10^{-6}\right)}{\left(10^{-8}\right)\times\left(0.798\times10^{-3}\right)}$$
$$= 835.08$$

As the obtained Reynolds number is less than 2300, the flow is laminar.

Step 4 of 6

Therefore Nusselt number, $Nu = 3.66$

Heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D_h}$$
$$= \frac{0.615\times3.66}{80\times10^{-6}}$$
$$= 28136.25\,\text{W/m}^2\,°\text{C}$$

Heat transfer surface area,

$$A_s = 2wL + 2HL$$
$$= 2\left(50\times10^{-6}\right)(0.012) + 2\left(200\times10^{-6}\right)(0.012)$$
$$= 1.2\times10^{-6} + 4.8\times10^{-6}$$
$$= 6\times10^{-6}\,\text{m}^2$$

Step 5 of 6

Exit temperature of the water is

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}C_p}\right)$$
$$= 350 - (350-290)\exp\left(-\frac{28136.25\times6\times10^{-6}}{8.33\times10^{-5}\times4178}\right)$$
$$= 350 - (60\times0.61565)$$
$$= 313\,\text{K}$$

Therefore the exit temperature is $\boxed{T_e = 313\text{K}}$

Step 6 of 6

Rate of power dissipated,

$$\dot{W}_e = NQ$$
$$= N\dot{m}c_p(T_e - T_i)$$
$$= 60\times8.333\times10^{-5}\times4178\times(350-313)$$
$$= 772.9\,\text{W}$$

Therefore total power dissipated is $\boxed{\dot{W}_e = 772.9\,\text{W}}$

Question: 19.129P

Water is heated at a rate of 10 kg/s from a temperature of 15°C to 35°C by passing it through five identical tubes, each 5.0 cm in diameter, whose surface temperature is 60.0°C. Estimate (*a*) the steady rate of heat transfer and (*b*) the length of tubes necessary to accomplish this task.

Step 1 of 6

Given data:

Mass flow rate of water in 5 tubes is 10 kg/s

Diameter of each pipe, $D = 0.05$ m

Number of pipes, $n = 5$

Surface temperature, $T_s = 60$ C

Step 2 of 6

Bulk temperature, $T_b = \dfrac{15+35}{2}$

$$T_b = 25°C$$

Properties of saturated water at 25° C (from saturated water tables) is

$\rho = 997$ kg/m^3

$k = 0.607$ W/mC

$\mu = 0.891 \times 10^{-3}$ m^2/s

$c_p = 4180$ J/kgC

$P_r = 6.14$

Step 3 of 6

Cross-sectional area of the pipe, $A_c = \dfrac{\pi}{4}(D^2)$

$$A_c = \frac{\pi}{4}(0.05)^2$$

$$A_c = 1.9635 \times 10^{-3} \text{ m}^2$$

Mass flow rate, $\dot{m} = \rho A_c V_{avg}$

So $V_{avg} = \dfrac{\dot{m}}{\rho A_c}$

$$V_{avg} = \frac{10/5}{997 \times 1.9635 \times 10^{-3}}$$

$$V_{avg} = 1.0216 \text{ m/s}$$

Step 4 of 6

Reynolds number, $\text{Re} = \dfrac{\rho V_{avg} D}{\mu}$

$$\text{Re} = \frac{997 \times 1.0216 \times 0.05}{0.891 \times 10^{-3}}$$

$$\text{Re} = 57156.86$$

This Reynolds number shows that the flow is turbulent

Nusselt number, $\text{Nu} = 0.023\text{Re}^{0.8}\text{Pr}^{0.3}$

$$= 0.023 \times (57156.86)^{0.8} \times (6.14)^{0.3}$$

$$= 253.415$$

But $\text{Nu} = \dfrac{hD}{k}$

so $h = \dfrac{\text{Nu} \times k}{D}$

$$h = \frac{253.415 \times 0.607}{0.05}$$

$$h = 3076.458 \text{ W/m}^2\text{sec}$$

Step 5 of 6

(a) Heat transfer is $\dot{Q} = \dot{m}C_p(T_e - T_i)$

$$\dot{Q} = 10 \times 4180 \times (35-15)$$

$$\dot{Q} = 836000 \text{ W}$$

Hence steady rate of heat transfer, $\boxed{\dot{Q} = 836000 \text{ W}}$

Step 6 of 6

(b) Using the relation for convective heat transfer, $Q = hA_s(T_s - T_b)$

Hence $A_s = \dfrac{Q}{h \times (T_s - T_b)}$

$$A_s = \frac{836000}{3076.458 \times (60-25)}$$

$$A_s = 7.764 \text{ m}^2$$

But surface area is $A_s = n \times \pi DL$

So $L = \dfrac{A_s}{n \times \pi D}$

$$L = \frac{7.764}{5 \times \pi \times 0.05}$$

$$L = 9.88 \text{ m}$$

Hence the length of each tube has to be $\boxed{L = 9.88 \text{ m}}$

Question: 19.130P

Repeat Prob. 19-129 for a flow rate of 20 kg/s.

Step 1 of 6

Given data:

Mass flow rate of water in 5 tubes is 20 kg/s

Diameter of each pipe, $D = 0.05$ m

Number of pipes, $n = 5$

Surface temperature, $T_s = 60$ C

Step 2 of 6

Bulk temperature, $T_b = \dfrac{15+35}{2}$

$T_b = 25°C$

Properties of saturated water at 25° C (from saturated water tables) is

$\rho = 997$ kg/m^3

$k = 0.607$ W/mC

$\mu = 0.891 \times 10^{-3}$ m^2/s

$c_p = 4180$ J/kgC

$P_r = 6.14$

Step 3 of 6

Cross-sectional area of the pipe is $A_c = \dfrac{\pi}{4}(D^2)$

$A_c = \dfrac{\pi}{4}(0.05)^2$

$A_c = 1.9635 \times 10^{-3}$ m^2

Mass flow rate, $\dot{m} = \rho A_c V_{avg}$

So $V_{avg} = \dfrac{\dot{m}}{\rho A_c}$

$V_{avg} = \dfrac{20/5}{997 \times 1.9635 \times 10^{-3}}$

$V_{avg} = 2.0432$ m/s

Step 4 of 6

Reynolds number, $\mathrm{Re} = \dfrac{\rho V_{avg} D}{\mu}$

$\mathrm{Re} = \dfrac{997 \times 2.0432 \times 0.05}{0.891 \times 10^{-3}}$

$\mathrm{Re} = 114313.715$

This Reynolds number value shows that the flow is turbulent

Nusselt number, $\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.3}$

$= 0.023 \times (114313.715)^{0.8} \times (6.14)^{0.3}$

$= 441.222$

But $\mathrm{Nu} = \dfrac{hD}{k}$ so $h = \dfrac{\mathrm{Nu} \times k}{D}$

$h = \dfrac{441.222 \times 0.607}{0.05}$

$h = 5356.435$ W/m^{2}sec

Step 5 of 6

(a) Heat transfer is $\dot{Q} = \dot{m} C_p (T_e - T_i)$

$\dot{Q} = 20 \times 4180 \times (35 - 15)$

$\dot{Q} = 1672000$ W

Hence steady rate of heat transfer, $\boxed{\dot{Q} = 1672000 \text{ W}}$

Step 6 of 6

(b) Using the relation for convective heat transfer, $Q = hA_s(T_s - T_b)$

Hence $A_s = \dfrac{Q}{h \times (T_s - T_b)}$

$A_s = \dfrac{1672000}{5356.435 \times (60 - 25)}$

$A_s = 8.918$ m^2

But surface area is $A_s = n \times \pi DL$

So $L = \dfrac{A_s}{n \times \pi D}$

$L = \dfrac{8.918}{5 \times \pi \times 0.05}$

$L = 11.35$ m

Hence the length of each tube has to be $\boxed{L = 11.35 \text{ m}}$

Question: 19.131P

Water at 1800 kg/h and 10°C enters a 10-mm diameter smooth tube whose wall temperature is maintained al 49°C. Calculate (*a*) the tube length necessary to heat the water to 40°C. and (*b*) the water outlet temperature if the tube length is doubled, Assume average water properties to be the same as in (*a*).

Step 1 of 8

Given data:

Mass flow rate, $\dot{m} = 1800 \text{kg/h}$

Inlet temperature of water $T_i = 10°\text{C}$

Outlet temperature of water, $T_e = 40°\text{C}$

Surface temperature of tube, $T_s = 49°\text{C}$

Diameter of the tube, $D = 0.01\text{m}$

Step 2 of 8

(*a*)

Average temperature,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{10+40}{2}$$
$$= 25°\text{C}$$

Properties of water at average temperature,

Specific heat, $c_p = 4180 \text{J/kgK}$

Density, $\rho = 997.0 \text{kg/m}^3$

Thermal conductivity, $k = 0.607 \text{ W/m} \cdot \text{K}$

Dynamic viscosity, $\mu = 0.891 \times 10^{-3} \text{kg/m} \cdot s$

Prandtl number, $\text{Pr} = 6.14$

Step 3 of 8

Average velocity at which the water flows is

$$V_{avg} = \frac{4\dot{m}}{\rho \pi D^2}$$
$$= \frac{4 \times 0.5}{997 \times \pi (0.01)^2}$$
$$= 6.385 \text{ m/s}$$

Rate of heat transfer of the water is

$$\dot{Q} = \dot{m} C_p (T_e - T_i)$$
$$= 0.5 \times 4180 \times (40 - 10)$$
$$= 62700 \text{W}$$

Step 4 of 8

Reynolds number for the flow is

$$\text{Re} = \frac{\rho V_{avg} D}{\mu}$$
$$= \frac{997 \times 6.385 \times 0.01}{0.891 \times 10^{-3}}$$
$$= 71446.07$$

As the Reynolds number is greater than 10,000 the flow is turbulent flow.

$$Nu = 0.023 \times \text{Re}^{0.8} \times \text{Pr}^{0.4}$$
$$= 0.023 (71446.07)^{0.8} (6.14)^{0.4}$$
$$= 363.23$$

Heat transfer coefficient for the water flowing through the tube is

$$h = \frac{k \cdot Nu}{D}$$
$$= \frac{0.607 \times 363.23}{0.01}$$
$$= 22047.86 \text{W/m}^2 °\text{C}$$

Step 5 of 8

Logarithmic mean temperature,

$$\Delta T_{lm} = \frac{T_i - T_e}{\ln\left[\frac{T_s - T_e}{T_s - T_i}\right]}$$
$$= \frac{10-40}{\ln\left[\frac{49-40}{49-10}\right]}$$
$$= 20.46°\text{C}$$

Surface area along which the heat transfer takes place is

$$A_s = \pi D L$$
$$= \pi (0.01) L$$
$$= 0.03142\, L$$

Step 6 of 8

Thus the heat transfer rate is expressed by using the following relation,

$$\dot{Q} = h A_s \Delta T_{lm}$$
$$62700 = 22047.86 \times 0.03142\, L \times 20.46$$
$$L = \frac{62700}{14173.537}$$
$$L = 4.42 \text{ m}$$

Therefore the length of the tube is $\boxed{L = 4.42\text{m}}$

Step 7 of 8

(*b*)

By considering double the length,

$$L = 4.42 \times 2$$
$$= 8.84\text{m}$$

Surface area along which the heat transfer takes place is

$$A_s = \pi D L$$
$$= \pi (0.01) \times 8.84$$
$$= 0.278\text{m}^2$$

Step 8 of 8

Thus the heat transfer rate is expressed by using the following relation,

$$\dot{Q} = h A_s \Delta T_{lm}$$
$$\dot{m} c_p (T_e - T_i) = h A_s \Delta T_{lm}$$
$$(T_e - T_i) = \frac{h A_s}{\dot{m} c_p} \Delta T_{lm}$$
$$(T_e - T_i) = \frac{22047.86 \times 0.278}{0.5 \times 4180} \times \left[\frac{T_e - 10}{\ln\left(\frac{49 - T_e}{49 - 10}\right)}\right]$$
$$(T_e - 10) = \frac{6129.3}{2090} \left[\frac{T_e - 10}{\ln\left(\frac{49 - T_e}{49 - 10}\right)}\right]$$

The solution of the above equation shows that $T_e = -683.37°\text{C}$, which is not practically feasible. So increasing the length is not necessary.

Question: 19.132P

Liquid water enters a 10-m-long smooth rectangular tube with a =50 mm and b= 25 mm. The surface temperature is maintained constant, and water enters the tube at 20°C with a mass How rate of 0.25 kg/s. Determine the tube surface temperature necessary to heat the water to the desired outlet temperature of 80°C.

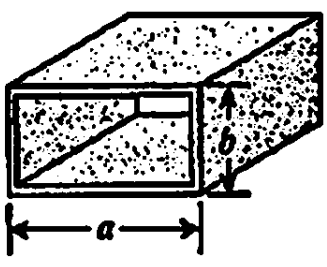


FIGURE P19–132

Step 1 of 5

Given data:

Length of smooth rectangular tube, $L = 10\,\text{m}$

Width of the tube $a = 50\,\text{mm} = 0.05\text{m}$

Height of the tube $b = 25\,\text{mm} = 0.025\text{m}$

Water Inlet temperature, $T_i = 20°\text{C}$

Mass flow rate, $\dot{m} = 0.25\,\text{kg/s}$

Outlet temperature, $T_e = 80°\text{C}$

Step 2 of 5

Average temperature,

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{20+80}{2}$$
$$= 50°\text{C}$$

Properties of water at average temperature from properties table,

Density, $\rho = 988.1\,\text{kg/m}^3$

Specific heat, $c_p = 4181\,\text{J/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.644\,\text{W/m}\cdot\text{K}$

Dynamic viscosity, $\mu = 0.547\times10^{-3}\,\text{kg/m}\cdot\text{s}$

Prandtl number, $\text{Pr} = 3.55$

Step 3 of 5

Hydraulic Diameter of the tube is

$$D_h = \frac{4A_c}{p}$$
$$= \frac{4ab}{2(a+b)}$$
$$= \frac{4\times0.05\times0.025}{2(0.05+0.025)}$$
$$= 0.0333\text{m}$$

Reynolds number for the flow is

$$\text{Re} = \frac{\rho V_{avg} D_h}{\mu}$$
$$= \frac{\rho(\dot{m}/\rho A_c)D_h}{\mu}$$
$$= \frac{\dot{m}\times D_h}{A_c\times\mu}$$
$$= \frac{0.25\times0.0333}{(0.05\times0.025)\times\left(0.547\times10^{-3}\right)}$$
$$= 12187.7$$

Step 4 of 5

The calculated value of Reynolds number is greater than 10000 therefore it is a turbulent flow. Now the entry length is determined by

$$L_h \approx L_t$$
$$\approx 10D_h$$
$$= 10\times0.0333$$
$$= 0.333\text{m}$$

As the entry length is less than given length, the flow is fully developed turbulent flow. Thus by Dittus-Boelter equation, Nusselt number is

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$
$$= 0.023(12187.7)^{0.8}(3.55)^{0.4}$$
$$= 70.89$$

Heat transfer coefficient for the flow is

$$h = \frac{k\cdot Nu}{D_h}$$
$$= \frac{0.644\times70.89}{0.0333}$$
$$= 1370.87\,\text{W/m}^2°\text{C}$$

Step 5 of 5

Surface area along which the heat transfer takes place is

$$A_s = 2\times L\times(a+b)$$
$$= 2\times10(0.05+0.025)$$
$$= 1.5\text{m}^2$$

Therefore from the relation for exit temperature,

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right)$$
$$80 = T_s - (T_s - 20)\exp\left(-\frac{1370.87\times1.5}{0.25\times4181}\right)$$
$$80 = T_s - 0.1398T_s + 2.7967$$
$$T_s = 89.75°\text{C}$$

Therefore the tube surface temperature is $\boxed{T_s = 89.75°\text{C}}$

Question: 19.133P

Air is flowing through a smooth thin-walled 4-in diameter copper tube that is submerged in water. The water maintains a constant temperature of 60°F and a convection heat transfer coefficient of 176 Btu/h-ft2•R. If air (1 atm) enters the copper tube at a mean temperature of 90°F with an average velocity of 8 ft/s, determine the necessary copper tube length so that the outlet mean temperature of the air is 70°F.

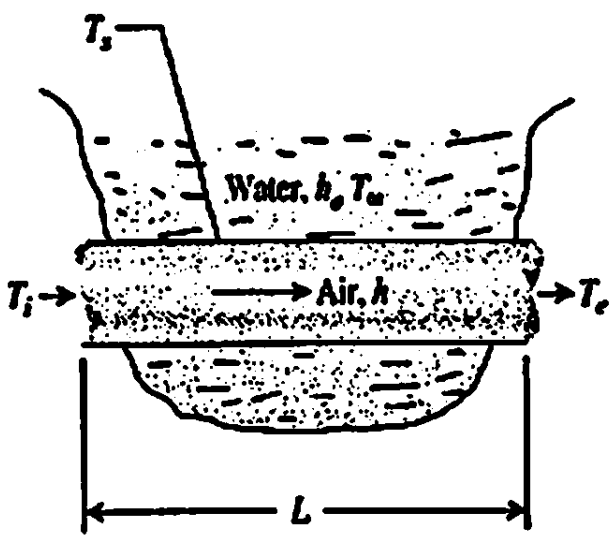


FIGURE P19–133E

Step 1 of 7

Given data:

Diameter of copper tube, $D = 4\,\text{inch} = 0.333\,\text{ft}$

Water temperature, $T_\infty = 60°\text{F}$

Convection coefficient, $h_0 = 176\,\text{Btu}/\text{ft}^2 \cdot \text{R}$

Air inlet temperature, $T_i = 90°\text{F}$

Velocity average, $V_{avg} = 8\,\text{ft}/s$

Outlet air temperature, $T_e = 70°\text{F}$

Step 2 of 7

Mean temperature is

$$T_{avg} = \frac{T_i + T_e}{2}$$
$$= \frac{90+70}{2}$$
$$= 80°\text{F}$$

From the properties of the various gases at atmospheric pressure

Density, $\rho = 0.07350\,\text{lbm}/\text{ft}^3$

Specific heat, $c_p = 0.2404\,\text{Btu}/\text{lbm} \cdot \text{R}$

Thermal conductivity, $k = 0.01481\,\text{Btu}/\text{h} \cdot \text{ft} \cdot \text{R}$

Kinematic viscosity, $\nu = 1.697 \times 10^{-4}\,\text{ft}^2/s$

Prandtl number, $\text{Pr} = 0.7290$

Step 3 of 7

Reynolds number,

$$\text{Re} = \frac{V_{avg} \cdot D}{\nu}$$
$$= \frac{8 \times 0.333}{1.697 \times 10^{-4}}$$
$$= 15714$$

This is greater than 10,000 , so the flow is turbulent and also assumes that the flow is a fully developed one.

Step 4 of 7

By using the Dittus – Boelter equation, Nusselt number can be expressed as,

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$
$$= 0.023 \times (15714)^{0.8} \cdot (0.7290)^{0.4}$$
$$= 46.11$$

Thus the Nusselt number (Nu) is expressed as

$$Nu = \frac{hD}{k}$$
$$h = \frac{Nu \times k}{D}$$
$$= \frac{46.11 \times 0.01481}{0.333}$$
$$h = 2.051\,\text{Btu}/\text{h} \cdot \text{ft}^2 \cdot °\text{F}$$

Step 5 of 7

Calculating mass flow rate, $\dot{m} = \rho A_c V_{avg}$

Where,

A_c – Cross-sectional area of tube, $\frac{\pi D^2}{4}$

Thus the mass flow rate is

$$\dot{m} = \rho \times \frac{\pi D^2}{4} \times V_{avg}$$
$$= \frac{0.07350 \times \pi \times (0.333)^2 \times 8}{4}$$
$$= 0.0512\,\text{lbm}/s$$

Converting the mass flow rate per hour,

$$\dot{m} = 0.0573 \times 3600\,\text{lbm}/\text{hour}$$
$$= 184.36\,\text{lbm}/\text{h}$$

Step 6 of 7

Therefore heat transfer to lower temperature,

$$\dot{Q} = \dot{m}C_p(T_i - T_e)$$
$$= 184.36 \times 0.2404 \times (90-70)$$
$$= 886.39\,\text{Btu}/\text{hr}$$

Now calculating surface temperature T_s per 1 ft length of tube by the equation

$$\dot{Q} = h_o A_s (T_s - T_\infty)$$

Surface area along which the heat transfer takes place is

$$A_s = \pi DL$$
$$= \pi \times 0.333 \times 1$$
$$= 1.047\,\text{ft}^2$$

Step 7 of 7

Thus from the steady conditions we get the heat given by air will be equal to the heat transferred to water.

$$\dot{Q} = h_o A_s (T_s - T_\infty)$$
$$886.39 = 176 \times 1.047 (T_s - 60)$$
$$T_s - 60 = 4.81$$
$$T_s = 64.81°\text{F}$$

Now, putting the value of T_s & all other quantities in the equation

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{h\pi DL}{\dot{m}C_p}\right)$$

$$70 = 64.81 - (64.81-90)\exp\left(\frac{-2.051 \times 1.047\,L}{184.36 \times 0.2404}\right) \qquad [\because A_s = \pi DL]$$

$$L = 32.60\,\text{ft}$$

Therefore, length of the pipe is $\boxed{L = 32.60\,\text{ft}}$

Question: 19.134P pg 1

Hot water at 90°C enters a 15-m section of a cast iron pipe (k= 52 W/m·K) whose inner and outer diameters are 4 and 4.6 cm, respectively, at an average velocity of 1.2m/s. The outer surface of the pipe, whose emissivity is 0.7, is exposed to the cold air at 10°C in a basement, with a convection heat transfer coefficient of 12 W/m2•K. Taking the walls of the basement to be at 10°C also, determine (*a*) the rate of heat loss from the water and (*b*) the temperature at which the water leaves the basement.

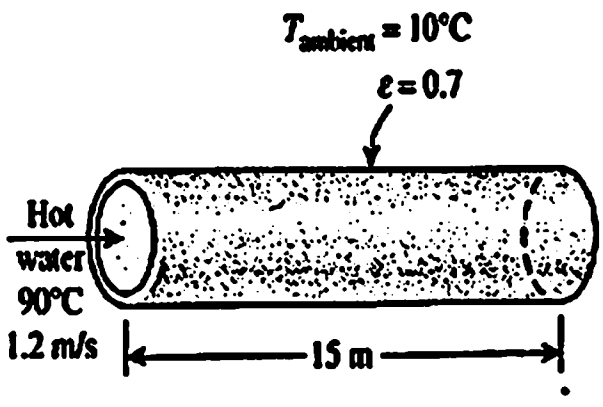
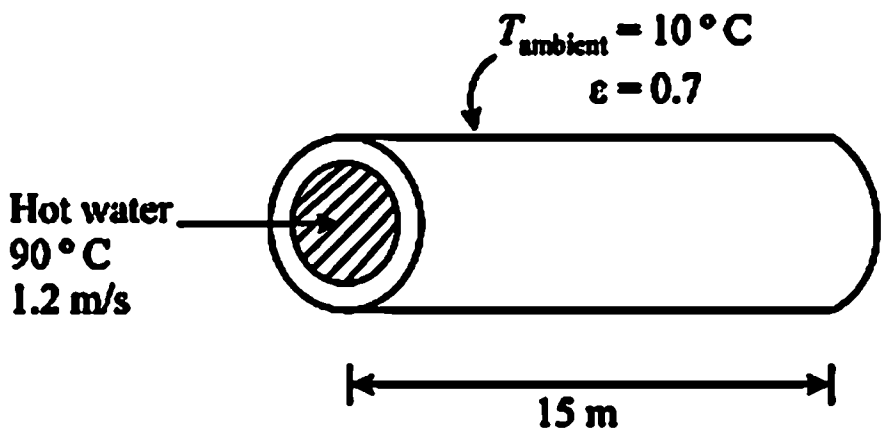


FIGURE P19–134

Step 1 of 13

The schematic diagram shows, hot water flows through cast iron pipe, and exposed to ambient temperature.



Step 2 of 13

Assume average temperature of the water is $T_{avg} = 90°C$.

Take properties of water from the table, 'properties of saturated water' at $T_{avg} = 90°C$

Density, $\rho = 965.3 kg/m^3$

Specific heat, $c_p = 4206 J/kg \cdot K$

Thermal conductivity, $k = 0.675 W/m \cdot K$

Dynamic viscosity, $\mu = 0.315 \times 10^{-3} N \cdot sec/m^2$

Prandtl number, $Pr = 1.96$

Step 3 of 13

Calculate kinematic viscosity of water by using the expression,

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.315 \times 10^{-3}}{965.3}$$
$$= 3.263 \times 10^{-7} m^2/s$$

Step 4 of 13

Calculate the Reynolds number by using the expression,

$$Re = \frac{V_{avg} D_i}{\nu}$$

Here, V_{avg} is average velocity of water flows in a pipe and D_i is inside diameter of cast iron pipe.

Substitute $1.2 m/s$ for V_{avg} and $0.04 m$ for D_i

$$Re = \frac{1.2 \times 0.04}{3.263 \times 10^{-7}}$$
$$= 147093.33$$

This is greater than $10,000$, therefore, the flow of water is turbulent flow.

Step 5 of 13

Check the entry length of hydrodynamic and turbulent boundary length by using the expression,

$$L_h \approx L_t$$
$$\approx 10 D_i$$
$$= 10 \times 0.04$$
$$= 0.4 m$$

This is much smaller than the given length $L = 15 m$

Therefore, it is a fully developed turbulent flow.

Step 6 of 13

From the table of 'equivalent roughness values for the new commercial pipes'
The roughness factor of cast iron pipe is,

$$\varepsilon = 0.26 mm$$
$$= 0.26 \times 10^{-3} m$$

Calculate the roughness ratio by using the equation,

$$\frac{\varepsilon}{D_i} = \frac{0.26 \times 10^{-3}}{0.04}$$
$$= 6.5 \times 10^{-3}$$

Step 7 of 13

Take friction factor f, from Moody's chart by interpolating the values between $(0.005 - 0.01)$

$$f = 0.0305 + \frac{0.0380 - 0.0305}{(0.01 - 0.005)}(0.0065 - 0.005)$$
$$= 0.03275$$

Calculate Nusselt number by using Chilton – Colburn analogy expression for rough tubes,

$$Nu = \frac{hD_i}{k} = 0.125 f . Re . Pr^{1/3}$$

$=0.03275$

Calculate Nusselt number by using Chilton – Colburn analogy expression for rough tubes,

$$\mathrm{Nu}=\frac{hD_i}{k}=0.125\,f.\mathrm{Re}.\mathrm{Pr}^{1/3}$$
$$=0.125\times0.03275\times147093.33\times(1.96)^{1/3}$$
$$=751.896$$

Step 8 of 13

Then calculate the heat transfer coefficient by using Nusselt number,

$$\mathrm{Nu}=\frac{hD_i}{k}$$
$$h=\frac{k\cdot\mathrm{Nu}}{D_i}$$
$$=\frac{0.675\times751.896}{0.04}$$
$$=12688.245\,\mathrm{w/m^2\cdot{}^\circ C}$$

Step 9 of 13

Calculate thermal conduction resistance in this pipe,

$$R=\frac{\ln(D_o/D_i)}{2\pi k_{CI}L}$$

Here, D_o is outer diameter of the pipe, L is length of the pipe and k_{CI} is thermal conductivity of cast iron pipe.

Substitute $0.046\,\mathrm{m}$ for D_o, $15\,\mathrm{m}$ for L and $52\,\mathrm{W/m\text{-}K}$ for k_{CI}

$$R=\frac{\ln(0.046/0.04)}{2\pi\times52\times15}$$
$$=2.85\times10^{-5}\,^\circ\mathrm{C/W}$$

Thermal conduction resistance is very less; therefore, so outer and inner temperature will be almost equal.

Therefore, assume the surface temperature is $T_s=90^\circ\mathrm{C}$

Step 10 of 13

Calculate the heat transfer surface area by using the equation,

$$A_s=\pi D_oL$$
$$=\pi\times0.046\times15$$
$$=2.168\,\mathrm{m^2}$$

Step 11 of 13

(a)

Calculate the total heat transfer due to convection and radiation to atmosphere from outer pipe surface.

$$Q_{\text{total}}=Q_{\text{conv}}+Q_{\text{rad}}$$
$$=h_oA_s(T_s-T_{\text{amb}})+\varepsilon A_s\sigma\left[T_s^4-T_{\text{surr}}^4\right]$$

Here, h_o is convection heat transfer coefficient, T_{amb} is ambient temperature of air, T_{surr} is surrounding temperature of walls, ε is emissivity of pipe surface and σ is Stefan Boltzmann's constant,

Substitute $10^\circ\mathrm{C}$ for T_{surr}, T_{amb}, 0.7 for ε, $12\,\mathrm{W/m^2\cdot K}$ for h_o and $5.67\times10^{-8}\,\mathrm{W/m^2\cdot K^4}$ for σ

$$Q_{total}=12\times2.168(90-10)+0.7\times2.168\times5.67\times10^{-8}\left[(90+273)^4-(10+273)^4\right]$$
$$=2081.28+942.12$$
$$=3023.4\,\mathrm{W}$$

Therefore, total heat loss from the water Q_{total} is $\boxed{3023.4\,\mathrm{W}}$

Step 12 of 13

(b)

Calculate mass flow rate of water by using the equation,

$$\dot{m}=\rho A_cV_{avg}$$

Here, A_c cross sectional area

Substitute $\frac{\pi D_i^2}{4}$ for A_c

$$\dot{m}=\frac{\rho\times\pi D_i^2}{4}\times V_{avg}$$
$$=\frac{965.3\times\pi\times(0.04)^2\times1.2}{4}$$
$$=1.456\,\mathrm{kg/s}$$

Step 13 of 13

Apply energy balance equation by equating; Total heat lost by pipe is equal to the heat gain by water flows through the pipe, $\dot{Q}$

$$Q_{total}=mc_p(T_i-T_e)$$

Here, T_i is water inlet temperature and T_e is water exit temperature.

Substitute $90^\circ\mathrm{C}$ for T_i

$$3023.4=1.456\times4206(90-T_e)$$
$$T_e=89.51^\circ\mathrm{C}$$

Therefore, exit temperature of the water T_e is $\boxed{89.51^\circ\mathrm{C}}$

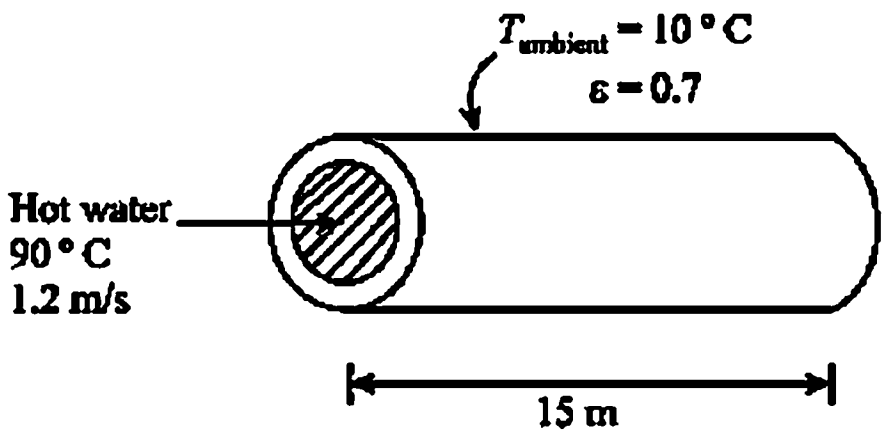
Also our assumed mean temperature is right as there is not much difference between inlet and outlet temperature.

Question: 19.135P pg 1

Repeat Prob. 19-134 for a pipe made of copper (k =386 W/m·K) instead of cast iron.

Step 1 of 12

The schematic diagram shows, hot water flows through cast iron pipe, and exposed to ambient temperature.



Step 2 of 12

Assume average temperature of the water is $T_{avg} = 90°C$.

Take properties of water from the table 'properties of saturated water' at $T_{avg} = 90°C$

Density, $\rho = 965.3 kg/m^3$

Specific heat, $c_p = 4206 J/kg \cdot K$

Thermal conductivity, $k = 0.675 W/m \cdot K$

Dynamic viscosity, $\mu = 0.315 \times 10^{-3} N \cdot sec/m^2$

Prandtl number, $Pr = 1.96$

Step 3 of 12

Calculate kinematic of viscosity of water by using the expression,

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.315 \times 10^{-3}}{965.3}$$
$$= 3.263 \times 10^{-7}\ m^2/s$$

Step 4 of 12

Calculate the Reynolds number by using the expression,

$$\mathrm{Re} = \frac{V_{avg} D_i}{\nu}$$

Here V_{avg} is average velocity of water flows in a pipe and D_i is inside diameter of cast iron pipe.

Substitute $1.2 m/s$ for V_{avg} and $0.04 m$ for D_i

$$\mathrm{Re} = \frac{1.2 \times 0.04}{3.263 \times 10^{-7}}$$
$$= 147093.33$$

This is greater than $10,000$, Therefore, the flow of water is turbulent flow.

Step 5 of 12

Check the entry length of hydrodynamic and turbulent boundary length by using the expression,

$$L_h \approx L_t$$
$$\approx 10 D_i$$
$$= 10 \times 0.04$$
$$= 0.4 m$$

This is much smaller than the given length $L = 15 m$

Therefore, it is a fully developed turbulent flow.

Step 6 of 12

From the table of 'equivalent roughness values for the new commercial pipes'
The roughness factor of cast iron pipe is,

$$\varepsilon = 0.0015 mm$$
$$= 1.5 \times 10^{-6} m$$

Calculate the roughness ratio by using the equation,

$$\frac{\varepsilon}{D_i} = \frac{1.5 \times 10^{-6}}{0.04}$$
$$= 3.75 \times 10^{-5}$$

Step 7 of 12

Take friction factor f, from Moody's chart at $\varepsilon / D = 3.75 \times 10^{-5}$

$f = 0.0119$

Calculate Nusselt number by using Chilton – Colburn analogy expression for rough tubes,

$$\mathrm{Nu} = \frac{hD_i}{k} = 0.125 f . \mathrm{Re} . \mathrm{Pr}^{1/3}$$
$$= 0.125 \times 0.0119 \times 147093.33 \times (1.96)^{1/3}$$
$$= 273.208$$

Then calculate the heat transfer coefficient by using Nusselt number,

$$\mathrm{Nu} = \frac{hD_i}{k}$$
$$h = \frac{k \cdot \mathrm{Nu}}{D_i}$$
$$= \frac{0.675 \times 273.208}{0.04}$$
$$= 4610.385\ W/m^2 \cdot {}^{\circ}C$$

Step 8 of 12

Calculate thermal conduction resistance in this pipe,

$$R = \frac{\ln(D_o/D_i)}{}$$

Step 8 of 12

Calculate thermal conduction resistance in this pipe,

$$R = \frac{\ln(D_o/D_i)}{2\pi k_{copper} L}$$

Here, D_o is outer diameter of the pipe, k_{copper} is thermal conductivity of copper and L is length of the pipe.

Substitute 0.046 m for D_o, 386 W/m·K for k_{copper} and 15 m for L

$$R = \frac{\ln(0.046/0.04)}{2\pi \times 386 \times 15}$$
$$= 3.84 \times 10^{-6}\ ^\circ\text{C/W}$$

Thermal conduction resistance is very less; therefore, so outer and inner temperature will be almost equal.

Therefore, assume the surface temperature is $T_s = 90^\circ\text{C}$

Step 9 of 12

Calculate the heat transfer surface area by using the equation,

$$A_s = \pi D_o L$$
$$= \pi \times 0.046 \times 15$$
$$= 2.168\ \text{m}^2$$

Step 10 of 12

(a)

Calculate the total heat transfer due to convection and radiation to atmosphere from outer pipe surface.

$$Q_{\text{total}} = Q_{\text{conv}} + Q_{\text{rad}}$$
$$= h_o A_s (T_s - T_{\text{amb}}) + \varepsilon A_s \sigma \left[T_s^4 - T_{\text{surr}}^4\right]$$

Here, h_o is convection heat transfer coefficient, T_{amb} is ambient temperature of air, T_{surr} is surrounding temperature of walls, ε is emissivity of pipe surface and σ is Stefan Boltzmann's constant,

Substitute 10°C for T_{surr}, T_{amb}, 0.7 for ε, 12 W/m²·K for h_o and 5.67×10^{-8} W/m²·K⁴ for σ

$$Q_{total} = 12 \times 2.168(90-10) + 0.7 \times 2.168 \times 5.67 \times 10^{-8} \left[(90+273)^4 - (10+273)^4\right]$$
$$= 2081.28 + 942.12$$
$$= 3023.4\ \text{W}$$

Therefore, total heat loss from the water Q_{total} is $\boxed{3023.4\ \text{W}}$

Step 11 of 12

(b)

Calculate mass flow rate of water by using the equation,

$$\dot{m} = \rho A_c V_{avg}$$

Here, A_c cross sectional area

Substitute $\frac{\pi D_i^2}{4}$ for A_c

$$\dot{m} = \frac{\rho \times \pi D_i^2}{4} \times V_{avg}$$
$$= \frac{965.3 \times \pi \times (0.04)^2 \times 1.2}{4}$$
$$= 1.456\ \text{kg/s}$$

Step 12 of 12

Apply energy balance equation by equating; Total heat lost by pipe is equal to the heat gain by water flows through the pipe, $\dot{Q}$

$$Q_{total} = mc_p(T_i - T_e)$$

Here, T_i is water inlet temperature and T_e is water exit temperature.

Substitute 90°C for T_i

$$3023.4 = 1.456 \times 4206(90 - T_e)$$
$$T_e = 89.51^\circ\text{C}$$

Therefore, exit temperature of the water T_e is $\boxed{89.51^\circ\text{C}}$

Also our assumed mean temperature is right as there is not much difference between inlet and outlet temperature.

Question: 19.136P

Hot exhaust gases leaving a stationary diesel engine at 450°C enter a 15-cm-diameter pipe at an average velocity of 7.2 m/s. The surface temperature of the pipe is 180°C. Determine the pipe length if the exhaust gases are to leave the pipe at 250°C after transferring heat to water in a heat recovery unit. Use properties of air for exhaust gases.

Step 1 of 9

Given data:

Diameter of the pipe, $D = 15\,\text{cm} = 0.15\,\text{m}$

Inlet temperature of gas, $T_i = 450°\text{C}$

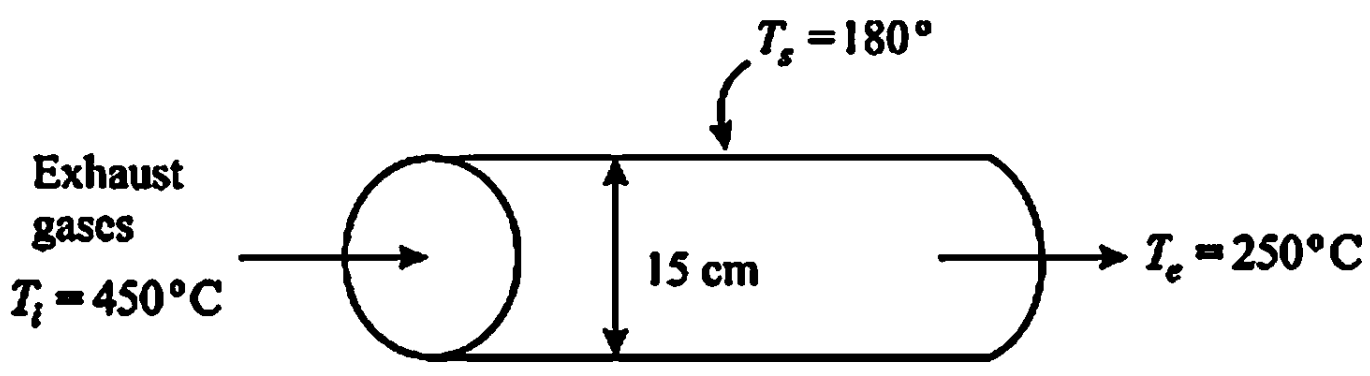
Outlet temperature of gas, $T_e = 250°\text{C}$

Average entrance velocity, $V_{avg} = 7.2\,\text{m/s}$

Surface temperature of tube, $T_s = 180°\text{C}$

Step 2 of 9

Schematic diagram of pipe flow:



Step 3 of 9

Mean film temperature of the air is

$$T_{avg} = (T_i + T_e)/2$$
$$= (450+250)/2$$
$$= 350°\text{C}$$

From the properties of the air table

Density, $\rho = 0.5664\,\text{kg/m}^3$

Specific heat at constant pressure, $c_p = 1056\,\text{J/kg}\cdot\text{k}$

Thermal conductivity, $k = 0.04721\,\text{W/m}\cdot\text{k}$

Kinematic viscosity, $\nu = 5.475\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.6937$

Step 4 of 9

Thus the Reynolds number for the flow is

$$\text{Re} = \frac{V_{avg}\cdot D}{\nu}$$
$$= \frac{7.2\times0.15}{5.475\times10^{-5}}$$
$$= 19726.03$$

Since it is greater than 10,000 , therefore it is turbulent flow

Therefore assuming it is a fully developed one.

Step 5 of 9

By Dittus – Boelter equation.

Nusselt number,

$$Nu = \frac{hD}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.3}$$
$$= 0.023(19726.03)^{0.8}(0.6937)^{033}$$
$$= 55.63$$

Then, Heat transfer coefficient is

$$Nu = \frac{hD}{k}$$
$$h = \frac{k\cdot Nu}{D}$$
$$= \frac{0.04721\times55.63}{0.15}$$
$$= 17.50\,\text{W/m}^2\cdot°\text{C}$$

Step 6 of 9

Therefore logarithmic mean temperature,

$$\Delta T_{lm} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}$$
$$= \frac{250-450}{\ln\left(\frac{180-250}{180-450}\right)}$$
$$= 148.16°\text{C}$$

Step 7 of 9

Heat transfer surface Area per meter length,

$$A_s = \pi DL$$
$$= \pi\times0.15\times1$$
$$= 0.4712\,\text{m}^2$$

Heat transfer per meter length of tube,

$$Q_m = hA_s\Delta T_{lm}$$
$$= 17.50\times0.4712\times148.16$$
$$= 1221..727\,\text{W}$$

Step 8 of 9

Now we will calculate total heat transfer in pipe by the expression

$$Q_t = \dot{m}C_p(T_i - T_e) \qquad \ldots\ldots (1)$$

Where,

Mass flow rate is $\dot{m} = \rho A_c V_{avg}$

A_c – Cross section area of pipe $= \frac{\pi D^2}{4}$

Then the mass flow rate is expressed as

$$\dot{m} = \frac{\rho\times\pi D^2\times V_{avg}}{4}$$

Step 9 of 9

Thus substituting the known values in equation (1) we get the heat transfer rate as

$$Q_t = \frac{\rho\times\pi D^2\times V_{avg}}{4}\times c_p\times(T_i - T_e)$$
$$= \frac{0.5664\times\pi\times(0.15)^2\times7.2\times1056\times(450-250)}{4}$$
$$= 15220.26\,\text{W}$$

Thus the length of the pipe is determined by

$$L = \frac{\text{Total heat transfer, } Q_t}{\text{heat transfer/m, } Q_m}$$
$$= \frac{15220.26}{1235.79}$$
$$= 12.31\text{m}$$

Therefore, length of the pipe is $\boxed{L = 12.31\text{m}}$

Question: 19.137P

Geothermal steam at 165°C condenses in the shell side of a heat exchanger over the tubes through which water flows. Water enters the 4-cm-diameter, 14-m-long tubes at 20°C at a rate of 0.8 kg/s. Determine the exit temperature of water and the rate of condensation of geothermal steam.

Step 1 of 10

Given data:

Diameter of tube, $D = 4\,\text{cm}$

Length of tube, $L = 14\,\text{m}$

Inlet temperature of water, $T_i = 20°\text{C}$

Mass flow rate, $\dot{m} = 0.8\,\text{kg/s}$

Steam temperature, $T_s = 165°\text{C}$

Step 2 of 10

As given, the steam at 165°C is condensing over the tubes.

Therefore assuming the surface temperature of tube is also 165°C

Therefore $T_s = 165°\text{C}$

Therefore as tube surface $T_s = 165°\text{C}$ & entrance temperature of water is 20°C

Therefore assuming mean temperature as 90°C

Taking properties of water at 90°C

Density of the water, $\rho = 965.3\,\text{kg/m}^3$

Specific heat of the water, $C_p = 4206\,\text{J/kg}\cdot\text{k}$

Thermal conductivity, $k = 0.675\,\text{w/m}\cdot\text{k}$

Dynamic viscosity, $\mu = 0.315\times10^{-3}\,\text{kg/m}\cdot\text{s}$

Prandtl number, $\text{Pr} = 1.96$

Step 3 of 10

As we know kinematic viscosity,

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{0.315\times10^{-3}}{965.3}$$

$$= 3.263\times10^{-7}\,\text{m}^2/\text{s}$$

Step 4 of 10

Now average velocity,

$$V_{avg} = \frac{\dot{m}}{\rho A_c}$$

Mass flow rate, $\dot{m} = \rho A_c V_{avg}$

Cross sectional area of pipe, $A_c = \frac{\pi D^2}{4}$

Then,

$$V_{avg} = \frac{4\dot{m}}{\rho\pi D^2}$$

$$= \frac{4\times0.8}{965.3\times\pi\times(0.04)^2}$$

$$= 0.6595\,\text{m/s}$$

Step 5 of 10

Therefore, Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{0.6595\times0.04}{3.263\times10^{-7}}$$

$$= 80845.84$$

This is larger than 10,000

Step 6 of 10

Let's check entry length

Therefore

$$L_h \approx L_t$$

$$\approx 10\text{D}$$

$$\approx 10\times0.04$$

$$\approx 0.4\,\text{m}$$

This is less than the given length, $L = 14\,\text{m}$

Therefore flow is a fully developed turbulent flow.

By the Dietus-Boelter equation,

Nusselt number,

$$Nu = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}$$

$$= 0.023(80845.84)^{0.8}(1.96)^{0.4}$$

$$= 253.96$$

Step 7 of 10

Also,

$$Nu = \frac{hD}{k}$$

Therefore heat transfer coefficient,

$$h = \frac{k\cdot Nu}{D}$$

$$= \frac{0.675\times253.96}{0.04}$$

$$= 4285.49\,\text{W/m}^2\,°\text{C}$$

Step 8 of 10

Heat transfer surface area,

$$A_s = \pi DL$$

$$= \pi\times0.04\times14$$

$$= 1.7593\,\text{m}^2$$

Now by the energy balance relation,

$$T_e = T_s - (T_s - T_i)\exp\left(\frac{-hA_s}{\dot{m}C_p}\right)$$

$$= 165 - (165-20)\exp\left(\frac{-4285.49\times1.7593}{0.8\times4206}\right)$$

$$= 149.57°\text{C}$$

Therefore, the exit temperature of the water, $\boxed{T_e = 149.57°\text{C}}$

Step 9 of 10

Therefore assumed mean temperature is nearly equal.

As we know rate of condensation,

$$m_R = \frac{\text{Total heat transfer to steam}}{\text{Enthalpy of vapourisation}}$$

$$= \frac{\dot{Q}}{h_{fg}}$$

Calculating $\dot{Q}$ from the expression,

$$\dot{Q} = \dot{m}C_p(T_e - T_i)$$

$$= 0.8\times4206\times(149.57-20°\text{C})$$

$$= 435977.136\,\text{W}$$

Step 10 of 10

Taking value of h_{fg} at 165°C from water properties table,

Therefore

$$h_{fg} = \frac{2083+2050}{2}$$

$$= \frac{4133}{2}$$

$$= 2066.5\,\text{kJ/kg}$$

Therefore

$$\dot{Q} = 435.977\,\text{KW}$$

Now,

$$\dot{m}_R = \frac{435.977}{2066.5}$$

Therefore, the rate of condensation of geothermal steam, $\boxed{\dot{m}_R = 0.2109\,\text{kg/s}}$

Question: 19.138P

Cold air at 5°C enters a 12-cm-diameter 20-m-long isothermal pipe at a velocity of 2.5 m/s and leaves at 19°C. Estimate the surface temperature of the pipe.

Step 1 of 9

Given data:

Diameter of pipe, $D = 12\,\text{cm}$

Length of the pipe, $L = 20\,\text{m}$

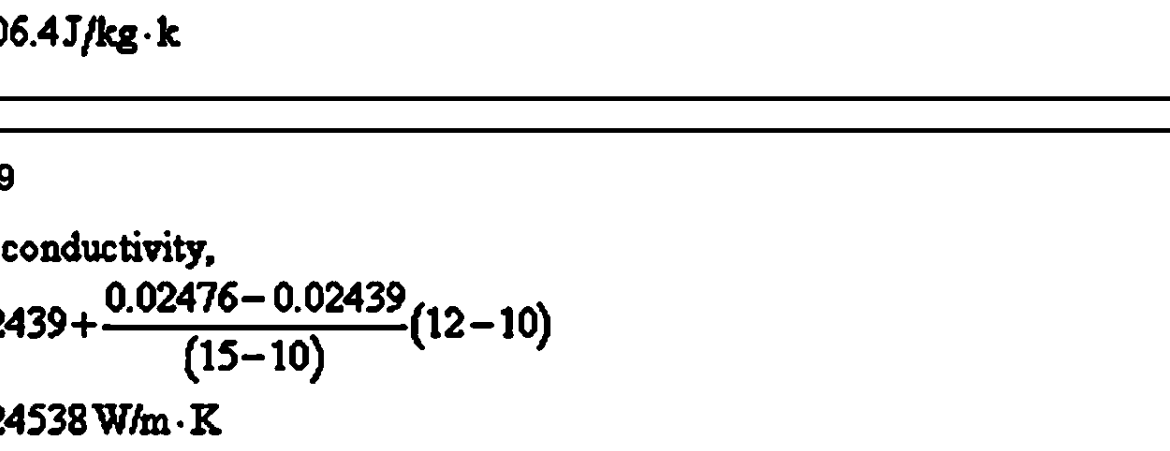
Inlet temperature of air, $T_{inlet} = 5^\circ\text{C}$

Outlet temperature of air, $T_{exit} = 19^\circ\text{C}$

Average velocity, $V_{avg} = 2.5\,\text{m/s}$

Step 2 of 9

Schematic diagram of pipe flow:



Step 3 of 9

Mean film temperature is

$$T_{avg} = \frac{(T_i + T_e)}{2} = \frac{(5+19)}{2} = 12^\circ\text{C}$$

From the properties of ideal gas of air at $T_{avg} = 12^\circ\text{C}$

Density,

$$\rho = 1.246 + \frac{1.225 - 1.246}{15 - 10} \times (12 - 10) = 1.2376\,\text{kg/m}^3$$

Specific heat,

$$C_p = 1006 + \frac{1007 - 1006}{15 - 10} \times (12 - 10) = 1006.4\,\text{J/kg}\cdot\text{k}$$

Step 4 of 9

Thermal conductivity,

$$k = 0.02439 + \frac{0.02476 - 0.02439}{(15 - 10)}(12 - 10) = 0.024538\,\text{W/m}\cdot\text{K}$$

Kinematic viscosity,

$$\nu = 1.426 \times 10^{-5} + \frac{(1.470 - 1.426)10^{-5}}{(15 - 10)}(12 - 10) = 1.4436 \times 10^{-5}\,\text{m}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = 0.7336 + \frac{0.7323 - 0.7336}{15 - 10}(12 - 10) = 0.73308$$

Step 5 of 9

Calculating Reynolds number,

$$\text{Re} = \frac{V_{avg} D}{\nu} = \frac{2.5 \times 0.12}{1.4436 \times 10^{-5}} = 20781.38$$

Which is greater than 10,000.

Let us check the entry lengths

$$L_h \approx L_t \approx 10\text{D} \approx 10 \times 0.12 = 1.2\,\text{m}$$

This is shorter than the given length of $L = 20\,\text{m}$

Therefore flow is a fully developed turbulent flow.

Step 6 of 9

By Dietus – Boelter equation,

Nusselt number,

$$Nu = 0.023(\text{Re})^{0.8}(\text{Pr})^{0.4} = 0.023(20781.38)^{0.8}(0.73308)^{0.4} = 57.8$$

Also we know, $Nu = \frac{hD}{k}$

Therefore, heat transfer coefficient,

$$h = \frac{k \cdot Nu}{D} = \frac{0.024538 \times 57.8}{0.12} = 11.819\,\text{W/m}^2{}^\circ\text{C}$$

Step 7 of 9

Mass flow rate of the air, $\dot{m} = \rho A_c V_{avg}$

Cross sectional area of pipe, $A_c = \frac{\pi D^2}{4}$

$$\dot{m} = p \times \frac{\pi D^2}{4} \times V_{avg} = \frac{1.2376 \times \pi \times (0.12)^2 \times 2.5}{4} = 0.03499\,\text{kg/s}$$

Step 8 of 9

The rate of heat transfer for air,

$$\dot{Q} = \dot{m} C_p (T_e - T_i) = 0.03499 \times 1006.4 \times (19 - 5) = 493.03\,\text{W}$$

Therefore heat transfer surface area,

$$A_s = \pi DL = \pi \times 0.12 \times 20 = 7.54\,\text{m}^2$$

We know that,

Heat lost by cold air is equal to the heat gain by the surface of the pipe.

$$Q = hA\Delta T_m$$

$$493.03 = 11.819 \times 7.54 \times \Delta T_m$$

$$\Delta T_m = 5.53^\circ\text{C}$$

Step 9 of 9

Then, log mean temperature difference,

$$\Delta T_m = \frac{T_{exit} - T_{inlet}}{\ln\left(\frac{T_s - T_{exit}}{T_s - T_{inlet}}\right)}$$

$$5.53 = \frac{19 - 5}{\ln\left(\frac{T_s - 19}{T_s - 5}\right)}$$

$$T_s = 3.99^\circ\text{C}$$

Therefore, the surface temperature of the pipe, $\boxed{T_s = 3.99^\circ\text{C}}$

Question:

Oil at 15°C is to be heated by saturated steam at 1 atm in a double-pipe heat exchanger to a temperature of 25°C. The inner and outer diameters of the annular space are 3 cm and 5 cm, respectively, and oil enters at with a mean velocity of 1.2 m/s. The inner tube may be assumed to be isothermal at 100°C, and the outer tube is well insulated. Assuming fully developed (low for oil, determine the tube length required to heat the oil to the indicated temperature. In reality, will you need a shorter or longer lube? Explain.

Step 1 of 7

Calculate the average temperature of the oil using the following equation

$$T_{avg} = (T_i + T_e)/2$$

Here, T_i is the inlet temperature of the oil and T_e is the exit temperature of the oil.

Substitute $15°C$ for T_i and $25°C$ for T_e

$$T_{avg} = \frac{(15+25)}{2}$$
$$= 20° C$$

Obtain the properties of engine oil at temperature $T_{avg} = 20°C$ from the table: "Properties of liquids – Engine oil (unused)".

Density, $\rho = 888.1\ kg/m^3$

Specific heat, $C_p = 1881\ J/kg \cdot K$

Thermal conductivity, $k = 0.1450\ w/m \cdot K$

Kinematic viscosity, $\nu = 9.429 \times 10^{-4}\ m^2/s$

Prandtl number, $Pr = 10,863$

Step 2 of 7

Calculate the hydraulic diameter using the following equation

$$D_h = \frac{4A_c}{P}$$
$$= \frac{4 \times \frac{\pi}{4}\left(D_o^2 - D_i^2\right)}{\pi(D_o + D_i)}$$
$$= D_o - D_i$$

Here, A_c is the cross-sectional area of the annulus, P is the perimeter of the annulus and D_o, D_i are the outer and inner diameters of the annular space respectively.

Substitute 0.05 m for D_o and 0.03 for D_i

$$D_h = 0.05 - 0.03$$
$$= 0.02\ m$$

Step 3 of 7

Calculate the Reynolds number using the following equation

$$Re = \frac{V_{avg} \times D_h}{\nu}$$

Here, V_{avg} is the mean velocity.

Substitute $1.2\ m/s$ for V_{avg} and

$$Re = \frac{1.2 \times 0.02}{9.429 \times 10^{-4}}$$
$$= 25.45$$

Therefore, the flow of oil is laminar.

Calculate the Hydrodynamic entry length using the following equation

$$L_h = 0.05(Re)(D_h)$$
$$= (0.05)(25.45)(0.02)$$
$$= 0.025\ m$$

Calculate the thermal boundary entry length using the following relation

$$L_t = L_h \cdot Pr$$
$$= 0.025 \times 10,863$$
$$= 271.5\ m$$

Step 4 of 7

The oil is flowing through the inner pipe.

Calculate the cross – sectional area of oil flow

$$A_c = \frac{\pi D_i^2}{4}$$
$$= \frac{\pi (0.03)^2}{4}$$
$$= 7.06 \times 10^{-4}\ m^2$$

Calculate the mass flow rate using the following equation

$$\dot{m} = \rho A_c V_{avg}$$
$$= 888.1 \times 7.06 \times 10^{-4} \times 1.2$$
$$= 0.7533\ kg/s$$

Step 5 of 7

Calculate the heat transferred rate using the following equation

$$\dot{Q} = \dot{m} C_p (T_e - T_i)$$
$$= 0.7533 \times 1881(25 - 15)$$

Calculate the heat transferred rate using the following equation

$$\begin{aligned}\dot{Q} &= \dot{m}C_p(T_e - T_i)\\ &= 0.7533\times 1881(25-15)\\ &= 14169.573\,\text{W}\end{aligned}$$

Calculate the heat transfer surface area using the following equation

$$\begin{aligned}A_s &= \pi D_i L\\ &= \pi\times 0.03\times L\\ &= 0.0942\,\text{L}\end{aligned}$$

Calculate the logarithmic mean temperature using the following equation

$$\begin{aligned}\Delta T_{lm} &= \frac{T_i - T_e}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)}\\ &= \frac{15-25}{\ln\left(\frac{100-25}{100-15}\right)}\\ &= 79.9^\circ\text{C}\end{aligned}$$

Calculate the ratio $\frac{D_i}{D_o}$

$$\begin{aligned}\frac{D_i}{D_o} &= \frac{0.03}{0.05}\\ &= 0.6\end{aligned}$$

Step 6 of 7

Obtain the value of Nusselt number from the table, "Nusselt number for fully developed laminar flow in an annulus with one surface isothermal and the other adiabatic - Nu_i " at $\frac{D_i}{D_o} = 0.6$.

Nusselt number, $\text{Nu} = 5.564$

Consider the equation for Nusselt number.

$$\text{Nu} = \frac{hD_h}{k}$$

$$h = \frac{\text{Nu}\cdot k}{D_h}$$

Here, h is the convective heat transfer coefficient.

$$\begin{aligned}h &= \frac{5.564\times 0.1450}{0.02}\\ &= 40.339\,\text{W/m}^2\,^\circ\text{C}\end{aligned}$$

Step 7 of 7

Consider the equation for heat transfer

$$Q = hA_s DT_{lm}$$

$$14169.573 = 40.339\times 0.0942\,L\times(79.9)$$

$$L = 46.66\,\text{m}$$

Here, the length of the pipe (46.66 m) is less than the thermal boundary entry length (271.5). Hence, the length of the pipe should be longer for the flow to be fully developed.

Question: 19.140P

100 kg/s of a crude oil is heated from 20°C to 40°C through the tube side of a multitube heat exchanger. The crude oil flow is divided evenly among all 100 tubes in the tube bundle. The ID of each tube is 10 mm, and the inside tube-wall temperature is maintained at 100°C. Average properties of the crude oil are ρ =950 kg/m3, cp =1.9 kJ/kg·K, k= 0.25 W/m·K, μ =12 mPa·s, and μw= 4 mPa·s. Estimate the rate of heat transfer and the tube length.

Step 1 of 7

Given data:

Internal Diameter of tube, $D = 10\,\text{mm}$

Inlet temperature of crude oil, $T_i = 20°\text{C}$

Outlet temperature of crude oil, $T_e = 40°\text{C}$

Wall temperature of tube, $T_s = 100°\text{C}$

Mass flow rate, $\dot{m} = 100\,\text{kg/s}$ i.e. for $N = 100$

Average properties of crude oil:

Density, $\rho = 950\,\text{kg/m}^3$

Specific heat, $C_p = 1.9\,\text{kJ/kg°k}$

Thermal conductivity, $k = 0.25\,\text{w/m} - \text{k}$

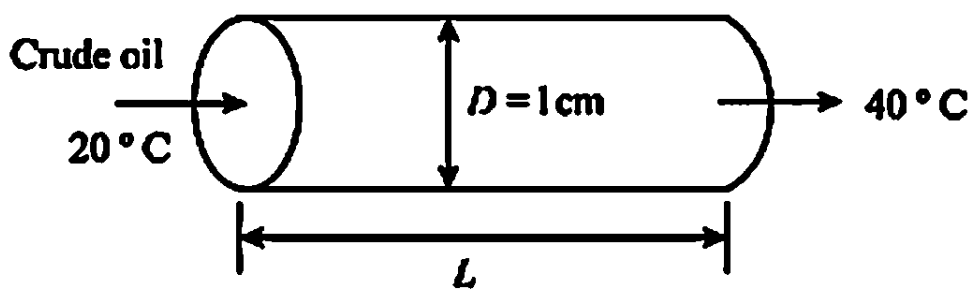
Dynamic viscosity, $\mu = 12\,\text{mPa} \cdot \text{s}$

Dynamic viscosity water $\mu_w = 4\,\text{mP}_a \cdot \text{s}$

$\mu_w = 4 \times 10^{-3}\,\text{kg/m} - \text{s}$

Step 2 of 7

Schematic diagram of pipe flow:



Step 3 of 7

As we know the rate of heat transfer can be expressed as:

$$\dot{Q} = \dot{m}C_p(T_e - T_i)$$
$$= 100 \times 1900(40 - 20)$$
$$= 38 \times 10^5\,\text{W}$$

Therefore, rate of heat transfer, $\boxed{\dot{Q} = 3.8 \times 10^6\,\text{W}}$

Step 4 of 7

As mass flow rate for tube is

$$\dot{m} = \frac{m}{N}$$
$$= \frac{100}{100}$$
$$= 1\,\text{kg/m} - \text{s}$$

Therefore,

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu}$$
$$= \frac{4 \times 1}{\pi \times 0.01 \times 0.012}$$
$$= 10610.33$$

Since it is more than 10,000 Therefore it is a turbulent flow.

Step 5 of 7

By the Dittus – Boelter equation

$$Nu = 0.23\text{Re}^{0.8}\text{Pr}^{0.4}$$

As P_r is not known calculating Prandtl number from

$$\text{Pr} = \frac{\mu C_p}{k}$$
$$= \frac{0.012 \times 1900}{0.25}$$
$$= 91.2$$

Therefore,

$$Nu = \frac{hD}{k} = 0.023(\text{Re})^{0.8}(\text{Pr})^{0.4}$$
$$= 0.023(10610.33)^{0.8}(91.2)^{0.4}$$
$$= 232.44$$

Step 6 of 7

Also, nusselt number

$$Nu = \frac{hD}{k}$$

Heat transfer coefficient,

$$h = \frac{k \cdot Nu}{D}$$
$$= \frac{0.25 \times 232.44}{0.01}$$
$$= 5811\,\text{W/m}^2 \cdot °\text{C}$$

Step 7 of 7

By the Newton's law of cooling

$$Q = hA_s(T_s - T_b)$$

Where

$$T_b = \frac{T_i + T_e}{2}$$
$$= \frac{40 + 20}{2}$$
$$= 30°\text{C}$$

Therefore putting the values in equation above, we get

$$3.8 \times 10^6 = 5811 \times \pi \times 0.01 \times L \times (100 - 30°)$$

$$\boxed{L = 297.36\,\text{m}}$$

This length L is the total length of tube bunch as the total calculated heat transfer was for all 100 tubes.

Therefore, length of each tube, $L = 2.97\,\text{m}$

Question: 19.141P

A fluid (ρ =1000 kg/m3. μ = 1.4 ×10−3 kg/m·s, cp = 4.2 kJ/kg·K, and k =0.58 W/m·K) flows with an average velocity of 0.3 m/s through a 14-m long tube with inside diameter of 0.01 m. Heat is uniformly added to the entire tube at the rate of 1500 W/m2. Determine (*a*) the value of convection heat transfer coefficient at the exit, (*b*) the value of $Ts - Tm$, and (*c*) the value of $Te - Ti$.

Step 1 of 7

Given data:

Average velocity of fluid flow, $V_{avg} = 0.3\,\text{m/s}$

Length of tube, $L = 14\,\text{m}$

Inside diameter of tube, $D_i = 0.01\,\text{m}$

Heat flux rate, $\dot{q}_s = 1500\ \text{W/m}^2$

Fluid properties:

Density, $\rho = 1000\ \text{kg/m}^3$

Dynamic viscosity, $\mu = 1.4\times10^{-3}\ \text{kg/m}\cdot\text{s}$

Specific heat, $c_p = 4.2\ \text{kJ/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.58\ \text{W/m}\cdot\text{K}$

Step 2 of 7

(a) The Reynolds number for the flow,

$$\text{Re} = \frac{\rho V_{avg} D}{\mu}$$

$$= \frac{1000\times0.3\times0.01}{1.4\times10^{-3}}$$

$$\text{Re} = 2142.8$$

Since the Reynolds number is less than 2300 the flow is said to be laminar flow.

Therefore it's a laminar flow.

The prandtl number ,

$$\text{Pr} = \frac{\mu c_p}{k}$$

$$= \frac{1.4\times10^{-3}\times4200}{0.58}$$

$$\text{Pr} = 10.14$$

Step 3 of 7

The hydrodynamic entry length,

$$L_h = 0.05\,\text{Re}\,D$$

$$= 0.05\times2142.8\times0.01$$

$$L_h = 1.071\,\text{m}$$

The thermal entry length,

$$L_t = L_h \times \text{Pr}$$

$$= 1.071\times10.14$$

$$L_t = 10.85$$

Since both the thermal entry lengths L_h and L_t are less than given length $L = 14\,\text{m}$

The flow is said to be fully developed laminar flow.

Step 4 of 7

From the graph named Local Nusselt numbers in the entry and fully developed regions for laminar flow in a circular tube for hydro dynamically developed and thermally developing flow.

For constant surface heat flux,

Nusselt number, $Nu = 4.36$

The heat transfer coefficient is determined as

$$Nu = \frac{hD}{k}$$

$$4.36 = \frac{h\times0.01}{0.58}$$

$$h = 252.88\ \text{W/m}^2\cdot\text{K}$$

Therefore, the value of convective heat transfer coefficient at the exit,

$$\boxed{h = 252.88\ \text{W/m}^2\cdot\text{K}}$$

Step 5 of 7

(b) From the Newton's law of cooling the heat flux ,

$$\dot{q} = h(T_s - T_m)$$

$$1500 = 252.88(T_s - T_m)$$

$$(T_s - T_m) = 5.93^\circ\text{C}$$

Therefore, the value of the temperature difference, $\boxed{T_s - T_m = 5.93^\circ\text{C}}$

Step 6 of 7

(c) surface area of the heat flow,

$$A_s = \pi DL$$

$$= \pi\times0.01\times14$$

$$A_s = 0.4398\ \text{m}^2$$

The mass flow rate of the fluid is determined as

$$\dot{m} = \rho V_{avg} A_c$$

$$= \rho\times V_{avg}\times\frac{\pi D^2}{4}$$

$$= \frac{1000\times0.3\times\pi\times(0.01)^2}{4}$$

$$\dot{m} = 0.02356\ \text{kg/s}$$

Step 7 of 7

The heat transfer rate can be expressed by

$$\dot{Q} = \dot{m}c_p(T_e - T_i)$$

$$\dot{q}\times A_s = 0.02356\times4.2\times10^3\times(T_e - T_i)$$

$$1500\times0.4398 = 0.02356\times4.2\times(T_e - T_i)$$

$$(T_e - T_i) = 6.67^\circ\text{C}$$

Therefore, the value of the temperature difference, $\boxed{(T_e - T_i) = 6.67^\circ\text{C}}$

Question: 19.142P

Air (1 atm) entered into a 5-mm diameter circular tube at an average velocity of 5 m/s. The tube wall is maintained at a constant surface temperature. Determine the convection heat transfer coefficient for (*a*) a 10-cm long tube and (*b*) a 50-cm long tube. Evaluate the air properties at 50°C.

Step 1 of 5

Given data:

Diameter of circular tube, $D = 5\,\text{mm}$

Average velocity of air, $V_{avg} = 5\ \text{m/s}$

Pressure of the air, $P = 1\,\text{atm}$

Temperature of the air, $T = 50°\text{C}$

Step 2 of 5

The properties of air table, at $P = 1\,\text{atm}$ and $T = 50°\text{C}$

Density, $\rho = 1.092\,\text{kg/m}^3$

Specific heat, $c_p = 1007\ \text{J/kg}\cdot\text{K}$

Thermal conductivity, $k = 0.02735\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

The Reynolds number for the flow is determined as

$$\text{Re} = \frac{V_{avg}D}{\nu}$$

$$= \frac{5\times5\times10^{-3}}{\left(1.798\times10^{-5}\right)}$$

$$\text{Re} = 1390.43$$

Since the Reynolds number is less than 2300, the flow is said to laminar flow.

Step 3 of 5

(a) For a tube length, $L = 10\ \text{cm}$

The thermal entry lengths are determined as

$$L_h = 0.05\,\text{Re}\times D$$

$$= 0.05\times1390.43\times5\times10^{-3}$$

$$L_h = 0.348\,\text{m}$$

$$L_t = L_h\times\text{Pr}$$

$$= 0.348\times0.7228$$

$$L_t = 0.2515\ \text{m}$$

Since the thermal entry lengths L_h and L_t are greater than the given length $L = 0.1\,\text{m}$,

The flow is fully developed laminar flow.

Step 4 of 5

The Nusselt number for the laminar flow in a circular tube,

$$Nu = 1.86\left(\frac{\text{Re}\cdot\text{Pr}\cdot D}{L}\right)\left(\frac{\mu_b}{\mu_s}\right)^{0.14}$$

From the Dynamic viscosity tables at $T = 50°\text{C}$

$\mu_b = 1.963\times10^{-5}\ \text{kg/m}\cdot\text{s}$

At surface temperature $T_s = 160°\text{C}$

$\mu_s = 2.420\times10^{-5}\ \text{kg/m}\cdot\text{s}$

The Nusselt number is determined as

$$Nu = 1.86\left(\frac{1390.43\times0.7228\times5\times10^{-3}}{0.1}\right)^{1/3}\left(\frac{1.963\times10^{-5}}{2.420\times10^{-5}}\right)^{0.14}$$

$$= 6.6655$$

The convective heat transfer coefficient is determined as

$$h = \frac{k\times Nu}{D}$$

$$= \frac{0.02735\times6.6655}{5\times10^{-3}}$$

$$= 36.46\ \text{W/m}^2\cdot\text{K}$$

Therefore, the convective heat transfer coefficient of air, $\boxed{h = 36.46\ \text{W/m}^2\cdot\text{K}}$

Step 5 of 5

(b) For a tube length, $L = 50\ \text{cm}$

Since both the thermal entry lengths L_h and L_t are less than given length $L = 0.5\text{m}$

The flow is said to be fully developed laminar flow.

From the graph named Local Nusselt numbers in the entry and fully developed regions for laminar flow in a circular tube for hydro dynamically developed and thermally developing flow.

For constant surface temperature,

$Nu = 3.66$

The convective heat transfer coefficient is determined as

$$h = \frac{k\times Nu}{D}$$

$$= \frac{0.02735\times3.66}{5\times10^{-3}}$$

$$h = 20.02\ \text{W/m}^2\cdot\text{K}$$

Therefore, the convective heat transfer coefficient of air is $\boxed{h = 20.02\ \text{W/m}^2\cdot\text{K}}$

CHAPTER 21

SOLUTIONS

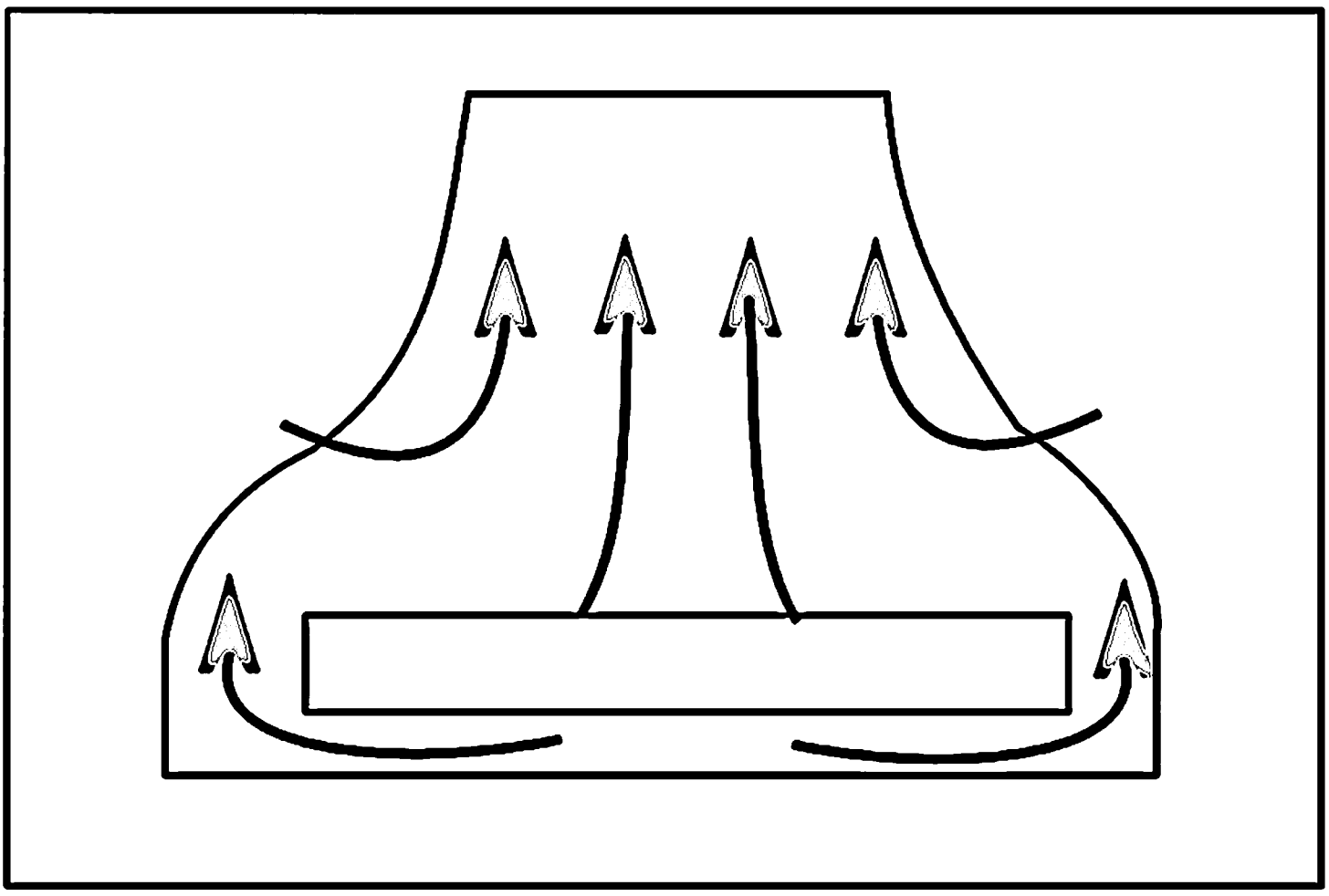
Question: 20.001P

What is natural convection? How does it differ from forced convection? What force causes natural convection currents?

Step 1 of 2

When a temperature difference produces density variations which results in mass movement of fluids, then the process is called free or natural convection.

The following sketch shows the free convection currents around a horizontal plate:

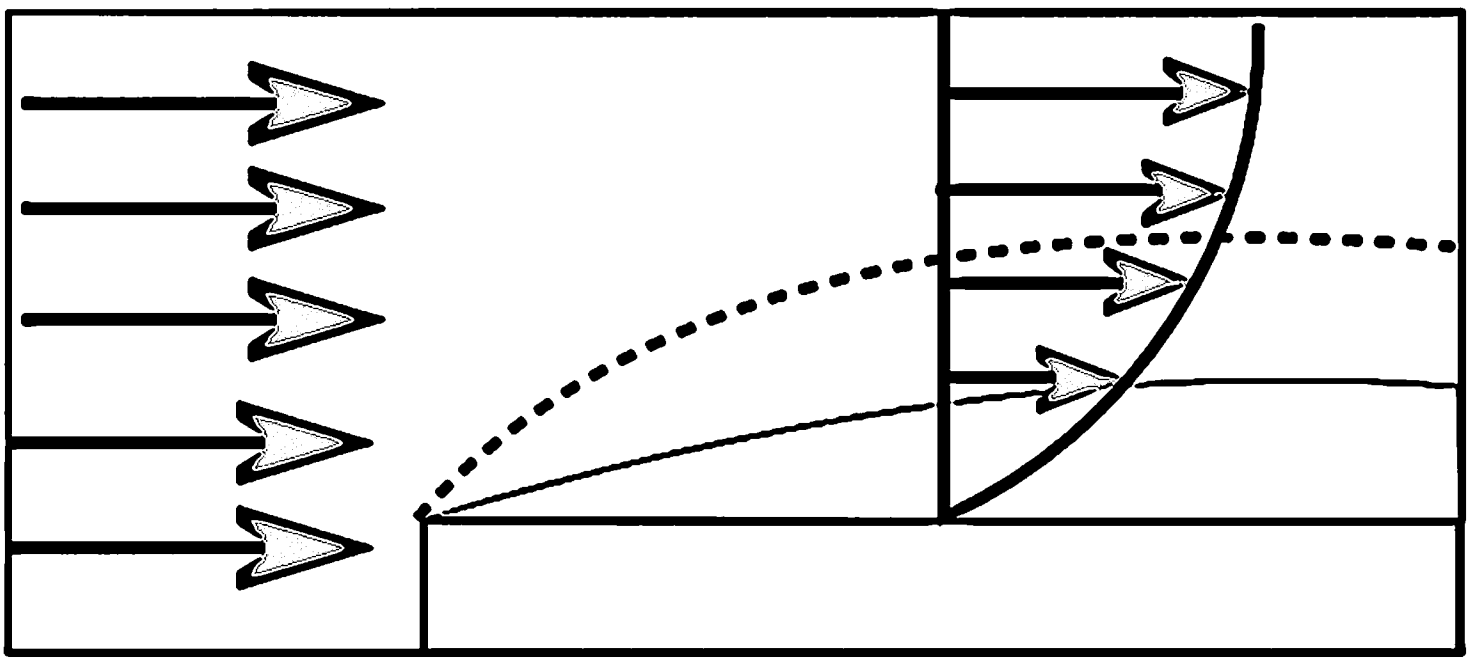


The natural convection occurs, when a free flowing air air sorrounded by a hot plate passes over it and rises from the plate .

Step 2 of 2

When the mass motion of the fluid is caused by an external device like a pump, compressor, blower or fan, the process is called forced convection.

The following sketch shows the forced convection currents:



The figure represents the relation between flow velocity, direction, and temperature for a hot, horizontal surface under forced convection.

Forced convection is generally more efficient than natural convection due to the rapid velocity of the currents. In forced convection, buoyancy has small effect on the direction of flow. Thus buoyancy force causes natural convection currents.

Question: 20.002P

In which mode of heat transfer is the convection heat transfer coefficient usually higher, natural convection or forced convection? Why?

Step 1 of 1

Velocity magnitude of forced convection is greater than that of the natural convection so that the heat transfer coefficient of forced convection is higher.

Question: 20.003P

What is buoyancy force? Compare the relative magnitudes of the buoyancy force acting on a body immersed in these mediums; (*a*) air, (*b*) water, (*c*) mercury, and (*d*) an evacuated chamber.

Step 1 of 1

The upward force exerted by a fluid on a body completely or partially immersed in it is called the buoyancy. The buoyancy force is proportional to the density of the medium. Therefore, the buoyancy force is the largest in mercury, followed by in water, air, and the evacuated chamber. Note that in an evacuated chamber there will be no buoyancy force because of absence of any fluid in the medium.

Question: 20.004P

Consider a hot boiled egg in a spacecraft that is filled with air at atmospheric pressure and temperature at all times. Will the egg cool faster or slower when the spacecraft is in space instead of on the ground? Explain.

Step 1 of 1

Natural convection is mainly based on buoyancy force which is due to the gravity effect. In case of boiled egg in the aircraft which is at space, there is no gravity effect acting on the egg, which means there is no buoyancy effect on egg. So the egg takes more time to cool down in the space. Hence egg cools faster when the spacecraft is on ground.

Question: 20.005P

When will the hull of a ship sink in water deeper: when the ship is sailing in fresh water or in seawater? Why?

Step 1 of 1

The buoyancy force is proportional to the density of the medium, and thus is larger in sea water than it is in fresh water. Therefore, the hull of a ship will sink deeper in fresh water because of the smaller buoyancy force acting upwards.

Question: 20.006P

Consider two fluids, one with a large coefficient of volume expansion and the other with a small one. In what fluid will a hot surface initiate stronger natural convection currents? Why? Assume the viscosity of the fluids to be the same.

Step 1 of 1

The greater the volume expansion coefficient, greater will be the change in density with temperature. And with greater change in density, the buoyancy force will be greater, and thus the natural convection currents are greater. So for the fluid with greater volume expansion coefficient, the natural convection currents are greater.

Question: 20.007P

Physically, what does the Grashof number represent? How does the Grashof number differ from the Reynolds number?

Step 1 of 1

Physically, Grashof number represents the ratio of the buoyancy force to the viscous force acting on a fluid. In Reynolds number, if the inertia force is replaced by the buoyancy forces then it becomes Grashof number.

Question: 20.008P

Show that the volume expansion coefficient of an ideal gas is $\beta = 1/T$, where T is the absolute temperature.

Step 1 of 1

The volume expansion coefficient is $\beta = \frac{-1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p$

But for an ideal gas, $P = \rho RT$ or

$$\rho = \frac{P}{RT}$$

Then the volume expansion coefficient becomes

$$\beta = -\frac{1}{\rho}\left(\frac{\partial}{\partial T}(P/RT)\right)_p$$

$$\beta = -\frac{1}{\rho}\left(\frac{P}{R} \times \frac{\partial}{\partial T}\left(\frac{1}{T}\right)\right)_p$$

$$\beta = \frac{-1}{\rho}\left(\frac{P}{R} \times \frac{-1}{T^2}\right)$$

$$\beta = \frac{-1}{\rho}\left(\frac{-P}{RT^2}\right)$$

$$\beta = \frac{1}{\rho T}\left(\frac{P}{RT}\right)$$

$$\beta = \frac{1}{\rho T}(\rho)$$

Hence volume expansion coefficient of ideal gas is $\boxed{\beta = \frac{1}{T}}$

Question: 20.009P

Using its definition and the values listed in Table A -15.determine the volume expression coefficient of saturated liquid water at 70°C. Compare the result with the value tabulated in Table A-15.

Step 1 of 3

Given data:

Temperature of water, $T = 70°C$

We assume that density is dependent upon temperature only.

Consider the density of the water to be ρ

A property that represents the variations of density of a fluid with temperature at constant pressure, is volume expansion co-efficient (β) is given as;

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p$$

$$= -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p$$

$$\beta = -\frac{1}{\rho}\left(\frac{\rho_1 - \rho_2}{T_1 - T_2}\right) \qquad \ldots\ldots (1)$$

Step 2 of 3

Let us take variation of temperature between 65 °C to 75 °C and respective densities values from the table Properties of saturated water

At temperature, $T_1 = 65°C$

Density, $\rho_1 = 980.4 kg/m^3$

At temperature, $T_2 = 75°C$

Density, $\rho_2 = 974.7 kg/m^3$

At temperature, $T = 70°C$

Average Density, $\rho = 977.5 kg/m^3$

Step 3 of 3

Now substituting the above values in the equation (1),

$$\beta = -\frac{1}{\rho}\left(\frac{\rho_1 - \rho_2}{T_1 - T_2}\right)$$

$$= -\frac{1}{977.5}\left(\frac{980.4 - 974.7}{65 - 75}\right)$$

$$= 5.83 \times 10^{-4} K^{-1}$$

Therefore, the volume expansion coefficient is $\boxed{\beta = 5.83 \times 10^{-4} K^{-1}}$

From the Properties of saturated water table, at $T = 70°C$

Volume expansion coefficient value, $\beta = 5.78 \times 10^{-4} K^{-1}$

By comparing both the values, approximately the calculated value is 1% higher than tabular value.

Question: 20.010P

The density of liquid water can be correlated as $\rho(T) = 1000 - 0.0736T - 0.00355T2$ where ρ and T are in kg/m and °C, respectively. Determine the volume expansion coefficient at 70°C. Compare the result with the value tabulated in Table A–15.

Step 1 of 3

Given data:

Density of the liquid water relation with the temperature,

$\rho(T) = 1000 - 0.0736T - 0.00355T^2$

Density units, ρ kg/m^3

Temperature units, T°C

Volume expansion co-efficient, β

Step 2 of 3

Volume expansion co-efficient relationship is written as

$$\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p \qquad \ldots\ldots (1)$$

By substituting the density relation in the equation (1) and differentiating with respective to the T;

$$\beta = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p$$

$$= -\frac{1}{\rho}\left[\frac{\partial}{\partial T}\left(1000 - 0.0736T - 0.00355T^2\right)\right]_p$$

$$= \frac{1}{\rho}\left(-0.0736 - 0.0071T\right)_p$$

$$= \frac{\left(-0.0736 - 0.0071T\right)_p}{1000 - 0.0736T - 0.00355T^2}$$

Step 3 of 3

Now at $T = 70°C$, the volume expansion co-efficient β is given by

$$\beta = -\frac{\left(-0.0736 - 0.0071T\right)}{\left(1000 - 0.0736T - 0.00355T^2\right)}$$

$$\beta_{T=70°C} = -\frac{\left(-0.0736 - 0.0071\times 70\right)}{\left[1000 - 0.0736\times 70 - 0.00355\times(70)^2\right]}$$

$$= \frac{0.5733}{977.453}$$

$$= 5.86\times 10^{-4}\,\mathrm{K}^{-1}$$

Therefore, the volume expansion co-efficient β is $\boxed{\beta_{T=70°C} = 5.86\times 10^{-4}\,\mathrm{K}^{-1}}$

From the Properties of saturated water table, at $T = 70°C$

Volume expansion coefficient value, $\beta = 5.78\times 10^{-4}\,\mathrm{K}^{-1}$

By comparing both the values, approximately the calculated value is 1% higher than tabular value.

Question: 20.011P

A 10 cm × 10cm plate has a constant surface temperature of 150°C. Determine the Grashof number if the chip is placed in the following fluids: air (1 atm, 30°C), liquid water (30°C), engine oil (10°C). Discuss how the Grashof number affects the natural convection flow.

Step 1 of 7

Given data:

Plate dimensions $= 10\,\text{cm} \times 10\,\text{cm}$

$= 0.1 \times 0.1\ \text{m}^2$

Surface temperature of plate, $T_s = 150°\text{C}$

Step 2 of 7

When chip is placed in air $(1\text{atm}, T_\infty = 30°\text{C})$:

As fluid properties are to be evaluated at film temperature T_f;

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(150+30)}{2}$$
$$= 90°\text{C}$$

Density, $\rho = 0.9718\ \text{kg/m}^3$

Volume expansion coefficient, $\beta = 2.755 \times 10^{-3}\ \text{K}^{-1}$

Dynamic viscosity, $\mu = 2.139 \times 10^{-5}\ \text{kg/m} \cdot s$

Characteristic length from plate dimension, $L_c = 0.1\text{m}$

Gravitational constant, $g = 9.81\ \text{m/s}^2$

Step 3 of 7

Now calculating the kinematic viscosity of the air,

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{2.139 \times 10^{-5}}{0.9718}$$
$$= 2.201 \times 10^{-5}\ \text{m}^2/s$$

Grashof number is given by

$$(\text{Gr}_L)_{air} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$
$$= \frac{9.81 \times 2.755 \times 10^{-3}(150-30)(0.1)^3}{\left(2.201 \times 10^{-5}\right)^2}$$
$$= 6694710$$

Thus, Grashof number when chip is placed in air is $\boxed{(\text{Gr}_L)_{air} = 6694710}$

Step 4 of 7

When chip is placed in Liquid water $(T_\infty = 30°\text{C})$:

By calculating fluid properties at film temperature,

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(150+30)}{2}$$
$$= 90°\text{C}$$

Density, $\rho = 965.3\text{kg/m}^3$

Volume expansion coefficient, $\beta = 0.702 \times 10^{-3}\text{K}^{-1}$

Dynamic viscosity, $\mu = 0.315 \times 10^{-3}\ \text{kg/m} \cdot s$

Step 5 of 7

By calculating the kinematic viscosity of the liquid:

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.315 \times 10^{-3}}{965.3}$$
$$= 3.263 \times 10^{-7}\ \text{m}^2/s$$

So, Grashof number is calculated as

$$(\text{Gr}_L)_w = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$
$$= \frac{9.81 \times 0.702 \times 10^{-3} \times (150-30)(0.1)^3}{\left(3.263 \times 10^{-7}\right)^2}$$
$$= 7761635041$$

Hence, Grashof number when chip is placed in liquid is $\boxed{(\text{Gr}_L)_w = 7761635041}$

Step 6 of 7

When chip is placed in Liquid water Engine oil $(T_\infty = 10°\text{C})$

Properties of engine oil at film temperature T_f

$$T_f = (T_s + T_\infty)/2$$
$$= (150+10)/2$$
$$= 80°\text{C}$$

Density, $\rho = 852\text{kg/m}^3$

Volume expansion coefficient, $\beta = 0.700 \times 10^{-3}\text{K}^{-1}$

Dynamic viscosity, $\mu = 3.232 \times 10^{-2}\ \text{kg/m} \cdot s$

Kinematic viscosity,
kinematic viscosity

$$\nu = \frac{\mu}{\rho}$$
$$= 3.793 \times 10^{-3}\ \text{m}^2/s$$

Step 7 of 7

Grashof number is given by

$$(Gr_L)_O = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$
$$= \frac{9.81 \times 0.7 \times 10^{-3} \times (150-10) \times (0.1)^3}{\left(3.793 \times 10^{-5}\right)^2}$$
$$= 668235$$

Hence, Grashof number when chip is placed in oil is $\boxed{(Gr_L)_O = 668235}$

Question:

A long 2-in.-diameter rod with surface temperature of 200°F is submerged in a bath of fluid. Determine the Grashof and Raylcigh numbers if the fluid is (*a*) liquid water at 40°F, (*b*) liquid ammonia at 40°F, (*c*) engine oil at 50°F, and (*d*) air at 40°F(1 atm).

Step 1 of 14

Given data:-

A long rod is submerged in a bath of fluid

Diameter of rod, $D = 2\text{in.}$

Surface temperature of rod, $T_s = 200\,^\circ\text{F}$

Step 2 of 14

Let us consider the following parameters;

g – Acceleration due to gravity, ft/s^2

β – Co-efficient of volume expansion, R^{-1}

T_∞ – Temperature of fluid, °F

ν – Kinematic viscosity, ft^2/s

L_c – Characteristic length, ft

Step 3 of 14

(a) If the fluid is Liquid water at $T_\infty = 40^\circ\text{F}$

By calculating properties of liquid water at film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{200+40}{2}$$

$$= 120^\circ\text{F}$$

Acceleration due to gravity, $g = 32.2\,\text{ft}/s^2$

Density, $\rho = 61.71\,\text{lbm}/\text{ft}^3$

Dynamic viscosity, $\mu = 3.744\times10^{-4}\,\text{lbm}/\text{ft}\cdot s$

Prandtl number, $\text{Pr} = 3.63$

Co-efficient of volume expansion, $\beta = 0.246\times10^{-3}\,\text{R}^{-1}$

Step 4 of 14

By calculating kinematic viscosity:

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{3.744\times10^{-4}}{61.71}$$

$$= 6.067\times10^{-6}\,\text{ft}^2/s$$

As the rod is placed horizontally, characteristic length becomes:

L_c = Diameter of rod

$D = 0.1667\,\text{ft}$

Step 5 of 14

Now substituting the values in the Grashof and Rayleigh number then

$$\text{Gr} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

$$= \frac{32.2\times0.246\times10^{-3}\times(200-40)\times(0.1667)^3}{\left(6.067\times10^{-6}\right)^2}$$

$$= 1.60\times10^8$$

$$\text{Ra} = \text{Gr}\cdot\text{Pr}$$

$$= 1.60\times10^8\times3.63$$

$$= 5.79\times10^8$$

Thus, Grashof and Rayleigh number are

$\boxed{\text{Gr} = 1.60\times10^8}$

and

$\boxed{\text{Ra} = 5.79\times10^8}$

Step 6 of 14

(b) If the fluid is Liquid Ammonia at $T_\infty = 40\,^\circ\text{F}$

By taking properties of Ammonia at film temperature;

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{200+40}{2}$$

$$= 120\,^\circ\text{F}$$

Density, $\rho = 35.26\,\text{lbm}/\text{ft}^3$

Dynamic viscosity, $\mu = 7.444\times10^{-5}\,\text{lbm}/\text{ft}\cdot s$

Prandtl number, $\text{Pr} = 1.313$

Co-efficient of volume expansion, $\beta = 1.74\times10^{-3}\,\text{R}^{-1}$

Step 7 of 14

By calculating kinematic viscosity:

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{7.444\times10^{-5}}{35.26}$$

$$= 2.11\times10^{-6}\,\text{ft}^2/s$$

As the rod is placed horizontally, characteristic length becomes:

L_c = Diameter of rod

$D = 0.1667\,\text{ft}$

Step 8 of 14

Now substituting the values in the Grashof and Rayleigh number then

$$\text{Gr} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

Step 8 of 14

Now substituting the values in the Grashof and Rayleigh number then

$$Gr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

$$= \frac{32.2 \times 0.174 \times 10^{-4} \times (200-40) \times (0.1667)^3}{(2.11 \times 10^{-6})^2}$$

$$= 9.33 \times 10^7$$

$$Ra = Gr \cdot Pr$$

$$= 1.968 \times 10^8 \times 1.313$$

$$= 1.22 \times 10^9$$

Thus, Grashof and Rayleigh number are

$$\boxed{Gr = 9.33 \times 10^7}$$

and

$$\boxed{Ra = 1.22 \times 10^9}$$

Step 9 of 14

(c) If the fluid is Engine oil at $T_\infty = 50°F$

By taking properties of engine oil at film temperature;

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{200+50}{2}$$

$$= 125°F$$

Density, $\rho = 54.24 \text{ lbm/ft}^3$

Dynamic viscosity, $\mu = 7.617 \times 10^{-2} \text{ lbm/ft} \cdot s$

Prandtl number, $Pr = 1607$

Co-efficient of volume expansion, $\beta = 0.389 \times 10^{-3} R^{-1}$

Step 10 of 14

By calculating kinematic viscosity:

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{7.617 \times 10^{-2}}{54.24}$$

$$= 1.404 \times 10^{-3} \text{ ft}^2/s$$

As the rod is placed horizontally, characteristic length becomes:

L_c = Diameter of rod

$D = 0.1667 \text{ ft}$

Step 11 of 14

Now substituting the values in the Grashof and Rayleigh number then

$$Gr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

$$= \frac{32.2 \times 0.389 \times 10^{-3} \times (200-50) \times (0.1667)^3}{(1.404 \times 10^{-3})^2}$$

$$= 4.42 \times 10^3$$

$$Ra = Gr \cdot Pr$$

$$= 6.619 \times 10^3 \times 1607$$

$$= 7.71 \times 10^6$$

Thus, Grashof and Rayleigh number are

$$\boxed{Gr = 4.42 \times 10^3}$$

and

$$\boxed{Ra = 7.1 \times 10^6}$$

Step 12 of 14

(d) If the fluid is Air at $T_\infty = 40°F$

By taking properties of air at film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{200+40}{2}$$

$$= 120°F$$

Density, $\rho = 0.06843 \text{ lbm/ft}^3$

Dynamic viscosity, $\mu = 1.316 \times 10^{-5} \text{ lbm/ft} \cdot s$

Prandtl number, $Pr = 0.723$

Co-efficient of volume expansion, $\beta = 1.72 \times 10^{-3} R^{-1}$

Step 13 of 14

By calculating kinematic viscosity:

$$\nu = \frac{\mu}{\rho}$$

$$= \frac{1.316 \times 10^{-5}}{0.06843}$$

$$= 1.923 \times 10^{-4} \text{ ft}^2/s$$

As the rod is placed horizontally, characteristic length becomes:

L_c = Diameter of rod

$D = 0.1667 \text{ ft}$

Step 14 of 14

Now substituting the values in the Grashof and Rayleigh number then

$$Gr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

$$= \frac{32.2 \times 1.72 \times 10^{-3} \times (200-40) \times (0.1667)^3}{}$$

Now substituting the values in the Grashof and Rayleigh number then

$$\mathrm{Gr} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

$$= \frac{32.2 \times 1.72 \times 10^{-3} \times (200-40) \times (0.1667)^3}{\left(1.923 \times 10^{-4}\right)^2}$$

$$= 1.11 \times 10^6$$

$$\mathrm{Ra} = \mathrm{Gr} \cdot \mathrm{Pr}$$

$$= 2.13 \times 10^6 \times 0.723$$

$$= 8.03 \times 10^5$$

Thus, Grashof and Rayleigh number are

$$\boxed{\mathrm{Gr} = 1.11 \times 10^6}$$

and

$$\boxed{\mathrm{Ra} = 8.03 \times 10^5}$$

Question: 20.013P

Consider laminar natural convection from a vertical hot-plate. Will the heat flux be higher at the top or at the bottom of the plate? Why?

Step 1 of 1

For a vertical hot plate, the boundary layer is thinner at the bottom than at the top. As the boundary layer thickness represents thermal resistance, the resistance at bottom is less so the plate has higher heat flux at bottom.

Question: 20.014P

Under what conditions can the outer surface of a vertical cylinder be treated as a vertical plate in natural convection calculations?

Step 1 of 1

In natural convection calculations, the vertical cylinder can be treated as a vertical plate if the diameter of the cylinder is large. The curvature effects are negligible if the diameter of the cylinder is large. The dimeter of the cylinder should satisfy the following relation:

$$D \geq \frac{35L}{\mathrm{Gr}^{1/4}}$$

Here, D is the diameter of the cylinder, L is the length of the cylinder, and Gr is the Grashof number.

Question: 20.015P

Will a hot horizontal plate whose back side is insulated cool faster or slower when its hot surface is facing down instead of up?

Step 1 of 1

A hot horizontal plate surface will not cool faster when the un-insulated surface is facing down. The reason is that the warmer air cannot rise and escape easily if the plate is faced down.

Question: 20.016P

An aluminum soda can of 150-mm in length and 60-mm in diameter is placed horizontally inside a refrigerator compartment that maintains a temperature of 4°C. If the surface temperature of the can is 36°C, estimate heat transfer rate from the can. Neglect the heat transfer from the ends of the can.

Step 1 of 6

Given data:-

Aluminum soda can placed horizontally inside a refrigerator.

Diameter of aluminum soda can, $D = 60\,\text{mm}$

Length of soda can, $L = 150\,\text{mm}$

Surface temperature of can, $T_s = 36\,^\circ\text{C}$

Refrigerator compartment temperature, $T_\infty = 4\,^\circ\text{C}$

Step 2 of 6

Heat transfer rate from the soda-can can be calculated by

$$\dot{Q} = hA_s\left(T_s - T_\infty\right)$$

Heat transfer co-efficient is given by the formula:

$$h = \frac{k}{D}N_u$$

Calculating film temperature:

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{36+4}{2}$$

$$= 20^\circ\text{C}$$

Step 3 of 6

By taking properties of the air from table at film temperature T_f:

Thermal conductivity, $k = 0.02514\,\text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.516\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7309$

Expansion co-efficient, $\beta = 0.003413\,\text{K}^{-1}$

Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$

Surface area of the soda-can is calculated as:

$$A_s = \pi DL$$

$$= 3.142\times0.06\times0.15$$

$$= 0.02827\,\text{m}^2$$

Step 4 of 6

Using the above values, the Rayleigh number is calculated as;

$$\text{Ra}_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}.\text{Pr}$$

$$= \frac{9.81\times0.003413(36-4)(0.06)^3}{\left(1.516\times10^{-5}\right)^2}\times(0.7309)$$

$$= \frac{1.69\times10^{-4}}{1.516\times10^{-10}}$$

$$= 735984.4$$

Step 5 of 6

Using the Rayleigh number, by calculating Nu (Nusselt number) for horizontal plate:

$$\text{Nu} = \left[0.6 + \frac{0.387\,\text{Ra}_D^{\frac{1}{6}}}{\left[1+\left(\frac{0.559}{\text{Pr}}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2$$

$$= \left[0.6 + \frac{0.387(735984.4)^{\frac{1}{6}}}{\left[1+(0.559/0.7309)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2$$

$$= [0.6+3.059]^2$$

$$= 13.391$$

Step 6 of 6

By calculating heat transfer co-efficient:

$$h = \frac{k}{D}\text{Nu}$$

$$= \frac{0.02514}{0.06}\times13.391$$

$$= 5.61\,\text{W/m}^2\cdot\text{K}$$

Now the heat transfer rate is calculated as;

$$\dot{Q} = hA_s\left(T_s - T_\infty\right)$$

$$= 5.61\times0.02827(36-4)$$

$$= 5.0757\,\text{W}$$

Hence, the heat transfer rate of the soda-can is $\boxed{\dot{Q} = 5.0757\,\text{W}}$

Question: 20.017P

A can of engine oil with a length of 150 mm and a diameter of 100 mm is placed vertically in the trunk of a car. In a hot summer day, the temperature in the trunk is 43°C. If the surface temperature of the can is 17°C, determine heat transfer rate from the can surface. Neglect the heat transfer from the ends of the can.

Step 1 of 5

Given data:

A can of engine oil is placed vertically.

Length of the can, $L = 150\,\text{mm}$

Diameter of can, $D = 100\,\text{mm}$

Temperature of trunk, $T_\infty = 43°\text{C}$

Surface temperature of can, $T_s = 17°\text{C}$

Step 2 of 5

Calculating film temperature:

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{43+17}{2}$$
$$= 30°\text{C}$$

By taking the air properties at film temperature,

Thermal conductivity, $k = 0.02588\,\text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.608\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

Volume expansion co-efficient, $\beta = 0.0033\,\text{K}^{-1}$

Acceleration due to gravity, $g = 9.81\,\text{m/s}^2$

Step 3 of 5

Surface area of the can is calculated as:

$$A_s = \pi DL$$
$$= \pi\times0.1\times0.15$$
$$= 0.047\,\text{m}^2$$

Using the above values, the Rayleigh number is given by,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr}$$
$$= \frac{9.81\times0.0033(43-17)(0.15)^3}{\left(1.608\times10^{-5}\right)^2}\times0.7282$$
$$= \frac{2.0686\times10^{-3}}{\left(1.608\times10^{-5}\right)^2}$$
$$= 8000343.94$$

Step 4 of 5

From table of empirical correlations for the average Nusselt number for natural convection over surfaces, the Nusselt number for vertical plate is given by

$$\text{Nu} = 0.59\,\text{Ra}_L^{1/4}$$
$$= 0.59\times(8000343.94)^{0.25}$$
$$= 31.378$$

Heat transfer co-efficient is calculated as:

$$h = \frac{k}{D}\text{Nu}$$
$$= \frac{0.02588}{0.15}\times31.378$$
$$= 5.4137\ \text{W/m}^2\cdot\text{K}$$

Step 5 of 5

Heat transfer rate is calculated as:

$$\dot{Q} = hA_s(T_\infty - T_s)$$
$$= 5.4138\times0.047\times(43-17)$$
$$= 6.6156\,\text{W}$$

Hence, the rate of heat transfer is $\boxed{\dot{Q} = 6.633\,\text{W}}$

Question: 20.018P pg 1

Consider a thin 24-cm-long and 20-cm-wide horizontal plate suspended in air at 20°C. The plate is equipped with electric resistance heating elements with a rating of 20 W. Now the heater is turned on and the plate temperature rises. Determine the temperature of the plate when steady operating conditions are reached. The plate has an emissivity of 0.90 and the surrounding surfaces arc at 17°C.

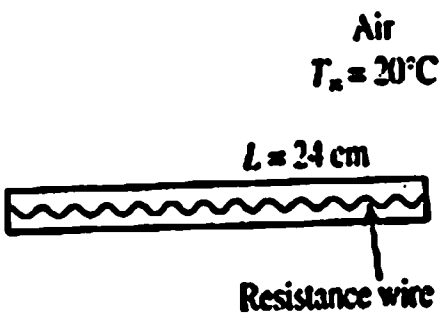


FIGURE P20–18

Step 1 of 10

Assume surface temperature of the plate is $T_s = 40°C$

Find the film temperature of air.

$$T_f = \frac{T_s + T_\infty}{2}$$

Here, T_s is surface temperature of the plate, and T_∞ is surrounding air temperature.

Substitute $40°C$ for T_s and $20°C$ for T_∞

$$T_f = \frac{40+20}{2}$$
$$= 30°C$$

Now take the properties of air at film temperature, $T_f = 30°C$ from table A-22, "properties of air" in the text book.

Thermal conductivity, $k = 0.02588\ \mathrm{W/m°C}$

Prandtl number, $\mathrm{Pr} = 0.7282$

Kinematic viscosity, $v = 1.608\times10^{-5}\ \mathrm{m^2/s}$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{30+273}$$
$$= 3.3\times10^{-3}\ \mathrm{K^{-1}}$$

Step 2 of 10

Find the characteristic length.

$$L_c = \frac{A_s}{p}$$

Here, A_s is surface area and p is perimeter.

Substitute $(0.24\times0.2)\mathrm{m^2}$ for A_s and $2(0.24+0.2)$ for p.

$$L_c = \frac{0.24\times0.20}{2(0.24+0.20)}$$
$$= 0.05455\ \mathrm{m}$$

Step 3 of 10

Find the Rayleigh number by using the following equation:

$$\mathrm{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2}\cdot\mathrm{Pr}$$

Here, g is acceleration due to gravity.

Substitute $9.81\ \mathrm{m/s^2}$ for g, $3.3\times10^{-3}\ \mathrm{K^{-1}}$ for β, $40°C$ for T_s, $20°C$ for T_∞, 0.05455 m for L_c, $1.608\times10^{-5}\ \mathrm{m^2/s}$ for v, and 0.7282 for Pr.

$$\mathrm{Ra} = \frac{9.81\times3.3\times10^{-3}(40-20)(0.05455)^3}{\left(1.608\times10^{-5}\right)^2}\times0.7282$$
$$= 2.959\times10^5$$

Step 4 of 10

Find the heat transfer from the top surface.

Find the Nusselt number by using the following equation:

$$\mathrm{Nu} = 0.54\,\mathrm{Ra}^{0.25}$$

Substitute 2.959×10^5 for Ra.

$$\mathrm{Nu} = 0.54\left(2.959\times10^5\right)^{0.25}$$
$$= 12.5944$$

Find the heat transfer coefficient by using the following equation:

$$\mathrm{Nu} = \frac{h_{top}L_c}{k}$$
$$h_{top} = \frac{k\cdot\mathrm{Nu}}{L_c}$$

Substitute 12.5944 for Nu, 0.05455 m for L_c, and $0.02588\ \mathrm{W/m°C}$ for k.

$$h_{top} = \frac{0.02588\times12.594}{0.05455}$$
$$= 5.9749\ \mathrm{W/m^2.K}$$

$$h_{top} = \frac{0.02588 \times 12.394}{0.05455}$$
$$= 5.9749\ \text{W/m}^2\text{.K}$$

Step 5 of 10

Find the heat transfer from the top surface.

$$\dot{Q}_{top} = h A_s (T_s - T_\infty)$$

Substitute $5.9749\ \text{W/m}^2\text{.K}$ for h, $(0.24 \times 0.2)\text{m}^2$ for A_s, and $20°\text{C}$ for T_∞

$$\dot{Q}_{top} = 5.9749 \times 0.24 \times 0.2 (T_s - 20)$$
$$= 0.28679 (T_s - 20)$$

Step 6 of 10

Calculate the heat transfer from bottom surface:

Find the Nusselt number by using the following relation:

$$\text{Nu} = 0.27\,\text{Ra}^{0.25}$$

Substitute 2.959×10^5 for Ra.

$$\text{Nu} = 0.27 \left(2.959 \times 10^5\right)^{0.25}$$
$$= 6.297$$

Step 7 of 10

Find the heat transfer coefficient at bottom of the surface by using the following equation:

$$\text{Nu} = \frac{h_{bottom} L_c}{k}$$
$$h_{bottom} = \frac{k \cdot \text{Nu}}{L_c}$$

Substitute 6.297 for Nu, 0.05455 m for L_c, and $0.02588\ \text{W/m}°\text{C}$ for k.

$$h_{bottom} = \frac{0.02588 \times 6.297}{0.05455}$$
$$= 2.987\ \text{W/m}^2\text{.K}$$

Step 8 of 10

Find the heat transfer from the bottom surface.

$$\dot{Q}_{bottom} = h_{bottom} A_s (T_s - T_\infty)$$

Substitute $2.987\ \text{W/m}^2\text{.K}$ for h_{bottom}, $(0.24 \times 0.2)\text{m}^2$ for A_s, and $20°\text{C}$ for T_∞

$$\dot{Q}_{bottom} = 2.987 \times 0.24 \times 0.2 (T_s - 20)$$
$$= 0.14337 (T_s - 20)$$

Step 9 of 10

Write the expression for heat transfer due to radiation,

$$\dot{Q}_{rad} = \varepsilon (2A_s) \sigma \left(T_s^4 - T_{amb}^4\right)$$

Here, ε is emissivity of the plate and σ is Stefan Boltzmann's constant.

Here, $2A_s$ indicates radiation heat loss from top and bottom surfaces.

Substitute 0.9 for ε, (0.24×0.2) for A_s, $5.67 \times 10^{-8}\ \text{W/m}^2\text{.K}^4$ for σ, and $(17 + 273)\text{K}$ for T_{amb}

$$Q_{rad} = 0.9 \times 2 \times 0.24 \times 0.2 \times 5.67 \times 10^{-8} \left((T_s + 273)^4 - (17 + 273)^4\right)$$
$$= 0.4898 \times 10^{-8} \left((T_s + 273)^4 - (17 + 273)^4\right)$$

Step 10 of 10

Apply energy balance.

$$\dot{Q} = \dot{Q}_{rad} + \dot{Q}_{top} + \dot{Q}_{bottom}$$

Substitute 20W for $\dot{Q}$, $0.28679(T_s - 20)$ for $\dot{Q}_{top}$, $0.14337(T_s - 20)$ for $\dot{Q}_{bottom}$, and $0.4898 \times 10^{-8} \left((T_s + 273)^4 - (17 + 273)^4\right)$ for $\dot{Q}_{rad}$

$$20 = 0.4898 \times 10^{-9} \left[(T_s + 273)^4 - (290)^4\right] + (T_s - 20)[0.28679 + 0.14337]$$
$$T_s = 39.05°\text{C}$$

The surface temperature is approximately equal to $40°\text{C}$. So, our assumption of surface temperature is correct.

Therefore, the surface temperature of the plate is $\boxed{39.05°\text{C}}$

Question: 20.019P

Flue gases from an incinerator are released to atmosphere using a stack that is 0.6 m in diameter and 10.0 m high. The outer surface of the stack is at 40°C and the surrounding air 1 is at 10°C. Determine the rate of heat transfer from the stack I assuming (*a*) there is no wind and (*b*) the stack is exposed to 20 km/h winds.

Step 1 of 7

Diameter of the stack, $D = 0.6\text{m}$

Height of the stack, $H = 10\text{m}$

Temperature outside of stack, $T_s = 40°\text{C}$

Temperature of surrounding air, $T_{air} = 10°\text{C}$

Step 2 of 7

(a) The heat convected by the stack can be calculated only if the heat transfer coefficient is known. The heat transfer coefficient is determined from the Nusselt number. For determining the Nusselt number, we need the Rayleigh number and the Raleigh number can be calculated if the properties of the surrounding medium are known at film temperature.

The film temperature is given by the formula

$$T_f = \frac{T_s + T_{air}}{2}$$
$$= \frac{40+10}{2}$$
$$= 25°\text{C}$$

The properties of the air at 25°C are

Thermal conductivity, $k = 0.02551\text{W/m°C}$

Prandtl number, $\text{Pr} = 0.7296$

Kinematic viscosity, $\nu = 1.562\times10^{-5}\text{m}^2/\text{s}$

Step 3 of 7

The Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{25+273}$$
$$= 3.356\times10^{-3}\text{K}^{-1}$$

The characteristic length of the plate is

$$L_c = \frac{A_s}{p}$$
$$= \frac{\pi DH}{\pi D}$$
$$= H$$
$$= 10\text{m}$$

Step 4 of 7

The formula for Rayleigh number is

$$\text{Ra} = \frac{g\beta(T_s - T_{air})L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{(9.81)\left(3.356\times10^{-3}\right)(40-10)(10)^3}{\left(1.562\times10^{-5}\right)^2}(0.7296)$$
$$= 2.953\times10^{12}$$

Let us check if the stack can be assumed as a vertical plate. In-order to treat the stack as vertical plate, the condition that should be satisfied is $\frac{35L}{Gr^{0.25}} < 0.6$

$$\frac{35L_c}{Gr^{0.25}} = \frac{35\times L_c}{(\text{Ra}/\text{Pr})^{0.25}}$$
$$= \frac{35\times10}{\left(2.953\times10^{12}/0.7296\right)^{0.25}}$$
$$= 0.246$$

Since the obtained value of ratio is less than 0.6, we can treat the stack as vertical plate.

Step 5 of 7

Now calculate the Nusselt number using the relation for vertical plate

$$\text{Nu} = 0.1\text{Ra}^{1/3}$$
$$= 0.1\left(2.953\times10^{12}\right)^{1/3}$$
$$= 1433.3$$

The heat transfer coefficient is calculated using the formula

$$h = \frac{\text{Nu}\times k}{L_c}$$
$$= \frac{1433.3\times0.02551}{10}$$
$$= 3.656\ \text{W/m}^2\text{K}$$

The heat convected from the plate is given by

$$Q_{conv} = hA_s(T_s - T_{air})$$
$$= 3.656\times(\pi\times0.6\times10)\times(40-10)$$
$$= 2067.6\ \text{W}$$

Thus the heat convected from the stack is $\boxed{Q_{conv} = 2067.6\ \text{W}}$

Step 6 of 7

(b) When the stack is exposed to wind at 20 km/hr (5.55 m/s), the heat transfer from stack is due to forced convection. For forced convection, the heat transfer coefficient is determined from the Nusselt number which in turn is determined depending on the value of Reynolds number.

The formula for Reynolds number is

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{(5.55)(0.6)}{\left(1.562\times10^{-5}\right)}$$
$$= 2.134\times10^5$$

Now calculate the Nusselt number using the relation

$$\text{Nu} = 0.027(\text{Re})^{0.805}(\text{Pr})^{0.333}$$
$$= 0.027\left(2.134\times10^5\right)^{0.805}(0.7296)^{0.333}$$
$$= 474$$

Step 7 of 7

The heat transfer coefficient is calculated using the formula

$$h = \frac{\text{Nu}\times k}{D}$$
$$= \frac{474\times0.02551}{0.6}$$
$$= 20.153\ \text{W/m}^2\text{K}$$

The heat convected from the plate is given by

$$Q_{conv} = hA_s(T_s - T_{air})$$
$$= 20.153\times(\pi\times0.6\times10)\times(40-10)$$
$$= 11396.2\ \text{W}$$

Thus the heat convected from the stack when the wind blows is $\boxed{Q_{conv} = 11396.2\ \text{W}}$

Question: 20.020P

Thermal energy generated by the electrical resistance of a 5-mm-diameter and 4-m-long bare cable is dissipated to the surrounding air at 20°C. The voltage drop and the electric current across the cable in steady operation are measured to be 60 V and 1.5 A, respectively. Disregarding radiation, estimate the surface temperature of the cable.

Step 1 of 5

Thermal energy generated by electric resistance of cable

Diameter of cable $D = 5\text{ mm} = 5\times10^{-3}\text{ m}$

Length of cable $L = 4\text{ m}$

Voltage drop across the cable $dV = 60\text{ V}$

Electric current across the cable $I = 1.5\text{ A}$

Surrounding temperature $T_\infty = 20°\text{C}$

Step 2 of 5

The surface temperature of the cable can be determined by considering the heat convected by the cable. This can be calculated only if the heat transfer coefficient is known. The heat transfer coefficient is determined from the Nusselt number. For determining the Nusselt number, we need the Rayleigh number and the Raleigh number can be calculated if the properties of the surrounding medium are known at film temperature.

Let us assume the surface temperature as 100°C for calculating the film temperature

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{100+20}{2}$$
$$= 60°\text{C}$$

The properties of the air at 60°C are

Thermal conductivity, $k = 0.02808\text{ W/m°C}$

Prandtl number, $\text{Pr} = 0.7202$

Kinematic viscosity, $\nu = 1.896\times10^{-5}\text{ m}^2/\text{s}$

Step 3 of 5

The Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{60+273}$$
$$= 3.003\times10^{-3}\text{ K}^{-1}$$

The characteristic length is

$$L_c = D$$
$$= 5\times10^{-3}\text{ m}$$

Step 4 of 5

The formula for Rayleigh number is

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times\left(3.003\times10^{-3}\right)(100-20)\left(5\times10^{-3}\right)^3}{\left(1.896\times10^{-5}\right)^2}\times0.7202$$
$$= 590.202$$

Nusselt number can be calculated using

$$\text{Nu} = \left(0.6 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right)^2$$
$$= \left(0.6 + \frac{0.387(590.202)^{1/6}}{\left[1+(0.559/0.7202)^{9/16}\right]^{8/27}}\right)^2$$
$$= 2.3456$$

Step 5 of 5

The heat transfer coefficient is calculated using the formula

$$h = \frac{\text{Nu}\times k}{D}$$
$$= \frac{2.3456\times0.02808}{0.005}$$
$$= 13.173\text{ W/m}^2\text{K}$$

The heat convected from the cable is given by

$$Q = hA_s(T_s - T_\infty)$$
$$T_s = \frac{Q}{hA_s} + T_\infty$$
$$= \frac{dV\times I}{hA_s} + T_\infty$$
$$= \frac{60\times1.5}{13.173\times\left(\pi\times5\times10^{-3}\times4\right)} + 20$$
$$= 128.74°\text{C}$$

Thus the surface temperature of the cable is $\boxed{T_s = 128.74°\text{C}}$

Question: 20.021P

A 12-m-long section of a 5-cm-diameter horizontal hot-water pipe passes through a large room whose temperature is 27°C. If the temperature and the emissivity of the outer surface of the pipe are 73°C and 0.8. respectively, determine the rate of heat loss from the pipe by (*a*) natural convection and (*b*) radiation.

Step 1 of 6

Given data:

Heat loss through the pipe is due to the convection and radiation.

Length of the pipe section, $L = 12\,\text{m}$

Diameter of pipe section, $D = 5\,\text{cm}$

Ambient (Room) temperature, $T_\infty = 27\,^\circ\text{C}$

Surface temperature of pipe, $T_s = 73^\circ\text{C}$

Emissivity, $\varepsilon = 0.8$

Step 2 of 6

(a) Heat transfer by natural convection:

Heat transfer coefficient can be calculated after finding the Rayleigh and Nusselt number.

By taking properties from the table at film temperature;

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(73+27)}{2}$$

$$= 50^\circ\text{C}$$

Prandlt number, $\text{Pr} = 0.7228$

Thermal conductivity, $k = 0.02735\,\text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\,\text{m/s}$

Step 3 of 6

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{(50+273)}$$

$$= 3.0959\times10^{-3}\,\text{K}^{-1}$$

Also characteristic length;

$$L_C = D$$

$$= 0.05\,\text{m}$$

Step 4 of 6

Now calculating the Rayleigh,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\cdot\text{Pr}$$

$$= \frac{9.81\times3.0959\times10^{-3}\times(73-27)\times(0.05)^3}{\left(1.798\times10^{-5}\right)^2}\times0.7228$$

$$= 390456.9$$

As pipe is placed horizontally, the Nusselt number is calculated as

$$\text{Nu} = \left[0.6+\frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.6+\frac{0.387(390456.9)^{\frac{1}{6}}}{\left[1+\left(\frac{0.559}{0.7228}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2$$

$$= 11.2257$$

Step 5 of 6

Now, heat transfer co-efficient is calculated as,

$$h = \frac{k}{D}\text{Nu}$$

$$= \frac{0.02735}{0.05}\times11.2257$$

$$= 6.1405\,\text{W/m}^2\cdot\text{K}$$

So, heat transfer by natural convection is calculated as,

$$\dot{Q} = hA(T_s - T_\infty)$$

$$= h\times\pi\times D\times L(T_s - T_\infty)$$

$$= 6.1405\times\pi\times0.05\times12(73-27)$$

$$= 532.43\,\text{W}$$

Hence, heat transfer by natural convection is $\boxed{\dot{Q} = 532.43\,\text{W}}$

Step 6 of 6

(b) Heat transfer by radiation is given by

$$Q_{rad} = EA\sigma\left(T_s^4 - T_\infty^4\right)$$

$$= 0.8\times\pi\times0.05\times12\times5.67\times10^{-8}\left[(346)^4-(300)^4\right]$$

$$= 8.55\times10^{-8}\left(6.2319\times10^9\right)$$

$$= 532.83\,\text{W}$$

Hence, heat transfer by radiation is $\boxed{Q_{rad} = 532.83\,\text{W}}$

Question: 20.022P

An average person generates heat at a rate of 240 Btu/h while resting in a room at 70°F. Assuming one-quarter of this heat is lost from the head and taking the emissivity of the skin to be 0.9, determine the average surface temperature of the head when it is not covered. The head can be approximated as a 12-in-diameter sphere, and the interior surfaces of the room can be assumed to be at the room temperature.

Step 1 of 6

Heat generated by person $\dot{Q} = 240\,\text{Btu/h}$

Room temperature $T_\infty = 70°\text{F}$

Heat lost from head $\dot{Q}_{\text{lost}} = 0.25\dot{Q} = 60\,\text{Btu/h}$

Emissivity of the skin $\varepsilon = 0.9$

Diameter of head $D = 12\text{in} = 1\text{ft}$

Temperature of room surface $T_{\text{amb}} = 70°\text{F}$

Step 2 of 6

Assume that the average surface temperature is $T_S = 85°\text{F}$

Calculate film temperature T_f,

$$T_f = (T_S + T_\infty)/2$$
$$= (85+70)/2$$
$$= 77.5°\text{F}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{77.5+460}$$
$$= 1.86\times10^{-3}\,\text{R}^{-1}$$

Step 3 of 6

Now take the properties of Air at 1atm and film temperature T_f,

Thermal conductivity,

$$k = 0.01457 + \frac{0.01481-0.01457}{(80-70)}(77.5-70)$$
$$= 0.01475\,\text{Btu/h}\cdot\text{ft}\cdot°\text{F}$$

Prandtl number,

$$\text{Pr} = 0.7306 + \frac{0.7290-0.7306}{(80-70)}(77.5-70)$$
$$= 0.7294$$

Kinematic viscosity,

$$\nu = 1.643\times10^{-4} + \frac{(1.697-1.643)10^{-4}}{(80-70)}(77.5-70)$$
$$= 1.6835\times10^{-4}\,\text{ft}^2/\text{s}$$

Step 4 of 6

Characteristic length,

$$L_c = D$$
$$= 1\text{ft}$$

Calculate the Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_S-T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{32.2\times1.86\times10^{-3}(85-70)(1)^3}{\left(1.6835\times10^{-4}\right)^2}\times(0.7294)$$
$$= 2.312\times10^7$$

Step 5 of 6

Nusselt number for sphere,

$$\text{Nu} = 2 + \frac{0.589\text{Ra}^{0.25}}{\left[1+(0.69/\text{Pr})^{9/16}\right]^{4/9}}$$
$$= 2 + \frac{0.589\left(2.312\times10^7\right)^{0.25}}{\left[1+(0.69/0.7294)^{9/16}\right]^{4/9}}$$
$$= 33.609$$

Neat transfer coefficient,

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$= \frac{0.01475}{1}\times33.609$$
$$= 0.496\,\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

Step 6 of 6

Surface Area,

$$A_S = \pi D^2$$
$$= \pi(1)^2$$
$$= 3.142\,\text{ft}^2$$

The total heat loss is due to both convection and radiation.

As, we know that the heat transfer rate by convection and radiation can be expressed as

$$\dot{Q}_{\text{lost}} = Q_{\text{conv}} + Q_{\text{rad}}$$
$$60 = hA_S(T_S-T_\infty) + \varepsilon A_S\sigma\left(T_S^4 - T_{\text{amb}}^4\right)$$
$$60 = 0.496(3.142)(T_S-70) + 0.9(3.142)\left(0.1714\times10^{-8}\right)\left[(T_S+460)^4-(70+460)^4\right]$$
$$60 = (1.558)(T_S-70) + \left(4.847\times10^{-9}\right)\left[(T_S+460)^4-(530)^4\right]$$

The temperature from the above equation is $T_S = 85°\text{F}$

Therefore the average surface temperature of the head when it is not covered is

$$\boxed{T_S = 83.18°\text{F}}$$

Question: 20.023P

An incandescent lightbulb is an inexpensive but highly inefficient device that converts electrical energy into light. It converts about 5 percent of the electrical energy it consumes into light while converting the remaining 95 percent into heat. The glass bulb of the lamp heats up very quickly as a result of absorbing all that heat and dissipating it to the surroundings by

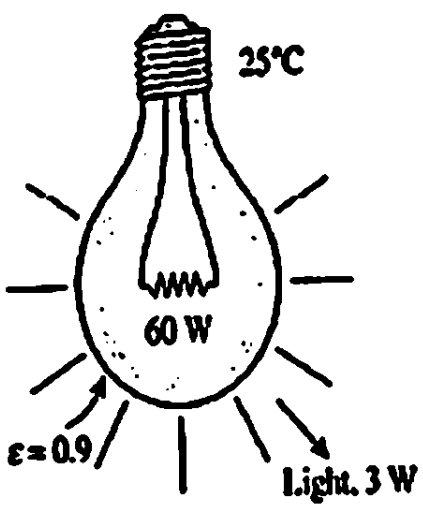


FIGURE P20–23

convection and radiation. Consider an 8-cm-diameter 60-W lightbulb in a room at 25°C. The emissivity of the glass is 0.9. Assuming that 5 percent of the energy passes through the glass bulb as light with negligible absorption and the rest of the energy is absorbed and dissipated by the bulb itself by natural convection and radiation, determine the equilibrium temperature of the glass bulb. Assume the interior surfaces of the room lo be at room temperature.

Step 1 of 7

Given data:

Diameter of bulb, $D = 8\text{cm}$

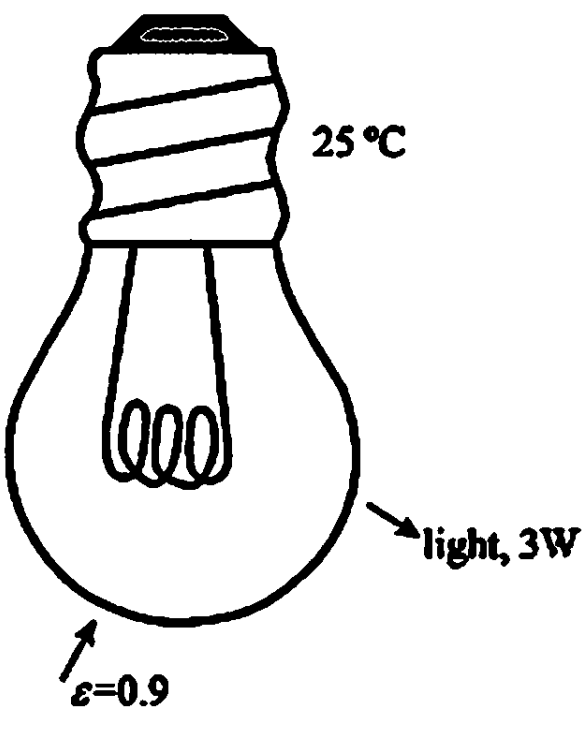
Wattage of bulb, $\dot{Q}_e = 60\,\text{W}$

Ambient temperature, $T_\infty = 25°\text{C}$

Emissivity of the glass, $\varepsilon = 0.9$

Step 2 of 7

Given figure:



Step 3 of 7

As it is converting 95% in to heat energy under steady condition heat generated,

Heat loss,

$\dot{Q} = 0.95 \times 60$

$\dot{Q} = 57\,\text{W}$

Assumptions:

(i) Steady conditions

(ii) Properties take at 1atm pressure.

(iii) Radiation and convection considered.

(iv) Assuming $T_S = 175°\text{C}$

To find T_s from the relation $\dot{Q} = hA(T_s - T_\infty)$ we need to calculate heat transfer Co-efficient (h)

By calculating Rayleigh and Nusselt number, we get

Film temperature of the given system,

$$T_f = \frac{(T_S + T_\infty)}{2}$$

$$= \frac{(175+25)}{2}$$

$$T_f = 100°\text{C}$$

Step 4 of 7

Taking properties from the table at $T_f = 100°\text{C}$

Prandtl number, $\text{Pr} = 0.7111$

Thermal conductivity, $k = 0.03095\ \text{W/m}\cdot°\text{C}$

Kinematic viscosity, $\nu = 2.306\times10^{-5}\,\text{m}^2/\text{s}$

Volume expansion coefficient,

$$\beta = 1/T_f$$

$$= \frac{1}{273+100}$$

$$= 2.6809\times10^{-3}$$

Characteristic length = Diameter of the bulb

= 0.08m

Step 5 of 7

Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_S - T_\infty)D^3}{\nu^2}\times\text{Pr}$$

$$= \frac{9.81\times2.6809\times10^{-3}\times(175-25)\times(0.08)^3}{(2.306\times10^{-5})^2}\times0.7111$$

$$= 2701055.25$$

Nusselt number considering sphere from the table,

Nusselt number,

$$\text{Nu} = 2 + \frac{0.589\text{Ra}^{1/4}}{\left(1+(0.469/\text{Pr})^{9/16}\right)^{4/9}}$$

$$= 2 + \frac{0.589\times(2701055.25)^{1/4}}{\left(1+\left(\frac{0.469}{0.7111}\right)^{9/16}\right)^{4/9}}$$

$$= 20.428$$

Step 6 of 7

Heat transfer coefficient,

$$h = \frac{k}{D}\text{Nu}$$

$$= \frac{0.03095\times20.428}{0.08}$$

$$= 7.903\,\text{W/m}^2\cdot°\text{C}$$

Surface area,

$$A_s = \pi D^2$$

$$= \pi(0.08)^2$$

$$= 0.02011\text{m}^2$$

Step 7 of 7

From the relation of the heat transfer, we get

$\dot{Q} = hA_S(T_S - T_\infty) + \varepsilon.A_S.\sigma\left(T_S^4 - T_\infty^4\right)$ (Because Radiation is also considered)

$$57 = 7.903\times0.02011(T_S - 25) + 0.9\times0.02011$$

$$\times5.67\times10^{-8}\left[(T_S+273)^4 - (298)^4\right]$$

$$57 = 0.15893(T_S - 25) + 1.0262\times10^{-9}\left[(T_S+273)^4 - (298)^4\right]$$

$$T_S = 174.84°\text{C}$$

This is approximately equal to assumed T_S that is 175°C

Therefore the equilibrium temperature of the glass tube, $\boxed{T_S = 174.84°\text{C}}$

Question:

A 450-W cylindrical resistance heater is 0.75 m long and 0.5 cm in diameter. The resistance wire is placed horizontally in a fluid at 20°C. Determine the outer surface temperature of the resistance wire in steady operation if the fluid is (*a*) air and (*b*) water. Ignore any heat transfer by radiation. Use properties at 500°C for air and 40°C for water.

Step 1 of 7

a) Calculate the surface temperature of the wire if the fluid is air:

Obtain the properties for air from properties of air table as follows:

Consider the properties of air are taken at $T_f = 500°C$.

At $T_f = 500°C$;

The thermal conductivity k is $0.05572\ W/m \cdot K$.

The kinematic viscosity ν is $7.806 \times 10^{-5}\ m^2/s$.

The Prandtl number Pr is 0.6986.

Calculate the value for expansion coefficient β:

$$\beta = \frac{1}{T_f}$$

Here, T_f is the average temperature.

Substitute 773 K $(500+273)$ K for T_f

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{(500+273)}$$
$$= 0.001294\ K^{-1}$$

Step 2 of 7

Calculate the Rayleigh Number:

Obtain the surface area of the heater as follows;

$$A_s = \pi DL$$

Here, *D* is the diameter of the heater and *L* is the length of the heater.

Substitute 0.005 m for *D* and 0.75 m for *L*

$$= \pi \times 0.005 \times 0.75$$
$$= 0.01178\ m^2$$

Obtain the characteristic length of the wire as follows.

$$L_c = D = 0.005\ m.$$

Assume the surface temperature to be $T_s = 1680°C$

Rayleigh Number,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} Pr$$

Here, g is the acceleration due to gravity, β is the expansion coefficient, T_s is the surface temperature, T_∞ is the temperature of the fluid, D is the diameter of the heater, ν is the kinematic viscosity and Pr is the Prandtl number.

Substitute, $9.81\ m/s$ for *g*, 0.001294 for β, $1680°C$ for T_s, $20°C$ for T_∞, $0.005\ m$ for *D*, $7.806 \times 10^{-5}\ m^2/s$ for *v* and 0.6986 for Pr.

$$Ra = \frac{9.81 \times 0.001294 \times (1680-20) \times 0.005^3}{(7.806 \times 10^{-5})^2}(0.6986)$$
$$= \frac{2.634 \times 10^{-6}}{6.09 \times 10^{-9}}(0.6986)$$
$$= 302.153$$

Step 3 of 7

Calculate the Nusselt number by using the formula as follows:

$$Nu = \left\{0.6 + \frac{0.387 Ra^{1/6}}{\left[1+(0.559/Pr)^{9/16}\right]^{8/27}}\right\}^2$$

Here, Ra is the Rayleigh number and Pr is the Prandtl number.

Substitute 302.153 for Ra and 0.6986 for Pr

$$= \left\{0.6 + \frac{0.387(302.153)^{1/6}}{\left[1+(0.559/0.6986)^{9/16}\right]^{8/27}}\right\}^2$$
$$= \left\{0.6 + \frac{1.0024}{1.20608}\right\}^2$$
$$= 2.048$$

Calculate the heat transfer coefficient using the formula as follows:

$$Nu = \frac{hD}{k}$$

Here, *h* is the heat transfer coefficient, *D* ids the diameter of the heater and Nu is the Nusselt number.

Substitute $0.05572\ W/m \cdot K$ for *k*, 0.005 m for *d* and 2.048 for Nu.

$$h = \frac{k}{D} Nu$$
$$= \frac{0.05572}{0.005} \times 2.048$$
$$= 22.822\ W/m^2 \cdot K$$

$$h = \frac{k}{D}\text{Nu}$$
$$= \frac{0.05572}{0.005} \times 2.048$$
$$= 22.822\ \text{W/m}^2 \cdot \text{K}$$

Calculate the heat transfer:

$$\dot{Q} = hA_s(T_s - T_\infty)$$

Here, T_s is the surface temperature, T_∞ is the temperature of the fluid, A_s is the surface area and h is the heat transfer coefficient.

Substitute $1680°C$ for T_s, $20°C$ for T_∞, $22.822\ \text{W/m}^2 \cdot \text{K}$ for h and $0.01178\,\text{m}^2$ for A_s.

$$450 = 22.822 \times 0.01178 \times (T_s - 20)$$

$$T_s = 1693.83°\text{C}$$

Since assumed surface temperature is approximately equal to the obtained value.

Hence the outer surface temperature T_s of the wire if the fluid is air is $\boxed{1693.83°\text{C}}$.

Step 4 of 7

b) Calculate the surface temperature of the wire if the fluid is water:

Obtain the properties for water from properties of saturated water table:

Consider the properties of air are taken at $T_f = 500°\text{C}$.

At $T_f = 40°\text{C}$;

The thermal conductivity k is $0.631\ \text{W/m} \cdot \text{K}$.

The kinematic viscosity ν is follows.

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.653 \times 10^{-3}}{992.1}$$
$$= 0.6582 \times 10^{-6}\ \text{m}^2/\text{s}$$

The Prandtl number Pr is 4.32.

Calculate the value for expansion coefficient β:

$$\beta = \frac{1}{T_f}$$

Here, T_f is the average temperature.

Substitute $313\ \text{K}(40 + 273)\ \text{K}$ for T_f

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{40 + 273}$$
$$= 0.00319\ \text{K}^{-1}$$

Step 5 of 7

Calculate the Rayleigh Number:

Obtain the surface area of the heater as follows;

$$A_s = \pi DL$$

Here, D is the diameter of the heater and L is the length of the heater.

Substitute 0.005 m for D and 0.75 m for L

$$= \pi \times 0.005 \times 0.75$$
$$= 0.01178\,\text{m}^2$$

Obtain the characteristic length of the wire as follows.

$$L_c = D = 0.005\ \text{m}.$$

Assume the surface temperature to be $T_s = 40°\text{C}$

Rayleigh Number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr}$$

Here, g is the acceleration due to gravity, β is the expansion coefficient, T_s is the surface temperature, T_∞ is the temperature of the fluid, D is the diameter of the heater, ν is the kinematic viscosity and Pr is the Prandtl number.

Substitute, $9.81\,\text{m/s}$ for g, 0.00319 for β, $40°C$ for T_s, $20°C$ for T_∞, $0.005\ \text{m}$ for D, $0.6582 \times 10^{-6}\,\text{m}^2/\text{s}$ for ν and 4.32 for Pr.

$$= \frac{9.81 \times 0.00319(40 - 20) \times 0.005^3}{(0.6582 \times 10^{-6})^2} \times 4.32$$
$$= \frac{78.23 \times 10^{-9}}{4.3323 \times 10^{-13}} \times 4.32$$
$$= 780078.94$$

Step 6 of 7

Calculate the Nusselt number by using the formula as follows:

$$\text{Nu} = \left\{0.6 + \frac{0.387\text{Ra}^{1/6}}{\left[1 + (0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

Here, Ra is the Rayleigh number and Pr is the Prandtl number.

Substitute 780078.94 for Ra and 4.32 for Pr

$$\mathrm{Nu}=\left\{0.6+\frac{0.387\mathrm{Ra}^{1/6}}{\left[1+(0.559/\mathrm{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

.024P pg 3

Here, Ra is the Rayleigh number and Pr is the Prandtl number.

Substitute 780078.94 for Ra and 4.32 for Pr

$$=\left\{0.6+\frac{0.387\times780078.94^{1/6}}{\left[1+(0.559/4.32)^{9/16}\right]^{8/27}}\right\}^2$$

$$=\left\{0.6+\frac{3.713}{1.085}\right\}^2$$

$$=16.177$$

Calculate the heat transfer coefficient using the formula as follows:

$$\mathrm{Nu}=\frac{hD}{k}$$

Here, h is the heat transfer coefficient, D ids the diameter of the heater and Nu is the Nusselt number.

Substitute $2041.53\ \mathrm{W/m\cdot K}$ for k, 0.005 m for d and 16.177 for Nu.

$$h=\frac{k}{D}\mathrm{Nu}$$

$$=\frac{0.631}{0.005}\times16.177$$

$$=2041.53\ \mathrm{W/m^2\cdot K}$$

Step 7 of 7

Calculate the heat transfer:

$$\dot{Q}=hA_s(T_s-T_\infty)$$

Here, T_s is the surface temperature, T_∞ is the temperature of the fluid, A_s is the surface area and h is the heat transfer coefficient.

Substitute $20°C$ for T_∞, $22.822\ \mathrm{W/m^2\cdot K}$ for h and $0.01178\ \mathrm{m^2}$ for A_s.

$$450=2041.53\times0.01178\times(T_s-20)$$

$$=\frac{450}{24.049}+20$$

$$T_s=38.711°\mathrm{C}$$

Since assumed surface temperature is approximately equal to the obtained value.

Hence the outer surface temperature T_s of the wire if the fluid is water is $\boxed{38.711°\mathrm{C}}$.

Question: 20.025P

In a plant that manufactures canned aerosol paints, the cans are temperature-tested in water baths at 60°C before they are shipped to ensure that they withstand temperatures up to 55°C during transportation and shelving. The cans, moving on a conveyor, enter the open hot water bath, which is 0.5 m deep, 1 m wide, and 3.5 m long, and move slowly in the hot water toward the other end. Some of the cans fail the test and explode in the water bath. The water container is made of sheet metal, and the entire container is at about the same temperature as the hot water. The emissivity of the outer surface of the container is 0.7. If the temperature of the surrounding air and surfaces is 20°C, determine the rate of heat loss from the four side surfaces of the container (disregard the top surface, which is open).

The water is heated electrically by resistance heaters, and the cost of electricity is $0.085/kWh. If the plant operates 24 h a day 365 days a year and thus 8760 h a year, determine the annual cost of the heat losses from the container for this facility.

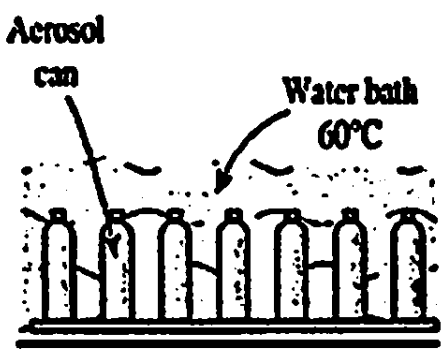


FIGURE P20–25

Step 1 of 7

Given data:

Water bath temperature, $T_s = 60°C$

Depth of water bath, $L = 0.5m$

Surrounding air temperature, $T_\infty = 20°C$

Emissivity, $\varepsilon = 0.7$

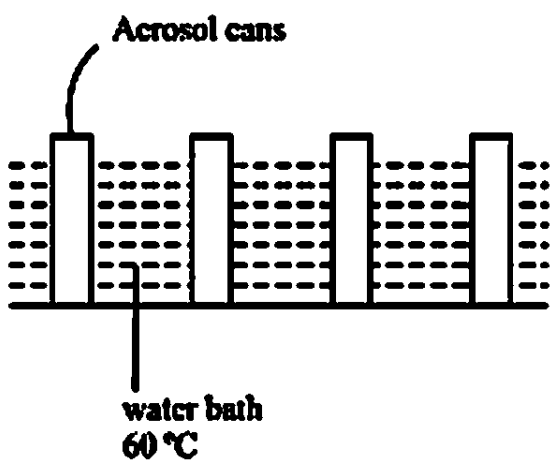
Container dimensions, $3.5m \times 1m$

Cost of electricity is $\$0.085/kwh$

Operations hours, $t = 8760h$

Step 2 of 7

Schematic representation of the given system,



Step 3 of 7

Heat transfer rate of the given system,

$$\dot{Q} = hA_s(T_s - T_\infty)$$

h And A_s are unknown here, so that find the heat transfer coefficient

Taking properties of air at film temperature,

$$T_f = \frac{60+20}{2}$$

$$= 40°C$$

Thermal conductivity, $k = 0.02662 W/m \cdot °C$

Prandtl number, $Pr = 0.7255$

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{(273+40)}$$

$$= 3.19488 \times 10^{-3} K^{-1}$$

Kinematic viscosity, $\nu = 1.702 \times 10^{-5} m^2/s$

Characteristic length, $L_c = 0.5m$

Step 4 of 7

Rayleigh number,

$$Ra_L = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2} \times Pr$$

$$= \frac{9.81 \times 3.19488 \times 10^{-3} \times (60-20) \times (0.5)^3}{(1.702 \times 10^{-5})^2} \times 0.7255$$

$$= 3.9247 \times 10^8$$

Considering vertical plate,

Nusselt number,

$$Nu = \left(0.825 + \frac{0.387 Ra^{1/6}}{(1+(0.492/Pr)^{9/16})^{8/27}}\right)^2$$

$$= \left(0.825 + \frac{0.387(3.9247 \times 10^8)^{1/6}}{\left(1+\left(\frac{0.492}{0.7255}\right)^{9/16}\right)^{8/27}}\right)^2$$

$$= 92.495$$

Step 5 of 7

Heat transfer co-efficient of the given system,

$$h = \frac{k}{L_c} Nu$$

$$= \frac{0.02662}{0.5} \times 92.495$$

$$= 4.9244 W/m^2 \cdot °C$$

Surface area of container with the given dimensions,

$$A_s = 2[(3.5 \times 0.5) + (1 \times 0.5)]$$

$$= 4.5m^2$$

Step 6 of 7

Heat lost by Convection,

$$\dot{Q}_{conv} = hA(T_s - T_\infty)$$

$$= 4.9244 \times 4.5 \times (60-20)$$

$$= 886.392 W$$

Heat lost by Radiation,

$$Q_{rad} = EA\sigma(T_s^4 - T_\infty^4)$$

$$= 0.7 \times 4.5 \times 5.67 \times 10^{-8}[(333)^4 - (293)^4]$$

$$= 879.865W$$

Step 7 of 7

Total heat loss of the given system,

$$Q_{total} = Q_{conv} + Q_{rad}$$

$$= 886.392 + 879.865$$

$$= 1766.26 W$$

$$= 1.7662 kW$$

Total Heat lost in a year,

$$Q = Q_{total} \times 8760$$

$$= 1.7662 \times 8760$$

$$= 15471.91 kW \cdot h$$

Total annual cost of the heat lost

cost = Heat lost in a year × cost of electricity

$$= 15471.91 \times 0.085$$

$$= \$1315.11$$

Therefore total annual cost of the heat lost, $\boxed{\text{cost} = \$1315.11}$

Question: 20.026P

Reconsider Prob. 20-25. In order to reduce the heating cost of the hot water, it is proposed to insulate the side and bottom surfaces of the container with 5-cm-thick fiberglass insulation (k= 0.035 W/m·K) and to wrap the insulation with aluminum foil (ε = 0.1) in order to minimize the heat loss by radiation. An estimate is obtained from a local insulation contractor, who proposes to do the insulation job for $350, including materials and labor. Would you support this proposal? How long will it take for the insulation to pay for itself from the energy it saves?

Step 1 of 8

Calculate heat transfer rate

$$\dot{Q}=hA_s\left(T_s-T_\infty\right)$$

Here,

The Convection heat coefficient is h

The cross-section area is A_s

Taking the properties by interpolation

At film temperature $T_f=22.5^\circ\text{C}$, take the properties of air,

Thermal conductivity,

$$k=0.02514+\frac{(0.02551-0.02514)}{(25-20)}(22.5-20)$$

$$=0.025325\,\text{W/m}^\circ\text{C}$$

Step 2 of 8

Kinematic viscosity,

$$\nu=1.516\times10^{-5}+\frac{10^{-5}(1.562-1.516)}{(25-20)}(22.5-20)$$

$$=1.539\times10^{-5}\,\text{m}^2/\text{s}$$

Prandtl number,

$$\text{Pr}=0.7309+\frac{(0.7296-0.7309)}{(25-20)}(22.5-20)$$

$$=0.73025$$

And

$$\beta=\frac{1}{T_f}$$

$$=\frac{1}{(22.5+273)}$$

$$=3.3841\times10^{-3}\text{K}^{-1}$$

Step 3 of 8

Calculate Rayleigh number,

$$Ra_L=\frac{g\beta(T_s-T_\infty)L_c^{\,3}}{\nu^2}\cdot\text{Pr}$$

$$=\frac{9.81\times3.841\times10^{-3}(25.0-20)(0.5)^3}{\left(1.539\times10^{-5}\right)^2}$$

$$=6.395\times10^7$$

Calculate Nusselt number,

$$Nu=\left[0.825+\frac{0.387(Ra)^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$=\left[0.825+\frac{0.387\left(6.397\times10^7\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.73025}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$=53.68$$

Step 4 of 8

Calculate Heat transfer coefficient,

$$h=\frac{k}{L_c}Nu$$

$$=\frac{0.025325}{0.5}\times53.68$$

$$=2.72\,\text{W/m}^2{}^\circ\text{C}$$

Calculate the area of container,

$$A_s=2(0.5\times3.6+0.5\times1.1)$$

$$=4.7\,\text{m}^2$$

Calculate Heat loss rate by convection

$$Q_{conv}=hA_s\left(T_s-T_\infty\right)$$

$$=2.72\times4.7(25-20)$$

$$=63.89\,\text{W}$$

Step 5 of 8

Calculate Heat lost by Radiation

$$Q_{rad}=\varepsilon\sigma\text{A}\left(T_s^4-T_\infty^4\right)$$

$$=0.1\times5.67\times10^{-8}\times4.7\left((25+273)^4-(20+273)^4\right)$$

$$=13.753\,\text{w}$$

Calculate total heat loss,

$$Q_{total}=Q_{conv}+Q_{rad}$$

$$=63.89+13.753$$

$$Q_{total}=77.64\,\text{W}$$

Step 6 of 8

Now to verify that the assumed surface temperature (T_s) is now much accurate we till find T_s by conduction between tank and insulation and that is equal to Heat loss by insulation exposed surface under steady conditions.

Conduction between hot tank and insulation = Total heat loss by outer surface

$$kA_s\frac{T_{hot\,bath}-T_s}{L}=77.64$$

$$\frac{0.035\times4.7(60-T_s)}{0.05}=77.64$$

$$T_s=36.4^\circ\text{C}$$

Not very close but ok.

Step 7 of 8

Now calculate heat loss in year

$$Q_{year}=Q_{total}\times hrs\,in\,year$$

$$=77.64\times8760$$

$$=680126\,\text{W-h}$$

The cost of the power, $cost=0.085/\text{kWh}$

Therefore,

$$\text{Totalcost}=680.126\times0.085$$

$$=\$57.81$$

Money saved in one year $=1315-57.81$

$$=\$1257.189$$

Money to be spent to insulation $=\$350$

Step 8 of 8

Clear from figures obtained that pay back will take less than one year.

$$\text{Pay back duration}=\frac{350}{1257.189}\times365$$

$$=101.61\text{Days}$$

Therefore, the Payback period is $\boxed{101.61\text{Days}}$

I would support the proposal.

Question: 20.027P pg 1

Consider a 15-cm × 20-cm printed circuit board (PCB) that has electronic components on one side. The board is placed in a room at 20°C. The heat loss from the back surface of the board is negligible. If the circuit board is dissipating 8 W of power in steady operation, determine the average temperature of the hot surface of the board, assuming the board is (*a*) vertical; (*b*) horizontal with hot surface facing up; and (c) horizontal with hot surface facing down. Take the emissivity of the surface of the board to be 0.8 and assume the surrounding surfaces to be at the same temperature as the air in the room.

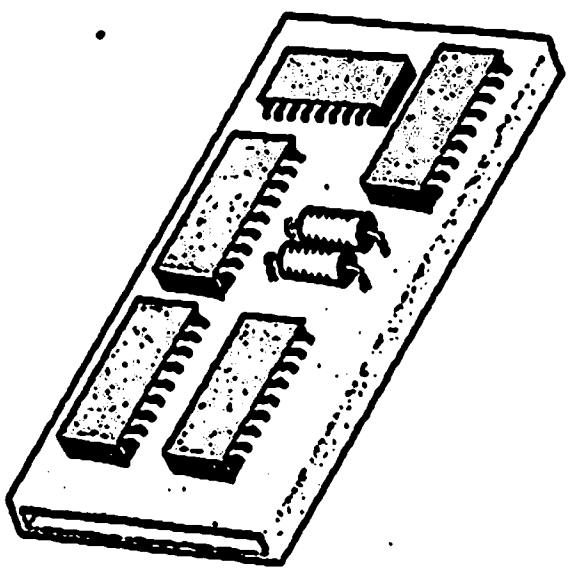


FIGURE P20–27

Step 1 of 12

Length of the Pinned circuit board, $L = 20\text{cm} = 0.2\text{m}$

Width of the Pinned circuit $W = 15\text{cm} = 0.15\text{m}$

Room temperature, $T_\infty = 20°\text{C}$

Heat dissipated, $\dot{Q} = 8\text{W}$

Emissivity $\varepsilon = 0.8$

Ambient temperature $T_{amb} = 20°\text{C}$

Step 2 of 12

Assuming the air temperature in the room as $T_s = 45$, calculate mean film temperature of the air

$$T_f = \frac{45+20}{2}$$
$$= 32.5°\text{C}$$

From the properties of air at 1 atm pressure and $T_f = 32.5°\text{C}$

By using the method of interpolation,

Thermal conductivity is

$$k = 0.02588 + \frac{(0.02625-0.02588)}{(35-30)}(32.5-30)$$
$$= 0.026065\ \text{W/m}\cdot°\text{C}$$

Prandtl number is

$$\text{Pr} = 0.7282 + \frac{0.7268-0.7282}{(35-30)}(32.5-30)$$
$$= 0.7275$$

Kinematic viscosity is

$$\nu = 1.608\times10^{-5} + \frac{(1.655-1.608)^{10^{-5}}}{(35-30)}\times(32.5-30)$$
$$= 1.6315\times10^{-5}\ \text{m}^2/\text{s}$$

Step 3 of 12

Volume expansion coefficient is

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{(273+32.5)}$$
$$= 3.273\times10^{-3}\text{K}^{-1}$$

Characteristic length

$$(L_c) = L$$
$$= 0.2\text{m}$$

Calculation Rayleigh number

$$\text{Ra}_L = \frac{g\beta(T_s-T_\infty)L_c^3}{\nu^2}\times\text{Pr}$$
$$= \frac{9.81\times3.273\times10^{-3}(45-20)(0.2)^3}{\left(1.6315\times10^{-5}\right)^2}\times0.7275$$
$$= 175.528\times10^5$$

Step 4 of 12

(a) Now Nusselt N_g for vertical printed circuit board is

$$\text{Nu} = \left[0.825 + \frac{0.387(\text{Ra}_L)^{1/6}}{\left[1+(0.492/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387(175.528)^{1/6}}{\left[1+(0.492/0.7275)^{9/16}\right]^{8/27}}\right]^2$$
$$= 36.777$$

Heat transfer co-efficient,

$$h = \frac{k\times\text{Nu}}{L_c}$$
$$= \frac{0.026065}{0.2}\times36.777$$
$$= 4.7931\ \text{W/m}^2\cdot°\text{C}$$

Step 5 of 12

$= 4.7931\ \text{W/m}^2\cdot^\circ\text{C}$

Step 5 of 12

Total heat transfer by convection and radiation,

$$Q = Q_{conv} + Q_{rad}$$

$$Q = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_{amb}^4\right)$$

$$8 = 4.7931(0.03)(T_s - 20) + 0.8\left(5.67\times10^{-8}\right)(0.03)\left[(T_s + 273)^4 - (20+273)^4\right]$$

By solving the above expression, we get $T_s = 46^\circ\text{C}$

Therefore average temperature of the hot surface board is $\boxed{T_s = 46^\circ\text{C}}$

Step 6 of 12

(b) Horizontal circuit board with hot surface facing up

Characteristic length in this set up

$$L_c = \frac{A_s}{p}$$

$$= \frac{(0.20\times0.15)}{2(0.2+0.15)}$$

$$= 0.04285\text{m}$$

Therefore calculate Rayleigh number.

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$

$$= \frac{9.81\times3.273\times10^{-3}\times(45-20)(0.04285)^3}{\left(1.6315\times10^{-5}\right)^2}\times0.7275$$

$$= 172696.48$$

Step 7 of 12

Calculate the Nusselt number by using the below equation

$$\text{Nu} = 0.54\text{Ra}^{1/4}$$

$$= 0.54(172696.48)^{1/4}$$

$$= 11.008$$

Therefore heat transfer coefficient,

$$h = \frac{k\times\text{Nu}}{L_c}$$

$$= \frac{0.026065\times11.008}{0.04285}$$

$$= 6.696\ \text{W/m}^2\cdot^\circ\text{C}$$

Step 8 of 12

Total heat transfer by convection and radiation,

$$Q = Q_{conv} + Q_{rad}$$

$$Q = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_{amb}^4\right)$$

$$8 = 6.696(0.03)(T_s - 20) + 0.8\left(5.67\times10^{-8}\right)(0.03)\left[(T_s + 273)^4 - (20+273)^4\right]$$

By solving the above expression, we get $T_s = 42^\circ\text{C}$

Therefore average temperature of the hot surface when board facing up is $\boxed{T_s = 42^\circ\text{C}}$

Step 9 of 12

(c) Horizontal circuit board with hot surface facing down

Calculation made by assuming $T_s = 45^\circ\text{C}$ Are not sufficiently accurate for this set up so assuming $T_s = 50^\circ\text{C}$ In this case, calculate mean film temperature of the air.

$$T_f = (T_s + T_\infty)/2$$

$$= (50+20)/2$$

$$= 35^\circ\text{C}$$

From the properties of air at 1 atm pressure and $T_f = 35^\circ\text{C}$

Thermal conductivity is $k = 0.02625\ \text{W/m}\cdot^\circ\text{C}$

Prandtl number is $\text{Pr} = 0.7268$

Kinematic viscosity is $\nu = 1.655\times10^{-5}\ \text{m}^2/\text{s}$

Step 10 of 12

Volume expansion coefficient is

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{(50+273)}$$

$$= 3.2467\times10^{-3}\ \text{K}^{-1}$$

Characteristic length

$$L_c = \frac{A_s}{p}$$

$$= \frac{(0.20\times0.15)}{2(0.2+0.15)}$$

$$= 0.04285\text{m}$$

Step 11 of 12

Calculate the Rayleigh number.

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$

$$= \frac{9.81\left(3.2467\times10^{-3}\right)(50-20)(0.04285)^3}{\left(1.655\times10^{-5}\right)^2}\times0.7268$$

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot \mathrm{Pr}$$

$$= \frac{9.81\left(3.2467\times10^{-3}\right)(50-20)(0.04285)^3}{\left(1.655\times10^{-5}\right)^2}\times 0.7268$$

$$= 199482$$

Calculate the Nusselt number.

$$\mathrm{Nu} = 0.27(\mathrm{Ra}_L)^{0.25}$$

$$= 0.27(199482)^{0.25}$$

$$= 5.706$$

Step 12 of 12

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\mathrm{Nu}$$

$$= \frac{0.02625}{0.04285}\times 5.706$$

$$= 3.496\,\mathrm{W/m^2\cdot^\circ C}$$

Total heat transfer by convection and radiation,

$$Q = Q_{conv} + Q_{rad}$$

$$Q = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_{amb}^4\right)$$

$$8 = 3.496(0.03)(T_s - 20) + 0.8\left(5.67\times10^{-8}\right)(0.03)\left[(T_s+273)^4 - (20+273)^4\right]$$

By solving the above expression, we get $T_s = 50.3^\circ C$

Therefore average temperature of the hot surface when board faces down is $\boxed{T_s = 50.3^\circ C}$

Question: 20.028P

Reconsider Prob. 20-27. Using EES (or other) software, investigate the effects of the room temperature and the emissivity of the board on the temperature of the hot surface of the board for different orientations of the board. Let the room temperature vary from 5°C to 35°C and the emissivity from 0.1 to 1.0. Plot the hot surface temperature for different orientations of the board as the functions of the room temperature and the emissivity, and discuss the results.

Step 1 of 1

Given

Power dissipation is maintained constant at 60 W

Calculate the Convection heat transfer coefficient,

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$h = \frac{\dot{Q}}{A_s(T_s - T_\infty)}$$

Here,

Where heat transfer surface area is calculated as

$$A = \pi DL$$

$$= \pi(0.02\text{ m})(0.8\text{ m})$$

$$= 0.05027\text{ m}^2$$

Therefore heat transfer coefficient becomes

$$h = \frac{60\text{ W}}{(0.05027\text{ m}^2)(120-20)^\circ\text{C}}$$

$$\therefore h = 11.9355\text{ W/m}^2.^\circ\text{C}$$

The radiation heat loss from the cylinder is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$

$$= (0.1)(0.05027\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)[(120+273\text{ K})^4 - (20+273\text{ K})^4]$$

$$= 4.7\text{ W}$$

Therefore, the fraction of heat loss by radiation is

$$\text{Radiation fraction} = \frac{\dot{Q}_{radiation}}{\dot{Q}_{total}}$$

$$= \frac{4.7\text{ W}}{60\text{ W}}$$

$$= 0.07833$$

$$= 7.833\%$$

This is greater than 5%. Therefore, the radiation effect is still more than acceptable, and corrections must be made for the radiation effect.

Question: 20.029P

Consider a vertical plate with length L, placed in quiescent air. If the Him temperature is 20°C and the average Nusselt number in natural convection is of the form $\mathrm{Nu} = \mathrm{CR}_L^n$, show that the average heat transfer coefficient can be expressed as

$h = 1.51(\Delta T/L)^{1/4}$ $\quad 10^4 < \mathrm{Ra}_L < 10^9$

$h = 1.19\Delta T^{1/3}$ $\quad 10^{10} < \mathrm{Ra}_L < 10^{13}$

Step 1 of 7

Given data:

Length of vertical plate is L

Mean film temperature, $T_f = 20°\mathrm{C}$

Nusselt number, $\mathrm{Nu} = C \times \mathrm{Ra}_L^n$

Step 2 of 7

From the ideal gas properties of the air at 1atm pressure,

At $T_f = 20°\mathrm{C}$

Thermal conductivity, $k = 0.02514\,\mathrm{w/m{\cdot}°C}$

Prandtl number, $\mathrm{Pr} = 0.7309$

Kinematic viscosity, $\nu = 1.516\times10^{-5}\,\mathrm{m^2/s}$

Volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{298}$$

$$= 3.413\times10^{-3}\mathrm{k}^{-1}$$

Characteristic length is the length of vertical plate (L)

Step 3 of 7

Rayleigh number,

$$\mathrm{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\times\mathrm{Pr}$$

$$= \frac{9.81\times3.413\times10^{-3}\times\Delta T\times L^3}{\left(1.516\times10^{-5}\right)^2}\times0.7309$$

$$= 1.0648\times10^8\,\Delta T\times L^3$$

where,

$\Delta T = T_s - T_\infty$ is the temperature difference

Step 4 of 7

From the empirical correlation for the average Nusselt number for natural convention over surfaces

For vertical plate,

Nusselt number for range of $\mathrm{Ra}\left(10^4 - 10^9\right)$

As given Nu in form of $C\times\mathrm{Ra}_L^n$

$$\mathrm{Nu} = 0.59\times\mathrm{Ra}_L^{0.25}$$

$$= 0.59\left(1.0648\times10^8\,\Delta T\times L^3\right)^{0.25}$$

Step 5 of 7

Now heat transfer coefficient,

$$h = \frac{k}{L_c}\mathrm{Nu}$$

$$= \frac{0.02514}{L}\times\left(0.59\left(1.0648\times10^8\,\Delta T\times L^3\right)^{0.25}\right)$$

$$= 1.51\left(\frac{\Delta T}{L}\right)^{1/4}$$

Therefore average heat transfer coefficient is $\boxed{h = 1.51\left(\frac{\Delta T}{L}\right)^{1/4}}$

Step 6 of 7

From the empirical correlation for the average Nusselt number for natural convention over surfaces

For vertical plate

Nusselt number for range of $\mathrm{Ra}\left(10^{10} - 10^{13}\right)$

$$\mathrm{Nu} = 0.1\times\mathrm{Ra}_L^{1/3}$$

$$= 0.1\times\left(1.0648\times10^8\,\Delta T\times L^3\right)^{1/3}$$

$$= 47.398\left(\Delta T\times L^3\right)^{1/3}$$

Step 7 of 7

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\mathrm{Nu}$$

$$= \frac{0.02514}{L}\times\left(47.398\left(\Delta T\times L^3\right)^{1/3}\right)$$

$$= 1.19(\Delta T)^{1/3}$$

Therefore average heat transfer coefficient is $\boxed{h = 1.19(\Delta T)^{1/3}}$

Question: 20.030P

A circular grill of diameter 0.25 m has an emissivity of 0.8. If the surface temperature is maintained at 150°C, determine the required electrical power when the room air and surroundings are at 30°C.

Step 1 of 8

Given data:

Grill diameter, $d = 0.25\text{m}$

$$r = \frac{0.25}{2}$$

$$= 0.125\text{m}$$

Emissivity, $\varepsilon = 0.8$

Surface temperature, $T_s = 150°\text{C}$

Surrounding Ambient air Temperature, $T_\infty = 30°\text{C}$

Step 2 of 8

Considering the grill as a horizontal plate, we have

Mean film temperature of the air,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{150+30}{2}$$

$$= 90°\text{C}$$

From the ideal gas properties of the air 1 atm pressure

At $T_f = 90°\text{C}$

Thermal conductivity, $k = 0.03024\,\text{w/m}\,°\text{k}$

Prandtl number, $\text{Pr} = 0.7132$

Kinematic viscosity, $\nu = 2.201\times10^{-5}\,\text{m}^2/\text{s}$

Step 3 of 8

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{363}$$

$$= 2.7548\times10^{-3}\text{K}^{-1}$$

Step 4 of 8

Characteristic length for horizontal plate

$$L_c = \frac{A_s}{p}$$

$$= \frac{\pi r^2}{2\pi r}$$

where

p is the perimeter

Therefore,

$L_c = r/2$

$= 0.0625\text{m}$

Step 5 of 8

Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\times\text{Pr}$$

$$= \frac{9.81\times2.7548\times10^{-3}\times(150-30)(0.0625)^3}{\left(2.201\times10^{-5}\right)^2}\times0.7132$$

$$= 1.166\times10^6$$

Step 6 of 8

From the empirical correlation for the average Nusselt number for natural convention over surfaces

For horizontal plate:

Nusselt number,

$$\text{Nu} = 0.54\,\text{Ra}_L^{0.25}$$

$$= 0.54\times\left(1.166\times10^6\right)^{0.25}$$

$$= 17.743$$

Step 7 of 8

Now heat transfer co-efficient,

$$h = \frac{k}{L_c}\text{Nu}$$

$$= \frac{0.03024}{0.0625}\times17.743$$

$$= 8.585\,\text{W/m}^2.°\text{C}$$

Step 8 of 8

Total rate of heat transfer,

$$Q_{total} = Q_{conv} + Q_{rad}$$

$$= hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_{Amb}^4\right)$$

$$= 8.585\times\pi\times(0.125)^2\times(150-30)$$

$$+0.8\times5.67\times10^{-8}\times\pi\times(0.125)^2\left((423)^4-(303)^4\right)$$

$$= 103.08\,\text{W}$$

Therefore total rate of heat transfer, $\boxed{Q_{total} = 103.08\,\text{W}}$

Question:

A manufacturer makes absorber plates that are 1.2 m × 0.8 m in size for use in solar collectors. The back side of the plate is heavily insulated, while its front surface is coated with black chrome, which has an absorptivity of 0.87 for solar radiation and an emissivity of 0.09. Consider such a plate placed horizontally outdoors in

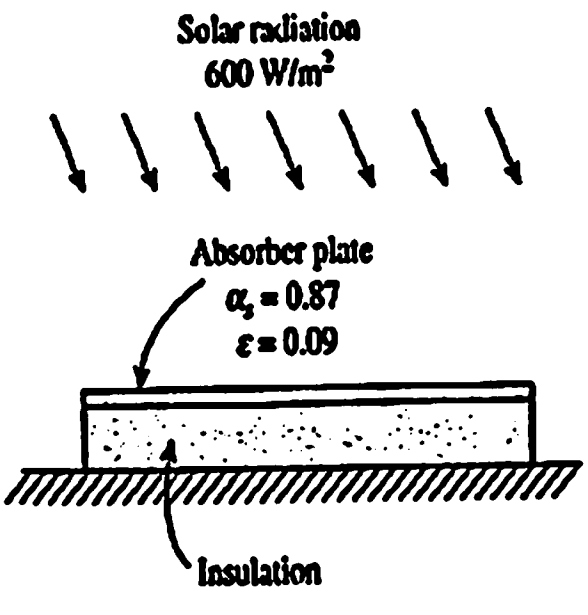


FIGURE P20–31

calm air at 25°C. Solar radiation is incident on the plate at a rate of 600 W/m2. Taking the effective sky temperature to be 10°C, determine the equilibrium temperature of the absorber plate. What would your answer be if the absorber plate is made of ordinary aluminum plate that has a solar absorptivity of 0.28 and an emissivity of 0.07?

Step 1 of 14

Given data:

Length of absorber plate, $L = 1.2\text{m}$

Width of absorber plate, $W = 0.8\text{m}$

Absorptivity of front side, $(\alpha_s) = 0.87$

Emissivity of the plate, $\varepsilon = 0.09$

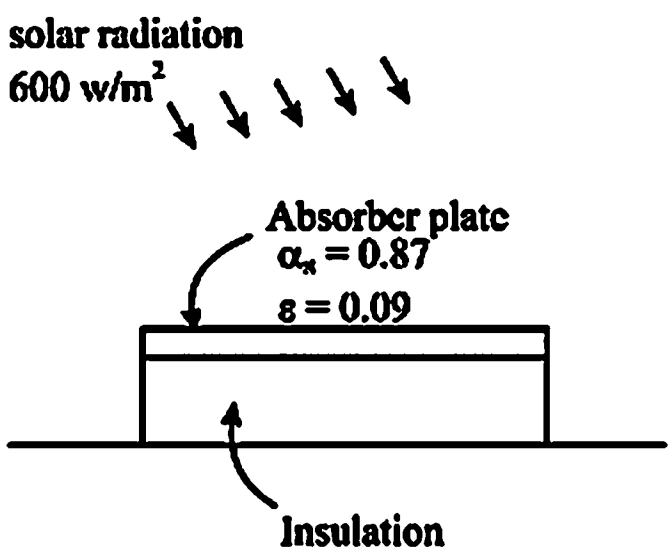
Ambient air temperature, $T_\infty = 25°\text{C}$

Solar radiation rate, $\dot{q} = 600\,\text{w/m}^2$

Effect sky temperature, $(T_{amb}) = 10°\text{C}$

Step 2 of 14

Schematic representation of the given system,



Step 3 of 14

Let us assume the surface temperature of the plate is $T_s = 100°\text{C}$

Therefore mean film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= 62.5°\text{C}$$

Step 4 of 14

From the ideal gas properties of the air 1 atm pressure,

At $T_f = 62.5°\text{C}$

By Using the Method of Interpolation

Thermal conductivity,

$$k = 0.02808 + \frac{0.02881 - 0.02808}{(70 - 60)}(62.5 - 60)$$

$$= 0.0282625\,\text{w/m}\cdot°\text{C}$$

Prandtl number,

$$\text{Pr} = 0.7202 + \frac{0.7177 - 0.7202}{(70 - 60)}(62.5 - 60)$$

$$= 0.719575$$

Kinematic viscosity,

$$\nu = 1.896 \times 10^{-5} + \frac{(1.995 - 1.896)10^{-5}}{(70 - 60)} \times (62.5 - 60)$$

$$= 1.92075 \times 10^{-5}\,\text{m}^2/\text{s}$$

Step 5 of 14

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= 2.9806 \times 10^{-3}\,\text{K}^{-1}$$

Characterize length = length of the plate

$$= A_s / p$$

where

p = Perimeter of plate

A_s = Area of plate

Therefore characteristic length,

$$L_c = \frac{1.2 \times 0.8}{2(1.2 + 0.8)}$$

$$= 0.24\,\text{m}$$

Step 6 of 14

Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \times \text{Pr}$$

$$9.81 \times 2.9806 \times 10^{-3}(100 - 25)(0.24)^3$$

Rayleigh number,

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \times \mathrm{Pr}$$

$$= \frac{9.81 \times 2.9806 \times 10^{-3}(100-25)(0.24)^3}{\left(1.92075 \times 10^{-5}\right)^2}$$

$$= 5.9129 \times 10^7$$

From the empirical correlation for the average Nusselt number for natural convention over surfaces

For horizontal plate of upper surface

Nusselt number for range $\left(10^7 - 10^{11}\right)$

$$\mathrm{Nu} = 0.15\left(\mathrm{Ra}_L\right)^{1/3}$$

$$= 0.15 \times \left(5.9129 \times 10^7\right)^{1/3}$$

$$= 58.44$$

Step 7 of 14

Heat transfer co-efficient,

$$h = \frac{k}{L_c} \times Nu$$

$$= \frac{0.028265}{0.24} \times 58.44$$

$$= 6.882 \mathrm{w/m^2 \cdot ^\circ C}$$

Surface area of the plate,

$$A_s = 0.12 \times 0.8$$

$$= 0.96 \mathrm{m^2}$$

Step 8 of 14

Heat again = heat lost by convection and radiation

Heat gain by solar radiation,

$$\dot{Q} = \alpha_s \cdot \dot{q} \cdot A_s$$

$$= 0.87 \times 600 \times 0.96$$

$$\dot{Q} = 501.12 \mathrm{w}$$

Step 9 of 14

To calculate temperature (T_s), we will use heat transfer equation

$$Q_{total} = h A_s (T_s - T_\infty) + \varepsilon A \sigma \left(T_s^4 - T_{Amb}^4\right)$$

$$501.12 = hA(T_s - T_\infty) + \sigma A E \left(T_s^4 - T_{amb}^4\right)$$

$$= 6.882 \times 0.96 (T_s - 25) + 5.67 \times 10^{-8} \times 0.96 \times 0.09$$

$$\left((T_s + 273)^4 283^4\right)$$

$$501.12 = 6.607(T_s - 25) + 4.9 \times 10^{-9}\left((T_s + 273)^4 - (283)^4\right)$$

$$T_s = 92.38^\circ \mathrm{C}$$

Step 10 of 14

If solar absorptivity, $\alpha_s = 0.28$

And emissivity, $\varepsilon = 0.07$

Then heat gain by solar radiation,

$$\dot{Q} = \alpha_s \cdot \dot{q} \cdot A_s$$

$$= 0.28 \times 600 \times 0.96$$

$$= 161.28 \mathrm{W}$$

Step 11 of 14

On substituting the known values in the above equation Q_{total}, we get

$$161.28 = 6.882 \times 0.96 (T_s - 25) + 5.67 \times 10^{-8}$$

$$\times 0.96 \times 0.07\left((T_s + 273)^4 - (283)^4\right)$$

$$161.28 = 6.6067(T_s - 25) + 3.81024 \times 10^{-9}\left((T_s + 273)^4 - (283)^4\right)$$

$$T_s = 47.058^\circ \mathrm{C}$$

This is not close to assumed one i.e. 100°C

Step 12 of 14

We need to assume T_s again for calculation near 50 say.

Assuming the surface temperature of the plate, $T_s = 55^\circ$

Mean film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= 40^\circ \mathrm{C}$$

Thermal conductivity, $k = 0.02662 \mathrm{W/m \cdot ^\circ C}$

Prandtl number, $\mathrm{Pr} = 0.7255$

Kinematic viscosity, $\nu = 1.702 \times 10^{-5} \mathrm{m^2/s}$

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= 3.1949 \times 10^{-3} \mathrm{K^{-1}}$$

Step 13 of 14

Nusselt, Rayleigh numbers and heat transfer co-efficient, h

$$\mathrm{Ra}_L = \frac{9.81 \times 3.1949 \times 10^{-3}(55-25)(0.24)^3}{\left(1.702 \times 10^{-5}\right)^2} \times 0.7255$$

$$= 3.255 \times 10^{-7}$$

Nusselt number,

$$\mathrm{Nu} = 0.15\left(3.255 \times 10^7\right)^{1/3}$$

$$= 47.895$$

$= 3.255 \times 10^{-7}$

Nusselt number,

$$Nu = 0.15\left(3.255 \times 10^{7}\right)^{1/3}$$

$$= 47.895$$

Heat transfer co-efficient,

$$h = \frac{0.02662}{0.24} \times 47.895$$

$$= 5.312\ W/m^2 \cdot K$$

Step 14 of 14

Now putting the values in the heat transfer rate equation

$$\dot{Q} = h A_s (T_s - T_\infty) + E A_s \sigma \left(T_s^4 - T_{amb}^4\right)$$

$$161.28 = 5.312 \times 0.96 (T_s - 25) + 5.67 \times 10^{-8} \times 0.96 \times 0.07 \left((T_s + 273)^4 - (283)^4\right)$$

$$161.28 = 5.0999 (T_s - 25) + 3.31024 \times 10^{-9} \left((T_s + 273)^4 - (283)^4\right)$$

$$T_s = 53.418^\circ C$$

Therefore the surface temperature of the plate, $\boxed{T_s = 53.418^\circ C}$

Question:

Repeat Prob. 20-31 for an aluminum plate painted flat black (solar absorptivity 0.98 and emissivity 0.98) and also for a plate painted white (solar absorptivity 0.26 and emissivity 0.90).

Step 1 of 12

Let us assume the surface temperature of the plate is $T_s = 70°C$

Then,

Mean film temperature is

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{70+25}{2}$$
$$= 47.5°C$$

Step 2 of 12

From the ideal gas properties of the air 1 atm pressure,

At $T_f = 47.5°C$

Calculate the thermal conductivity of the system,

$$k = 0.2699 + \frac{0.02735 - 0.02699}{(50-45)}(47.5-45)$$
$$= 0.02717\ W/m\cdot°C$$

Step 3 of 12

Calculate the Prandtl number,

$$Pr = 0.7241 + \frac{0.7228 - 0.7241}{(50-45)}(47.5-45)$$
$$= 0.7235$$

Calculate the Kinematic viscosity,

$$\nu = 1.750\times10^{-5} + \frac{(1.798-1.750)10^{-5}}{(50-45)}\times(47.5-45)$$
$$= 1.774\times10^{-5}\ m^2/s$$

Step 4 of 12

Calculate the Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

Where,

T_f is the mean film temperature

Substitute the respective values in the above equation,

$$\beta = \frac{1}{(47.5+273)}$$
$$= 3.1201\times10^{-3}\ k^{-1}$$

Step 5 of 12

Characterize length = length of the plate

$$L_c = \frac{A_s}{p}$$

Where as,

p = Perimeter of plate

A_s = Area of plate

$$L_c = \frac{1.2\times0.8}{2(1.2+0.8)}$$
$$= 0.24\ m$$

Step 6 of 12

Calculate the Rayleigh number,

$$Ra_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}Pr$$

Where,

T_∞ is Ambient air temperature and

g is acceleration due to gravity.

Substitute the respective values in the above equation,

$$Ra_L = \frac{9.81\times3.1201\times10^{-3}(70-25)(0.24)^3}{\left(1.774\times10^{-5}\right)^2}\times0.7235$$
$$= 4.379\times10^7$$

Step 7 of 12

For horizontal plate of upper surface, the Nusselt number,

$$Nu = 0.15\left(Ra_L\right)^{1/4}$$
$$= 0.54\times\left(4.379\times10^7\right)^{1/4}$$
$$= 43.93$$

Step 8 of 12

Calculate the Heat transfer co-efficient,

$$h = \frac{k}{L_c}\times Nu$$
$$= \frac{0.02717}{0.24}\times43.93$$
$$= 4.973\ W/m^2\cdot°C$$

Calculate the Surface area of the plate,

$$A_s = 0.12\times0.8$$
$$= 0.96\ m^2$$

Step 9 of 12

$A_s = 0.12 \times 0.8$

$= 0.96\text{m}^2$

Step 9 of 12

As it's a steady operation,
Heat gain = heat lost by convection and radiation
Heat gain by solar radiation

$$\dot{Q} = \alpha_s \dot{q} A_s$$

Where,
$\dot{q}$ is the Solar radiation rate and
α_s is the Absorbivity of front side

Substitute $\left(600\text{w/m}^2\right)$ for $\dot{q}$, 0.98 for α_s and the respective values in the above equation,

$$\dot{Q} = 0.98 \times 600 \times 0.96$$
$$= 564.48\text{W}$$

Step 10 of 12

To calculate temperature (T_s), we can use heat transfer equation,

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A \sigma\left(T_s^4 - T_{amb}^4\right)$$

Where,
T_{amb} is the Effect sky temperature,
ε is the Emissivity of the plate,

Substitute 10°C for T_{amb}, 0.98 for ε and the respective values in the above equation,

$$564.48 = hA(T_s - T_\infty) + \sigma AE\left(T_s^4 - T_{amb}^4\right)$$
$$= 4.973 \times 0.96(T_s - 25) + 5.67 \times 10^{-8} \times 0.96 \times 0.98\left[(T_s + 273)^4 - (10 + 273)^4\right]$$
$$T_s = 66.4886°\text{C}$$

The temperature of the absorber plate is close to the assumed value,
Therefore, the temperature of the black colored absorber plate is $\boxed{66.4886°\text{C}}$.

Step 11 of 12

If the plate is painted with white color, then the solar absorptivity is $\alpha_s = 0.26$ and emissivity is $\varepsilon = 0.9$
Then heat gain by solar radiation,

$$\dot{Q} = \alpha_s \dot{q} A_s$$

Substitute the respective values in the above equation,

$$\dot{Q} = 0.26 \times 600 \times 0.96$$
$$= 149.76\text{W}$$

Step 12 of 12

To calculate temperature (T_s), we can use heat transfer equation,

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A \sigma\left(T_s^4 - T_{amb}^4\right)$$
$$149.76 = 4.973 \times 0.96 \times (T_s - 25) + 5.67 \times 10^{-8} \times 0.96 \times 0.9 \times \left[(T_s + 273)^4 - (10 + 273)^4\right]$$
$$T_s = 32.6411°\text{C}$$

Therefore, the temperature of the white colored absorber plate is $\boxed{32.6411°\text{C}}$.

Question: 20.033P

A room is to be heated by a coal-burning stove, which is a cylindrical cavity with an outer diameter of 50 cm and a height of 120 cm. The rate of heat loss from the room is estimated to be 1.5 kW when the air temperature in the room is maintained constant at 24°C. The emissivity of the stove surface is 0.85, and the average temperature of the surrounding wall surfaces is 14°C. Determine the surface temperature of the stove. Neglect the heat transfer from the bottom surface and take the heat transfer coefficient at the top surface to be the same as that on the side surface.

The heating value of the coal is 30,000 kJ/kg, and the combustion efficiency is 65 percent. Determine the amount of coal burned a day if the stove operates 14 h a day.

Step 1 of 9

Given data:

Outer diameter of stove, $D = 50\,\text{cm} = 0.5\,\text{m}$

Height of stove, $L = 120\,\text{cm} = 1.2\,\text{m}$

Air temperature in room, $T_\infty = 24°\text{C}$

Surrounding wall temperature, $T_{amb} = 14°\text{C}$

Emissivity of stove surface, $\varepsilon = 0.85$

Heating value of coal $= 30000\,\text{kJ/kg}$

Combustion efficiency, $\eta = 65\% = 0.65$

Operations hours $= 14$

Rate of heat loss, $Q_{total} = 1.5\text{kW}$

Step 2 of 9

Let us assume the surface temperature of the cylinder, $T_s = 76°\text{C}$

Then mean film temperature,

$$T_f = \frac{(T_s + T_\infty)}{2}$$

$$= \frac{(76+24)}{2}$$

$$= 50°\text{C}$$

From the properties of air at 1 atm pressure table, the properties of air at 50°C,

Thermal conductivity, $k = 0.02735\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7228$

Step 3 of 9

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{(273+50)}$$

$$= 3.0960\times10^{-3}\,\text{k}^{-1}$$

Characteristic length = height of vertical cylinder

$$L_c = L$$

$$= 1.2\,\text{m}$$

Step 4 of 9

Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\times\text{Pr}$$

$$= \frac{9.81\times3.096\times10^{-3}(76-24)(1.2)^3}{\left(1.793\times10^{-5}\right)^2}\times0.7228$$

$$= 6.101\times10^9$$

Step 5 of 9

By using the condition for checking if we can treat cylinder as a vertical plate or not, we have

$$D \geq \frac{35L}{\text{Gr}_L^{0.25}}$$

$$\geq \frac{35L}{\left(\frac{\text{Ra}_L}{\text{Pr}}\right)^{0.25}} \qquad \left(\text{since, } \text{Gr}_L = \frac{\text{Ra}_L}{\text{Pr}}\right)$$

$$\geq \frac{35L}{\left(\frac{6.101\times10^9}{0.7228}\right)^{0.25}}$$

$$\geq \frac{35\times1.2}{303.115}$$

$$D \geq 0.13856$$

Given that, $D = 0.5\text{m}$

As it is satisfying the equation, we will treat the outer curvature as a vertical plate.

Step 6 of 9

By using empirical correlations for the average Nusselt number for vertical plate, we get

$$\text{Nu} = \left[0.825 + \frac{0.387\left(R_{a_L}\right)^{1/6}}{\left[1+\left(\frac{0.492}{p_r}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.825 + \frac{0.387\left(6.101\times10^9\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.7228}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= 216.42$$

Step 7 of 9

Using formula for heat transfer co-efficient, we get

$$h = \frac{k}{L_c}\text{Nu}$$

$$= \frac{0.02735}{1.2}\times216.42$$

$$= 4.9326\,\text{W/m}^2\cdot\text{k}$$

Step 8 of 9

By using heat loss rate equation (considering both convection and radiation), we get

$$Q_{total} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma\left(T_s^4 - T_{amb}^4\right)$$

$$1500 = 4.9326\times2.081\times(T_s - 24) + 0.85\times2.081$$

$$\times5.67\times10^{-8}\times\left((T_s+273)^4 - (287)^4\right)$$

$$1500 = 10.265(T_s - 24) + 1.003\times10^{-7}\left((T_s+273)^4 - (287)^4\right)$$

$$T_s = 81.72°\text{C}$$

Therefore, surface temperature of stove is $\boxed{T_s = 81.72°\text{C}}$.

Step 9 of 9

Total heat production $= 1.5\text{kW} = 1.5\text{kJ/s}$

14 operating hours per day is given

Therefore,

Amount of heat produced,

$$Q = 1.5\times14\times3600$$

$$Q = 75600\,\text{kJ}$$

Now heat value of coal $= 30000\,\text{kJ/kg}$

By using the formula for coal consumed, we get

$$m = \frac{Q/\eta}{\text{heating value}}$$

$$= \frac{75600/0.65}{30000}$$

$$= 3.88\,\text{kg}$$

Therefore, amount of cola burned per day is $\boxed{m = 3.88\,\text{kg}}$.

Question:

Thick fluids such as asphalt and waxes and the pipes in which they flow are often heated in order to reduce the viscosity of the fluids and thus to reduce the pumping costs. Consider the flow of such a fluid through a 100-m-long pipe of outer diameter 30 cm in calm ambient air at 0°C. The pipe is heated electrically, and a thermostat keeps the outer surface temperature of the pipe constant at 25°C. The emissivity of the outer surface of the pipe is 0.8, and the effective sky temperature is –30°C. Determine the power rating of the electric resistance heater, in kW, that needs to be used. Also, determine the cost of electricity associated with heating the pipe during a 15-h period under the above conditions if the price of electricity is $0.09/kWh.

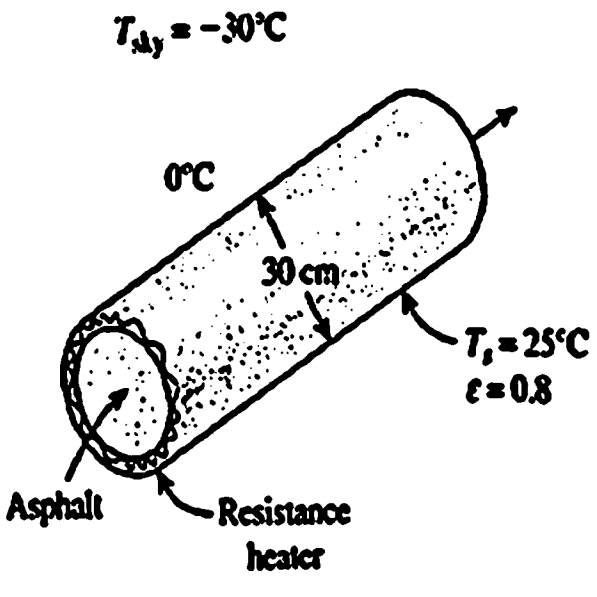


FIGURE P20–34

Step 1 of 9

Given data:

Length of the pipe, $L = 100\,\text{m}$

Diameter of the pipe, $D = 30\,\text{cm} = 0.3\text{m}$

Ambient air, $T_\infty = 0°C$

Surface temperature of pipe, $T_s = 25°C$

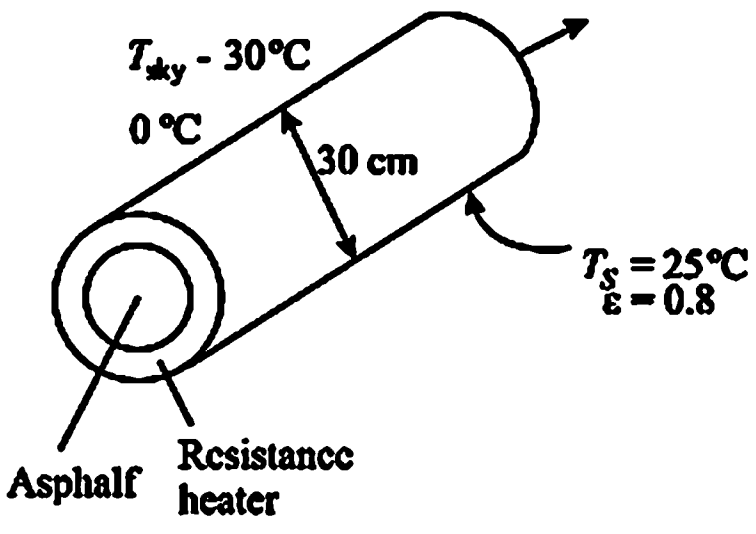
Emissivity, $\varepsilon = 0.8$

Effective sky temperature, $T_{amb} = -30°C$

Price of electricity $= \$0.09/\text{kWh}$

Step 2 of 9

The pipe is as shown in figure,



Step 3 of 9

By using Mean film temperature formula, we get

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= 12.5°C$$

From the properties of air at mean film temperature and 1 atm pressure by using the method of interpolation,

The thermal conductivity,

$$k = (0.02476 + 0.02439)/2$$

$$= 0.024575\,\text{W/m}\cdot\text{K}$$

The Prandtl number,

$$\text{Pr} = (0.7323 + 0.7336)/2$$

$$= 0.73295$$

Kinematic viscosity

$$\nu = (1.470 + 1.426)/2 \times 10^{-5}$$

$$= 1.448 \times 10^{-5}\,\text{m}^2/\text{s}$$

Step 4 of 9

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{285.5}$$

$$= 3.5026 \times 10^{-3}\,\text{K}^{-1}$$

Characteristic length = diameter of pipe

$$L_c = D$$

$$= 0.3\text{m}$$

Step 5 of 9

Now, Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \times \text{Pr}$$

$$= \frac{9.81 \times 3.5026 \times 10^{-3}(25-0)(0.3)^3}{(1.448 \times 10^{-5})^2} \times (0.73295)$$

$$= 8.108 \times 10^7$$

Step 6 of 9

By using Nusselt number formula, we get

$$\text{Nu} = \left[0.6 + \frac{0.387(\text{Ra}_L)^{1/6}}{\left[1 + \left(\frac{0.559}{\text{Pr}}\right)^{9/16}\right]^{8/27}} \right]^2$$

$$= \left[0.6 + \frac{0.387(8.108 \times 10^7)^{1/6}}{\left[1 + \left(\frac{0.559}{0.73295}\right)^{9/16}\right]^{8/27}} \right]^2$$

$$\left[1+\left(\frac{0.559}{\mathrm{Pr}}\right)^{9/16}\right]^{20}\Bigg] \quad \text{034P pg 2}$$

$$= \left[0.6+\frac{0.387\left(8.108\times10^{7}\right)^{1/6}}{\left[1+\left(\frac{0.559}{0.73295}\right)^{9/16}\right]^{8/27}}\right]^{2}$$

$$= 53.296$$

Step 7 of 9

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\mathrm{N_u}$$

$$= \frac{0.024575}{0.3}\times(53.296)$$

$$= 4.366\,\mathrm{W/m^2\cdot K}$$

Surface area of the plate,

$$A_s = \pi\Delta L$$

$$= \pi\times0.3\times100$$

$$= 94.248\,\mathrm{m^2}$$

Step 8 of 9

Power rating of electric resistance heater = heat loss rate from cylinder by convection and radiation

$$\dot{Q} = hA_s\left(T_s - T_\infty\right) + \varepsilon A_s\sigma\left(T_s^4 - T_{amb}^4\right)$$

$$= 4.366\times94.28\times(25-0) + 0.8\times94.248 \times5.67\times10^{-8}\left(298^4 - 243^4\right)$$

$$= 29098.344\,\mathrm{W}$$

Therefore, power rating of the electrical resistance is $\boxed{\dot{Q} = 29.098\,\mathrm{kW}}$.

Step 9 of 9

During 15 hours period heat loss will be

$$Q = \dot{Q}\times\text{number of hours}$$

$$= 29.098\times15$$

$$Q = 436.48\ \mathrm{kWh}$$

The cost of electricity,

$$\text{cost} = Q\times\text{cost/hour}$$

$$= 436.48\times0.09$$

$$= \$39.282$$

Therefore, cost of the electricity is $\boxed{\text{cost} = \$39.282}$

Question:

Reconsider Prob. 20-34. To reduce the heating cost I of the pipe, it is proposed to insulate it with sufficiently thick fiberglass insulation (k= 0.035 W/m·K) wrapped with aluminum foil (ε = 0.1) to cut down the heat losses by 85 percent. Assuming the pipe temperature to remain constant at 25°C, determine the thickness of the insulation that needs to be used. How much money will the insulation save during this 15-h period?

Step 1 of 13

Loading solution. Please wait ...

Step 2 of 13

Loading solution. Please wait ...

Step 3 of 13

From the properties of air at 1 atm pressure table, the properties of air at 12.5°C by using interpolation,

Thermal conductivity,

$$\frac{12.5-10}{15-10}=\frac{k-0.02439}{0.02476-0.02439}$$

$$k=0.02458\ \text{W/m}\cdot\text{K}$$

Kinematic viscosity,

$$\frac{12.5-10}{15-10}=\frac{v-1.426\times10^{-5}}{1.47\times10^{-5}-1.426\times10^{-5}}$$

$$v=1.448\times10^{-5}\ \text{m/s}$$

Step 4 of 13

Prandtl number,

$$\frac{12.5-10}{15-10}=\frac{\text{Pr}-0.7336}{0.7326-0.7336}$$

$$\text{Pr}=0.7331$$

Rayleigh's number,

$$\text{Ra}=\frac{g\beta(T_s-T_\infty)L_c^3}{v^2}\times\text{Pr}$$

$$=\frac{9.81\times0.003503\times(25-0)\times0.3^3}{(1.448\times10^{-5})^2}\times0.7330$$

$$=8.106\times10^7$$

Step 5 of 13

$$\text{Pr}=0.7331$$

Rayleigh's number,

$$\text{Ra}=\frac{g\beta(T_s-T_\infty)L_c^3}{v^2}\times\text{Pr}$$

$$=\frac{9.81\times0.003503\times(25-0)\times0.3^3}{(1.448\times10^{-5})^2}\times0.7330$$

$$=8.106\times10^7$$

Step 5 of 13

Nusselt number,

$$\text{Nu}=\left\{0.6+\frac{0.387\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

$$=\left\{0.6+\frac{0.387(8.106\times10^7)^{1/6}}{\left[1+(0.559/0.7330)^{9/16}\right]^{8/27}}\right\}^2$$

$$=53.29$$

But $\text{Nu}=\frac{hL_c}{k}$, so heat transfer coefficient is

$$h=\frac{k}{L_c}\text{Nu}$$

$$=\frac{0.02458}{0.3\ \text{m}}\times(53.29)$$

$$=4.366\ \text{W/m}^2\cdot\text{K}$$

Step 6 of 13

Calculate the Convection heat transfer rate,

$$\dot{Q}_{conv}=hA_s(T_s-T_\infty)$$

$$=4.366\times94.25\times(25-0)$$

$$=10{,}287\ \text{W}$$

Calculate the Radiation heat transfer rate,

$$\dot{Q}_{rad}=\varepsilon A_s\sigma\left(T_s^4-T_{sur}^4\right)$$

Where,

ε is the Emissivity of the outer surface of pipe,

T_{sur} is the Surrounding temperature,

Step 7 of 13

Substitute $T_{sur}=-30°\text{C}$, $\varepsilon=0.8$ and the respective values in the above equation,

$$\dot{Q}_{rad}=0.8\times94.25\times5.67\times10^{-8}\times[(25+273)^4-(-30+273)^4]$$

$$=18{,}808\ \text{W}$$

Calculate the total hear transfer rate,

$$\dot{Q}_{total}=\dot{Q}_{conv}+\dot{Q}_{rad}$$

$$=10{,}287+18{,}808$$

$$=29.094\ \text{kW}$$

Step 8 of 13

After insulation, the heat losses gets reduced by 85%, so

$$\dot{Q}=(1-0.85)\times\dot{Q}_{\text{no insulation}}$$

Step 8 of 13

After insulation, the heat losses gets reduced by 85%, so

$$\dot{Q} = (1-0.85)\times \dot{Q}_{\text{no insulation}}$$
$$= 0.15\times 29,094$$
$$= 4364 \text{ W}$$

Calculate the average film temperature,

$$T_f = 5^\circ\text{C}$$
$$= 5+273$$
$$= 278^\circ\text{K}$$

Step 9 of 13

From properties of air table, the properties of air at 5°C are

$k = 0.02401 \text{ W/m}\cdot\text{K}$

$\upsilon = 1.382\times 10^{-5} \text{ m}^2/\text{s}$

$\text{Pr} = 0.7350$

$$\beta = \frac{1}{T_f} = 0.003597 \text{ K}^{-1}$$

Calculate the Surface area of the pipe,

$$A_s = \pi D_0 L$$
$$= \pi(0.3+2t_{ins})\times 100$$

Calculate the characteristic length after insulation,

$$L_c = D+2t_{ins}$$
$$= 0.3+2t_{ins}$$

Step 10 of 13

Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s-T_\infty)L_c^3}{\upsilon^2}\text{Pr}$$
$$= \frac{9.81\times 0.003597\times[(T_s-273)](0.3+2t_{ins})^3}{(1.382\times 10^{-5})^2}(0.7350)$$

Nusselt number,

$$\text{Nu} = \left\{0.6+\frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$
$$= \left\{0.6+\frac{0.387Ra^{1/6}}{\left[1+(0.559/0.7350)^{9/16}\right]^{8/27}}\right\}^2$$

Step 11 of 13

But $\text{Nu} = \frac{hL_c}{k}$ so heat transfer coefficient is

$$h = \frac{k}{L_c}\text{Nu}$$
$$= \frac{0.02401}{L_c}\text{Nu}$$

Calculate the total hear transfer rate,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}$$
$$= hA_s(T_s-T_\infty)+\varepsilon A_s\sigma(T_s^4-T_{\text{sur}}^4)$$
$$4364 = \{h\times A_s\times(T_s-273)\}+\{0.1\times A_s\times 5.67\times 10^{-8}\times[T_s^4-(-30+273)^4]\} \quad \ldots\ldots (1)$$

Step 12 of 13

In steady state operation, the heat lost by the side surfaces of the pipe must be equal to the heat lost from the exposed surface of the insulation by convection and radiation, therefore,

$$\dot{Q}_{\text{ins}} = \frac{2\pi kL(T_{\text{sink}}-T_s)}{\ln(D_o/D)}$$
$$4364 = \frac{2\pi\times 0.035\times 100\times(298-T_s)}{\ln\left(\frac{0.3+2t_{\text{ins}}}{0.3}\right)} \quad \ldots\ldots (2)$$

Solve equations (1) and (2) by using equation solver, we get

$T_s = 281.5 \text{ K} = 8.5^\circ\text{C}$

$t_{\text{ins}} = 1.3 \text{ cm}$

Therefore, the final insulation has thickness is $\boxed{t_{\text{ins}} = 1.3 \text{ cm}}$

Step 13 of 13

Calculate the amount money insulation will save during a 10 hr period,

$$Q_{\text{saved, total}} = \dot{Q}_{\text{saved}}\Delta t$$
$$= (0.85\times 29.094 \text{ kW})(15 \text{ h})$$
$$= 370.948 \text{ kWh}$$

Money saved = (Energy saved)(Unit cost of energy)

$$= (370.948 \text{ kWh})(\$0.09/\text{kWh})$$
$$= \$33.385$$

Therefore, the total money saved due to insulation is $\boxed{\text{Money saved} = \$33.385}$

Question: 20.036P pg 1

Consider a 1.2-m-high and 2-m-wide glass window with a thickness of 6 mm, thermal conductivity k =0.78 W/m-K, and emissivity ε = 0.9. The room and the walls that face the window are maintained at 25°C, and the average temperature of the inner surface of the window is measured to be 5°C. If the temperature of the outdoors is –5°C, determine (*a*) the convection heat transfer coefficient on the inner surface of the window, (*b*) the rate of total heat transfer through the window, and (*c*) the combined natural convection and radiation heat transfer coefficient on the outer surface of the window. Is it reasonable to neglect the thermal resistance of the glass in this case?

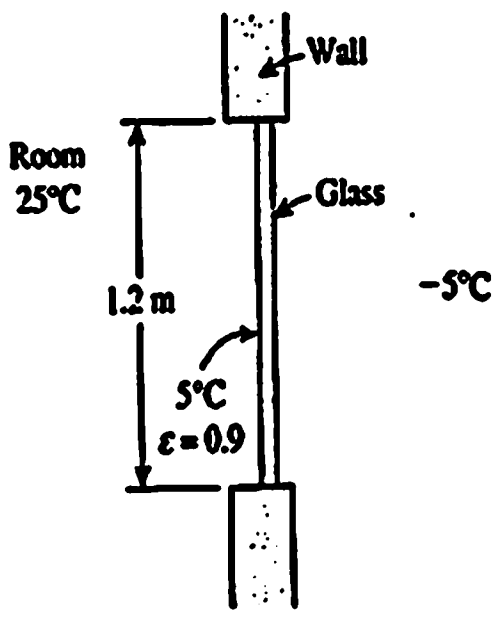


FIGURE P20–36

Step 1 of 12

Find the average film temperature.

$$T_f = \frac{T_s + T_\infty}{2}$$

Here, T_s is average inner surface temperature and T_∞ is ambient temperature.

Step 2 of 12

Step 3 of 12

Substitute $5°C$ for T_s and $25°C$ for T_∞.

$$T_f = \frac{5+25}{2}$$
$$= 15°C$$
$$= 288\,K$$

Obtain the following properties of the air from the table A-22 "properties of air at 1 atm pressure "at $15°C$.

Step 4 of 12

Thermal concavity, $k = 0.02476\ W/m°C$

Kinematic viscosity, $\upsilon = 1.471\times10^{-5}\ m^2/s$

Prandit number, $Pr = 0.7323$

Step 5 of 12

a)

Find the co-efficient of volume expansion.

$$\beta = \frac{1}{T_f}$$

Substitute $288\,K$ for T_f.

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{288}$$
$$= 0.003472\ K^{-1}$$

Find the surface area of the window.

$$A_s = (1.2\ m \times 2\ m)$$
$$= 2.4\ m^2$$

Find the characteristic length.

$$L_c = L$$
$$= 1.2\ m$$

Step 6 of 12

Find the Rayleigh number using the equation.

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2} Pr$$

Substitute $0.003472\ K^{-1}$ for β, $9.81\ m/s^2$ for g, 0.7323 for Pr, $1.2\ m$ for L, $1.471\times10^{-5}\ m^2/s$ for υ, $5°C$ for T_s and $25°C$ for T_∞.

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2} Pr$$
$$= \frac{9.81\times0.003472\times(25-5)\times1.2^3}{(1.417\times10^{-5})^2}\times(0.7323)$$
$$= 3.986\times10^9$$

Find the Nusselt number using the number.

$$Nu = \left\{0.825 + \frac{0.387Ra^{1/6}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu=\left\{0.825+\frac{0.387Ra^{1/6}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{8/27}}\right\}$$

Substitute 0.7323 for Pr, and 3.986×10^9 for Ra.

$$Nu=\left\{0.825+\frac{0.387\times\left(3.986\times10^9\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.7323}\right)^{9/16}\right]^{8/27}}\right\}^2$$
$$=189.7$$

Step 7 of 12

Find the heat transfer coefficient of inner surface using the equation.

$$Nu=\frac{hL_c}{k}$$

$$h=\frac{k}{L_c}Nu$$

Substitute $0.02476\ W/m°C$ for k, $1.2\ m$ for L_c and 189.7 for Nu.

$$h=\frac{k}{L_c}Nu$$
$$=\frac{0.02476}{1.2}\times189.7$$
$$=3.915\ W/m^2°C$$

Therefore, the heat transfer coefficient of inner surface is $\boxed{3.915\ W/m^2°C}$.

Step 8 of 12

b)

Find the convection heat transfer rate using the equation.

$$\dot{Q}_{conv}=hA_s(T_\infty-T_s)$$

Substitute $3.915\ W/m^2°C$ for h, $2.4\ m^2$ for A_s, $5°C$ for T_s and $25°C$ for T_∞.

$$\dot{Q}_{conv}=3.915\times2.4\times(25-5)$$
$$=187.9\ W$$

Find the radiation heat transfer rate using the equation.

$$\dot{Q}_{rad}=\varepsilon A_s\sigma\left(T_{surr}^4-T_s^4\right)$$

Here. ε is emissivity and σ is the Stefan-Boltzmann constant.

Substitute

$$\dot{Q}_{rad}=0.9\times2.4\times5.67\times10^{-8}\times\left[(25+273)^4-(5+273)^4\right]$$
$$=234.3\ W$$

Find the total heat transfer rate using the equation.

$$\dot{Q}_{total}=\dot{Q}_{conv}+\dot{Q}_{rad}$$

Substitute $234.3\ W$ for $\dot{Q}_{conv}$ and 234.3 W for $\dot{Q}_{rad}$.

$$\dot{Q}_{total}=187.9+234.3$$
$$=422.2\ W$$

Therefore, the rate of heat transfer through window is $\boxed{422.2\ W}$.

Step 9 of 12

c)

Find the temperature using conduction heat transfer through the window glass.

$$\dot{Q}_{cond}=\frac{k_gA_s}{t}\left(T_{s,i}-T_{s,o}\right)$$

$$T_{s,o}=T_{s,i}-\left(\frac{\dot{Q}_{total}t}{k_gA_s}\right)$$

Here, t is thickness and k_g is the thermal conductivity of glass.

Substitute $422.2\ W$ for $\dot{Q}_{total}$, $2.4\ m^2$ for A_s, $5°C$ for $T_{s,i}$, $0.78\ W/m°C$ for k_g, and 0.006 m for t.

$$T_{s,o}=T_{s,i}-\left(\frac{\dot{Q}_{total}t}{k_gA_s}\right)$$
$$=5-\left(\frac{422.2\times0.006}{0.78\times2.4}\right)$$
$$=3.65°C$$

Step 10 of 12

Find the heat transfer due to conduction is equal to the heat transfer due to combined effect of convection and radiation.

$$\dot{Q}_{cond}=h_{combined}A_s\left(T_{s,o}-T_{\infty,o}\right)$$

$$h_{combined}=\frac{\dot{Q}_{total}}{A_s\left(T_{s,o}-T_{\infty,o}\right)}$$

$$h_{combined}=\frac{\dot{Q}_{total}}{A_s(T_{s,o}-T_{\infty,o})}$$

$$h_{combined}=\frac{346}{2.4\times[3.65-(-5)]}$$

$$h_{combined}=20.35\ W/m^2°C$$

Hence coefficient of heat transfer due to combined effect of radiation and convection is

$$h_{combined} = \frac{346}{2.4\times[3.65-(-5)]}$$

20.036P pg 3

$$h_{combined} = 20.35\ \text{W/m}^2\text{°C}$$

Hence coefficient of heat transfer due to combined effect of radiation and convection is

$$\boxed{h_{combined} = 20.35\ \text{W/m}^2\text{°C}}$$

Step 11 of 12

Note that $\Delta T = \dot{Q}R$ and thus the thermal resistance R of a layer is proportional to the temperature drop across that layer. Therefore, the fraction of thermal resistance of the glass is equal to the ratio of the temperature drop across the glass to the overall temperature difference,

$$\frac{R_{glass}}{R_{total}} = \frac{\Delta T_{glass}}{\Delta T_{total}} = \frac{5-3.65}{25-(-5)} = 0.045 \quad \text{(or 4.5\%)}$$

Step 12 of 12

It is reasonable to neglect the thermal resistance of the glass.

Question: 20.037P

A 3-mm-diameter and 12-m-long electric wire is tightly wrapped with a 1.5-mm-thick plastic cover whose thermal conductivity and emissivity are k = 0.20 W/m·K and ε = 0.9. Electrical measurements indicate that a current of 10 A passes through the wire and there is a voltage drop of 7 V along the wire. If the insulated wire is exposed to calm atmospheric air at T_∞ = 30°C, determine the temperature at the interface of the wire and the plastic cover in steady operation. Take the surrounding surfaces to be at about the same temperature as the air.

Step 1 of 6

Given

Length of the electric wire, $L = 12\text{ m}$

Diameter of the electric wire, $d = 0.003\text{ m}$

Thickness of the electric wire, $t = 0.0015\text{ m}$

Emissivity of the plastic cover is $\varepsilon = 0.9$

Ambient temperature $T_\infty = 30^0\text{C}$

Step 2 of 6

Assuming the surface temperature as $T_s = 55^0\text{C}$

Average film temperature, $T_f = \dfrac{T_s + T_\infty}{2}$

$$T_f = \frac{55+30}{2}$$

$$T_f = 42.5^0\text{C}$$

$$T_f = 315.5^0\text{K}$$

Properties of the air at 42.5^0C

$k = 0.02892\text{ W/m.°C}$

$\upsilon = 1.752\times10^{-5}\text{ m}^2/s$

$\text{Pr} = 0.73$

$$\beta = \frac{1}{T_f} = 0.0031695\text{ K}^{-1}$$

Step 3 of 6

Rate of heat transfer, $\dot{Q} = VI$

$$\dot{Q} = (7\text{ V})(10\text{ A})$$

$$\dot{Q} = 70\text{ W}$$

Surface area of the electric wire, $A_s = \pi DL$

$$A_s = \pi(0.003\text{ m})(12\text{ m})$$

$$A_s = 0.1131\text{ m}^2$$

Characteristic length, $L_c = D = 0.003\text{ m}$

Rayleighs number, $Ra = \dfrac{g\beta(T_s - T_\infty)D^3}{\upsilon^2}\text{Pr}$

$$Ra = \frac{9.81\times0.0031695\times(55-30)\times0.003^3}{(1.752\times10^{-5})^2}(0.73)$$

$$Ra = 56.3888$$

Step 4 of 6

Nusselt number, $Nu = \left\{0.6 + \dfrac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$

$$Nu = \left\{0.6 + \frac{0.387(56.3888)^{1/6}}{\left[1+(0.559/0.73)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = 1.72$$

But $Nu = \dfrac{hL_c}{k}$ So heat transfer coefficient is

$$h = \frac{k}{D}Nu$$

$$h = \frac{0.02892}{0.003}(1.72)$$

$$h = 15.897\text{ W/m}^2\text{°C}$$

Step 5 of 6

Total heat transfer rate,

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$

$$70 = \{15.897\times0.1131\times(T_s - 30)\}$$

$$+\left\{0.9\times0.1131\times5.67\times10^{-8}\times[(T_s+273)^4 - (30+273)^4]\right\}$$

$$70 = \{1.79795\times(T_s - 30)\} + \left\{5.7715\times10^{-9}\times[(T_s+273)^4 - (303)^4]\right\}$$

$$T_s = 56.6°\text{C}$$

Step 6 of 6

Conduction heat transfer through the wire is

$$\dot{Q} = \frac{2\pi kL}{\ln(D_2/D_1)}(T_i - T_s)$$

$$T_i = T_s + \frac{\dot{Q}\ln(D_2/D_1)}{2\pi kL}$$

$$T_i = 56.6 + \frac{70\times\ln(3/1.5)}{2\pi\times0.2\times12}$$

$$T_i = 59.8^0\text{C}$$

Temperature at interface, $\boxed{T_i = 59.8^0\text{C}}$

Question:

During a visit to a plastic sheeting plant, it was observed that a 45-m-long section of a 2-in nominal (6.03-em-outer-diameter) steam pipe extended from one end of the plant to the other with no insulation on it. The temperature measurements at several locations revealed that the average temperature of the exposed surfaces of the steam pipe was 170°C, while the temperature of the surrounding air was 20°C. The outer surface of the pipe appeared to be oxidized, and its emissivity can be taken to be 0.7. Taking the temperature of the surrounding surfaces to be 20°C also, determine the rate of heat loss from the steam pipe.

Steam is generated in a gas furnace that has an efficiency of 84 percent, and the plant pays $1.10 per therm (1 therm = 105,500 kJ) of natural gas. The plant operates 24 h a day 365 days a year, and thus 8760 h a year. Determine the annual cost of the heat losses from the steam pipe for this facility.

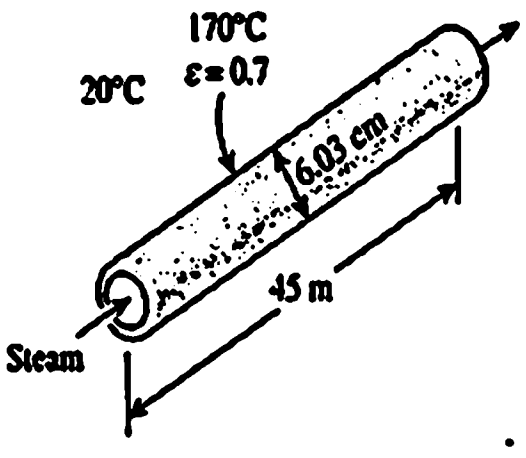


FIGURE P20–38

Step 1 of 11

Given data:

Length of steam pipe, $L = 45\text{m}$

Diameter of steam pipe, $D_0 = 6.03\text{cm} = 0.0603\text{m}$

Average surface temperature, $T_s = 170°\text{C}$

Air temperature, $T_\infty = 20°\text{C}$

Emissivity of surface, $\varepsilon = 0.7$

Surrounding surface Temperature, $T_{amb} = 20°\text{C}$

Efficiency of the gas furnace, $\eta_{gasfurnace} = 84\%$

Operations hours,

$t = 8760\,\text{hours}$

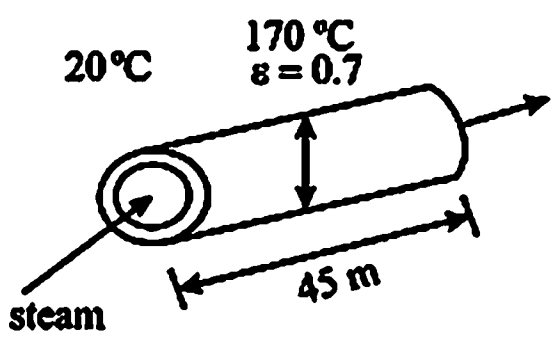
$= 8760 \times 3600$

$= 31536000\,\text{sec}$

Amount paid = $1.10 per therm

Step 2 of 11

The steam pipe is as shown in figure,



Step 3 of 11

Mean film temperature,

$$T_f = (T_s + T_\infty)/2$$

$$= (170+20)/2$$

$$= 95°\text{C}$$

Step 4 of 11

From the properties of the ideal gas of air at 1 atm pressure and mean film temperature,

Thermal conductivity,

$$k = (0.03095+0.03024)/2$$

$$= 0.030595$$

Prandtl number,

$$\text{Pr} = (0.7132+0.7111)/2$$

$$= 0.71215$$

Kinematic viscosity,

$$\nu = (2.201+2.306)/2\times10^{-5}$$

$$= 2.2535\times10^{-5}\,\text{m}^2/\text{s}$$

Step 5 of 11

Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{368}$$

$$= 2.7174\times10^{-3}\,\text{K}^{-1}$$

Characteristic length for horizontal = diameter of cylinder

$L_c = 0.0603\text{m}$

Step 6 of 11

Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\times\text{Pr}$$

$$= \frac{9.81\times2.7174\times10^{-3}(170-20)(0.0603)^3}{(2.254\times10^{-5})^2}\times0.71215$$

$$= 1.229\times10^6$$

Step 7 of 11

Nusselt number,

$$\text{Nu} = \left\{0.6 + \frac{0.387(R_{a_L})^{1/6}}{\left[\left(0.559\right)^{9/16}\right]^{8/27}}\right.$$

Nusselt number,

$$\mathrm{Nu}=\left[0.6+\frac{0.387\left(R_{a_L}\right)^{1/6}}{\left[1+\left(\frac{0.559}{P_r}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$=\left[0.6+\frac{0.387\left(1.229\times10^{6}\right)^{1/6}}{\left[1+\left(\frac{0.559}{0.71215}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$=15.412$$

Step 8 of 11

Heat transfer co-efficient,

$$h=\frac{k}{L_c}\mathrm{Nu}$$

$$=\frac{0.030595}{0.0603}\times15.412$$

$$=7.82\,\mathrm{W/m^2\cdot K}$$

Surface Area of the cylinder,

$$A_s=\pi DL$$

$$=\pi\times0.0603\times45$$

$$=8.525\,\mathrm{m^2}$$

Step 9 of 11

By using total heat losses formula, we get

$$\dot{Q}=hA_s\left(T_s-T_\infty\right)+\varepsilon A_s\sigma\left(T_s^4-T_{amb}^4\right)$$

$$=7.82\times8.525(170-20)+0.7\times8.525$$

$$\times5.67\times10^{-8}\left(443^4-293^4\right)$$

$$=20537.49\,\mathrm{W}$$

Therefore, the total heat loss from the pipe is $\boxed{\dot{Q}=20537.49\,\mathrm{W}}$.

Step 10 of 11

Total heat consumption from gas,

$$Q_{gas}=Q\times\text{operations hours}$$

$$=20537.49\times31536000$$

$$=6.47\times10^{11}\,\mathrm{J/s}$$

$$=6.47\times10^{8}\,\mathrm{kJ/s}$$

As efficiency is 84%

Therefore, actual heat transfer is

$$Q_{gas}=\frac{6.47\times10^{8}}{\eta_{\text{gas furnance}}}$$

$$=\frac{6.47\times10^{8}}{0.84}$$

$$=7.7\times10^{8}\,\mathrm{kJ/s}$$

Step 11 of 11

Note: $1\text{therm}=105500\,\mathrm{kJ}$

Let us convert into therms,

Therefore

$$Q_{gas}=\frac{7.7\times10^{8}}{105500}$$

$$=7298.57\,\text{therms}$$

$\text{cost/therm}=\$1.10$

Then, Total cost (annual) of heat loses

$$\text{Total cost}=\text{cost/therm}\times Q_{gas}$$

$$=1.10\times7298.57$$

$$=\$8028.427$$

Therefore, annual cost of heat losses is $\boxed{\text{total cost}=\$8028.427}$.

Question: 20.039P

Reconsider Prob. 20-38. Using EES (or other) software, investigate the effect of the surface temperature of the steam pipe on the rate of heat loss from the pipe and the annual cost of this heat loss. Let the surface temperature vary from 100°C to 200°C. Plot the rate of heat loss and the annual cost as a function of the surface temperature, and discuss the results.

Step 1 of 5

```
"GIVEN"
L=45 "[m]"
D=0.0603 "[m]"
"T_s=170 [C], parameter to be varied from 100[C] to 200[C]"
T_infinity=20 "[C]"
epsilon=0.7
T_surr=T_infinity
eta_furnace=0.84 "Efficiency"
UnitCost=1.1 "[$/therm]"
time=8760 "[hr]"
```

Step 2 of 5

```
"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
beta=1/(T_film+273)
sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzmann constant"
g=9.807 "[m/s^2], gravitational acceleration"
```

Step 3 of 5

```
"FORMULAE"
T_film=1/2*(T_s+T_infinity)
delta=D
Ra=(g*beta*(T_s-T_infinity)*delta^3)/nu^2*Pr
Nusselt=(0.6+(0.387*Ra^(1/6))/(1+(0.559/Pr)^(9/16))^(8/27))^2
h=k/delta*Nusselt
A=pi*D*L
Q_dot=h*A*(T_s-T_infinity)+epsilon*A*sigma*((T_s+273)^4-(T_surr+273)^4)
Q_gas=(Q_dot*time)/eta_furnace*Convert(h, s)*Convert(J, kJ)*Convert(kJ, therm)
Cost=Q_gas*UnitCost
```

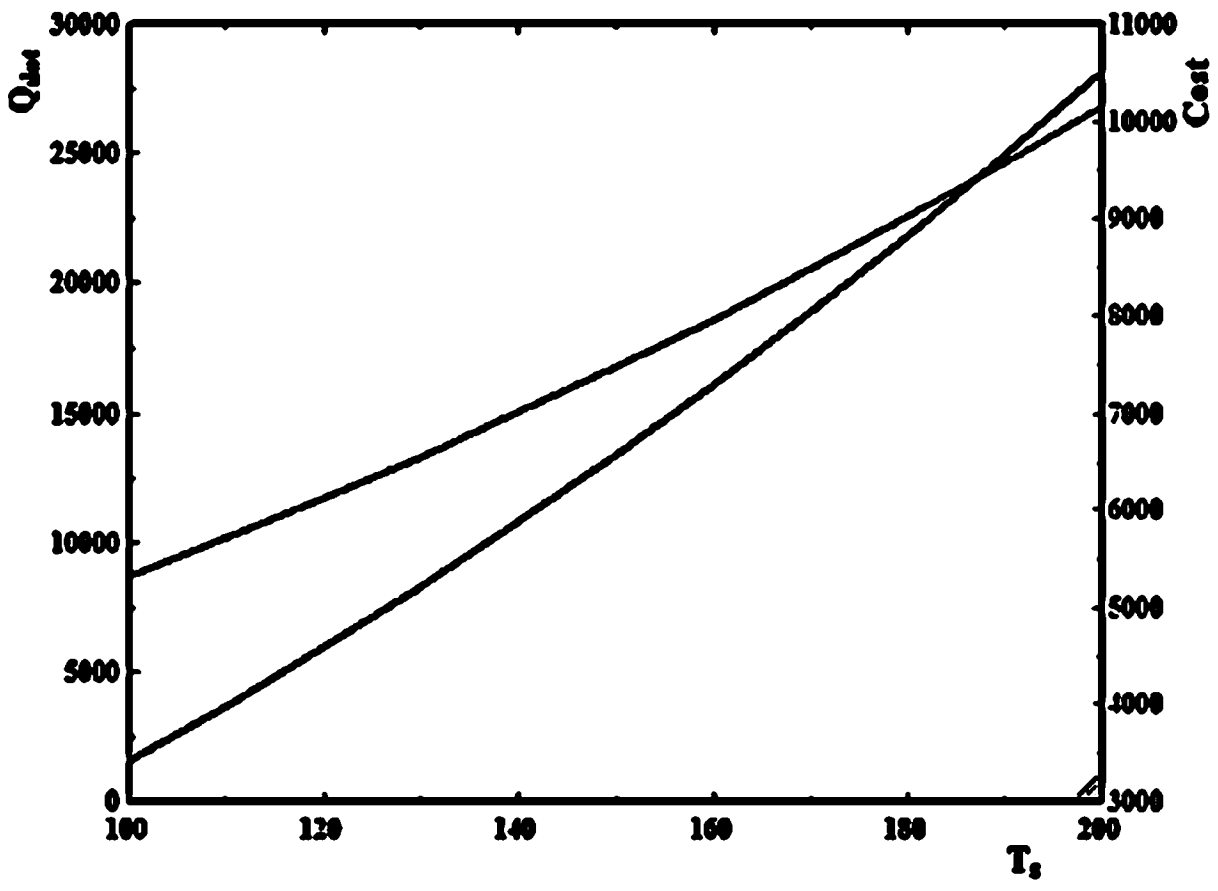
Step 4 of 5

The variation of heat loss and annual cost with surface temperature is as shown below.

▷ 1..11	1 T_s	2 $\dot{Q}$	3 Cost
Run 1	100	8726	3416
Run 2	110	10183	3986
Run 3	120	11714	4585
Run 4	130	13320	5214
Run 5	140	15005	5873
Run 6	150	16768	6563
Run 7	160	18613	7286
Run 8	170	20542	8041
Run 9	180	22558	8830
Run 10	190	24664	9654
Run 11	200	26862	10514

Step 5 of 5

Plot showing the variation of heat loss and annual cost with surface temperature:



Thus both the heat loss and annual cost increases with increase in the surface temperature.

Question:

Reconsider Prob. 20-38. In order to reduce heat losses, it is proposed to insulate the steam pipe with 5-cm-thick fiberglass insulation (k =0.038 W/m·K) and to wrap it with aluminum foil ($\varepsilon = 0.1$) in order to minimize the radiation losses. Also, an estimate is obtained from a local insulation contractor, who proposed to do the insulation job for $750. including materials and labor. Would you support this proposal? How long will it take for the insulation to pay for itself from the energy it saves? Assume the temperature of the steam pipe to remain constant at 170°C.

Step 1 of 11

Before insulation,

Calculate the mean film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

Where,

T_s is the pipe outer surface temperature,

T_∞ is the ambient temperature,

Substitute $T_s = 170°C$ and $T_\infty = 20°C$ in the above equation,

$$T_f = \frac{170+20}{2}$$
$$= 95°C$$
$$= 368\ K$$

Step 2 of 11

From properties of air table, the properties of at 95°C

$$k = 0.0306\ W/m\cdot K$$
$$\upsilon = 2.252\times10^{-5}\ m^2/s$$
$$Pr = 0.7121$$
$$\beta = \frac{1}{T_f} = 0.002717\ K^{-1}$$

Step 3 of 11

Surface area of the pan,

$$A_s = \pi D_1 L$$

Where,

D_1 is the diameter of the steam pipe before insulation,

L is the Length of the steam pipe,

Substitute $D_1 = 0.0603\ m$ and $L = 45\ m$ in the above equation,

$$A_s = \pi(0.0603\ m)(45\ m)$$
$$= 8.524\ m^2$$

Characteristic length, $L_c = D = 0.0603\ m$

Rayleigh number,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\upsilon^2}\times Pr$$
$$= \frac{9.81\times0.002717\times(170-20)\times0.0603^3}{(2.252\times10^{-5})^2}(0.7121)$$
$$= 1.231\times10^6$$

Step 4 of 11

Nusselt number,

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/Pr)^{9/16}\right]^{8/27}}\right\}^2$$
$$= \left\{0.6 + \frac{0.387(1.231\times10^6)^{1/6}}{\left[1+(0.559/0.7121)^{9/16}\right]^{8/27}}\right\}^2$$
$$= 15.42$$

Step 5 of 11

But, $Nu = \frac{hL_c}{k}$, so heat transfer coefficient is

$$h = \frac{k}{D_1}Nu$$
$$= \frac{0.0306}{0.0603}\times(15.42)$$
$$= 7.823\ W/m^2\cdot K$$

Calculate the total heat loss,

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^4 - T_\infty^4)$$

Where,

ε is the emissivity of surface of pipe,

Substitute $\varepsilon = 0.7$ and the respective values in the above equation,

$$\dot{Q} = \{7.823\times8.524\times(170-20)\}$$
$$+\left\{0.7\times8.524\times5.67\times10^{-8}\times[(170+273)^4-(20+273)^4]\right\}$$
$$= 20538.921\ W$$
$$= 20.538\ kW$$

Step 6 of 11

After insulation, let the surface temperature be assumed to 35°C, therefore $T_s = 35°C$

Then the mean film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{35+20}{2}$$
$$= 27.5°C$$
$$= 300.5\ K$$

Step 7 of 11

From properties of air table, the properties of at 27.5°C

$$k = 0.0257\ W/m\cdot K$$
$$\upsilon = 1.584\times10^{-5}\ m^2/s$$

Step 7 of 11

From properties of air table, the properties of at 27.5°C

$k = 0.0257 \text{ W/m-K}$

$\upsilon = 1.584\times10^{-5} \text{ m}^2/\text{s}$

$\text{Pr} = 0.7289$

$$\beta = \frac{1}{T_f} = 0.003328 \text{ K}^{-1}$$

Surface area of the pan,

$$A_s = \pi D_2 L$$

Where,

D_2 is the diameter of the steam pipe after insulation,

Substitute $D_2 = 0.1603$ m and the respective values in the above equation,

$$A_s = \pi(0.1603 \text{ m})(45 \text{ m})$$
$$= 22.661 \text{ m}^2$$

Step 8 of 11

Characteristic length, $L_c = D = 0.1603$ m.

Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\upsilon^2}\text{Pr}$$
$$= \frac{9.81\times0.003328\times(35-20)\times0.1603^3}{(1.584\times10^{-5})^2}(0.7289)$$
$$= 5.856\times10^6$$

Nusselt number,

$$\text{Nu} = \left\{0.6 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$
$$= \left\{0.6 + \frac{0.387(5.856\times10^6)^{1/6}}{\left[1+(0.559/0.7289)^{9/16}\right]^{8/27}}\right\}^2$$
$$= 24.23$$

Step 9 of 11

But $\text{Nu} = \frac{hL_c}{k}$, so heat transfer coefficient is

$$h = \frac{k}{D}\text{Nu}$$
$$= \frac{0.0257}{0.1603}\times(24.23)$$
$$= 3.884 \text{ W/m}^2\text{-K}$$

Calculate the Total heat loss,

$$\dot{Q} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^4 - T_{sur}^4)$$
$$= \{3.884\times22.661\times(35-20)\} +$$
$$\{0.1\times22.661\times5.67\times10^{-8}\times[(35+273)^4 - (20+273)^4]\}$$
$$= 1529.553 \text{ W}$$
$$= 1.529 \text{ kW}$$

Step 10 of 11

In steady state operation, the heat lost by the side surfaces of the pipe must be equal to the heat lost from the exposed surface of the insulation by convection and radiation, which must be equal to the heat conducted through the insulation.

$T_{s,1}$ is the temperature on the surface of the pipe, and $T_{s,1} = 170°\text{C}$

T_s is the temperature on the surface of the insulation.

$$\dot{Q}_{ins} = \frac{T_{s,1} - T_s}{R_{ins}}$$

$$\dot{Q}_{ins} = \frac{T_{s,1} - T_s}{\frac{\ln(D_2/D_1)}{2\pi kL}}$$

$$1529.553 = \frac{(170 - T_s)}{\left[\frac{\ln(16.03/6.03)}{2\pi\times0.038\times45}\right]}$$

$$170 - T_s = 1529.553\times0.091$$

$$T_s = 30.812°\text{C}$$

Calculate the reduction heat loss,

$$\dot{Q}_{saved} = 20.538 - 1.529$$
$$= 19.009 \text{ kW}$$

Calculate the annual cost of the container from the heat loss,

$$Q_{gas} = \frac{\dot{Q}\Delta t}{\eta}$$
$$= \frac{19.009 \text{ kJ/s}}{0.84}\left(\frac{1 \text{ therm}}{105{,}500 \text{ kJ}}\right)(8760 \text{ h/yr}\times3600 \text{ s/h})$$
$$= 6764.475 \text{ therms/yr}$$

$$\text{Cost} = (6764.475 \text{ therms/yr})\times(\$1.10/\text{therm})$$
$$\text{Cost} = \$\,7440.922 \text{ per yr}$$

Step 11 of 11

Calculate the time required for the insulation to pay for itself from the energy it saves is

$$t = \frac{\$750}{\$7440.992 \text{ per yr}}$$
$$= 0.1432\times365 \text{ days}$$
$$= 36.78 \text{ days}$$

Therefore, in $\boxed{36.78 \text{ days}}$ cost of insulation gets paid by itself.

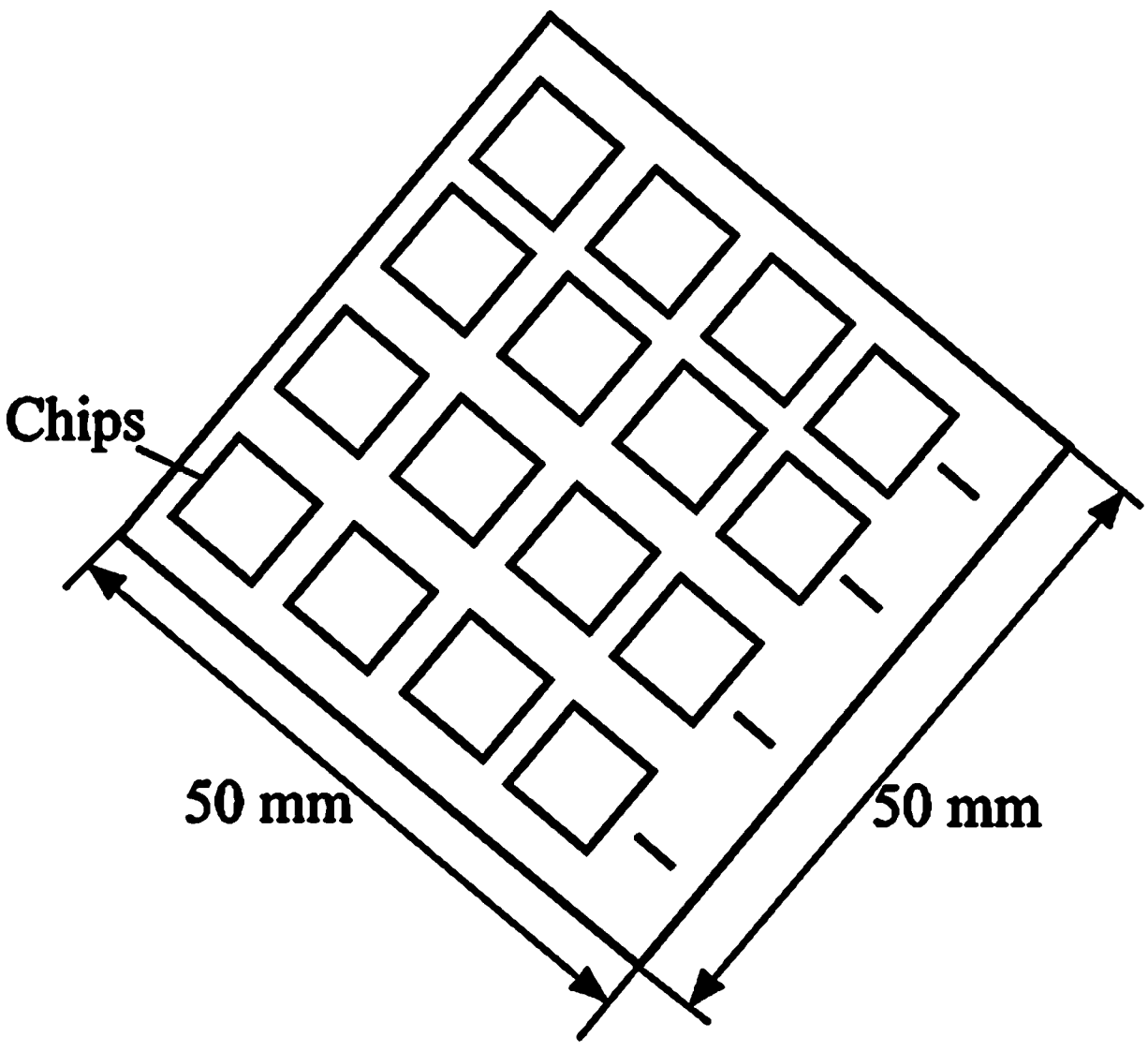
Therefore, in 36.78 days cost of [illegible] gets paid by itself.

Question:

A 50-cm × 50-cm circuit board that contains 121 square chips on one side is to be cooled by combined natural convection and radiation by mounting it on a vertical surface in a room at 25°C. Each chip dissipates 0.18 W of power, and the emissivity of the chip surfaces is 0.7. Assuming the heat transfer from the back side of the circuit board to be negligible, and the temperature of the surrounding surfaces to be the same as the air temperature of the room, determine the surface temperature of the chips.

Step 1 of 9

Sketch the circuit board with square chips cooled by natural convection as follows:



Step 2 of 9

Assume the surface temperature of the circuit board as $35°C$.

$$T_s = 35°C$$

Calculate the mean flam temperature of the air as follows:

$$T_f = \frac{T_s + T_\infty}{2}$$

Substitute $35°C$ for T_s and $25°C$ for T_∞.

$$T_f = \frac{35+25}{2}$$
$$= 30°C$$

Step 3 of 9

Calculate the coefficient of volume expansion as follows:

$$\beta = \frac{1}{T_f}$$

Substitute $30°C$ for T_f.

$$\beta = \frac{1}{30}$$
$$= \frac{1}{(30+273)\ \mathrm{K}}$$
$$= 3.3\times10^{-3}\mathrm{K}^{-1}$$

Obtain the properties of air from table A-15, "Properties of air at 1 atm pressure" at $T = 30°C$ as follows:

Thermal conductivity, $k = 0.02588\ \mathrm{w/m°C}$

Kinematic viscosity, $\nu = 1.608\times10^{-5}\ \mathrm{m^2/s}$

Prandtl number, $\mathrm{Pr} = 0.7282$

Step 4 of 9

Calculate the characteristic length L_c.

$$L_c = \frac{A_s}{p}$$

Here, A_s is the surface area of the board, and p is the parameter of the board.

Substitute $(L\times w)$ for A_s and $2(L+w)$ for p.

$$L_c = \frac{(L\times w)}{2(L+w)}$$

Here, L and w is the length and width of the board.

Substitute 50 mm for L and 50 mm for w .

$$L_c = \frac{(50\times50)\ \mathrm{mm^2}}{2(50+50)\ \mathrm{mm}}$$
$$= \frac{\left(2500\times10^{-6}\right)\ \mathrm{m^2}}{2\left(100\times10^{-3}\right)\mathrm{m}}$$
$$= 0.125\,\mathrm{m}$$

Step 5 of 9

Calculate the Rayleigh number as follows:

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\mathrm{Pr}$$

Calculate the Rayleigh number as follows:

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot \mathrm{Pr}$$

Here, g is the acceleration due gravity, T_s is the surface temperature, T_∞ is the ambient temperature, Pr is the Prandtl number, ν is the kinematic viscosity.

Substitute $9.81\ \mathrm{m/s^2}$ for g, $3.3\times10^{-3}\mathrm{K^{-1}}$ for β, $35°\mathrm{C}$ for T_s, $25°\mathrm{C}$ for T_∞, $0.125\,\mathrm{m}$ for L_c, $1.608\times10^{-5}\ \mathrm{m^2/s}$ for ν and 0.7282 for Pr.

$$\mathrm{Ra}_L = \frac{9.81\times3.3\times10^{-3}(35-25)(0.125)^3}{\left(1.608\times10^{-5}\right)^2}(0.7282)$$

$$= 178.07\times10^4$$

Step 6 of 9

Calculate the Nusselt number as follows:

From the empirical correlation for the average Nusselt number for natural convention over surfaces, as it is a upper surface hot plate and lies in rang $\left(10^4 - 10^7\right)$, hence the relation for Nusselt number becomes as follows:

$$\mathrm{Nu} = 0.54\left(\mathrm{Ra}_L\right)^{0.25}$$

Substitute 178.07×10^4 for Ra_L.

$$\mathrm{Nu} = 0.54\times\left(178.07\times10^4\right)^{0.25}$$

$$= 19.726$$

Calculate the heat transfer coefficient h.

$$h = \frac{k}{L_c}\times\mathrm{Nu}$$

Substitute $0.02588\,\mathrm{w/m°C}$ for k, $0.125\,\mathrm{m}$ for L_c, and 19.726 for Nu.

$$h = \frac{0.02588}{0.125}\times19.726$$

$$= 4.084\,\mathrm{w/m^2°C}$$

Step 7 of 9

Calculate the total heat transfer dissipated from the board.

$$\dot{Q} = \text{heat dissipated by one chip} \times \text{number of chips}$$

$$= 0.18\ \mathrm{W}\times121$$

$$= 21.78\ \mathrm{W}$$

a)

Calculate the surface temperature of the chip, if the chips are facing up as follows:

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon\sigma A\left((T_s)^4 - \left(T_\infty^4\right)\right)$$

Here, ε is the emissivity, σ is the Stephen Boltzmann's constant.

Substitute 21.78 W for $\dot{Q}$, $4.084\,\mathrm{w/m^2°C}$ for h, $25°\mathrm{C}$ for T_∞, $5.67\times10^{-8}\ \mathrm{W/m^2\cdot K}$ for σ, 0.7 for ε.

$$21.78 = 4.084\times0.25(T_s - 25) + 0.7\times5.67\times10^{-8}\times0.25\left((T_s+273)^4 - (298)^4\right)$$

$$21.78 = 1.01(T_s - 25) + 9.9225\times10^{-9}\left((T_s+273)^4 - (298)^4\right)$$

$$21.78 = 1.01T_s - 25.25 + 9.9225\times10^{-9}(T_s+273)^4 - 78.25$$

$$125.28 = 1.01T_s + 9.9225\times10^{-9}(T_s+273)^4$$

$$T_s = 35.675°\mathrm{C}$$

Therefore, the surface temperature of the chip, if the chips are facing up is $\boxed{35.675°\mathrm{C}}$.

Step 8 of 9

b)

If the chips are facing down as follows:

Calculate the Nusselt number as follows:

From the empirical correlation for the average Nusselt number for natural convention over surfaces, as it is downward with hot surface down range $\left(10^5 - 10^{11}\right)$, hence the relation for Nusselt number becomes as follows:

$$\mathrm{Nu} = 0.27\left(\mathrm{Ra}_L\right)^{1/4}$$

Substitute 178.07×10^4 for Ra_L.

$$\mathrm{Nu} = 0.27\times\left(178.07\times10^4\right)^{0.25}$$

$$= 9.3631$$

Calculate the heat transfer coefficient h.

$$h = \frac{k}{L_c}\times\mathrm{Nu}$$

Substitute $0.02588\,\mathrm{w/m°C}$ for k, $0.125\,\mathrm{m}$ for L_c, and 9.3631 for Nu.

$$h = \frac{0.02588}{0.125}\times9.8631$$

$$= 2.042\,\mathrm{w/m^2°C}$$

Step 9 of 9

Calculate the surface temperature of the chip, if the chips are facing downward as follows:

Step 9 of 9

Calculate the surface temperature of the chip, if the chips are facing downward as follows:

$$\dot{Q} = h A_s (T_s - T_\infty) + \varepsilon \sigma A \left((T_s)^4 - (T_\infty^4) \right)$$

Substitute 21.78 W for $\dot{Q}$, $2.042\,\mathrm{w/m^2{}^\circ C}$ for h, 25°C for T_∞, $5.67\times10^{-8}\ \mathrm{W/m^2 \cdot K}$ for σ, 0.7 for ε.

$$21.78 = 2.042\times0.25(T_s - 25) + 0.7\times5.67\times10^{-8}\times0.25\left((T_s+273)^4 - (298)^4\right)$$

$$21.78 = 0.5105(x-25) + 9.9225\times10^{-9}\left((x+273)^4 - (298)^4\right)$$

$$21.78 = 0.5105T_s - 12.76 + 9.9225\times10^{-9}(T_s+273)^4 - 78.25$$

$$112.8 = 1.01T_s + 9.9225\times10^{-9}(T_s+273)^4$$

$$T_s = 38.33^\circ C$$

Therefore, the surface temperature of the chip, if the chips are facing up is **38.33°C**.

Question: 20.042P

Repeat Prob. 20–41 assuming the circuit board to be positioned horizontally with (*a*) chips facing up and (*b*) chips facing down.

Step 1 of 8

Length of circuit board, $L = 50\text{cm} = 0.5\text{m}$

Breadth of circuit board, $w = 50\text{cm} = 0.5\text{m}$

Room temperature, $T_\infty = 25°\text{C}$

Emissivity of chip surface, $\varepsilon = 0.7$

Temperature of surrounding surface is, $T_{amb} = 25°\text{C}$

Step 2 of 8

a) Assuming the surface temperature of the circuit board is $T_s = 30°\text{C}$, calculate the mean film temperature of the air.

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{30+25}{2}$$
$$= 30°\text{C}$$

From the properties of air,

Thermal conductivity, $k = 0.02588\ \text{W/m-°C}$

Kinematic viscosity, $\nu = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

Step 3 of 8

Coefficient of volume expansion is

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{303}$$
$$= 3.3\times10^{-3}\text{K}^{-1}$$

Characteristic length,

$$L_c = \frac{A_s}{p}$$
$$= \frac{L\times w}{2(L+w)}$$
$$= \frac{(0.5)^2}{2(0.5+0.5)}$$
$$= 0.125\text{m}$$

Step 4 of 8

Calculate the Rayleigh number

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.3\times10^{-3}(35-25)(0.125)^3}{\left(1.608\times10^{-5}\right)^2}(0.7282)$$
$$= 178.07\times10^4$$

From the empirical correlation for the average Nusselt number for natural convention over surfaces,

$$\text{Nu} = 0.54(\text{Ra})^{0.25}$$
$$= 0.54\times\left(178.07\times10^4\right)^{0.25}$$
$$= 19.726$$

Step 5 of 8

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$= \frac{0.02588}{0.125}\times19.726$$
$$= 4.084\ \text{W/m}^2\text{-°C}$$

Therefore total heat transfer here will be

$$\dot{Q} = (\text{heat dissipated per chip})\times(\text{number of chips})$$
$$= 0.18\times121$$
$$= 21.78$$

Step 6 of 8

Now consider the total heat transfer as the sum of convection and radiation heat transfers.

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon\sigma A\left[(T_s)^4 - \left(T_{amb}^4\right)\right]$$

$$21.78 = 4.084(0.25)(T_s - 25) + 0.7\left(5.67\times10^{-8}\right)(0.25)\left[(T_s+273)^4 - (298)^4\right]$$

$$21.78 = 1.01(T_s - 25) + 9.9225\times10^{-9}\left[(T_s+273)^4 - (298)^4\right]$$

The value of temperature obtained by solving the above equation is $T_s = 35.675°\text{C}$

Therefore surface temperature of the circuit board is $\boxed{T_s = 35.675°\text{C}}$

Step 7 of 8

(b) Hot surface down facing. Consider the Rayleigh number which is calculated in above case.

$$\text{Ra}_L = 178.07\times10^4$$

Calculate the Nusselt number form the equation

$$\text{Nu} = 0.27\text{Ra}^{1/4}$$
$$= 0.27\times\left(178.07\times10^4\right)^{0.25}$$
$$= 9.3631$$

Heat transfer co-efficient

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$= \frac{0.02588}{0.125}\times9.8631$$
$$= 2.042\ \text{W/m}^2\text{-°C}$$

Step 8 of 8

Now consider the total heat transfer as the sum of convection and radiation heat transfers.

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon\sigma A\left[(T_s)^4 - \left(T_{amb}^4\right)\right]$$

$$21.78 = 2.042(0.25)(T_s - 25) + 0.7\left(5.67\times10^{-8}\right)(0.25)\left[(T_s+273)^4 - (298)^4\right]$$

$$21.78 = 1.01(T_s - 25) + 9.9225\times10^{-9}\left[(T_s+273)^4 - (298)^4\right]$$

The value of temperature obtained by solving the above equation is $T_s = 38.33°\text{C}$

Therefore surface temperature of the circuit board when facing down is $\boxed{T_s = 38.33°\text{C}}$

Question:

The side surfaces of a 3-m-high cubic industrial furnace burning natural gas are not insulated, and the temperature at the outer surface of this section is measured to be 110°C. The temperature of the furnace room, including its surfaces, is 30°C, and the emissivity of the outer surface of the furnace is 0.7. It is proposed that this section of the furnace wall be insulate with glass wool insulation (k= 0.038 W/m·K) wrapped by a reflective sheet (ε = 0.2) in order to reduce the heat loss by 90 percent. Assuming the outer surface temperature of the metal section still remains at about 110°C, determine the thickness of the insulation that needs to be used.

The furnace operates continuously throughout the year and has an efficiency of 78 percent. The price of the natural gas is $1.10/therm (1 therm = 105,500 kJ of energy content). If the installation of the insulation will cost $550 for materials and labor, determine how long it will take for the insulation to pay for itself from the energy it saves.

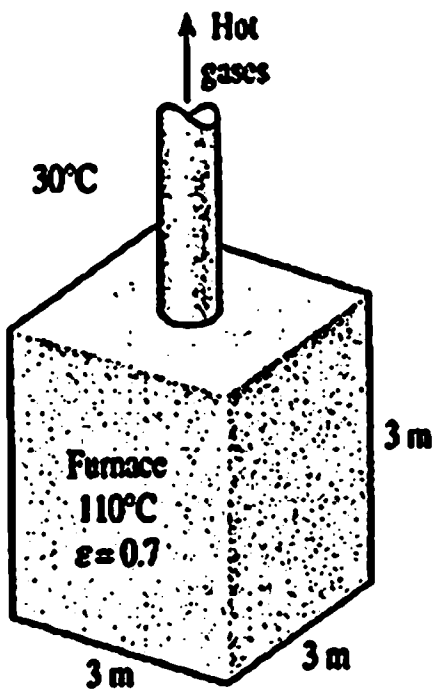


FIGURE P20–43

Step 1 of 10

Calculate the film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

Where,

T_s is the outer surface temperature of the furnace wall,

T_∞ is the temperature of the room.

Substitute $T_\infty = 303\,\text{K}$ and $T_s = 383\,\text{K}$ in the above equation,

$$T_f = \frac{110^\circ\text{C} + 30^\circ\text{C}}{2}$$
$$= 70^\circ\text{C}$$
$$\beta = \frac{1}{T_f + 273}$$
$$= \frac{1}{70 + 273}$$
$$= 0.002915\,\text{K}^{-1}$$

The properties of air at film temperature and atmospheric pressure are

Thermal conductivity, $k = 0.02881\,\text{W/m}^\circ\text{C}$

Kinematic viscosity, $\nu = 1.995 \times 10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7177$

Step 2 of 10

The Rayleigh number is given by

$$\text{Ra} = \text{Gr} \times \text{Pr}$$
$$= \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \times \text{Pr}$$

Where,

L is the side of the cube shaped furnace,

g is the acceleration due to gravity.

Subtitute $g = 9.81\,\text{m/s}^2$ and 3 m for L in the above equation,

$$\text{Ra} = \frac{9.81 \times 0.002915(110 - 30)(3)^3}{\left(1.995 \times 10^{-5}\right)^2} \times 0.7177$$
$$= 1.1138 \times 10^{11}$$

The relation for Nusselt number,

$$\text{Nu} = \left[0.825 + \frac{0.387 \times \text{Ra}^{1/6}}{\left(1 + (0.492/\text{Pr})^{9/16}\right)^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387 \times \left(1.1138 \times 10^{11}\right)^{1/6}}{\left(1 + (0.492/0.7177)^{9/16}\right)^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{26.844}{1.192}\right]^2$$
$$= 544.99$$

Step 3 of 10

Calculate the heat transfer coefficient,

$$h = \frac{\text{Nu} \times k}{L}$$
$$= \frac{544.99 \times 0.02881}{3}$$
$$= 5.234\ \text{W/m}^2\text{C}$$

Calculate the lateral surface area of the furnace,

$$A_s = 4 \times \left(L^2\right)$$
$$= 4 \times 3^2$$
$$= 36\ \text{m}^2$$

Step 4 of 10

Calculate the total heat of the given system,

$$\dot{Q} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_\infty^4\right)$$

Step 4 of 10

Calculate the total heat of the given system,

$$\begin{aligned}\dot{Q} &= \dot{Q}_{conv} + \dot{Q}_{rad} \\ &= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_\infty^4\right) \\ &= \left[5.234 \times 36(383-303)\right] + \left[0.7 \times 36 \times \left(5.67 \times 10^{-8}\right)\left(383^4 - 303^4\right)\right] \\ &= 33.77\ \text{kW}\end{aligned}$$

Calculate the quantity of heat lost,

$$\begin{aligned}\dot{Q}_{lost} &= (1-0.9)\dot{Q} \\ &= (0.1)(33.77\ \text{kW}) \\ &= 3.377\ \text{kJ/s}\end{aligned}$$

Calculate the quantity of heat saves,

$$\begin{aligned}\dot{Q}_{saved} &= 0.9\dot{Q} \\ &= (0.9)(33.77\ \text{kW}) \\ &= 30.39\ \text{kJ/s}\end{aligned}$$

Step 5 of 10

The cost of energy saved by controlling the heat loss is given by

$$\begin{aligned}\text{Cost} &= \left(\frac{\dot{Q}_{saved}}{\eta_f}\right) \times (\Delta t) \times (\text{Unit cost}) \\ &= \left(\frac{30.39}{0.78}\ \text{kJ/s} \times \frac{1\ \text{therm}}{105500\ \text{kJ}}\right) \times (365 \times 24 \times 60 \times 60\ \text{s/yr}) \times (\$1.1/\text{therm}) \\ &= \$12811\end{aligned}$$

The duration in which the saved energy repays the installation cost is given by

$$\begin{aligned}\text{time} &= \frac{\text{installation cost}}{\text{Cost saved per yr}} \\ &= \frac{\$550}{\$12811/\text{yr}} \\ &= 0.0429\ \text{yr} \\ &= 15.67\ \text{days}\end{aligned}$$

Therefore the time required for the saved cost to repay its installation charges is

$\boxed{15.67\ \text{days}}$

Step 6 of 10

Now by considering the insulation, the calculation should be repeated for determining the surface temperature T_{ins} of the insulation.

Consider the surface temperature of the insulation as $T_{ins} = 50^\circ\text{C}$

Calculate the film temperature,

$$\begin{aligned}T_f &= \frac{T_{ins} + T_\infty}{2} \\ &= \frac{50^\circ\text{C} + 30^\circ\text{C}}{2} \\ &= 40^\circ\text{C} \\ \beta &= \frac{1}{T_f + 273} \\ &= \frac{1}{40 + 273} \\ &= 0.003195\ \text{K}^{-1}\end{aligned}$$

The properties of air at film temperature and atmospheric pressure are

Thermal conductivity, $k = 0.02662\ \text{W/m}^\circ\text{C}$

Kinematic viscosity, $\nu = 1.702 \times 10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7255$

Step 7 of 10

The Rayleigh number is given by

$$\begin{aligned}\text{Ra} &= \text{Gr} \times \text{Pr} \\ &= \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \times \text{Pr} \\ &= \frac{9.81 \times 0.003195(50-30)(3)^3}{\left(1.702 \times 10^{-5}\right)^2} \times 0.7255 \\ &= 4.2388 \times 10^{10}\end{aligned}$$

The relation for Nusselt number is

$$\begin{aligned}\text{Nu} &= \left[0.825 + \frac{0.387 \times Ra^{1/6}}{\left(1 + (0.492/\text{Pr})^{9/16}\right)^{8/27}}\right]^2 \\ &= \left[0.825 + \frac{0.387 \times \left(4.2388 \times 10^{10}\right)^{1/6}}{\left(1 + (0.492/0.7255)^{9/16}\right)^{8/27}}\right]^2 \\ &= \left[0.825 + \frac{22.8517}{1.191}\right]^2 \\ &= 400.479\end{aligned}$$

Step 8 of 10

Calculate the heat transfer coefficient,

$$\begin{aligned}h &= \frac{\text{Nu} \times k}{L} \\ &= \frac{400.479 \times 0.02662}{3} \\ &= 3.55\ \text{W/m}^2\text{C}\end{aligned}$$

Calculate the total surface area of the insulation,

$$\begin{aligned}A_s &= 4 \times (L)(L + 2t_{ins}) \\ &= 4 \times (3)(3 + 2t_{ins}) \\ &= 12(3 + 2t_{ins})\ \text{m}^2\end{aligned}$$

Calculate the total surface area of the insulation,

20.043P pg 3

$$A_s = 4\times(L)(L+2t_{ins})$$
$$= 4\times(3)(3+2t_{ins})$$
$$= 12(3+2t_{ins})\,\text{m}^2$$

Step 9 of 10

Calculate the quantity of heat lost,

$$\dot{Q}_{lost} = \frac{k_{ins}A_{ins}(T_s - T_{ins})}{t_{ins}}$$

$$t_{ins} = \frac{k_{ins}A_{ins}(T_s - T_{ins})}{\dot{Q}_{lost}}$$
$$= \frac{0.038\times\left[12(3+2t_{ins})\right](110-T_{ins})}{3.377\ \text{kJ/s}}$$
$$= 0.135(3+2t_{ins})(110-T_{ins}) \qquad \ldots\ldots (1)$$

Step 10 of 10

Calculate the total heat loss of the given system by the energy balance equation,

$$\dot{Q}_{lost} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$3.377 = hA_{ins}(T_{ins} - T_\infty) + \varepsilon_r A_{ins}\sigma\left(T_{ins}^4 - T_\infty^4\right)$$
$$3.377 = \left[3.55\left[12(3+2t_{ins})\right](T_{ins}-303)\right]$$
$$+\left[0.2\left[12(3+2t_{ins})\right]\left(5.67\times10^{-8}\right)\left(T_{ins}^4 - 303^4\right)\right]$$
$$3.377 = \left[42.6(T_{ins}-303) + \left(13.608\times10^{-8}\right)\left(T_{ins}^4 - 303^4\right)\right](3+2t_{ins}) \ldots\ldots (2)$$

solve the equations (1) and (2),

$$t_s = 3.18\ \text{cm}$$

Therefore the thickness of the insulation is $\boxed{3.18\text{cm}}$

Question: 20.044P

A 1.5-m-diameter, 4-m-long cylindrical propane tank is initially filled with liquid propane, whose density is 581 kg/m3. The tank is exposed to the ambient air at 25°C in calm weather. The outer surface of the tank is polished so that the radiation heat transfer is negligible. Now a crack develops at the top of the tank, and the pressure inside drops to 1 atm while the temperature drops to –42°C. which is the boiling temperature of propane at 1 atm. The heat of vaporization of propane at 1 atm is 425 kJ/kg. The propane is slowly vaporized as a result of the heat transfer from the ambient air into the tank, and the propane vapor escapes the tank at –42°C through the crack. Assuming the propane tank to be at about the same temperature as the propane inside at all times, determine how long it will take for the tank to empty if it is not insulated.

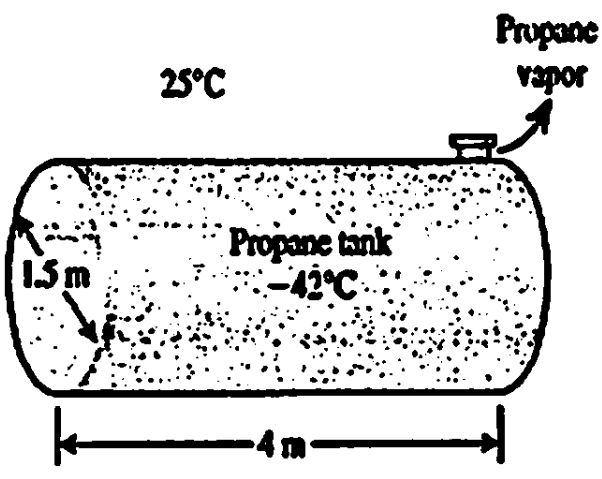


FIGURE P20–44

Step 1 of 4

The time for emptying the tank can be determined by finding the time for heating all the propane to its vapor state. In-order to find the time for heating the propane, determine the convectional heat input by considering the properties of air at film temperature.

The average film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

Substitute 25°C for T_∞ and –42°C for T_S

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{-42 + 25}{2}$$
$$= -8.5°C$$

The volume expansion coefficient,

$$\beta = \frac{1}{T_f + 273}$$
$$= \frac{1}{-8.5 + 273}$$
$$= 0.003781\ K^{-1}$$

Step 2 of 4

The properties of air at –8.5°C are

$$k = 0.02299\ W/m \cdot °C$$
$$\nu = 1.265 \times 10^{-5}\ m^2/s$$
$$Pr = 0.7383$$

Calculate the Rayleigh number. Substitute $9.81\ m/s^2$ for g, $0.003781 K^{-1}$ for β, –42°C for T_s, 25°C for T_∞, 1.5 m for D, $1.265 \times 10^{-5}\ m^2/s$ for ν and 0.7383 for Pr

$$Ra = \frac{g\beta(T_\infty - T_s)D^3}{\nu^2} Pr$$
$$= \frac{9.81 \times 0.003781 \times (25 + 42) \times 1.5^3}{\left(1.265 \times 10^{-5}\right)^2} \times 0.7383$$
$$= 3.869 \times 10^{10}$$

Now determine the Nusselt number. Substitute 3.869×10^{10} for Ra and 0.7383 for Pr

$$Nu = \left[0.6 + \frac{0.387 \times Ra^{1/6}}{\left(1 + (0.559/Pr)^{9/16}\right)^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{0.387 \times \left(3.869 \times 10^{10}\right)^{1/6}}{\left(1 + (0.559/0.7383)^{9/16}\right)^{8/27}}\right]^2$$
$$= 374.1$$

Step 3 of 4

Estimate the heat transfer coefficient from the Nusselt number. Substitute 374.1 for Nu, 4 m for L and 0.02299 W/m·°C for k

$$h = \frac{Nu \times k}{L}$$
$$= \frac{374.1 \times 0.02299}{4}$$
$$= 5.733\ W/m^2 \cdot K$$

Determine the surface area. Substitute 1.5 m for D and 4 m for L

$$A_s = (\pi \times D \times L) + 2\left(\frac{\pi}{4} \times D^2\right)$$
$$= (\pi \times 1.5 \times 4) + 2\left(\frac{\pi}{4} \times 1.5^2\right)$$
$$= 22.38\ m^2$$

Determine the total heat transfer from the circuit board. Substitute $5.733\ W/m^2 \cdot K$ for h, $22.38\ m^2$ for A_s, –42°C for T_s and 25°C for T_∞

$$\dot{Q} = hA_s(T_\infty - T_s)$$
$$= 374.1 \times 22.38 \times (25 - 42)$$
$$= 8598\ W$$
$$= 8.598\ kW$$

Step 4 of 4

Find the total mass of propane. Substitute $581\ kg/m^3$ for ρ, 1.5 m for D and 4 m for L

$$m = \rho \times V$$
$$= \rho \times \left(\frac{\pi}{4} \times D^2 \times L\right)$$
$$= 581 \times \left(\frac{\pi}{4} \times 1.5^2 \times 4\right)$$
$$= 4107\ kg$$

Determine the heat required for heating all the propane to its vapor state.

$$Q = m \times h_{fg}$$
$$= 4107 \times 425$$
$$= 1745475\ kJ$$

Calculate the time for emptying the tank. Substitute 1745475 kJ for Q and 8.598 kW for $\dot{Q}$

$$t = \frac{Q}{\dot{Q}}$$
$$= \frac{1745475\ kJ}{8.598\ kJ/s}$$
$$= 203009.42\ s$$
$$= 56.39\ hr$$

Thus the time for emptying the tank is **56.39 hr**

Question:

Consider a 2-ft · 2-ft thin square plate in a room at 75°F. One side of the plate is maintained at a temperature of 130°F, while the other side is insulated. Determine the rate of heat transfer from the plate by natural convection if the plate is (*a*) vertical; (*b*) horizontal with hot surface facing up: and (*c*) horizontal with hot surface facing down.

Step 1 of 9

Given

Ambient temperature $T_\infty = 75^0\text{F}$

Pipe outer surface temperature is $T_s = 130^0\text{F}$

Step 2 of 9

Average film temperature, $T_f = \dfrac{T_s + T_\infty}{2}$

$$T_f = \frac{75+130}{2}$$

$$T_f = 102.5^0\text{F} = 562.5^0\text{R}$$

Properties of the air at 102.5^0F, $k = 0.01535$ Btu/h.ft.°F

$\upsilon = 0.1823\times10^{-3}$ ft²/s

$\text{Pr} = 0.7256$

$$\beta = \frac{1}{T_f} = 0.001778\ \text{R}^{-1}$$

Step 3 of 9

a)

Surface area of the plate is $A_s = L^2$

$= (2\text{ ft})^2$

$= 4\text{ ft}^2$

Characteristic length is the height of the plate so $L_c = L = 2$ ft.

Rayleigh Number, $Ra = \dfrac{g\beta(T_s - T_\infty)L^3}{\upsilon^2}\text{Pr}$

$$Ra = \frac{32.2\times0.001778\times(130-75)\times2^3}{(0.1823\times10^{-3})^2}(0.7256)$$

$$Ra = \frac{25.19\times0.7256}{3.323329\times10^{-8}}$$

$$Ra = 5.503\times10^8$$

Step 4 of 9

Nusselt number, $Nu = \left\{0.825 + \dfrac{0.387\text{Ra}^{1/6}}{\left[1+\left(\dfrac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2$

$$Nu = \left\{0.825 + \frac{0.387(5.503\times10^8)^{1/6}}{\left[1+\left(\frac{0.492}{0.7256}\right)^{9/16}\right]^{8/27}}\right\}^2$$

$$Nu = \left\{0.825 + \frac{11.078}{1.190967}\right\}^2$$

$$Nu = 102.6$$

Step 5 of 9

But $Nu = \dfrac{hL}{k}$ So heat transfer coefficient,

$$h = \frac{k}{L}Nu$$

$$h = \frac{0.01535}{2}(102.6)$$

$h = 0.7869$ Btu/h.ft²°F

Heat transfer rate, $\dot{Q} = hA_s(T_s - T_\infty)$

$$\dot{Q} = 0.7869\times4\times(130-75)$$

$$\dot{Q} = 173.1\text{ Btu/hr}$$

$\boxed{\dot{Q} = 173.1\text{ Btu/hr}}$

Step 6 of 9

b)

the characteristic length, $L_s = \dfrac{A_s}{P}$

$$L_s = \frac{L}{4}$$

$$L_s = 0.5\text{ ft}$$

Rayleigh Number, $Ra = \dfrac{g\beta(T_s - T_\infty)L_s^3}{\upsilon^2}\text{Pr}$

$$Ra = \frac{32.2\times0.001778\times(130-75)\times(0.5)^3}{\left(0.1823\times10^{-3}\right)^2}(0.7256)$$

$$Ra = \frac{0.3936}{3.32333\times10^{-8}}(0.7256)$$

$$Ra = 8.598\times10^6$$

Nusselt number, $Nu = 0.54Ra^{1/4}$

$$Nu = 0.54(8.598\times10^6)^{1/4}$$

$$Nu = 29.24$$

Step 7 of 9

But $Nu = \dfrac{hL}{k}$ So transfer coefficient,

$$h = \frac{k}{L_c}Nu$$

$$h = \frac{0.01535}{(29.24)}$$

But $Nu = \frac{hL}{k}$ So transfer coefficient,

$$h = \frac{k}{L_c} Nu$$

$$h = \frac{0.01535}{0.5}(29.24)$$

$$h = 0.8975 \text{ Btu/h.ft}^2.^\circ\text{F}$$

Heat transfer rate, $\dot{Q} = hA_s(T_s - T_\infty)$

$$\dot{Q} = 0.8975 \times 4 \times (130 - 75)$$

$$\dot{Q} = 197.4 \text{ btu/hr}$$

$$\boxed{\dot{Q} = 197.4 \text{ btu/hr}}$$

Step 8 of 9

c)

the characteristic length, $L_s = \frac{A_s}{P}$

$$L_s = \frac{L}{4}$$

$$L_s = 0.5 \text{ ft}$$

Rayleigh Number, $Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\upsilon^2}\text{Pr}$

$$Ra = \frac{32.2 \times 0.001778 \times (130 - 75) \times (0.5)^3}{\left(0.1823 \times 10^{-3}\right)^2}(0.7256)$$

$$Ra = \frac{0.3936}{3.32333 \times 10^{-8}}(0.7256)$$

$$Ra = 8.598 \times 10^6$$

Nusselt number, $Nu = 0.27Ra^{1/4}$

$$Nu = 0.54(8.598 \times 10^6)^{1/4}$$

$$Nu = 14.62$$

Step 9 of 9

But $Nu = \frac{hL}{k}$ So transfer coefficient,

$$h = \frac{k}{L_c} Nu$$

$$h = \frac{0.01535}{0.5}(14.62)$$

$$h = 0.4487 \text{ Btu/h.ft}^2{}^\circ\text{F}$$

Heat transfer rate, $\dot{Q} = hA_s(T_s - T_\infty)$

$$\dot{Q} = 0.4487 \times 4 \text{ ft}^2 \times (130 - 75)$$

$$\dot{Q} = 98.7 \text{ Btu/h}$$

$$\boxed{\dot{Q} = 98.7 \text{ Btu/h}}$$

Question:

Reconsider Prob. 20–45E. Using EES (or other) software, plot the rate of natural convection heat transfer for different orientations of the plate as a function of the plate temperature as the temperature varies from 80°F to 180°F, and discuss the results.

Step 1 of 7

Side length of the thin square plate, $s = 2\,\text{ft}$

Outside room temperature, $T_\infty = 75°\text{F}$

Higher surface temperature, $T_s = 130°\text{F}$

Step 2 of 7

This given problem is solved in EES "Engineering Equation Solver" in the following method.

Open a new window in the EES software by ceasing " File > New " and type the program as shown below.

Equations Window

```
P_1=1 [atm]

"Properties of the air at the temperature T_f"
rho=Density(Air,T=T_f,P=P_1)
k=Conductivity(Air,T=T_f)*convert(R,F)
Pr=Prandtl(Air,T=T_f)
mu=Viscosity(Air,T=T_f)
nu=mu/rho*convert(ft^2/hr,ft^2/s)
beta=1/(converttemp('F','R',T_f))          "Co-efficient of volumetric expansion "
A_s=L^2                                    "Surface area of the plate "
p=4*L

"Plate is held vertical position"

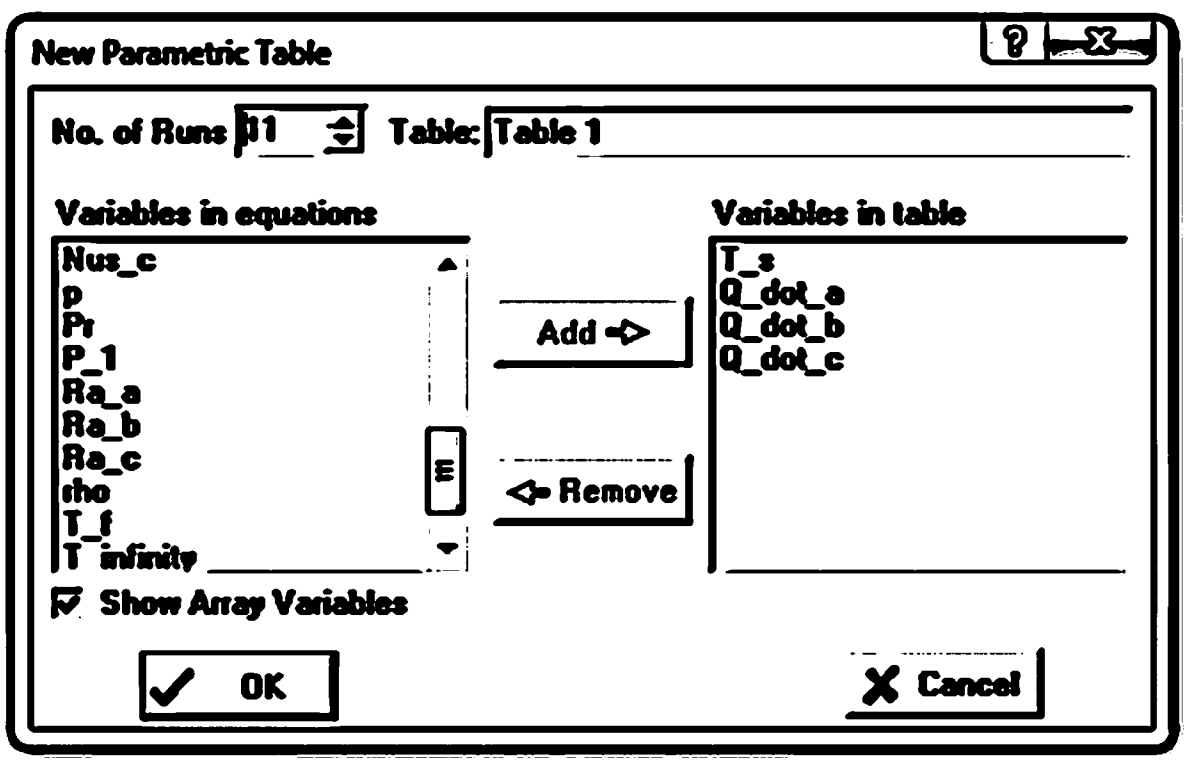
L_c_a=L                                    "Characteristic length in case a"
Ra_a=(g#*beta*(T_s-T_infinity)*L_c_a^3)/nu^2*Pr
Nus_a=(0.825+((0.387*Ra_a^(1/6))/(1+(0.492/Pr)^(9/16))^(8/27)))^2
h_a=k/L*Nus_a                              "Heat transfer coefficient "
Q_dot_a=h_a*A_s*(T_s-T_infinity)           "Heat transfer rate"

"The plate is in horizontal position with hot surface facing up"
L_c_b=A_s/p                                "Characteristic length in case b"
Ra_b=(g#*beta*(T_s-T_infinity)*L_c_b^3)/nu^2*Pr
Nus_b=0.54*Ra_b^(1/4)
h_b=k/L*Nus_b                              "Heat transfer coefficient "
Q_dot_b=h_b*A_s*(T_s-T_infinity)           "Heat transfer rate"

"The plate is in horizontal position with hot surface facing down"
L_c_c=A_s/p                                "Characteristic length in case b"
Ra_c=(g#*beta*(T_s-T_infinity)*L_c_c^3)/nu^2*Pr
Nus_c=0.27*Ra_c^(1/4)
h_c=k/L*Nus_c                              "Heat transfer coefficient "
Q_dot_c=h_c*A_s*(T_s-T_infinity)           "Heat transfer rate"
```

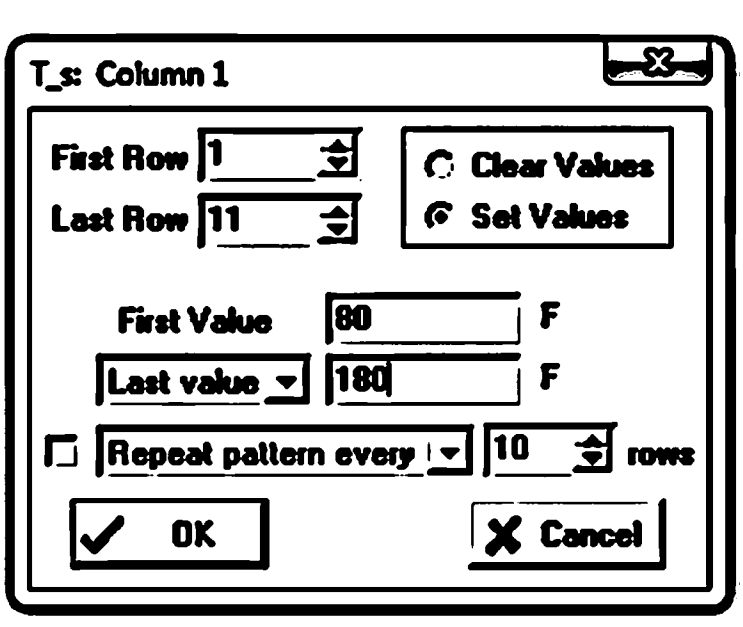
Step 3 of 7

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,



Step 4 of 7

To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Step 5 of 7

Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

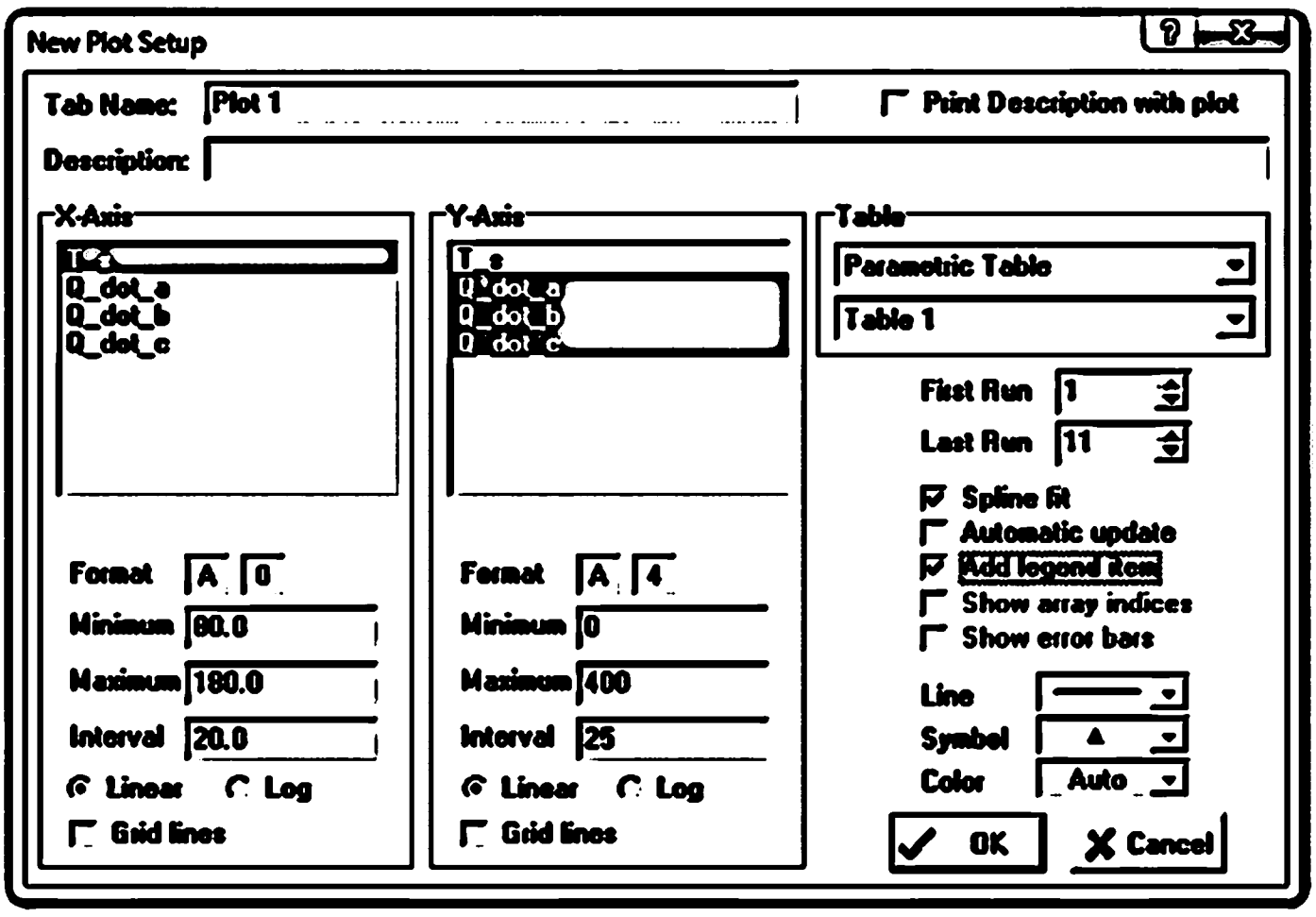
Table 1

1..11	T_s [F]	$\dot{Q}_a$	$\dot{Q}_b$	$\dot{Q}_c$
Run 1	80	7.825	2.495	1.247
Run 2	90	32.43	9.823	4.912
Run 3	100	62.75	18.55	9.277
Run 4	110	96.83	28.18	14.09

Run 1	80	04 6[illegible]25	g 22.495	1.247
Run 2	90	32.43	9.823	4.912
Run 3	100	62.75	18.55	9.277
Run 4	110	96.83	28.18	14.09
Run 5	120	133.8	38.49	19.24
Run 6	130	173	49.33	24.67
Run 7	140	214.1	60.64	30.32
Run 8	150	256.9	72.34	36.17
Run 9	160	301.1	84.38	42.19
Run 10	170	346.5	96.73	48.37
Run 11	180	393.1	109.4	54.68

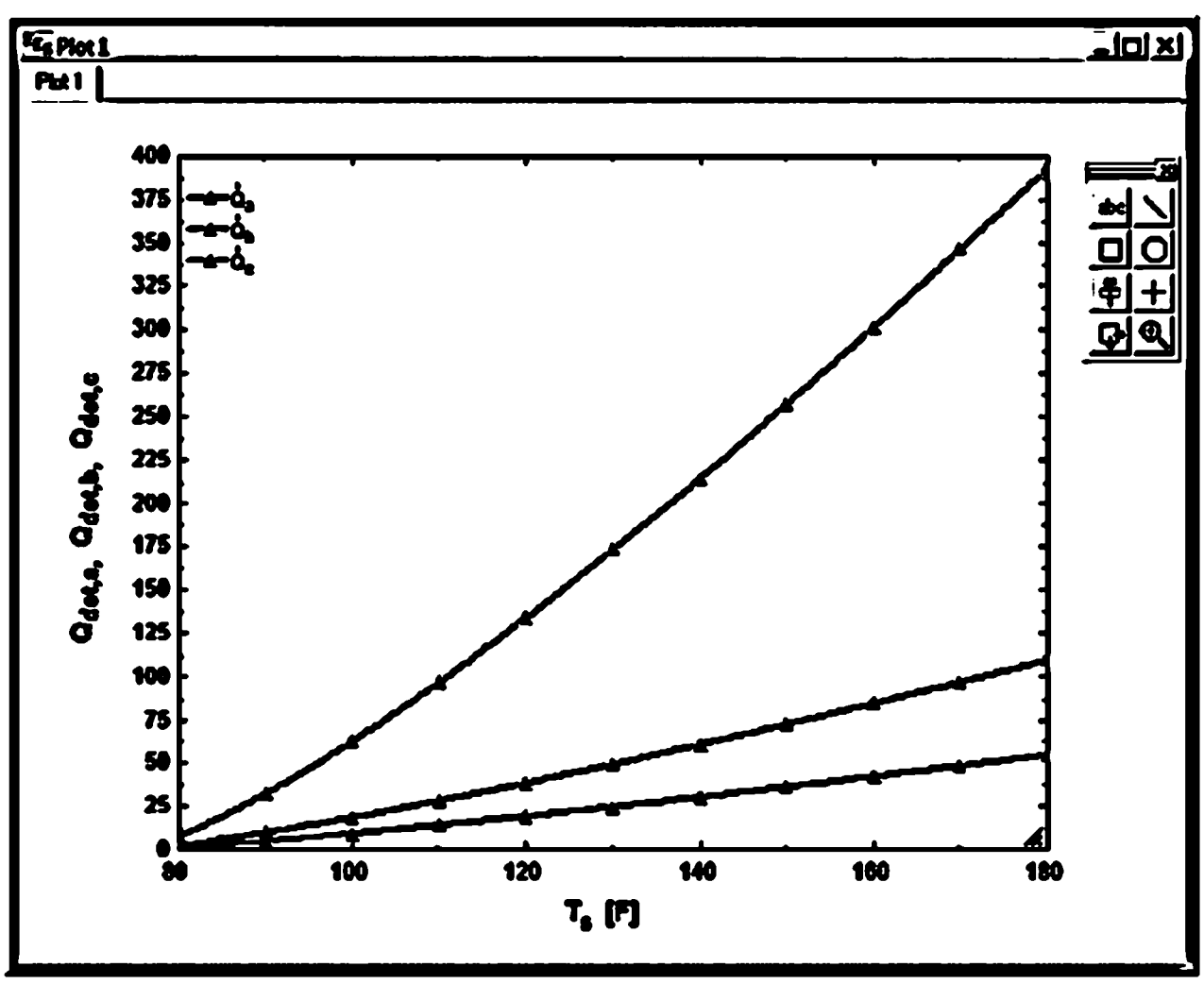
Step 6 of 7

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 7 of 7

By clicking on OK the graph will be generated as shown,



Question: 20.047P

A 28-cm-high, 18-cm-long, and 18-cm-wide rectangular container suspended in a room at 24°C is initially filled with cold water at 2°C. The surface temperature of the container is observed to be nearly the same as the water temperature inside. The emissivity of the container surface is 0.6, and the temperature of the surrounding surfaces is about the same as the air temperature. Determine the water temperature in the container after 3 h, and the average rate of heat transfer to the water. Assume the heat transfer coefficient on the top and bottom surfaces to be the same as that on the side surfaces.

Step 1 of 7

Height of container, $H = 28\text{cm} = 0.28\text{m}$

Length of container, $L = 18\text{cm} = 0.18\text{m}$

Width of container, $w = 18\text{cm} = 0.18\text{m}$

Temperature outside the room, $T_\infty = 24°\text{C}$

Emissivity of container surface $\varepsilon = 0.6$

Outside surface temperature $T_{amb} = 24°\text{C}$

Cold water temperature $T_1 = 2°\text{C}$

Step 2 of 7

Assume the surface temperature as 6°C. Now the film temperature is

$$T_f = (T_s + T_\infty)/2$$
$$= (6+24)/2$$
$$= 15°\text{C}$$

Volume expansion Co-efficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{15+273}$$
$$= 3.472\times10^{-3}\text{K}^{-1}$$

Characteristic length,

$$L_c = H$$
$$= 0.28\text{m}$$

Step 3 of 7

Take the properties of air at 1 atm and temperature, 15°C

Thermal conductivity, $k = 0.02476\,\text{W/m}\cdot°\text{C}$

Prandtl Number, $\text{Pr} = 0.7323$

Kinematic Viscosity, $\nu = 1.470\times10^{-5}\,\text{m}^2/\text{s}$

Rayleigh number

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.472\times10^{-3}(24-6)(0.28)^3}{\left(1.470\times10^{-5}\right)^2}(0.7323)$$
$$= 4.56\times10^7$$

Step 4 of 7

Nusselt number for entire range

$$\text{Nu} = \left[0.825 + \frac{0.387\,Ra^{1/6}}{\left[1+(0.492/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387\left(4.56\times10^7\right)^{1/6}}{\left[1+(0.492/0.7323)^{9/16}\right]^{8/27}}\right]^2$$
$$= 48.596$$

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$= \frac{0.02476}{0.28}\times48.596$$
$$= 4.297\,\text{W/m}^2\cdot\text{K}$$

Step 5 of 7

Calculate the surface area.

$$A_s = 4(0.28\times0.18) + 2(0.18\times0.18)$$
$$= 0.2664\,\text{m}^2$$

Rate of heat transfer $\dot{Q}$ can be written as (radiation considered),

$$\dot{Q} = hA_s\left(T_\infty - \frac{T_1+T_2}{2}\right) + \varepsilon\sigma A_s\left(T_{amb}{}^4 - \left(\frac{T_1+T_2}{2}\right)^4\right)$$

$$\dot{Q} = 4.297(0.2664)\left(297 - \left(\frac{275+T_2}{2}\right)\right) + 0.6\left(5.67\times10^{-8}\right)0.2664\left(297^4 - \left(\frac{275+T_2}{2}\right)^4\right)$$

$$\dot{Q} = 1.1447\left(297 - \left(\frac{275+T_2}{2}\right)\right) + 9.0629\times10^{-9}\left(297^4 - \left(\frac{275+T_2}{2}\right)^4\right) \quad \text{..... (1)}$$

Step 6 of 7

Mass of water,

$$m = \rho v$$
$$= 1000\times(0.28\times0.18\times0.18)$$
$$= 9.072\,\text{kg}$$

Specific heat of water, $c_p = 4214\,\text{J/kg}°\text{C}$

Heat transfer to water,

$$Q = mc_p(T_2 - T_1)$$
$$= 9.072\times4214(T_2 - 275)$$
$$= 38229.41(T_2 - 275)$$

Average rate of heat transfer after 3 hours,

$$\dot{Q} = \frac{Q}{\Delta t}$$
$$\dot{Q} = \frac{38229.41}{3\times3600}(T_2 - 275)$$
$$\dot{Q} = 3.54(T_2 - 275) \quad \text{..... (2)}$$

Step 7 of 7

Consider equations (1) and (2). And calculate the final temperature.

By solving the temperatures, the final temperature becomes. $T_2 = 11.75°\text{C}$

Hence water temperature after 3 hours, is $\boxed{T_2 = 11.75°\text{C}}$

Average rate of heat transfer to water is obtained by substituting the value of T_2 in equation (2)

$$\dot{Q} = 3.54(284.75 - 275)$$
$$= 34.514\ \text{W}$$

The average rate of heat transfer to the water is $\boxed{\dot{Q} = 34.514\ \text{W}}$

Question:

Reconsider Prob. 20–47. Using EES (or other) software, plot the water temperature in the container as a function of the heating time as the time varies from 30 min to 10 h. and discuss the results.

Step 1 of 6

Loading solution. Please wait ...

Step 2 of 6

This given problem is solved in EES "Engineering Equation Solver" in the following method.
Open a new window in the EES software by ceasing " File > New " and type the program as shown below,

```
"Given data"
H=0.28 [m]
L=0.18 [m]
W=0.18 [m]
T_infinity=297 [K]
epsilon=0.6
T_water_i=275[K]
L_c=0.28 [m]
P_1=101.3 [kPa]
T_s=279 [K]                                   "Assumption of film temprature"
t=3 [hour]*convert(hour,second)

"Properties of air and water at given conditions"
A_s=2*((H*L)+(W*H)+(W*L))
T_f=(T_s+T_infinity)/2
T_w_avg=(T_water_i+T_water_f)
k=Conductivity(Air,T=T_f)
Pr=Prandtl(Air,T=T_f)
mu=Viscosity(Air,T=T_f)
cp=Cp(water,T=T_water_i,P=P_1)
rho=Density(Air,T=T_f,P=P_1)
nu=mu/rho
rho_w=density(water,T=T_water_i,P=P_1)

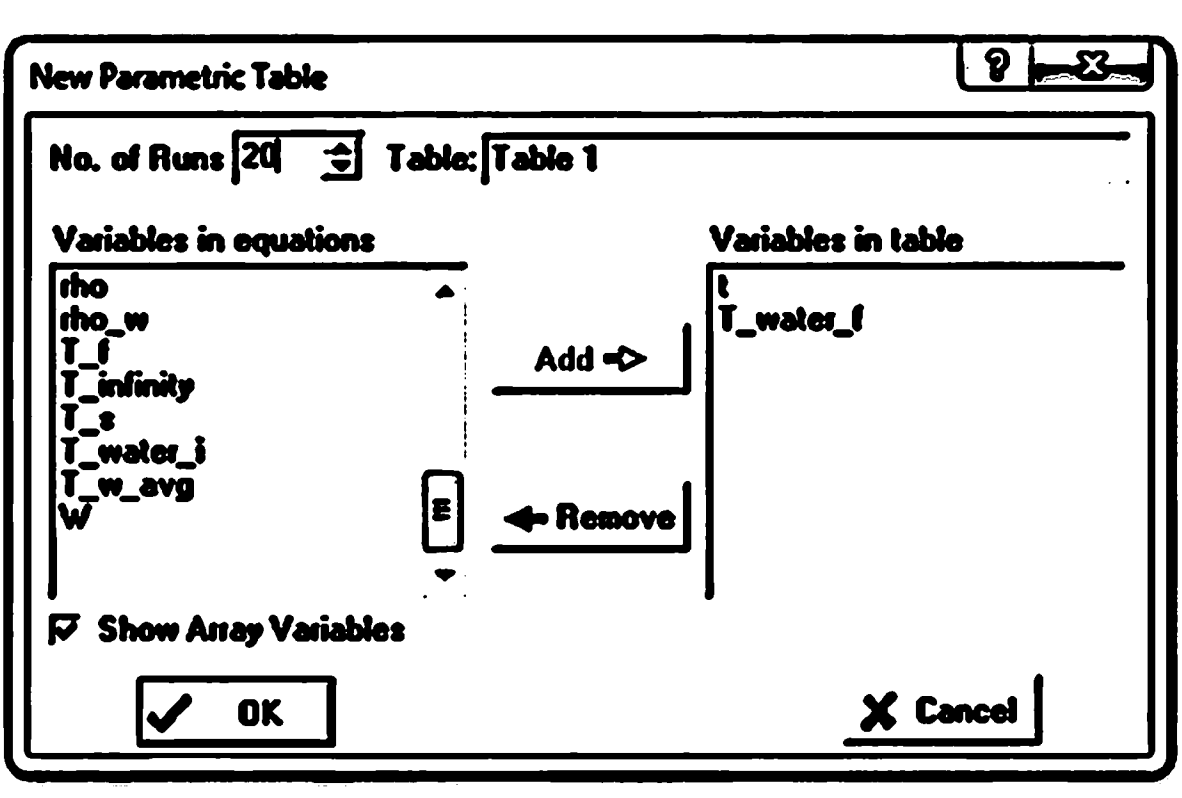
beta=1/T_f                                    "Co-efficient of Volume expansion "

Ra=(g)*beta*(T_infinity-T_s)*L_c^3)/nu^2*Pr
Nus=(0.825+((0.387*Ra^(1/6))/(1+(0.492/Pr)^(9/16)^(8/27))))   "Nusselt number "
h_bar=(k/L_c)*Nus

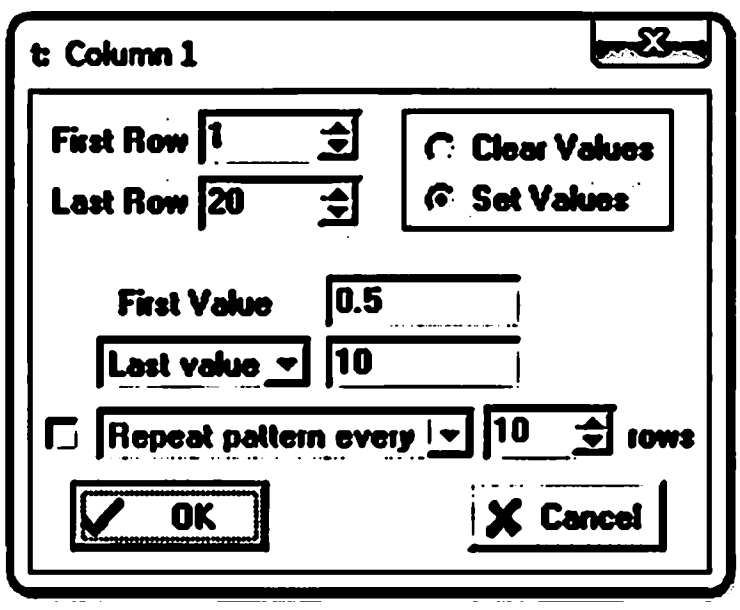
"Rate of heat transfer "
Q_dot=h_bar*A_s*(T_infinity- T_w_avg)+(epsilon*sigma#*A_s*((T_s)^4)-( T_w_avg)^4)
m=rho_w*(H*L*W)                               "Mass of water "
Q=m*cp*( T_water_f- T_water_i)                "Heat transfer to water"
Q_dot=Q/t                                     "Average rate of heat transfer after 3 hours"
```

Step 3 of 6

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,

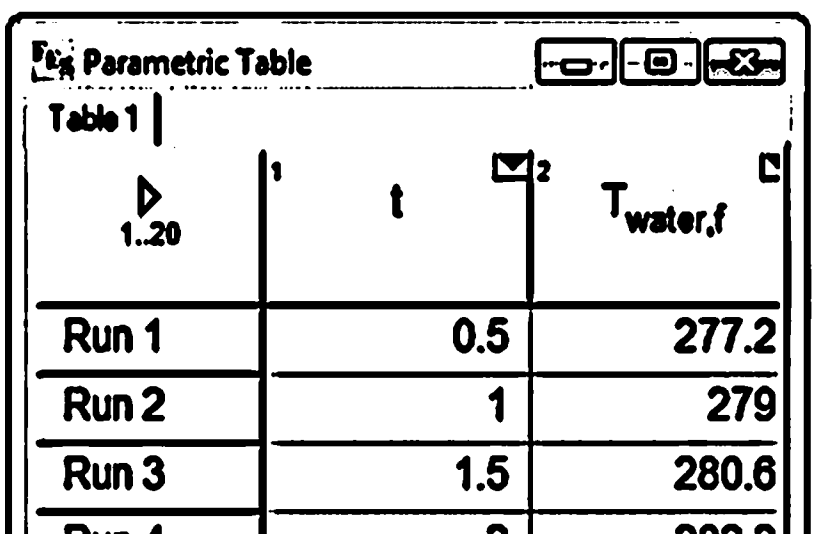


To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Step 4 of 6

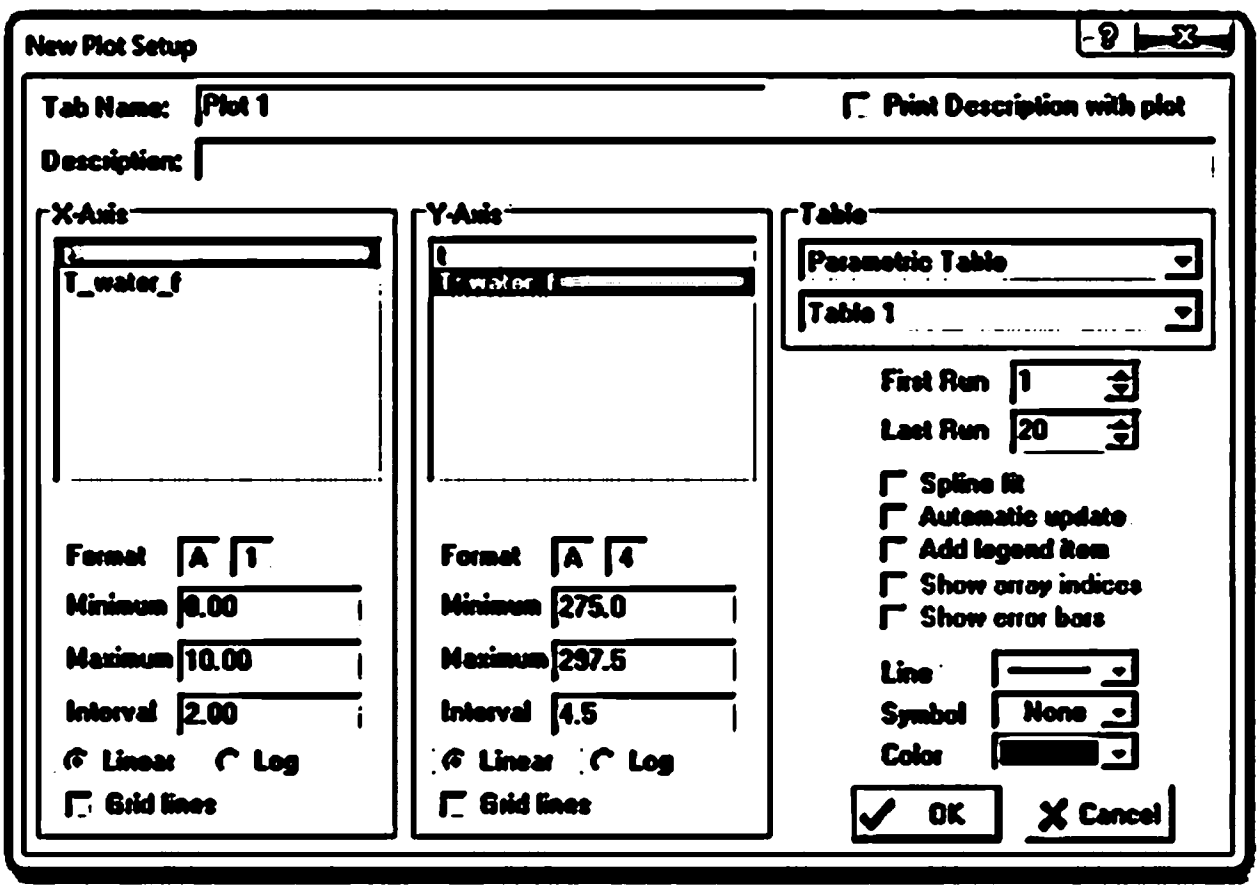


1..20	t	$T_{water,f}$
Run 1	0.5	277.2
Run 2	1	279
Run 3	1.5	280.6

Run 1	0.5	277.2
Run 2	1	279
Run 3	1.5	280.6
Run 4	2	282.2
Run 5	2.5	283.6
Run 6	3	284.8
Run 7	3.5	286
Run 8	4	287.1
Run 9	4.5	288.2
Run 10	5	289.1
Run 11	5.5	290
Run 12	6	290.8
Run 13	6.5	291.6
Run 14	7	292.3
Run 15	7.5	293.1
Run 16	8	293.7
Run 17	8.5	294.4
Run 18	9	295
Run 19	9.5	295.5
Run 20	10	296.1

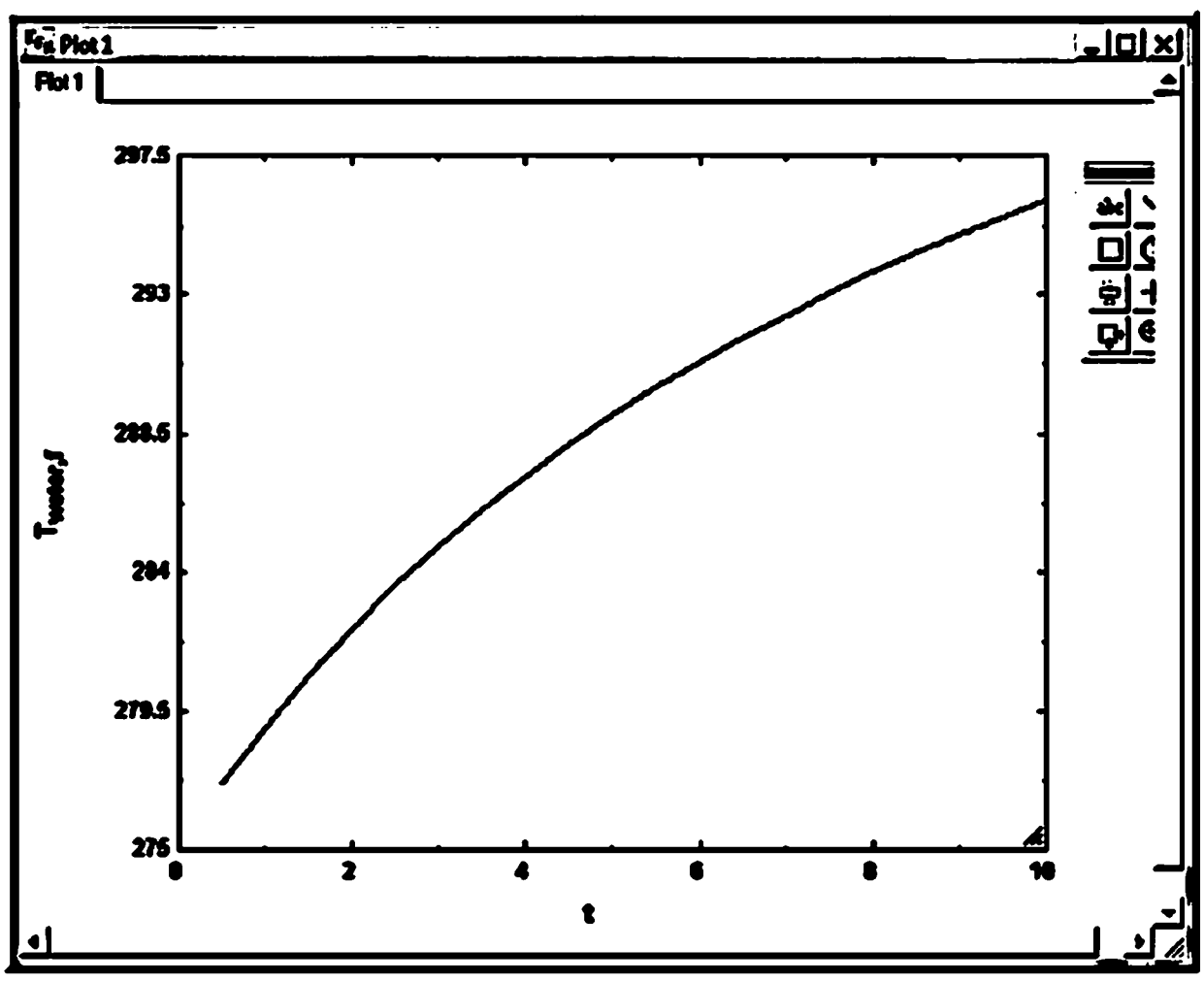
Step 5 of 6

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 6 of 6

By clicking on OK the graph will be generated as shown,



Question: 20.049P

The following experiment is conducted to determine the natural convection heat transfer coefficient for a horizontal cylinder that is 80 cm long and 2 cm in diameter. A 8 cm long resistance heater is placed along the centerline of the cylinder, and the surfaces of the cylinder are polished to minimize the radiation effect. The two circular side surfaces of the cylinder are well insulated. The resistance heater is turned on and the power dissipation is maintained constant at 60 W. If:; the average surface temperature of the cylinder is measured to 1 be 120°C in the 20°C room air when steady operation is I reached, determine the natural convection heat transfer coefficient. If the emissivity of the outer surface of the cylinder is 0.1 and a 5 percent error is acceptable, do you think we used to do any correction for the radiation effect? Assume the surrounding surfaces to be at 20°C also.

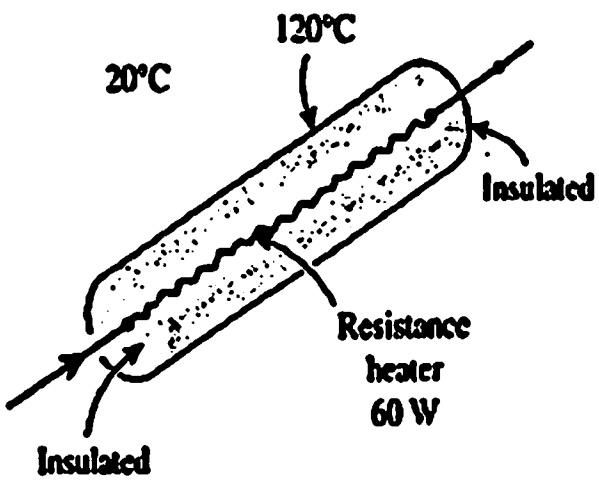


FIGURE P20–49

Step 1 of 3

Loading solution. Please wait ...

Step 2 of 3

Surface area, $A = \pi DL$

$A = \pi(0.02\ \text{m})(0.8\ \text{m})$

$A = 0.05027\ \text{m}^2$

Heat transfer due to convection, $\dot{Q} = hA_s(T_s - T_\infty)$

Now $h = \dfrac{\dot{Q}}{A_s(T_s - T_\infty)}$

$h = \dfrac{60\ \text{W}}{0.05027 \times (120 - 20)}$

$h = 11.9355\ \text{W/m}^2\text{C}$

Step 3 of 3

Radiation heat loss from the cylinder,

$\dot{Q}_{rad} = \varepsilon A_s \sigma(T_s^4 - T_{surr}^4)$

$\dot{Q}_{rad} = 0.1 \times 0.05027 \times 5.67 \times 10^{-8} \times [(120+273)^4 - (20+273)^4]$

$\dot{Q}_{rad} = 4.7\ \text{W}$

The fraction of heat loss by radiation,

Radiation fraction $= \dfrac{\dot{Q}_{radiation}}{\dot{Q}_{total}}$

$= \dfrac{4.7\ \text{W}}{60\ \text{W}}$

$= 0.07833$

$= 7.833\%$

This is greater than 5%. Therefore, the radiation effect is still considerable and necessary corrections must be made to control the radiation effect.

Question: 20.050P

The water in a 40-L tank is to be heated from 15°C to 45°C by a 6-cm-diameter spherical heater whose surface temperature is maintained at 85°C. Determine how long the heater should be kept on.

Step 1 of 7

Loading solution. Please wait ...

Step 2 of 7

Average temperature of water,

$$T_\infty = \frac{T_1 + T_2}{2}$$
$$= \frac{15+45}{2}$$
$$= 30°C$$

Mean film temperature of water

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{(85+30)}{2}$$
$$= 57.5°C$$

Step 3 of 7

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{57.5+273}$$
$$= 3.026\times10^{-3}\,K^{-1}$$

Take the properties of water by interpolating the values between $(55-60)°C$

Thermal conductivity, $k = 0.6515\,W/m\cdot°C$

Prandtl number, $Pr = 3.12$

Kinematic viscosity, $\nu = 4.933\times10^{-7}\,m^2/s$

Step 4 of 7

Properties of water at 30°C (Average temperature)

Density, $\rho = 996\,kg/m^3$

Specific heat, $c_p = 4178\,J/kg\,°C$

Characteristic length,

$$L_c = D$$
$$= 0.06\,m$$

Calculate the Rayleigh

$$Ra_D = \frac{g\beta(T_s - T_\infty)L_c^3}{(\nu)^2}\cdot Pr$$
$$= \frac{9.81\times\left(3.026\times10^{-3}\right)(85-30)(0.06)^3}{\left(4.933\times10^{-7}\right)^2}\times(3.12)$$
$$= 45.211\times10^8$$

Step 5 of 7

Calculate the Nusselt number for sphere $\left(Ra_D \le 10^{11} \text{ and } Pr \ge 0.7\right)$

$$Nu = 2 + \frac{0.589\,Ra_D^{1/4}}{\left[1+(0.469/Pr)^{9/16}\right]^{4/9}}$$
$$= 2 + \frac{0.589\left(45.211\times10^8\right)^{1/4}}{\left[1+(0.469/3.12)^{9/16}\right]^{4/9}}$$
$$= 136$$

Heat transfer co-efficient,

$$h = \frac{k}{L_c}\cdot Nu$$
$$= \frac{0.6515}{0.06}\times136$$
$$= 1475.72\,W/m^2\cdot°C$$

Step 6 of 7

Rate of heat transfer by convection

$$Q_{conv} = hA_s(T_s - T_\infty)$$
$$= h\left(\pi D^2\right)(T_s - T_\infty)$$
$$= 1475.72\left(\pi0.06^2\right)(85-30)$$
$$= 917.95\,W$$

Total amount of heat transfer to water:

$$Q = mc_p(T_2 - T_1)$$
$$= (\rho V)c_p(T_2 - T_1)$$
$$= (996\times0.04)\times4178\times(45-15)$$
$$= 4993545.6\,J$$

$$= (\rho V)c_p(T_2 - T_1)$$
$$= (996\times0.04)\times4178\times(45-15)$$
$$= 4993545.6\,J$$

Step 7 of 7

Time taken by the heater,

$$t = \frac{\text{amount of heat needed}}{\text{Rate of heat transfer}}$$
$$= \frac{Q}{Q_{conv}}$$
$$= \frac{4993545.6}{917.95}$$
$$= 5439.88\,sec$$
$$= 1.51\,h$$

The heater should be kept on for $\boxed{t = 1.51\,hours}$

Question: 20.052P

The upper and lower compartments of a well-insulated container are separated by two parallel sheets of glass with an air space between them. One of the compartments is to be filled with a hot fluid and the other with a cold fluid. If it is desired that heat transfer between the two compartments be minimal, would you recommend putting the hot fluid into the upper or the lower compartment of the container? Why?

Step 1 of 1

The convection currents get developed if the hotter fluid (lighter) is below the colder (denser) fluid. Hence it is better to have colder fluid in the bottom compartment and the hotter fluid in the upper compartment of a well-insulated container.

Question: 20.053P

Someone claims that the air space in a double-pane window enhances the heat transfer from a house because of the natural convection currents that occur in the air space and recommends that the double-pane window be replaced by a single sheet of glass whose thickness is equal to the sum of the thicknesses of the two glasses of the double-pane Window to save energy. Do you agree with this claim?

Step 1 of 1

By removing the air space and replacing the double pane with a single glass sheet, we eliminate the resistance offered by the air. As air has low thermal conductivity, the resistance offered by the air to flow of heat is more. Hence by removing the air space and by providing a single glass sheet, the heat transfer gets enhanced.

Question: 20.054P

Consider a double-pane window consisting of two glass sheets separated by a 1-cm-wide air space. Someone suggests inserting a thin vinyl sheet in the middle of the two glasses to form two 0.5-cm-wide compartments in the window in order to reduce natural convection heat transfer through the window. From a heat transfer point of view, would you be in favor of this idea to reduce heat losses through the window?

Step 1 of 1

Dividing the air space into thinner compartments would reduce the convection currents of the air. Hence by placing a thin vinyl sheet in the middle of two glasses to divide the air compartment into thinner compartments, heat losses from the window can be reduced.

Question: 20.055P

What does the effective conductivity of an enclosure represent? How is the ratio of the effective conductivity to thermal conductivity related to the Nusselt number?

Step 1 of 3

The rate of heat transfer through the enclosure is

$$\dot{Q} = hA_s(T - T_2)$$

$$= kNuA_s\left(\frac{T_1 - T_2}{L_c}\right) \qquad \text{.....(1)}$$

Since $h = \frac{kNu}{L}$

The rate of steady heat conduction across the layer of thickness L_c and A_s, and thermal conductivity k is expressed as

$$\dot{Q} = kA_s\left(\frac{T_1 - T_2}{L_c}\right) \qquad \text{.....(2)}$$

Where

T_1 and T_2 Is the temperature on the both sides of the layer.

Step 2 of 3

A compression the relation 2 with 1 is reveals that the convention heat transfer in an enclosure is analogue to heat conduction across the fluid layer in the enclosure provided that the thermal conductivity k is replaces with kNu .it is due to convention currents. So the quantity kNu is called effective thermal conductivity.

Step 3 of 3

The effective thermal conductivity of the rectangular enclosure represents the product of the thermal conductivity and the Nusselt number as a result of convection currents.
The ratio of effective thermal conductivity to thermal conductivity equals to the Nusselt number.

$$\frac{k_{eff}}{k} = Nu$$

Question: 20.056P

Consider a 3-m high rectangular enclosure consisting of two surfaces separated by a 0.1-m air gap at 1 atm. If the surface temperatures across the air gap are 30°C and -10°C, determine the ratio of the heat transfer rate for the horizontal orientation (with hotter surface at the bottom) to that for vertical orientation.

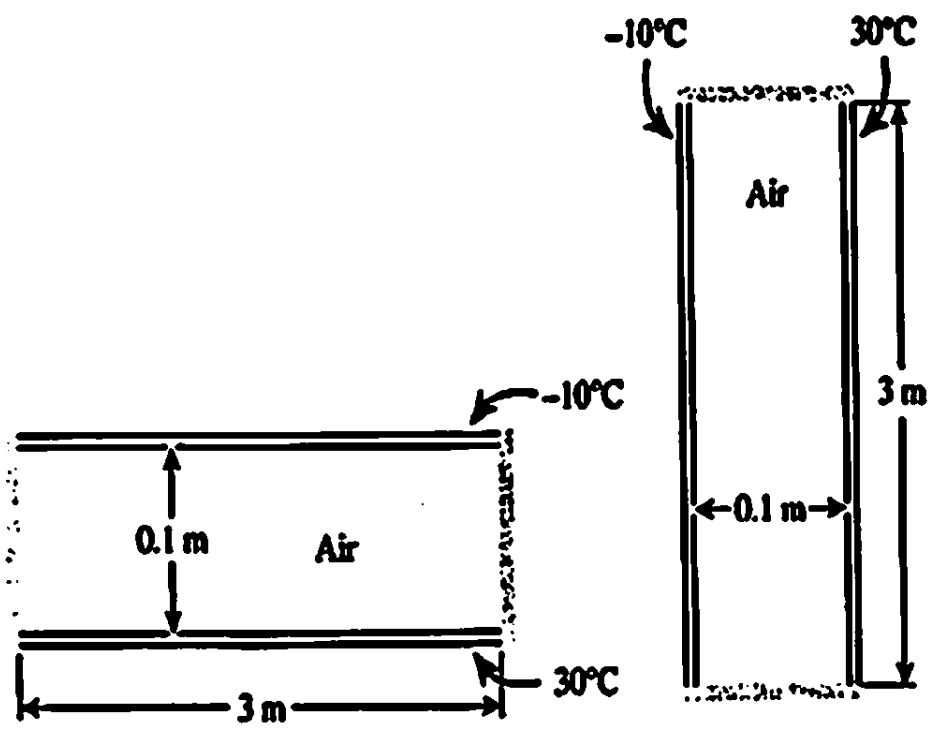


FIGURE P20–56

Step 1 of 5

Calculate the average temperature of two surfaces by using the equation,

$$T_f = \frac{T_{s1} + T_{s2}}{2}$$

Here, T_{s1} and T_{s2} are the surface temperatures.

Substitute $30°C$ for T_{s1} and $-10°C$ for T_{s2}

$$T_f = \frac{-10+30}{2}$$
$$= 10°C$$

Find coefficient of volume expansion by using the equation,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{10+273}$$
$$= 0.00353$$

Obtain the properties of air at 1atm and $10°C$ from properties of air table,

Thermal conductivity, $k = 0.02288\ \text{W/m°·C}$

Prandtl number, $\text{Pr} = 0.7387$

Kinematic viscosity, $\nu = 1.252\times10^{-5}\ \text{m}^2/\text{s}$

Step 2 of 5

Find the Rayleigh number when the plates are placed in horizontal position,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$

Here, β is coefficient of volume expansion, L_c is characteristic length

Substitute 0.1 m for L_c . 9.81m/s^2 for g

$$\text{Ra}_L = \frac{g\beta\Delta T L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times0.00353\times[30-(-10)]\times0.1^3}{(1.252\times10^{-5})^2}\times0.7387$$
$$= 6.5277\times10^6\ (<10^8)$$

For horizontal position, calculate the Nusselt number by using the expression,

$$\text{Nu}_h = 1+1.44\left[1-\frac{1708}{\text{Ra}_L}\right]^+ + \left[\frac{\text{Ra}_L^{1/3}}{18}-1\right]^+$$
$$= 1+1.44\left[1-\frac{1708}{6.5277\times10^6}\right]^+ + \left[\frac{(6.5277\times10^6)^{1/3}}{18}-1\right]^+$$
$$= 1+1.44\times0.999+9.377$$
$$= 11.81$$

Step 3 of 5

For vertical orientation, find the aspect ratio,

$$\frac{H}{L}$$

Here, H is height of the vertical plate and L is distance between two plates.

Substitute 3 m for H and 0.1 m for L

$$\frac{H}{L} = \frac{3}{0.1}$$
$$= 30$$

For vertical position, find the Nusselt number at the aspect ratio $\frac{H}{L} = 30$

$$\text{Nu} = 0.42\text{Ra}_L^{1/4}\,\text{Pr}^{0.012}\left(\frac{H}{L}\right)^{-0.3}$$
$$= 0.42\times(6.5277\times10^6)^{1/4}\times0.7387^{0.012}\times30^{-0.3}$$
$$= 7.624$$

Step 4 of 5

Find the heat transfer coefficient in horizontal position,

$$h_h = \frac{\text{Nu}_h\times k}{L_c}$$
$$= \frac{11.81\times0.02288}{0.1}$$
$$= 2.7021\ \text{W/m}^2\cdot°\text{C}$$

Heat transfer coefficient for vertical position,

$$h_v = \frac{\text{Nu}_v\times k}{L_C}$$
$$= \frac{7.624\times0.02288}{0.1}$$
$$= 1.744\ \text{W/m}^2\cdot°\text{C}$$

Find the heat transfer in horizontal position by using the equation,

$$Q_h = h_h A_s(T_{s1}-T_{s2})$$
$$= 2.7021\times A_s(30-(-10))$$
$$= 108.084A_s\ \text{W}$$

Find the heat transfer in vertical position by using the equation,

$$Q_v = h_v A_s(T_{s1}-T_{s2})$$
$$= 1.744\times A_s(30-(-10))$$
$$= 69.76A_s$$

Step 5 of 5

Find the ratio of heat transfer between horizontal orientations to the vertical orientation of the surfaces,

$$\frac{\dot{Q}_h}{\dot{Q}_v} = \frac{108.084A_s}{69.76A_s}$$
$$= 1.54$$

Hence the ratio of heat transfer rate for horizontal to vertical is $\boxed{\frac{\dot{Q}_h}{\dot{Q}_v} = 1.54}$

Question: 20.057P

Two concentric spheres with radii of 5 cm and 10 cm are having the surface temperatures maintained at 100°C and 200°C, respectively. The enclosure between the two concentric spherical surfaces is filled with nitrogen gas at 1 atm. Determine the rate of heat transfer through the enclosure.

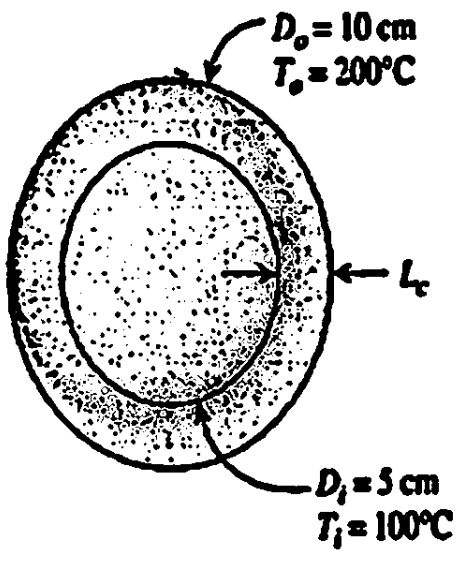


FIGURE P20–57

Step 1 of 6

Given data:

Diameter of outer sphere, $D_0 = 10\,\text{cm} = 0.1\text{m}$

Diameter of inner sphere, $D_i = 5\,\text{cm} = 0.05\text{m}$

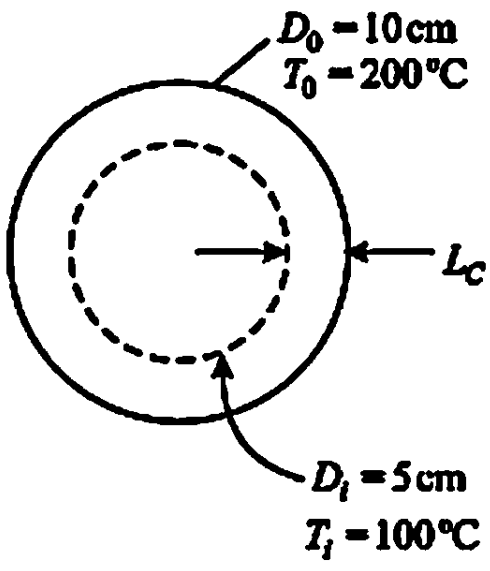
Temperature of outer surface, $T_o = 200°\text{C}$

Temperature of inner surface, $T_i = 100°\text{C}$

Pressure of Nitrogen filled in the sphere, $P = 1\,\text{atm}$

Step 2 of 6

Given figure:



Step 3 of 6

Characteristic length,

$$L_C = \frac{D_0 - D_i}{2}$$
$$= \frac{10-5}{2}$$
$$= 2.5\,\text{cm}$$
$$= 0.025\text{m}$$

Film temperature of Nitrogen,

$$T_f = \frac{T_o + T_i}{2}$$
$$= \frac{200+100}{2}$$
$$= 150°\text{C}$$

Step 4 of 6

Properties of Nitrogen at 1 atm and 150°C

Thermal conductivity, $k = 0.03416\,\text{w/m °C}$

Kinematic viscosity, $v = 2.851\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7025$

Now Rayleigh number,

$$Ra_L = \frac{g\beta(T_o - T_i)L_C^3}{v^2}\cdot\text{Pr}$$
$$= \frac{g(1/T_f)(T_o - T_i)L_C^3}{v^2}\cdot\text{Pr}$$
$$= \frac{9.81(200-100)\times0.025^3}{(150+273)\left(2.851\times10^{-5}\right)^2}\times(0.7025)$$
$$= 3.1317\times10^4$$

Step 5 of 6

Effective thermal conductivity,

$$F_{sph} = \frac{L_C}{(D_iD_0)^4\left(D_i^{-7/5} + D_0^{-7/5}\right)^5}$$
$$= \frac{0.025}{(0.1\times0.05)^4\left((0.05)^{-7/5} + (0.1)^{-7/5}\right)^5}$$
$$= 6.2682\times10^{-3}$$
$$k_{eff} = 0.74k\left(\frac{\text{Pr}}{0.861+\text{Pr}}\right)^{1/4}\left(F_{sph}Ra_L\right)^{1/4}$$
$$= 0.74\times0.03416\left(\frac{0.7025}{0.861+0.7025}\right)^{1/4}(6.2682\times3.1317\times10)^{1/4}$$
$$= 0.07747\,\text{W/m}\cdot\text{K}$$

Step 6 of 6

For concentric spheres, heat transfer through gap by natural convection is

$$\dot{Q} = k_{eff}\left(\frac{\pi D_iD_0}{L_C}\right)(T_o - T_i)$$
$$= 0.07747\times\left(\frac{\pi\times0.05\times0.1}{0.025}\right)(200-100)$$
$$= 4.867\ \text{W}$$

Hence the heat transfer through gap is $\boxed{\dot{Q} = 4.867\ \text{W}}$

Question: 20.058P

Show that the thermal resistance of a rectangular enclosure can be expressed as $R = L_c/(Ak\,\mathrm{Nu})$, where k is the thermal conductivity of the fluid in the enclosure.

Step 1 of 2

The rate of heat transfer through the enclosure is

$$\dot{Q} = hA_s(T - T_2)$$

$$= kNuA_s\left(\frac{T_1 - T_2}{L_c}\right) \qquad \ldots\ldots(1)$$

Since $h = \frac{kNu}{L}$

The rate of steady heat conduction across the layer of thickness L_c and A_s, and thermal conductivity k is expressed as

$$\dot{Q} = kA_s\left(\frac{T_1 - T_2}{L_c}\right) \qquad \ldots\ldots(2)$$

Where

T_1 and T_2 Is the temperature on the both sides of the layer.

Step 2 of 2

A compression the relation 2 with 1 is reveals that the convention heat transfer in an enclosure is analogue to heat conduction across the fluid layer in the enclosure provided that the thermal conductivity k is replaces with kNu .it is due to convention currents.

$$\dot{Q} = hA_s(T_s - T_\infty)$$

$$= kNuA_s\frac{(T_s - T_\infty)}{L_c}$$

$$= \frac{(T_s - T_\infty)}{\left(\frac{L_c}{kNuA_s}\right)}$$

From the above relation thermal resistance through enclosure is

$$R = \left(\frac{L_c}{kNuA_s}\right)$$

$$\boxed{R = \left(\frac{L_c}{kNuA_s}\right)}$$

Question: 20.059P

A vertical 1.5-m-high and 3.0-m-wide enclosure consists of two surfaces separated by a 0.4-m air gap at atmospheric pressure. If the surface temperatures across the air gap are measured to be 280 K and 336 K and the surface emissivities to be 0.15 and 0.90. determine the fraction of heat transferred through the enclosure by radiation.

Step 1 of 7

Height of enclosure, $H = 1.5\,\text{m}$

Width of enclosure, $w = 3.0\,\text{m}$

Air gap between surfaces, $L_C = 0.4\,\text{m}$

Temperature of surface-1, $T_1 = 336\,\text{K}$

Temperature of surface-2, $T_2 = 280\,\text{K}$

Emissivity of surface-1, $\varepsilon_1 = 0.90$

Emissivity of surface-2, $\varepsilon_2 = 0.15$

Step 2 of 7

The average temperature,

$$T_{avg} = \frac{(T_1 + T_2)}{2}$$
$$= \frac{(336 + 280)}{2}$$
$$= 308\,\text{K}$$

Considering the properties of Air at 1atm and 308 K,

Thermal conductivity, $k = 0.02625\ \text{W/m}^\circ\text{C}$

Prandtl number, $\text{Pr} = 0.7268$

Kinematic viscosity, $v = 1.655 \times 10^{-5}\ \text{m}^2/\text{s}$

Characteristic length, L_C = Distance between two surfaces $= 0.4\,\text{m}$

Step 3 of 7

Now Rayleigh number,

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2} \cdot \text{Pr}$$
$$= \frac{g(T_s - T_\infty)L_c^3}{(T_f) \times (v^2)} \cdot \text{Pr}$$
$$= \frac{9.81 \times (336 - 280)(0.4)^3}{308 \times \left(1.655 \times 10^{-5}\right)^2} \times (0.7268)$$
$$= 3.029 \times 10^8$$

Step 4 of 7

As $\text{Ra}_L < 10^{10}$, the relation for Nusselt number will be

$$\text{Nu} = 0.22\left(\frac{\text{Pr}}{0.22 + \text{Pr}} \times \text{Ra}_L\right)^{0.28}\left(\frac{H}{2}\right)^{-1/4}$$
$$= 0.22\left(\frac{0.7268}{0.22 + 0.7268} \times 3.029 \times 10^8\right)^{0.28}(3.75)^{-0.25}$$
$$= 34.795$$

Heat transfer coefficient,

$$h = \frac{k}{L_C} \times \text{Nu}$$
$$= \frac{0.02625}{0.4} \times 34.795$$
$$= 2.283\ \text{W/m}^2 \cdot ^\circ\text{C}$$

Step 5 of 7

Surface area,

$$A_S = H \times w$$
$$= 1.5 \times 3.0$$
$$= 4.5\ \text{m}^2$$

Rate of Heat transfer by convection,

$$Q_{conv} = hA_S(T_1 - T_2)$$
$$= 2.283 \times 4.5(336 - 280)$$
$$= 575.32\ \text{W}$$

Step 6 of 7

Effective emissivity,

$$\varepsilon_{eff} = \frac{1}{(1/\varepsilon_1) + (1/\varepsilon_2) - 1}$$
$$= \frac{1}{(1/0.9) + (1/0.15) - 1}$$
$$= 0.1475$$

Rate of Heat transfer by Radiation,

$$Q_{rad} = \varepsilon_{eff}\sigma A_S\left(T_1^4 - T_2^4\right)$$
$$= 0.1475 \times \left(5.67 \times 10^{-8}\right) \times 4.5\left(336^4 - 280^4\right)$$
$$= 248.35\ \text{W}$$

Step 7 of 7

Total heat transfer,

$$Q_{tot} = Q_{conv} + Q_{rad}$$
$$= 575.32 + 248.35$$
$$= 823.67\ \text{W}$$

Radiation heat transfer fraction in total heat transfer,

$$\text{Fraction} = \frac{Q_{rad}}{Q_{tot}}$$
$$= \frac{248.35}{823.67}$$
$$= 0.3015$$

Hence fraction of radiation heat transfer in total heat transfer is $\boxed{\text{Fraction} = 0.3015}$

Question:

A vertical 1.5-m-high, 2.8-m-wide double-pane window consists of two layers of glass separated by a 2.0-cm air gap at atmospheric pressure. The room temperature is 26°C while the inner glass temperature is 18°C. Disregarding radiation heat transfer, determine the temperature of the outer glass layer and the rate of heat loss through the window by natural convection.

Step 1 of 9

Given data:

Height of enclosure, $H = 1.5\text{m}$

Width of enclosure, $w = 2.8\text{ m}$

Air gap between surfaces, $L_c = 20\text{ cm} = 0.02\text{m}$

Temperature of the inner glass, $T_1 = 18°\text{C}$

Temperature of room, $T_\infty = 26°\text{C}$

Step 2 of 9

Film temperature,

$$T_f = \frac{T_1 + T_\infty}{2}$$
$$= \frac{18+26}{2}$$
$$= 22°\text{C}$$

The volume expansion coefficient is given by,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{22+273}$$
$$= 3.390\times10^{-3}\,\text{K}^{-1}$$

Characteristic length,

$$L_c = H$$
$$= 1.5\text{m}$$

Step 3 of 9

Properties of air at 1atm and film temperature, by interpolation,

Thermal conductivity,

$$k = 0.02514 + \frac{0.02551-0.02514}{25-20}(22-20)$$
$$= 0.025288\ \text{W/m}\cdot\text{K}$$

Prandtl Number,

$$\text{Pr} = 0.7309 + \frac{0.7296-0.7309}{25-20}(22-20)$$
$$= 0.7305$$

Kinematic viscosity,

$$\nu = 1.516\times10^{-5} + \frac{(1.562-1.516)10^{-5}}{(25-20)}(22-20)$$
$$= 1.5344\times10^{-5}\,\text{m}^2/\text{s}$$

Step 4 of 9

Rayleigh number id given by,

$$\text{Ra}_L = \frac{g\beta(T_S - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.390\times10^{-3}(26-18)(1.5)^3}{\left(1.5344\times10^{-5}\right)^2}\times0.7305$$
$$= 2.786\times10^9$$

Nusselt Number is given by,

$$\text{Nu} = \left[0.825 + \frac{0.387\left(\text{Ra}_L\right)^{1/6}}{\left[1+\left(0.492/\text{Pr}\right)^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387\left(2.786\times10^9\right)^{1/6}}{\left[1+\left(0.492/0.7305\right)^{9/16}\right]^{8/27}}\right]^2$$
$$= 169.524$$

Step 5 of 9

Heat transfer coefficient is given by,

$$h = \frac{k}{L_c}\text{Nu}$$
$$= \frac{0.025288}{1.5}\times169.524$$
$$= 2.858\ \text{W/m}^2\cdot\text{K}$$

Surface area of the plate is given by,

$$A_s = H\times w$$
$$= 1.5\times2.8$$
$$= 4.2\,\text{m}^2$$

Rate of heat transfer from room to inner glass surface is given by,

$$\dot{Q}_{conv} = hA_s(T_1 - T_\infty)$$
$$= 2.858\times(4.2)(26-18)$$
$$= 96.028\,\text{W}$$

Step 6 of 9

Now, considering convection between two glass surfaces,

Let us assume the outer glass temperature, $T_2 = 2°\text{C}$,

The properties of Air, at pressure 1 atm and average temperature,

Thermal conductivity, $k = 0.02439\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.426\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7336$

Characteristic length, $L_c = L = 0.020\text{m}$

The properties of Air, at pressure 1 atm and average temperature,

Thermal conductivity, $k = 0.02439\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 1.426\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7336$

Characteristic length, $L_C = L = 0.020\text{m}$

Average temperature is given by,

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{18+2}{2}$$
$$= 10^\circ\text{C}$$

Step 7 of 9

The volume expansion coefficient is given by,

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{273+10}$$
$$= 3.534\times10^{-3}\ \text{K}^{-1}$$

Rayleigh number is given by,

$$\text{Ra}_L = \frac{g\beta(T_1 - T_2)L_C^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.534\times10^{-3}(18-2)(0.020)^3}{\left(1.426\times10^{-5}\right)^2}(0.7336)$$
$$= 16009.06$$

Step 8 of 9

Nusselt Number is given by,

$$\text{Nu} = 0.42\ \text{Ra}_L^{1/4}\ \text{Pr}^{0.012}\left(\frac{H}{L}\right)^{-0.3}$$
$$= 0.42\times(16009.06)^{0.25}\times(0.7336)^{0.012}\times\left(\frac{1.5}{0.020}\right)^{-0.3}$$
$$= 1.2888$$

Heat transfer coefficient is given by,

$$h = \frac{k}{L_C}\times\text{Nu}$$
$$= \frac{0.02439}{0.02}\times1.2888$$
$$= 1.572\ \text{W/m}^2{}^\circ\text{C}$$

Step 9 of 9

Rate of heat transfer between two glass surfaces is given by,

$$\dot{Q} = hA_s(T_1 - T_2)$$
$$\dot{Q} = 1.572\times4.2(18 - T_2)$$

As the heat transfer rate between room and inner glass surface is equal to heat transfer rate between glass surfaces. Therefore,

$$\dot{Q}_{conv} = \dot{Q}$$
$$96.028 = 1.572\times4.2(18 - T_2)$$
$$T_2 = 3.456^\circ\text{C}$$

Therefore, the outer glass surface temperature is $\boxed{T_2 = 3..456^\circ\text{C}}$

Question: 20.061P

Two concentric spheres of diameters 15 cm and 25 cm are separated by air at 1 atm pressure. The surface temperatures of the two spheres enclosing the air are $T1$, = 350 K and $T2$ =275 K, respectively. Determine the rate of heat transfer from the inner sphere to the outer sphere by natural convection.

Step 1 of 6

Diameter of outer sphere, $D_0 = 25\text{cm} = 0.25\text{m}$

Diameter of inner sphere, $D_i = 15\text{cm} = 0.15\text{m}$

Temperature of inner sphere, $T_1 = 350\text{ K}$

Temperature of Outer sphere, $T_2 = 275\text{ K}$

Step 2 of 6

Average temperature is given by,

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{350 + 275}{2}$$
$$= 312.5\text{ K}$$
$$= 39.5^\circ\text{C}$$

The volume expansion coefficient is given by,

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{312.5}$$
$$= 3.2\times10^{-3}\text{ K}^{-1}$$

Step 3 of 6

Properties of air at pressure 1 atm and average temperature by interpolation,

Thermal conductivity,

$$k = (0.02625) + \left(\frac{0.02662 - 0.02625}{40 - 35}\right)(39.5 - 35)$$
$$= 0.026583\text{ W/m}^\circ\text{C}$$

Prandtl Number,

$$\text{Pr} = (0.7268) + \left(\frac{0.7255 - 0.7268}{40 - 35}\right)(39.5 - 35)$$
$$= 0.72563$$

Kinematic viscosity,

$$\nu = \left(1.655\times10^{-5}\right) + \frac{(1.702 - 1.655)10^{-5}}{(40 - 35)}(39.5 - 35)$$
$$= 1.6973\times10^{-5}\text{ m}^2/\text{s}$$

Step 4 of 6

Characteristic length is given by,

$$L_C = \frac{D_0 - D_i}{2}$$
$$= \frac{0.25 - 0.15}{2}$$
$$= 0.05\text{ m}$$

Rayleigh Number is given by,

$$\text{Ra}_L = \frac{g\beta(T_1 - T_2)L_C^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.2\times10^{-3}(350 - 275)(0.05)^3}{\left(1.6973\times10^{-5}\right)^2}\times0.72563$$
$$= 741290.218$$

Step 5 of 6

Geometric factor is given by,

$$F_{sph} = \frac{L_C}{(D_i D_0)^4\left(D_i^{-7/5} + D_0^{-7/5}\right)^5}$$
$$= \frac{0.05}{(0.15\times0.25)^4\left(0.15^{-7/5} + 0.25^{-7/5}\right)^5}$$
$$= 0.0058998$$

Effective heat transfer coefficient is given by,

$$k_{eff} = 0.74\left(\frac{\text{Pr}}{0.861 + \text{Pr}}\right)^{1/4}\left(F_{sph}\text{Ra}_L\right)^{1/4}\cdot k$$
$$= 0.74\left(\frac{0.72563}{0.861 + 0.72563}\right)^{0.25}\times\left[(0.0058998)\times(741290.218)\right]^{0.25}\times0.026583$$
$$= 0.1315\text{ W/m}^2{}^\circ\text{C}$$

Step 6 of 6

Rate of heat transfer from inner to outer sphere can be expressed by the equation.

$$\dot{Q}_{conv} = k_{eff}\frac{\pi D_i D_0}{L_C}(T_1 - T_2)$$
$$= 0.1315\frac{(\pi\times0.25\times0.15)}{0.05}(350 - 275)$$
$$= 23.24\text{ W}$$

Therefore, the rate of heat transfer is $\boxed{\dot{Q}_{conv} = 23.24\text{ W}}$

Question: 20.062P

Reconsider Prob. 20–61. Using EES (or other) software, plot the rate of natural convection heat transfer as a function of the hot surface temperature of the sphere as the temperature varies from 250 K to 450 K, and discuss the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

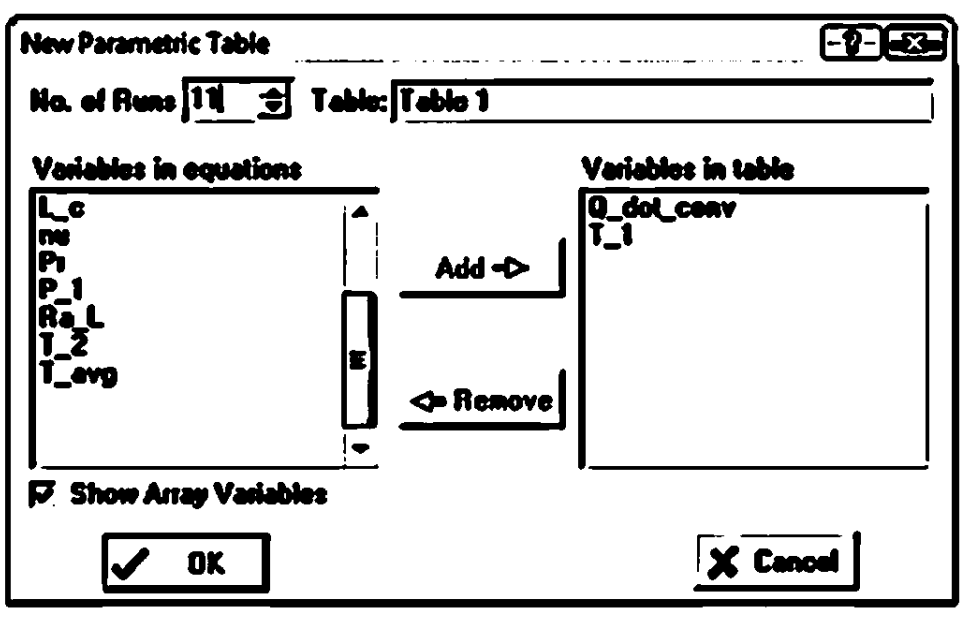
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"Given Data"
D_o=0.25[m]                          "Diameter of outer sphere"
D_i=0.15[m]                          "Diameter of inner sphere"
"T_1=350[K]"                         "Temperature of inner sphere"
T_2=275[K]                           "Temperature of outer sphere"
P_1=1[atm]*convert(atm,kPa)          "Pressure of air"

T_avg=converttemp(K,C,((T_1)+(T_2))/2)
                                     "Average Temperature"
beta=1/(converttemp(C,K,T_avg))      "Volume expansion"
k=Conductivity(Air,T=T_avg)          "properties of air at T_avg temperature"
nu= Viscosity(Air,T=T_avg)/ Density(Air,T=T_avg,P=P_1)
Pr=Prandtl(Air,T=T_avg)
L_c=(D_o-D_i)/2                      "Characteristic Length"
Ra_L=g#*beta*abs(T_1-T_2)*L_c^3*Pr/nu^2
                                     "Rayleigh Number"
F_sph=L_c/((D_i*D_o)^4*(D_i^(-7/5)+D_o^(-7/5))^5)
                                     "Geometric Factor"
k_eff=0.74*(Pr/(0.861+Pr))^0.25*(F_sph*Ra_L)^0.25*k
                                     "Effective heat transfre coefficient"
Q_dot_conv=k_eff*pi*D_i*D_o*(T_1-T_2)/L_c
                                     "Rate of heat Transfer"
```

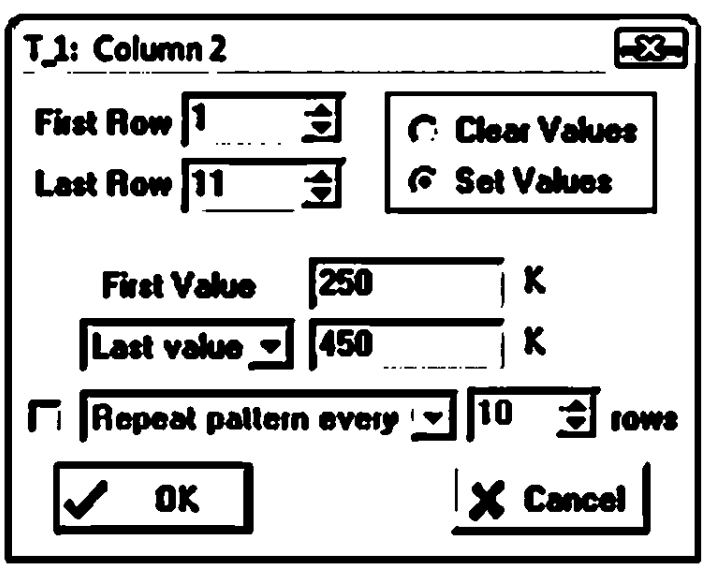
Step 2 of 5

Now to solve the equation for a range of hot surface temperature values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "Q_dot_conv" and "T_1" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 5

To enter the values in the "T_1" column click on the drop tab in "T_1" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 5

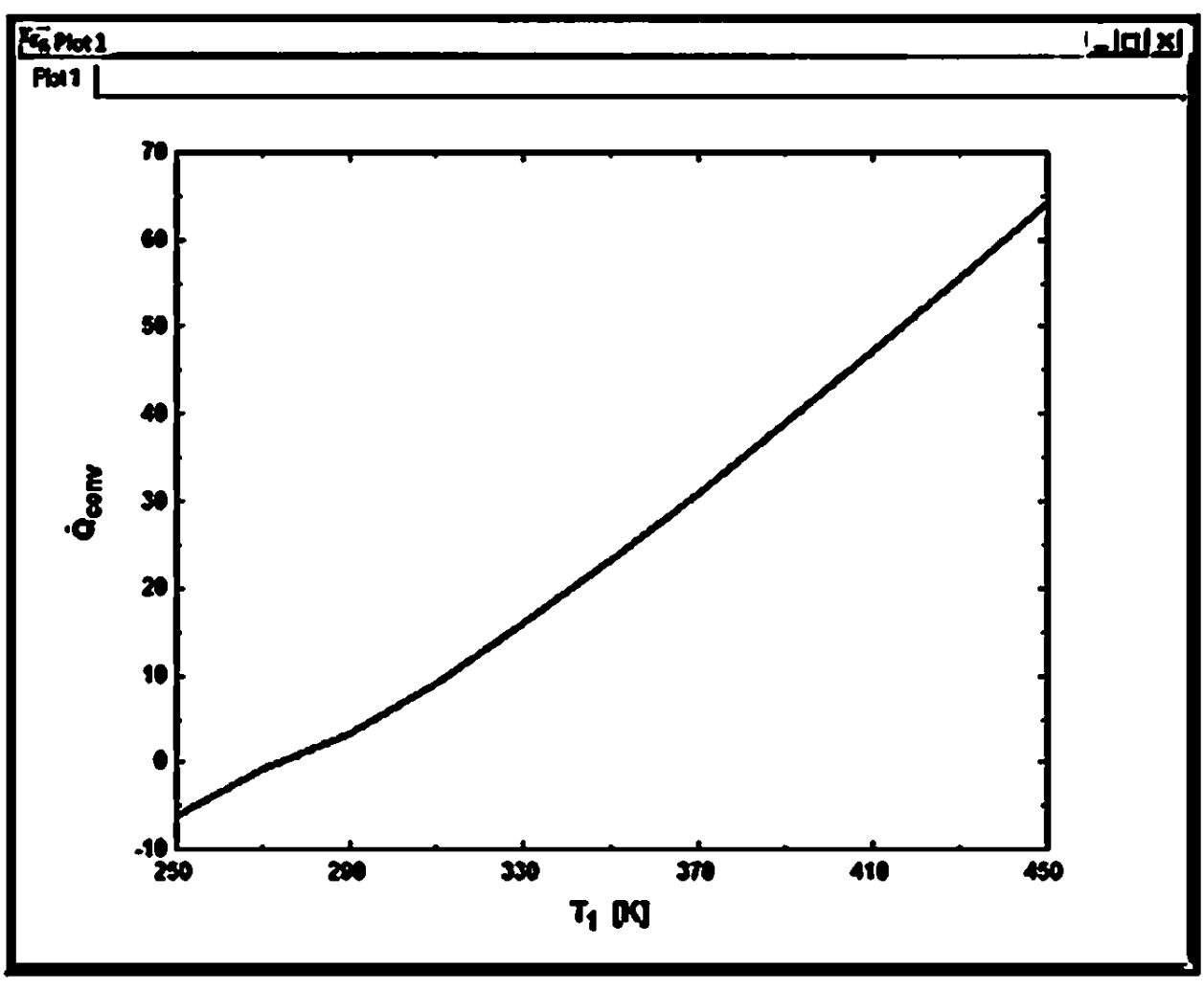
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	$\dot{Q}_{conv}$ [1]	T_1 [K] [2]
Run 1	-6.198	250
Run 2	-0.8198	270
Run 3	3.202	290
Run 4	9.14	310
Run 5	15.92	330
Run 6	23.24	350
Run 7	30.94	370
Run 8	38.93	390
Run 9	47.15	410
Run 10	55.57	430
Run 11	64.14	450

Step 5 of 5

Inorder to plot the rate of natural convection heat transfer as a function of hot surface temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as "T_1" and Y-axis as "Q_dot_conv" and press 'OK'
The Graph will be plotted as shown below:



From the above graph we can interpret that as the temperature of the hot surface increases, the rate of convection heat transfer also increases.

Question: 20.063P

Flat-plate solar collectors are often tilted up toward the sun in order to intercept a greater amount of direct solar radiation, The tilt angle from the horizontal also affects the rate of heat loss from the collector. Consider a 1.5-m-high and 3-m-wide solar collector that is tilted at an angle θ from the horizontal. The back side of the absorber is heavily insulated. The absorber plate and the glass cover, which are spaced 2.5 cm from each other, are maintained at temperatures of 80°C and 40°C, respectively. Determine the rate of heat loss from the absorber plate by natural convection for θ =0°, 30°, and 90°.

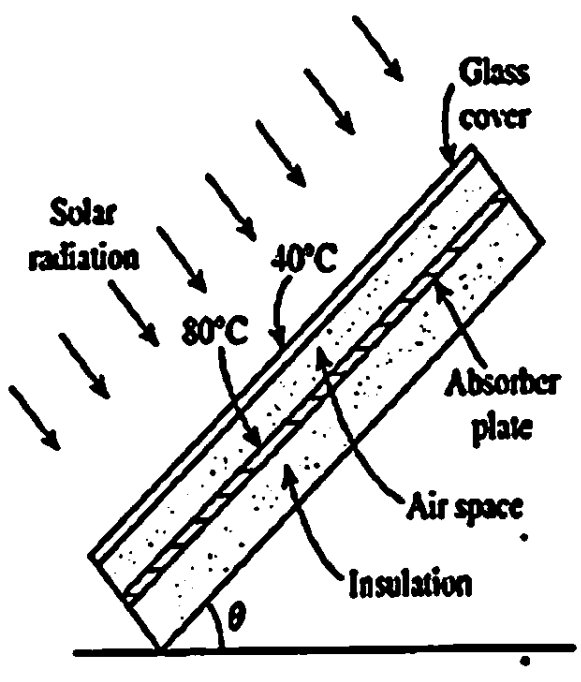


FIGURE P20-63

Step 1 of 8

Height of collector, $H = 1.5\text{m}$

Width of collector, $w = 3\text{m}$

Temperature of absorber plate, $T_1 = 80°\text{C}$

Temperature of Glass plate, $T_2 = 40°\text{C}$

Space between the plates, $L = 2.5\text{ cm} = 0.025\text{m}$

Step 2 of 8

At inclination of $\theta = 0°$:

The set up will become horizontal enclosure.

Average temperature,

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{40 + 80}{2}$$
$$= 60^0\text{C}$$

Inverse of temperature,

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{60 + 273}$$
$$= 3.003 \times 10^{-3}\ \text{K}^{-1}$$

Step 3 of 8

Properties of Air at 1atm & average temperature,

Thermal conductivity, $k = 0.02808\ \text{w/m°C}$

Prandtl number, $P_r = 0.7202$

Kinematic viscosity, $v = 1.896 \times 10^{-5}\ \text{m}^2/\text{s}$

Characteristic length $L_C = 0.025\text{m}$

Area of the surface,

$$(A_S) = H \times w$$
$$= 1.5 \times 3$$
$$= 4.5\ \text{m}^2$$

Step 4 of 8

Rayleigh number,

$$Ra_L = \frac{g\beta(T_1 - T_2)L_C^3}{v^2} \cdot \text{Pr}$$
$$= \frac{9.81 \times 3.003 \times 10^{-3} \times (80 - 40) \times (0.025)^3}{\left(1.896 \times 10^{-5}\right)^2} \times (0.7202)$$
$$= 36887.63$$

Nusselt number,

$$Nu = 1 + 1.44\left[1 - \frac{1708}{Ra_L}\right]^+ + \left[\frac{Ra_L^{1/3}}{18} - 1\right]^+$$
$$= 1 + 1.44\left[1 - \frac{1708}{36887.63}\right]^+ + \left[\frac{(36887.63)^{1/3}}{18} - 1\right]^+$$
$$= 3.223$$

Step 5 of 8

Now heat transfer,

$$\dot{Q}_{conv} = k\,Nu\,A_S\frac{T_1 - T_2}{L_C}$$
$$= \frac{0.02808 \times 3.223 \times 4.5 \times (80 - 40)}{0.025}$$
$$= 651.5\ \text{W}$$

Hence the rate of heat loss is $\boxed{Q_{conv} = 651.5\ \text{W}}$

Step 6 of 8

At inclination of $\theta^o = 30°$:

Nusselt number,

$$Nu = 1 + 1.44\left[1 - \frac{1708}{R_{a_L}\cos\theta}\right]^+\left(1 - \frac{1708(\sin 1.8\theta)^{1.6}}{R_{a_L}\cos\theta}\right) + \left[\frac{(R_{a_L}\cos\theta)^{1/3}}{18} - 1\right]^+$$
$$= 1 + 1.44\left[1 - \frac{1708}{36887.6\cos 30°}\right]\left(1 - \frac{1708(\sin 1.8 \times 30)^{1.6}}{36887.6\cos 30°}\right) + \left[\frac{(36887.6\cos 30°)^{1/3}}{18} - 1\right]$$
$$= 3.092$$

Step 7 of 8

Now heat transfer,

$$\dot{Q}_{conv} = k\,Nu\,A_S\frac{T_1 - T_2}{L_C}$$
$$= \frac{0.02808 \times 3.092 \times 4.5 \times (80 - 40)}{0.025}$$
$$= 625.2\ \text{W}$$

Hence the rate of heat loss is $\boxed{Q_{conv} = 625.2\ \text{W}}$

Step 8 of 8

At inclination of $\theta = 90°$:

Set up will be vertical rectangular enclosure. Therefore Nusselt number, Nu for vertical enclosure based on Ra_L calculated is

$$Nu = 0.42R_{a_L^{0.25}}P_r^{0.012}\left(\frac{H}{C}\right)^{-0.3}$$
$$= 0.42Ra_L^{0.25}P_r^{0.012}\left(\frac{1.5}{0.025}\right)^{-0.3}$$
$$= 1.6975$$

Now heat transfer,

$$\dot{Q}_{conv} = k\,Nu\,A_S\frac{T_1 - T_2}{L_C}$$
$$= \frac{0.02808 \times 1.6975 \times 4.5 \times (80 - 40)}{0.025}$$
$$= 343.2\ \text{W}$$

Hence the rate of heat loss is $\boxed{Q_{conv} = 343.2\ \text{W}}$

Question:

A simple solar collector is built by placing a 5-cm-diameter clear plastic lube around a garden hose whose outer diameter is 1.6 cm. The hose is painted black to maximize solar absorption, and some plastic rings are used to keep the spacing between the hose and the clear plastic cover constant. During a clear day, the temperature of the hose is measured to be 65°C, while the ambient air temperature is 26°C. Determine the rate of heat loss from the water in the hose per meter of its length by natural convection. Also, discuss how the performance of this solar collector can be improved.

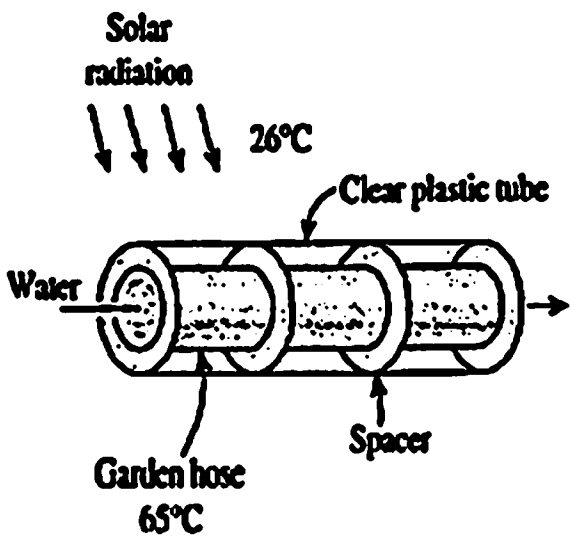


FIGURE P20–64

Step 1 of 10

Diameter of plastic tube, $D_0 = 5\text{cm} = 0.05\text{m}$

Diameter of garden Hose, $D_i = 1.6\text{cm} = 0.016\text{m}$

Ambient Air temperature, $T_\infty = 26°\text{C}$

Temperature of Garden Hose, $T_1 = 65°\text{C}$

Step 2 of 10

Let us assume the temperature of clear plastic tube to be $T_2 = 35°$

Average temperature,

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{65+35}{2}$$
$$= 50°\text{C}$$

Inverse of temperature,

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{323}$$
$$= 3.0960\times10^{-3}\,\text{K}^{-1}$$

Step 3 of 10

The properties of air at average temperature,

Thermal conductivity, $k = 0.02735\,\text{w/m}°\text{C}$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\,\text{m}^2/\text{s}$

Prandtl number, $P_r = 0.7228$

Rayleigh number,

$$Ra_L = \frac{g\beta(T_1 - T_2)L_C^3}{\nu^2}.\text{Pr}$$
$$= \frac{9.81\times3.096\times10^{-3}(65.35)(0.017)^3}{\left(1.798\times10^{-5}\right)^2}\times(0.7228)$$
$$= 10008.6$$

Step 4 of 10

Characteristic length

$$L_C = \frac{\Delta_0 - \Delta_i}{2}$$
$$= \frac{0.05 - 0.016}{2}$$
$$= 0.017\,\text{m}$$

Geometric factor,

$$F_{cyl} = \frac{\left[\ln(\Delta_0/\Delta_i)\right]^4}{L_C^3\left(\Delta_i^{-3/5} + \Delta_0^{-3/5}\right)^2}$$
$$= \frac{\left[\ln(0.05/0.016)\right]^4}{(0.017)^3\left[(0.016)^{-3/5} + (0.05)^{-3/5}\right]^5}$$
$$= 0.1821$$

Step 5 of 10

Effective thermal conductivity,

$$k_{eff} = 0.386\left(\frac{P_r}{0.861+P_r}\right)^{0.25}\left(F_{cyl}\cdot Ra_L\right)^{0.25}\times k$$
$$= 0.386\left(\frac{0.7228}{0.861+0.7228}\right)^{0.25}\times(0.1821\times10008.6)^{0.25}\times0.02735$$
$$= 0.31726\times6.5339\times0.02735$$
$$= 0.0567\ \text{W/m}^0\text{C}$$

Rate of heat loss through concentric cylinders can be expressed as

$$\dot{Q} = \frac{2\pi k_{eff}}{\ln(\Delta_0/\Delta_i)}(T_i - T_2)$$
$$\dot{Q} = \frac{2\pi\times0.0567}{\ln(0.05/0.016)}(65 - T_2)$$
$$\dot{Q} = 0.3126(65 - T_2) \qquad \ldots\ldots (1)$$

Step 6 of 10

Heat transfer from garden hose to plastic will be equal to heat transfer from plastic pipe to martinet air. So

By assuming the surface temperature to be $T_S = 35°$

Film temperature,

Heat transfer from garden hose to plastic will be equal to heat transfer from plastic pipe to martinet air. So

By assuming the surface temperature to be $T_S = 35°$

Film temperature,

$$T_f = \frac{T_S + T_\infty}{2}$$
$$= \frac{35+26}{2}$$
$$= 30.5°C$$

Temperature inverse,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{273+30.5}$$
$$= 3.2948\times10^{-3}\ K^{-1}$$

Step 7 of 10

At film temperature, the properties using interpolation are

Thermal conductivity

$$k = 0.02588 + \frac{0.02625-0.02588}{(35-30)}\times(30.5-30)$$
$$= 0.025917\ W/m°C$$

Prandtl number,

$$P_r = 0.7282 + \frac{0.7268-0.7282}{35-30}\times(30.5-30)$$
$$= 0.72806$$

Kinematic viscosity,

$$\nu = \left(1.608\times10^{-5}\right) + \frac{(1.655-1.608)10^{-5}}{35-30}\times(30.5-30)$$
$$= 1.6127\times10^{-5}\ m^2/s$$

Characteristic length, $L_C = D_0 = 0.05m$

Step 8 of 10

Rayleigh number

$$Ra_L = \frac{g\beta(T_S-T_\infty)L_C^3}{\nu^2}\cdot P_r$$
$$= \frac{9.81\times3.2948\times10^{-3}(35-26)(0.05)^3}{\left(1.6127\times10^{-5}\right)^2}\times(0.72806)$$
$$= 1.0179\times10^5$$

Nusselt number for horizontal cylinder,

$$Nu = \left[0.6 + \frac{0.387R_a^{1/6}}{\left[1+(0.559/P_r)^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{0.387\left(1.0179\times10^5\right)^{1/6}}{\left[1+(0.559/0.72806)^{9/16}\right]^{8/27}}\right]^2$$
$$= 7.8377$$

Step 9 of 10

Heat transfer coefficient,

$$h = \frac{k}{L_C}Nu$$
$$= \frac{0.025917}{0.05}\times7.8377$$
$$= 4.0626\ W/m^2°C$$

Surface area,

$$A_S = \pi\Delta_0 L$$
$$= \pi\times0.05\times1$$
$$= 0.15708\ m^2$$

Step 10 of 10

Now rate of heat transfer from pipe to outside air is

$$\dot{Q} = hA_S(T_S-T_\infty)$$
$$\dot{Q} = 4.0626\times0.15708(T_S-26)$$
$$\dot{Q} = 0.6382(T_S-26) \qquad \text{..... (2)}$$

From equations (1) and (2), we get

$$0.3126(65-T_2) = 0.6382(T_2-26)$$
$$(65-T_2) = \frac{0.6382}{0.3126}(T_2-26)$$
$$65-T_2 = 2.0416T_2 - 53.08$$
$$T_2 = 38.82°C$$

As the obtained temperature is very close to the assumed value, we can consider the heat transfer according to equation (1) as

$$\dot{Q} = 0.3126(65-38.82)$$
$$\dot{Q} = 8.183\ W$$

Hence the rate of heat loss is $\boxed{\dot{Q} = 8.183\ W}$

Question: 20.065P

Reconsider Prob. 20-64. Using EES (or other) eHsoftware, plot the rate of heat loss from the water by natural convection as a function of the ambient air temperature as the temperature varies from 4°C to 40°C, and discuss the results.

Step 1 of 3

Solve the problem with EES software by following the below steps.

1. **Open EES software**
2. **Crease a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in the new window.**

```
"GIVEN DATA"
D_1=0.016 [m]
D_2=0.05 [m]
T_1=65 "[C]"
"T_infinity=26 [C]"
Length=1 [m] " consider unit length of the tube"

"PROPERTIES for enclosure"
Fluid$='air'
k_1=Conductivity(Fluid$, T=T_ave)
Pr_1=Prandtl(Fluid$, T=T_ave)
rho_1=Density(Fluid$, T=T_ave, P=101.3)
mu_1=Viscosity(Fluid$, T=T_ave)
nu_1=mu_1/rho_1
beta_1=1/(T_ave+273)
T_ave=1/2*(T_1+T_2)
g=9.807 [m/s^2]
"ANALYSIS for enclosure"
L=(D_2-D_1)/2
Ra_1=(g*beta_1*(T_1-T_2)*L^3)/nu_1^2*Pr_1
F_cyl=(ln(D_2/D_1))^4/(L^3*(D_1^(-3/5)+D_2^(-3/5))^5)
k_eff=0.386*k_1*(Pr_1/(0.861+Pr_1))^0.25*(F_cyl*Ra_1)^0.25
Q_dot=(2*pi*k_eff)/ln(D_2/D_1)*(T_1-T_2)

"PROPERTIES for convection on the outer surface"
k_2=Conductivity(Fluid$, T=T_film)
Pr_2=Prandtl(Fluid$, T=T_film)
rho_2=Density(Fluid$, T=T_film, P=101.3)
mu_2=Viscosity(Fluid$, T=T_film)
nu_2=mu_2/rho_2
beta_2=1/(T_film+273)
T_film=1/2*(T_2+T_infinity)
"ANALYSIS for convection on the outer surface"
delta=D_2
Ra_2=(g*beta_2*(T_2-T_infinity)*delta^3)/nu_2^2*Pr_2
Nusselt=(0.6+(0.387*Ra_2^(1/6))/(1+(0.559/Pr_2)^(9/16))^(8/27))^2
h=k_2/delta*Nusselt
A=pi*D_2*Length
Q_dot=h*A*(T_2-T_infinity)
```

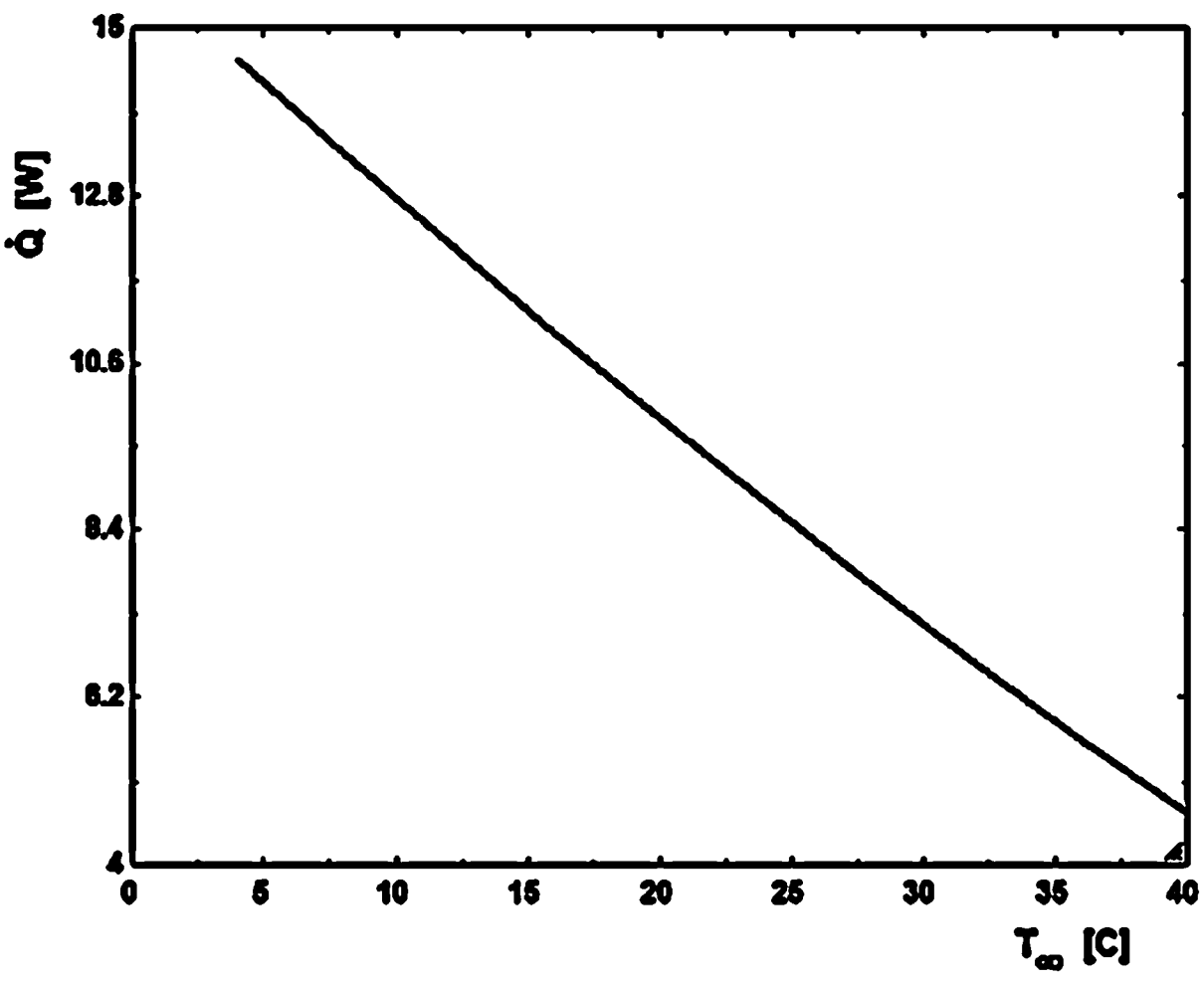
Step 2 of 3

Now to solve the equation for a range of 'temperature' values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as 'T_infinity' and 'Q_dot' press 'Add' and then press 'ok'
A table will appear and enter the values of "T_ infinity" as 4, 8,....40 in runs 1 to 10
Then from click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

▷ 1..10	1 T_∞ [C]	2 $\dot{Q}$ [W]
Run 1	4	14.6
Run 2	8	13.37
Run 3	12	12.18
Run 4	16	11.01
Run 5	20	9.871
Run 6	24	8.764
Run 7	28	7.688
Run 8	32	6.647
Run 9	36	5.641
Run 10	40	4.675

Step 3 of 3

Then click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as 'T_infinity' and Y-axis as 'Q_dot'. Press OK
The Graph will be plotted as shown below.



This graph represents that by increasing the ambient temperature, heat transfer rate will decrease.

Question:

A vertical 5-ft-high and 6-ft-wide double-pane window consists of two sheets of glass separated by a 0.8-in air gap at atmospheric pressure. If the glass surface temperatures across the air gap are measured to be 65°F and 40°F, determine the rate of heat transfer through the window by (*a*) natural convection and (*b*) radiation. Also, determine the *R*-value of insulation of this window such that multiplying the

.

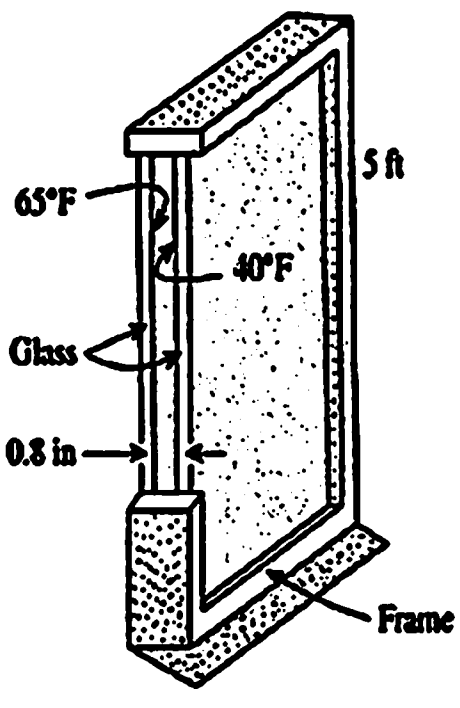


FIGURE P20–66E

inverse of the *R*-value by the surface area and the temperature difference gives the total rate of heat transfer through the window. The effective emissivity for use in radiation calculations between two large parallel glass plates can be taken to be 0.82.

Step 1 of 9

Given data:

Height of window, $H = 5\text{ft}$

Width of window, $W = 6\text{ ft}$

Glass separation gap $L = 0.8\text{in} = 0.0667\text{ft}$

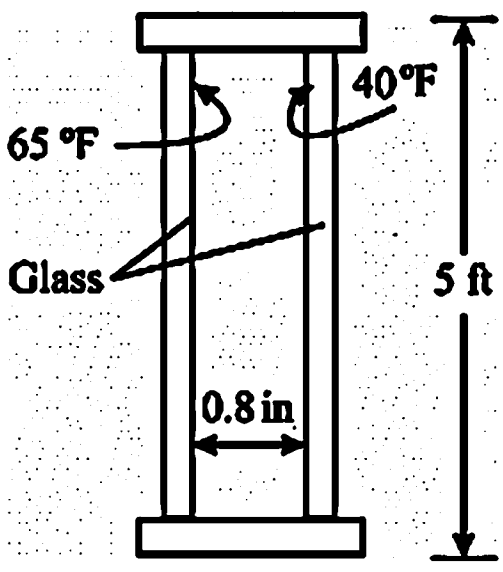
Temperature of inner glass surface, $T_i = 65°\text{F}$

Temperature of outer glass surface, $T_o = 40°\text{F}$

Effective emissivity, $\varepsilon = 0.82$

Step 2 of 9

Figure showing the double pane window:



Step 3 of 9

Average temperature is given by,

$$T_{avg} = \frac{T_i + T_o}{2}$$
$$= \frac{65+40}{2}$$
$$= 52.5°\text{F}$$

Volume expansion co-efficient is given by,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{(52.5+460)}$$
$$= 1.9512\times10^{-3}\ \text{R}^{-1}$$

Characteristic length is given by,

$$L_c = L$$
$$= 0.8\text{in}$$
$$= 0.0667\text{ft}$$

Step 4 of 9

From properties table of air, at 1 atm and average temperature by interpolation,

Thermal conductivity,

$$k = (0.01409) + \left(\frac{0.01433-0.01409}{60-50}\right)(52.5-50)$$
$$= 0.01415\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$$

Kinematic viscosity,

$$\nu = \left(1.535\times10^{-4}\right) + \left(\frac{1.588\times10^{-4}-1.535\times10^{-4}}{60-50}\right)(52.5-50)$$
$$= 0.1548\times10^{-3}\ \text{ft}^2/\text{s}$$

Prandtl number,

$$\text{Pr} = (0.7336) + \left(\frac{0.7321-0.7336}{60-50}\right)(52.5-50)$$
$$= 0.7332$$

Step 5 of 9

Rayleigh number is given by,

$$Ra_L = \frac{g\beta(T_i-T_o)L_c^3}{\nu^2}\times\text{Pr}$$
$$= \frac{32.2\left(1.9512\times10^{-3}\right)(65-40)(0.0667)^3}{\left(0.1548\times10^{-3}\right)^2}(0.7332)$$
$$= 14261.188$$

The aspect ratio is given by,

$$(H/L) = \frac{5}{}$$

$$= 14261.188$$

The aspect ratio is given by,

$$(H/L) = \frac{5}{0.0667}$$
$$= 75$$

Step 6 of 9

From the Rayleigh number and aspect ratio, the appropriate relation for Nusselt number is given by,

$$\mathrm{Nu} = 0.42 Ra_L^{0.25} \mathrm{Pr}^{0.012} \left(\frac{H}{L}\right)^{-0.3}$$

$$= 0.42(14261.188)^{0.25} (0.7332)^{0.012} \left(\frac{5}{0.0667}\right)^{-0.3}$$

$$= 1.2523$$

The surface area is given by,

$$A_s = H \times W$$
$$= 5 \times 6$$
$$= 30 \text{ ft}^2$$

Step 7 of 9

(a)

Rate of heat transfer by convection is given by,

$$\dot{Q}_{conv} = k \,\mathrm{Nu}\, A_s \frac{(T_i - T_o)}{L_c}$$

$$= 0.01415 \times 1.2523 \times 30 \times \left(\frac{65-40}{0.0667}\right)$$

$$= 199.25 \text{ Btu/h}$$

Therefore, the rate of heat transfer by convection is $\boxed{\dot{Q}_{conv} = 199.25 \text{ Btu/h}}$

Step 8 of 9

(b)

Rate of heat transfer by radiation is given by,

$$\dot{Q}_{rad} = \varepsilon A_s \sigma \left(T_i^4 - T_o^4\right)$$

$$= 0.82 \times 30 \times \left(0.1714 \times 10^{-8}\right)\left[(65+460)^4 - (40+460)^4\right]$$

$$= 4.21644 \times 10^{-8}\left(525^4 - 500^4\right)$$

$$= 567.92 \text{ Btu/h}$$

Therefore, the rate of heat transfer by radiation is $\boxed{\dot{Q}_{rad} = 567.92 \text{ Btu/h}}$

Step 9 of 9

Total heat transfer rate,

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= 199.25 + 567.92$$
$$= 767.17 \text{ Btu/h}$$

As per given condition,

$$\dot{Q}_{total} = R_{value}^{-1} \times A_s \times (T_i - T_o)$$

$$\dot{Q}_{total} = \frac{A_s \times (T_i - T_o)}{R_{value}}$$

$$R_{value} = \frac{A_s (T_i - T_o)}{\dot{Q}_{total}}$$

$$R_{value} = \frac{30 \times (65-40)}{767.17}$$

$$R_{value} = 0.978 \text{ ft}^2 \cdot \text{hr} \cdot \text{R/Btu}$$

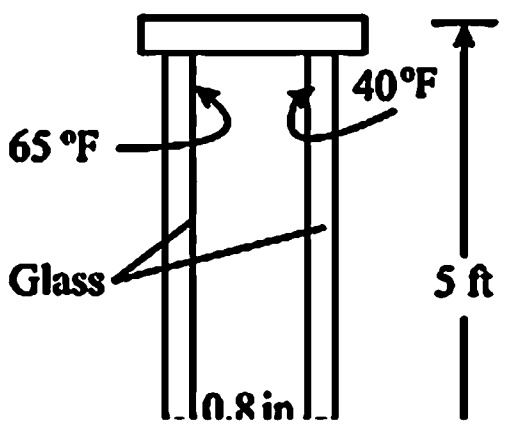
Therefore, the R-value of insulation of the window is $\boxed{R_{value} = 0.978 \text{ ft}^2 \cdot \text{hr} \cdot \text{R/Btu}}$

Question: 20.067P pg 1

Reconsider Prob. 20–66E. Using EES (or other) software, investigate the effect of the air gap thickness on the rates of heat transfer by natural convection and radiation, and the *R*-value of insulation. Let the air gap thickness vary from 0.2 in to 2.0 in. Plot the rates of I heat transfer by natural convection and radiation, and the value of insulation as a function of the air gap thickness, and discuss the results.

Step 1 of 8

Draw the figure of double pane window:

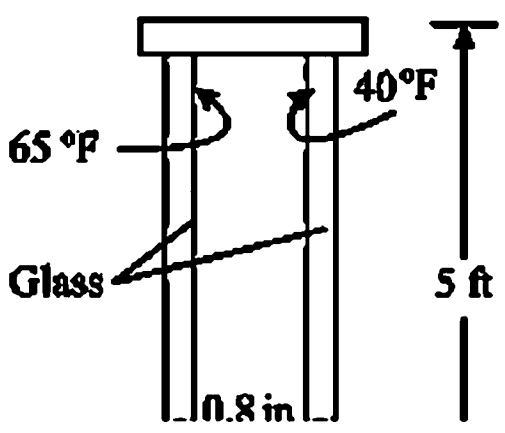


Question:

Reconsider Prob. 20–66E. Using EES (or other) software, investigate the effect of the air gap thickness on the rates of heat transfer by natural convection and radiation, and the *R*-value of insulation. Let the air gap thickness vary from 0.2 in to 2.0 in. Plot the rates of I heat transfer by natural convection and radiation, and the value of insulation as a function of the air gap thickness, and discuss the results.

Step 1 of 8

Draw the figure of double pane window:

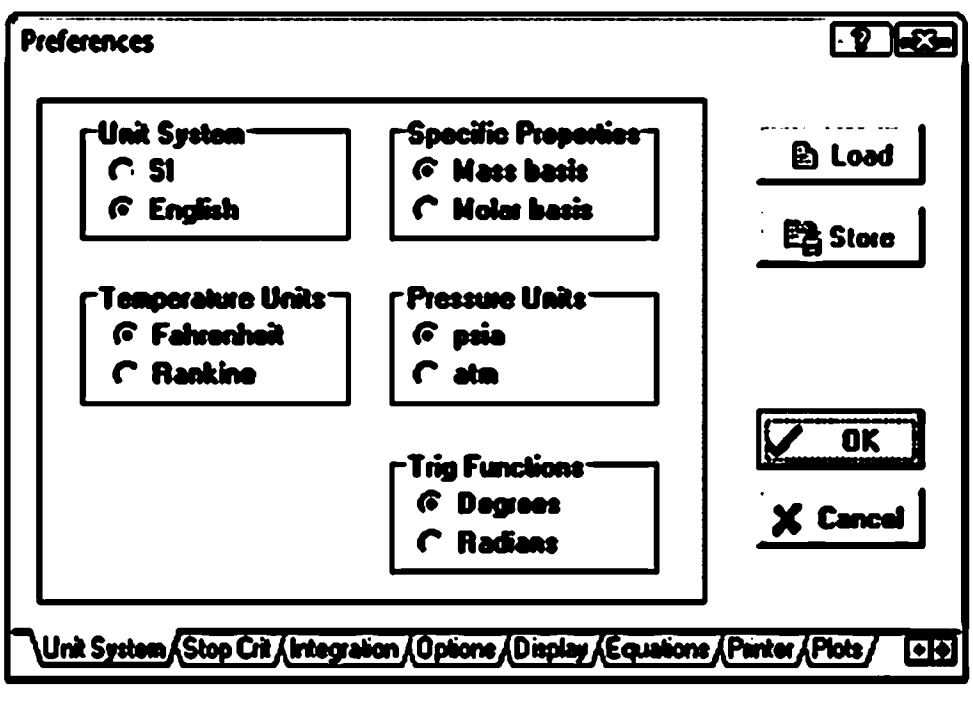



```
Ra_L=g#*beta*(T_i-T_o)*L_c^3*Pr/nu^2"Rayleigh number"
Nus=0.42*Ra_L^0.25*Pr^0.012*(H/L)^(-0.3)
                                        "Nusselt Number"
A_s=H*W                                 "Surface area"

Q_dot_conv=k*Nus*A_s*(T_i-T_o)/L_c"Rate of heat transfer due to convection"
Q_dot_rad=epsilon*A_s*sigma#*(T_i^4-T_o^4)
                                        "Rate of heat transfer due to radiation"
Q_dot_total=Q_dot_conv+Q_dot_rad
Q_dot_total=R_value^(-1)*A_s*(T_i-T_o)
```

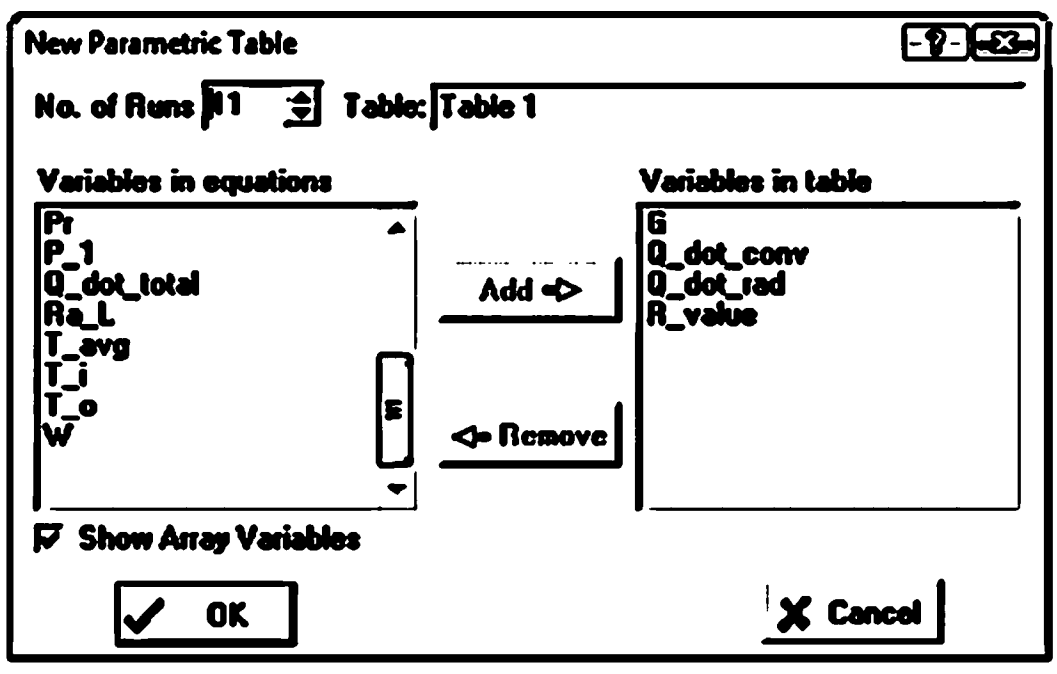
Step 3 of 8

Since, the units are in english system. Therefore, the unit system must be changed to english system by clicking on the 'Unit system' of 'Options' menu and change the radio button to English. The screenshot of the 'Preferences' window is shown below:



Step 4 of 8

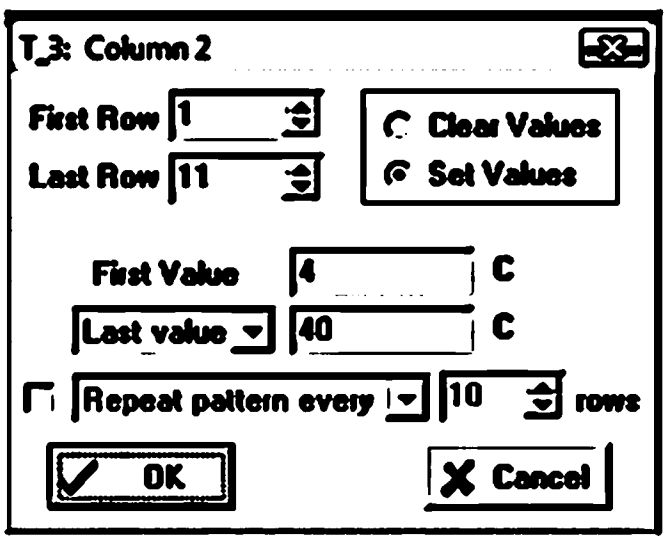
Now to solve the equation for a range of air gap values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "G", "Q_dot_conv", "Q_dot_rad" and "R_value" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 5 of 8

To enter the values in the "G" column, click on the drop tab in "G" column. A new windows pops up, fill in the details as shown below and then press 'OK'

To enter the values in the "G" column, click on the drop tab in "G" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 6 of 8

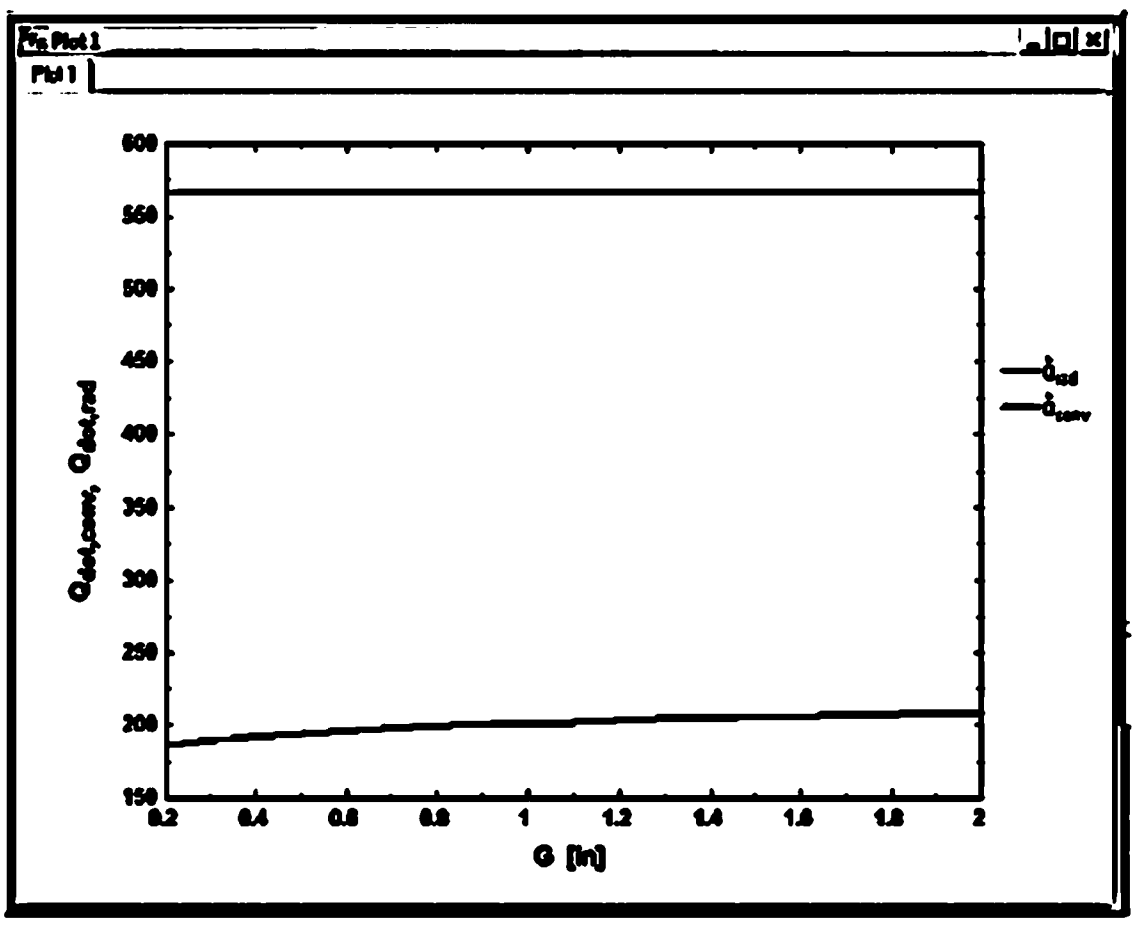
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	G [in]	$\dot{Q}_{conv}$	$\dot{Q}_{rad}$	R_{value}
Run 1	0.2	185.8	566.2	0.9974
Run 2	0.38	191.9	566.2	0.9894
Run 3	0.56	195.6	566.2	0.9845
Run 4	0.74	198.4	566.2	0.981
Run 5	0.92	200.5	566.2	0.9782
Run 6	1.1	202.3	566.2	0.9759
Run 7	1.28	203.9	566.2	0.9739
Run 8	1.46	205.2	566.2	0.9722
Run 9	1.64	206.4	566.2	0.9707
Run 10	1.82	207.5	566.2	0.9694
Run 11	2	208.5	566.2	0.9682

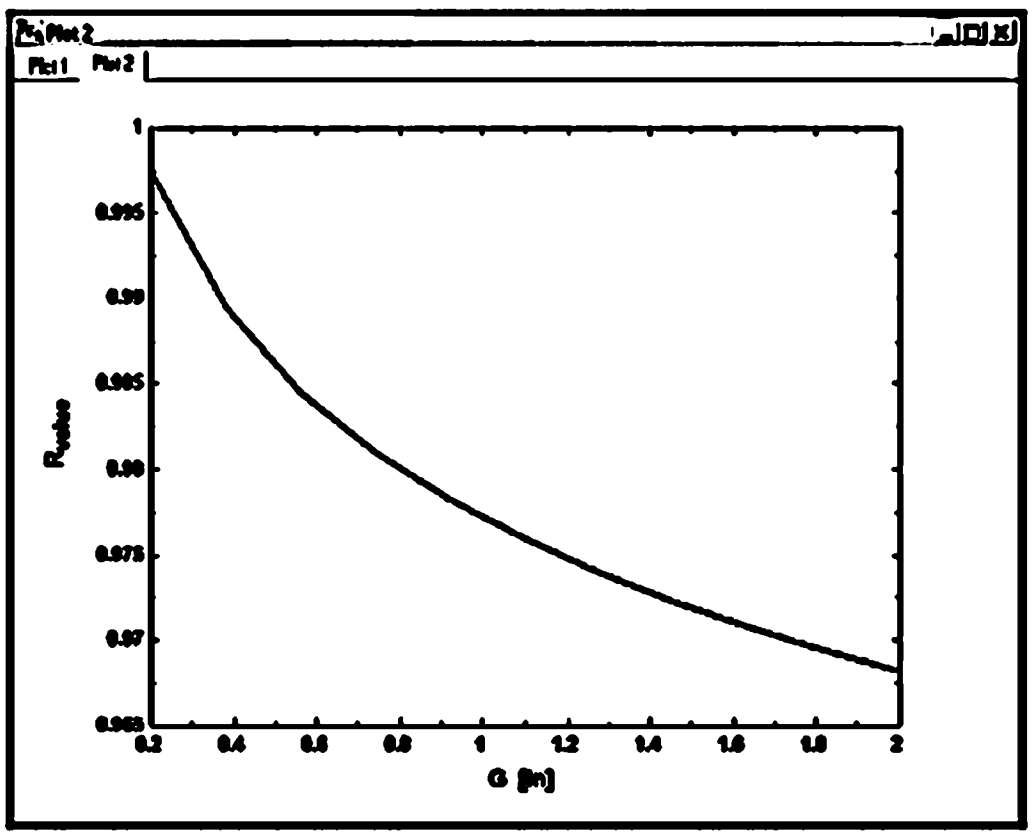
Step 7 of 8

Inorder to plot the rate of heat loss from water by natural convection as a function of ambient air temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Select X-axis as "G" and Y-axis as "Q_dot_conv","Q_dot_rad" and press 'OK'
The Graph will be plotted as shown below:



Step 8 of 8

Now to plot a graph for "G" and "R_value", select X-axis as "G" and Y-axis as "R_value" and press 'OK'
The Graph will be plotted as shown below:



From the above graph we can interpret that as the air gap thickness increases, the rate of heat transfer by convection (Q_{conv}) increases and the R_{value} decreases.

Question:

Consider two concentric horizontal cylinders of diameters 55 cm and 65 cm, and length 125 cm. The surfaces of the inner and outer cylinders are maintained at 54°C and 106°C, respectively. Determine the rate of heat transfer between the cylinders by natural convection if the annular space is filled with (*a*) water and (*b*) air.

Step 1 of 9

The diameter of inner cylinder, $D_i = 55 \text{ cm} = 0.55 \text{ m}$

The diameter of outer cylinder, $D_o = 65 \text{ cm} = 0.65 \text{ m}$

Length of cylinders, $L = 125 \text{ cm} = 1.25 \text{ m}$

Step 2 of 9

(*a*)

The average temperature is given by,

$$T_{avg} = \frac{T_i + T_o}{2}$$
$$= \frac{54 + 106}{2}$$
$$= 80°\text{C}$$

Characteristic length is given by,

$$L_c = \frac{D_o - D_i}{2}$$
$$= \frac{65 - 55}{2}$$
$$= 0.05 \text{ m}$$

Step 3 of 9

The annular space is filled with water:

From properties of saturated water tables, considering the properties of water at average temperature, $T_{avg} = 80°\text{C}$

Thermal conductivity, $k = 0.670 \text{ W/m-K}$

Density, $\rho = 971.8 \text{ kg/m}^3$

Dynamic viscosity, $\mu = 0.355 \times 10^{-3} \text{ kg/m-s}$

Prandtl number, $\text{Pr} = 2.22$

Volume expansion co-efficient, $\beta = 0.653 \times 10^{-3} \text{ K}^{-1}$

Kinematic viscosity is given by,

$$\nu = \frac{\mu}{\rho}$$
$$= \frac{0.355 \times 10^{-3}}{971.8}$$
$$= 3.653 \times 10^{-7} \text{ m}^2/\text{s}$$

Step 4 of 9

Rayleigh numbers is given by,

$$\text{Ra}_L = \frac{g\beta(T_o - T_i)L_c^3}{\nu^2} \cdot \text{Pr}$$
$$= \frac{9.81 \times 0.653 \times 10^{-3} \times (106 - 54) \times (0.05)^3}{\left(3.653 \times 10^{-7}\right)^2}(2.22)$$
$$= 6.927 \times 10^8$$

Geometric factor is given by,

$$F_{cyl} = \frac{\left[\ln(D_o / D_i)\right]^4}{L_c^3\left(D_i^{-3/5} + D_o^{-3/5}\right)^5}$$
$$= \frac{\left[\ln(0.65/0.55)\right]^4}{(0.05)^3\left((0.55)^{-3/5} + (0.65)^{-3/5}\right)^5}$$
$$= 0.041358$$

Step 5 of 9

Effective thermal conductivity is given by,

$$k_{eff} = 0.386k\left(\frac{\text{Pr}}{0.861 + \text{Pr}}\right)^{1/4}\left(F_{cyl} \times \text{Ra}_L\right)^{1/4}$$
$$= 0.386 \times 0.670 \times \left(\frac{2.22}{0.861 + 2.22}\right)^{1/4}\left(0.041357 \times 6.927 \times 10^8\right)^{1/4}$$
$$= 0.386 \times 0.670 \times \left(\frac{2.22}{3.081}\right)^{1/4}(28{,}647{,}994)^{1/4}$$
$$= 17.43 \text{ W/m} \cdot \text{K}$$

Step 6 of 9

Rate of heat transfer when fluid is water,

$$\dot{Q}_{water} = \frac{2\pi \times k_{eff}}{\ln(D_o / D_i)}(T_o - T_i)$$
$$= \frac{2\pi \times 17.43}{\ln(0.65/0.55)}(106 - 54)$$
$$= 34.09 \times 10^3 \text{ W}$$
$$= 34.09 \text{ kW}$$

Therefore, rate of heat transfer when fluid is water is $\boxed{\dot{Q}_{water} = 34.09 \text{ kW}}$

Step 7 of 9

(*b*)

From properties of air tables, considering the properties of air at pressure of 1 atm and average temperature, $T_{avg} = 80°\text{C}$

Thermal conductivity, $k = 0.02953 \text{ W/m} \cdot \text{K}$

Kinematic viscosity, $\nu = 2.097 \times 10^{-5} \text{ m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7154$

From properties of air tables, considering the properties of air at pressure of 1 atm and average temperature, $T_{avg} = 80°C$

Thermal conductivity, $k = 0.02953\ \text{W/m}\cdot\text{K}$

Kinematic viscosity, $\nu = 2.097\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7154$

Volume expansion co-efficient is given by,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{353}$$
$$= 2.833\times10^{-3}\ \text{K}^{-1}$$

Step 8 of 9

Rayleigh number is given by,

$$\text{Ra}_L = \frac{g\beta(T_o - T_i)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times2.833\times10^{-3}\times(106-54)\times(0.05)^3}{\left(2.097\times10^{-5}\right)^2}\times(0.7154)$$
$$= 2.94\times10^5$$

Geometric factor is given by,

$$F_{cyl} = \frac{\left[\ln(D_o/D_i)\right]^4}{L_c^3\left(D_i^{-3/5} + D_o^{-3/5}\right)^5}$$
$$= \frac{\left[\ln(0.65/0.55)\right]^4}{(0.05)^3\left((0.55)^{-3/5} + (0.65)^{-3/5}\right)^5}$$
$$= 0.041358$$

Step 9 of 9

Effective thermal conductivity is given by,

$$k_{eff} = 0.386k\left(\frac{\text{Pr}}{0.861+\text{Pr}}\right)^{0.25}\left(F_{cyl}\cdot\text{Ra}_L\right)^{1/4}$$
$$= 0.386\times0.02953\left(\frac{0.7154}{0.861+0.7154}\right)^{0.25}\left(0.041358\times2.94\times10^5\right)^{1/4}$$
$$= 0.09824\ \text{W/m}\,°\text{C}$$

Rate of heat transfer when the fluid is air is given by,

$$\dot{Q}_{Air} = \frac{2\pi k_{eff}}{\ln(D_o/D_i)}(T_o - T_i)$$
$$= \frac{2\pi\times0.09824}{\ln(0.65/0.55)}(106-54)$$
$$= 192.14\ \text{W}$$

Therefore, rate of heat transfer when fluid is air, $\boxed{\dot{Q}_{\text{Air}} = 192.14\,\text{W}}$

Question: 20.069P

Consider an $L\times L$ horizontal plate that is placed in quiescent air with the hot surface facing up. If the film temperature is 20°C and the average Nusselt number in natural convection is of the form $\mathrm{Nu} = C\mathrm{Ra}_L^n$, show that the average heat transfer coefficient can be expressed as

$h = 1.95(\Delta T/L)^{1/4} \quad 10^4 < \mathrm{Ra}_L < 10^7$

$h = 1.79\Delta T^{1/3} \quad 10^7 < \mathrm{Ra}_L < 10^{11}$

Step 1 of 4

Given data:

Length of horizontal plate is L

Width of horizontal plate is L

Film temperature, $T_f = 20°\mathrm{C}$

Step 2 of 4

The properties of air at the given film temperature,

Thermal conductivity, $k = 0.02514\ \mathrm{W/m\text{-}°C}$

Prandtl number, $\mathrm{Pr} = 0.7309$

Kinematic viscosity, $\nu = 1.516\times10^{-5}\ \mathrm{m^2/s}$

Inverse of film temperature,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{298}$$
$$= 3.413\times10^{-3}\ \mathrm{K^{-1}}$$

Step 3 of 4

Characteristic length,

$$(L_c) = \frac{A_s}{P}$$
$$= \frac{L\times L}{2(L+L)}$$
$$= \frac{L}{4}$$

The expression for Rayleigh number,

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot \mathrm{Pr}$$
$$= \frac{9.81\times3.413\times10^{-3}(\Delta T)L^3}{64\nu^2}\times0.7309$$
$$= 1.664\times10^6(\Delta T)L^3$$

Step 4 of 4

From empirical correlations table, Nusselt number for range $\left(10^4 - 10^7\right)$

$$Nu = 0.54\mathrm{Ra}_L^{0.25}$$
$$= 0.54\left(1.664\times10^6(\Delta T)L^3\right)^{0.25}$$

Now heat transfer co-efficient,

$$h = \frac{k}{L_c}Nu$$
$$= \frac{0.02514\times4}{L}\left[0.54\left(1.664\times10^6(\Delta T)L^3\right)^{0.25}\right]$$
$$= \frac{1.95025}{L}\left(\Delta T L^3\right)^{0.25}$$
$$= 1.95\left(\frac{\Delta T}{L}\right)^{1/4}$$

Hence heat transfer co-efficient for range $\left(10^4 - 10^7\right)$ is $\boxed{h = 1.95\left(\frac{\Delta T}{L}\right)^{1/4}}$

From empirical correlations table, Nusselt number for range $\left(10^7 - 10^{11}\right)$

$$Nu = 0.15\mathrm{Ra}_L^{1/3}$$
$$= 0.15\left(1.664\times10^6(\Delta T)L^3\right)^{1/3}$$
$$= 17.775\left((\Delta T)L^3\right)^{1/3}$$

Now heat transfer co-efficient,

$$h = \frac{k}{L_c}\cdot Nu$$
$$= \frac{0.02514\times4}{L}\times17.775\left(\Delta T L^3\right)^{1/3}$$
$$= 1.787(\Delta T)^{1/3}$$

Hence heat transfer co-efficient for range $\left(10^7 - 10^{11}\right)$ is $\boxed{h = 1.79(\Delta T)^{1/3}}$

Question: 20.070P

A plate (0.5 m × 0.5 m) is inclined at an angle of 30°. The top surface of the plate is well insulated. Estimate the rate of heat loss from the plate when the bottom surface is maintained at 60°C and the surrounding atmospheric quiescent air is at 0°C.

Step 1 of 6

Given data:

Length of Plate, $L = 0.5\text{m}$

Width of Plate, $W = 0.5\text{m}$

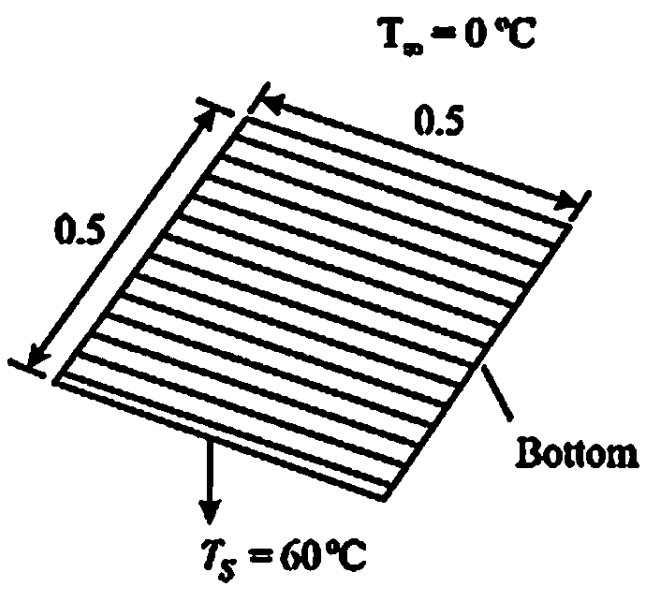
Therefore outside, $T_\infty = 0°\text{C}$

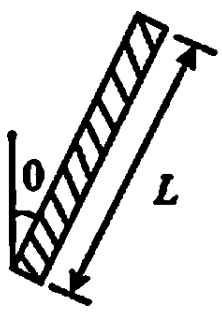
Surface temperature of plate, $T_S = 60°\text{C}$

Inclination angle, $\theta = 30°\text{C}$

Step 2 of 6

Figure showing the inclined plate:





Step 3 of 6

Film temperature,

$$T_f = (T_S + T_\infty)/2$$
$$= (60+0)/2$$
$$= 30°\text{C}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{303}$$
$$= 3.3\times10^{-3}\ \text{K}^{-1}$$

Characteristic length,

$$L_C = L$$
$$= 0.5\text{m}$$

Step 4 of 6

From the properties of air at 1atm pressure and film temperature

Thermal conductivity, $k = 0.02588\ \text{W/m-°C}$

Kinematics viscosity, $\nu = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7282$

Now Rayleigh Number,

$$Ra_L = \frac{g\cos\theta\beta(T_S - T_\infty)L_C^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times\cos30°\times3.3\times10^{-3}(60-0)(0.5)^3}{\left(1.608\times10^{-5}\right)^2}\times(0.7282)$$
$$= 5.92\times10^8$$

Step 5 of 6

Therefore using vertical plate & Ra_L entre range, Nusselt number,

$$Nu = \left[0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.492/\text{Pr})^{9/16}\right]^{8/27}}\right]$$

$$Nu = \left[0.825 + \frac{0.387\left(5.92\times10^8\right)^{1/6}}{\left[1+(0.492/0.728)^{9/16}\right]^{8/27}}\right]^2$$

$$Nu = 104.94$$

Heat transfer coefficient,

$$h = \frac{k}{L_C}\cdot Nu$$
$$= \frac{0.02588}{0.5}\times104.94$$
$$= 5.432\ \text{W/m}^2\text{-°C}$$

Step 6 of 6

Area of surface,

$$A_S = L\times W$$
$$= 0.5\times0.5$$
$$= 0.25\ \text{m}^2$$

The rate of Heat loss,

$$\dot{Q} = hA_S(T_S - T_\infty)$$
$$= 5.432\times0.25\times(60-0)$$
$$= 5.432\times0.25\times60$$
$$= 81.47\ \text{W}$$

Hence the rate of Heat loss is $\boxed{\dot{Q} = 81.47\ \text{W}}$

Question: 20.071P

A 12-cm-diameter and 15-m-long cylinder with a surface temperature of 10°C is placed horizontally in air at 40°C. Calculate the steady rate of heat transfer for the cases of (*a*) free-stream air velocity of 10 m/s due to normal winds and (*b*) no winds and thus a free stream velocity of zero.

Step 1 of 8

Given data:

Diameter of cylinder, $D = 0.12\,\text{m}$

Length of cylinder, $L = 15\,\text{m}$

Outside air temperature, $T_\infty = 40°\text{C}$

Surface temperature of cylinder, $T_S = 10°\text{C}$

Step 2 of 8

Figure showing the Cylinder:



Step 3 of 8

(*a*)

The free stream air velocity, $V = 10\,\text{m/s}$

Area of cylinder surface,

$$A = \pi DL$$
$$= \pi \times 0.12 \times 15$$
$$= 5.65\,\text{m}^2$$

Film temperature,

$$T_f = \frac{T_S + T_\infty}{2}$$
$$= \frac{10+40}{2}$$
$$= 25°\text{C}$$

Step 4 of 8

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{298}$$
$$= 3.356 \times 10^{-3}\,\text{K}^{-1}$$

Characteristic length

$$L_C = D$$
$$= 0.12\,\text{m}$$

Properties of air at 1atm and film temperature,

Thermal conductivity, $k = 0.02551\,\text{W/m}\,°\text{C}$

Prandtl number, $\text{Pr} = 0.7296$

Kinematic viscosity, $\nu = 1.562 \times 10^{-5}\,\text{m}^2/\text{s}$

Step 5 of 8

Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$
$$= \frac{10 \times 0.12}{1.562 \times 10^{-5}}$$
$$= 76824.58$$

Therefore Nusselt number,

$$Nu = 0.027.\text{Re}^{0.805}.\text{Pr}^{1/3}$$
$$= 0.027 \times (76824.58)^{0.805} \times (0.7296)^{1/3}$$
$$= 208.23$$

Step 6 of 8

Heat transfer coefficient,

$$h = \frac{k \times Nu}{L_C}$$
$$= \frac{0.02551 \times 208.23}{0.12}$$
$$= 44.27\,\text{W/m}^2\,°\text{C}$$

The heat transfer rate,

$$\dot{Q}_{FC} = hA(T_\infty - T_S)$$
$$= 44.27 \times 5.65(40-10)$$
$$= 7503.29\,\text{W}$$

Hence the heat transfer rate by forced convection is $\boxed{\dot{Q}_{FC} = 7503.29\,\text{W}}$

Step 7 of 8

(*b*)

Rayleigh number,

$$Ra_D = \frac{g\beta(T_\infty - T_S)}{\nu^2} \cdot \text{Pr}$$
$$= \frac{9.81 \times 3.356 \times 10^{-3} \times (40-10) \times (0.12)^3}{(1.562 \times 10^{-5})^2} \times (0.7296)$$
$$= 5.104 \times 10^6$$

Therefore Nusselt number for horizontal cylinder for obtained Nusselt number range,

$$Nu = \left[0.6 + \frac{0.387\,Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{0.387 \times (5.104 \times 10^6)^{1/6}}{\left[1+(0.559/0.7296)^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{5.078}{1.2}\right]^2$$
$$= 23.345$$

Step 8 of 8

Heat transfer coefficient,

$$h = \frac{k}{L_C} \cdot Nu$$
$$= \frac{0.02551}{0.12} \times (23.345)$$
$$= 4.963\,\text{W/m}^2\,{}^0\text{C}$$

Heat transfer rate by natural convection,

$$\dot{Q}_{NC} = hA(T_\infty - T_S)$$
$$= 4.948 \times 5.65 \times (40-10)$$
$$= 838.71\,\text{W}$$

Hence the heat transfer rate by natural convection is $\boxed{\dot{Q}_{NC} = 838.71\,\text{W}}$

Question: 20.072P pg 1

A spherical vessel, with 30.0-cm outside diameter, is used as a reactor for a slow endothermic reaction. The vessel is completely submerged in a large water-filled tank, held at a constant temperature of 30°C. The outside surface temperature of the vessel is 20°C. Calculate the rate of heat transfer in steady operation for the following cases: (*a*) the water in the tank is still, (*b*) the water in the tank is still (as in a part a), however, the buoyancy force caused by the difference in water density is assumed to be negligible, and (*c*) the water in the lank is circulated at an average velocity of 20 cm/s.

Step 1 of 9

Determine the Film temperature,

$$T_f = \frac{T_S + T_\infty}{2}$$
$$= \frac{30+20}{2}$$
$$= 25°C$$

The properties of water at film temperature,

Thermal conductivity, $k = 0.607\,W/m°C$

Dynamic viscosity, $\mu = 0.891\times10^{-3}\,kg/m-s$

Prandtl number, $Pr = 6.14$

Volume expansion coefficient, $\beta = 0.247\times10^{-3}\,K^{-1}$

Density, $\rho = 997.0 kg/m^3$

Step 2 of 9

Therefore kinematic viscosity,

$$v = \mu/\rho$$
$$= \frac{0.891\times10^{-3}}{997}$$
$$= 8.94\times10^{-7}\,m^2/s$$

Characteristic length,

$$L_C = D$$
$$= 0.3\,m$$

Step 3 of 9

(*a*)

When water in the tank is still, only Natural convection prevails.

Rayleigh number,

$$Ra_D = \frac{g\beta(T_\infty - T_S)L_C^3}{v^2}.Pr$$
$$= \frac{9.81\times0.247\times10^{-3}\times(30-20)\times(0.3)^3}{\left(8.94\times10^{-7}\right)^2}\times(6.14)$$
$$= 5.026\times10^9$$

Nusselt number for the obtained range of Raleigh's number is,

$$Nu = 2 + \frac{0.589Ra_D^{1/4}}{\left[1+\left(\frac{0.469}{Pr}\right)^{9/16}\right]^{4/9}}$$
$$= 2 + \frac{0.589\times\left(5.026\times10^9\right)^{0.25}}{\left[1+\left(\frac{0.469}{6.14}\right)^{9/16}\right]^{4/9}}$$
$$= 144.77$$

Step 4 of 9

Heat transfer coefficient,

$$h = \frac{k\cdot Nu}{L_C}$$
$$= \frac{0.607\times144.77}{0.3}$$
$$= 293\ W/m^2°C$$

Now heat transfer rate by natural convection,

$$Q_{NC} = hA_S(T_\infty - T_S)$$
$$= h\pi D^2(T_\infty - T_S)$$
$$= 293\times\pi\times0.3^2\times(30-20)$$
$$= 828.44\,W$$

Hence heat transfer rate by natural convection is $\boxed{Q_{NC} = 828.44\,W}$

Step 5 of 9

(b)

If the Buoyancy force due to difference in water density is negligible, then the Rayleigh number can be neglected $\left(Ra_D = 0\right)$.

Thus Nusselt number is

$$Nu = 2 + \frac{0.589\,Ra_D^{1/4}}{\left[1+(0.469/Pr)^{9/16}\right]^{4/9}}$$
$$= 2 + \frac{0.589\times(0)^{1/4}}{\left[1+(0.469/Pr)^{9/16}\right]^{4/9}}$$
$$= 2$$

Step 6 of 9

Heat transfer coefficient,

$$= h\pi D^2(T_\infty - T_S)$$
$$= 293\times\pi\times0.3^2\times(30-20)$$
$$= 828.44\,W$$

Hence heat transfer rate by natural convection is $\boxed{Q_{NC} = 828.44\,W}$

20.072P pg 2

$= 828.44\,\text{W}$

Hence heat transfer rate by natural convection is $\boxed{Q_{NC} = 828.44\,\text{W}}$

Step 5 of 9

(b)

If the Buoyancy force due to difference in water density is negligible, then the Rayleigh number can be neglected $\left(Ra_D = 0\right)$.

Thus Nusselt number is

$$Nu = 2 + \frac{0.589\,Ra_D^{1/4}}{\left[1 + \left(0.469/\text{Pr}\right)^{9/16}\right]^{4/9}}$$

$$= 2 + \frac{0.589 \times (0)^{1/4}}{\left[1 + \left(0.469/\text{Pr}\right)^{9/16}\right]^{4/9}}$$

$$= 2$$

Step 6 of 9

Heat transfer coefficient,

$$\mu_\infty = 0.798 \times 10^{-3}\,\text{kg/m-s}$$

Step 8 of 9

Now Reynolds number,

$$\text{Re} = \frac{VD}{\nu}$$

$$= \frac{VD\rho}{\mu}$$

$$= \frac{0.2 \times 0.3 \times 998}{\left(1.002 \times 10^{-3}\right)}$$

$$= 59760.96$$

Nusselt number,

$$Nu = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_s}{\mu_\infty}\right)^{0.25}$$

$$= 2 + \left[0.4(59760.96)^{0.5} + 0.06(59760.96)^{2/3}\right](7.01)^{0.4}\left(\frac{1.002 \times 10^{-3}}{0.798 \times 10^{-3}}\right)^{0.25}$$

$$= 2 + [189.49 \times 2.179 \times 1.059]$$

$$= 439.27$$

Step 9 of 9

Heat transfer coefficient,

$$h = \frac{k \cdot Nu}{D}$$

$$= \frac{0.598 \times 439.27}{0.3}$$

$$= 875.61\,\text{W/m}^2\,^\circ\text{C}$$

Rate of heat transfer by forced convection,

$$Q = hA_S\left(T_\infty - T_S\right)$$

$$= 875.61 \times 0.283 \times (30 - 20)$$

$$= 2477.98\,\text{W}$$

Therefore rate of heat transfer by forced convection, $\boxed{Q = 2477.98\,\text{W}}$

Question:

Hot water is flowing at an average velocity of 4 ft/s through a cast iron pipe (k= 30 Btu/h·ft·°F) whose inner and outer diameters are 1.0 in and 1.2 in, respectively. The pipe passes through a 50-ft-long section of a basement whose temperature is 60°F. The emissivity of the outer surface of the pipe is 0.5, and the walls of the basement are also at about 60°F. If the inlet temperature of the water is 150°F and the heat transfer coefficient on the inner surface of the pipe is 30 Btu/h ·°F, determine the temperature drop of water as it passes through the basement.

Step 1 of 7

Length of pipe, $L = 50\,\text{ft}$

Emissivity of pipe surface, $\varepsilon = 0.5$

Temperature of basement, $T_\infty = 60°\text{F}$

Inlet water flow temperature, $T_w = 150°\text{F}$

Average velocity of water flow, $V = 4\ \text{ft/s}$

Inside Diameter of pipe, $D_i = 1.0\,\text{in} = 0.083\,\text{ft}$

Outside Diameter of pipe, $D_o = 1.2\,\text{in} = 0.1\,\text{ft}$

Temperature of outside surfaces, $T_{amb} = 60°\text{F}$

Thermal conductivity of pipe, $k = 30\ \text{Btu/h}\cdot\text{ft}°\text{F}$

Heat transfer coefficient for inner pipe surface, $h_i = 30\,\text{Btu/h}\cdot\text{ft}^2°\text{F}$

Step 2 of 7

Calculate the Film temperature,

$$T_f = \frac{T_S + T_\infty}{2}$$
$$= \frac{150+60}{2}$$
$$= 105°\text{F}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{460+105}$$
$$= 1.77\times10^{-3}\ \text{R}^{-1}$$

Step 3 of 7

Take properties of air at 1atm pressure and film temperature,

Thermal conductivity, $k = 0.015405\,\text{Btu/h}-\text{ft}°\text{F}$

Prandtl number, $\text{Pr} = 0.72525$

Kinematic viscosity, $\nu = 1.8375\times10^{-4}\ \text{ft}^2/\text{s}$

Characteristic length, $L_C = 0.1\ \text{ft}$

Inner surface Area,

$$A_i = \pi D_i L$$
$$= \pi\times0.083\times50$$
$$= 13.04\ \text{ft}^2$$

Outer surface Area,

$$A_o = \pi D_o L$$
$$= \pi\times0.1\times50$$
$$= 15.71\ \text{ft}^2$$

Step 4 of 7

Calculate Rayleigh number,

$$Ra = \frac{g\beta(T_\infty - T_S)L_C^3}{\nu^2}\times\text{Pr}$$
$$= \frac{32.2\times\left(1.77\times10^{-3}\right)\times(150-60)\times(0.1)^3}{\left(1.8375\times10^{-4}\right)^2}\times(0.72525)$$
$$= 1.102\times10^5$$

Calculate Nusselt number,

$$Nu = \left[0.6 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{0.387(1.102\times10^5)\text{Ra}^{1/6}}{\left[1+(0.559/0.72525)^{9/16}\right]^{8/27}}\right]^2$$
$$= 7.998$$

Step 5 of 7

Heat transfer coefficient for outside surface,

$$h_o = \frac{k}{L_C}\cdot Nu$$
$$= \frac{0.015405}{0.1}\times7.998$$
$$= 1.232\ \text{Btu/h-ft}^2\text{-°F}$$

Calculating heat transfer coefficient for radiation from outside surface,

$$h_{rad} = \varepsilon\sigma(T_S + T_{amb})\left(T_S^2 + T_{amb}^2\right)$$
$$= 0.5\times\left(0.1714\times10^{-8}\right)\left[(150+460)+(60+460)\right]$$
$$\left[(150+460)^2+(60+460)^2\right]$$
$$= 0.6222\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

Step 6 of 7

Therefore combined heat transfer coefficient,

$$h_{com} = h_0 + h_{rad}$$
$$= 1.232 + 0.6222$$

Step 6 of 7

Therefore combined heat transfer coefficient,

$$h_{com} = h_0 + h_{rad}$$
$$= 1.232 + 0.6222$$
$$= 1.854 \text{ Btu/h-ft}^2\text{-R}$$

Overall resistance to heat transfer,

$$R_{tot} = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{4\pi kL} + \frac{1}{h_o A_o}$$
$$= \frac{1}{(30)(13.04)} + \frac{\ln(0.1/0.0833)}{4\pi(30)(50)} + \frac{1}{(1.854)(15.71)}$$
$$= \left(2.56\times10^{-3}\right) + \left(9.69\times10^{-6}\right) + (0.0343)$$
$$= 0.03687 \ ^0\text{R-h/Btu}$$

Heat transfer due to convection and radiation,

$$\dot{Q} = \frac{T_w - T_\infty}{R_{tot}}$$
$$= \frac{150 - 60}{0.03687}$$
$$= 2441.28 \text{Btu/h}$$

Step 7 of 7

Density of water from property table, at 150°F, is $\rho = 61.2 \text{lbm/ft}^3$

Now calculate mass flow rate,

$$\dot{m} = \rho \times A_i \times V$$
$$= 61.21 \times 13.04 \times 4$$
$$= 1.3245 \text{ lbm/s}$$
$$= 4768.2 \text{ lbm/hour}$$

Calculate heat transfer rate,

$$\dot{Q} = \dot{m} c_p (\Delta T)$$

$$\Delta T = \frac{\dot{Q}}{\dot{m} c_p}$$
$$= \frac{2441.28}{4768.2 \times 1.0}$$
$$= 0.512$$

Hence the temperature Drop across pipe is, $\boxed{\Delta T = 0.512°\text{F}}$

$$= 2441.28 \text{Btu/h}$$

Step 7 of 7

Question: 20.074P

Consider a flat-plate solar collector placed horizontally on the flat roof of a house. The collector is 1.5 m wide and 4.5 m long, and the average temperature of the exposed surface of the collector is 42°C. Determine the rate of heat loss from the collector by natural convection during a calm day when the ambient air temperature is 8°C. Also, determine the heat loss by radiation by taking the emissivity of the collector surface to be 0.85 and the effective sky temperature to be –15°C.

Step 1 of 7

Given data:

Length of collector, $L = 4.5\text{m}$

Width of collector $w = 1.5\text{m}$

Temperature of collector surface $T_s = 42°\text{C}$

Ambient air temperature $T_\infty = 8°\text{C}$

Effective sky temperature $T_{sky} = -15°\text{C}$

Emissivity of the collector surface $\varepsilon = 0.85$

Step 2 of 7

(a)Rate of heat transfer by convection, $\dot{Q}_{conv}$ is expressed as:-

$$Q_{conv} = hA_s\left(T_s - T_\infty\right) \quad \ldots\ldots (1)$$

Let us find the unknown quantities

Heat transfer surface Area,

$$A_s = L \times W$$
$$= 4.5 \times 1.5$$
$$= 6.75\text{m}^2$$

The average temperature of air

$$T_f = \left(T_s + T_\infty\right)/2$$
$$T_f = (42+8)/2$$
$$T_f = 25°\text{C}$$

Step 3 of 7

The properties of air at 1atm and average temperature at $T_f = 25°\text{C}$ are

Thermal conductivity, $k = 0.02551\,\text{W/m}\cdot\text{k}$

Prandtl Number, $\text{Pr} = 0.7296$

Kinematic viscosity, $v = 1.562\times10^{-5}\,\text{m}^2/\text{s}$

Volume expansion coefficient

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{298}$$
$$= 3.36\times10^{-3}\,\text{K}^{-1}$$

Step 4 of 7

The characteristic length for horizontal surface is calculated as,

$$L_c = \frac{A_s}{p}$$

Where A_s – Surface area

p = Perimeter of surface

$$L_c = \frac{A_s}{2(L+w)}$$
$$= \frac{6.75}{2(1.5+4.5)}$$
$$= 0.5625\text{m}$$

Step 5 of 7

The Rayleigh number becomes

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.36\times10^{-3}\times(42-8)\times(0.5625)^3}{\quad}\times(0.7296)$$
$$= \frac{1}{298}$$
$$= 3.36\times10^{-3}\,\text{K}^{-1}$$

Step 4 of 7

The characteristic length for horizontal surface is calculated as,

$$L_c = \frac{A_s}{p}$$

Where A_s – Surface area

p = Perimeter of surface

$$L_c = \frac{A_s}{2(L+w)}$$
$$= \frac{6.75}{2(1.5+4.5)}$$
$$= 0.5625\text{m}$$

Step 5 of 7

The Rayleigh number becomes

$$\text{Ra}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.36\times10^{-3}\times(42-8)\times(0.5625)^3}{\quad}\times(0.7296)$$

(b)Rate of heat transfer by radiation, $\dot{Q}_{rad}$ can be calculated as

$$\dot{Q}_{rad} = \varepsilon A_s\sigma\left(T_s^4 - T_{sky}^4\right)$$

Where

Stefan Boltzmann constant $\sigma = 5.67\times10^{-8}\,\text{W/m}^2\text{K}^4$

Then

$$\dot{Q}_{rad} = 0.85\times6.75\times5.67\times10^{-8}\left((42+273)^4 - (-15+273)^4\right)$$
$$= 1761.53\,\text{W}$$

Question: 20.076P

A 6-m-internal-diameter spherical tank made of 1.5-cm-thick stainless steel (k =15 W/m·K) is used to store iced water at 0°C in a room at 20°C. The walls of the room are also at 20°C. The outer surface of the tank is black (emissivity ε= 1), and heat transfer between the outer surface of the tank and the surroundings is by natural convection and radiation. Assuming the entire steel tank to be at 0°C and thus the thermal resistance of the tank to be negligible, determine (*a*) the rate of heat transfer to the iced water in the tank and (*b*) the amount of ice at 0°C that melts during a 24-h period. The heat of fusion of water is 333.7 kJ/kg.

Step 1 of 6

As given the heat transfer is by convection and radiation both, therefore

$$Q_{total} = Q_{conv} + Q_{rad}$$

$$Q_{total} = hA_s(T_s - T_\infty) + \sigma A_s \varepsilon\left(T_s^4 - T_w^4\right) \quad \text{..... (1)}$$

The average temperature of air is

$$T_f = (T_s + T_\infty)/2$$
$$= (0+20)/2$$
$$= 10°C$$

Taking properties of air at 1 atm and T_f =10°C is

Thermal conductivity, $k = 0.02439\,W/m°C$

Prandtl number, $Pr = 0.7336$

Kinematic viscosity, $\nu = 1.426\times10^{-5}\,m^2/s$

Step 2 of 6

Determine the Volume expansion,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{10+273}$$
$$= 3.534\times10^{-3}K^{-1}$$

Characteristic length of spherical tank is equal to the diameter of the spherical tank

$$L_c = D_o = 6.03\,m$$

Step 3 of 6

Therefore Rayleigh number

$$Ra_L = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2}\cdot Pr$$
$$= \frac{9.81\times3.534\times10^{-3}(20-0)(6.03)^3}{\left(1.426\times10^{-5}\right)^2}\times(0.7336)$$
$$= 5.4845\times10^{11}$$

From table "Empirical correlations for the average Nusselt number for natural convection" in case of spheres Nusselt number is

$$Nu = 2 + \frac{0.589Ra_L^{0.25}}{\left[1+\left(\frac{0.469}{Pr}\right)^{9/16}\right]^{4/9}}$$
$$= 2 + \frac{0.589\times\left(5.4845\times10^{11}\right)^{0.25}}{\left[1+\left(\frac{0.469}{0.7336}\right)^{9/16}\right]^{4/9}}$$
$$= 394.53$$

Step 4 of 6

Heat transfer coefficient,

$$h = \frac{Nu\cdot k}{L_c}$$
$$= \frac{394.53\times0.02439}{6.03}$$
$$= 1.596\,W/m^2°C$$

Area of spherical surface,

$$A_s = \pi D_0^2$$
$$= \pi(6.03)^2$$
$$= 114.23\,m^2$$

Step 5 of 6

(a) Substituting known values in equation (1)

$$Q_{total} = hA_s(T_s - T_\infty) + \sigma A_s\varepsilon\left(T_s^4 - T_w^4\right)$$
$$= 1.596\times114.23(20-0) + 5.67\times10^{-8}\times114.23\times1\left(293^4 - 273^4\right)$$
$$= 3646.2216 + 11758.568$$
$$= 15404.789\,W$$

Therefore heat transfer to the iced water in the tank is $\boxed{15.404\,kW}$

Step 6 of 6

(b) Heat transfer per second Q_{total} =15404.789 J/s

Therefore heat transfer in 24 hours,

$$Q = 15404.789\times3600\times24$$
$$Q = 1330.97\times10^3\,kJ$$

Amount of ice melt,

$$m = \frac{\text{Total heat transfer in 24 hours}}{\text{Heat of fusion of water, } h_f}$$
$$m = \frac{1330.97\times10^3}{333.7}$$

Thus the mass of ice melt is $\boxed{3988.522\,kg}$

Question: 20.077P

Consider a 1.2-m-high and 2-m-wide double-pane window consisting of two 3-mm-thick layers of glass (k =0.78 W/m · K} separated by a 2.5-cm-wide air space. Determine the steady rate of heat transfer through this window and the temperature of its inner surface for a day during which the room is maintained at 20°C while the temperature of the outdoors is 0°C. Take the heat transfer coefficients on the inner and outer surfaces of the window to be $h1$, = 10 W/m K and $h2$ =25 W/m2.K and disregard any heat transfer by radiation.

Step 1 of 7

Loading solution. Please wait ...

Step 2 of 7

Assumptions:

(i) Steady state heat transfer

(ii) Radiation effects are neglected

(iii) Inner glass temperature, $T_1 = 18°C$

(iv) Outer glass surface temperature, $T_2 = 2°C$

(v) Let us take the air properties at 1atm and at average temperature T_{avg}

Average temperature of air is

$$T_{avg} = \frac{T_1 + T_2}{2}$$

$$= \frac{18+2}{2}$$

$$T_{avg} = 10°C$$

Thermal conductivity, $k = 0.02439\,w/m°C$

Prandtl number, $Pr = 0.7336$

Kinematic viscosity, $\nu = 1.426\times10^{-5}\,m^2/s$

Step 3 of 7

Volume expansion coefficient,

$$\beta = \frac{1}{T_{avg}}$$

$$= \frac{1}{(273+10)}$$

$$= 3.534\times10^{-3}\,K^{-1}$$

Heat transfer through window, Q can be expressed as

$$Q = hA_s(T_1 - T_2) \qquad \ldots\ldots (1)$$

Heat transfer rate $\dot{Q}$ is also given by.

$$\dot{Q} = \frac{T_{\infty_i} - T_{\infty_o}}{R_{inside} + R_{Glass} + R_{airspace} + R_{outside}} \qquad \ldots\ldots (2)$$

Step 4 of 7

Characteristic length L_c = Airspace between Glasses,

$$L_c = L = 0.025\,m$$

Calculating Rayleigh and Nusselt number for heat transfer coefficient h,
Rayleigh number,

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}\cdot Pr$$

$$= \frac{9.81\times3.534\times10^{-3}(18-2)(0.025)^3}{(1.426\times10^{-5})^2}\times(0.7336)$$

$$= 31267.7$$

Nusselt number

$$Nu = 0.42Ra^{0.25}Pr^{0.012}\left(\frac{H}{L}\right)^{-0.3}$$

$$= 0.42\times(31267.7)^{0.25}\times(0.7336)^{0.012}\times\left(\frac{1.2}{0.025}\right)^{-0.3}$$

$$= 1.742$$

Step 5 of 7

Heat transfer coefficient,

$$h_{air} = \frac{k}{L_c}\cdot Nu$$

$$= \frac{0.02439}{0.025}\times1.742$$

$$= 1.7\,W/m^2°C$$

Cross sectional area of the Glass,

$$A_s = H\times W$$

$$= 1.2\times2$$

$$= 2.4\,m^2$$

Step 6 of 7

Substituting the known values in equation (2)

$$\dot{Q} = \frac{T_{\infty_i} - T_{\infty_o}}{R_{inside} + R_{Glass} + R_{airspace} + R_{outside}}$$

$$= \frac{T_{\infty_i} - T_{\infty_o}}{\frac{1}{h_1A_s} + \frac{2L_g}{k_gA_s} + \frac{1}{h_{air}A_s} + \frac{1}{h_2\cdot A}}$$

$$= \frac{20-0}{\frac{1}{10\times2.4} + \frac{2\times0.003}{0.78\times2.4} + \frac{1}{1.7\times2.4} + \frac{1}{25\times2.4}}$$

$$= \frac{20}{0.3066}$$

$$= 65.22\,W$$

Step 7 of 7

Now putting the values in equation (1) to check whether the assumed values are correct or not

$$Q = hA_s(T_1 - T_2)$$

$$65.22 = 1.7\times2.4(T_1 - T_2)$$

$$T_1 - T_2 = 15.98°C$$

$T_1 - T_2 = 16°C$ This is exactly same as we assumed.

Question: 20.078P pg 1

An electric resistance space heater is designed such that it resembles a rectangular box 50 cm high, 80 cm long, and 15 cm wide filled with 45 kg of oil. The heater is to be placed against a wall, and thus heat transfer from its back surface is negligible. The surface temperature of the heater is not to exceed 75°C in a room at 25°C for safety considerations. Disregarding heat transfer from the bottom and top surfaces of the heater in anticipation that the top surface will be used as a shelf, determine the power rating of the heater in W. Take the emissivity of the outer surface of the heater to be 0.8 and the average temperature of the ceiling and wall surfaces to be the same as the room air temperature.

Also, determine how long it will take for the heater to reach steady operation when it is first turned on (i.e., for the oil temperature to rise from 25°C to 75°C). State your assumptions in the calculations.

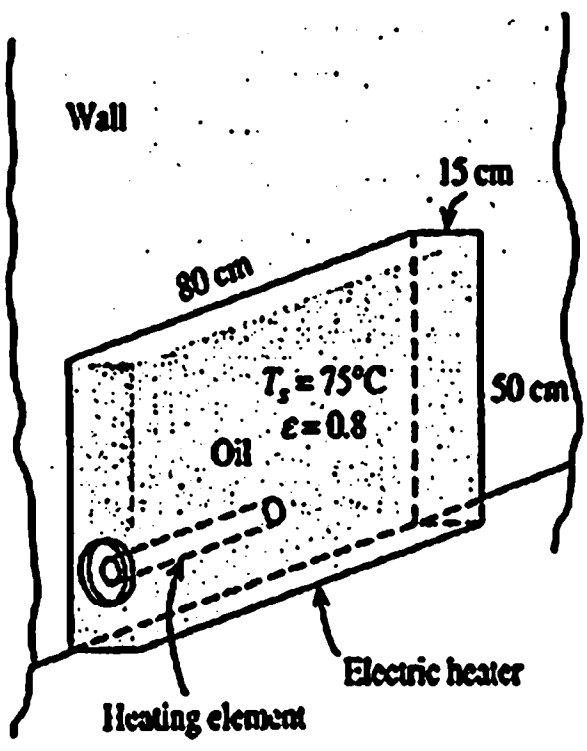


FIGURE P20–78

Step 1 of 7

Height of rectangular box, $H = 50\text{cm} = 0.5\text{m}$

Width of rectangular box, $W = 15\text{cm} = 0.15\text{m}$

Length of rectangular box, $L = 80\text{cm} = 0.8\text{m}$

Quantity of oil, $m = 45\text{kg}$

Surface temperature of heater, $T_s = 75°\text{C}$

Room temperature $T_\infty = 25°\text{C}$

Emissivity of Heater surface, $\varepsilon = 0.8$

Outside wall temperature, $T_w = 25°\text{C}$

Step 2 of 7

The heat loss is due to both convection and radiation. Therefore heat transfer rate will be expressed as

$$\dot{Q} = hA_s\left(T_s - T_\infty\right) + \varepsilon A_s \sigma\left(T_s^4 - T_w^4\right) \qquad \ldots\ldots (1)$$

Let us take average temperature of air for determining the properties of air

$$T_f = \left(T_s + T_\infty\right)/2$$
$$= (75+25)/2$$
$$= 50°\text{C}$$

Taking air properties at 1atm pressure and average temperature $T_f = 50°\text{C}$

Thermal conductivity, $k = 0.02735\,\text{W/m}°\text{C}$

Prandtl number, $\text{Pr} = 0.7228$

Kinematic viscosity, $\nu = 1.798\times10^{-5}\,\text{m}^2/\text{s}$

Step 3 of 7

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{50+273}$$
$$= \frac{1}{323}$$
$$= 3.0960\times10^{-3}\,\text{K}^{-1}$$

Characteristic length is height of vertical side of box

$L_c = 0.5\text{m}$

Step 4 of 7

Calculating the Rayleigh number,

$$\text{Ra} = \frac{g\beta\left(T_s - T_\infty\right)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.0960\times10^{-3}\times(75-25)\times(0.5)^3}{\left(1.798\times10^{-5}\right)^2}\times(0.7228)$$
$$= 4.244\times10^8$$

Nusselt number for vertical plate and complete range of Ra

$$\text{Nu} = \left[0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.492/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387\left(4.244\times10^8\right)^{1/6}}{\left[1+(0.492/0.7228)^{9/16}\right]^{8/27}}\right]^2$$
$$= 94.68$$

Step 5 of 7

Therefore Heat transfer coefficient,

$$h = \frac{k\times\text{Nu}}{L_c}$$
$$= \frac{0.02735\times94.68}{0.5}$$
$$= 5.179\,\text{W/m}^2°\text{C}$$

$$= \frac{0.02735 \times 94.68}{0.5}$$

$$= 5.179\ \text{W/m}^2\,^\circ\text{C}$$

The heat transfer takes place from side surfaces of the rectangular box and the front surface of the rectangular box. Thus the surface Area of heat flow is

$$A_s = (0.5 \times 0.8) + 2(0.15 \times 0.5)$$

$$= 0.55\ \text{m}^2$$

From equation (1), the total heat loss is

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_w^4\right)$$

$$= 5.179(0.55)(75-25) + 0.8(0.55)\left(5.67 \times 10^{-8}\right)\left[(75+273)^4 - (25+273)^4\right]$$

$$= 142.4225 + 169.148$$

Thus the total heat loss is $\boxed{\dot{Q} = 311.57\ \text{W}}$

Step 6 of 7

Time taken to raise oil temperature from 25°C to 75°C:

Let us calculate the heat required to raise the oil temperature. It can be expressed as

$$Q_T = mc_p(T_2 - T_1)$$

Assuming specific heat of oil as $c_p = 2006\ \text{J/kg.°C}$

$T_2 = T_s$ = Surface temperature of heater

$T_1 = T_\infty$ = Ambient room temperature

Therefore

$$Q_T = 45 \times 2006 \times (75-25)$$

$$= 4.5135 \times 10^6\ \text{J}$$

Step 7 of 7

And the heat generated is power rating of the heater

$$\dot{Q} = 311.57\ \text{W}$$

$$= 311.57\ \text{J/s}$$

Therefore time taken to rise the oil temperature from 25°C to 75°C

$$t = \frac{\text{Heat required to rise temperature}}{\text{Heat generated}}$$

$$= \frac{Q_T}{\dot{Q}}$$

$$= \frac{4.5135 \times 10^6}{311.57}$$

$$= 14486.31\ \text{seconds}$$

$$= 4.024\ \text{hours}$$

Therefore time taken to rise the oil temperature is $\boxed{t = 4.024\ \text{hours}}$

Question: 20.079P pg 1

During a plant visit, it was observed that a 2-m-high and 1-m-wide section of the vertical front section of a natural gas furnace wall was too hot to touch. The temperature measurements on the surface revealed that the average temperature of the exposed hot surface was 110°C, while the temperature of the surrounding air was 25°C. The surface appeared to be oxidized, and its emissivity can be taken to be 0.6. Taking the temperature of the surrounding surfaces to be 25°C also, determine the rate of heat loss from, this furnace.

The furnace has an efficiency of 79 percent, and the plant pays $ 1.20 per therm of natural gas. If the plant operates 10 h a day, 310 days a year, and thus 3100 h a year, determine the annual cost of the heat loss from this vertical hot surface on the front section of the furnace wall.

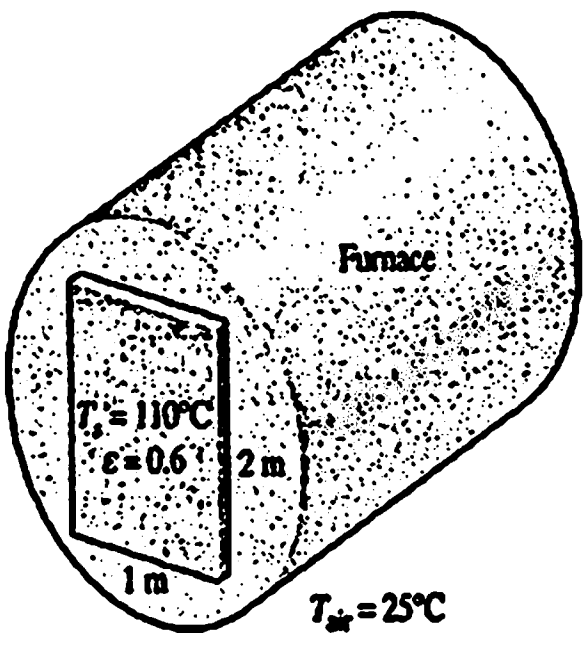


FIGURE P20–79

Step 1 of 8

Given data:

Height of section, $H = 2\text{m}$

Width of section, $W = 1\text{m}$

Temperature of hot surface, $T_s = 110°C$

Temperature of surrounding air, $T_\infty = 25°C$

Emissivity of surface, $\varepsilon = 0.6$

Surrounding surface temperature, $T_w = 25°C$

Furnace efficiency

$\eta_{\text{furnace}} = 79\%$

$= 0.79$

Operating Hours, $N = 3100\text{h}$

Cost of natural gas, $C_N = \$1.20/\text{therm}$

Step 2 of 8

The total rate of heat loss,

$$Q_T = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_w^4\right) \quad \ldots\ldots (1)$$

The average film temperature of air,

$T_f = (T_s + T_\infty)/2$

$= (110 + 25)/2$

$T_f = 67.5°C$

Taking air properties at 1atm pressure and film temperature $T_f = 67.5°C$

Interpolating air properties between $(60 - 70)°C$

Thermal conductivity,

$$k = 0.02808 + \frac{(0.02881 - 0.02808)}{(70 - 60)}(67.5 - 60)$$

$k = 0.02863\text{w/m} \cdot \text{K}$

Prandtl number, $\text{Pr} = 0.71833$

Step 3 of 8

Kinematic viscosity,

$$\nu = 1.896\times10^{-5} + \frac{(1.995 - 1.896)\times10^{-5}}{(70 - 60)} \times (67.5 - 60)$$

$\nu = 1.97025\times10^{-5}\,\text{m}^2/\text{s}$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{273 + 67.5}$$

$\beta = 2.937\times10^{-3}\,\text{K}^{-1}$

Step 4 of 8

Characteristic length is Height of section, $L_C = H = 2\text{m}$

Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \cdot \text{Pr}$$

$$= \frac{9.81\times2.937\times10^{-3}\times(110 - 25)\times(2)^3}{\left(1.97025\times10^{-5}\right)^2} \times (0.71833)$$

$\text{Ra} = 3.626\times10^{10}$

Nusselt number for vertical plate and entire range of Ra,

$$\text{Nu} = \left[0.825 + \frac{0.387\,Ra^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.825 + \frac{0.387\left(3.626\times10^{10}\right)^{1/6}}{\left[1 + \left(\frac{0.492}{0.71833}\right)^{9/16}\right]^{8/27}}\right]^2$$

$\text{Nu} = 380.44$

Step 5 of 8

$$\left[\left(\frac{}{0.71833}\right)^{}\right]^{}$$ 20.079P pg 2

$\mathrm{Nu} = 380.44$

Step 5 of 8

Heat transfer coefficient,

$$h = \frac{k}{L_c} \times \mathrm{Nu}$$

$$= \frac{0.02863}{2} \times 380.44$$

$$h = 5.45\,\mathrm{W/m^2.^oC}$$

Surface Area of the heat transfer

$$A_s = H \times W$$

$$= 2 \times 1$$

$$A_s = 2\,\mathrm{m^2}$$

Step 6 of 8

On substituting the known values in the equation (1), we get

$$Q_T = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s\left(T_s^4 - T_w^4\right)$$

$$= 5.45 \times 2 \times (110-25) + 0.6 \times 2 \times 5.67 \times 10^{-8}\left[(273+110)^4 - (273+25)^2\right]$$

$$Q_T = 926.5 + 927.488$$

Therefore the rate of heat loss from this furnace, $\boxed{Q_T = 1853.988\,\mathrm{W}}$

Step 7 of 8

Annual cost of heat loss (C_A)

Rate of heat loss, $Q_T = 1853.32\,\mathrm{J/s}$

Total heat loss in a year

$$Q_{year} = \frac{Q_T \times N}{\eta_{furnance}}$$

$$= \frac{1853.988 \times 3100 \times 3600}{0.79}$$

$$Q_{year} = 2.619 \times 10^{10}\,\mathrm{J}$$

$$Q_{year} = 2.619 \times 10^{7}\,\mathrm{kJ}$$

Step 8 of 8

Let us convert into therm of natural gas

We know $1\mathrm{therm} = 105500\,\mathrm{kJ}$

Therefore,

$$Q_{year} = \frac{2.619 \times 10^7}{105500}$$

$$Q_{year} = 248.246\,\mathrm{therms}$$

The annual cost,

$$C_A = Q_{year} \times C_N$$

$$= 248.246 \times 1.20$$

$$C_A = \$297.895$$

Therefore the annual cost, $\boxed{C_A = \$297.8}$

Question: 20.080P

A solar collector consists of a horizontal copper tube of outer diameter 5 cm enclosed in a concentric thin glass tube of 9 cm diameter. Water is heated as it flows through the tube, and the annular space between the copper and glass tube is filled with air at 1 atm pressure. During a clear day, the temperatures of the tube surface and the glass cover are measured to be 60°C and 32°C, respectively. Determine the rate of heat loss from the collector by natural convection per meter length of the tube.

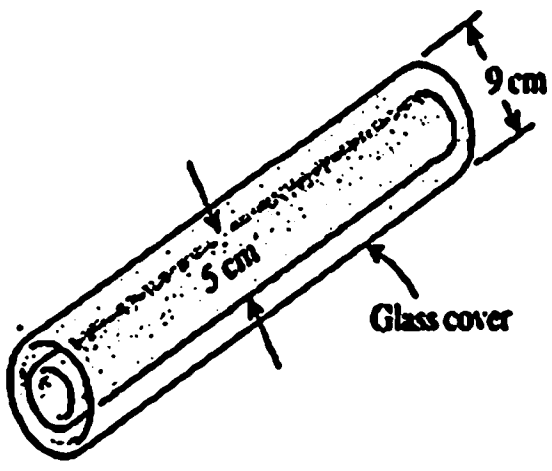


FIGURE P20–80

Step 1 of 3

The rate of heat loss from the collector due to convection is determined by estimating the effective thermal conductivity. In order to determine the effective thermal conductivity, first take the properties at average temperature and calculate the Rayleigh number then use the relation for shape factor.

Calculate the average temperature. Substitute 60°C for T_1 and 32°C for T_2

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{60+32}{2}$$
$$= 46°C$$

Determine the volume expansion coefficient.

$$\beta = \frac{1}{T_{avg} + 273}$$
$$= \frac{1}{46+273}$$
$$= 0.003135\ K^{-1}$$

Step 2 of 3

Calculate the characteristic length. Substitute 0.09 m for D_2 and 0.05 m for D_1

$$L_c = \frac{D_1 - D_2}{2}$$
$$= \frac{0.09-0.05}{2}$$
$$= 0.02\ m$$

Calculate the shape factor. Substitute 0.02 m for L_c, 0.09 m for D_2 and 0.05 m for D_1

$$F = \frac{\left[\ln(D_1/D_2)\right]^4}{L_c^3\left[D_1^{-3/5} + D_2^{-3/5}\right]^5}$$
$$= \frac{\left[\ln(0.09/0.05)\right]^4}{0.02^3\left[0.09^{-3/5} + 0.05^{-3/5}\right]^5}$$
$$= 0.1303$$

Step 3 of 3

Calculate the Rayleigh number. Substitute 9.81 m/s^2 for g, 0.7238 for Pr, 0.003135 K^{-1} for β, 0.02 m for L, 1.76×10^{-5} m^2/s for v, 60°C for T_1 and 32°C for T_2

$$\text{Ra} = \frac{g\beta(T_1 - T_2)D^3}{v^2} \times \text{Pr}$$
$$= \frac{9.81\times0.003135\times(60-32)(0.02)^3}{\left(1.76\times10^{-5}\right)^2} \times 0.7238$$
$$= 16100$$

Determine the effective thermal conductivity. Substitute 0.1303 for F, 0.7238 for Pr, 16100 for Ra and 0.02706 W/m·°C for k

$$k_{eff} = 0.386\times k\times\left(\frac{\text{Pr}}{0.861+\text{Pr}}\right)^{0.25}(F\times\text{Ra})^{0.25}$$
$$= 0.386\times0.02706\left(\frac{0.7238}{0.861+0.7238}\right)^{0.25}(0.1303\times16100)^{0.25}$$
$$= 0.05811\ \text{W/m}\cdot°C$$

Calculate the heat loss from the collector. Substitute 0.05811 W/m·°C for k_{eff}, 0.09 m for D_2, 0.05 m for D_1, 60°C for T_1 and 32°C for T_2

$$\dot{Q} = \frac{2\pi k_{eff}(T_1 - T_2)}{\ln(D_2/D_1)}$$
$$= \frac{2\pi(0.05811)(60-32)}{\ln(0.09/0.05)}$$
$$= 17.4\ \text{W}$$

Thus the heat loss from the collector is **17.4 W**

Question:

A solar collector consists of a horizontal aluminum tube of outer diameter 5 cm enclosed in a concentric thin glass tube of 7 cm diameter. Water is heated as it flows through the aluminum tube, and the annular space between the aluminum and glass tubes is filled with air at 1 atm pressure. The pump circulating the water fails during a clear day, and the water temperature in the tube starts rising. The aluminum tube absorbs solar radiation at a rate of 20 W per meter length, and the temperature of the ambient air outside is 30°C. Approximating the surfaces of the tube and the glass cover as being black (emissivity $\varepsilon = 1$) in radiation calculations and taking the effective sky temperature to be 20°C, determine the temperature of the aluminum tube when equilibrium is established (i.e., when the net heat loss from the tube by convection and radiation equals the amount of solar energy absorbed by the tube).

Step 1 of 13

Let assume the outside surface temperature of glass tube, $T_s = 35°C$

Calculate the Mean film temperature of the air,

$$T_f = (T_s + T_\infty)/2$$

Here,

Outside air temperature, T_∞

Substitute $30°C$ for T_∞ $35°C$ for T_s

$$T_f = (T_s + T_\infty)/2$$

$$= (35 + 30)/2$$

$$= 32.5°C$$

Step 2 of 13

From the properties of the air,

At $T_f = 32.5°C$

Use the method of interpolating properties between $(30 - 35)°C$

Thermal conductivity,

$$k = \frac{0.02588 + 0.02625}{2}$$

$$= 0.026065\ W/m \cdot °C$$

Prandtl number,

$$Pr = \frac{0.7268 + 0.7282}{2}$$

$$= 0.7275$$

Kinematic viscosity,

$$\nu = \frac{(1.608 + 1.655)10^{-5}}{2}$$

$$= 1.6315 \times 10^{-5}\ m^2/s$$

Calculate the Coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{32.5 + 273}$$

$$\beta = 3.273 \times 10^{-3}\ K^{-1}$$

Step 3 of 13

Characteristic length (L_c) = Diameter of Horizontal cylinder

$$L_c = 0.07\ m$$

Calculate Rayleigh number,

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \cdot Pr$$

Here,

Acceleration due gravity is g

Substitute $9.81\ m/s^2$ for g, $3.273 \times 10^{-3}\ K^{-1}$ for β, $0.07\ m$ for L_c 0.7275 for Pr, $1.6315 \times 10^{-5}\ m^2/s$ for ν

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \cdot Pr$$

$$= \frac{9.81 \times 3.273 \times 10^{-3} \times (35 - 30)(0.07)^3}{(1.6315 \times 10^{-5})^2} \times (0.7275)$$

$$Ra = 150500.512$$

Step 4 of 13

From table "empirical correlations for the average Nusselt number for natural convection over surfaces" Nusselt number for horizontal cylinders is

$$Nu = \left[0.6 + \frac{0.387\,Ra^{1/6}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right]^2$$

Substitute 150500.512 for Ra

$$Nu = \left[0.6 + \frac{0.387\,Ra^{1/6}}{\left[1 + \left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$Nu = \left[0.6 + \frac{0.387\,Ra^{1/6}}{\left[1+\left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.6 + \frac{0.387(150500.512)^{1/6}}{\left[1+\left(\frac{0.559}{0.7275}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.6 + \frac{2.8225}{1.2022}\right]^2$$

$$Nu = 8.689$$

Calculate Heat transfer coefficient,

$$h = \frac{k}{L_c} Nu$$

Here,

Heat transfer coefficient is h

Substitute 8.689 for Nu, $0.07\,m$ for L_c, $0.026065\,W/m \cdot K$ for k

$$h = \frac{k}{L_c} Nu$$

$$= \frac{0.026065 \times 8.689}{0.07}$$

$$h = 3.235\,W/m^2 \cdot {}^\circ C$$

Step 5 of 13

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$$1K = 1°C + 273$$

Calculate the rate of heat loss,

$$Q = Q_{conv} + Q_{rad}$$

$$= hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_{eff}^4\right)$$

$$= h(\pi DL)(T_s - T_\infty) + \varepsilon(\pi DL)\sigma\left(T_s^4 - T_{eff}^4\right)$$

Here,

Rate of solar radiation, Q_{rad}

Outside air temperature, T_∞

Emissivity of surface, ε

Effective sky temperature, T_{eff}

Substitute $20°C$ for T_{eff}, $30°C$ for T_∞, 1 for ε, $20\,W/m$ for Q_{rad}

$$20 = 3.235 \times \pi \times 0.07 \times 1 \times (T_s - 30)$$

$$+ (1)(\pi \times 0.07 \times 1) \times 5.67 \times 10^{-8} \times \left[(T_s + 273)^4 - (30 + 273)^4\right]$$

$$20 = 0.7114(T_s - 30) + 1.25 \times 10^{-8}\left[(T_s + 273)^4 - (303)^4\right]$$

$$T_s = 39.23°C$$

Therefore surface temperature of the glass tube, $T_s = 39.23°C$.

Step 6 of 13

Therefore temperature of glass tube surface,

$$T_s = T_2 = 39.23°C$$

Now calculating heat transfer through the air space between Aluminum tube and glass cover

Assuming Aluminum tube surface temperature,

$$T_1 = 45°C$$

Calculate the Mean film temperature of the aluminum,

$$T_{avg} = (T_1 + T_2)/2$$

$$= (45 + 39.23)/2$$

$$= 42.115°C$$

Step 7 of 13

From properties of air table at $T_{avg} = 42.115°C$

Thermal conductivity,

$$k = 0.02662 + \frac{0.02699 - 0.02662}{(45 - 40)}(42.115 - 40)$$

$$= 0.02678\,W/m \cdot {}^\circ C$$

Prandtl number,

$$Pr = 0.7255 + \frac{0.7241 - 0.7255}{(45 - 40)}(42.115 - 40)$$

$$= 0.7249$$

Kinematic viscosity,

$$v = 1.702 \times 10^{-5} + \frac{(1.750 - 1.702) \times 10^{-5}}{(45 - 40)}(42.115 - 40)$$

$$= 1.7224 \times 10^{-5}\,m^2/s$$

Step 8 of 13

$$=1.7224\times10^{-5}\ \mathrm{m^2/s}$$

Step 8 of 13

Calculate the Coefficient of volume expansion,

$$\beta=\frac{1}{T_{avg}}$$
$$=\frac{1}{273+42.115}$$
$$=3.1734\times10^{-3}\ \mathrm{K^{-1}}$$

Step 9 of 13

Find the Characteristic length,

$$(L_C)=(D_o-D_i)/2$$
$$=(0.07-0.05)/2$$
$$=0.01\mathrm{m}$$

Calculate Rayleigh number,

$$\mathrm{Ra}=\frac{g\beta(T_1-T_2)L_C^3}{\nu^2}\times\mathrm{Pr}$$
$$=\frac{9.81\times3.1734\times10^{-3}(45-39.23)(0.01)^3}{\left(1.7224\times10^{-5}\right)^2}\times(0.7249)$$
$$=438.914$$

Calculate the geometric factor for concentric cylinders,

$$F_{cyl}=\frac{\left[\ln\frac{D_o}{D_i}\right]^4}{L_C^3\left[D_i^{-3/5}+D_o^{-3/5}\right]^5}$$

Here,

Geometric factor for concentric cylinders are F_{cyl}

Substitute $0.07\,\mathrm{m}$ for D_0, $0.05\,\mathrm{m}$ for D_i, $0.01\,\mathrm{m}$ for L_c

$$F_{cyl}=\frac{\left[\ln\frac{D_o}{D_i}\right]^4}{L_C^3\left[D_i^{-3/5}+D_o^{-3/5}\right]^5}$$
$$=\frac{\left(\ln\frac{0.07}{0.05}\right)^4}{(0.01)^3\left[(0.05)^{-3/5}+(0.07)^{-3/5}\right]^5}$$
$$=\frac{0.012817}{0.1585}$$
$$=0.08086$$

Step 10 of 13

Calculate effective thermal conductivity,

$$k_{eff}=0.386k\left[\frac{\mathrm{Pr}}{0.861+\mathrm{Pr}}\right]^{0.25}\left(F_{cyl}\cdot\mathrm{Ra}\right)^{0.25}$$

Substitute 0.7249 for Pr, $0.02678\,\mathrm{W/m\cdot K}$ for k, 438.914 for Ra, 0.08086 for F_{cyl}

$$k_{eff}=0.386k\left[\frac{\mathrm{Pr}}{0.861+\mathrm{Pr}}\right]^{0.25}\left(F_{cyl}\cdot\mathrm{Ra}\right)^{0.25}$$
$$=0.386\times(0.02678)\left[\frac{0.7249}{0.861+0.7249}\right]^{0.25}\times(0.08086\times438.914)^{1/4}$$
$$=0.00849\times2.4407$$
$$=0.02072\,\mathrm{W/m\cdot{}^\circ C}$$

Step 11 of 13

Calculate the heat transfer due to convection between the cylinders is

$$Q_{conv}=\frac{2\pi k_{eff}}{\ln\left(\frac{D_o}{D_i}\right)}(T_1-T_2)$$
$$=\frac{2\pi\times0.02072}{0.33647}(T_1-39.23)$$
$$=0.386(T_1-39.23)$$

Step 12 of 13

Calculate Surface area of the plate per unit length of the tube,

$$A_s=\pi D_iL$$

Here,

Length of the plate is L

Substitute $1\,\mathrm{m}$ for L

$$A_s=\pi D_iL$$
$$=\pi\times(0.05)\times1$$
$$A_s=0.1571\,\mathrm{m^2}$$

Calculate Heat transfer through radiation,

$$Q_{rad}=\varepsilon A_s\sigma\left(T_s^4-T_{amb}^4\right)$$

Here,

Emissivity is ε

$$Q_{rad} = \varepsilon A_s \sigma \left(T_s^4 - T_{amb}^4 \right)$$

Here,

Emissivity is ε

Stefan Boltzmann constant is σ

substitute $5.67 \times 10^{-8}\ \mathrm{W/m^2 \cdot K^4}$ for σ, 1 for ε

$$Q_{rad} = 1 \times 0.1571 \times 5.67 \times 10^{-8} \left[(T_1 + 273)^4 - (273 + 39.23)^4 \right]$$

$$= 8.91 \times 10^{-9} \left[(T_1 + 273)^4 - (312.23)^4 \right]$$

Step 13 of 13

Calculate the total heat transfer,

$$Q = Q_{conv} + Q_{rad}$$

$$20 = 0.386 (T_1 - 39.23) + 8.91 \times 10^{-9} \left[(T_1 + 273)^4 - (312.23)^4 \right]$$

Solve for T_1

$$T_1 = 52.2137^\circ\mathrm{C}$$

Therefore temperature of the aluminum tube when equilibrium is established, $\boxed{52.2137^\circ\mathrm{C}}$

Question:

The components of an electronic system dissipating 150 W are located in a 5-ft-long horizontal duct whose J cross section is 6 in × 6 in. The components in the duct are J cooled by forced air, which enters at 85°F at a rate of 22 cfm and leaves at 100°F. The surfaces of the sheet metal duct are 1 not painted, and thus radiation heat transfer from the outer surfaces is negligible. If the ambient air temperature is 80°F determine (*a*) the heat transfer from the outer surfaces of the 1 duct to the ambient air by natural convection and (*b*) the average temperature of the duct.

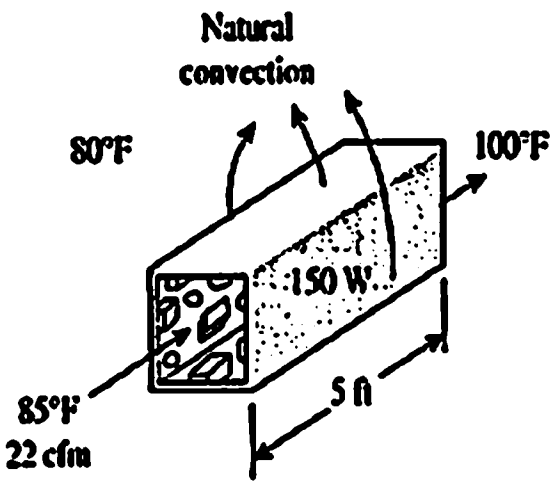


FIGURE P20–82E

Step 1 of 11

Given data:

Heat dissipation by system, $Q = 150\,\text{W}$

Length of Horizontal duct, $L = 5\,\text{ft}$

Width of duct, $W = 6\,\text{in} = 0.5\,\text{ft}$

Height of duct, $H = 6\,\text{in} = 0.5\,\text{ft}$

Forced air temperature, $T_{in} = 85°\text{F}$

Flow rate of air, $V = 22\,\text{cfm}\left(\text{ft}^3/\text{min}\right)$

Exit temperature of air, $T_{exit} = 100°\text{F}$

Ambient air temperature, $T_\infty = 80°\text{F}$

Step 2 of 11

Assuming surface temperature of the duct is $T_s = 90°\text{F}$

Film temperature,

$$T_f = (T_s + T_\infty)/2$$

$$= (90+80)/2$$

$$T_f = 85°\text{F}$$

Heat transfer by natural convection,

$$Q_{conv} = hA_s(T_s - T_\infty) \qquad \ldots\ldots (1)$$

Taking properties of air at 1atm and at temperature $T_f = 85°\text{F}$

Thermal conductivity, $k = 0.01493\,\text{Btu/h}\cdot\text{ft}\cdot°\text{F}$

Prandtl number, $\text{Pr} = 0.72825$

Kinematic viscosity, $\nu = 1.725\times10^{-4}\,\text{ft}^2/\text{s}$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{85+460}$$

$$\beta = 1.835\times10^{-3}\,\text{R}^{-1}$$

Step 3 of 11

Considering Horizontal top surface:

Characteristic length,

$$L_c = \frac{A_s}{p}$$

where, A_s – Surface Area

p = Perimeter of surface

$$L_c = \frac{5\times0.5}{2(5+0.5)}$$

$$L_c = 0.2273\,\text{ft}$$

Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$

$$= \frac{32.2\times1.835\times10^{-3}\times(90-80)\times(0.2273)^3}{\left(1.725\times10^{-4}\right)^2}(0.72825)$$

$$\text{Ra} = 1.698\times10^5$$

Step 4 of 11

Nusselt number,

$$\text{Nu} = 0.54\,\text{Ra}^{0.25}$$

$$= 0.54\left(1.698\times10^5\right)^{0.25}$$

$$\text{Nu} = 10.96$$

Heat transfer coefficient on top surface,

$$h_{top} = \frac{k}{L_c}\cdot\text{Nu}$$

$$= \frac{0.01493\times10.96}{0.2273}$$

$$h_{top} = 0.72\,\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

Surface Area,

$$A_s = 5\times0.5$$

$$A_s = 2.5\,\text{ft}^2$$

$$Q_{top} = h_{top}A_s(T_s - T_\infty)$$

$$Q_{top} = 0.72\times2.5(T_s - 80)$$

$$Q_{top} = 1.8(T_s - 80) \qquad \ldots\ldots (2)$$

Step 5 of 11

Considering Horizontal bottom surface:

The Nusselt number in case of Horizontal bottom hot surface

Step 5 of 11

Considering Horizontal bottom surface:
The Nusselt number in case of Horizontal bottom hot surface,

$$\mathrm{Nu} = 0.27(\mathrm{Ra})^{0.25}$$

$$= 0.27\left(1.698\times10^{5}\right)^{0.25}$$

$$\mathrm{Nu} = 5.48$$

Step 6 of 11

Heat transfer coefficient on bottom surface:-

$$h_{\text{bottom}} = \frac{k}{L_c}\times\mathrm{Nu}$$

$$= \frac{0.01493\times5.48}{0.2273}$$

$$h_{\text{bottom}} = 0.36\,\mathrm{Btu/h\cdot ft^2\cdot{}^oF}$$

Heat transfer through bottom surface by natural convection,

$$Q_{\text{bottom}} = h_{\text{bottom}} A_s (T_s - T_\infty)$$

$$= 0.36\times2.5(T_s - 80)$$

$$Q_{\text{bottom}} = 0.9(T_s - 80) \qquad \text{..... (3)}$$

Step 7 of 11

Considering vertical side surface (Right, left):
The characteristic length $L_c = H = 0.5\,\mathrm{ft}$
Rayleigh number,

$$\mathrm{Ra} = \frac{g\beta(T_s - T_\infty)L_C^3}{\nu^2}\cdot\mathrm{Pr}$$

$$= \frac{32.2\times1.835\times10^{-3}(90-80)(0.5)^3}{\left(1.725\times10^{-4}\right)^2}(0.72825)$$

$$\mathrm{Ra} = 1.808\times10^{6}$$

Nusselt number in case of vertical surfaces,

$$\mathrm{Nu} = \left[0.825 + \frac{0.387\mathrm{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\mathrm{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.825 + \frac{0.387\times\left(1.808\times10^{6}\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.72825}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$\mathrm{Nu} = 19.47$$

Step 8 of 11

Therefore heat transfer coefficient on vertical sides,

$$h_{\text{vertical}} = \frac{k}{L_c}\times\mathrm{Nu}$$

$$= \frac{0.01493}{0.2273}(19.47)$$

$$h_{\text{vertical}} = 1.2788\,\mathrm{Btu/h\cdot ft^2\cdot{}^oF}$$

Surface Area of the vertical surface,

$$A_s = 2\times5\times0.5$$

$$= 5\,\mathrm{ft^2}$$

Heat transfer by vertical sides by natural convection, Q_{vertical}.

$$Q_{\text{vertical}} = h_{\text{vertical}}\cdot A_s(T_s - T_\infty)$$

$$= 1.2788\times5(T_s - 80)$$

$$Q_{\text{vertical}} = 6.394(T_s - 80) \qquad \text{..... (4)}$$

The amount of energy dissipated,

$$\dot{Q}_{\text{total}} = 150\,\mathrm{W}$$

$$= 150\times3.412\,\mathrm{Btu/h}$$

$$\dot{Q}_{\text{total}} = 511.8\,\mathrm{Btu/h}$$

Step 9 of 11

(a)Calculating heat transfer by forced air flow,

$$Q_{\text{forced}} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}})$$

Properties of air taken at average temperature

$$T_{avg} = \frac{T_{\text{in}} + T_{\text{out}}}{2}$$

$$= \frac{85+100}{2}$$

$$T_{avg} = 92.5^\circ\mathrm{F}$$

At this average temperature properties of air,

$$\rho = 0.07217 + (0.07088 - 0.07217)\left(\frac{92.5-90}{100-90}\right)$$

$$\rho = 0.07184\,\mathrm{lbm/ft^3}$$

$$c_p = 0.2404\,\mathrm{Btu/lbm^\circ F}$$

Mass flow rate of air,

$$\dot{m} = \rho V$$

$$= 0.07184\times22$$

$$\dot{m} = 1.58048\,\mathrm{lbm/min}$$

Step 10 of 11

Therefore forced convection heat transfer is

$$Q_{\text{forced}} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}})$$

$$= 1.58048\times0.2404(100-85)$$

$$Q_{\text{forced}} = 5.6992\,\mathrm{Btu/min}$$

Therefore forced convection heat transfer is

$$Q_{forced} = \dot{m} c_p (T_{out} - T_{in})$$

$$= 1.58048 \times 0.2404 (100 - 85)$$

$$Q_{forced} = 5.6992 \text{ Btu/min}$$

$$Q_{forced} = 341.952 \text{ Btu/h}$$

Total heat transfer,

$$Q_{total} = Q_{forced} + Q_{conv}$$

$$Q_{conv} = Q_{total} - Q_{forced}$$

$$= 511.8 - 341.952$$

$$Q_{conv} = 169.848 \text{ Btu/h}$$

Therefore heat transfer due to convection, $\boxed{Q_{conv} = 169.848 \text{ Btu/h}}$

Step 11 of 11

(b)

The average temperature of the duct

$$Q_{conv} = Q_{top} + Q_{bottom} + Q_{vertical}$$

$$169.848 = 1.8(T_s - 80) + 0.9(T_s - 80) + 6.394(T_s - 80)$$

$$169.848 = (T_s - 80)[9.094]$$

$$T_s = 98.676^\circ\text{F}$$

The value obtained is closer to the assumed value.

Therefore the surface temperature of the duct, $\boxed{T_s = 98.676^\circ\text{F}}$

Question: 20.083P

Repeat Prob. 20–82E for a circular horizontal duct of diameter 4 in.

Step 1 of 9

Given data:

Dissipation of heat from the system, $Q = 150\,\text{W}$

Diameter of duct,

$D = 4\,\text{in.}$

$= 0.333\,\text{ft}$

Length of the duct, $L = 5\,\text{ft}$

Temperature of air entering into the duct, $T_{in} = 85°\text{F}$

Exit temperature of air from the duct, $T_{exit} = 100°\text{F}$

Flow rate of air, $V = 22\,\text{cfm}$

Ambient air temperature, $T_\infty = 80°\text{F}$

Step 2 of 9

Assume surface temperature of the duct is $T_s = 120°\text{F}$

Then film temperature of air,

$T_f = (T_s + T_\infty)/2$

$= 120 + 80/2$

$T_f = 100°\text{F}$

From properties of air table at 1atm and $T_f = 100°\text{F}$

Thermal conductivity

$k = 0.01529\,\text{Btu/h-ft-°F}$

Prandtl number,

$\text{Pr} = 0.7260$

Kinematic viscosity,

$\nu = 1.809\times10^{-4}\,\text{ft}^2/\text{s}$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{100+460}$$

$\beta = 1.786\times10^{-3}\,\text{R}^{-1}$

Step 3 of 9

Heat transfer by natural convection,

$Q_{conv} = hA_s(T_s - T_\infty)$ ..... (1)

As given, total heat dissipation from the system,

$\dot{Q}_{total} = 150\,\text{W}$

$= 150\times3.412$

$\dot{Q}_{total} = 511.8\,\text{Btu/h}$

Heat transfer by forced air flow,

$Q_{forced} = \dot{m}c_p(T_{out} - T_{in})$

Step 4 of 9

Mass flow rate of air,

$\dot{m} = \rho v$

Taking Density, ρ at average temperature of air,

$T_{avg} = (T_{in} + T_{out})/2$

$= 85 + 100/2$

$T_{avg} = 92.5°\text{F}$

From properties of air table,

$$\rho_{92.5°\text{C}} = 0.07217 + (0.07088 - 0.07217)\left(\frac{92.5-90}{100-90}\right)$$

$\rho_{92.5°\text{C}} = 0.07184\,\text{lbm/ft}^3$

Step 5 of 9

The mass flow rate of air,

$\dot{m} = \rho V$

$= 0.07184\times22$

$\dot{m} = 1.58048\,\text{lbm/min}$

$\dot{m} = 94.828\,\text{lbm/h}$

Specific heat at constant pressure, $c_p = 0.2404\,\text{Btu/lbm°F}$

Step 6 of 9

Heat transfer by forced air flow,

$Q_{forced} = \dot{m}c_p(T_{out} - T_{in})$

$= 94.828\times0.2404\times(100-85)$

$= 341.949\,\text{Btu./h}$

Total heat transfer,

$Q_{total} = Q_{conv} + Q_{forced}$

$Q_{conv} = Q_{total} - Q_{forced}$

$= 511.8 - 341.949$

$= 169.851\,\text{Btu/h}$

Therefore convection heat transfer, $\boxed{Q_{conv} = 169.85\,\text{Btu/h}}$

Step 7 of 9

(b)

Calculating Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_C^3}{\nu^2}\times\text{Pr}$$

$$= \frac{32.2\times1.786\times10^{-3}(120-80)(0.33)^3}{(1.809\times10^{-4})^2}(0.7260)$$

$= 1.833\times10^6$

Nusselt number for circular duct

$$\text{Nu} = \left[0.6 + \frac{0.387\,\text{Ra}^{1/6}}{\left[1+\left(\frac{0.559}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.6 + \frac{0.387(1.833\times10^6)^{1/6}}{\left[1+\left(\frac{0.559}{0.7260}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.6 + \frac{4.2808}{1.202}\right]^2$$

$= 17.31$

Step 8 of 9

Heat transfer coefficient,

$$h = \frac{k}{L_C}\times\text{Nu}$$

$$= \frac{0.01529\times17.31}{0.33}$$

$h = 0.802\,\text{Btu/h-ft}^2\text{-°F}$

Surface Area,

$A_s = \pi DL$

$= \pi\times0.33\times5$

$A_s = 5.184\,\text{ft}^2$

Step 9 of 9

Then from the equation (1), we have

Convective heat transfer,

$Q_{conv} = hA_s(T_s - T_\infty)$

$169.85 = 0.802\times5.184(T_s - 80)$

$T_s = 120.8532°\text{F}$

The surface temperature is approximately equal to the assumed value.

Therefore the actual surface temperature of the duct, $\boxed{T_s = 120.8532°\text{F}}$

Question:

Repeat Prob. 20–82E assuming the fan fails and thus the entire heat generated inside the duel must be rejected to the ambient air by natural convection through the outer surfaces of the duel.

Step 1 of 10

Consider that the surface temperature of the duct (T_s) is $135°F$.

Calculate the mean film temperature (T_f) by using the following relation:

$$T_f = \frac{T_s + T_\infty}{2}$$

Here, T_∞ is the temperature of the surrounding air and T_s is the surface temperature of the duct.

Substitute $135°F$ for T_s, and $80°F$ for T_∞.

$$T_f = \frac{135+80}{2}$$
$$= 107.5°F$$

Calculate the volume expansion coefficient (β).

$$\beta = \frac{1}{t_f}$$

Substitute $(107.5+460)\ R$ for t_f.

$$\beta = \frac{1}{107.5+460}$$
$$= 0.001680\ R^{-1}$$

Step 2 of 10

Use interpolation technique and obtain the following properties of air at $1\ atm$ and $107.5°F$ from table A-15E (Properties of air at $1\ atm$) in the text book.

Interpolate the Thermal conductivity.

$$k = 0.01529 + \frac{0.01552-0.01529}{110-100}(107.5-100)$$
$$= 0.01546\ Btu/h \cdot ft \cdot R$$

Interpolate the Kinematic viscosity.

$$\nu = 1.809\times10^{-4} + \frac{1.866\times10^{-4}-1.809\times10^{-4}}{110-100}(107.5-100)$$
$$= 1.8517\times10^{-4}\ ft^2/s$$

Interpolate the Prandtl number.

$$Pr = 0.7260 - \frac{0.7260-0.7245}{110-100}(107.5-100)$$
$$= 0.7848$$

Step 3 of 10

a)

As the heat transferred through radiation is negligible, and because of the failure of the fan, the total heat generated must be dissipated by natural convection only.

Therefore, the amount of heat dissipated by natural convection $(Q_{natural})$ is $\boxed{150\ W}$.

Step 4 of 10

b)

Calculate the horizontal top surface area of the duct from the geometry of the duct.

$$A_{top} = 5\times\frac{6}{12}$$
$$= 2.5\ ft^2$$

Calculate the perimeter of the top surface of the duct from the geometry of the duct.

$$P_{top} = 2\times\left(5+\frac{6}{12}\right)$$
$$= 11\ ft$$

Calculate the characteristic length for horizontal surface of the duct.

$$L_c = \frac{A_{top}}{P_{top}}$$

Here, A is the area of horizontal surface and P is the perimeter of horizontal surface.

Substitute $2.5\ ft^2$ for A_{top}, and $11\ ft$ for P_{top}.

$$L_c = \frac{2.5}{11}$$
$$= 0.2273\ ft$$

Step 5 of 10

Calculate the Rayleigh number for horizontal surface of the duct.

$$Ra = \frac{g\beta(T_s-T_\infty)L_c^3}{\nu^2}Pr$$

Here, g is the acceleration due to gravity, volume expansion coefficient is, and characteristic length of the duct is L_c.

Substitute $32.2\ ft/s^2$ for g, $0.001680\ R^{-1}$ for β, $135°F$ for T_s, $80°F$ for T_∞, $0.2273\ ft$ for L_c, $1.8517\times10^{-4}\ ft^2/s$ for ν, and 0.7848 for Pr.

$$Ra = \frac{32.2\times0.001680(135-80)\times0.2273^3}{\left(1.8517\times10^{-4}\right)^2}\times0.7848$$

1.8517×10^{-4} ft^2/s for ν, and 0.7848 for Pr.

$$\text{Ra} = \frac{32.2\times0.001680(135-80)\times0.2273^3}{\left(1.8517\times10^{-4}\right)^2}\times0.7848$$
$$= 7.99\times10^5$$

Calculate the Nusselt number for horizontal top surface by using the following relation:

$$\text{Nu}_{top} = 0.54\text{Ra}^{1/4}$$

Substitute 7.99×10^5 for Ra.

$$\text{Nu}_{top} = 0.54\times\left(7.99\times10^5\right)^{1/4}$$
$$= 16.144$$

Step 6 of 10

Calculate the heat transfer coefficient for top surface of the duct by using the following relation:

$$h_{top} = \frac{k}{L_c}\text{Nu}_{top}$$

Substitute 0.01546 $\text{Btu/h}\cdot\text{ft}\cdot\text{R}$ for k, 16.144 for Nu_{top}, and 0.2273 ft for L_c.

$$h_{top} = \frac{0.01546}{0.2273}\times16.144$$
$$= 1.098 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Calculate the Nusselt number for horizontal bottom surface by using the following relation:

$$\text{Nu}_{bot} = 0.27Ra^{1/4}$$

Substitute 7.99×10^5 for Ra.

$$\text{Nu}_{bot} = 0.27\times\left(7.99\times10^5\right)^{1/4}$$
$$= 8.072$$

Step 7 of 10

Calculate the heat transfer coefficient for bottom surface of the duct by using the following relation:

$$h_{bot} = \frac{k}{L_c}\text{Nu}_{bot}$$

Substitute 0.01546 $\text{Btu/h}\cdot\text{ft}\cdot\text{R}$ for k, 8.072 for Nu_{bot}, and 0.2273 ft for L_c.

$$h_{bot} = \frac{0.01546}{0.2273}\times8.072$$
$$= 0.549 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

The horizontal bottom surface area of the duct is equal to the top surface.

$$A_{bot} = A_{top}$$

Substitute 2.5 ft^2 for A_{top}.

$$A_{bot} = 2.5 \text{ ft}^2$$

Step 8 of 10

Calculate the characteristic length for vertical surface of the duct.

$$L_c = L$$

Here, length of the vertical surface is L.

Substitute $\frac{6}{12}$ ft for L.

$$L_c = \frac{6}{12}$$
$$= 0.5 \text{ ft}$$

Calculate the Rayleigh number for vertical side surfaces of the duct.

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\text{Pr}$$

Here, acceleration due to gravity is g, volume expansion coefficient is β, and characteristic length for the vertical surface of the duct is L_c.

Substitute 32.2 ft/s^2 for g, 0.001680 R^{-1} for β, $135°\text{F}$ for T_s, $80°\text{F}$ for T_∞, 0.5 ft for L_c, 1.8517×10^{-4} ft^2/s for ν, and 0.7848 for Pr.

$$\text{Ra} = \frac{32.2\times0.001680(135-80)\times0.5^3}{\left(1.8517\times10^{-4}\right)^2}\times0.7848$$
$$= 8.512\times10^6$$

Step 9 of 10

Calculate the Nusselt number for vertical side surfaces by using the following relation:

$$\text{Nu}_{side} = \left\{0.825 + \frac{0.387Ra^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2$$

Substitute 8.512×10^6 for Ra, and 0.7848 for Pr.

$$\text{Nu}_{side} = \left\{0.825 + \frac{0.387\times\left(8.512\times10^6\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.7848}\right)^{9/16}\right]^{8/27}}\right\}^2$$
$$= 30.1948$$

$$\mathrm{Nu}_{side}=\left\{0.825+\frac{\left(\quad\right)^{2}}{\left[1+\left(\frac{0.492}{0.7848}\right)^{9/16}\right]^{8/27}}\right\}^{2}$$

20384P pg 3

$$=30.1948$$

Calculate the heat transfer coefficient for side surfaces of the duct by using the following relation:

$$h_{side}=\frac{k}{L_c}\mathrm{Nu}_{side}$$

Substitute $0.01546\ \mathrm{Btu/h\cdot ft\cdot R}$ for k, 30.1948 for Nu_{side}, and $0.5\ \mathrm{ft}$ for L_c.

$$h_{side}=\frac{0.01546}{0.5}\times 30.1948$$
$$=0.9336\ \mathrm{Btu/h\cdot ft^2\cdot ^\circ F}$$

Calculate the vertical side surface area of the duct.

$$A_{side}=2\times 5\times 0.5$$
$$=5\ \mathrm{ft^2}$$

Step 10 of 10

Calculate the average temperature of the duct by using the following relation:

$$Q_{total}=Q_{top}+Q_{bottom}+Q_{side}$$
$$=\left(h_{top}A_{top}+h_{bot}A_{bot}+h_{side}A_{side}\right)\left(T_s-T_\infty\right)$$

Substitute $150\ \mathrm{W}$ for Q_{total}, $1.098\ \mathrm{Btu/h\cdot ft^2\cdot ^\circ F}$ for h_{top}, $2.5\ \mathrm{ft^2}$ for A_{top}, $0.549\ \mathrm{Btu/h\cdot ft^2\cdot ^\circ F}$ for h_{bot}, $2.5\ \mathrm{ft^2}$ for A_{bot}, $0.9336\ \mathrm{Btu/h\cdot ft^2\cdot ^\circ F}$ for h_{side}, $5\ \mathrm{ft^2}$ for A_{side}, and $80^\circ\mathrm{F}$ for T_∞.

$$150\times 3.41214=\left(1.098\times 2.5+0.549\times 2.5+0.9336\times 5\right)\left(T_s-80\right)$$
$$150\times 3.41214=\left(1.098\times 2.5+0.549\times 2.5+0.9336\times 5\right)\left(x-80\right)$$
$$T_s=138.257^\circ\mathrm{F}$$

The calculated average temperature of the duct is very close to the assumed temperature and hence there is no need to repeat the calculation.

Therefore, the average temperature of the duct is $\boxed{138.257^\circ\mathrm{F}}$.

Question:

Consider a cold aluminum canned drink that is initially at a uniform temperature of 5°C. The can is 12.5 cm high and has a diameter of 6 cm. The emissivity of the outer surface of the can is 0.6. Disregarding any heat transfer from the bottom surface of the can, determine how long will take for the average temperature of the drink to rise to 7°C if the surrounding air and surfaces are at 25°C.

Step 1 of 9

Diameter of can, $D = 6\,\text{cm} = 0.06\,\text{m}$

Ambient air temperature, $T_\infty = 25^\circ\text{C}$

Ambient surface temperature, $T_{amb} = 25^\circ\text{C}$

Emissivity of the outer surface of the can, $\varepsilon = 0.6$

Cold drink temperature, $T_1 = 5^\circ\text{C}$

Temperature rise of the drink, $T_2 = 7^\circ\text{C}$

Height of can,

$$H = 12.5\,\text{cm}$$
$$= 0.125\,\text{m}$$

Step 2 of 9

The average surface temperature of can surface,

$$T_s = \frac{5+7}{2}$$
$$T_s = 6^\circ\text{C}$$

The film temperature,

$$T_f = (T_s + T_\infty)/2$$
$$= (6+25)/2$$
$$T_f = 15.5^\circ\text{C}$$

The heat transfer rate,

$$Q = hA_s(T_\infty - T_s) + \varepsilon A_s \sigma\left(T_{amb}^4 - T_s^4\right) \quad \ldots\ldots (1)$$

From properties of air table the properties of air at 1atm and at film temperature,

Thermal conductivity,

$$k = 0.02476 + \frac{0.02514 - 0.02476}{(20-15)}(15.5-15)$$
$$k = 0.024798\,\text{W/m}^\circ\text{C}$$

Step 3 of 9

Prandtl number,

$$\text{Pr} = 0.7323 + \frac{0.7309 - 0.7323}{(20-15)}(15.5-15)$$
$$\text{Pr} = 0.73216$$

Kinematic viscosity,

$$\nu = 1.470\times10^{-5} + \frac{(1.516-1.470)\times10^{-5}}{(20-15)}(15.5-15)$$
$$\nu = 1.4746\times10^{-5}\,\text{m}^2/\text{s}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{15.5+273}$$
$$\beta = 3.46\times10^{-3}\,\text{K}^{-1}$$

Step 4 of 9

Characteristic length L_c = Height of can, $H = 0.125\,\text{m}$

Calculating Rayleigh number

$$\text{Ra} = \frac{g\beta(T_\infty - T_s)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.46\times10^{-3}(25-6)(0.125)^3}{\left(1.4746\times10^{-5}\right)^2}\times0.73216$$
$$\text{Ra} = 4.25\times10^6$$

Grashoff number,

$$\text{Gr} = \frac{\text{Ra}}{\text{Pr}}$$
$$= \frac{4.25\times10^6}{0.73216}$$
$$\text{Gr} = 5.8047\times10^6$$

Step 5 of 9

Checking whether the cylindrical can be treated as vertical plate,

If

$$D \geq \frac{35L}{\text{Gr}^{0.25}}$$

$$\frac{35L}{\text{Gr}^{0.25}} = \frac{35\times(0.125)}{\left(5.8047\times10^6\right)^{0.25}}$$

$$\frac{35L}{\text{Gr}^{0.25}} = 0.08913$$

As $D = 0.125$

Therefore,

$D \geq \frac{35L}{\text{Gr}^{0.25}}$ We can treat it as vertical plate.

Step 6 of 9

Therefore Nusselt number in case of vertical plates,

$$\text{Nu} = \left[0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$

$$Nu=\left[0.825+\frac{0.387Ra^{1/6}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{8/27}}\right]^{2}$$ 085P pg 2

$$=\left[0.825+\frac{0.387\left(4.25\times10^{6}\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.73216}\right)^{9/16}\right]^{8/27}}\right]^{2}$$

$Nu=24.64$

Heat transfer coefficient,

$$h=\frac{k}{L_c}Nu$$

$$=\frac{0.024798\times24.64}{0.125}$$

$$h=4.89\,W/m^2{}^\circ C$$

Step 7 of 9

Area, A_s = Area of cylindrical surface + Area of top of the can

$$A_s=\pi DH+\frac{\pi D^2}{4}$$

$$=\pi\left(0.06\times0.125+\frac{(0.06)^2}{4}\right)$$

$$A_s=0.02639\,m^2$$

On substituting the known values in equation (1), we get

$$Q=hA_s\left(T_\infty-T_s\right)+\varepsilon A_s\sigma\left(T_{amb}^4-T_s^4\right)$$

$$=\left[4.89\times0.02639(25-6)\right]+\left[0.6\times(0.02639)\times\left(5.67\times10^{-8}\right)\times\left[(25+273)^4-(6+273)^4\right]\right]$$

$$=2.45189+1.64019$$

$$\dot{Q}=4.092\,W$$

Step 8 of 9

Heat required raising temperature from 5°C to 7°C

$$Q_{req}=mc_p\left(T_2-T_1\right)$$

Let us take average temperature of water $T_{av}=6°C$

At $T_{av}=6°C$ from saturated properties of water

$$c_p=4203\,J/kg\cdot K$$

$$\rho=999.9\,kg/m^3$$

Mass of water,

$$m=\rho V$$

$$=999.9\times\frac{\pi D^2}{4}\times L$$

$$=\frac{999.9\times\pi\times(0.06)^2\times0.125}{4}$$

$$m=0.3534\,kg$$

Step 9 of 9

Heat required raising temperature from 5°C to 7°C,

$$Q_{req}=mc_p\left(T_2-T_1\right)$$

$$=0.3534\times4203\times(7-5)$$

$$Q_{req}=2970.7\,W$$

Time taken to increase the temperature of water from 5°C to 7°C,

$$t=\frac{Q_{req}}{Q}$$

$$=\frac{2970.7}{4.092}$$

$$t=725.97\,s$$

$$t=725.97\times\frac{1}{60}$$

$$t=12.099\,min$$

Therefore time taken to increase the temperature of water from 5°C to 7°C,

$$\boxed{t=12.099\,min}$$

Question: 20.086P

A group of 25 power transistors, dissipating 1.5 W each, are to be cooled by attaching them to a black-anodized square aluminum plate and mounting the plate on the wall of a room at 30°C. The emissivity of the transistor and the plate surfaces is 0.9. Assuming the heat transfer from the back side of the plate to be negligible and the temperature of the surrounding surfaces to be the same as the air temperature of the room, determine the size of the plate if the average surface temperature of the plate is not to exceed 50°C.

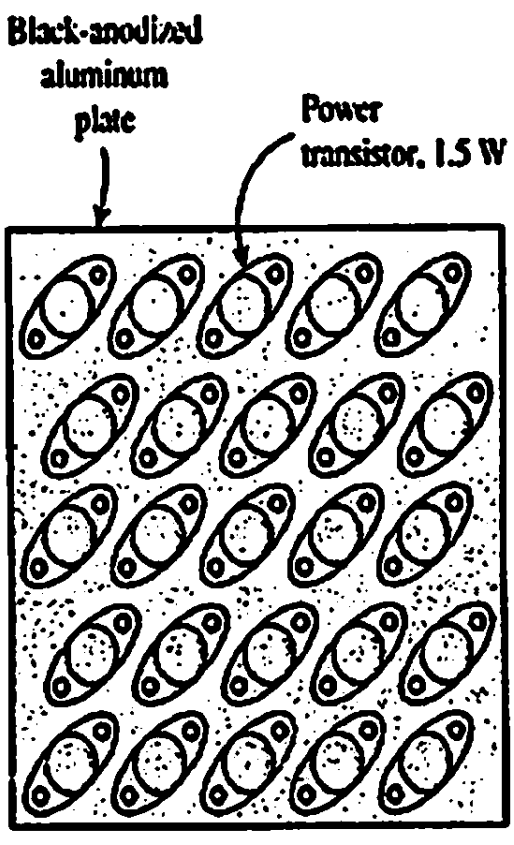


FIGURE P20–86

Step 1 of 8

Number of power transistors, $N = 25$
Dissipation of heat by a transistor, $Q = 1.5\,\text{W}$
Room temperature, $T_\infty = 30^\circ\text{C}$
Emissivity of transistor surface $\varepsilon = 0.9$
Temperature of surrounding $T_{amb} = 30^\circ\text{C}$
Average surface temperature of the plate, $T_s = 50^\circ\text{C}$

Step 2 of 8

The total heat transfer rate $\dot{Q}_t$ is given by.

$$\dot{Q}_t = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma\left(T_s^4 - T_\infty^4\right) \quad \ldots\ldots (1)$$

Taking the properties of air at 1atm and film temperature T_f,

$$T_f = (T_s + T_\infty)/2$$
$$= (30+50)/2$$
$$T_f = 40^\circ\text{C}$$

Step 3 of 8

Thermal conductivity,
$k = 0.02662\,\text{W/m}^\circ\text{K}$
Prandtl number, $\text{Pr} = 0.7255$
Kinematic viscosity, $\nu = 1.702\times10^{-5}\,\text{m}^2/\text{s}$
The coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{273+40}$$
$$\beta = 3.195\times10^{-3}\,\text{K}^{-1}$$

Step 4 of 8

Characteristic length, L_c = length of plate $= L$
Therefore Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.195\times10^{-3}(50-30)L^3}{\left(1.702\times10^{-5}\right)^2}\cdot(0.7255)$$
$$\text{Ra} = 1.57\times10^9\,L^3$$

Step 5 of 8

The Nusselt number for vertical plate is given by,

$$\text{Nu} = \left[0.825 + \frac{0.387(\text{Ra})^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{0.387\left(1.57\times10^9\,L^3\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.7255}\right)^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.825 + \frac{13.19L^{1/2}}{1.19}\right]^2$$
$$\text{Nu} = \left(0.825 + 11.08L^{1/2}\right)^2$$

Heat transfer coefficient,

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$h = \frac{0.02662}{L}\left(0.825+11.08\sqrt{L}\right)^2$$

Step 6 of 8

Surface Area of the square plate is
$A_s = L^2$
Now substituting values in equation (1)

$$\dot{Q}_t = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma\left(T_s^4 - T_\infty^4\right)$$
$$= \frac{0.02662}{L}\left(0.825+11.08\sqrt{L}\right)^2\times L^2\times(50-30)$$
$$+0.9\times L^2\times5.67\times10^{-8}\left[(273+50)^4 - (273+30)^4\right]$$
$$= 0.02662\left(0.825+11.08\sqrt{L}\right)^2 20L + 125.3L^2$$
$$\dot{Q}_t = 0.5324L\left(0.825+11.08\sqrt{L}\right)^2 + 125.3L^2 \quad \ldots\ldots (2)$$

Step 7 of 8

We know
$\dot{Q}_t = N\times Q$
Where
N =Number of transistor
Q =power dissipated by each transistor
Then,
$\dot{Q}_t = 25\times1.5\,\text{W}$

Step 8 of 8

Then substituting in equation (2),

$$\dot{Q}_t = 0.5324L\left(0.825+11.08\sqrt{L}\right)^2 + 125.3L^2$$
$$25\times1.5 = 0.5324L\left(0.825+11.07\sqrt{L}\right)^2 + 125.3L^2$$
$$37.5 = 0.5324L\left(0.825+11.07\sqrt{L}\right)^2 + 125.3L^2$$

By solving the above equation
$L = 0.4264\,\text{m}$
Therefore, size of the square aluminum plate is $\boxed{42.64\,\text{cm}\times42.64\,\text{cm}}$

Question: 20.087P

Repeat Prob. 20-86 assuming the plate to be positioned horizontally with (*a*) transistors facing up and (*b*) transistors facing down.

Step 1 of 7

Number of power transistors, $N = 25$
Dissipation of heat by a transistor, $Q = 1.5\,\text{W}$
Room temperature, $T_\infty = 30^\circ\text{C}$
Emissivity of transistor surface, $\varepsilon = 0.9$
Temperature of surrounding, $T_{\text{surr}} = 30^\circ\text{C}$
Average surface temperature of the plate, $T_s = 50^\circ\text{C}$

Step 2 of 7

The total heat transfer rate $\dot{Q}_t$ is given by.

$$\dot{Q}_t = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma\left(T_s^4 - T_\infty^4\right) \quad \ldots\ldots (1)$$

Taking the properties of air at 1atm and film temperature T_f, we have,

$$T_f = (T_s + T_\infty)/2$$
$$= (30+50)/2$$
$$T_f = 40^\circ\text{C}$$

Thermal conductivity, $k = 0.02662\,\text{W/m}^\circ\text{K}$
Prandtl number, $\text{Pr} = 0.7255$
Kinematic viscosity, $\nu = 1.702\times10^{-5}\,\text{m}^2/\text{s}$
The coefficient of volume expansion,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{273+40}$$
$$= 3.195\times10^{-3}\,\text{K}^{-1}$$

Step 3 of 7

Characteristic length, L_C for a Horizontal plate:

$$L_c = \frac{A_s}{p}$$
$$= \frac{L^2}{4L}$$
$$= \frac{L}{4}$$

Total heat dissipated by transistors,

$$\dot{Q}_t = N\times Q$$

where N is the number of transistor
Q is the power dissipated by each transistor
Hence,

$$\dot{Q}_t = 25\times1.5\,\text{W}$$

Step 4 of 7

(i) Plate to be positioned horizontally with transistors facing up.
Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.195\times10^{-3}(50-30)\left(\frac{L}{4}\right)^3}{\left(1.702\times10^{-5}\right)^2}\cdot(0.7255)$$
$$= 2.453\times10^7\,L^3$$

Assuming Rayleigh number ranges as $Ra = 10^4 \sim 10^7$, from empirical correlations of the Nusselt number,

$$\text{Nu} = 0.54\,\text{Ra}^{1/4}$$
$$= 0.54\left(2.453\times10^7\,L^3\right)^{1/4}$$
$$= 38L^{3/4}$$

Step 5 of 7

Heat transfer coefficient,

$$h = \frac{k}{L_c}\text{Nu}$$
$$= \frac{0.02662}{L/4}\times38L^{3/4}$$
$$= \frac{4.05}{L^{1/4}}$$

Surface Area of the square plate is

$$A_s = L^2$$

Substitute the known values in equation (1).

$$\dot{Q}_t = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma\left(T_s^4 - T_\infty^4\right)$$
$$= 4.05L^{-0.25}\times L^2(50-30) + 0.9\times5.67\times10^{-8}\times L^2\left[(273+50)^4 - (273+30)^4\right]$$
$$\dot{Q}_t = 81L^{1.75} + 125.311L^2$$
$$25\times15 = 81L^{1.75} + 125.311L^2$$
$$37.5 = 81L^{1.75} + 125.311L^2$$

By solving the above equation

$$L = 0.4066\,\text{m}$$

Therefore, the size of the square aluminum plate is $\boxed{40.66\times40.66\ \text{cm}^2}$.

Step 6 of 7

(ii) Plate to be positioned horizontally with transistors facing down.
Rewriting the above calculated values, we have,

$$\text{Ra} = 2.453\times10^7\,L^3$$

Then from empirical correlations of the Nusselt number,

$$\text{Nu} = 0.27\,\text{Ra}^{0.25}$$
$$= 0.27\left(2.453\times10^7\,L^3\right)^{1/4}$$
$$= 19.0L^{3/4}$$

Then heat transfer coefficient,

$$h = \frac{k}{L_c}\cdot\text{Nu}$$
$$= \frac{0.02662}{L/4}\left(19.0L^{3/4}\right)$$
$$h = 2.023L^{-1/4}$$

Step 7 of 7

Substitute the known values in equation (1).

$$\dot{Q}_t = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma\left(T_s^4 - T_\infty^4\right)$$
$$37.5 = 2.023L^{-1/4}\cdot L^2(50-30) + 0.9\times5.67\times10^{-8}\times L^2\left[(273+50)^4 - (273+30)^4\right]$$
$$37.5 = 40.46L^{1.75} + 125.3L^2$$

By solving the above equation,

$$L = 0.4638\,\text{m}$$

Therefore size of the square aluminum plate is $\boxed{46.38\times46.38\ \text{cm}^2}$.

Question: 20.088P

A 0.1-W small cylindrical resistor mounted on a lower part of a vertical circuit board is 0.3 in long and has a diameter of 0.2 in. The view of the resistor is largely blocked by another circuit board facing it, and the heat transfer through the connecting wires is negligible. The air is free to flow through the large parallel flow passages between the boards as a result of natural convection currents. If the air temperature at the vicinity of the resistor is 120°F, determine the approximate surface temperature of the resistor.

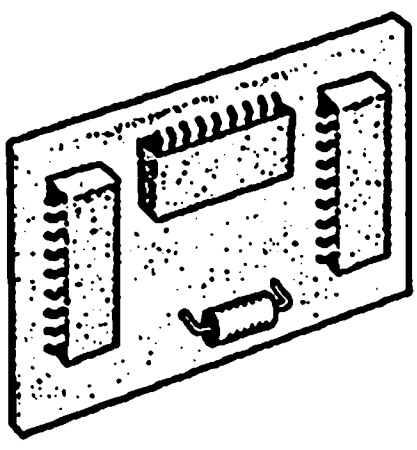


FIGURE P20–88E

Step 1 of 4

The cooling of the resistor is due to the natural convective heat transfer. In order to find the surface temperature of the resistor from the energy balance equation, we first need to find the Rayleigh's number and then the Nusselt number and heat transfer coefficient. But to find the Rayleigh number, we need the properties of air at mean film temperature. So assume the surface temperature to be 220°F. Now substitute 220°F for T_s and 120°F for T_∞

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{220+120}{2}$$
$$= 170°\text{F}$$

Step 2 of 4

The properties of air at mean film temperature are
Thermal conductivity, $k = 0.01692$ Btu/h·ft·°F
Prandtl number, $\text{Pr} = 0.7161$
Kinematic viscosity, $\nu = 0.222\times10^{-3}$ ft^2/s
Volume expansion coefficient,

$$\beta = \frac{1}{T_f + 273}$$
$$= \frac{1}{170+460}$$
$$= 0.001587\ \text{R}^{-1}$$

Step 3 of 4

Calculate the Rayleigh number. Substitute 32.2 ft/s^2 for g, 0.001587 R^{-1} for β, 220°F for T_s, 120°F for T_∞, 0.2/12 ft for D, 0.222×10^{-3} ft^2/s for ν and 0.7161 for Pr

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\times\text{Pr}$$
$$= \frac{32.2\times0.001587\times(220-120)(0.2/12)^3}{\left(0.222\times10^{-3}\right)^2}\times0.7161$$
$$= 343.8$$

Determine the Nusselt number. Substitute 343.8 for Ra and 0.7161 for Pr

$$\text{Nu} = \left[0.6 + \frac{0.387\,\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$
$$= \left[0.6 + \frac{0.387(343.8)^{1/6}}{\left[1+(0.559/0.7161)^{9/16}\right]^{8/27}}\right]^2$$
$$= 2.105$$

Step 4 of 4

Estimate the heat transfer coefficient. Substitute 2.105 for Nu, 0.01692 Btu/h·ft·°F for k, and 0.2/12 ft for D

$$h = \frac{\text{Nu}\times k}{D}$$
$$= \frac{2.105\times0.01692}{0.2/12}$$
$$= 2.138\ \text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

Determine the heat transfer rate. Substitute 2.138 Btu/h·ft^2·°F for h, 0.2/12 ft for D, 0.3/12 ft for L, (0.1×3.412) Btu/h for $\dot{Q}$ and 120°F for T_∞

$$\dot{Q} = h\left(\pi DL + 2\frac{\pi D^2}{4}\right)(T_s - T_\infty)$$

$$T_s = T_\infty + \frac{\dot{Q}}{h\left(\pi DL + 2\frac{\pi D^2}{4}\right)}$$
$$= 120 + \frac{0.1\times3.412}{2.138\left(\pi\times\frac{0.2}{12}\times\frac{0.3}{12} + 2\frac{\pi(0.2/12)^2}{4}\right)}$$
$$= 211°\text{F}$$

Thus the surface temperature of the resistor is $\boxed{211°\text{F}}$

Question: 20.089P pg 1

An ice chest whose outer dimensions are 30 cm X 40 cm × 40 cm is made of 2-cm-thick Styrofoam (k =0.033 W/m · K). Initially, the chest is filled with 35 kg of ice at 0°C, and the inner surface temperature of the ice chest can be taken to be 0°C at all times. The heat of fusion of water at 0°C is 337.7 kJ/kg, and the surrounding ambient air is at 20CC. Disregarding any heat transfer from the 40 cm × 40 cm base of the ice chest, determine how long it will take for the ice in the chest to melt completely if the ice chest is subjected to (*a*) calm air and (*b*) winds at 50 km/h. Assume the heat transfer coefficient on the front, back, and lop surfaces to be the same as that on the side surfaces.

Step 1 of 13

Given data:

Mass of ice, $m = 35\,\text{kg}$

Heat of fusion, $h_f = 337.7\,\text{kJ/kg}$

Surface temperature inside, $T_{in} = 0°\text{C}$

Surrounding Air temperature, $T_\infty = 20°\text{C}$

Length of the ice-chest,

$L = 40\,\text{cm}$

$= 0.4\,\text{m}$

Breadth of the ice-chest,

$B = 40\,\text{cm}$

$= 0.4\,\text{m}$

Width of the ice-chest,

$W = 30\,\text{cm}$

$= 0.3\,\text{m}$

Thermal conductivity of Styrofoam, $k_s = 0.033\,\text{W/m} \cdot \text{K}$

Step 2 of 13

(*a*)

Let us assume the surface temperature as $T_s = 12°\text{C}$

Film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$

$$= \frac{12+20}{2}$$

$$= 16°\text{C}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{16+273}$$

$$= 3.46\times10^{-3}\ \text{K}^{-1}$$

Step 3 of 13

Taking properties from the table of air, at 1atm and at film temperature $T_f = 16°\text{C}$

Thermal conductivity, $k = 0.02484\ \text{W/m°C}$

Prandtl number, $\text{Pr} = 0.73202$

Kinematic viscosity, $\nu = 1.4792\times10^{-5}\,\text{m}^2/\text{s}$

Characteristic length $L_C = H = 0.3\text{m}$

Rayleigh Number,

$$Ra = \frac{g\beta(T_s - T_\infty)L_C^3}{\nu^2}.\text{Pr}$$

$$= \frac{9.81\times3.46\times10^{-3}(20-12)(0.3)^3}{\left(1.4792\times10^{-5}\right)^2}(0.73202)$$

$$= 2.45\times10^7$$

Step 4 of 13

Nusselt number,

$$Nu = \left[0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.492/\text{Pr})^{9/16}\right]^{8/27}}\right]^2$$

$$= \left[0.825 + \frac{0.387\left(2.45\times10^7\right)^{1/6}}{\left[1+(0.492/0.73202)^{9/16}\right]^{8/27}}\right]^2$$

$$= 40.53$$

Heat transfer coefficient,

$$h = \frac{k}{L_C}\cdot\text{Nu}$$

$$= \frac{0.0.2484\times40.53}{0.3}$$

$$= 3.356\ \text{W/m}^2\text{°C}$$

Step 5 of 13

Surface Area of chest,

$$A_s = 4[0.3\times0.4]+[0.4\times0.4]$$

$$= 0.64\text{m}^2$$

Total resistance to heat transfer,

$$R_{total} = R_{cond} + R_{conv}$$

$$= \frac{t}{kA_s} + \frac{1}{hA_s}$$

$$= \frac{0.02}{(0.033\times0.64)} + \frac{1}{(3.356\times0.64)}$$

$$= 1.41255\ \text{K/W}$$

Step 6 of 13

The heat transfer rate is,

$$\dot{Q} = \frac{T_\infty - T_{in}}{R_{total}}$$

$$= \frac{20-0}{}$$

The heat transfer rate is,

$$\dot{Q} = \frac{T_\infty - T_{in}}{R_{total}}$$
$$= \frac{20-0}{1.41255}$$
$$= 14.158\,\text{W}$$

The relation for heat transfer due to convection is

$$\dot{Q} = hA_s(T_\infty - T_s)$$
$$14.158 = 3.356\times 0.64(20 - T_s)$$
$$T_s = 20 - \frac{14.158}{3.356\times 0.64}$$
$$= 13.40^\circ\text{C}$$

Step 7 of 13

As the obtained value of surface temperature is very close to assumed value, our assumptions are valid and heat transfer calculated can be used for obtaining rate of melting. So the Rate of ice melting is

$$\dot{m} = \frac{\dot{Q}}{h_f}$$
$$= \frac{14.158\times 10^{-3}}{337.7}$$
$$= 4.243\times 10^{-5}\ \text{kg/s}$$

Time taken for melting,

$$t = \frac{m}{\dot{m}}$$
$$= \frac{35}{4.243\times 10^{-5}}$$
$$= 824937\,\text{sec}$$
$$= 229.15\ \text{h}$$

Hence, the time taken for melting the ice is $\boxed{t = 229.15\ \text{h}}$

Step 8 of 13

(b)

When chest is subjected to winds at 50 km/hr, the forced convection prevails.

Let us assume the surface temperature as $T_s = 19^\circ\text{C}$

Now film temperature becomes,

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{20+19}{2}$$
$$= 19.5^\circ\text{C}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{(2730+19.5)}$$
$$= 3.419\times 10^{-3}\,\text{K}^{-1}$$

Step 9 of 13

Taking properties from the table of air, at 1atm and at film temperature $T_f = 19.5^\circ\text{C}$

Thermal conductivity, $k = 0.025102\,\text{W/m}^\circ\text{C}$

Prandtl number, $\text{Pr} = 0.73104$

Kinematic viscosity, $\nu = 1.5114\times 10^{-5}\,\text{m}^2/\text{s}$

Characteristic length, $L_C = W = 0.4\,\text{m}$

Reynolds number

$$R_e = V\frac{L}{\nu}$$
$$= \left(50\times\frac{1000}{3600}\right)\frac{0.4}{1.5114\times 10^{-5}}$$
$$= 367576.8$$

The calculated value of Reynolds number is less than the critical Reynolds number so the flow is laminar.

Step 10 of 13

Nusselt number,

$$Nu = 0.664\times \text{R}_e^{0.5}\times p_r^{1/3}$$
$$= 0.664\times(367576.8)^{0.5}(0.73104)^{1/3}$$
$$= 362.65$$

Heat transfer coefficient,

$$h = \frac{k}{L_C}\times Nu$$
$$= \frac{0.025102}{0.4}\times 362.65$$
$$= 22.758\ \text{W/m}^2{}^\circ\text{C}$$

Step 11 of 13

Rate of heat transfer,

$$\dot{Q} = \frac{T_\infty - T_{in}}{R_{chestwall} + R_{conv}}$$
$$= \frac{T_\infty - T_{in}}{\left(\frac{t}{kA_s}\right)+\left(\frac{1}{hA_s}\right)}$$
$$= \frac{(20-0)}{\left(\frac{0.02}{0.033\times 0.64}\right)+\left(\frac{1}{22.76\times 0.64}\right)}$$
$$= 19.69\ \text{W}$$

Step 12 of 13

$= 19.69\ \text{W}$

Step 12 of 13

The relation for heat transfer due to convection is

$$\dot{Q} = hA_s(T_\infty - T_s)$$

$$19.69 = 22.758 \times 0.64(20 - T_s)$$

$$T_s = 20 - \frac{19.69}{22.758 \times 0.64}$$

$$= 18.65^\circ\text{C}$$

As the obtained value of surface temperature is very close to assumed value, our assumptions are valid and heat transfer calculated can be used for obtaining rate of melting. So the Rate of ice melting is

$$\dot{m} = \frac{\dot{Q}}{h_f}$$

$$= \frac{19.69 \times 10^{-3}}{337.7}$$

$$= 5.8306 \times 10^{-5}\ \text{kg/s}$$

Step 13 of 13

Time taken for melting the ice,

$$t = \frac{m}{\dot{m}}$$

$$= \frac{35}{5.8306 \times 10^{-5}}$$

$$= 600279.3\ \text{sec}$$

$$= 166.74\ \text{h}$$

Hence, the time taken for melting the ice is $\boxed{t = 166.74\ \text{h}}$

Question: 20.090P pg 1

Solar radiation is incident on the glass cover of a solar collector at a rate of 650 W/m2. The glass transmits 88 percent of the incident radiation and has an emissivity of 0.90. The hot water needs of a family in summer can be met completely by a collector 1.5 m high and 2 m wide, and tilted 40° from the horizontal. The temperature of the glass cover is measured to be 40°C on a calm day when the surrounding air temperature is 20°C. The effective sky temperature for radiation exchange between the glass cover and the open sky is −40°C. Water enters the tubes attached to the absorber plate at a rate of 1 kg/min. Assuming the back surface of the absorber plate to be heavily insulated and the only heat loss occurs through the glass cover, determine (*a*) the total rate of heat loss from the collector; (*b*) the collector efficiency, which is the ratio of the amount of heat transferred to the water to the solar energy incident on the collector; and (*c*) the temperature rise of water as it flows through the collector.

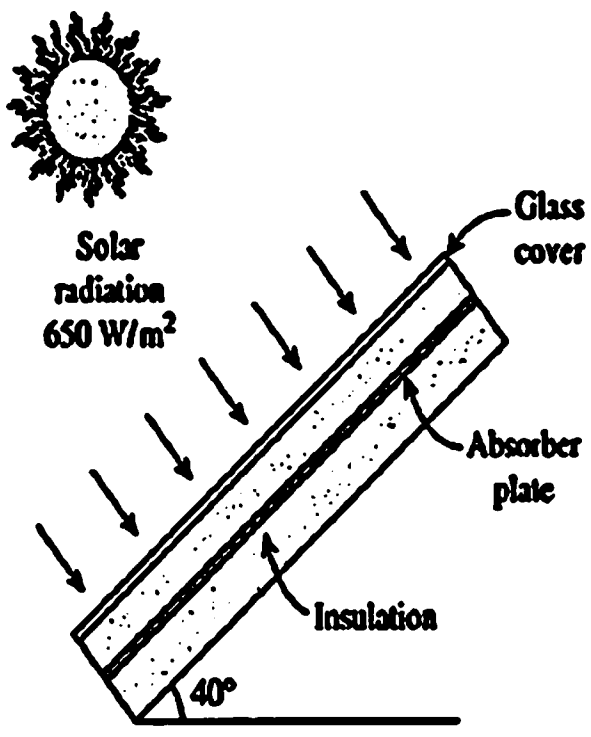


FIGURE P20–90

Step 1 of 8

Solar radiation incident rate, $\dot{q} = 650\ \text{W/m}^2$

Emissivity of the glass, $\varepsilon = 0.9$

Transmissivity of the glass, $\tau = 0.88$

Height of the collector, $H = 1.5\text{m}$

Width of the collector, $W = 2\text{m}$

Angle of tilt, $\theta = 40°$

Temperature of glass cover, $T_s = 40°\text{C}$

Surrounding air temperature, $T_\infty = 20°\text{C}$

Open sky temperature $T_{sky} = -40°\text{C}$

Water flow rate, $\dot{m} = 1\ \text{kg/min}$

Step 2 of 8

Film temperature,

$$T_f = \frac{T_s + T_\infty}{2}$$
$$= \frac{40+20}{2}$$
$$= 30°\text{C}$$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$
$$= \frac{1}{273+30}$$
$$= 3.3\times10^{-3}\ \text{K}^{-1}$$

Step 3 of 8

Take air properties at 1atm pressure and film temperature T_f

Thermal conductivity, $k = 0.02588\ \text{W/m°C}$

Prandtl number, $\text{Pr} = 0.7282$

Kinematic viscosity, $\nu = 1.608\times10^{-5}\ \text{m}^2/\text{s}$

Characteristic length,

$$L_C = \frac{A_s}{p}$$
$$= \frac{(1.5\times2)}{2(1.5+2)}$$
$$= 0.429\text{m}$$

Surface Area,

$$A_s = 1.5\times2$$
$$= 3\text{m}^2$$

Step 4 of 8

Rayleigh number,

$$\text{Ra} = \frac{g\beta\cos\theta(T_s - T_\infty)L_C^3}{\nu^2}\cdot\text{Pr}$$
$$= \frac{9.81\times3.3\times10^{-3}\times\cos40°(40-20)(0.429)^3}{\left(1.608\times10^{-5}\right)^2}$$
$$= 1.103\times10^8$$

Nusselt number,

$$Nu = 0.15\text{Ra}^{1/3}$$
$$= 0.15\left(1.103\times10^8\right)^{1/3}$$
$$= 71.93$$

Step 5 of 8

Heat transfer coefficient,

$$h = \frac{k}{L_C}\cdot Nu$$
$$= \frac{0.02588}{0.429}\times71.93$$
$$= 4.34\ \text{W/m}^2\text{°C}$$

$$h = \frac{k}{L_C} \cdot Nu$$
$$= \frac{0.02588}{0.429} \times 71.93$$
$$= 4.34\ \text{W/m}^2\,^\circ\text{C}$$

Heat transfer rate by convection,

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty)$$
$$= 4.34 \times 3 \times (40 - 20)$$
$$= 260.4\ \text{W}$$

Step 6 of 8

Heat transfer rate by radiation,

$$\dot{Q}_{rad} = \sigma \varepsilon A_s \left(T_s^4 - T_{amb}^4\right)$$
$$= 0.9 \times 5.67 \times 10^{-8} \times 3\left((273+40)^4 - (273-40)^4\right)$$
$$= 1018.14\ \text{W}$$

Total heat transfer rate,

$$\dot{Q}_{tot} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= 260.4 + 1018.14$$
$$= 1278.5\ \text{W}$$

Hence the total heat transfer rate is $\boxed{\dot{Q} = 1278.5\ \text{W}}$

Step 7 of 8

(b)

Heat transmitted by the glass,

$$Q_{in} = \tau \dot{q} A_s$$
$$= 0.88 \times 650 \times 3$$
$$= 1716\ \text{W}$$

Heat transferred to water,

$$Q_w = Q_{in} - Q$$
$$= 1716 - 1278.5$$
$$= 437.5\ \text{W}$$

Step 8 of 8

Collector efficiency,

$$\eta_C = \frac{Q_w}{Q_{in}} \times 100$$
$$= \frac{437.5}{1716} \times 100$$
$$= 25.5\%$$

Therefore the collector efficiency is $\boxed{\eta_C = 25.5\%}$

Heat required to raise temperature is given by

$$Q_w = mc_p(\Delta T)$$
$$437.5 = \frac{1}{60} \times 4180(\Delta T)$$
$$\Delta T = \frac{437.5 \times 60}{4180}$$
$$= 6.28^0\text{C}$$

Hence the temperature difference is $\boxed{\Delta T = 6.28^\circ\text{C}}$

Question: 20.091P

Two concentric spheres with radii of 5 cm and 10 cm are having the surface temperatures maintained at 200°C and 100°C, respectively. The enclosure between the two concentric spherical surfaces is filled with nitrogen gas at 1 atm. If both concentric sphere surfaces tire black (ε= 1), determine the rate of heat transfer on the inner surface.

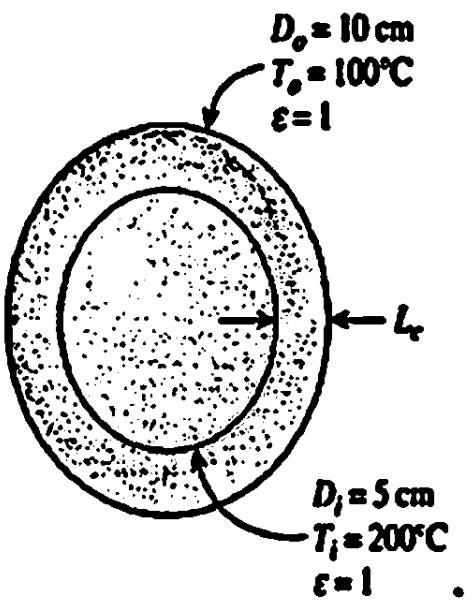


FIGURE P20–91

Step 1 of 6

Diameter of the outer cylinder, $D_0 = 10\text{cm} = 0.1\text{m}$

Temperature of the outer cylinder, $T_0 = 100°\text{C}$

Diameter of the inner cylinder, $D_i = 5\text{cm} = 0.05\text{m}$

Temperature of the inner cylinder, $T = 200°\text{C}$

Pressure, $P = 1\text{atm}$

Step 2 of 6

Mean film temperature of the concentric cylinder is

$$T_{avg} = \frac{T_i + T_0}{2}$$
$$= \frac{100+200}{2}$$
$$= 150°\text{C}$$

From the table of "properties of air at 1atm pressure "

At $T_{avg} = 150°\text{C}$

Thermal conductivity

$$k = 0.03374 + \frac{(150-140)(0.03511-0.03374)}{160-140}$$
$$= 0.034425$$

Prandtl number is

$$\text{Pr} = 0.7041 + \frac{(150-140)(0.7014-0.7041)}{(160-140)}$$
$$= 0.70275$$

Kinematic viscosity

$$\nu = 2.745\times10^{-5} + \frac{(150-140)\left((2.975-2.745)10^{-5}\right)}{160-140}$$
$$= 2.86\times10^{-5}\ \text{m}^2/\text{s}$$

And

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{(150+273\text{K})}$$
$$= \frac{1}{423\text{K}}$$

Step 3 of 6

Characteristic length of the spherical enclosure is

$$L_C = \frac{D_0 - D_i}{2}$$
$$= \frac{0.1-0.05}{2}$$
$$= 0.025\,\text{m}$$

The Rayleigh number is

$$\text{Ra}_L = \frac{g\beta(T_i - T_0)L_c^3}{\nu^2}\text{Pr}$$
$$= \frac{9.81\times\frac{1}{423}(200-100)(0.025)^3(0.70275)}{\left(2.86\times10^{-5}\right)^2}$$
$$= 31.132\times10^3$$

Step 4 of 6

$$F_{sph} = \frac{L_c}{(D_iD_o)^4\left(D_i^{-7/5} + D_0^{-7/5}\right)^5}$$
$$= \frac{0.025}{(0.05\times0.1)^4\left(0.05^{-7/5} + 0.1^{-7/5}\right)^5}$$
$$= \frac{0.025}{6.25\times10^{-10}(6381455116)}$$
$$= 0.006268$$

Step 5 of 6

Then, effective thermal conductivity is

$$k_{eff} = (0.74)k\left(\frac{\text{Pr}}{0.861+\text{Pr}}\right)^{1/4}\left(F_{sph}\text{Ra}_L\right)^{1/4}$$
$$= 0.74\times0.034425\left(\frac{0.70275}{0.861+0.70275}\right)^{1/4}\left(31.132\times10^3\times0.006268\right)^{1/4}$$
$$= 0.02085\times3.737$$
$$= 0.0779\,\text{W/m}\cdot\text{K}$$

Step 6 of 6

The rate of heat transfer becomes

$$\dot{Q} = k_{eff}\frac{\pi D_0 D_i}{L_c}(T_i - T_0)$$
$$= 0.0779\frac{\pi\times0.05\times0.1}{0.025}(200-100)$$
$$= 4.8\,\text{kW}$$

Therefore, Heat Transfer Rate is $\boxed{\dot{Q} = 4.8\,\text{kW}}$

Question: 20.092P

Exhaust gases from a manufacturing plant are being discharged through a 10-m tall exhaust stack with outer diameter of i m. The exhaust gases are discharged at a rate of 0.125 kg/s, while temperature drop between inlet and exit of the exhaust stack is 30°C, and the constant pressure-specific heat of the exhaust gases is 1600 J/kg·K. On a particular calm day, the surrounding quiescent air temperature is 33°C. Solar radiation

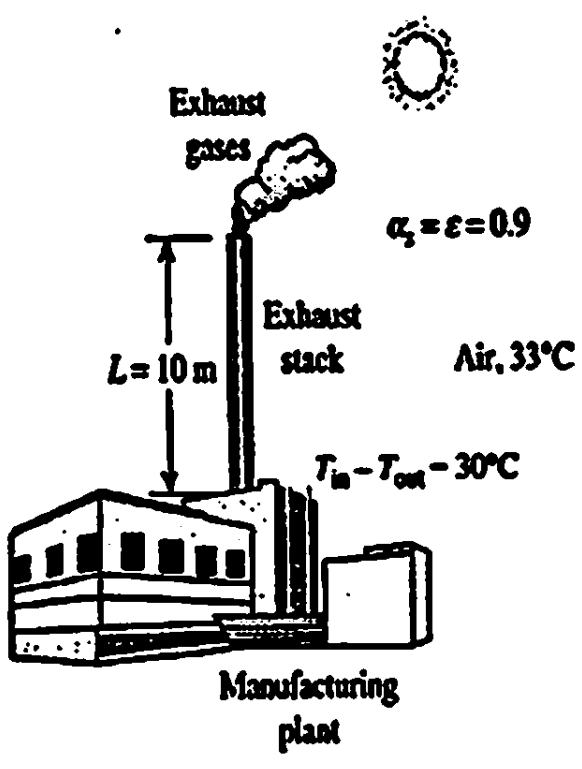


FIGURE P20–92

is incident on the exhaust stack outer surface at a rate of 500W/m2, and both the emissivity and solar absorptivity 0f the outer surface are 0.9. Determine the exhaust stack outer surface temperature. Assume the film temperature is 60°C.

Step 1 of 8

Given data:

Height of exhaust stack, $L = 10\,\text{m}$

Diameter of stack, $D = 1\,\text{m}$

Discharge rate of gases, $m = 0.125\,\text{kg/s}$

Temperature drop, $(T_{in} - T_{out}) = 30°C$

Specific heat of gas, $c_p = 1600\,\text{J/kg}\cdot\text{K}$

Ambient air temperature, $T_\infty = 33°C$

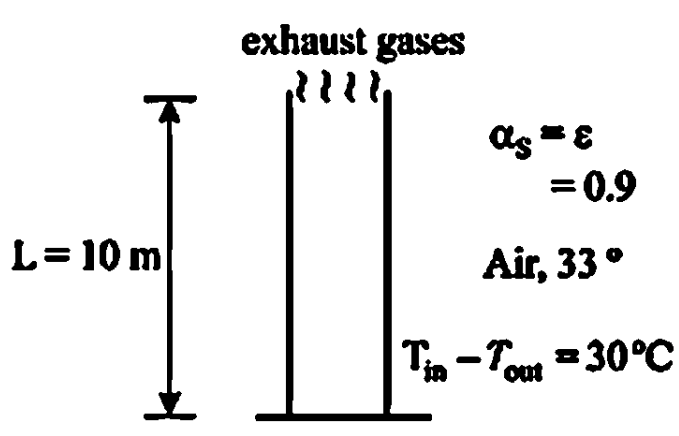
Solar radiation incidence $\dot{q} = 500\,\text{W/m}^2$

Emissivity of stack surface, $\varepsilon = 0.9$

Film temperature, $T_f = 60°C$

Step 2 of 8

Figure showing the exhaust stack:



Step 3 of 8

Film temperature, $T_f = 60°C$

Volume expansion coefficient,

$$\beta = \frac{1}{T_f}$$

$$= \frac{1}{273+60}$$

$$= 3.0\times10^{-3}\ \text{K}^{-1}$$

Taking properties from the table of air at film temperature $T_f = 60°C$ and 1 atm pressure

Thermal conductivity, $k = 0.02808\,\text{W/m}\,°C$

Prandtl number, $\text{Pr} = 0.7202$

Kinematic viscosity, $\nu = 1.896\times10^{-5}$

Characteristic length, $L_C = L = 10\,\text{m}$

Step 4 of 8

Calculating Rayleigh number,

$$Ra = \frac{g\beta(T_s - T_\infty)L_C^3}{\nu^2}\cdot\text{Pr}$$

$$= \frac{9.81\times\left(3.0\times10^{-3}\right)(T_s - 306)(10)^3}{\left(1.896\times10^{-5}\right)^2}(0.7202)$$

$$= 5.896\times10^{10}(T_s - 306)$$

Therefore as per calculation Ra value seems to be in the range $10^{10} - 10^{13}$

Therefore Nusselt number,

$$Nu = 0.1\text{Ra}^{1/3}$$

$$= 0.1\left(5.86\times10^{10}(T_s - 306)\right)^{1/3}$$

$$= 389.2(T_s - 306)^{1/3}$$

Step 5 of 8

Convective heat transfer coefficient,

$$h = \frac{k}{L_C}\cdot Nu$$

$$= \frac{0.02808}{10}\times389.2(T_s - 306)^{1/3}$$

$$= 1.093(T_s - 306)^{1/3}$$

Surface area of the stack,

$$A_s = \pi DL$$

$$= \pi\times1\times10$$

$$= 31.42\,\text{m}^2$$

Step 6 of 8

Convective heat transfer rate,

$$Q_{conv} = hA_s(T_s - T_\infty)$$

$$= 1.093(T_s - 306)^{1/3}\times31.42\times(T_s - 306)$$

$$= 34.342(T_s - 306)^{4/3}$$

Rate of heat loss from exhaust gases

$$Q_{exhaust} = m\times c_p\times(T_1 - T_2)$$

$$= 0.125\times1600\times30$$

$$= 6000\,\text{W}$$

Total heat gain from solar radiation,

$$Q_{total} = \alpha\times\dot{q}\times A_s$$

$$= 500\times0.9\times31.42$$

$$= 14139\ \text{W}$$

Step 7 of 8

Radiation heat transfer rate,

$$\dot{Q}_{rad} = \varepsilon\times A_s\times\sigma$$

$$= 0.9\times31.42\times\left(5.67\times10^{-8}\right)\times\left[(T_s + 273)^4 - (33+273)^4\right]$$

$$= 1.6\times10^{-6}\times\left((T_s + 273)^4 - (33+273)^4\right)$$

Step 8 of 8

Now total heat transfer rate is

$$Q_{total} = Q_{exhaust} + Q_{conv} + Q_{rad}$$

$$14139 = [6000] + \left[34.342(T_s - 306)^{4/3}\right] + \left[1.6\times10^{-6}\times\left(T_s^4 - 306^4\right)\right]$$

By solving the above expression,

The temperature is close to,

$$T_s = 360.5\ \text{K}$$

$$= 87.5°C$$

As this is near to assumed value, our assumption is valid and the surface temperature is

$$\boxed{T_s = 87.5°C}$$

CHAPTER 22

SOLUTIONS

Question: 21.001P

By what properties is an electromagnetic wave characterized? How are these properties related to each other?

Step 1 of 1

Electromagnetic waves are characterized by their frequency ν or wavelength λ.
These two properties in a medium are related by

$$\lambda = \frac{c}{\nu}$$

Where c is the speed of propagation of a wave in that medium, $c = \frac{c_0}{n}$

c_0 is the speed of light in a vacuum

n is the index of refraction of that medium.

Question: 21.002P

What is visible light? How does it differ from the other forms of electromagnetic radiation?

Step 1 of 1

Light is the visible portion of the electromagnetic spectrum that lies between 0.4 μm and 0.76 μm. Light is characteristically not different from other electromagnetic radiations, except that it triggers the sensation of seeing in the human eye. Light, or the visible spectrum, consists of narrow bands of color from violet (0.40 to 0.44 μm) to red (0.63 to 0.76 μm).

Question: 21.003P

How does microwave cooking differ from conventional cooking?

Step 1 of 1

Microwave ovens utilize electromagnetic radiation in the microwave region of the spectrum generated by microwave tubes called magnetrons. Microwaves in the range $10^2 - 10^5\ \mu m$ are very suitable for use in cooking since they are reflected by metals, transmitted by glass and plastics, and absorbed by food molecules. Thus in a microwave oven, the electrical energy is directly converted to radiation that is useful for cooking. But in conventional cooking, a considerable amount of supplied heat is lost to surroundings due conduction, convection and radiation. Hence the time for cooking also increases in conventional cooking.

Question: 21.004P

What is thermal radiation? How does it differ from the other forms of electromagnetic radiation?

Step 1 of 1

Thermal radiation emitted as a result of energy transitions of molecules, atoms, and electrons of a substance. Thermal radiation is also defined as the portion of the electromagnetic spectrum that extends from about 0.1 to 100 μm. Unlike other forms of electromagnetic radiation thermal radiation is continuously emitted by all matter whose temperature is above absolute zero.

Question: 21.005P

What is the cause of color? Why do some objects appear blue to the eye while others appear red? Is the color of a surface at room temperature related to the radiation it emits?

Step 1 of 1

The color of a surface depends on the absorption and reflection characteristics of the surface and is due to selective absorption and reflection of the incident visible radiation coming from a light source such as the sun or incandescent light bulb. For example, a surface that reflects radiation in the wavelength range 0.63 μm - 0.76c μm while absorbing the rest appears red to the eye. A surface that reflects all of the light appears white, while a surface that absorbs the entire light incident on it appears black.

Question: 21.006P

Why is radiation usually treated as a surface phenomenon?

Step 1 of 1

For opaque solids such as metals, wood, and rocks, radiation is considered to be a surface phenomenon, since the radiation emitted by the interior regions can never reach the surface, and the radiation incident on such bodies is usually absorbed within a few microns from the surface.

Question: 21.007P

How do ultraviolet and infrared radiation differ? Do you think your body emits any radiation in the ultraviolet range? Explain.

Step 1 of 1

The radiation emitted by bodies at room temperature falls into the infrared region of the spectrum, which extends from 0.76 to 100 μm.

The ultraviolet radiation includes the low wavelength end of the thermal radiation spectrum and lies between the wavelengths 0.01 and 0.4 μm.

Since the human body at room temperature emits radiation in the infrared region only so that it is not possible to emits radiation in the ultraviolet region.

Question: 21.008P

A cordless telephone is designed to operate at a frequency of 8.5 ×108 Hz. Determine the wavelength of these telephone waves.

Step 1 of 1

Given frequency of the cordless telephone, $\nu = 8.5 \times 10^8$ Hz

Wavelength of the cordless telephone is determined by using $\lambda = \frac{c}{\nu}$

Where λ is the wavelength in m

ν is the frequency of the waves in Hz

c is the speed of light in medium $= 3 \times 10^8$ m/s

Hence $\lambda = \frac{3 \times 10^8 \text{ m/s}}{8.5 \times 10^8 \text{ Hz}}$

$\lambda = 0.353$ m

Therefore wavelength is $\boxed{\lambda = 0.353 \text{ m}}$

Question: 21.009P

Electricity is generated and transmitted in power lines at a frequency of 50 Hz (1 Hz = 1 cycle per second). Determine the wavelength of the electromagnetic waves generated by the passage of electricity in power lines.

Step 1 of 1

Given data:

Frequency of power lines $\nu = 50$ Hz.

Question: 21.010P

A microwave oven is designed to operate at a frequency of 2.2 ×109 Hz. Determine the wavelength of these microwaves and the energy of each microwave.

Step 1 of 2

Given frequency of microwaves is $2.2\times10^{9}\ \text{Hz}$

The wavelength of these microwaves is determined by using,

$$\lambda = \frac{c}{\nu}$$

$$\lambda = \frac{2.9979\times10^{8}\ \text{m/s}}{2.2\times10^{9}\ \text{Hz}}$$

$$\lambda = 0.136\ \text{m}$$

Hence wavelength of these microwaves is $\boxed{\lambda = 0.136\ \text{m}}$

Step 2 of 2

The energy of each microwave is determined by using $e = h\nu$

Where e is the energy in Joules (J)

h is the Planck's constant $(6.626\times10^{-34}\ \text{J.s})$

ν is the frequency of the wave $(2.2\times10^{9}\ \text{Hz})$

Hence $e = h\nu$

$$e = (6.626\times10^{-34}\ \text{J.s})(2.2\times10^{9}\ \text{Hz})$$

$$e = 1.457\times10^{-24}\ \text{J}$$

Hence energy of each microwave is $\boxed{e = 1.457\times10^{-24}\ \text{J}}$

Question: 21.011P

A radio station is broadcasting radio waves at a wavelength of 150 m. Determine the frequency of these waves.

Step 1 of 1

Given data:

The wavelength of the radio waves is $\lambda = 150$ m.

Question: 21.012P

The speed of light in vacuum is given to be 3.0 ×108 m/s. Determine the speed of light in air (n = 1), in water (n = 1.33), and in glass (n = 1.5).

Step 1 of 2

Given

Speed of light in the vacuum is $c_0 = 3\times10^8$ m/s

Speed of light in medium is $c = \frac{c_0}{n}$

For air, $n = 1$

Hence, $c = \frac{c_0}{n}$

$$c = \frac{3\times10^8 \text{ m/s}}{1}$$

$$c = 3\times10^8 \text{ m/s}$$

Speed of light in air is $\boxed{c = 3\times10^8 \text{ m/s}}$

Step 2 of 2

For water, $n = 1.5$

Hence, $c = \frac{c_0}{n}$

$$c = \frac{3\times10^8 \text{ m/s}}{1.5}$$

$$c = 2\times10^8 \text{ m/s}$$

Speed of light in water is $\boxed{c = 2\times10^8 \text{ m/s}}$

Question: 21.013P

What is a blackbody? Does a blackbody actually exist?

Step 1 of 1

A black body is a perfect emitter and absorber of radiation. Black body is an ideal body and hence it does not actually exist.

Question: 21.014P

Why did we define the blackbody radiation function? What does it represent? For what is it used?

Step 1 of 1

The radiation energy emitted by a blackbody per unit area over a wavelength band from $\lambda = 0$ to λ is determined from

$$E_b = \sum_0^\infty E_{b\lambda}(\lambda, T)\, d\lambda \qquad (W/m^2)$$

But this summation does not have a simple closed form solution, and performing a numerical integration each time we need a value of $E_{b0-\lambda}$ is not practically feasible.

Therefore, a dimensionless quantity f_λ called the blackbody radiation function is introduced to represent the fraction of radiation emitted from a blackbody at temperature T in the wavelength band from $\lambda = 0$ to λ.

This function is used to determine the fraction of radiation in a wavelength range between λ_1 and λ_2.

Question: 21.015P

Define the total and spectral blackbody emissive powers. How are they related to each other? How do they differ?

Step 1 of 1

Total body emissive power (E_b) is the sum of the radiation emitted over entire range of wavelengths. Whereas Spectral body emissive power is the amount of radiation energy emitted by a blackbody at a thermodynamic temperature (T) per unit time, per unit surface area, and per unit wavelength about the specific wavelength λ.

The spectral body emissive power ($E_{b\lambda}$) over the entire wavelength spectrum gives the total blackbody emissive power (E_b).

$$E_b = \sum_0^\infty E_{b\lambda}(\lambda, T)\, d\lambda \qquad (W/m^2)$$

Total emissive power of the blackbody doesn't vary with wavelength. Spectral blackbody emissive power varies with wavelength.

Question: 21.016P

The temperature of the filament of an incandescent lightbulb is 2500 K. Assuming the filament to be a blackbody, determine the fraction of the radiant energy emitted by the filament that falls in the visible range. Also, determine the wavelength at which the emission of radiation from the filament peaks.

Step 1 of 4

Given data:

The temperature of the incandescent bulb $T = 2500\ \text{K}$.

Step 2 of 4

The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.4\ \mu\text{m}$ to $\lambda_2 = 0.76\ \mu\text{m}$.

Hence $\lambda_1 = 0.4\ \mu\text{m}$

$\lambda_2 = 0.76\ \mu\text{m}$

$$\lambda_1 T = (0.4\ \mu\text{m})(2500\ \text{K})$$
$$= 1000\ \mu\text{m}\cdot\text{K}$$

From blackbody radiation functions table at $\lambda_1 T = 1000\ \mu\text{m}\cdot\text{K}$, the radiation function is $f_{\lambda_1} = 0.000321$

$$\lambda_2 T = (0.76\ \mu\text{m})(2500\ \text{K})$$
$$= 1900\ \mu\text{mK}$$

From blackbody radiation functions table at $\lambda_2 T = 1900\ \mu\text{m}\cdot\text{K}$, the radiation function is $f_{\lambda_2} = 0.053035$

Step 3 of 4

Fraction of radiant energy emitted in the given range of wavelengths is given by,

$$f_{\lambda_2} - f_{\lambda_1} = 0.053035 - 0.000321$$
$$f_{\lambda_2} - f_{\lambda_1} = 0.05272$$
$$\boxed{f_{\lambda_2} - f_{\lambda_1} = 0.05272}$$

About 5% of the radiation emitted by the filament of the lightbulb falls in the visible range.

Step 4 of 4

The wavelength at which the emission of radiation from the filament peaks is easily determined from Wien's displacement law, which is given below.

$$(\lambda_1 T)_{\text{max_power}} = 2897.8\ \mu\text{m}\cdot\text{K}$$
$$(\lambda_1)_{\text{max_power}} = \frac{2897.8\ \mu\text{m}\cdot\text{K}}{2500\ \text{K}}$$
$$(\lambda_1)_{\text{max_power}} = 1.16\ \mu\text{m}$$

Therefore, the displacement of the wave form is determined to be $\boxed{(\lambda_1)_{\text{max_power}} = 1.16\ \mu\text{m}}$.

Question: 21.017P

Consider a black surface at 2060°F. Determine the radiation energy emitted by the black surface per unit area (in Btu/h•ft2) for ($\lambda \geq 4\ \mu m$.)

Step 1 of 1

Substitute $0.1714\times10^{-8}\,BTu/h\cdot ft^2\cdot R^4$ for σ and $(2060+460)$ for T.

$$E_b(T)=\sigma T^4$$

$$E_b(T)=0.1714\times10^{-8}\,BTu/h\cdot ft^2\cdot R^4(2060+460)^4$$

$$E_b=6.912\times10^4\ BTu/h\cdot ft^2$$

Therefore, radiation of energy emitted from the black surface is $\boxed{6.912\times10^4\ BTu/h\cdot ft^2}$.

Question: 21.018P

Consider a surface at a uniform temperature of 1000 K. Determine the maximum rate of thermal radiation that can be emitted by this surface, in W/m2.

Step 1 of 2

Given

The temperature of the surface $T = 1000\ \text{K}$.

Step 2 of 2

Maximum rate of thermal radiation emitted by the surface at temperature 1000 K can be determined by

$$E_b(T) = \sigma T^4 \ (\text{W/m}^2)$$

Where

Stefan-boltzmann constant $\sigma = 5.67 \times 10^{-8}\ \text{W/m}^2.\text{K}^4$

$$E_b(T) = (5.67 \times 10^{-8}\ \text{W/m}^2.\text{K}^4)(1000\ \text{K})^4$$
$$= 56700\ \text{W/m}^2$$

Therefore, maximum rate of thermal radiation emitted by the surface is $\boxed{E_b(T) = 56,700\ \text{W/m}^2}$.

Question: 21.019P

An incandescent lightbulb is desired to emit at least 15 percent of its energy at wavelengths shorter than 0.8 μm. Determine the minimum temperature to which the filament of the lightbulb must be heated.

Step 1 of 2

Given that the fraction of radiation for the light bulb is 15%

Hence $f_\lambda = 0.15$

Wavelength range of light bulb is from $0.0\,\mu m$ to $0.8\,\mu m$

Hence $\lambda_1 = 0.0\,\mu m$

$\lambda_2 = 0.8\,\mu m$

Step 2 of 2

From blackbody radiation functions table at $f_\lambda = 0.15$,

$\lambda T = 2445\,\mu m\text{K}$

For the wavelength range $\lambda_1 = 0.0\,\mu m$ to $\lambda_2 = 0.8\,\mu m$,

The minimum temperature to which the filament of the light bulb must be heated is

$$\lambda T = 2445\,\mu m\text{K}$$

$$T = \frac{2445\,\mu m\text{K}}{0.8\,\mu m}$$

$$T = 3056.25\,\text{K}$$

The minimum temperature is $\boxed{T = 3056.25\,\text{K}}$

Question: 21.020P

The sun can be treated as a blackbody at an effective surface temperature of 10,400 R. Determine the rate at which infrared radiation energy (λ = 0.76-100 μm) is emitted by the sun, in Btu/h·ft2.

Step 1 of 3

Given

Effective temperature Sun surface is $(T) = 10400\ \text{R}$
$= 5778\ \text{K}$

Let wavelength range be from λ_1 to λ_2

$\lambda_1 = 0.76\ \mu m$

$\lambda_2 = 100\ \mu m$

Step 2 of 3

Considering the product of the minimum wavelength and temperature,

$\lambda_1 T = (0.76\ \mu\text{m})(5778\ \text{K})$

$\lambda_1 T = 4391.3\ \mu\text{mK}$

From blackbody radiation functions table at $\lambda T = 4391.3$,

$f_{\lambda_1} = 0.547370$

Considering the product of the maximum wavelength and temperature,

$\lambda_2 T = (100\ \mu\text{m})(5778\ \text{K})$

$\lambda_2 T = 577{,}800\ \mu\text{mK}$

From blackbody radiation functions table at $\lambda T = 577800$,

$f_{\lambda_2} = 1.0$

Step 3 of 3

Fraction of radiation emitted between these two wavelengths becomes

$f_{\lambda_2} - f_{\lambda_1} = 1.0 - 0.547$

$= 0.453$

From the Stefan-Boltzmann law the total blackbody emissive power of the sun is

$E_b = \sigma T^4$

$= (0.1714 \times 10^{-8}\ \text{Btu/h.ft}^2\text{.R}^4)(10{,}400\ \text{R})^4$

$= 2.005 \times 10^7\ \text{Btu/h.ft}^2$

Then the infrared radiation energy emitted by the sun is

$E_{\text{infrared}} = (0.451) E_b$

$= (0.453)(2.005 \times 10^7\ \text{Btu/h.ft}^2)$

$= 9.08 \times 10^6\ \text{Btu/h.ft}^2$

$\boxed{E_{\text{infrared}} = 9.08 \times 10^6\ \text{Btu/h.ft}^2}$

Question: 21.021P

The sun can be treated as a blackbody at 5780 K. Using EES (or other) software, calculate and plot the spectral blackbody emissive power $E_{b\lambda}$ of the sun versus wavelength in the range of 0.01 μm to 1000 μm. Discuss the results.

Step 1 of 3

The given problem can be solved using EES software by following the below steps.

1. **Open EES software**
2. **Create a new file by clicking on the 'New' of the 'File menu'**

```
Equations Window
  "GIVEN"
  T=5780 "[K]"
  "lambda=0.01[micrometer],
..parameter to be varied"

  "ANALYSIS"
  E_b_lambda=C_1/(lambda^5*(exp
..(C_2/(lambda*T))-1))
  C_1=3.742E8 "[W-micrometer^4/m^
..2]"
  C_2=1.439E4 "[micrometer-K]"
```

Step 2 of 3

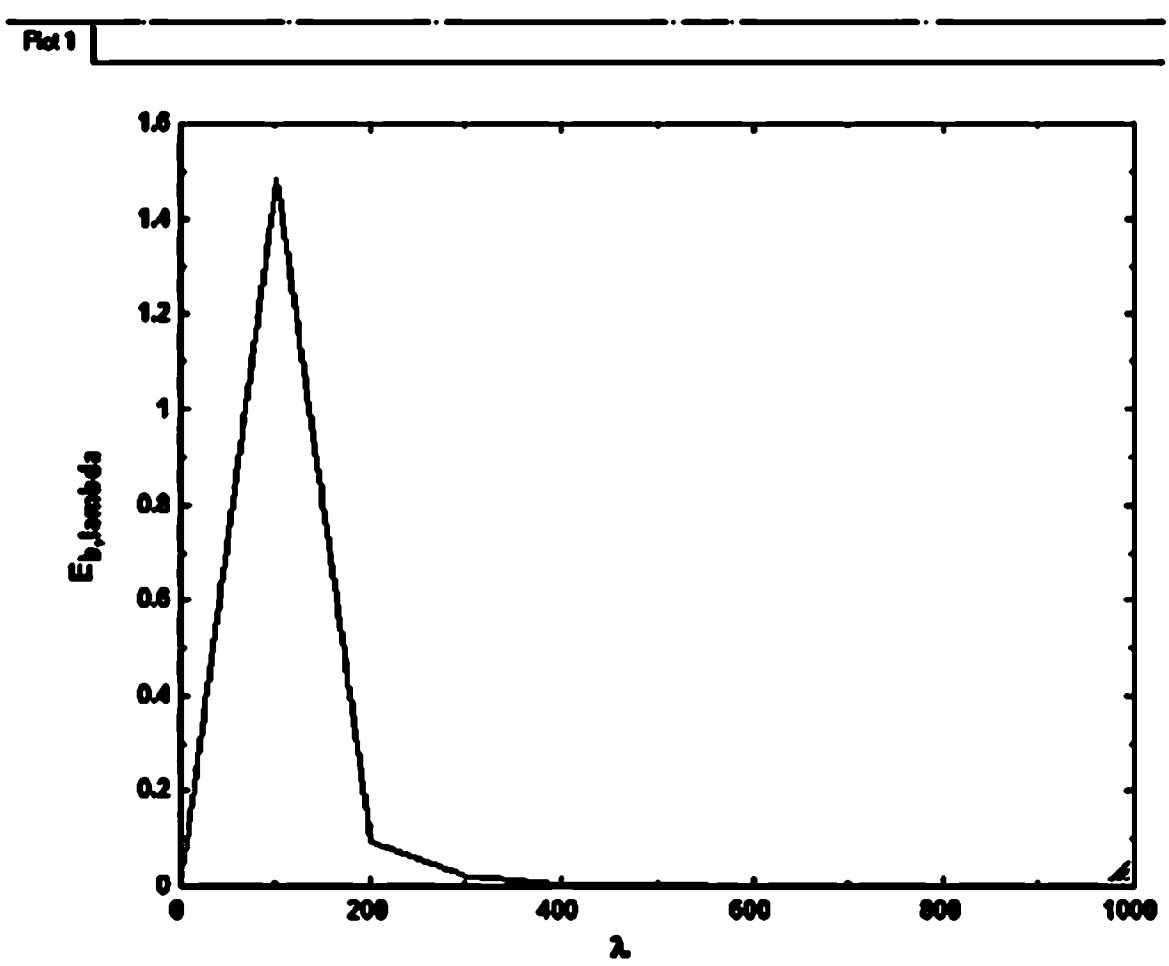
From the above code generate the parametric table.

Table 1

1..11	1 $E_{b,\lambda}$	2 λ
Run 1	2.820E-90	0.01
Run 2	1.484	100
Run 3	0.09334	200
Run 4	0.01848	300
Run 5	0.005853	400
Run 6	0.002399	500
Run 7	0.001157	600
Run 8	0.0006249	700
Run 9	0.0003664	800
Run 10	0.0002288	900
Run 11	0.0001501	1000

Step 3 of 3

Plot the graph



Question: 21.022P

The temperature of the filament of an incandescent lightbulb is 2800 K. Treating the filament as a blackbody, determine the fraction of the radiant energy emitted by the filament that falls in the visible range. Also, determine the wavelength at which the emission of radiation from the filament peaks.

Step 1 of 3

Given data:

The temperature of the incandescent bulb $T = 2800$ K.

Step 2 of 3

We know that,

The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.4$ μm to $\lambda_2 = 0.76$ μm

Hence $\lambda_1 = 0.4$ μm

$\lambda_2 = 0.76$ μm

$\lambda_1 T = (0.4\ \mu m)(2800\ K)$

$= 1120\ \mu mK$

From blackbody radiation functions table at $\lambda_1 T = 1120\ \mu mK$

Radiation function is $f_{\lambda_1} = 0.0014088$

$\lambda_2 T = (0.76\ \mu m)(2800\ K)$

$= 2128\ \mu mK$

From blackbody radiation functions table at $\lambda_2 T = 2128\ \mu mK$

Radiation function is $f_{\lambda_2} = 0.0885904$

Fraction of radiation emitted between these two wavelengths becomes

$f_{\lambda_2} - f_{\lambda_1} = 0.0885904 - 0.0014088$

$f_{\lambda_2} - f_{\lambda_1} = 0.0872$

$\boxed{f_{\lambda_2} - f_{\lambda_1} = 0.0872}$

About 8% of the radiation emitted by the filament of the lightbulb falls in the visible range.

Step 3 of 3

The wavelength at which the emission of radiation from the filament peaks is easily determined from Wien's displacement law to be

$(\lambda_1 T)_{max_power} = 2897.8\ \mu m.K$

$(\lambda_1)_{max_power} = \dfrac{2897.8\ \mu m.K}{2800}$

$(\lambda_1)_{max_power} = 1.035\ \mu m$

Therefore, the length wave length is $\boxed{(\lambda_1)_{max_power} = 1.035\ \mu m}$.

Question: 21.023P

Reconsider Prob. 21-22. Using EES (or other) software, investigate the effect of temperature on the fraction of radiation emitted in the visible range. Let the surface temperature vary from 1000 K to 4000 K, and plot fraction of radiation emitted in the visible range versus the surface temperature.

Step 1 of 3

The given problem can be solved using EES software by following the below steps.

1. **Open EES software**
2. **Create a new file by clicking on the 'New' of the 'File menu'**

Equations Window

```
"GIVEN"
"T=2800 [K], parameter to be varied"
lambda_1=0.40 "[micrometer]"
lambda_2=0.76 "[micrometer]"

"ANALYSIS"
E_b_lambda=C_1/(lambda^5*(exp(C_2/(lambda*T))-1))
C_1=3.742E8 "[W-micrometer^4/m^2]"
C_2=1.439E4 "[micrometer-K]"

f_lambda=integral(E_b_lambda, lambda, lambda_1, lambda_2)/E_b
E_b=sigma*T^4
sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzmann constant"
```

Step 2 of 3

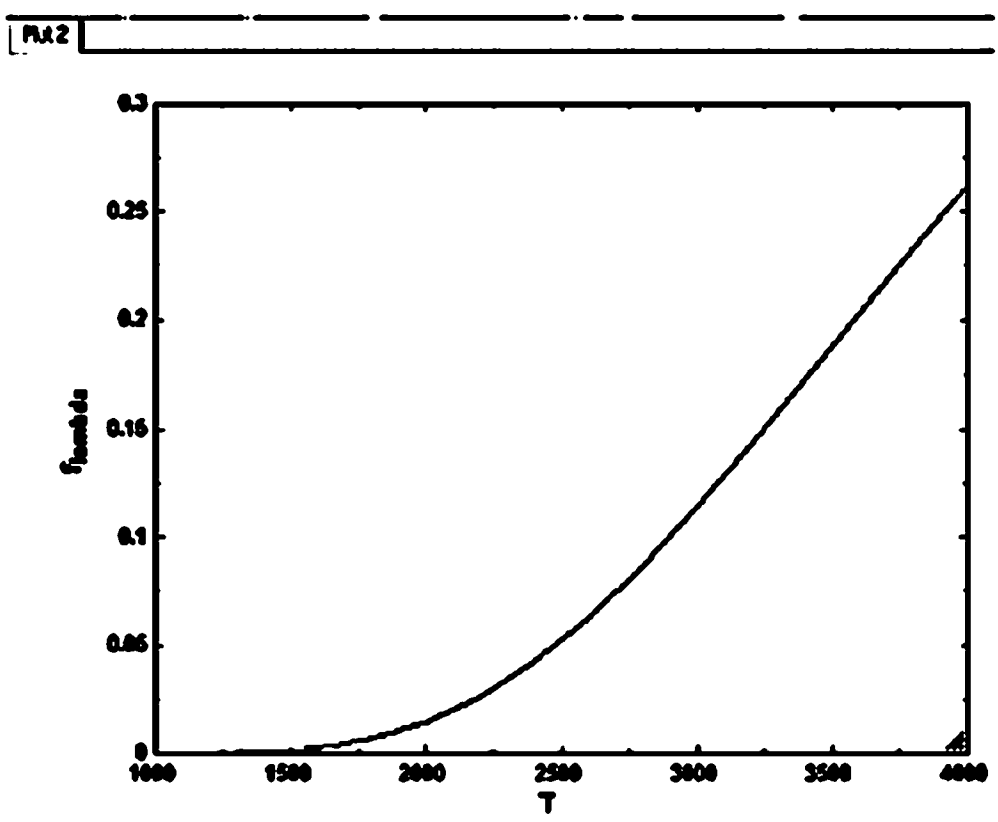
From the above code generate the parametric table.

Table 2

1..16	1 f_λ	2 T
Run 1	0.000007353	1000
Run 2	0.0001032	1200
Run 3	0.0006403	1400
Run 4	0.002405	1600
Run 5	0.006505	1800
Run 6	0.01404	2000
Run 7	0.02576	2200
Run 8	0.04198	2400
Run 9	0.06248	2600
Run 10	0.08671	2800
Run 11	0.1139	3000
Run 12	0.143	3200
Run 13	0.1732	3400
Run 14	0.2036	3600
Run 15	0.2336	3800
Run 16	0.2623	4000

Step 3 of 3

Plot the graph



Question: 21.024P

It is desired that the radiation energy emitted by a light source reach a maximum in the blue range (λ = 0.47 μm).Determine the temperature of this light source and the fraction of radiation it emits in the visible range (λ = 0.40–0.76 μm).

Step 1 of 3

Given that the radiation energy emitted by a light source reach a maximum in the blue range i.e. at $\lambda = 0.47\ \mu m$

The standard value for the maximum, $(\lambda T)_{max\ power} = 2897.8\ \mu m \cdot K$

The temperature of this light source is determined by using the relation,

$$T = \frac{(\lambda T)_{max\ power}}{\lambda}$$

$$T = \frac{2897.8\ \mu m \cdot K}{0.47\ \mu m}$$

$$T = 6167\ K$$

Temperature of this light source is $\boxed{T = 6167\ K}$

Step 2 of 3

Visible range of electromagnetic spectrum extends from $\lambda_1 = 0.4\ \mu m$ to $\lambda_2 = 0.76\ \mu m$

Considering the product of the minimum wavelength and temperature,

$\lambda_1 T = (0.4\ \mu m)(6167\ K)$

$\lambda_1 T = 2466\ \mu mK$

From blackbody radiation functions table at $\lambda T = 2466$,

$f_{\lambda_1} = 0.15444$

Considering the product of the maximum wavelength and temperature,

$\lambda_2 T = (0.76\ \mu m)(6167\ K)$

$\lambda_2 T = 4686\ \mu mK$

From blackbody radiation functions table at $\lambda T = 4686$,

$f_{\lambda_2} = 0.59141$

Step 3 of 3

Fraction of radiation emitted between these two wavelengths becomes

$$f_{\lambda_2} - f_{\lambda_1} = 0.59141 - 0.15444$$

$$f_{\lambda_2} - f_{\lambda_1} = 0.437$$

Fraction of radiation emitted is $\boxed{43.7\ \%}$

Question: 21.025P

A 3-mm-thick glass window transmits 90 percent of the radiation between λ = 0.3 and 3.0 μm and is essentially opaque for radiation at other wavelengths. Determine the rate of radiation transmitted through a 3-m × 3-m glass window from blackbody sources at (*a*) 5800 K and (*b*) 1000 K.

Step 1 of 5

(a)

Temperature of the plate is, $T = 5800\ \text{K}$

Take the product of the minimum wavelength, λ_1 and temperature *T*,

Substitute $0.3\ \mu\text{m}$ for λ_1 and 5800 K for *T*,

$$\begin{aligned}\lambda_1 T &= (0.3\ \mu\text{m})(5800\ \text{K})\\ &= 1740\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 1740$, find the value of fraction of radiation emitted from a black body at temperature *T* by using interpolation method,

$$\frac{1800-1600}{1800-1740} = \frac{0.039341-0.019718}{0.039341-f_{\lambda_1}}$$

$$f_{\lambda_1} = 0.03345$$

Take the product of the maximum wavelength and temperature,

Substitute $3\ \mu\text{m}$ for λ_1 and 5800 K for *T*,

$$\begin{aligned}\lambda_2 T &= (3.0\ \mu\text{m})(5800\ \text{K})\\ &= 17400\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 17400$, find the value of fraction of radiation emitted from a black body at temperature *T* by using interpolation method,

$$\frac{18000-16000}{18000-17400} = \frac{0.980860-0.973814}{0.980860-f_{\lambda_2}}$$ 00.

$$f_{\lambda_2} = 0.97875$$

Step 2 of 5

Find the fraction of radiation emitted between these two wavelengths,

$$\begin{aligned}f_{\lambda_2} - f_{\lambda_1} &= 0.97875-0.03345\\ \Delta f &= 0.9453\end{aligned}$$

Calculating the total blackbody radiation emission,

$$E_b(T) = \sigma T^4 A_s$$

Here, area of the square plate,

$$\begin{aligned}A_s &= 3\text{m}\times 3\text{m}\\ &= 9\text{m}^2\end{aligned}$$

Substitute $5.67\times10^{-8}\ \text{W/m}^2.\text{K}$ for σ

$$\begin{aligned}E_b(T) &= \sigma T^4 A_s\\ &= \left(5.67\times10^{-8}\right)(5800)^4(9)\\ &= 5.774\times10^5\ \text{kW}\end{aligned}$$

Step 3 of 5

Find the rate of radiation transmitted,

$$E_{\text{transmit}} = \varepsilon\times\Delta f\times E_b(T)$$

Here, ε is emissivity

Substitute 0.9 for ε

$$\begin{aligned}E_{\text{transmit}} &= 0.9\times0.9453\times5.774\times10^5\\ &= 491{,}234.598\ \text{kW}\end{aligned}$$

Therefore, the rate of radiation transmitted through a glass window is $\boxed{491{,}234.598\ \text{kW}}$

Step 4 of 5

(b)

Temperature of the plate is, $T = 1000\ \text{K}$

Take the product of the minimum wavelength, λ_1 and temperature *T*,

Substitute $0.3\ \mu\text{m}$ for λ_1 and 1000 K for *T*,

$$\begin{aligned}\lambda_1 T &= (0.3\ \mu\text{m})(1000\ \text{K})\\ &= 300\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 300$

$$f_{\lambda_1} = 0.0$$

Take the product of the maximum wavelength and temperature,

Substitute $3\ \mu\text{m}$ for λ_1 and 5800 K for *T*,

$$\begin{aligned}\lambda_2 T &= (3.0\ \mu\text{m})(1000\ \text{K})\\ &= 3000\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 3000$

$$f_{\lambda_2} = 0.273232$$

Step 5 of 5

Fraction of radiation emitted between these two wavelengths becomes

$$\begin{aligned}f_{\lambda_2} - f_{\lambda_1} &= 0.273232-0.0\\ \Delta f &= 0.273232\end{aligned}$$

Calculating the total blackbody radiation emission,

$$\begin{aligned}E_b(T) &= \sigma T^4 A_s\\ &= \left(5.67\times10^{-8}\right)(1000)^4(9)\\ &= 510.3\ \text{kW}\end{aligned}$$

Find the rate of radiation transmitted,

$$\begin{aligned}E_{\text{transmit}} &= \varepsilon\times\Delta f\times E_b(T)\\ &= 0.9\times0.273232\times510.3\\ &= 125.47\ \text{kW}\end{aligned}$$

Therefore, the rate of radiation transmitted through a glass window is $\boxed{125.47\ \text{kW}}$

Question: 21.026P

Daylight and candlelight may be approximated as a blackbody at the effective surface temperatures of 5800 K and 1800 K, respectively. Determine the radiation energy (in W/m2) emitted by both lighting sources (daylight and candlelight) within the visible light region (0.40 to 0.76 μm).

Step 1 of 6

Given data:

Effective surface temperature of daylight, $T = 5800\ \text{K}$

Effective surface temperature of candle light, $T_l = 1800\ \text{K}$

For daylight:

The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.4\ \mu\text{m}$ to $\lambda_2 = 0.76\ \mu\text{m}$.

For candlelight:

The visible range of the electromagnetic spectrum extends from $\lambda_1 = 0.4\ \mu\text{m}$ to $\lambda_2 = 0.76\ \mu\text{m}$.

Step 2 of 6

Considering the product of the wavelength and temperature,

$$\begin{aligned}\lambda_1 T &= (0.4\ \mu\text{m})(5800)\ \text{K}\\ &= 2320\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 2320\ \mu\text{m}\cdot\text{K}$

By using interpolation method,

Fraction of radiation emitted,

$$\frac{2400-2000}{2400-2320} = \frac{0.140256-0.100888}{0.140256-f_{\lambda_1}}$$

$$f_{\lambda_1} = 0.13382$$

Step 3 of 6

Considering the product of the wavelength and temperature,

$$\begin{aligned}\lambda_2 T &= (0.76\ \mu\text{m})(5800)\ \text{K}\\ &= 4408\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 4408\ \mu\text{m}\cdot\text{K}$

By using interpolation method,

Fraction of radiation emitted,

$$\frac{4600-4400}{4600-4408} = \frac{0.579280-0.516014}{0.579280-f_{\lambda_2}}$$

$$f_{\lambda_2} = 0.5185$$

Fraction of radiation emitted between these two wavelengths becomes

$$f_{\lambda_2} - f_{\lambda_1} = 0.5185 - 0.13382$$

$$\Delta f = 0.3846$$

Step 4 of 6

The total blackbody emissive power of the sun is determined from Stefan-Boltzmann Law to be

$$E_b(T) = \sigma T^4 \quad \ldots\ldots (1)$$

Where,

Stefan Boltzman constant is, $\sigma = 5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4$

Substituting the values $T = 5800\ \text{K}$ and σ in equation (1), we get

$$\begin{aligned}E_b(T) &= \sigma T^4\\ &= 5.67\times10^{-8}\times5800^4\\ &= 6.416\times10^7\ \text{W/m}^2\end{aligned}$$

Radiation energy of daylight,

$$\begin{aligned}E_{\text{Daylight}} &= \Delta f E_b(T)\\ &= (0.3846)\left(6.416\times10^7\right)\ \text{W/m}^2\\ &= 2.467\times10^7\ \text{W/m}^2\end{aligned}$$

Therefore, radiation energy of daylight is $\boxed{E_{\text{Daylight}} = 2.467\times10^7\ \text{W/m}^2}$

Step 5 of 6

Considering the product of the wavelength and temperature,

$$\begin{aligned}\lambda_1 T &= (0.4\ \mu\text{m})(1800)\ \text{K}\\ &= 720\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 720\ \mu\text{m}\cdot\text{K}$

By using interpolation method,

Fraction of radiation emitted,

$$\frac{800-600}{800-720} = \frac{0.000016-0}{0.000016-f_{\lambda_1}}$$

$$f_{\lambda_1} = 0.0000096$$

Step 6 of 6

Considering the product of the wavelength and temperature,

$$\begin{aligned}\lambda_2 T &= (0.76\ \mu\text{m})(1800)\ \text{K}\\ &= 1368\ \mu\text{m}\cdot\text{K}\end{aligned}$$

From blackbody radiation functions table, at $\lambda T = 1368\ \mu\text{m}\cdot\text{K}$

By using interpolation method,

Fraction of radiation emitted,

$$\frac{1400-1200}{1400-1368} = \frac{0.007790-0.002134}{0.007790-f_{\lambda_2}}$$

$$f_{\lambda_2} = 0.006885$$

Fraction of radiation emitted between these two wavelengths becomes

$$f_{\lambda_2} - f_{\lambda_1} = 0.0068854 - 0.0000096$$

$$\Delta f = 0.0068754$$

The total blackbody emissive power of the sun is determined from Stefan-Boltzmann Law to be

$$E_b(T) = \sigma T^4 \quad \ldots\ldots (2)$$

Where,

Stefan Boltzman constant is, $\sigma = 5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4$

Substituting the values $T = T_l = 1800\ \text{K}$ and σ in equation (2), we get

$$\begin{aligned}E_b(T) &= \sigma T^4\\ &= 5.67\times10^{-8}\times1800^4\\ &= 5.5952\times10^5\ \text{W/m}^2\end{aligned}$$

Radiation energy of daylight,

$$\begin{aligned}E_{\text{candlelight}} &= \Delta f E_b(T)\\ &= (0.006885)5.952\times10^5\ \text{W/m}^2\\ &= 4097.9\ \text{W/m}^2\end{aligned}$$

Therefore, radiation energy of daylight is $\boxed{E_{\text{candlelight}} = 4097.9\ \text{W/m}^2}$

Question: 21.027P

A flame from a match may be approximated as a blackbody at the effective surface temperature of 1700 K, while moonlight may be approximated as a blackbody at the effective surface temperature of 4000 K, respectively. Determine the peak spectral blackbody emissive power for both lighting sources (match flame and moonlight).

Step 1 of 4

Effective surface temperature of the match flame, $T_1 = 1700\ \text{K}$

Effective surface temperature of the moonlight, $T_2 = 4000\ \text{K}$

Step 2 of 4

The relation for the spectral blackbody emissive power $E_{b\lambda}$ is determined by the Planck's law which is expressed as

$$E_{b\lambda T}(\lambda, T) = \frac{C_1}{\lambda^5\left[\exp(C_2/\lambda T) - 1\right]} \qquad \ldots\ldots (1)$$

Where the value of the constants are,

$C_1 = 3.742\times10^8\ \text{W}\cdot\mu\text{m}^4/\text{m}^2$

$C_2 = 1.439\times10^4\ \mu\text{m}\cdot\text{K}$

Step 3 of 4

Now the value of the wavelength (λ) for the match flame is determined by using the Wien's displacement law which is expressed as

$(\lambda T)_{\text{max power}} = 2897.8\ \mu\text{m}\cdot\text{K}$

Re-arranging the above relation for wavelength we get

$$\lambda = \frac{2897.8}{T}$$
$$= \frac{2897.8}{1700}$$
$$= 1.704\ \mu\text{m} \qquad \ldots\ldots (2)$$

Similarly the value of the wavelength (λ) for the moon light is determined by using the Wien's displacement law

$$\lambda = \frac{2897.8}{T}$$
$$= \frac{2897.8}{4000}$$
$$= 0.724\ \mu\text{m} \qquad \ldots\ldots (3)$$

Step 4 of 4

Now substituting the value of λ for the match flame from equation (2) and the constants C_1, C_2 in equation (1) we get

$$E_{b\lambda T}(\lambda, T) = \frac{C_1}{\lambda^5\left[\exp(C_2/\lambda T) - 1\right]}$$
$$= \frac{3.742\times10^8}{1.704^5\left[\exp\left(1.439\times10^4/1.704\times1700\right) - 1\right]}$$
$$= \frac{3.742\times10^8}{2049.71}$$
$$= 0.182\times10^6\ \text{W}/\text{m}^2\cdot\mu\text{m}$$

Similarly substituting the value of λ for the moon light from equation (3) and the constants C_1, C_2 in equation (1) we get

$$E_{b\lambda T}(\lambda, T) = \frac{C_1}{\lambda^5\left[\exp(C_2/\lambda T) - 1\right]}$$
$$= \frac{3.742\times10^8}{0.724^5\left[\exp\left(1.439\times10^4/0.724\times4000\right) - 1\right]}$$
$$= \frac{3.742\times10^8}{28.421}$$
$$= 13.16\times10^6\ \text{W}/\text{m}^2\cdot\mu\text{m}$$

Therefore the peak spectral blackbody emissive power for the match flame is $\boxed{0.182\times10^6\ \text{W}/\text{m}^2\cdot\mu\text{m}}$ and for the moon light is $\boxed{13.16\times10^6\ \text{W}/\text{m}^2\cdot\mu\text{m}}$

Question: 21.028P

What is a graybody? How does it differ from a blackbody? What is a diffuse gray surface?

Step 1 of 1

A body whose surface properties i.e. emissivity and abosrptivity are independent of wavelength is said to be a graybody. The emissivity of a blackbody is one for all wavelengths; the emissivity of a graybody is between zero and one.

A surface for which emissivity and absorptivity are directional (θ) independent and wavelength (λ) independent are termed as diffusive gray surface.

Question: 21.029P

Define the properties emissivity and absorptivity. When are these two properties equal to each other?

Step 1 of 1

Emissivity of a surface represents the ratio of the radiation emitted by the surface at a given temperature to the radiation emitted by a blackbody at the same temperature. The fraction of irradiation absorbed by the surface is called the absorptivity. Both emissivity and absorptivity are equal under the condition that the surface temperature is equal to the temperature of the source of irradiation.

Question: 21.030P

What is the greenhouse effect? Why is it a matter of great concern among atmospheric scientists?

Step 1 of 1

Earth receives energy from the sun as radiation and gets heated. At nights, the earth's surface gets cooled by radiating this energy into deep space as Infrared radiation. But due to the presence of materials that are of non-gray characteristics, the Infrared radiation is reflected back to the surface and this eventually cause global warming. And this Heating effect is known as greenhouse effect. There is a great concern for this effect among atmospheric scientists, as this effect would bring drastic changes in weather patterns.

Question: 21.031P

Consider an opaque horizontal plate that is well insulated on the edges and the lower surface. The plate is maintained at 500 K, has a total hemispherical absorptivity of 0.51 and the following spectral emissivity function:

$$\varepsilon_\lambda = \begin{cases} \varepsilon_1 = 0.4, & 0 \le \lambda < 4\ \mu\text{m} \\ \varepsilon_2 = 0.8, & 4\ \mu\text{m} \le \lambda < \infty \end{cases}$$

If the plate is subjected to an irradiation of 5600 W/m2, determine (*a*) the total hemispherical emissivity and (*b*) the radiosity of the plate surface.

Step 1 of 4

Temperature of the plate, $T = 500\ \text{K}$
Absorptivity of the plate, $\alpha = 0.51$
Irradiation on the plate, $G = 5600\ \text{W/m}^2$
The emissivity of the plate for $0 < \lambda < 4\ \mu\text{m}$ is $\varepsilon_1 = 0.4$
The emissivity of the plate for $4\ \mu\text{m} < \lambda < \infty$ is $\varepsilon_2 = 0.8$

Step 2 of 4

(a)
Radiation function corresponding to the temperature of 500K,

$$\lambda T = (4)(500)$$
$$= 2000\ \mu\text{m}\cdot\text{K}$$

From the black body radiation function, at $\lambda T = 2000\ \mu\text{m}\cdot\text{K}$

$$f_\lambda = 0.066728$$

The emissivity of the plate is

$$\varepsilon = \varepsilon_1 f_\lambda + \varepsilon_2 (1 - f_\lambda)$$
$$= (0.4)(0.066728) + (0.8)(1 - 0.066728)$$
$$= 0.7733$$

Hence the emissivity of the plate is $\boxed{\varepsilon = 0.7733}$

Step 3 of 4

(b)
The radiation emitted by the plate is

$$E_{\text{emitted}} = \varepsilon \sigma T^4$$
$$= 0.7733 \times \left(5.67 \times 10^{-8}\right) \times 500^4$$
$$= 2740.38\ \text{W/m}^2$$

The energy reflected by the plate is

$$E_{\text{reflected}} = \rho G$$
$$= (1 - \alpha) G$$
$$= (1 - 0.51)5600$$
$$= 2744\ \text{W/m}^2$$

Step 4 of 4

The radiosity of the plate is

$$J = E_{\text{emitted}} + E_{\text{reflected}}$$
$$= 2740.38 + 2744$$
$$= 5484.38\ \text{W/m}^2$$

Thus the radiosity of the plate is $\boxed{J = 5484.38\ \text{W/m}^2}$

Question: 21.032P

The spectral emissivity function of an opaque surface at 1000 K is approximated as

$$\varepsilon_\lambda = \begin{cases} \varepsilon_1 = 0.4, & 0 \le \lambda < 3\ \mu m \\ \varepsilon_2 = 0.7, & 3\ \mu m \le \lambda < 6\ \mu m \\ \varepsilon_3 = 0.3, & 6\ \mu m \le \lambda < \infty \end{cases}$$

Determine the average emissivity of the surface and the rate of radiation emission from the surface, in W/m2.

Step 1 of 4

Given data:

The temperature of the surface $T = 1000\ \text{K}$

In order to find the average emissivity of the surface, the fraction of radiation is determined.

The fraction of radiation in the range of $3\mu m$ to $6\mu m$ is

$$\lambda_1 T = (3\ \mu\text{m})(1000\ \text{K})$$
$$= 3000\ \mu\text{m} \cdot \text{K}$$

From the blackbody radiation function f_λ

At $\lambda_1 T = 3000\ \mu\text{m} \cdot \text{K}$

$$f_{\lambda_1} = 0.273232$$

Step 2 of 4

$$\lambda_2 T = (6\ \mu\text{m})(1000\ \text{K})$$
$$= 6000\ \mu\text{m} \cdot \text{K}$$

From the blackbody radiation function f_λ

At $\lambda_2 T = 6000\ \mu\text{m} \cdot \text{K}$

$$f_{\lambda_2} = 0.737818$$

Step 3 of 4

The average emissivity of the surface is determined using the formula,

$$\varepsilon(T) = \varepsilon_1 f_{\lambda_1} + \varepsilon_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \varepsilon_3 \left(1 - f_{\lambda_2}\right)$$

Substituting the known values in the above relation, we have,

$$\varepsilon(T) = (0.4)(0.273232) + (0.7)(0.737818 - 0.273232) + 0.3(1 - 0.737818)$$
$$= 0.1092928 + 0.3252102 + 0.0786546$$
$$= 0.5131$$

Therefore, the average rate of emissivity of the given opaque surface is $\boxed{\varepsilon(T) = 0.513}$.

Step 4 of 4

The rate of emission from the surface is expressed as,

$$E = \varepsilon \sigma T^4$$

$$E = (0.513)\left(5.67 \times 10^{-8}\right)(1000)^4$$
$$= 29{,}096\ \text{W/m}^2$$
$$\approx 29.1\ \text{W/m}^2$$

Therefore, rate of radiation emission form the surface is $\boxed{E = 29.1\ \text{kW/m}^2}$.

Question: 21.033P

A 5-in-diameter spherical ball is known to emit radiation at a rate of 480 Btu/h when its surface temperature is 950 R. Determine the average emissivity of the ball at this temperature.

Step 1 of 2

Given

Diameter of the spherical ball $D = 5$ in

$= 5/12$ ft

Emit radiation rate $E = 480$ Btu/h

Surface temperature $T = 950$ R

Step 2 of 2

Surface area of the spherical ball is

$$A = \pi D^2$$

$$= \pi(5/12 \text{ ft})^2$$

$$= 0.5454 \text{ ft}^2$$

The average emissivity of the ball at this temperature is determined by

$$\varepsilon_{avg} = \frac{E}{A\sigma T^4}$$

$$= \frac{480 \text{ Btu/h}}{(0.5454 \text{ ft}^2)(0.1714 \times 10^{-8} \text{ Btu/h.ft}^2 \cdot \text{R}^4)(950 \text{ R})^4}$$

$$= 0.63$$

Therefore, average emissivity of the ball is $\boxed{\varepsilon_{avg} = 0.63}$.

Question: 21.034P

A furnace that has a 40-cm ×40-cm glass window can he considered to be a blackbody at 1200 K. If the transmissivity of the glass is 0.7 for radiation at wavelengths less than 3 μm and zero for radiation at wavelengths greater than 3 μm, determine the fraction and the rate of radiation coming from the furnace and transmitted through the window.

Step 1 of 3

Given

Glass window at temperature $T = 1200\ \text{K}$

Transmissivity of surface at wavelength less than $3\ \mu\text{m}$ is $\tau_1 = 0.7$

Transmissivity of surface at wavelength greater than $3\ \mu\text{m}$ is $\tau_2 = 0$

Surface area of the glass window $A_s = a^2$

$$= (0.4)^2$$

$$= 0.16\ \text{m}^2$$

Step 2 of 3

Considering the product of the wavelength and temperature,

$$\lambda T = (3\ \mu\text{m})(1200\ \text{K})$$

$$\lambda T = 3600\ \mu\text{mK}$$

From blackbody radiation functions table at $\lambda T = 3600$,

$f_{\lambda_1} = 0.403607$

Step 3 of 3

The fraction of radiation coming from the furnace and transmitted through the window is

$$\tau(T) = \tau_1 f_\lambda + \tau_2(1 - f_\lambda)$$

$$\tau(T) = (0.7)(0.403607) + (0)(1 - 0.403607)$$

$$\tau(T) = 0.283$$

The rate of radiation is $G_w = \tau A \sigma T^4$

$$G_w = 0.283(0.4 \times 0.4\ \text{m}^2)(5.67 \times 10^{-8}\ \text{W/m}^2.\text{K}^4)(1200\ \text{K})^4$$

$$G_w = 5323.7\ \text{W}$$

Rate of radiation coming from the furnace and transmitted through the window is

$\boxed{G_w = 5323.7\ \text{W}}$

Question: 21.035P

The emissivity of a tungsten filament can be approximated to be 0.5 for radiation at wavelengths less than 1 μm and 0.15 for radiation at greater than 1 μm. Determine the average emissivity of the filament at (*a*) 1500 K and (*b*) 2500 K. Also, determine the absorptivity and reflectivity of the filament at both temperatures.

Step 1 of 5

Given

The emissivity $\varepsilon_1 = 0.5$ for wavelength less than $1\ \mu m$

The emissivity $\varepsilon_2 = 0.15$ for wavelength greater than $1\ \mu m$

Step 2 of 5

(a) At $T = 1500\ K$

Average emissivity of the surface for the radiation at $T = 1500\ K$ is determined as follows

Fraction of radiation at wavelength less than 5 μm is calculated by

$$\lambda T = (1\ \mu m)(1500\ K)$$
$$= 1500\ \mu mK$$

From Blackbody radiation function f_λ

$$\lambda T = 1500\ \mu mK$$

$$f_\lambda = 0.01375$$

Average emissivity of the surface is determined by

$$\varepsilon(T) = \varepsilon_1 f_\lambda + \varepsilon_2(1 - f_\lambda)$$
$$\varepsilon(T) = (0.5)(0.01375) + (0.15)(1 - 0.01375)$$
$$\boxed{\varepsilon(T) = 0.155}$$

From Kirchhoff's law the absorptivity of the surface becomes $\boxed{\therefore \alpha = \varepsilon = 0.155}$

Step 3 of 5

Then the reflectivity of the surface is determined by

$$\alpha + \rho = 1$$

$$\rho = 1 - \alpha$$

$$= 1 - 0.155$$

$$\boxed{\rho = 0.854}$$

Step 4 of 5

(b) $T = 2500\ K$

Average emissivity of the surface for the radiation at $T = 1500\ K$ is determined as follows

Fraction of radiation at wavelength less than 5 μm is calculated by

$$\lambda T = (1\ \mu m)(2500\ K)$$
$$= 2500\ \mu mK$$

From Blackbody radiation function f_λ

At $\lambda T = 2500\ \mu mK$

$$f_\lambda = 0.1617$$

Average emissivity of the surface is determined by

$$\varepsilon(T) = \varepsilon_1 f_\lambda + \varepsilon_2(1 - f_\lambda)$$
$$\varepsilon(T) = (0.5)(0.1617) + (0.15)(1 - 0.1617)$$
$$\boxed{\varepsilon(T) = 0.2066}$$

From Kirchhoff's law the absorptivity of the surface becomes $\boxed{\alpha = \varepsilon = 0.2066}$

Step 5 of 5

Then the reflectivity of the surface is determined by

$$\alpha + \rho = 1$$

$$\rho = 1 - \alpha$$

$$= 1 - 0.2066$$

$$\boxed{\rho = 0.7934}$$

Question: 21.036P

The variations of the spectral emissivity of two surfaces are as given in Fig. P21-36. Determine the average emissivity of each surface at $T = 3000$ K. Also, determine the average absorptivity and reflectivity of each surface for radiation coming from a source at 3000 K. Which surface is more suitable to serve as a solar absorber?

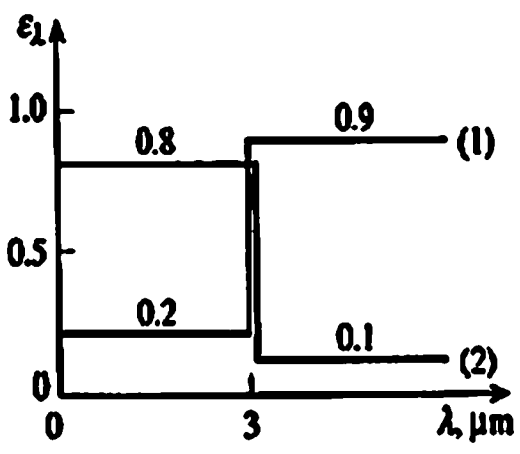


FIGURE P21–36

Step 1 of 9

Given temperature of the surface, $T = 3000$ K

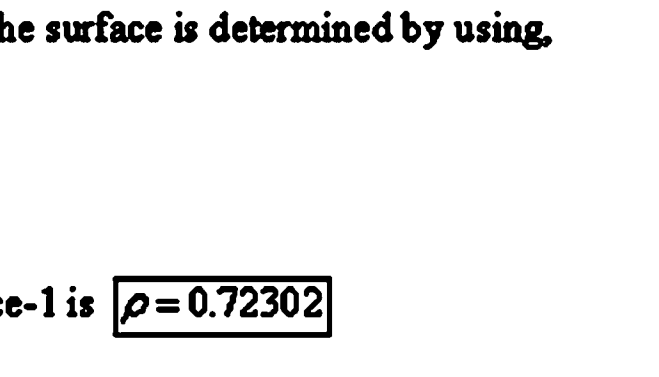
Emissivity of the surface-1 at wavelength less than 3 μm , $\varepsilon_{1a} = 0.2$

Emissivity of the surface-1 at wavelength greater than 3 μm , $\varepsilon_{1b} = 0.9$

Emissivity of the surface-2 at wavelength less than 3 μm , $\varepsilon_{2a} = 0.8$

Emissivity of the surface-2 at wavelength greater than 3 μm , $\varepsilon_{2b} = 0.1$

Step 2 of 9



Step 3 of 9

For the surface-1,

Considering the product of the wavelength and temperature,

$\lambda T = (3\ \mu m)(3000\ K)$

$\lambda T = 9{,}000\ \mu mK$

From blackbody radiation functions table at $\lambda T = 9000$,

$f_{\lambda 1} = 0.890029$

Step 4 of 9

Average emissivity of the surface is determined by using,

$$\varepsilon(T) = \varepsilon_{1a} f_{\lambda 1} + \varepsilon_{1b}(1 - f_{\lambda 1})$$

$$\varepsilon(T) = (0.2)(0.890029) + (0.9)(1 - 0.890029)$$

$$\varepsilon(T) = 0.27698$$

Hence emissivity of the surface-1, $\boxed{\varepsilon = 0.27698}$

From Kirchhoff's law, the absorptivity of the surface is $\boxed{\alpha = \varepsilon = 0.27698}$

Step 5 of 9

Then the reflectivity of the surface is determined by using,

$$\alpha + \rho = 1$$

$$\rho = 1 - \alpha$$

$$\rho = 1 - 0.27698$$

$$\rho = 0.72302$$

Reflectivity of the surface-1 is $\boxed{\rho = 0.72302}$

Step 6 of 9

For the surface-2,

Considering the product of the wavelength and temperature,

$\lambda T = (3\ \mu m)(3000\ K)$

$\lambda T = 9{,}000\ \mu mK$

From blackbody radiation functions table at $\lambda T = 9000$,

$f_{\lambda 2} = 0.890029$

Step 7 of 9

Average emissivity of the surface is determined by using,

$$\varepsilon(T) = \varepsilon_{2a} f_{\lambda 2} + \varepsilon_{2b}(1 - f_{\lambda 2})$$

$$\varepsilon(T) = (0.8)(0.890029) + (0.1)(1 - 0.890029)$$

$$\varepsilon(T) = 0.72302$$

Hence emissivity of the surface-2, $\boxed{\varepsilon = 0.72302}$

From Kirchhoff's law, the absorptivity of the surface is $\boxed{\alpha = \varepsilon = 0.72302}$

Step 8 of 9

Then the reflectivity of the surface is determined by using,

$$\alpha + \rho = 1$$

$$\rho = 1 - \alpha$$

$$\rho = 1 - 0.72302$$

$$\rho = 0.27698$$

Reflectivity of the surface-2 is $\boxed{\rho = 0.27698}$

Step 9 of 9

The second surface is more suitable to serve as a solar absorber since its absorptivity for short wavelength radiation (typical of radiation emitted by a high-temperature source such as the sun) is high, and its emissivity for long wavelength radiation (typical of emitted radiation from the absorber plate) is low.

Question: 21.037P

The emissivity of a surface coated with aluminum oxide can be approximated to be 0.15 for radiation at wavelengths less than 5 μm and 0.9 for radiation at wavelengths greater than 5 μm. Determine the average emissivity of this surface at (*a*) 5800 K and (*b*) 300 K. What can you say about the absorptivity of this surface for radiation coming from sources at 5800 K and 300 K?

Step 1 of 5

Given data:

The emissivity for the wave lengths less than $5\,\mu m$ is $\varepsilon_1 = 0.15$

The emissivity for the wave lengths greater than $5\,\mu m$ is $\varepsilon_2 = 0.9$

Step 2 of 5

a)

Average emissivity of the surface for the radiation at $T = 5800\ \text{K}$ is determined as

Fraction of radiation at wavelength $5\,\mu\text{m}$ is calculated by

$$\lambda T = (5\ \mu\text{m})(5800\ \text{K})$$
$$= 29{,}000\ \mu\text{mK}$$

From the black body radiation function f_λ table

To the corresponding λT value take black body radiation functions

$$f_\lambda = 0.992215 + (0.995340 - 0.992215)\left[\frac{29000 - 25000}{30000 - 25000}\right]$$

$$f_\lambda = 0.994715$$

Step 3 of 5

Average emissivity of the surface is determined by

$$\varepsilon(T) = \varepsilon_1 f_\lambda + \varepsilon_2 (1 - f_\lambda)$$
$$= (0.15)(0.994715) + (0.9)(1 - 0.994715)$$
$$= 0.153$$

$\therefore$ The average emissivity is $\boxed{\varepsilon(T) = 0.153}$

From Kirchhoff's law the absorptivity of the surface becomes $\boxed{\therefore\ \alpha = \varepsilon = 0.153}$

Step 4 of 5

b)

Average emissivity of the surface for the radiation at $T = 300\ \text{K}$ is determined as follows

Fraction of radiation at wavelength $5\ \mu\text{m}$ is calculated by

$$\lambda T = (5\ \mu\text{m})(300\ \text{K})$$
$$= 1500\ \mu\text{mK}$$

From the black body radiation function f_λ table

To the corresponding λT value take black body radiation functions

$$f_\lambda = 0.007790 + (0.019718 - 0.007790)\left[\frac{1500 - 1400}{1600 - 1400}\right]$$

$$f_\lambda = 0.013754$$

Step 5 of 5

Average emissivity of the surface is determined by

$$\varepsilon(T) = \varepsilon_1 f_\lambda + \varepsilon_2 (1 - f_\lambda)$$
$$= (0.15)(0.013754) + (0.9)(1 - 0.013754)$$

$\therefore$ The average emissivity is $\boxed{\varepsilon(T) = 0.889}$

From Kirchhoff's law the absorptivity of the surface becomes $\boxed{\therefore\ \alpha = \varepsilon = 0.889}$

Question: 21.038P

The variation of the spectral absorptivity of a surface is as given in Fig. P21-38. Determine the average absorptivity and reflectivity of the surface for radiation that originates from a source at $T = 2500$ K. Also, determine the average emissivity of this surface at 3000 K.

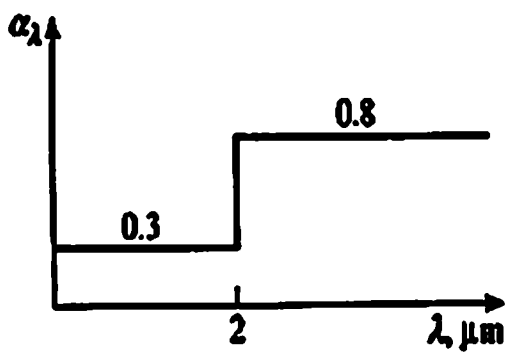
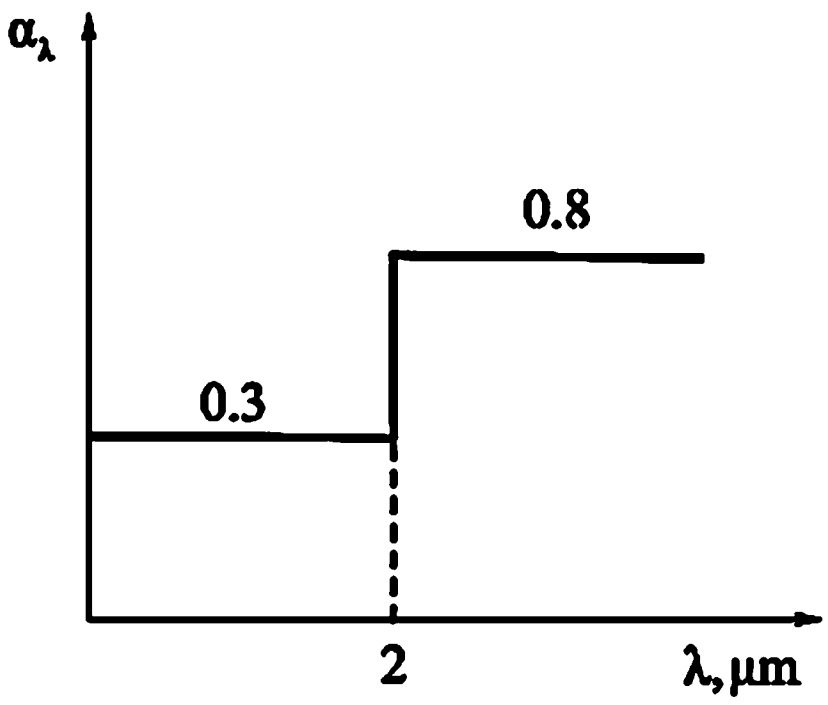


FIGURE P21–38

Step 1 of 4



Step 2 of 4

Considering the figure to determine the average absorptivity and reflectivity of the surface for radiation at T= 2500 K calculated as follows

Fraction of radiation at wavelength less than 2 μm is calculated by

$\lambda_1 T = (2\ \mu m)(2500\ K)$

$= 5000\ \mu mK$

From Blackbody radiation function f_λ

At $\lambda_1 T = 5000\ \mu mK$

$f_{\lambda_1} = 0.633747$

Average absorptivity of the surface is determined by

$$\alpha(T) = \alpha_1 f_\lambda + \alpha_2 (1 - f_\lambda)$$
$$= (0.3)(0.633747) + (0.8)(1 - 0.633747)$$
$$= 0.48$$

Therefore, average absorptivity of the surface is $\boxed{\alpha(T) = 0.48}$

Step 3 of 4

Then the reflectivity of the surface is determined by

$$\alpha + \rho = 1$$
$$\rho = 1 - \alpha$$
$$= 1 - 0.48$$
$$= 0.52$$

Therefore, average reflectivity of the surface is $\boxed{\rho = 0.52}$

Step 4 of 4

The average emissivity of the surface at T= 3000 K is determined using Kirchhoff's law i.e. $\alpha = \varepsilon$

$\lambda T = (2\ \mu m)(3000\ K)$

$= 6000\ \mu mK$

From Blackbody radiation function f_λ

At $\lambda T = 6000\ \mu mK$

$f_\lambda = 0.737818$

Average absorptivity of the surface is determined by

$$\varepsilon(T) = \varepsilon_1 f_\lambda + \varepsilon_2 (1 - f_\lambda)$$
$$= (0.3)(0.737818) + (0.8)(1 - 0.737818)$$
$$= 0.43$$

Therefore, average emissivity of the surface is $\boxed{\varepsilon(T) = 0.43}$.

Question: 21.039P

The reflectivity of aluminum coated with lead sulfate is 0.35 for radiation at wavelengths less than 3 μm and 0.95 for radiation greater than 3 μm. Determine the average reflectivity of this surface for solar radiation (T ≈5800 K) and radiation coming from surfaces at room temperature (T ≈ 300 K). Also, determine the emissivity and absorptivity of this surface at both temperatures. Do you think this material is suitable for use in solar collectors?

Step 1 of 7

Given

Reflectivity of surface at wavelength less than $3\,\mu m$ is $\rho_1 = 0.35$

Reflectivity of surface at wavelength greater than $3\,\mu m$ is $\rho_2 = 0.95$

Temperature of the solar radiation is $T = 5800\ K$

Considering the product of the wavelength and temperature,

$\lambda_1 T = (3\ \mu m)(5800\ K)$

$\lambda_1 T = 17400\ \mu mK$

From blackbody radiation functions table at $\lambda T = 17400$,

$f_{\lambda 1} = 0.978746$

Step 2 of 7

Average emissivity of the surface is determined by using,

$$\rho(T) = \rho_1 f_{\lambda 1} + \rho_2 (1 - f_{\lambda 1})$$
$$\rho(T) = (0.35)(0.978746) + (0.95)(1 - 0.978746)$$
$$\rho(T) = 0.362$$

$\boxed{\rho(T) = 0.362}$

Step 3 of 7

The reflectivity of the surface is determined by using,

$$\alpha + \rho = 1$$
$$\alpha = 1 - \rho$$
$$\alpha = 1 - 0.362$$
$$\alpha = 0.638$$

absorptivity of the surface, $\boxed{\alpha = 0.638}$

From Kirchhoff's law the emissivity of the surface becomes $\boxed{\alpha = \varepsilon = 0.638}$

Step 4 of 7

Temperature of the room is $T = 300\ K$

Considering the product of the wavelength and temperature,

$\lambda_1 T = (3\ \mu m)(300\ K)$

$\lambda_1 T = 900\ \mu mK$

From blackbody radiation functions table at $\lambda T = 900$,

$f_{\lambda 1} = 0.0001685$

Step 5 of 7

Average emissivity of the surface is determined by

$$\rho(T) = \rho_1 f_\lambda + \rho_2 (1 - f_\lambda)$$
$$\rho(T) = (0.35)(0.0001685) + (0.95)(1 - 0.0001685)$$
$$\rho(T) = 0.95$$

$\boxed{\rho(T) = 0.95}$

Step 6 of 7

Then the reflectivity of the surface is determined by

$$\alpha + \rho = 1$$
$$\alpha = 1 - \rho$$
$$\alpha = 1 - 0.95$$
$$\alpha = 0.05$$

absorptivity of the surface, $\boxed{\alpha = 0.05}$

From Kirchhoff's law the emissivity of the surface becomes $\boxed{\alpha = \varepsilon = 0.05}$

Step 7 of 7

The temperature of the aluminum plate is close to room temperature, and thus emissivity of the plate will be equal to its absorptivity at room temperature. That is,

$$\varepsilon = \varepsilon_{room} = 0.05$$
$$\alpha = \alpha_s = 0.638$$

This makes it suitable for using it as solar collector. ($\alpha_s = 1$ and $\varepsilon_{room} = 0$ for an ideal solar collector)

Question: 21.040P

The variation of the spectral transmissivity of a 0.6-cm-thick glass window is as given in Fig. P21-40. Determine the average transmissivity of this window for solar radiation ($T \approx 5800$ K) and radiation coming from surfaces at room temperature ($T \approx 300$ K). Also, determine the amount of solar radiation transmitted through the window for incident solar radiation of 650 W/m2.

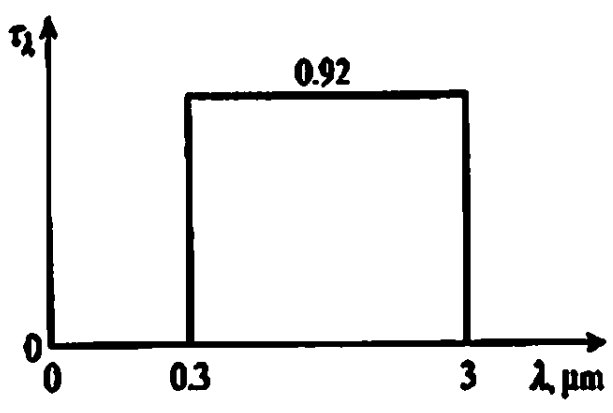


FIGURE P21–40

Step 1 of 7

Calculate the value of the λT for surface at temperature $5800\ \text{K}$ and wavelength $0.3\ \mu\text{m}$.

Substitute $0.3\ \mu\text{m}$ for λ and $5800\ \text{K}$ for T.

$$\lambda_1 T_1 = (0.3\ \mu\text{m})(5800\ \text{K})$$
$$\lambda_1 T_1 = 1740\ \mu\text{mK}$$

Determine the black body radiation function, f_{λ_1} of the window for $1740\ \mu\text{mK}$.

Take the black body radiation function by linear interpolating the value between $1600\ \mu\text{mK}$ and $1800\ \mu\text{mK}$.

$$f_{\lambda_1} = 0.019718 + (0.039341 - 0.019718)\left[\frac{1740-1600}{1800-1600}\right]$$
$$f_{\lambda_1} = 0.03345$$

Step 2 of 7

Calculate the value of the λT for surface at temperature $5800\ \text{K}$ and wavelength $3\ \mu\text{m}$.

Substitute $3\ \mu\text{m}$ for λ and $5800\ \text{K}$ for T.

$$\lambda_2 T_2 = (3\ \mu\text{m})(5800\ \text{K})$$
$$\lambda_2 T_2 = 17400\ \mu\text{mK}$$

Determine the black body radiation function, f_{λ_2} of the window for $17400\ \mu\text{mK}$.

Take the black body radiation function by linear interpolating the value between $16000\ \mu\text{mK}$ and $18000\ \mu\text{mK}$.

$$f_{\lambda 2} = 0.973814 + (0.980860 - 0.973814)\left[\frac{17400-16000}{18000-16000}\right]$$
$$f_{\lambda 2} = 0.9787462$$

Step 3 of 7

Calculate the average transmissivity of the surface at $5800\ \text{K}$.

$$\tau(T) = \tau_1\left(f_{\lambda_2} - f_{\lambda_1}\right)$$

Here, $\tau(T)$ is the average transmissivity of the surface and τ_1 is the transmissivity of the surface.

Substitute 0.9787462 for f_{λ_2} , 0.03345 for f_{λ_1} and 0.92 for τ_1

$$\tau(T) = (0.92)(0.9787462 - 0.03345)$$
$$\tau(T) = 0.87$$

Thus the transmissivity is $\boxed{0.87}$.

Step 4 of 7

Calculate the value of the λT for surface at temperature $300\ \text{K}$ and wavelength $0.3\ \mu\text{m}$.

Substitute $0.3\ \mu\text{m}$ for λ and $300\ \text{K}$ for T.

$$\lambda_1 T_1 = (0.3\ \mu\text{m})(300\ \text{K})$$
$$\lambda_1 T_1 = 90\ \mu\text{mK}$$

Determine the black body radiation function, $f_{\lambda 1}$ of the window for $90\ \mu\text{mK}$.

$$f_{\lambda 1} = 0$$

Step 5 of 7

Calculate the value of the λT for surface at temperature $300\ \text{K}$ and wavelength $0.3\ \mu\text{m}$.

Substitute $3\ \mu\text{m}$ for λ and $300\ \text{K}$ for T.

$$\lambda_2 T_2 = (3\ \mu\text{m})(300\ \text{K})$$
$$\lambda_2 T_2 = 900\ \mu\text{mK}$$

Determine the black body radiation function, $f_{\lambda 2}$ of the window for $900\ \mu\text{mK}$.

Take the black body radiation function by linear interpolating the value between $800\ \mu\text{mK}$ and $1000\ \mu\text{mK}$.

$$f_{\lambda_2} = 0.000016 + (0.000321 - 0.000016)\left[\frac{900-800}{1000-800}\right]$$
$$f_{\lambda_2} = 0.00001685$$

Step 6 of 7

Calculate the average transmissivity of the surface at $300\ \text{K}$.

$$\tau(T) = \tau_1\left(f_{\lambda_2} - f_{\lambda_1}\right)$$

Substitute 0.00001685 for f_{λ_2} , 0 for f_{λ_1} and 0.92 for τ_1

$$\tau(T) = (0.92)(0.00001685 - 0)$$
$$\tau(T) = 0.000016$$

Thus the transmissivity is $\boxed{0.000016}$.

Step 7 of 7

Calculate the amount of solar radiation transmitted through this glass.

$$G_{tr} = \tau G_{\text{incident}}$$

Here, G_{incident} is the incident solar radiation and τ is transmissivity of surface.

Substitute $650\ \text{W/m}^2$ for G_{incident} and 0.87 for τ

$$G_{tr} = 0.87(650\ \text{W/m}^2)$$
$$G_{tr} = 566\ \text{W/m}^2$$

Thus, the amount of solar radiation transmitted is $\boxed{566\ \text{W/m}^2}$.

Question: 21.041P

An opaque horizontal plate is well insulated on the edges and the lower surface. The irradiation on the plate is 3000 W/m2, of which 500 W/m2 is reflected. The plate has a uniform temperature of 700 K and has an emissive power of 5000 W/m2. Determine the total emissivity and absorptivity of the plate.

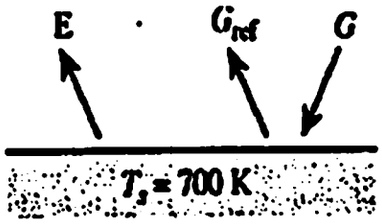


FIGURE P21-41

Step 1 of 3

Irradiation on the plate, $G = 3000\ \text{W/m}^2$

Reflected irradiation, $G_{ref} = 500\ \text{W/m}^2$

Temperature of the plate, $T = 700\ \text{K}$

Emissive power of the plate, $E = 5000\ \text{W/m}^2$

Step 2 of 3

The emissivity of the plate is given by

$$\varepsilon = \frac{E}{E_b}$$
$$= \frac{E}{\sigma T^4}$$
$$= \frac{5000}{\left(5.67\times10^{-8}\right)700^4}$$
$$= 0.367$$

Thus the emissivity of the plate is $\boxed{\varepsilon = 0.367}$

Step 3 of 3

The absorptivity of the plate is

$$\alpha = \frac{G_{abs}}{G}$$
$$= \frac{G - G_{ref}}{G}$$
$$= \frac{3000 - 500}{3000}$$
$$= 0.833$$

Thus the absorptivity of the plate is $\boxed{\alpha = 0.833}$

Question: 21.042P

A horizontal plate is experiencing uniform irradiation on the both upper and lower surfaces. The ambient air temperature surrounding the plate is 290 K with a convection heat

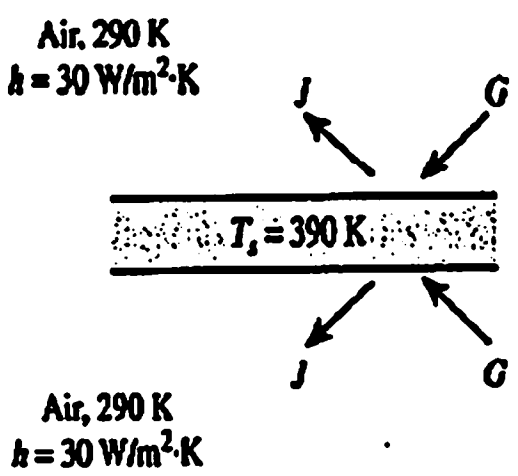


FIGURE P21–42

transfer coefficient of 30 W/m2•K. Both upper and lower surfaces of the plate have a radiosity of 4000 W/m2, and the plate temperature is maintained uniformly at 390 K. If the plate is not opaque and has an absorptivity of 0.527, determine the irradiation and emissivity of the plate.

Step 1 of 5

Temperature of the plate, $T = 390$ K
Temperature of the air, $T_\infty = 290$ K

The convective coefficient of air is $h = 30$ W/m^{2}K

Radiocity of the plate, $J = 4000$ W/m^2
Absorptivity of the plate, $\alpha = 0.527$
Let the irradiation on the plate be G
Let the Emissive power of the plate be E

Step 2 of 5

The upper and lower surfaces of the plate are subjected to similar conditions. Hence it is sufficient to consider either of the surfaces for energy balance. By considering the energy balance on upper surface of the plate, we get

$$Q_{abs} = Q_{con} + E$$
$$\alpha G = h(T - T_\infty) + E$$
$$E = (0.527 \times G) - 30(390 - 290)$$
$$E = (0.527 \times G) - 3000 \quad \text{..... (1)}$$

Step 3 of 5

From the definition of the radiocity, we have

$$J = Q_{ref} + E$$
$$E = J - Q_{ref}$$
$$E = J - \rho G$$
$$E = J - (1 - \alpha)G$$
$$E = 4000 - (1 - 0.527)G$$
$$E = 4000 - (0.473 \times G) \quad \text{..... (2)}$$

Step 4 of 5

By considering equations (1) and (2), we get

$$(0.527 \times G) - 3000 = 4000 - (0.473 \times G)$$
$$(0.527 + 0.473)G = 4000 + 3000$$
$$G = 7000 \text{ W/m}^2$$

Thus the irradiation on the surface is $\boxed{G = 7000 \text{ W/m}^2}$

Step 5 of 5

Substituting the values is equation (2), we get

$$E = 4000 - (0.473 \times G)$$
$$\varepsilon E = 4000 - (0.473 \times 7000)$$
$$\varepsilon = \frac{4000 - 3311}{E}$$
$$= \frac{4000 - 3311}{(5.67 \times 10^{-8})390^4}$$
$$= 0.525$$

Thus the emissivity of the plate is $\boxed{\varepsilon = 0.525}$

Question: 21.043P

What does the view factor represent? When is the view factor from a surface to itself not zero?

Step 1 of 1

View Factor is a geometric quantity. It does not depend on surface properties and temperature. The other names for view factor are shape factor, configuration factor, and angle factor.

The view factor is denoted as $F_{i \to j}$ or F_{ij} . It indicates the fraction of the radiation leaving surface *i* that strikes surface *j*.

For **concave surfaces**, view factor from a surface to itself not zero.

Question: 21.044P

What is the crossed-strings method? For what kind of geometries is the crossed-strings method applicable?

Step 1 of 1

Crossed strings method is applicable for the channels and ducts that are very long in one direction relative to the other directions. Such geometries can conveniently be considered to be two-dimensional, since any radiation interaction through their end surface is negligible. These geometries can subsequently be modeled as being infinitely long

View factor for the crossed strings method is expressed as

$$F_{i\to j}=\frac{\sum(\text{Crossed strings})-\sum(\text{Uncrossed strings})}{2\times(\text{String on surface } i)}$$

The crossed-string method is applicable even when the two surfaces considered share a common edge, as in a triangle. I such cases, the common edge can be treated as an imaginary string of zero length. The method can also be applied to surfaces that are partially blocked by other surfaces by allowing the strings to bend around the blocking surfaces

Question: 21.045P

What are the summation rule and the superposition rule for view factors?

Step 1 of 1

The sum of the view factors from surface i of an enclosure to all surfaces of the enclosure, including to itself, must equal unity. This is known as the summation rule for an enclosure and it is expressed as

$$\sum_{j=1}^{N} F_{i \to j} = 1$$

Superposition rule can be expressed as the view factor from a surface i to a surface j is equal to the sum of the view factors from surface i to the parts of surface j.

Question: 21.046P

How can you determine the view factor *F*12 when the view factor *F*21 and the surface areas are available?

Step 1 of 1

The view factors $F_{i\to j}$ and $F_{j\to i}$ are not equal to each other unless the areas of the two surfaces are. That is

$F_{j\to i} = F_{i\to j}$ when $A_i = A_j$

$F_{j\to i} \neq F_{i\to j}$ when $A_i \neq A_j$

The pair of view factors $F_{i\to j}$ and $F_{j\to i}$ are related to each other by the reciprocity rule

$A_i F_{ij} = A_j F_{ji}$

Where A_i is the area of the surface i and A_j is the area of the surface j.

Therefore,

$A_1 F_{12} = A_2 F_{21}$

$$F_{12} = \frac{A_2}{A_1} F_{21}$$

Question: 21.047P

Consider an enclosure consisting of five surfaces. How many view factors does this geometry involve? How many of these view factors can be determined by the application of the reciprocity and summation rules?

Step 1 of 1

An enclosure consisting of eight surfaces involves view factors are

$$N^2 = 5^2$$
$$= 25$$

The total number of view factors that need to be evaluated directly for the N- surfaces enclosure becomes

$$\frac{N(N-1)}{2} = \frac{5(5-1)}{2}$$
$$= 10$$

The remaining $25-10=15$ the view factors can be determined from the 15 equations that are obtained by the application of the reciprocity and summation rules.

Question: 21.048P

Consider a cylindrical enclosure with *A*1, *A*2, and *A*3 representing the internal base, top, and side surfaces, respectively. Using the length to diameter ratio, $K = L/D$, determine (*a*) the expression for the view factor between the base and the side surface *F*13 in terms of *K* and (*b*) the value of the view factor *F*13 for $L = D$.

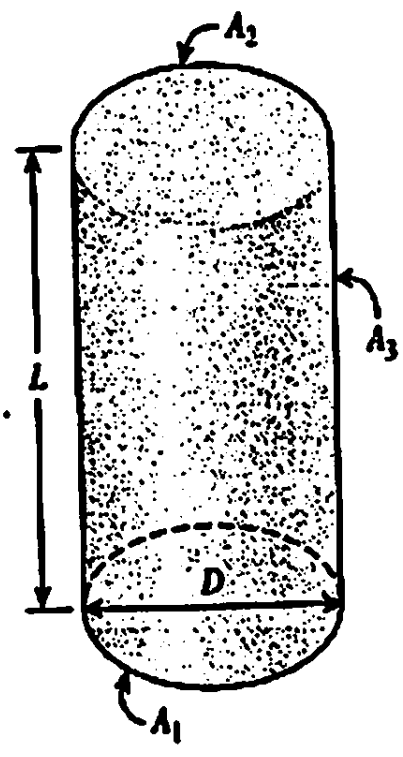


FIGURE P21–48

Step 1 of 5

Given data:

Number of surfaces in an enclosure, $N = 13$

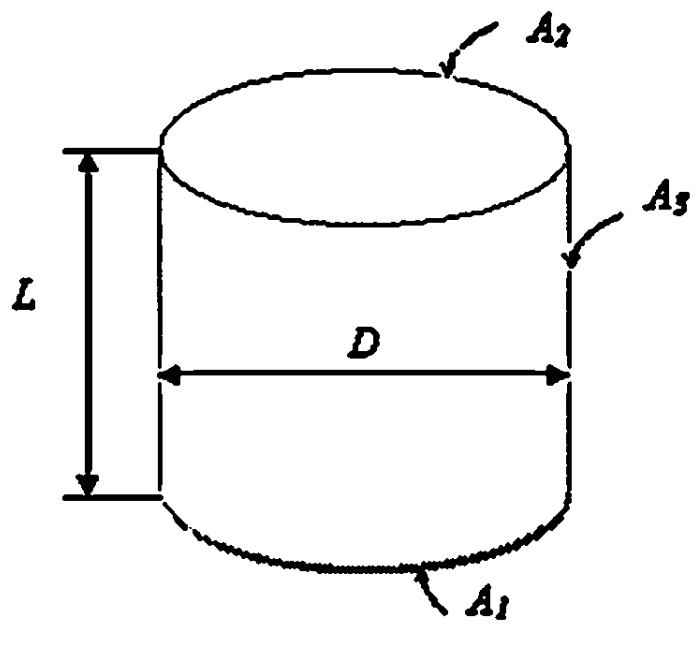
Surface area of the base, $A_1 = \frac{\pi}{4}D^2$

Surface area of the top, $A_2 = \frac{\pi}{4}D^2$

Surface area of the side surface, $A_3 = \pi DL$

Length to diameter ratio, $K = \frac{L}{D}$

Step 2 of 5



Step 3 of 5

a) The view factor from the base to the top can be determined from the table view factor expressions for common geometries

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

Where, $R = \frac{D}{2L}$

By rewriting the R in terms of K, $R = \frac{1}{2K}$

Therefore, the view factor from the base to the top written as

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

$$= 1 + \frac{1 - \sqrt{4(1/2K)^2 + 1}}{2(1/2K)^2}$$

$$= 1 + \frac{1 - \sqrt{1/K^2 + 1}}{1/2K^2}$$

$$= 1 + 2K\left(K - \sqrt{K^2 + 1}\right)$$

Step 4 of 5

The view factor from the base to the side surface is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1$$

$$F_{13} = 1 - F_{11} - F_{12}$$

Since the base surface is flat and thus $F_{11} = 0$

Therefore substituting the known values in the above expression the view factor from the base to the side surface can be expressed as

$$F_{13} = 1 - F_{11} - F_{12}$$

$$= 1 - 0 - \left(1 + 2K\left(K - \sqrt{K^2 + 1}\right)\right)$$

$$= 2K\sqrt{K^2 + 1} - 2K^2$$

$$\boxed{F_{13} = 2K\sqrt{K^2 + 1} - 2K^2}$$

Step 5 of 5

b) The value of the view factor from the base to the side surface for $L = D$ can be determined as

For $L = D$ the value of K is

$$K = \frac{L}{D}$$

$$= \frac{D}{D}$$

$$K = 1$$

And by substituting the value of K, the view factor F_{13} can be calculated as

$$F_{13} = 2K\sqrt{K^2 + 1} - 2K^2$$

$$= 2 \times 1 \times \sqrt{1^2 + 1} - 2 \times 1^2$$

$$F_{13} = 0.828$$

$$\boxed{F_{13} = 0.828}$$

Question: 21.049P

Two infinitely long parallel plates of width *w* are located at *w* distance apart, as shown in the figure. Using the Hottel's crossed-strings method, determine the view factor *F*12.

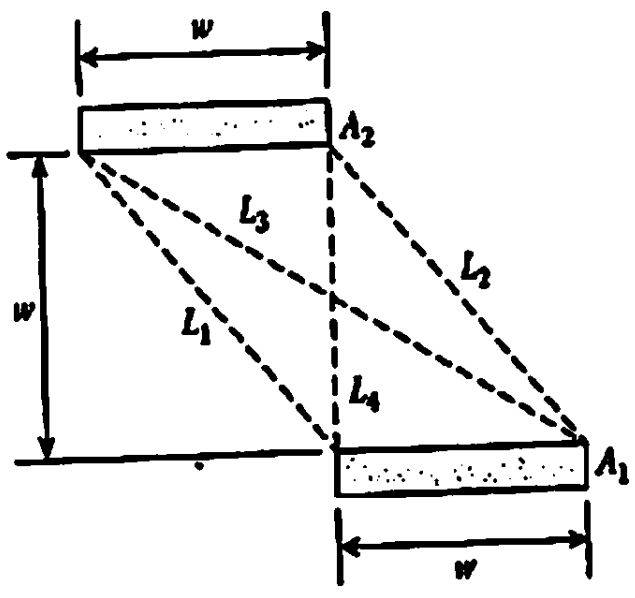


FIGURE P21–49

Step 1 of 10

Given data:

Width of the two plates is w and the view factors between two infinitely long parallel plates are to be determined using the Hottel's crossed-strings method.

Step 2 of 10

Assumptions:

The surfaces are diffuse emitters and reflectors.

Step 3 of 10



L_2

L_3

Step 4 of 10

w

L_1

Step 5 of 10

L_4

Step 6 of 10

Step 7 of 10

Step 8 of 10

Step 9 of 10

The view factor from the base plate to the top plate can be determined using the Hottel's cross-strings method as

$$F_{12} = \frac{\sum(\text{Crossed strings}) - \sum(\text{Uncrossed strings})}{2 \times (\text{String on surface 1})}$$

$$= \frac{(L_3 + L_4) - (L_1 + L_2)}{2w}$$

Where

$$L_4 = w$$

$$L_2 = \sqrt{L_4^2 + w^2}$$

$$= \sqrt{w^2 + w^2}$$

$$= \sqrt{2}w$$

$$L_1 = \sqrt{L_4^2 + w^2}$$

$$= \sqrt{w^2 + w^2}$$

$$= \sqrt{2}w$$

$$L_3 = \sqrt{(2w)^2 + (w)^2}$$

$$= \sqrt{5}w$$

Step 10 of 10

Substituting the known values in F_{12}

$$F_{12} = \frac{(L_3 + L_4) - (L_1 + L_2)}{2w}$$

$$= \frac{(\sqrt{5}w + w) - (\sqrt{2}w + \sqrt{2}w)}{2w}$$

$$= \frac{w\left[(\sqrt{5} + 1) - (\sqrt{2} + \sqrt{2})\right]}{2w}$$

$$= 0.204$$

$$\boxed{F_{13} = 0.204}$$

Question: 21.050P

Determine the view factors *F*13 and *F*23 between the rectangular surfaces shown in Fig. P21-50.

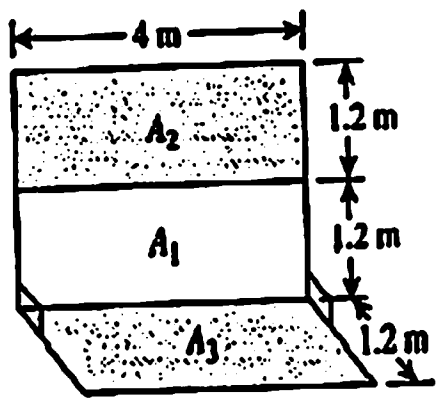
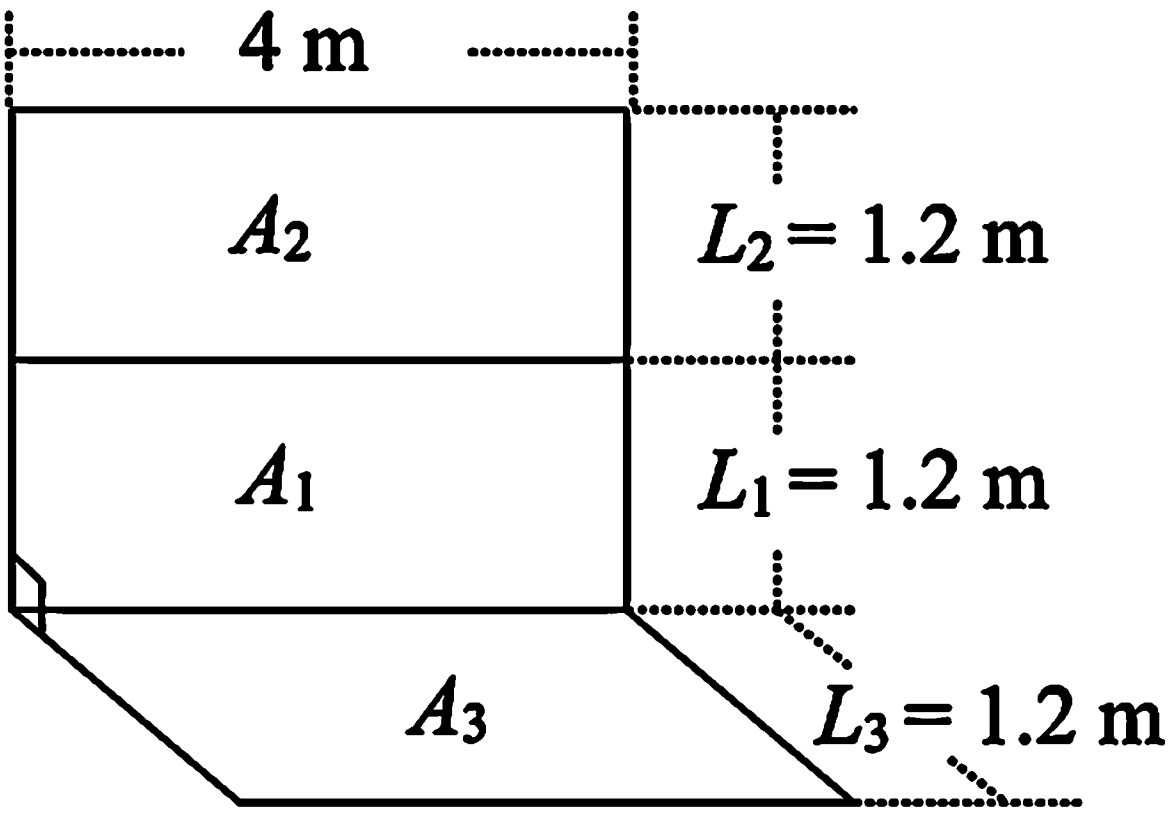


FIGURE P21–50

Step 1 of 4

Figure showing the geometry of the surfaces



Step 2 of 4

From the figure, the length to width ratio for surfaces 3 and 1 are

$$\frac{L_3}{w} = \frac{1.2}{4} = 0.3$$
$$\frac{L_1}{w} = \frac{1.2}{4} = 0.3$$

Where L_3 and L_1 are the lengths, and w is the width of surfaces 3 and 1 respectively

From the table of view factor between two perpendicular rectangles, at $(L_3/w) = 0.3$ and $(L_1/w) = 0.3$, the shape factor is

$$F_{31} = 0.26$$

Step 3 of 4

From the figure, the length to width ratio for entire vertical surface (surface-1 and surface-2) and horizontal surface (surface-3) is

$$\frac{L_1 + L_2}{w} = \frac{2.4}{4}$$
$$= 0.6$$

From the table of view factor between two perpendicular rectangles, at $(L_3/w) = 0.3$ and $(L_1 + L_2)/w = 0.6$, the shape factor is

$$F_{3\to(1+2)} = 0.32$$

Apply the reciprocity rule to the surfaces 1 and 3

$$A_1 F_{13} = A_3 F_{31}$$
$$A \times F_{13} = A \times F_{31}$$
$$F_{13} = F_{31}$$
$$= 0.26$$

Therefore the view factor $\boxed{F_{13} = 0.26}$

Step 4 of 4

Apply the superposition rule to the surfaces

$$F_{3\to(1+2)} = F_{31} + F_{32}$$
$$0.32 = 0.26 + F_{32}$$
$$F_{32} = 0.06$$

Apply the reciprocity rule to the surfaces 2 and 3

$$A_2 F_{23} = A_3 F_{32}$$
$$A \times F_{23} = A \times F_{32}$$
$$F_{23} = F_{32}$$
$$= 0.06$$

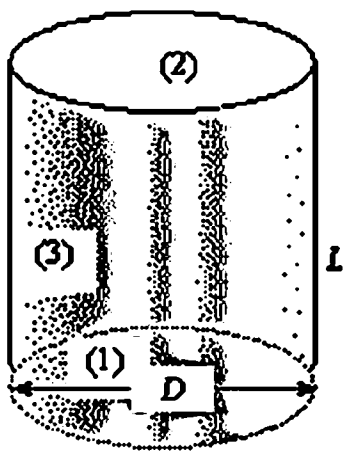
Therefore the view factor $\boxed{F_{23} = 0.06}$

Question: 21.051P

Consider a cylindrical enclosure whose height is twice the diameter of its base. Determine the view factor from the side surface of this cylindrical enclosure to its base surface.

Step 1 of 5

Given that,
Height is twice the diameter $L = 2D$



Step 2 of 5

The view factor from the side surface of this cylindrical enclosure can be calculated by using the table of view factor expressions for some common geometries of finite size (3-D)

$$R_1 = \frac{r_1}{L}$$
$$= \frac{r_1}{r_1}$$
$$= 1$$
$$\& R_2 = \frac{r_2}{L}$$
$$= \frac{r_2}{r_2}$$
$$= 1$$
$$S = 1 + \frac{1 + R_2^2}{R_1^2}$$
$$= 1 + \frac{1 + 1^2}{1^2}$$
$$= 3$$

Step 3 of 5

The view factor from surface 1-2 is given by

$$F_{12} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{R_2}{R_1}\right)^2\right]^{0.5}\right\}$$
$$= \frac{1}{2}\left\{3 - \left[3^2 - 4\left(\frac{1}{1}\right)^2\right]^{0.5}\right\}$$
$$= 0.3819$$

Step 4 of 5

From the summation rule
$F_{11} + F_{12} + F_{13} = 1$
$F_{11} = 0$
$0 + 0.3819 + F_{13} = 1$
$F_{13} = 0.618$

Step 5 of 5

From the reciprocity rule
$A_1 F_{13} = A_3 F_{31}$

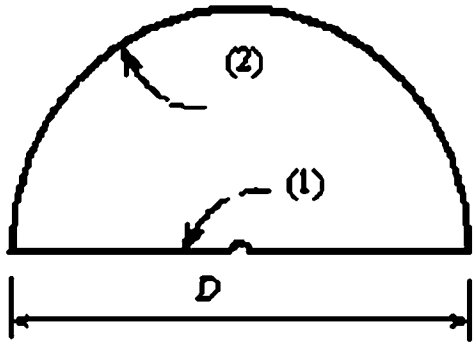
$$F_{31} = \frac{A_1}{A_3} F_{13}$$
$$= \frac{\pi r_1^2}{2\pi r_1 L} F_{13}$$
$$= \frac{\pi r_1^2}{2\pi r_1 (r_1)} F_{13}$$
$$= \frac{1}{2}(0.618)$$
$$= 0.309$$

∴ The view factor from the side surface to its base surface, $\boxed{F_{31} = 0.309}$

Question: 21.052P

Consider a hemispherical furnace with a flat circular base of diameter *D*. Determine the view factor from the dome of this furnace to its base.

Step 1 of 3



Step 2 of 3

From the reciprocity rule

$F_{11}+F_{12}=1$

View factor for the surface 1 it self is $F_{11}=0$

$0+F_{12}=1$

$F_{12}=1$

Step 3 of 3

From the reciprocity rule

$$A_1F_{12}=A_2F_{21}$$

$$F_{21}=\frac{A_1}{A_2}F_{12}$$

$$=\frac{\pi r^2}{2\pi r^2}F_{12}$$

$$=\frac{1}{2}(1)$$

$$=0.5$$

∴ The view factor from the dome of this surface to its base $\boxed{F_{21}=0.5}$

Question: 21.053P

Three infinitely long parallel cylinders of diameter D are located a distance s apart from each other. Determine the view factor between the cylinder in the middle and the surroundings.

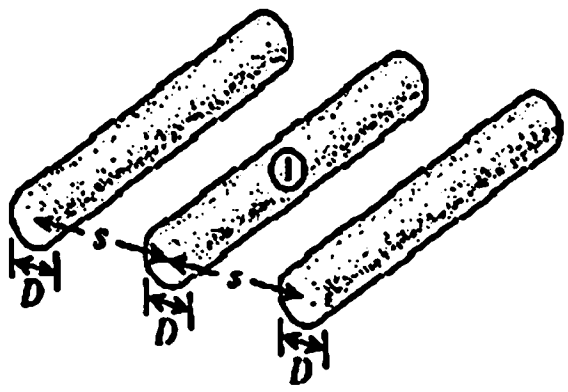
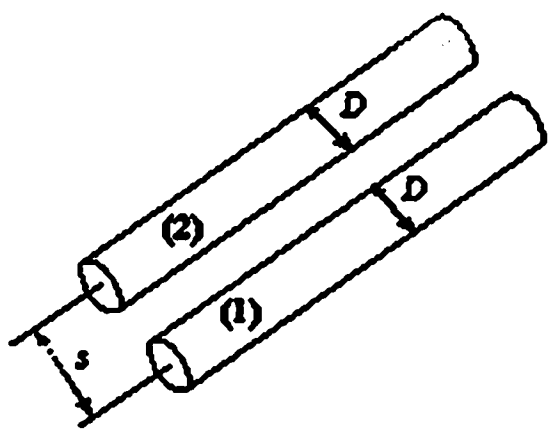


FIGURE P21–53

Step 1 of 4



Step 2 of 4

Using cross string method

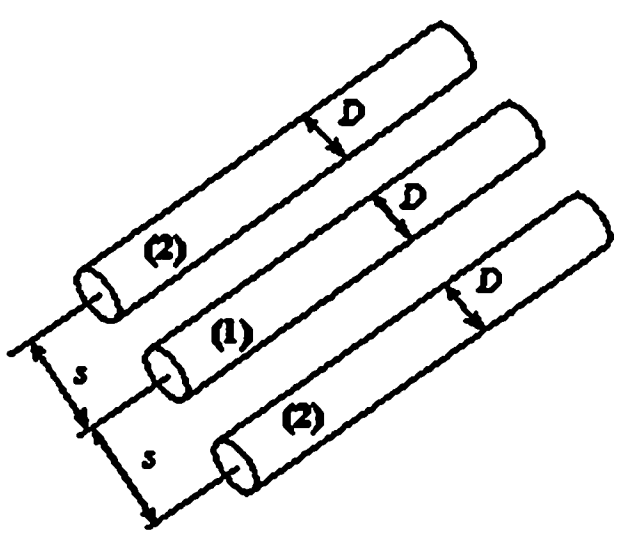
$$F_{1-2} = \frac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{String on surface 1}}$$

$$= \frac{2\sqrt{s^2 + D^2} - 2s}{2(\pi D / 2)}$$

∴ The view factor between two cylinders facing each other is

$$F_{1-2} = \frac{2\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}$$

Step 3 of 4



Step 4 of 4

Noting that the radiation leaving cylinder 1 that does not strike the cylinder will strike the surroundings, and this is also the case for the other half of the cylinder, the view factor between the cylinder in the middle and the surroundings becomes

$$F_{1-surr} = 1 - 2F_{1-2}$$

$$F_{1-surr} = 1 - 2\left(\frac{2\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}\right)$$

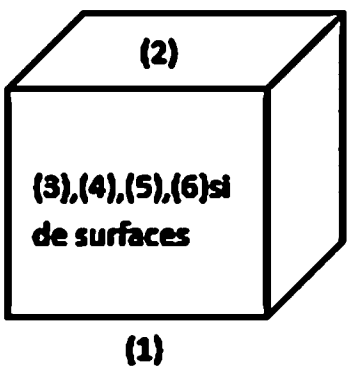
$$\therefore \boxed{F_{1-surr} = 1 - \frac{4\left(\sqrt{s^2 + D^2} - s\right)}{\pi D}}$$

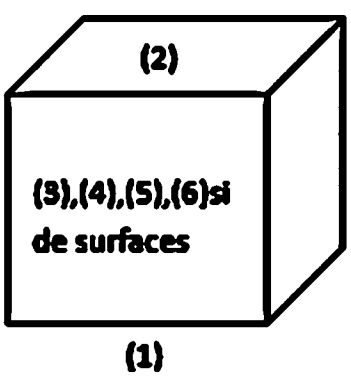
Question: 21.054P

Determine the view factors from the base of a cube to each of the other five surfaces.

Step 1 of 3

Draw the cubic box and indicate the sides of cube with numbers.





Step 2 of 3

From the figure the base of cube is side-1.

Side-2 is aligned parallel to the side-1, and the sides 3, 4, 5, and 6 are aligned perpendicular to the side-1.

Calculate the view factor side-2 from the base.

Obtain the ratio $\frac{L_1}{D}$ and the ratio $\frac{L_2}{D}$.

Here, L_1 is the length of cube, L_2 is the width of cube, and D is the height of cube.

For cubic box all sides are equal, $L_1 = L_2 = D$.

$$\frac{L_1}{D} = 1$$

$$\frac{L_2}{D} = 1$$

Obtain the view factor F_{12} at $\frac{L_1}{D} = 1$ and $\frac{L_2}{D} = 1$ from the plot " view factor between two aligned parallel rectangles of equal size" of the text book.

$$F_{12} = 0.2$$

Therefore, the view factor of side-2 from the base of cube is, $\boxed{0.2}$.

Step 3 of 3

Calculate the view factor side-3 from the base.

Obtain the ratio $\frac{L_1}{W}$ and $\frac{L_2}{W}$.

Here, L_1 is the length of cube, L_2 is the height of cube, and W is the width of cube.

For cubic box all sides are equal, $L_1 = L_2 = W$.

$$\frac{L_1}{W} = 1$$

$$\frac{L_2}{W} = 1$$

Obtain the view factor F_{12} at $\frac{L_1}{W} = 1$ and $\frac{L_2}{W} = 1$ from the plot " view factor between two aligned parallel rectangles of equal size" of the text book.

$$F_{13} = 0.2$$

Therefore, the view factor of side-3 from the base of cube is, $\boxed{0.2}$.

The sides 4, 5, and 6 are perpendicular to the base (side-1) as like side-3.

Therefore, the view factors of sides4, 5, and 6 from base are, $\boxed{F_{14} = F_{15} = F_{16} = 0.2}$.

Question: 21.055P

Consider a conical enclosure of height h and base diameter D. Determine the view factor from the conical side surface to a hole of diameter d located at the center of the base.

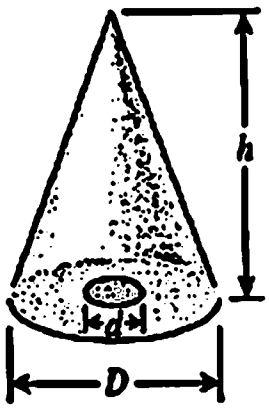
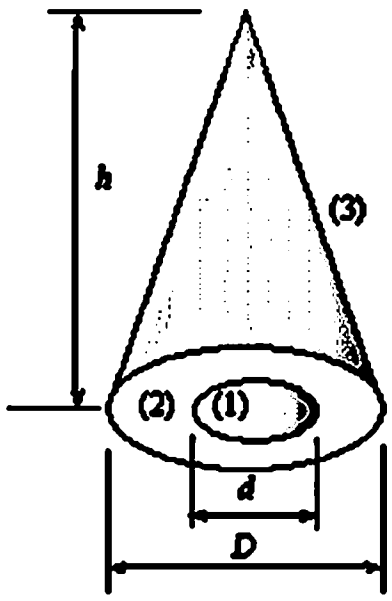


FIGURE P21–55

Step 1 of 3



Step 2 of 3

Surfaces 1 and 2 are flat, and they have no direct view of each other. Therefore,

$$F_{11} = F_{22} = F_{12} = F_{21} = 0$$

From the summation rule

$$F_{11} + F_{12} + F_{13} = 1$$

$$F_{13} = 1$$

Step 3 of 3

From the reciprocity rule

$$A_1 F_{13} = A_3 F_{31}$$

$$F_{31} = \frac{A_1}{A_3} F_{13}$$

$$= \frac{\pi \frac{d^2}{4}}{\frac{\pi D h}{2}} \times (1)$$

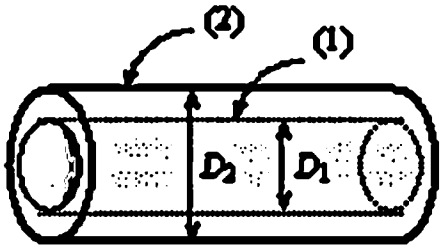
$$= \frac{d^2}{2dh}$$

∴ The view factor from the conical side surface to a hole of diameter d $\boxed{F_{31} = \frac{d^2}{2dh}}$

Question: 21.056P

Determine the four view factors associated with an enclosure formed by two very long concentric cylinders of radii $r1$ and $r2$. Neglect the end effects.

Step 1 of 3



Step 2 of 3

No radiation leaving surface 1 strikes itself and thus $\therefore F_{11} = 0$

All radiation leaving surface 1 strikes surface 2 and thus $\therefore F_{12} = 1$

From the reciprocity rule

$$A_1 F_{12} = A_2 F_{21}$$

$$F_{21} = \frac{A_1}{A_2} F_{12}$$

$$= \frac{2\pi r_1 h}{2\pi r_2 h} (1)$$

$\therefore$ The view factor for the surface 1 to 2 is $\boxed{F_{21} = \frac{r_1}{r_2}}$

Step 3 of 3

From the reciprocity rule

$$F_{21} + F_{22} = 1$$

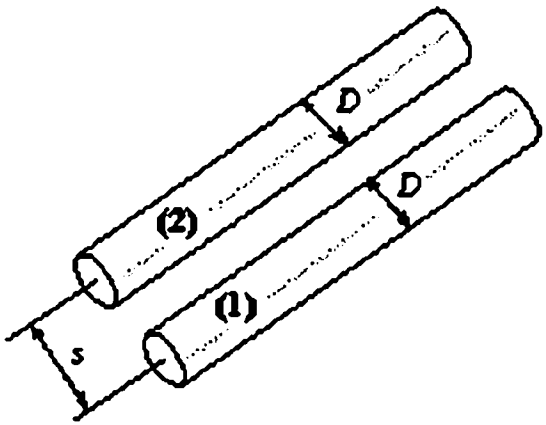
$$F_{22} = 1 - F_{21}$$

$\therefore$ The view factor for the surface 2 itself $\boxed{F_{22} = 1 - \frac{r_1}{r_2}}$

Question: 21.057P

Two infinitely long parallel cylinders of diameter D are located a distance s apart from each other. Determine the view factor $F12$ between these two cylinders.

Step 1 of 2



Step 2 of 2

Using cross string method

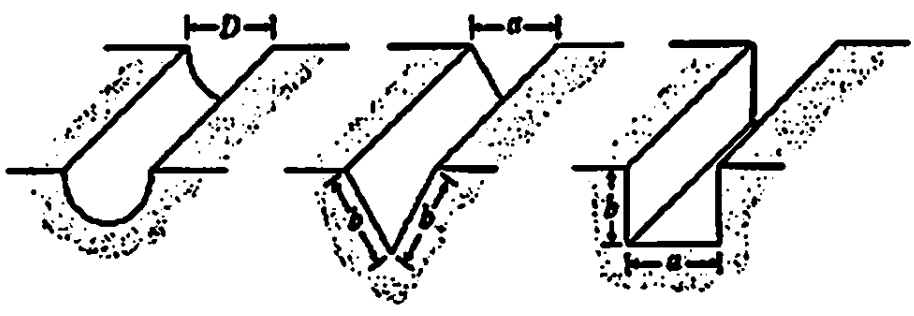
$$F_{1-2} = \frac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{String on surface 1}}$$

$$= \frac{2\sqrt{s^2 + D^2} - 2s}{2(\pi D / 2)}$$

$\therefore$ The view factor between two long cylinders is $\boxed{F_{1-2} = \frac{2\sqrt{s^2 + D^2} - 2s}{\pi D}}$

Question: 21.058P

Determine the view factors from the very long grooves shown in Fig. P21-58 to the surroundings without using any view factor tables or charts. Neglect end effects.

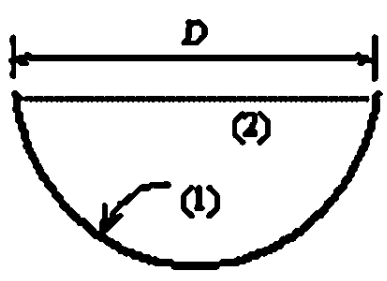


(a) Semicylindrical (b) Triangular groove (c) Rectangular groove

FIGURE P21–58

Step 1 of 7

a)



Step 2 of 7

From the reciprocity rule

$F_{21} + F_{22} = 1$

$F_{22} = 0$

Therefore $F_{21} = 1$

From the reciprocity rule

$A_1 F_{12} = A_2 F_{21}$

$$F_{12} = \frac{A_2}{A_1} F_{21}$$

$$= \frac{D}{\frac{\pi D}{2}} (1)$$

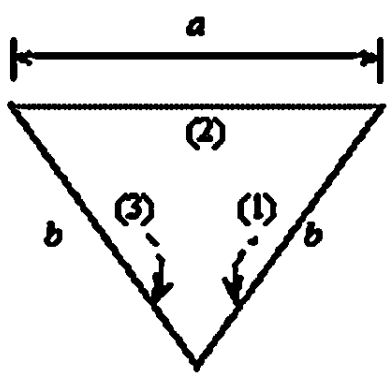
$$= \frac{2}{\pi}$$

$$= 0.636$$

∴ The view factor for the long circular groove is $\boxed{F_{12} = 0.636}$

Step 3 of 7

b)



Step 4 of 7

From the summation rule

$F_{21} + F_{22} + F_{23} = 1$

View factor for the surface itself $F_{22} = 0$

$0 + F_{21} + F_{23} = 1$

$F_{21} = F_{23}$ Because of symmetry

$F_{21} = F_{23} = 0.5$

Step 5 of 7

From the summation rule

$F_{22} + F_{2\to(1+3)} = 1$

$F_{2\to(1+3)} = 1$

From the reciprocity rule

$A_2 F_{2\to(1+3)} = A_{(1+3)} F_{(1+3)\to 2}$

$$F_{(1+3)} = \frac{A_2}{A_{(1+3)}} (1)$$

$$= \frac{a}{2b}$$

∴ The view factor for the long groove is $\boxed{F_{(1+3)} = \frac{a}{2b}}$

Step 6 of 7

c)

Step 7 of 7

From the summation rule

$F_{44} + F_{4\to(1+2+3)} = 1$

View factor for the surface itself $F_{44} = 0$

$F_{4\to(1+2+3)} = 1$

From the reciprocity rule

$A_4 F_{4\to(1+2+3)} = A_{(1+2+3)} F_{(1+2+3)\to 4}$

$$F_{(1+2+3)} = \frac{A_4}{A_{(1+2+3)}} (1)$$

$$= \frac{a}{a+2b}$$

∴ The view factor for the long rectangular groove is $\boxed{F_{(1+2+3)} = \frac{a}{a+2b}}$

Question: 21.059P

A circular cone of diameter D is positioned on a common axis with a circular disk, also of diameter, D, at a distance L, as shown in the figure. With a hypothetical area ($A3$) corresponding to the opening of the cone, determine the values of $F11$ and $F12$ for $L = D = 100$ mm.

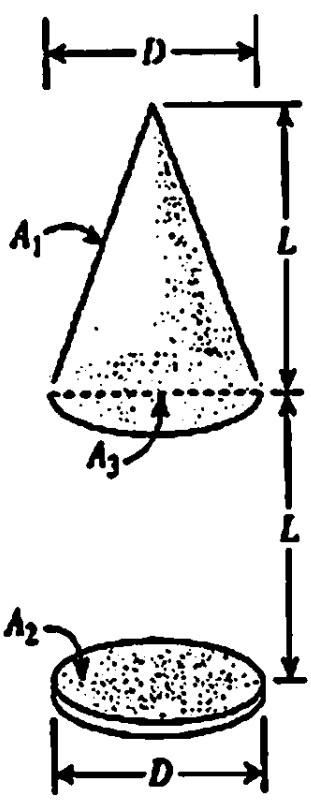


FIGURE P21-59

Step 1 of 5

Consider the surfaces 1 and 3 as enclosure.

So, use summation rule as follows:

$$F_{11} + F_{13} = 1$$
$$F_{11} = 1 - F_{13} \quad \text{..... (1)}$$

Here, F_{11} is the view factor for 1-1 surfaces, and F_{13} is the view factor for 1-3 surfaces.

Apply reciprocity relation between A_1 and A_3 as follows.

$$A_1F_{13} = A_3F_{31}$$
$$F_{13} = \left(\frac{A_3}{A_1}\right)F_{31}$$

Substitute $\left(\frac{A_3}{A_1}\right)F_{31}$ for F_{13} in equation (1).

$$F_{11} = 1 - F_{13}$$
$$F_{11} = 1 - \left(\frac{A_3}{A_1}\right)F_{31} \quad \text{..... (2)}$$
$$= 1 - \left(\frac{7853.89}{17562.0}\right)F_{31}$$
$$F_{11} = 1 - 0.44F_{31}$$

Step 2 of 5

Consider the following relation between view factors.

$$F_{31} + F_{33} = 1$$

Since, $F_{33} = 0$

Substitute 0 for F_{33} .

$$F_{31} + 0 = 1$$
$$F_{31} = 1$$

Substitute 1 for F_{31} in equation (2).

$$F_{11} = 1 - 0.44F_{31}$$
$$= 1 - 0.44$$
$$= 0.55$$

Therefore, the view factor 11 is $\boxed{0.55}$.

Step 3 of 5

The radiation leaving from surface A_2 is intercepted by A_1 and A_2 equally.

$$F_{21} = F_{23}$$

Apply reciprocity relation between A_1 and A_2 as follows.

$$A_1F_{12} = A_2F_{21}$$
$$F_{12} = \left(\frac{A_2}{A_1}\right)F_{21} \quad \text{..... (3)}$$
$$= \left(\frac{7853.89}{17562.0}\right)F_{21}$$
$$F_{12} = 0.44F_{21}$$

Step 4 of 5

Consider that coaxial parallel disks,
Obtain the view factor F_{23} from table 13-1 as follows.

$$F_{23} = \frac{1}{2}\left\{S - \left(S^2 - 4\left(\frac{D_3}{D_2}\right)^2\right)^{\frac{1}{2}}\right\}$$
$$F_{23} = \frac{1}{2}\left(S - \left(S^2 - 4\right)^{\frac{1}{2}}\right)$$

Here, R and S are constants.

Calculate the values of constants as follows:

$$R = \frac{D}{2L}$$
$$= \frac{100}{2(100)}$$
$$= 0.5$$

$$S = 1 + \frac{1 + R^2}{R^2}$$
$$= 2 + \frac{1}{R^2}$$
$$= 2 + 4$$
$$= 6$$

Step 5 of 5

Substitute 0.5 for R and 6 for S .

$$F_{23} = \frac{1}{2}\left(S - \left(S^2 - 4\right)^{\frac{1}{2}}\right)$$
$$= \frac{1}{2}\left(6 - \left(6^2 - 4\right)^{\frac{1}{2}}\right)$$
$$= 0.1716$$

Substitute 0.1716 for F_{23} in equation (3).

$$F_{12} = 0.44F_{21}$$
$$= 0.44 \times (0.1716)$$
$$= 0.0755$$

Therefore, the view factor 11 is $\boxed{0.755}$.

Question: 21.060P

Consider a cylindrical surface and a disk oriented coaxially as shown in the figure. The cylinder has a diameter D and a length L. The disk of diameter D is placed coaxially with the cylinder at a distance L. If $L = 2D$, determine the view factor $F12$ between the cylindrical surface (1) and the disk (2) facing it.

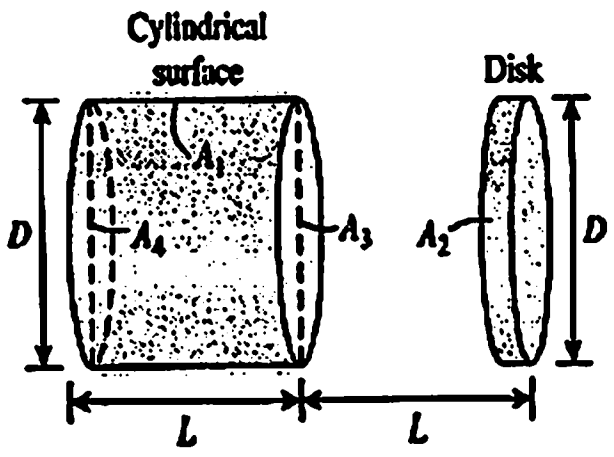


FIGURE P21-60

Step 1 of 5

Given data:

Length of the cylinder, $L = 2D$

Distance between the cylinder and the disc, $L = 2D$

Step 2 of 5

Given diagram:

Step 3 of 5

From the tables of the view factors of finite geometries, the view factor from the disc to the surface A_3 is given by,

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

where,

$$R = \frac{D}{2L}$$
$$= \frac{D}{2 \times 2D}$$
$$= 0.25$$

Substituting the value of R, we have,

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{\left(4 \times 0.25^2\right) + 1}}{2 \times 0.25^2}$$
$$= 0.05573$$

Step 4 of 5

Similarly, the view factor from the surface A_2 to the surface A_4 is,

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

where,

$$R = \frac{D}{2 \times 2L}$$
$$= \frac{D}{2 \times (2 \times 2D)}$$
$$= 0.125$$

By substituting the value of R, we have,

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{\left(4 \times 0.125^2\right) + 1}}{2 \times 0.125^2}$$
$$= 0.015155$$

Step 5 of 5

By using the summation rule, we have,

$$F_{21} + F_{22} + F_{23} + F_{24} = 1$$
$$F_{21} + 0 + 0.05573 + 0.015155 = 1$$
$$F_{21} = 0.93$$

By using the reciprocity rule, the view factor from the cylindrical surface, A_1 to the disc, A_2 can be determined as follows.

$$A_1 F_{12} = A_2 F_{21}$$

Therefore, the view factor F_{12} is given by,

$$F_{12} = \frac{A_2}{A_1} F_{21}$$
$$= \frac{(\pi/4)D^2}{\pi D \times 2D} \times 0.93$$
$$= 0.11625$$
$$F_{12} = 0.11625$$

The view factor F_{12} between the cylindrical surface and the disc facing is $\boxed{F_{12} = 0.11625}$

Question: 21.061P

Repeat Prob. 21-60 with $L = D$.

Step 1 of 5

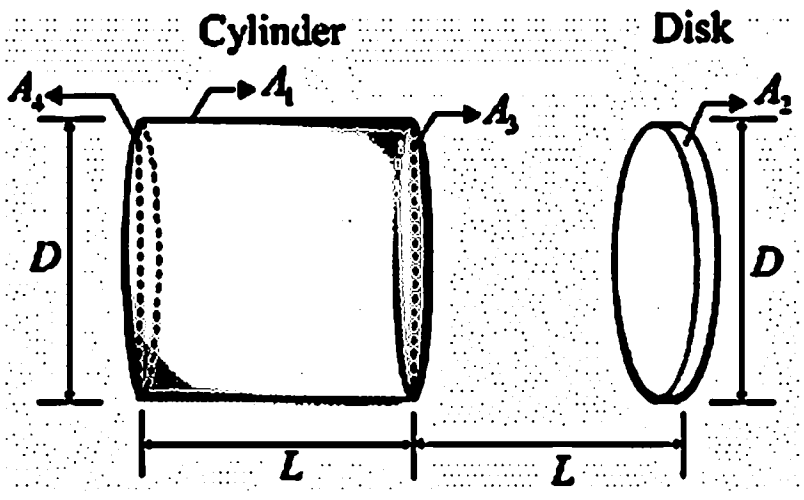
Given data:

Length of the cylinder, $L = 2D$

Distance between the cylinder and the disc, $L = 2D$

Step 2 of 5

Given diagram:



Step 3 of 5

From the tables of the view factors of finite geometries, the view factor from the disc to the surface A_3 is given by,

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

where,

$$R = \frac{D}{2L}$$
$$= \frac{D}{2 \times D}$$
$$= 0.5$$

Substituting the value of R, we have,

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{\left(4 \times 0.5^2\right) + 1}}{2 \times 0.5^2}$$
$$= 0.1715$$

Step 4 of 5

Similarly, the view factor from the surface A_2 to the surface A_4 is,

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

where,

$$R = \frac{D}{2 \times 2L}$$
$$= \frac{D}{2 \times (2 \times D)}$$
$$= 0.25$$

Substituting the value of R, we get

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{\left(4 \times 0.25^2\right) + 1}}{2 \times 0.25^2}$$
$$= 0.05573$$

Step 5 of 5

By using the summation rule, we have,

$$F_{21} + F_{22} + F_{23} + F_{24} = 1$$
$$F_{21} + 0 + 0.1715 + 0.05573 = 1$$
$$F_{21} = 0.772772$$

By using the reciprocity rule, the view factor from the cylindrical surface, A_1 to the disc, A_2 can be determined as follows.

$$A_1 F_{12} = A_2 F_{21}$$

Therefore, the view factor F_{12} is given by,

$$F_{12} = \frac{A_2}{A_1} F_{21}$$
$$= \frac{(\pi/4)D^2}{\pi D \times D} \times 0.772772$$
$$= 0.1932$$

$$\boxed{F_{12} = 0.1932}$$

The view factor F_{12} between the cylindrical surface and the disc facing is $F_{12} = 0.1932$.

Question: 21.062P

What are the two methods used in radiation analysis? How do these two methods differ?

Step 1 of 2

The two methods used in radiation analysis are the matrix and network methods. In matrix method, equations

Surface with specified net heat transfer rate $\dot{Q}_i = A_i \sum_{j=1}^{N} F_{i \to j}(J_i - J_j)$

Surface with specified temperature $\sigma T_i^4 = J_i + \frac{1-\varepsilon_i}{\varepsilon_i} \sum_{j=1}^{N} F_{i \to j}(J_i - J_j)$

Give N linear algebraic equations for the determination of the N unknown radiosities for an N-surface enclosure. Once the radiosities are available, the unknown surface temperatures and heat transfer rates can be determined from these equations respectively. This method involves the use of matrices especially when there are a large number of surfaces. Therefore this method requires some knowledge of linear algebra.

Step 2 of 2

The network method involves drawing a surface resistance associated with each surface of an enclosure and connecting them with space resistances. Then the radiation problem is solved by treating it as an electrical network problem where the radiation heat transfer replaces the current and the radiosity replaces the potential. The network method is not practical for enclosures with more than three or four surfaces due to the increased complexity of the network.

Question: 21.063P

How does radiosity for a surface differ from the emitted energy? For what kind of surfaces are these two quantities identical?

Step 1 of 1

Radiosity is defined as the total radiation energy leaving a surface per unit time per unit area. It is denoted by *J*. Radiosity includes the emitted radiation energy as well as the reflected energy.

Radiosity can be expressed through the following relation:

$$\begin{aligned} J_i &= \left(\text{Radiation emitted by surface } i\right) + \left(\text{Radiation reflected by surface } i\right) \\ &= \varepsilon_i E_{bi} + \rho_i G_i \end{aligned}$$

For a black body, $\left(\text{emissivity}: \varepsilon_i = 1 \text{ and reflectivity}: \rho_i = 0\right)$

Substituting it in the radiosity expression, we obtain

$$\begin{aligned} J_i &= (1) E_{bi} + (0) G_i \\ &= E_{bi} \\ &= \sigma T_i^4 \end{aligned}$$

Radiosity of a blackbody is equal to its emissive power since a blackbody does not reflect any radiation.

Therefore, radiosity and emitted energy are identical for black bodies.

Question: 21.064P

What are the radiation surface and space resistances? How are they expressed? For what kind of surfaces is the radiation surface resistance zero?

Step 1 of 1

During a surface interaction, a surface losses energy by emitting radiation and gains energy by absorbing radiation emitted by other surfaces. The surface resistance to radiation is expressed as

$$R_i = \frac{1-\varepsilon_i}{A_i \varepsilon_i}$$

It is zero for black surfaces.

The space resistance is the radiation resistance between two surfaces and is expressed as

$$R_{i \to j} = \frac{1}{A_i F_{i \to j}}$$

Question: 21.065P

Why is the radiation analysis of enclosures that consist of black surfaces relatively easy? How is the rate of radiation heat transfer between two surfaces expressed in this case?

Step 1 of 1

A blackbody is defined as a perfect emitter and absorber of radiation. Because of the absence of the reflection analysis of radiation exchange between black surfaces is relatively easy. Rate of radiation heat transfer between two surfaces expressed as

$$\dot{Q} = A_1 F_{12} \sigma \left(T_1^4 - T_2^4 \right)$$

Where A_1 is the surface area,
F_{12} is the view factor,
And T_1 and T_2 are the temperatures of two surfaces.

Question: 21.066P

What is a reradiating surface? What simplifications does a reradiating surface offer in the radiation analysis?

Step 1 of 1

A reradiating surface is insulated on one side and all the heat transfer it receives through radiation is emitted by radiation and convection. The net $\dot{Q}_i$ from a reradiating surface is zero. Approximated by surfaces that are well insulated on one side and for which convection is negligible on the opposite (radiating) side

Some surfaces encountered in numerous practical heat transfer applications are modeled as being adiabatic as the back sides of these surfaces are well insulated and net heat transfer through these surfaces is zero. When the convection effects on the front (heat transfer) side of such a surface is negligible and steady-state conditions are reached, the surface must lose as much radiation energy as it receives. Such a surface is called reradiating surface. In radiation analysis, the surface resistance of a reradiating surface is taken to be zero since there is no heat transfer through it.

Setting $\dot{Q}_i = 0$ yields

$$J_i = E_{bi} = \sigma T_i^4 \left(\mathrm{W/m^2}\right)$$

Question: 21.067P pg 1

Two aligned parallel rectangles with dimensions 6 m × 8 m are spaced apart by a distance of 2 m. If the two parallel rectangles are experiencing radiation heat transfer as black surfaces, determine the percentage of change in radiation heat transfer rate when the rectangles are moved 8 m apart.

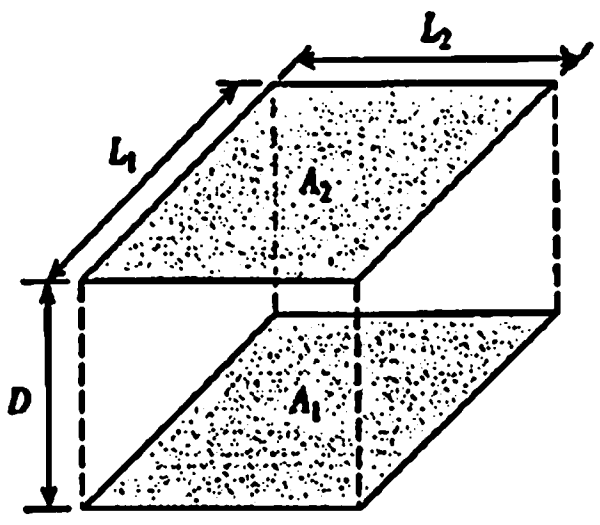
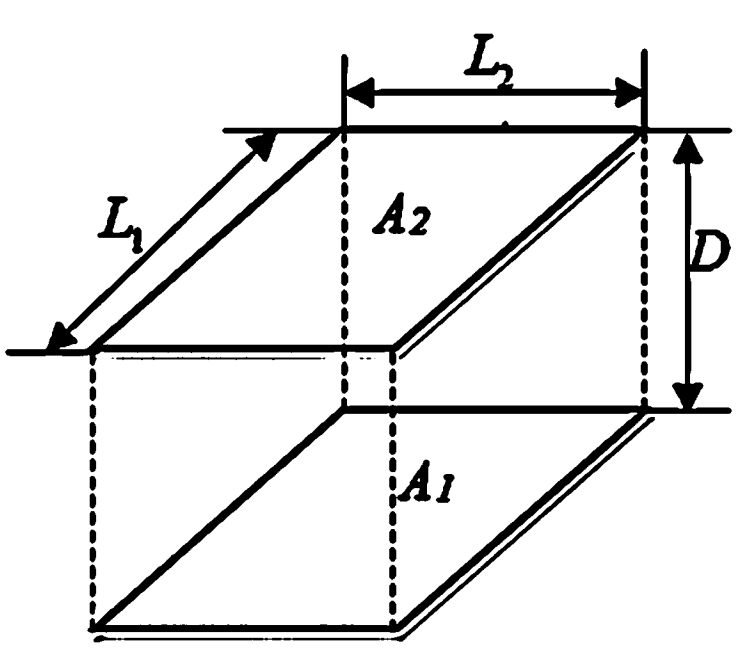


FIGURE P21–67

Step 1 of 7

The surfaces are diffuse emitters and reflectors.



Step 2 of 7

Given data:

Length of the rectangle, $L_1 = 8\,\text{m}$

Width of the rectangle, $L_2 = 6\,\text{m}$

Distance between the two rectangles, $D = 2\,\text{m}$

Find the *L/D* ratio of upper plate.

$$x = \frac{L_2}{D}$$

Here, Length of the rectangle L_1 and Distance between two rectangles is *D*.

Substitute, 8 m for L_1 and 2 m for *D*.

$$\begin{aligned} x &= \frac{L_2}{D} \\ &= \frac{6}{2} \\ &= 3 \end{aligned}$$

Step 3 of 7

Find the *L/D* ratio of upper plate.

$$y = \frac{L_1}{D}$$

Substitute, 8 m for L_1 and 2 m for *D*.

$$\begin{aligned} y &= \frac{L_1}{D} \\ &= \frac{8}{2} \\ &= 4 \end{aligned}$$

Step 4 of 7

Find the view factor of bottom surface to the top surface from the figure.

$$(F_{12})_{D=2\text{m}} = \frac{2}{\pi \times x \times y}\left\{\begin{array}{l} \ln\left(\sqrt{\frac{(1+x^2)(1+y^2)}{1+x^2+y^2}}\right) + x\sqrt{1+y^2}\tan^{-1}\left(\frac{x}{\sqrt{1+y^2}}\right) \\ +y\sqrt{1+x^2}\tan^{-1}\left(\frac{y}{\sqrt{1+x^2}}\right) - x\tan^{-1}(x) - y\tan^{-1}(y) \end{array}\right\}$$

Substitute, 3 for *x* and 4 for *y*.

$$\begin{aligned} (F_{12})_{D=2\text{m}} &= \frac{2}{\pi \times 3 \times 4}\left\{\begin{array}{l} \ln\left(\sqrt{\frac{(1+3^2)(1+4^2)}{1+3^2+4^2}}\right) + 3\sqrt{1+4^2}\tan^{-1}\left(\frac{3}{\sqrt{1+4^2}}\right) \\ +4\sqrt{1+3^2}\tan^{-1}\left(\frac{4}{\sqrt{1+3^2}}\right) - 3\tan^{-1}(3) - 4\tan^{-1}(4) \end{array}\right\} \\ &= \frac{2}{\pi \times 3 \times 4}\left\{\begin{array}{l} \ln(2.557) + 3\sqrt{1+4^2}\tan^{-1}(0.7276) \\ +4\sqrt{1+3^2}\tan^{-1}(1.265) - 3\tan^{-1}(3) - 4\tan^{-1}(4) \end{array}\right\} \\ &= \frac{2}{\pi \times 3 \times 4}(0.9388 + 7.78 + 11.384 - 3.747 - 5.303) \\ &= 0.586 \end{aligned}$$

Find the *L/D* ratio of upper plate.

$$x_2 = \frac{L_2}{D_2}$$

Here, Distance between two rectangles is *D*.

Find the *L/D* ratio of upper plate. 21.067P pg 2

$$x_2 = \frac{L_2}{D_2}$$

Here, Distance between two rectangles is D_2 .

Substitute, 8 m for L_1 and 8 m for *D*.

$$\begin{aligned} x_2 &= \frac{L_2}{D_2} \\ &= \frac{6}{8} \\ &= 0.75 \end{aligned}$$

Step 5 of 7

Find the *L/D* ratio of upper plate.

$$y_2 = \frac{L_1}{D_2}$$

Substitute, 8 m for L_1 and 8 m for D_2 .

$$\begin{aligned} y &= \frac{L_1}{D_2} \\ &= \frac{8}{8} \\ &= 1 \end{aligned}$$

Step 6 of 7

Find the view factor of bottom surface to the top surface from the figure.

$$(F_{12})_{D=8m} = \frac{2}{\pi \times x_2 \times y_2}\left\{\begin{array}{l} \ln\left(\sqrt{\frac{(1+x_2^2)(1+y_2^2)}{1+x_2^2+y_2^2}}\right)+x_2\sqrt{1+y_2^2}\tan^{-1}\left(\frac{x_2}{\sqrt{1+y_2^2}}\right) \\ +y_2\sqrt{1+x_2^2}\tan^{-1}\left(\frac{y_2}{\sqrt{1+x_2^2}}\right)-x_2\tan^{-1}(x_2)-y\tan^{-1}(y_2) \end{array}\right\}$$

Substitute, 0.75 for x_2 and 1 for y_2.

$$\begin{aligned} (F_{12}')_{D=8m} &= \frac{2}{\pi \times 0.75 \times 1}\left\{\begin{array}{l} \ln\left(\sqrt{\frac{(1+0.75^2)(1+1^2)}{1+0.75^2+1^2}}\right)+0.75\sqrt{1+1^2}\tan^{-1}\left(\frac{0.75}{\sqrt{1+1^2}}\right) \\ +1\sqrt{1+0.75^2}\tan^{-1}\left(\frac{1}{\sqrt{1+0.75^2}}\right)-0.75\tan^{-1}(0.75)-1\tan^{-1}(1) \end{array}\right\} \\ &= \frac{2}{\pi \times 0.75 \times 1}(0.099+0.5171+0.8895-0.9316-0.785) \\ &= 0.17 \end{aligned}$$

Step 7 of 7

Find the percentage change in radiation heat transfer rate can be determined using the radiation equation.

$$\dot{Q} = F_{12} A\sigma\left(T_1^4 - T_2^4\right)$$

Since the surface area, Stefan Boltzmann constant and the temperatures are remains same, the change in radiation heat transfer rate is mainly due to change in shape factor.

$$\begin{aligned} \%\text{change} &= \frac{\left(F_{12} - F_{12}'\right)}{F_{12}} \times 100 \\ &= \frac{(0.6-0.17)}{0.6} \times 100 \\ &= 71.6\% \end{aligned}$$

Due to increase in distance between the two surfaces from 2 to 8 m, the view factor has been reduced from 0.6 to 0.17 and hence the radiation heat transfer between two surfaces reduced by **71.6%**

Question: 21.068P

A solid sphere of 1 m diameter at 600 K is kept in an evacuated, long, equilateral triangular enclosure whose sides are 2 m long. The emissivity of the sphere is 0.45 and the temperature of the enclosure is 420 K. If heat is generated uniformly within the sphere at a rate of 3100 W, determine (*a*) the view factor from the enclosure to the sphere and (*b*) the emissivity of the enclosure.

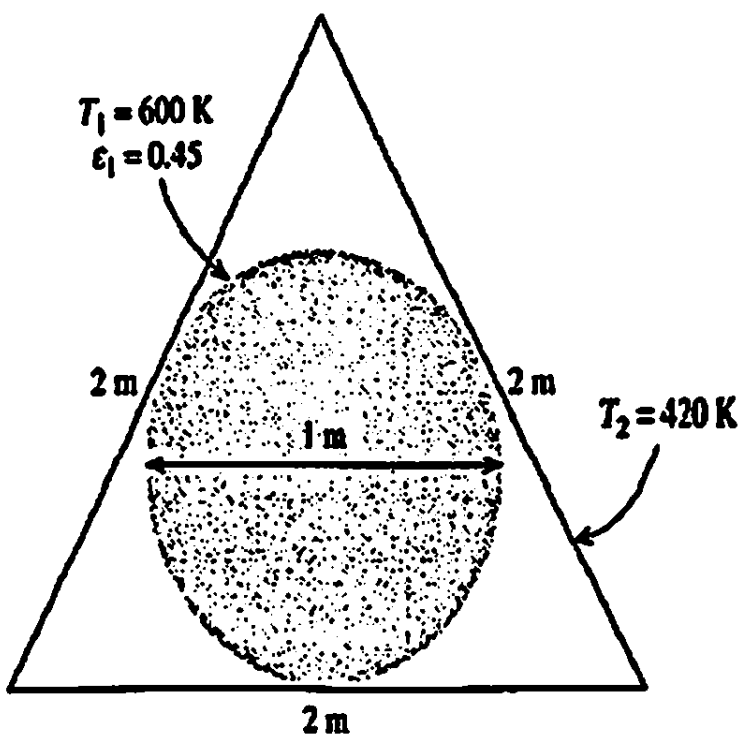
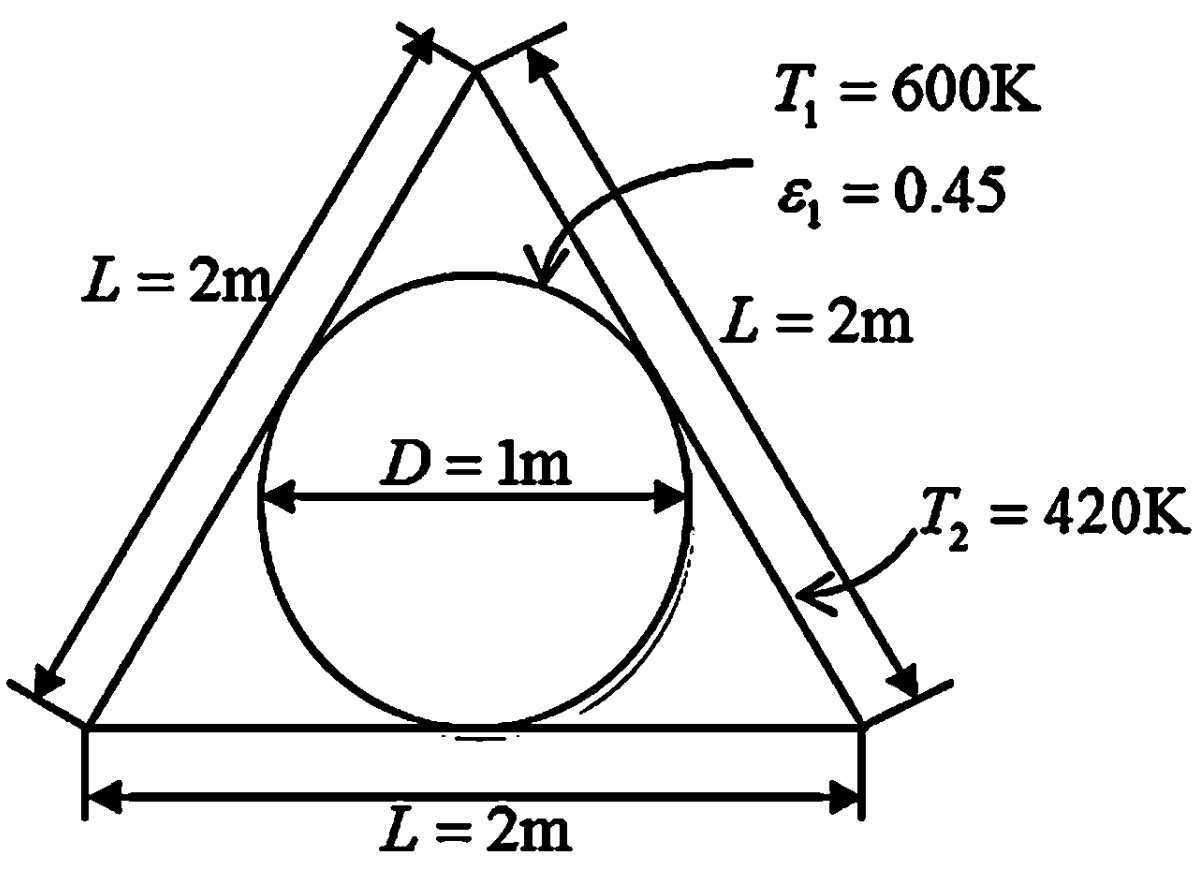


FIGURE P21–68

Step 1 of 3

Figure shows a solid sphere is kept in evacuated equilateral triangle enclosure:



Step 2 of 3

(a)

Find the cross sectional area sphere surface 1.

$$A_1 = \pi D^2$$

Here, D is the diameter of the sphere.

Substitute 1 m for D.

$$\begin{aligned} A_1 &= \pi D^2 \\ &= \pi \times 1^2 \\ &= 3.141\ \text{m}^2 \end{aligned}$$

Find the cross sectional area of equilateral triangular enclosure is 2.

$$A_2 = 3 \times \frac{L}{2}\sqrt{L^2 - D^2}$$

Here, L is the side of equilateral triangular enclosure.

Substitute 2 m for L.

$$\begin{aligned} A_2 &= 3 \times \frac{L}{2}\sqrt{L^2 - D^2} \\ &= 3 \times \frac{2}{2}\sqrt{2^2 - 1^2} \\ &= 5.196\ \text{m}^2 \end{aligned}$$

Since, the view factor for sphere surface with respect to equilateral triangular enclosure surface is unity because sphere is enclosed in equilateral triangular enclosure.

$$F_{12} = 1$$

Find the view factor for equilateral triangular enclosure 2 with respect to a sphere surface 1 by using the reciprocity relation.

$$\begin{aligned} A_1 F_{12} &= A_2 F_{21} \\ F_{21} &= \frac{A_1 F_{12}}{A_2} \end{aligned}$$

Substitute 1 for F_{12}, $5.196\ \text{m}^2$ for A_2, and $3.141\ \text{m}^2$ for A_1.

$$\begin{aligned} F_{21} &= \frac{A_1 F_{12}}{A_2} \\ &= \frac{3.141 \times 1}{5.196} \\ &= 0.604 \end{aligned}$$

Therefore, the view factor for equilateral triangular enclosure 2 with respect to a sphere surface 1 is $\boxed{0.604}$.

Step 3 of 3

(b)

Find the emissivity of the enclosure using net rate of radiation heat transfer for the two-surface enclosure.

$$\dot{Q} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

Here, T_2 is temperature of the enclosure, T_1 is the temperature of the sphere, ε_1 is emissivity of sphere, ε_2 is emissivity triangular, σ is Stefan Boltzmann constant and $\dot{Q}$ is net radiation heat transfer.

Substitute $5.67 \times 10^{-8} \times \text{W/m}^2 \cdot \text{K}^4$ for σ, 600 K for T_1, 420 K for T_2, 0.604 for F_{21}, ε_1

$$\dot{Q} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$3100\,\text{W} = \frac{5.67 \times 10^{-8} \times \left(600^4 - 420^4\right)}{\frac{1-0.45}{3.141 \times 0.45} + \frac{1}{3.141 \times 1} + \frac{1-\varepsilon_2}{5.196 \times \varepsilon_2}}$$

$$3100 = \frac{5583.98}{0.389 + 0.3183 + \left(\frac{1-\varepsilon_2}{5.196\varepsilon_2}\right)}$$

$$\frac{1-\varepsilon_2}{5.196\varepsilon_2} = 1.09398$$

$$\varepsilon_2 = 0.15$$

Therefore, the emissivity of the enclosure is $\boxed{0.15}$.

Question:

Consider a 10-ft ×10-ft ×10-ft cubical furnace whose top and side surfaces closely approximate black surfaces and whose base surface has an emissivity ε = 0.4. The base, top, and side surfaces of the furnace are maintained at uniform temperatures of 800 R, 1600 R, and 2400 R, respectively. Determine the net rate of radiation heat transfer between (*a*) the base and the side surfaces and (*b*) the base and the top surfaces. Also, determine the net rate of radiation heat transfer to the base surface.

Step 1 of 11

Given data:

Size of the cubical furnace, $L = 10\,\text{ft}$

Emissivity of the base surface, $\varepsilon_1 = 0.4$

Emissivity of the top surface, $\varepsilon_2 = 1$

Emissivity of the side surfaces, $\varepsilon_3 = 1$

Temperature of the base surface, $T_1 = 800\,\text{R}$

Temperature of the top surface, $T_2 = 1600\,\text{R}$

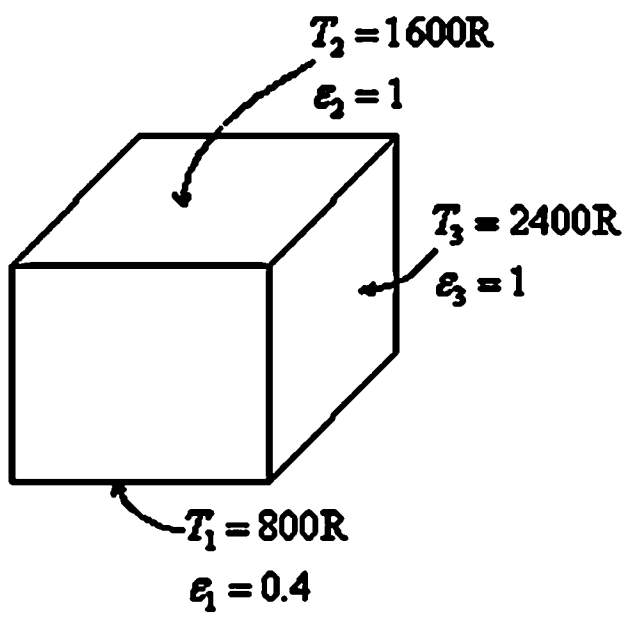
Temperature of the side surfaces, $T_3 = 2400\,\text{R}$

Step 2 of 11

Assumptions:

1. The surfaces are diffuse, gray and opaque
2. Steady operating conditions exist
3. Heat transfer from and to the surfaces is only due to Radiation

Step 3 of 11



Step 4 of 11

Consider the base surface to be surface 1, the top surface to be surface 2 and the side surfaces to be surface 3. The cubical furnace can be considered to be three-surface enclosure. The areas and blackbody emissive powers of surfaces can be calculated as

$$A_1 = 10\times10\,(\text{ft})^2$$
$$= 100\,\text{ft}^2$$
$$A_2 = 10\times10\,(\text{ft})^2$$
$$= 100\,\text{ft}^2$$
$$A_3 = 4\times10\times10\,(\text{ft})^2$$
$$= 400\,\text{ft}^2$$

Step 5 of 11

Emissive powers on each surface

$$E_{b1} = \sigma T_1^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times800^4\,(\text{R})^4$$
$$= 702.0544\,\text{Btu/h-ft}^2$$
$$E_{b2} = \sigma T_2^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times1600^4\,(\text{R})^4$$
$$= 11232.8704\,\text{Btu/h-ft}^2$$
$$E_{b3} = \sigma T_3^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times2400^4\,(\text{R})^4$$
$$= 56866.4064\,\text{Btu/h-ft}^2$$

Step 6 of 11

From figure, the view factor from the base to the top surface of the cube can be determined as

$$\frac{L_1}{D} = \frac{10(\text{ft})}{10(\text{ft})} = 1$$
$$\frac{L_2}{D} = \frac{10(\text{ft})}{10(\text{ft})} = 1$$
$$F_{12} = 0.2$$

By using the summation rule, the view factor from the base or top to the side surfaces can be determined as

$$F_{11} + F_{12} + F_{13} = 1$$

Since the base surface is flat and thus the view factor $F_{11} = 0$

Therefore, by rewriting the above equation we will get

$$F_{13} = 1 - F_{11} - F_{12}$$
$$= 1 - 0 - 0.2$$
$$= 0.8$$

Step 7 of 11

In order to calculate the radiation heat transfer, we need to determine the thermal resistances and these can be calculated as

$$R_1 = \frac{1-\varepsilon_1}{A_1\varepsilon_1}$$
$$= \frac{1-0.4}{100(\text{ft}^2)\times0.4}$$
$$= 0.015\,\text{ft}^{-2}$$
$$R_{12} = \frac{1}{A_1F_{12}}$$

$$= \frac{1}{100(\text{ft}^2)\times 0.4}$$

$$= 0.015\text{ft}^{-2}$$

$$R_{12} = \frac{1}{A_1 F_{12}}$$

$$= \frac{1}{100(\text{ft}^2)\times 0.2}$$

$$= 0.05\text{ft}^{-2}$$

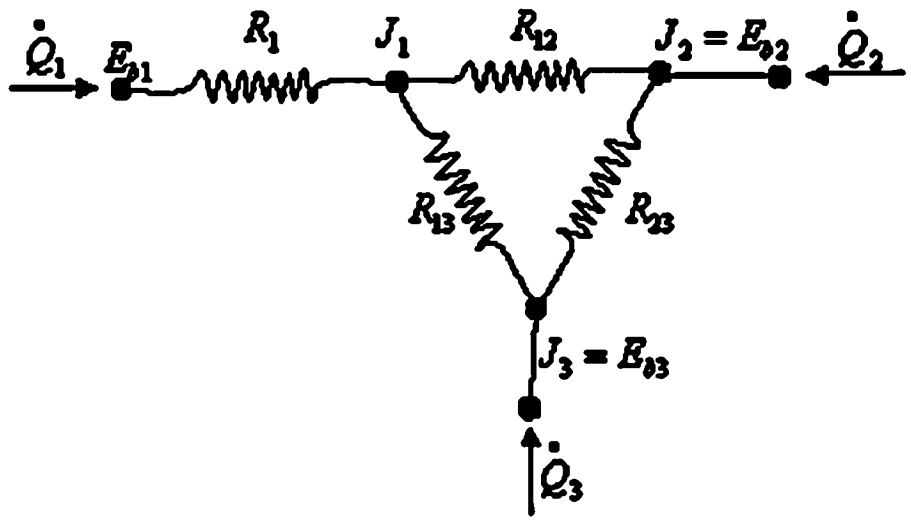
$$R_{13} = \frac{1}{A_1 F_{13}}$$

$$= \frac{1}{100(\text{ft}^2)\times 0.8}$$

$$= 0.0125\text{ft}^{-2}$$

Step 8 of 11

Since the side and the top surfaces are black, and thus their radiosities are equal to their emissive powers. The radiosity of the base surface is determined by using electrical network analogy that the algebraic sum of the currents (net radiation heat transfer) at each node must be equal to zero.



$$\frac{E_{b1}-J_1}{R_1}+\frac{E_{b2}-J_1}{R_{12}}+\frac{E_{b3}-J_1}{R_{13}}=0$$

$$\frac{702.0544-J_1}{0.015}+\frac{11232.8704-J_1}{0.05}+\frac{56866.4064-J_1}{0.0125}=0$$

$$J_1 = 28924.6413\text{Btu/h-ft}^2$$

Step 9 of 11

a) The net rate of radiation heat transfer between the base and the side surfaces can be determined as

$$\dot{Q}_{31} = \frac{E_{b3}-J_1}{R_{13}}$$

$$= \frac{56866.4064\,(\text{Btu/h-ft}^2)-28924.6413(\text{Btu/h-ft}^2)}{0.0125(\text{ft}^{-2})}$$

$$= 2.235\times 10^6\text{Btu/h}$$

$$\boxed{\dot{Q}_{31} = 2.235\times 10^6\text{Btu/h}}$$

Step 10 of 11

b) The net rate of radiation heat transfer between the base and the top surfaces is

$$\dot{Q}_{12} = \frac{J_1-E_{b2}}{R_{12}}$$

$$= \frac{28924.6413(\text{Btu/h-ft}^2)-11232.8704\,(\text{Btu/h-ft}^2)}{0.05\,(\text{ft}^{-2})}$$

$$= 3.538\times 10^5\text{Btu/h}$$

$$\boxed{\dot{Q}_{12} = 3.538\times 10^5\text{Btu/h}}$$

Step 11 of 11

c) The net rate of radiation heat transfer to the base surface can be determined as

$$\dot{Q}_1 = \dot{Q}_{12}+\dot{Q}_{13}$$

$$= 3.538\times 10^5(\text{Btu/h})+2.235\times 10^6\,(\text{Btu/h})$$

$$= 2.589\times 10^6\text{Btu/h}$$

$$\boxed{\dot{Q}_1 = 2.589\times 10^6\text{Btu/h}}$$

Question:

Reconsider Prob. 21-69E. Using EES (or other) software, investigate the effect of base surface emissivity on the net rates of radiation heat transfer between the base and the side surfaces, between the base and top surfaces, and to the base surface. Let the emissivity vary from 0.1 to 0.9. Plot the rates of heat transfer as a function of emissivity, and discuss the results.

Step 1 of 13

Given data:

Size of the cubical furnace, $L = 10\,\text{ft}$

Emissivity of the base surface, $\varepsilon_1 = 0.4$

Emissivity of the top surface, $\varepsilon_2 = 1$

Emissivity of the side surfaces, $\varepsilon_3 = 1$

Temperature of the base surface, $T_1 = 800\,\text{R}$

Temperature of the top surface, $T_2 = 1600\,\text{R}$

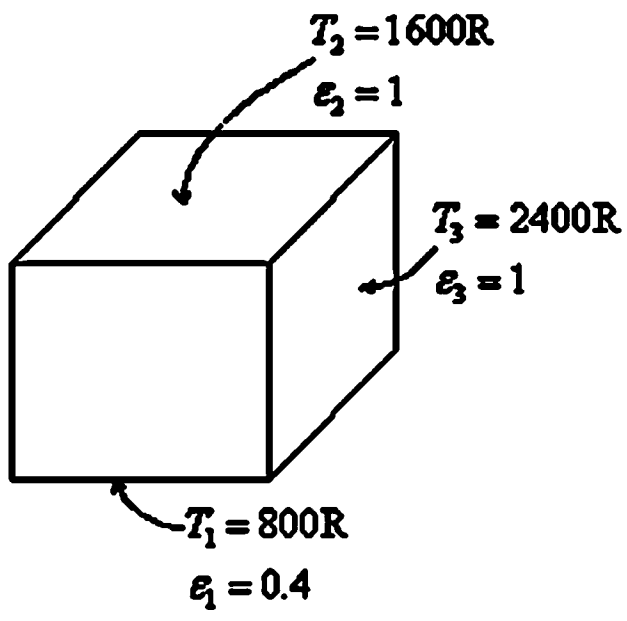
Temperature of the side surfaces, $T_3 = 2400\,\text{R}$

Step 2 of 13

Assumptions:

1. The surfaces are diffuse, gray and opaque
2. Steady operating conditions exist
3. Heat transfer from and to the surfaces is only due to Radiation

Step 3 of 13



Step 4 of 13

Consider the base surface to be surface 1, the top surface to be surface 2 and the side surfaces to be surface 3. The cubical furnace can be considered to be three-surface enclosure. The areas and blackbody emissive powers of surfaces can be calculated as

$$A_1 = 10\times10\,(\text{ft})^2$$
$$= 100\,\text{ft}^2$$
$$A_2 = 10\times10\,(\text{ft})^2$$
$$= 100\,\text{ft}^2$$
$$A_3 = 4\times10\times10\,(\text{ft})^2$$
$$= 400\,\text{ft}^2$$

Step 5 of 13

Emissive powers on each surface

$$E_{b1} = \sigma T_1^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times800^4\,(\text{R})^4$$
$$= 702.0544\,\text{Btu/h-ft}^2$$
$$E_{b2} = \sigma T_2^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times1600^4\,(\text{R})^4$$
$$= 11232.8704\,\text{Btu/h-ft}^2$$
$$E_{b3} = \sigma T_3^4$$
$$= 0.1714\times10^{-8}\,(\text{Btu/h-ft}^2\text{-R}^4)\times2400^4\,(\text{R})^4$$
$$= 56866.4064\,\text{Btu/h-ft}^2$$

Step 6 of 13

From figure, the view factor from the base to the top surface of the cube can be determined as

$$\frac{L_1}{D} = \frac{10(\text{ft})}{10(\text{ft})} = 1$$
$$\frac{L_2}{D} = \frac{10(\text{ft})}{10(\text{ft})} = 1$$
$$F_{12} = 0.2$$

By using the summation rule, the view factor from the base or top to the side surfaces can be determined as

$$F_{11} + F_{12} + F_{13} = 1$$

Since the base surface is flat and thus the view factor $F_{11} = 0$

Therefore, by rewriting the above equation we will get

$$F_{13} = 1 - F_{11} - F_{12}$$
$$= 1 - 0 - 0.2$$
$$= 0.8$$

Step 7 of 13

In order to calculate the radiation heat transfer, we need to determine the thermal resistances and these can be calculated as

$$R_1 = \frac{1-\varepsilon_1}{A_1\varepsilon_1}$$
$$= \frac{1-\varepsilon_1}{100(\text{ft}^2)\times\varepsilon_1}$$
$$R_{12} = \frac{1}{A_1 F_{12}}$$

$$= \frac{1-\varepsilon_1}{100(\text{ft}^2)\times\varepsilon_1}$$

$$R_{12} = \frac{1}{A_1 F_{12}}$$

$$= \frac{1}{100(\text{ft}^2)\times 0.2}$$

$$= 0.05\text{ft}^{-2}$$

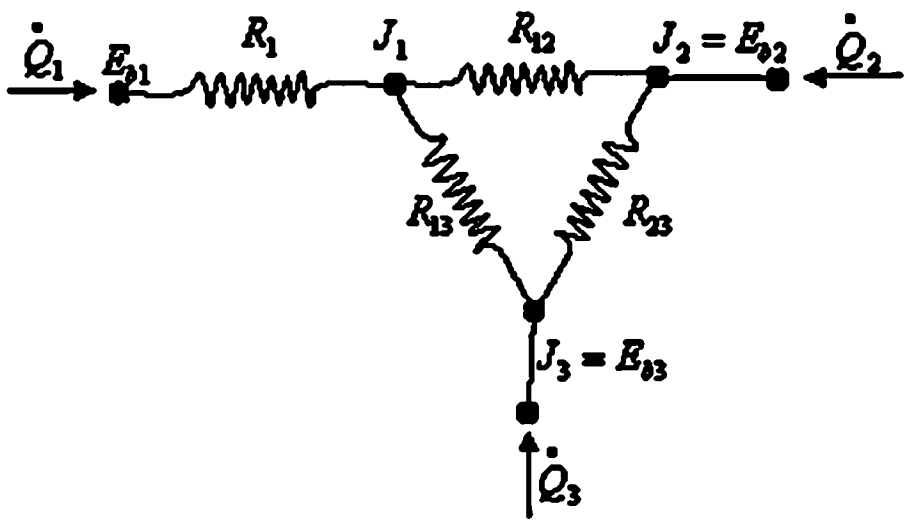
$$R_{13} = \frac{1}{A_1 F_{13}}$$

$$= \frac{1}{100(\text{ft}^2)\times 0.8}$$

$$= 0.0125\text{ft}^{-2}$$

Step 8 of 13

Since the side and the top surfaces are black, and thus their radiosities are equal to their emissive powers. The radiosity of the base surface is determined by using electrical network analogy that the algebraic sum of the currents (net radiation heat transfer) at each node must be equal to zero.



$$\frac{E_{b1}-J_1}{R_1}+\frac{E_{b2}-J_1}{R_{12}}+\frac{E_{b3}-J_1}{R_{13}}=0$$

$$\frac{702.0544-J_1}{R_1}+\frac{11232.8704-J_1}{0.05}+\frac{56866.4064-J_1}{0.0125}=0$$

By varying the emissivity of the bas surface, the thermal resistance of the base surface will vary and hence the radiosity of the base surface.

Step 9 of 13

a) The net rate of radiation heat transfer between the base and the side surfaces can be expressed as

$$\dot{Q}_{31} = \frac{E_{b3}-J_1}{R_{13}}$$

$$= \frac{56866.4064\,(\text{Btu/h-ft}^2)-J_1\,(\text{Btu/h-ft}^2)}{0.0125(\text{ft}^2)}$$

Step 10 of 13

b) The net rate of radiation heat transfer between the base and the top surfaces can be expressed as

$$\dot{Q}_{12} = \frac{J_1-E_{b2}}{R_{12}}$$

$$= \frac{J_1\,(\text{Btu/h-ft}^2)-11232.8704\,(\text{Btu/h-ft}^2)}{0.05\,(\text{ft}^2)}$$

Step 11 of 13

c) The net rate of radiation heat transfer to the base surface can be determined as

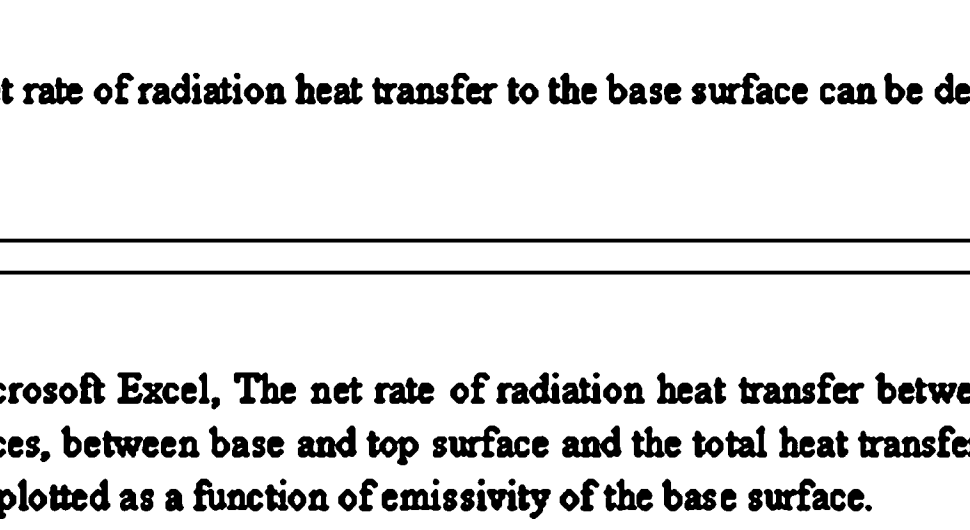
$$\dot{Q}_1 = \dot{Q}_{12}+\dot{Q}_{13}$$

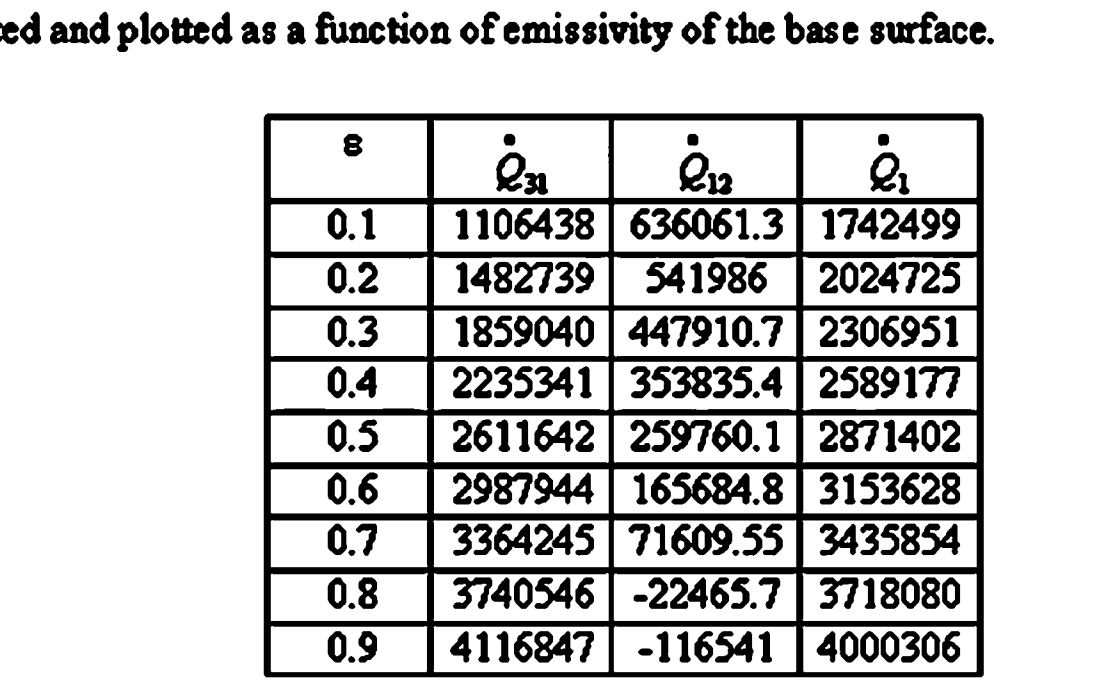
Step 12 of 13

Using the Microsoft Excel, The net rate of radiation heat transfer between the base and the side surfaces, between base and top surface and the total heat transfer to the base are tabulated and plotted as a function of emissivity of the base surface.

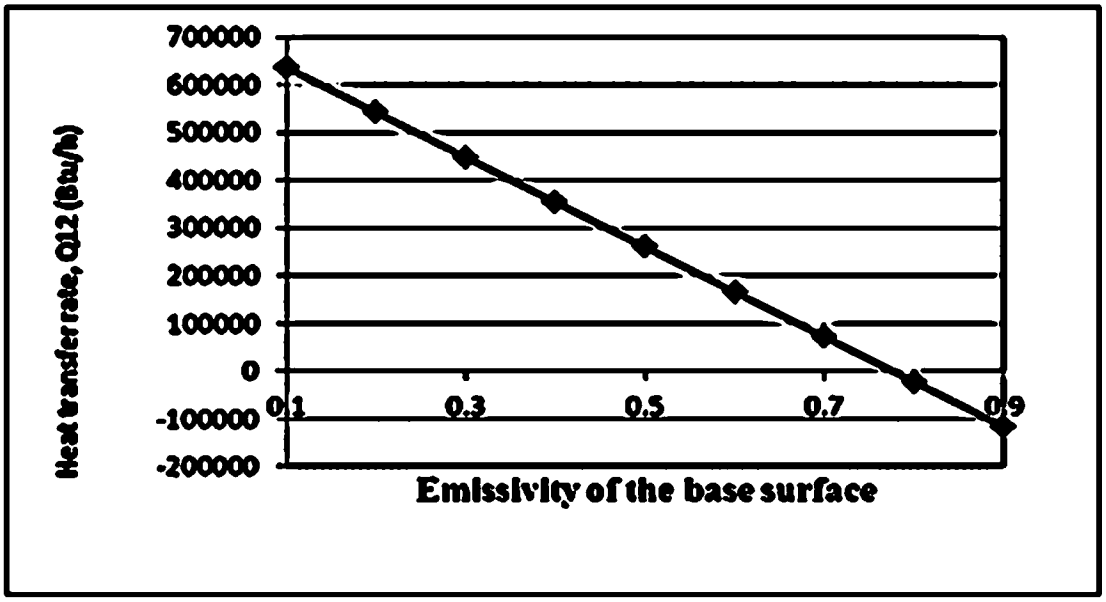
ε	$\dot{Q}_{31}$	$\dot{Q}_{12}$	$\dot{Q}_1$
0.1	1106438	636061.3	1742499
0.2	1482739	541986	2024725
0.3	1859040	447910.7	2306951
0.4	2235341	353835.4	2589177
0.5	2611642	259760.1	2871402
0.6	2987944	165684.8	3153628
0.7	3364245	71609.55	3435854
0.8	3740546	-22465.7	3718080
0.9	4116847	-116541	4000306

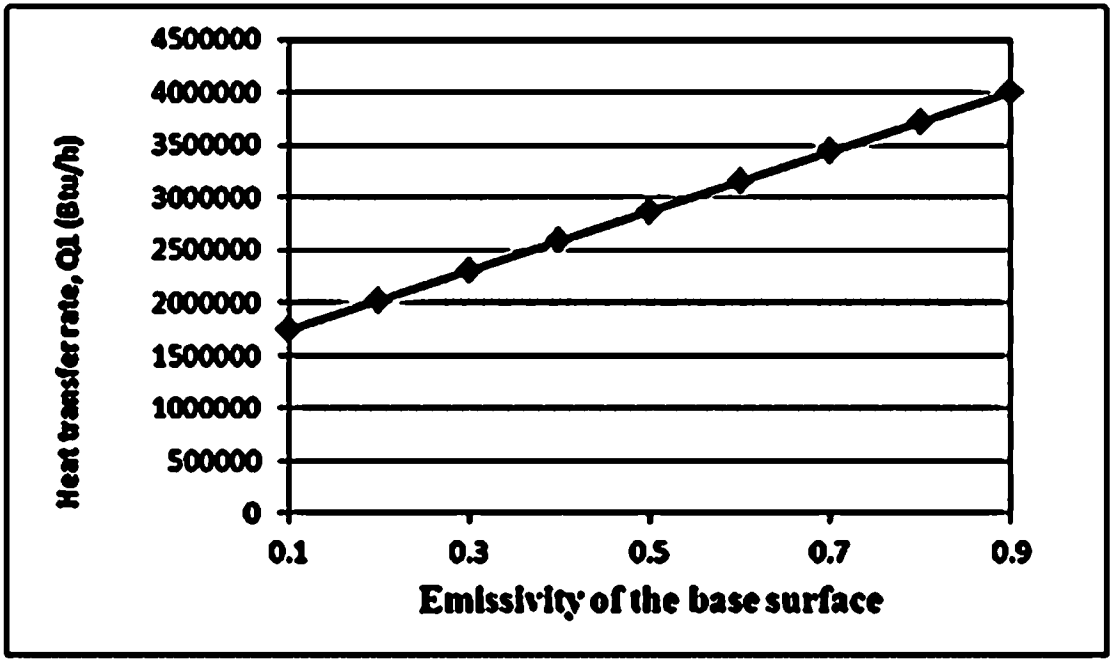
Step 13 of 13





Emissivity of the base surface





By increasing the emissivity of the base surface, the net rate of heat transfer between base and side surface increases, where as between base and top surface it is decreases and the total net rate of heat transfer to the base is increases.

Question: 21.071P

Two very large parallel plates are maintained at uniform temperatures of $T1 = 600$ K and $T2 = 400$ K and have emissivities $\varepsilon 1 = 0.5$ and $\varepsilon 2 = 0.9$, respectively. Determine the net rate of radiation heat transfer between the two surfaces per unit area of the plates.

Step 1 of 2

Given

Two plates maintained at temperatures $T_1 = 600\ K$

and $T_2 = 400\ \text{K}$

Emissivity of the two plates was $\varepsilon_1 = 0.5$

$\varepsilon_2 = 0.9$

Step 2 of 2

The net rate of radiation heat transfer is calculated as

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

$$= \frac{5.67 \times 10^{-8} \times \left(600^4 - 400^4\right)}{\frac{1}{0.5} + \frac{1}{0.9} - 1}$$

$$= 2793.2\ \text{W/m}^2$$

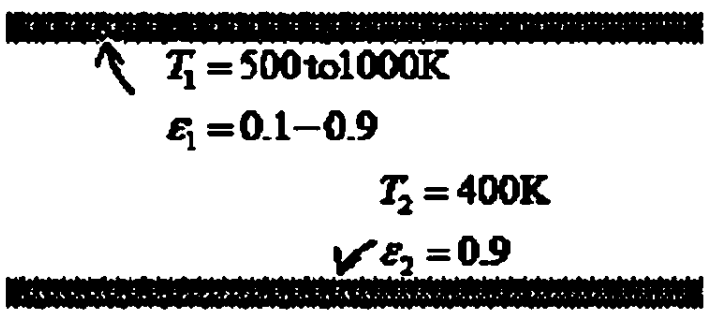
Net rate of radiation heat transfer per unit area, $\boxed{\dot{Q}_{12} = 2793.2\ \text{W/m}^2}$

Question: 21.072P

Reconsider Prob. 21-71. Using EES (or other) software, investigate the effects of the temperature and the emissivity of the hot plate on the net rate of radiation heat transfer between the plates. Let the temperature vary from 500 K to 1000 K and the emissivity from 0.1 to 0.9. Plot the net rate of radiation heat transfer as functions of temperature and emissivity, and discuss the results.

Step 1 of 6

The surfaces are diffuse, gray and opaque. Steady operating conditions exist. Heat transfer from and to the surfaces is only due to radiation.



Step 2 of 6

The net rate of radiation heat transfer between the two surfaces per unit area of the plates is determined by the expression,

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

T_1 is the temperature of the top plate

T_2 is the temperature of the bottom plate

ε_1 is the emissivity of the top plate

ε_2 is the emissivity of the bottom plate

Now, substitute 400K for T_2 , 0.9 for ε_2

Then the expression becomes,

$$\dot{Q}_{12} = \frac{5.67\times10^{-8}\,(\text{W/m}^2\cdot\text{K}^4)\times(T_1^4 - 400^4)(\text{K}^4)}{\frac{1}{\varepsilon_1} + \frac{1}{0.9} - 1}$$

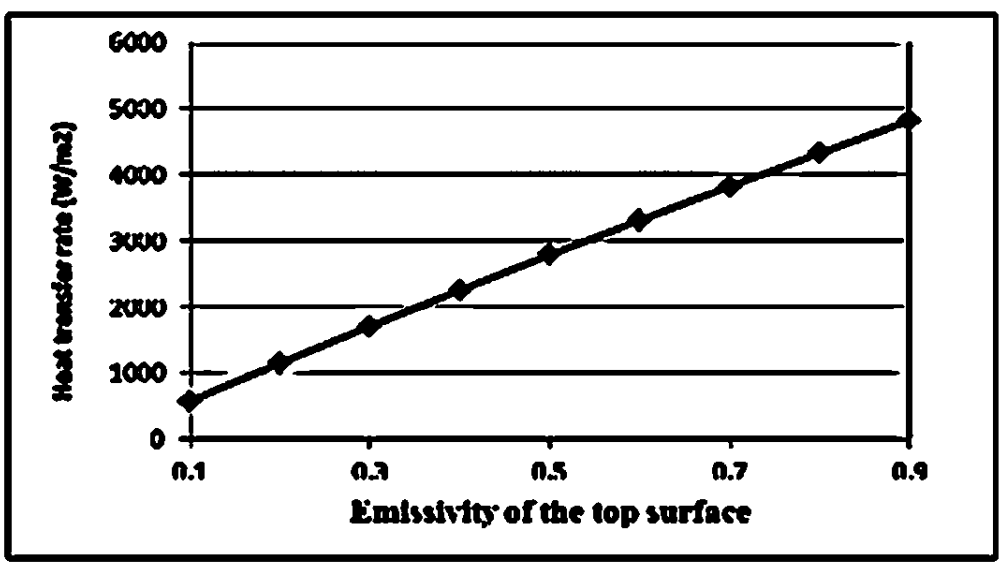
Step 3 of 6

By varying the emissivity of the top plate varied from 0.1 and 0.9, the EES software is developed the table for ε_1 corresponding to the $\dot{Q}_{12}$ values that are shown below,

ε_1	$\dot{Q}_{12}$
0.1	583.2
0.2	1153.722
0.3	1711.974
0.4	2258.349
0.5	2793.221
0.6	3316.95
0.7	3829.88
0.8	4332.343
0.9	4824.655

Step 4 of 6

The EES software is developing the graph between $\dot{Q}_{12}$ and ε_1 is as shown below,



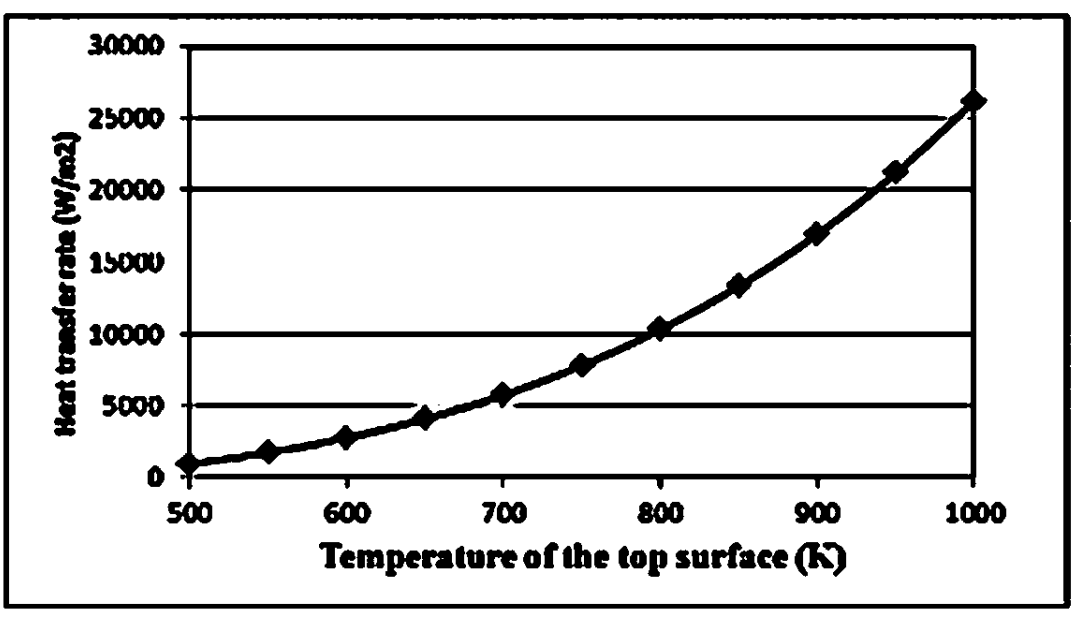
Step 5 of 6

When the temperature varied from 500 K and 1000 K, the EES software is developing the table for T corresponding to the $\dot{Q}_{12}$ values is as shown below,

T	$\dot{Q}_{12}$
500	991.0563
550	1770.103
600	2793.221
650	4106.74
700	5761.018
750	7810.444
800	10313.43
850	13332.43
900	16933.9
950	21188.36
1000	26170.33

Step 6 of 6

The EES software is developing the graph between $\dot{Q}_{12}$ and T is as shown below,



Question: 21.073P

Two infinitely long parallel plates of width w are located at w distance apart, as shown in the figure. The two plates behave as black surfaces, where surface $A1$ has a temperature of 700 K and surface $A2$ has a temperature of 300 K. Determine the radiation heat flux between the two surfaces.

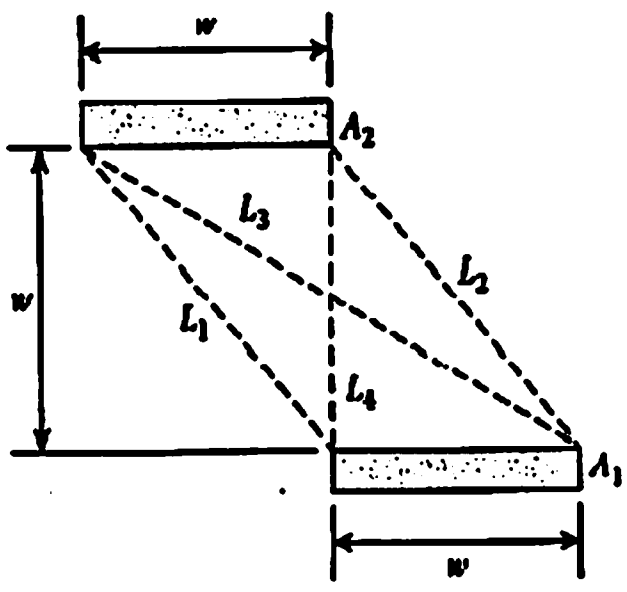
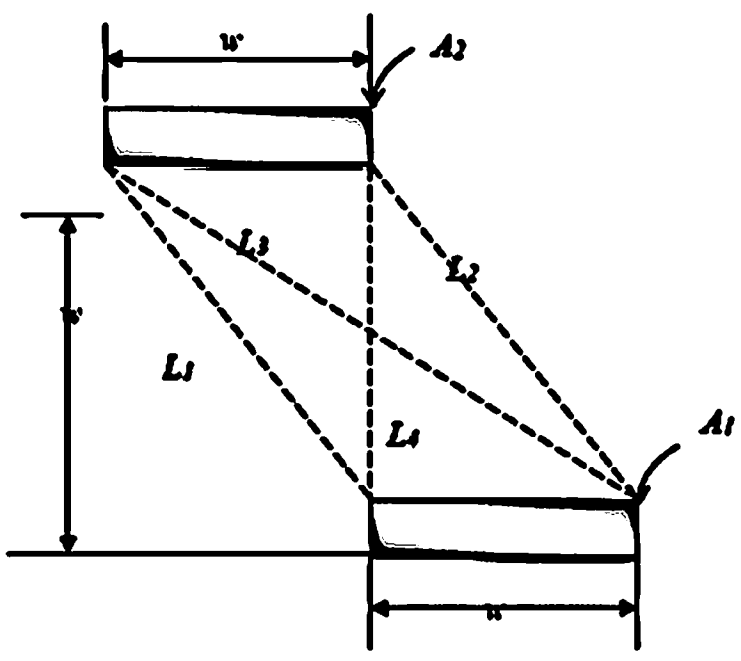


FIGURE P21–73

Step 1 of 4

Draw the schematic diagram of long parallel plates.



Step 2 of 4

Use Pythagoras theorem to calculate the crossed and uncrossed strings lengths.

$$L_4 = w$$

$$L_2 = \sqrt{L_4^2 + w^2}$$
$$= \sqrt{w^2 + w^2}$$
$$= \sqrt{2}w$$

$$L_1 = \sqrt{L_4^2 + w^2}$$
$$= \sqrt{w^2 + w^2}$$
$$= \sqrt{2}w$$

$$L_3 = \sqrt{(2w)^2 + (w)^2}$$
$$= \sqrt{5}w$$

Step 3 of 4

The view factor from the base plate to the top plate can be calculate by using the Hottel's cross-strings method.

$$F_{12} = \frac{\sum(\text{Crossed strings}) - \sum(\text{Uncrossed strings})}{2 \times (\text{String on surface 1})}$$
$$= \frac{(L_3 + L_4) - (L_1 + L_2)}{2w}$$

$$F_{12} = \frac{(L_3 + L_4) - (L_1 + L_2)}{2w}$$
$$= \frac{\left(\sqrt{5}w + w\right) - \left(\sqrt{2}w + \sqrt{2}w\right)}{2w}$$
$$= \frac{w\left[\left(\sqrt{5}+1\right) - \left(\sqrt{2}+\sqrt{2}\right)\right]}{2w}$$
$$= 0.204$$

Step 4 of 4

The net rate of heat flux $\dot{q}_{1\to 2}$ from surface 1 to surface 2 can be Calculate using the radiation equation.

$$\dot{q}_{1\to 2} = F_{1\to 2}\sigma\left(T_1^4 - T_2^4\right)$$

Here, $F_{1\to 2}$ is the view factor, σ is the Stefan Boltzmann constant, T_1 is the Temperature of the surface A_1 and T_2 is the temperature of the surface A_2.

Substitute 0.204 for $F_{1\to 2}$, $5.67\times10^{-8}\ \text{W/m}^2\text{-K}^4$ for σ, 700K for T_1 and 300K for T_2.

$$\dot{q}_{1\to 2} = F_{1\to 2}\sigma\left(T_1^4 - T_2^4\right)$$
$$= 0.204\times5.67\times10^{-8}(\text{W/m}^2\text{-K}^4)\times\left[(700\text{K})^4 - (300\text{K})^4\right]$$
$$= 2683.5\ \text{W/m}^2$$

Therefore, the net rate of heat flux is $\boxed{2683.5\ \text{W/m}^2}$.

Question: 21.074P

A dryer is shaped like a long semi-cylindrical duct of diameter 1.5 m. The base of the dryer is occupied with water soaked materials to be dried. The base is maintained at

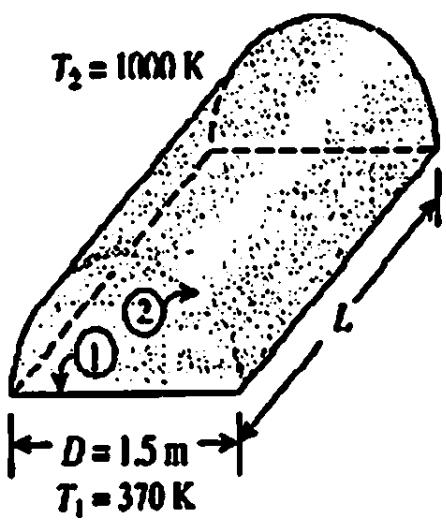


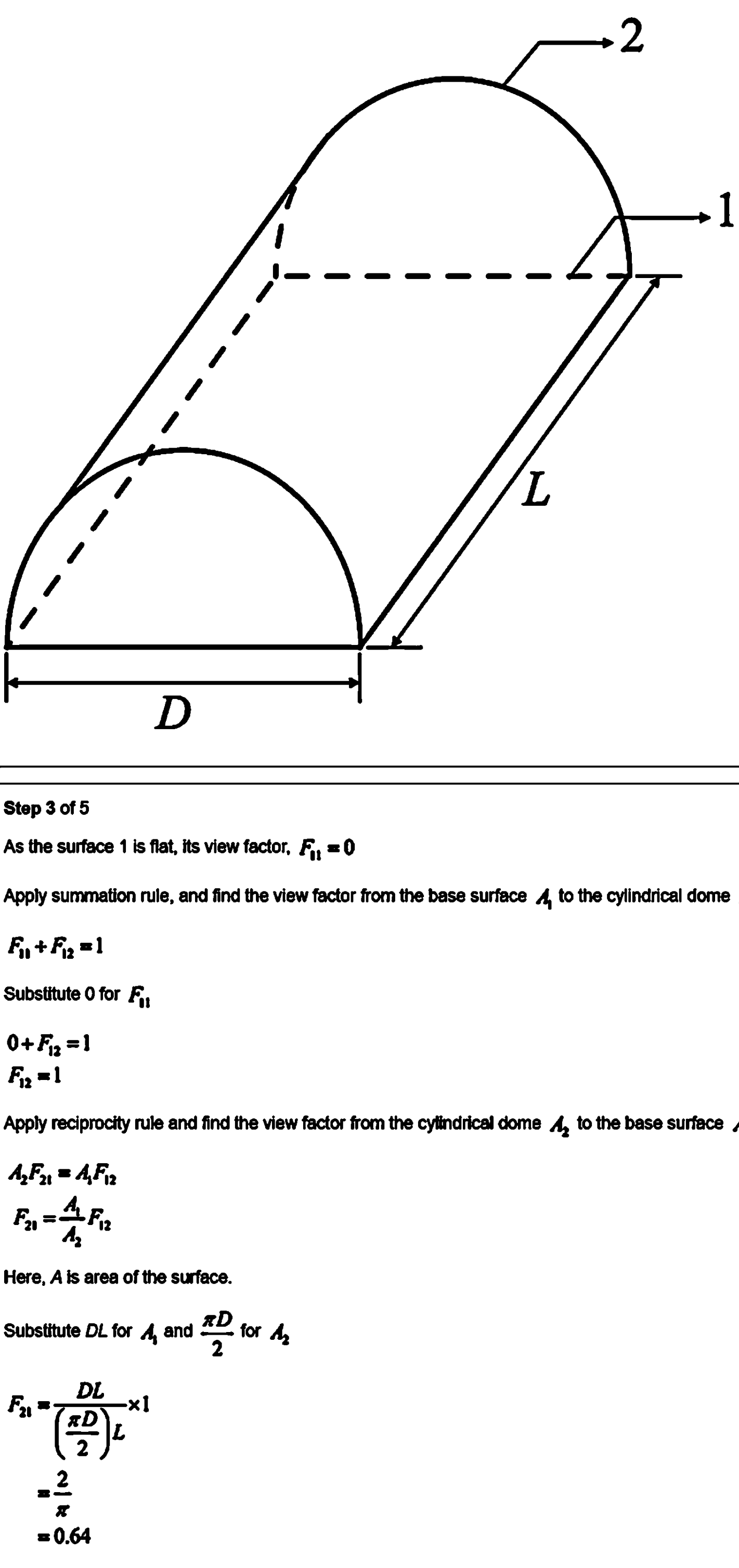
FIGURE P21–74

a temperature of 370 K, while the dome of the dryer is maintained at 1000 K. If both surfaces behave as blackbody, determine the drying rate per unit length experienced by the wet materials.

Step 1 of 5

Draw the semi cylindrical duct as shown:

Step 2 of 5



Step 3 of 5

As the surface 1 is flat, its view factor, $F_{11} = 0$

Apply summation rule, and find the view factor from the base surface A_1 to the cylindrical dome A_2

$$F_{11} + F_{12} = 1$$

Substitute 0 for F_{11}

$$0 + F_{12} = 1$$
$$F_{12} = 1$$

Apply reciprocity rule and find the view factor from the cylindrical dome A_2 to the base surface A_1.

$$A_2 F_{21} = A_1 F_{12}$$
$$F_{21} = \frac{A_1}{A_2} F_{12}$$

Here, *A* is area of the surface.

Substitute *DL* for A_1 and $\frac{\pi D}{2}$ for A_2

$$F_{21} = \frac{DL}{\left(\frac{\pi D}{2}\right)L} \times 1$$
$$= \frac{2}{\pi}$$
$$= 0.64$$

Step 4 of 5

Apply energy balance equation to the dryer.

$$\dot{Q}_{21} = \dot{Q}_{evap}$$

Calculate the radiation heat exchange between the dome and the base surface per unit length by using the following equation:

$$\dot{Q}_{21} = F_{21} A_2 \sigma \left(T_2^4 - T_1^4\right)$$
$$\dot{Q}_{21} = F_{21} \times \frac{\pi D}{2} \sigma \left(T_2^4 - T_1^4\right)$$

Here, σ is Stefan Boltzmann's constant, T_1 is base temperature and T_2 is temperature of the dome.

Substitute 0.64 for F_{21}, 1.5 m for *D*, $5.67 \times 10^{-8}\ \text{W/m}^2.\text{K}^4$ for σ, 370 K for T_1, and 1000 K for T_2

$$\dot{Q}_{21} = 0.64 \times \left(\frac{\pi}{2} \times 1.5\right) \times 5.67 \times 10^{-8} \times \left(1000^4 - 370^4\right)$$
$$= 83899.15\ \text{W/m}$$

Step 5 of 5

Find the mass flow rate of water by using amount of heat required for drying.

$$\dot{Q}_{21} = \dot{Q}_{evap}$$
$$\dot{Q}_{evap} = \dot{m} h_{fg}$$

Here, h_{fg} is enthalpy of vaporization and $\dot{m}$ is mass flow rate of water.

Substitute $83899.15\ \text{W/m}$ for $\dot{Q}_{evap}$ and $2257 \times 10^3\ \text{J/kg}$ for h_{fg}

$$83899.15 = \dot{m} \times 2257 \times 10^3$$
$$\dot{m} = 0.0371\ \text{kg/s.m}$$

Therefore, the drying rate per unit length is $\boxed{0.0371\ \text{kg/s} \cdot \text{m}}$.

Question: 21.075P

A furnace is of cylindrical shape with $R = H = 3$ m. The base, top, and side surfaces of the furnace are all black and are maintained at uniform temperatures of 500,700, and 1400 K, respectively. Determine the net rate of radiation heat transfer to or from the top surface during steady operation.

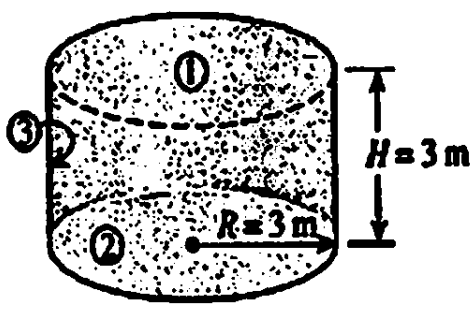


FIGURE P21–75

Step 1 of 6

Given data:

Radius of the furnace, $R = 3\,\text{m}$

Height of the furnace, $H = 3\text{m}$

Emissivity of the all furnace surfaces, $\varepsilon = 1$

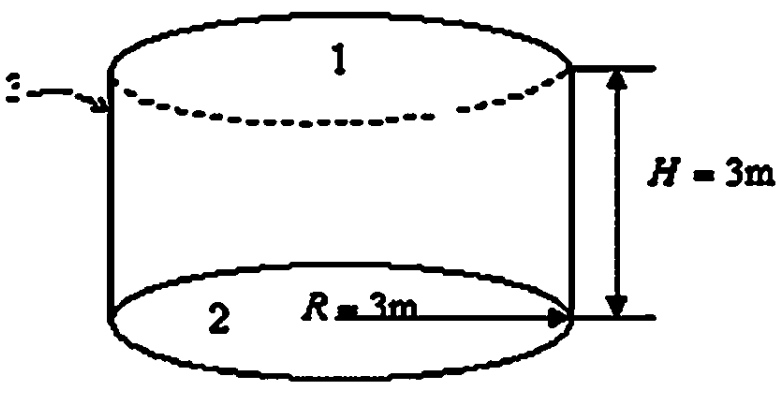
Temperature of the base surface, $T_2 = 500\,\text{K}$

Temperature of the top surface, $T_1 = 700\,\text{K}$

Temperature of the side surface, $T_3 = 1400\,\text{K}$

Step 2 of 6

Given figure:



Step 3 of 6

Consider the top surface to be surface 1, the base surface to be surface 2 and the side surface to be surface 3.

The area of the top surface is given by,

$$A_1 = \pi R^2$$
$$= \pi \times 3^2$$
$$= 28.27\,\text{m}^2$$

From the figure, the view factor from the base to the top surface is determined as

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

where,

$$R = \frac{L_1}{R_1}$$
$$= \frac{3}{3}$$
$$= 1$$

Substituting the value of R, we have,

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{\left(4 \times 1^2\right) + 1}}{2 \times 1^2}$$
$$F_{12} = 0.38$$

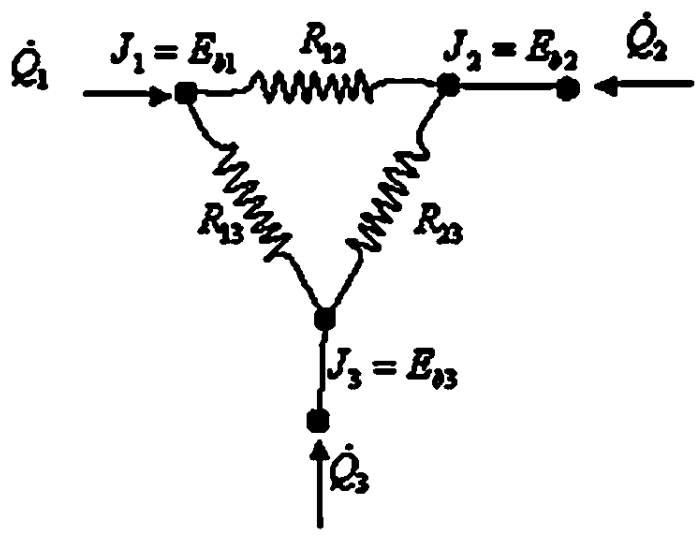
Step 4 of 6

The view factor from the base to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1$$
$$0 + 0.38 + F_{13} = 1$$
$$F_{13} = 1 - 0.38$$
$$= 0.62$$

Step 5 of 6

The cylindrical furnace can be considered to be three-surface enclosure. We assume that steady-state conditions exist. Since all the surfaces are black, the radiosities are equal to the emissive powers of surfaces, and the net rate of radiation heat transfer from the top surface can be determined by using electrical network analogy that the algebraic sum of the currents (net radiation heat transfer) at each node must be equal to zero.



Step 6 of 6

The net rate of radiation heat transfer from the top surface can be expressed as,

$$\dot{Q} = A_1 F_{12} \sigma\left(T_1^4 - T_2^4\right) + A_1 F_{13} \sigma\left(T_1^4 - T_3^4\right)$$
$$= 28.27\left(\text{m}^2\right) \times 0.38 \times 5.67 \times 10^{-8}\left(\text{W/m}^2 \cdot \text{K}^4\right)\left(700^4 - 500^4\right)\left(\text{K}^4\right)$$
$$+ 28.27\left(\text{m}^2\right) \times 0.62 \times 5.67 \times 10^{-8}\left(\text{W/m}^2 \cdot \text{K}^4\right)\left(700^4 - 1400^4\right)\left(\text{K}^4\right)$$
$$= -3471\,\text{kW}$$

$$\boxed{\dot{Q} = -3471\,\text{kW}}$$

The negative sign indicates that the net heat transfer is towards the top surface.

Therefore, the net rate of radiation heat transfer to the top surface is determined to be 3471 kW.

Question: 21.076P

A 9-ft-high room with a base area of 12 ft × 12 ft is to be heated by electric resistance heaters placed on the ceiling, which is maintained at a uniform temperature of 90°F at all times. The floor of the room is at 65°F and has an emissivity of 0.8. The side surfaces are well insulated. Treating the ceiling as a blackbody, determine the rate of heat loss from the room through the floor.

Step 1 of 7

Given data:

Base area of the room, $A_2 = A_1 = 12\text{ft} \times 12\text{ft}$

Height of the room, $D = 9\text{ft}$

Emissivity of the base surface, $\varepsilon_2 = 0.8$

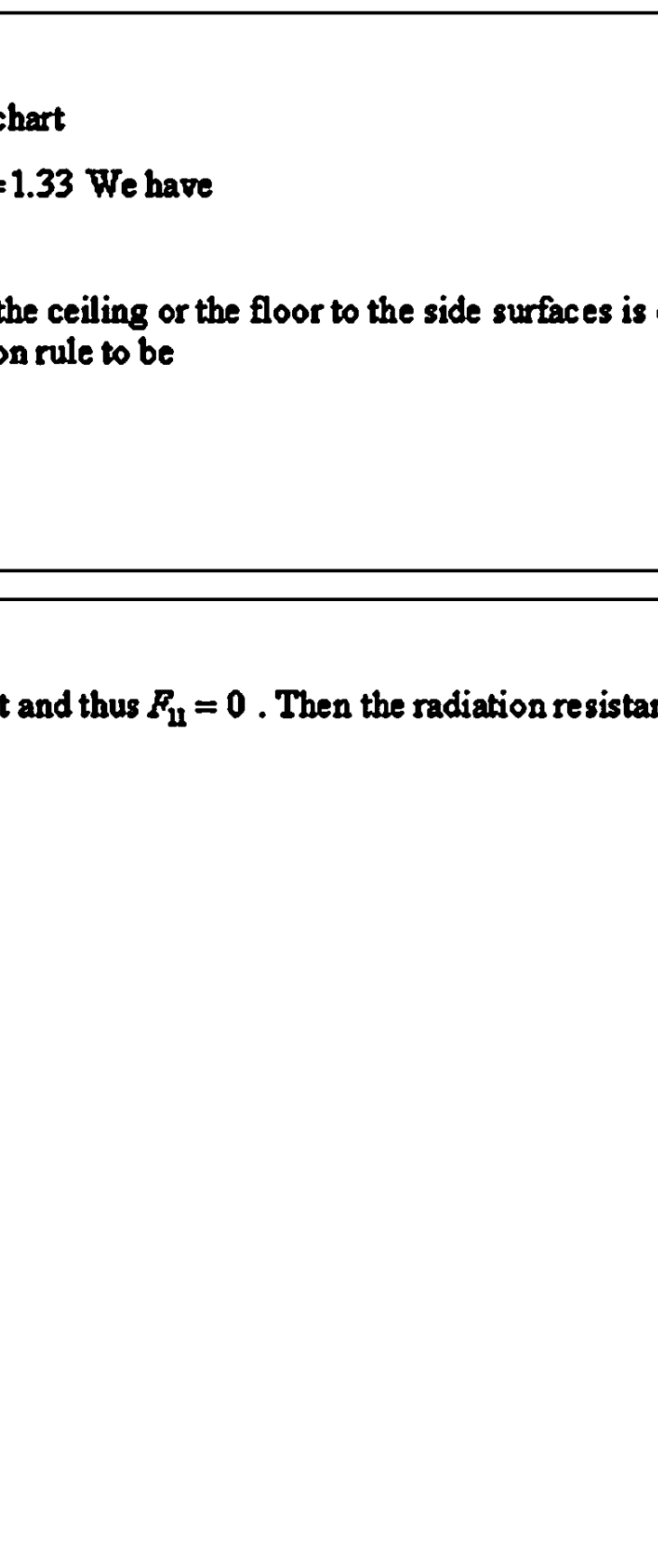
Emissivity of the top surface, $\varepsilon_1 = 1$

Temperature of the base surface, $T_2 = 65\,^{\circ}\text{F}$

Temperature of the top surface, $T_1 = 90^{\circ}\text{F}$

Step 2 of 7

Ceiling temperature and floor temperature of the room is as shown in figure:



Step 3 of 7

Consider the base surface to be surface 1, the top surface to be surface 2 and the side surfaces to be surface 3. The furnace can be considered to be three-surface enclosure. The areas and blackbody emissive powers of surfaces can be calculated as

$$A_1 = 12 \times 12$$
$$= 144\text{ft}^2$$
$$A_2 = 12 \times 12$$
$$= 144\text{ft}^2$$

From the figure, the view factor from the floor to the ceiling of the room is

$$\frac{L_1}{D} = \frac{12}{9}$$
$$= 1.33$$
$$\frac{L_2}{D} = \frac{12}{9}$$
$$= 1.33$$

Step 4 of 7

From the view factor chart

At $\frac{L_1}{D} = 1.33$ and $\frac{L_2}{D} = 1.33$ We have

$$F_{12} = 0.27$$

The view factor from the ceiling or the floor to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1$$
$$0 + 0.27 + F_{13} = 1$$
$$F_{13} = 0.73$$

Step 5 of 7

Since the ceiling is flat and thus $F_{11} = 0$. Then the radiation resistances are calculated as

$$R_2 = \frac{1-\varepsilon_2}{A_2\varepsilon_2}$$
$$= \frac{1-0.8}{144 \times 0.8}$$
$$= 0.001736\,\text{ft}^{-2}$$
$$R_{12} = \frac{1}{A_1 F_{12}}$$
$$= \frac{1}{144 \times 0.27}$$
$$= 0.0257\text{ft}^{-2}$$
$$R_{13} = R_{23}$$
$$= \frac{1}{A_1 F_{13}}$$
$$= \frac{1}{144 \times 0.73}$$
$$= 0.009513\text{ft}^{-2}$$

Step 6 of 7

The emissive powers of the ceiling and base surface are calculated as

$$E_{b1} = \sigma T_1^4$$
$$= 0.1714 \times 10^{-8} \times 550^4$$
$$= 156.84\ \text{Btu/h} \cdot \text{ft}^2$$
$$E_{b2} = \sigma T_2^4$$
$$= 0.1714 \times 10^{-8} \times 525^4$$
$$= 130.211\text{Btu/h} \cdot \text{ft}^2$$

Step 7 of 7

Since the side surfaces are reradiating, there is no heat transfer through them, and the entire heat lost by the ceiling must be gained by the floor. Then the rate of heat loss from the room through its floor can be determined as

$$\dot{Q}_{12} = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}}\right)^{-1} + R_2}$$
$$= \frac{156.84 - 130.211}{\left(\frac{1}{0.0257} + \frac{1}{2 \times 0.009513}\right)^{-1} + 0.001736}$$
$$= 2102\ \text{Btu/h}$$

Therefore, the rate of heat loss from the floor is $\boxed{\dot{Q}_{12} = 2102\text{Btu/h}}$.

Question: 21.077P

Two very long concentric cylinders of diameters $D1 = 0.35$ m and $D2 = 0.5$ m are maintained at uniform temperatures of $T1 = 950$ K and $T2 = 500$ K and have emissivities $\varepsilon 1 = 1$ and $\varepsilon 2 = 0.55$, respectively. Determine the net rate of radiation heat transfer between the two cylinders per unit length of the cylinders.

Step 1 of 3

Given data:

Diameter of inner cylinder, $D_1 = 0.35$ m

Diameter of outer cylinder, $D_2 = 0.5$ m

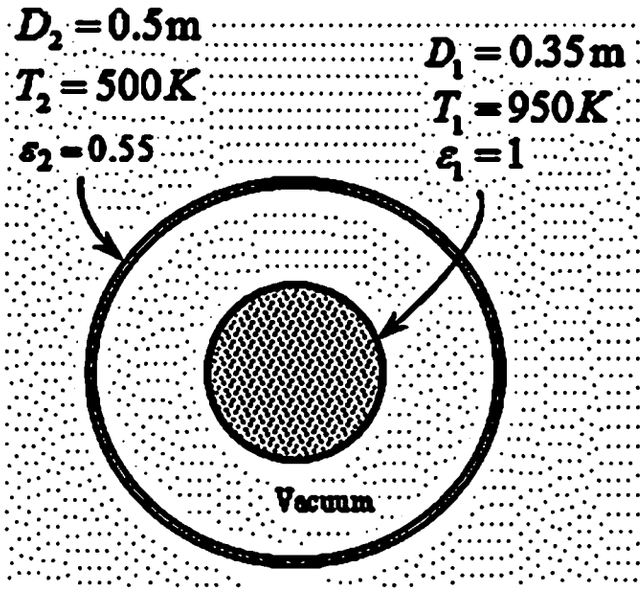
Temperature of inner cylinder, $T_1 = 950\ K$

Temperature of outer cylinder, $T_2 = 500$ K

Emissivities of the two cylinders are $\varepsilon_1 = 1$

And $\varepsilon_2 = 0.55$

Step 2 of 3



Step 3 of 3

Rate of radiation heat transfer between two cylinders per unit length is

$$\dot{Q}_{12} = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \left(\frac{1-\varepsilon_2}{\varepsilon_2}\right)\left(\frac{r_1}{r_2}\right)}$$

$$\frac{\dot{Q}_{12}}{L} = \frac{\pi \times 0.35 \times 5.67 \times 10^{-8} \times \left(950^4 - 500^4\right)}{\frac{1}{1} + \left(\frac{1-0.55}{0.55}\right)\left(\frac{0.35}{0.5}\right)}$$

$$= \frac{46883.76}{1.572}$$

$$= 29810.48\ \text{W}$$

Net rate radiation heat transfer per unit length, $\boxed{\dot{Q} = 29.81\ \text{kW}}$

Question: 21.078P pg 1

This question deals with steady-state radiation heat transfer between a sphere ($r1$ = 30 cm) and a circular disk ($r2$ = 120 cm), which are separated by a center-to-center distance h = 60 cm. When the normal to the center of disk passes through the center of the sphere, the radiation view factor is given by

$$F_{12} = 0.5\left\{1 - \left[1 + \left(\frac{r_2}{h}\right)^2\right]^{-0.5}\right\}$$

Surface temperatures of the sphere and the disk are 600°C and 200°C, respectively; and their emissivities are 0.9 and 0.5, respectively.

(*a*) Calculate the view factors *F*12 and *F*2l.

(*b*) Calculate the net rate of radiation heat exchange between the sphere and the disk.

(*c*) For the given radii and temperatures of the sphere and the disk, the following four possible modifications could increase the net rate of radiation heat exchange: paint each of the two surfaces to alter their emissivities, adjust the distance between them, and provide an (refractory) enclosure. Calculate the net rate of radiation heat exchange between the two bodies if the *best values* are selected for each of the above modifications.

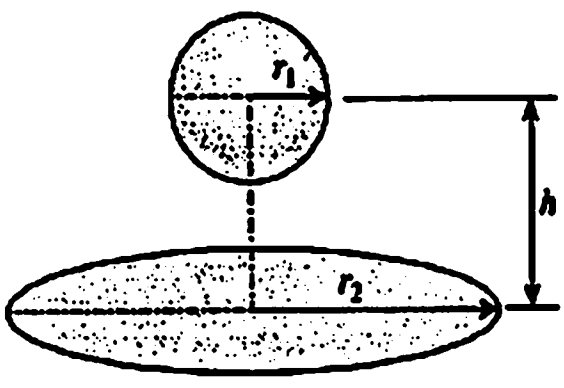


FIGURE P21–78

Step 1 of 10

(a) The radiation view factor from surface 1 to surface 2 is given as,

$$F_{12} = 0.5\left\{1-\left[1+\left(\frac{r_2}{h}\right)^2\right]^{-0.5}\right\}$$

Radius of disc is r_2

Center to center distance is h

Now, Substitute 120 cm for r_2 and 60 cm for h

$$F_{12} = 0.5\left\{1-\left[1+\left(\frac{120}{60}\right)^2\right]^{-0.5}\right\}$$
$$= 0.5\times 0.5527$$
$$= 0.27635$$

Therefore, the radiation view factor from surface 1 to surface 2 is $\boxed{F_{12} = 0.2764}$

Step 2 of 10

The view factor from surface 2 to surface 1 can be calculated using the reciprocity rule,

$$A_2F_{21} = A_1F_{12}$$

The area of the sphere is A_1

The area of the disc surface is A_2

Therefore, the view factor F_{21} calculated as,

$$F_{21} = \frac{A_1}{A_2}F_{12}$$
$$= \frac{\pi\times 0.6^2}{\frac{\pi}{4}\times 2.4^2}\times 0.2764$$
$$= 0.0691$$

Therefore, the view factor from 2 to 1 is $\boxed{0.0691}$

Step 3 of 10

(b) The net rate of radiation heat transfer between the surfaces can be determined by the expression,

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

Now substitute 5.67×10^{-8} W/m^2-K for σ , 873K for T_1 , 473K for T_2 , and 0.9 for ε_1 , 0.5 for ε_2

$$\dot{Q}_{12} = \frac{5.67\times10^{-8}\times(873^4 - 473^4)}{\frac{1-0.9}{1.131\times0.9} + \frac{1}{1.131\times0.2764} + \frac{1-0.5}{4.524\times0.5}}$$
$$= \frac{30095.56}{3.518}$$
$$= 8554.73\,\text{W}$$

Therefore, the net rate of radiation heat exchange between the sphere and the disk is $\boxed{8554.73\,\text{W}}$

Step 4 of 10

(c)The given two modifications could increase the net rate of radiation heat exchange, which are

1) Emissivity of the surfaces should be 1.
   It can be achieved by paint each of the two surfaces to alter their emissivities, and get maximum possible heat exchange.
2) By adjusting the distance between the two bodies which means that for $h = 0$ the maximum possible heat exchange will happen.

In this case, the view factor becomes,

$$F_{12} = 0.5\left\{1-\left[1+\left(\frac{r_2}{h}\right)^2\right]^{-0.5}\right\}$$

Now substitute, $h = 0$ and 120 cm for r_2

$$F_{12} = 0.5\left\{1-\left[1+\left(\frac{120}{0}\right)^2\right]^{-0.5}\right\}$$
$$= 0.5$$

$$F_{12}=0.5\left\{1-\left[1+\left(\frac{120}{0}\right)^2\right]^{-0.5}\right\}$$
$$=0.5$$

Step 5 of 10

The view factor F_{21} calculated as using the reciprocity rule as follows,

$$A_2F_{21}=A_1F_{12}$$
$$F_{21}=\frac{A_1}{A_2}F_{12}$$

Now substitute, $A_1=\pi\times0.6^2$, $A_2=\frac{\pi}{4}\times2.4^2$ and $F_{12}=0.5$

$$F_{21}=\frac{1.131}{4.524}\times0.5$$
$$=0.125$$

Step 6 of 10

(3) By providing a refractory enclosure and let us assume the enclosure surface is 3,

By using the summation rule,

$$F_{11}+F_{12}+F_{13}=1$$
$$F_{21}+F_{22}+F_{23}=1$$

Since the view factor with respect to the sphere surface and disc itself, $F_{11}=0$ and $F_{22}=0$

Therefore,

$$F_{11}+F_{12}+F_{13}=1$$
$$0+F_{12}+F_{13}=1$$
$$F_{13}=1-F_{12}$$

Substitute, 0.5 for F_{12}

$$F_{13}=1-F_{12}$$
$$=1-0.5$$
$$=0.5$$

Step 7 of 10

And for the surface 2

$$F_{21}+F_{22}+F_{23}=1$$
$$F_{21}+0+F_{23}=1$$
$$F_{23}=1-F_{21}$$

Now substitute 0.5 for F_{12}

$$F_{23}=1-F_{21}$$
$$=1-0.125$$
$$=0.875$$

Step 8 of 10

The surface resistance of the circular disk can be calculated as

$$R_2=\frac{1-\varepsilon_2}{A_2\varepsilon_2}$$

Now, substitute 0.5 for ε_2, $A_2=\frac{\pi}{4}\times2.4^2$

$$R_2=\frac{1-1}{\frac{\pi}{4}\times2.4^2\times1}$$
$$=0$$

The space resistance for surfaces $1\to2$ can be calculated as

$$R_{12}=\frac{1}{A_1F_{12}}$$

Now, substitute 0.5 for F_{12}, $A_1=\pi\times0.6^2$

$$R_{12}=\frac{1}{\left(\pi\times0.6^2\right)\times0.5}$$
$$=1.768\ \text{m}^{-2}$$

Step 9 of 10

The space resistance for surfaces $1\to3$ can be calculated as

$$R_{13}=\frac{1}{A_1F_{13}}$$

Now, substitute 0.5 for F_{13}, $A_1=\pi\times0.6^2$

$$R_{13}=\frac{1}{1.131\times0.5}$$
$$=1.768\ \text{m}^{-2}$$

The space resistance for surfaces $2\to3$ can be calculated as

$$R_{23}=\frac{1}{A_2F_{23}}$$

Now, substitute 0.875 for F_{23}, $A_2=\frac{\pi}{4}\times2.4^2$

$$R_{23}=\frac{1}{4.524\times0.875}$$
$$=0.25262\ \text{m}^{-2}$$

Step 10 of 10

Therefore, the net rate of radiation heat exchange can be calculated as,

$$\dot{Q}_{12}=\frac{E_{b1}-E_{b2}}{\left(\frac{1}{R_{12}}+\frac{1}{R_{13}+R_{23}}\right)^{-1}+R_2}$$

Emissive power of the surface 1 E_{b1}

Emissive power of the surface 1 E_{b2}

$$\dot{Q}_{12}=\frac{\sigma T_1^4-\sigma T_2^4}{}$$

Emissive power of the surface 1 E_{b1}

Emissive power of the surface 1 E_{b2}

$$\dot{Q}_{12} = \frac{\sigma T_1^4 - \sigma T_2^4}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13}+R_{23}}\right)^{-1} + R_2}$$

Now substitute, 0 for R_2 1.768 m^{-2} for R_{12} 1.768 m^{-2} for R_{13} 0.25262 m^{-2} for R_{23} and $5.67\times10^{-8}\ W/m^2\cdot K$ for σ , 873K for T_1 , 473K for T_2

$$\dot{Q}_{12} = \frac{5.67\times10^{-8}\times(873^4 - 473^4)}{\left(\frac{1}{1.768} + \frac{1}{(1.768+0.25262)}\right)^{-1} + 0}$$

$$= \frac{30095.56}{0.9429}$$

$$= 31916.59\ W$$

The net rate of radiation heat exchange between the two bodies, $\boxed{\dot{Q}_{12} = 31916.6\,W}$

Hence, the net rate of heat transfer is higher in part (c) which is almost 3 times higher than part 2 by using the best values.

Question: 21.079P

Two parallel disks of diameter $D = 0.6$ m separated by $L = 0.4$ m are located directly on top of each other. Both disks are black and are maintained at a temperature of 450 K. The back sides of the disks are insulated, and the environment that the disks are in can be considered to be a blackbody at 300 K. Determine the net rate of radiation heat transfer from the disks to the environment.

Step 1 of 4

Given

Diameter of the disk, $D = 0.6\text{ m}$

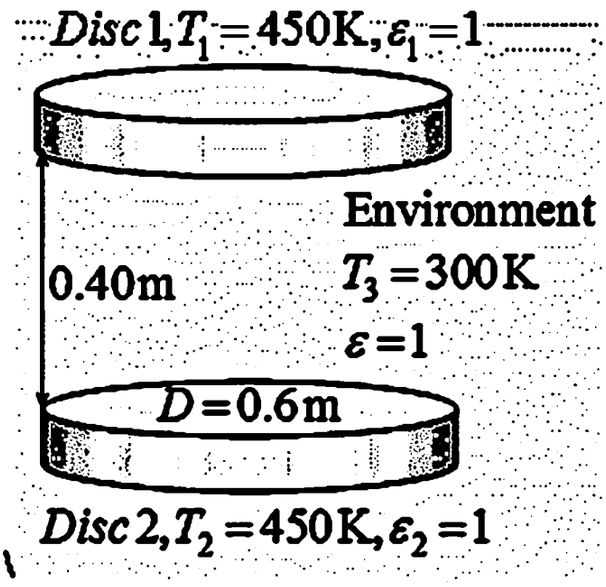
Distance between two discs, $L = 0.4\text{m}$

Temperatures of each disc 1, $T_1 = 450\text{K}$

Temperatures of each disc 2, $T_2 = 450\text{K}$

Environment temperature, $T_3 = 300\text{K}$

Step 2 of 4



Step 3 of 4

From figure, view factor between two coaxial parallel disks

For $L/r_1 = 1.33$

And $r_2/L = 0.75$

So view factor between base to top surface of the cylinder becomes $F_{12} = 0.26$ From the summation rule

$$F_{13} = 1 - 0.26$$
$$= 0.74$$

Step 4 of 4

The net rate of radiation heat transfer from the discs to the environment is calculated as

$$\dot{Q}_3 = \dot{Q}_{13} + \dot{Q}_{23}$$
$$= 2\dot{Q}_{13}$$
$$\dot{Q}_3 = 2F_{13}A_1\sigma(T_1^4 - T_3^4)$$
$$= 2 \times 0.74 \times \left(\pi \times 0.3^2\right) \times 5.67 \times 10^{-8} \times \left(450^4 - 300^4\right)$$
$$= 780.76\text{ W}$$

Net rate heat radiation, $\boxed{\dot{Q}_3 = 780.76\text{ W}}$

Question: 21.080P

Two black parallel rectangles with dimensions 3 ft × 5 ft are spaced apart by a distance of 1 ft. The two parallel rectangles are experiencing radiation heat transfer as black surfaces, where the top rectangle receives a total of 180,000 Btu/h radiation heat transfer rate from the bottom rectangle. If the top rectangle has a uniform temperature of 60°F, determine the temperature of the bottom rectangle.

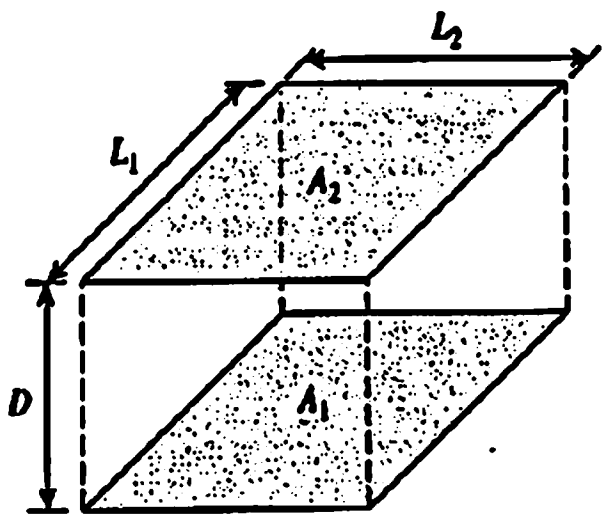


FIGURE P21–80E

Step 1 of 4

Size of the rectangle, $L_1 = 5\text{ft}$

Size of the rectangle, $L_2 = 3\text{ft}$

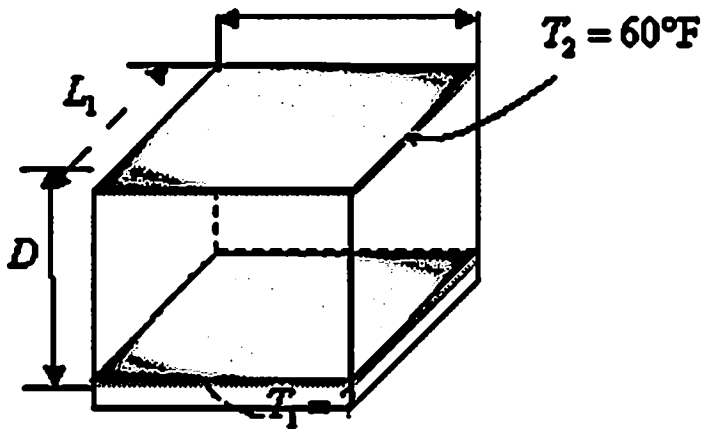
Distance between two rectangles, $D = 1\text{ft}$

The net rate of radiation heat transfer from surface A_1 to A_2 is, $\dot{Q}_{12} = 180,000\ \text{Btu/h}$

Temperature of the top rectangle, $T_2 = 60°\text{F}$

Step 2 of 4

Two black parallel rectangles with all dimensions as shown in figure:



Step 3 of 4

Consider the base surface to be surface 1 and the top surface to be surface 2.
From the figure, the view factor from the base surface to the top surface is

$$\frac{L_1}{D} = \frac{5}{1}$$
$$= 5$$
$$\frac{L_2}{D} = \frac{3}{1}$$
$$= 3$$

From the view factor chart

At $\frac{L_1}{D} = 5$ and $\frac{L_2}{D} = 5$ we have

$F_{12} = 0.6$

Step 4 of 4

Since the top and bottom surfaces are blackbody, the net rate of radiation heat transfer can be expressed as

$$\dot{Q}_{12} = A_1 F_{12} \sigma (T_1^4 - T_2^4)$$

$$180,000 = 15 \times 0.6 \times 0.1714 \times 10^{-8} \times \left(T_1^4 - (60+460)^4\right)$$

By rewriting the above equation for the temperature of the bottom rectangle, we will get

$T_1 = 1851\text{ R}$

$T_1 = 1391°\text{F}$

Therefore, the temperature of the bottom rectangle is $\boxed{T_1 = 1391°\text{F}}$.

Question: 21.081P

Two coaxial parallel disks of equal diameter 1 m are originally placed at a distance of 1 m apart. If both disks behave as black surfaces, determine the new distance between the disks such that there is a 75% reduction in radiation heat transfer rate from the original distance of 1 m.

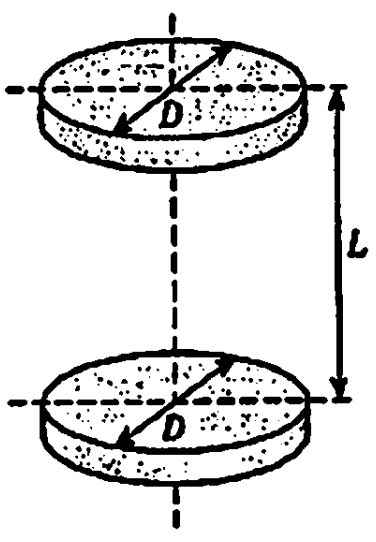


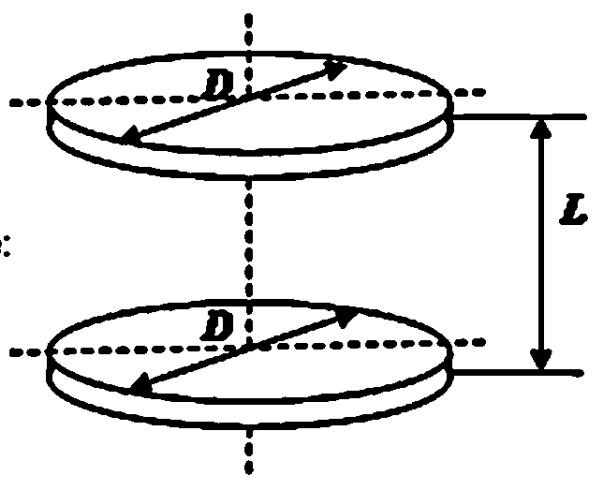
FIGURE P21–81

Step 1 of 5

Draw two coaxial parallel disks

I

of equal diameter and placed at a distance as shown in figure:



Step 2 of 5

Find the view factor from surface 1 to surface 2 by using the equation,

$$F_{12}=1+\frac{1-\sqrt{4R^2+1}}{2R^2}$$

Calculate R value as shown below:

$$R=\frac{D}{2L}$$

Substitute 1 m for D and 1 m for L

$$R=\frac{1}{2\times 1}$$
$$=0.5$$

Now substitute the known values in view factor

$$F_{12}=1+\frac{1-\sqrt{4R^2+1}}{2R^2}$$
$$=1+\frac{1-\sqrt{\left(4\times 0.5^2\right)+1}}{2\times 0.5^2}$$
$$=0.1715$$

Step 3 of 5

Express the net rate of radiation heat transfer between the surfaces by using the expression,

$$\dot{Q}_{12}=A_1F_{12}\sigma\left(T_1^4-T_2^4\right)$$

Since the area of the two discs and the temperature of two discs unchanged, the radiation heat transfer is directly proportional to the view factor, F_{12}

Since 75% reduction in radiation heat transfer rate from the original distance of

1 m will be resulted in the 75% reduction in view factor. Hence the new distance can be determined as a result of 75% reduction in view factor.

Then find the new view factor,

$$F_{12}=0.1715\times 0.25$$
$$=0.043$$

Step 4 of 5

Find the R value by using the equation,

$$F_{12}=1+\frac{1-\sqrt{4R^2+1}}{2R^2}$$
$$0.043=1+\frac{1-\sqrt{\left(4\times R^2\right)+1}}{2R^2}$$
$$\frac{1-\sqrt{\left(4\times R^2\right)+1}}{2R^2}=0.043-1$$
$$\frac{1-\sqrt{\left(4\times R^2\right)+1}}{2R^2}=-0.957$$
$$1-\sqrt{\left(4\times R^2\right)+1}=(-0.957)2R^2$$
$$1-\sqrt{\left(4\times R^2\right)+1}=-1.914R^2$$
$$1+1.914R^2=\sqrt{\left(4\times R^2\right)+1}$$
$$\left(1+1.914R^2\right)^2=\left(\sqrt{\left(4\times R^2\right)+1}\right)^2$$

Step 5 of 5

Squaring on both sides

$$1^2+\left(1.914R^2\right)^2+2\left(1.914R^2\right)=\left(4R^2\right)+1$$
$$1+3.663R^4+3.828R^2=\left(4R^2\right)+1$$
$$3.663R^4+3.828R^2-\left(4R^2\right)=0$$
$$3.663R^4-0.172R^2=0$$
$$\left(3.663R^2-0.172\right)R^2=0$$
$$3.663R^2-0.172=0$$
$$R^2=\frac{0.172}{3.663}$$
$$=0.04695$$
$$R=\pm 0.2166$$

Consider the positive value $R=0.2166$

Find the new distance between discs by using the equation,

$$R=\frac{D}{2L}$$
$$0.2166=\frac{1}{2\times L}$$
$$L=2.3084\,\text{m}$$

Therefore, the new distance between the discs is $\boxed{2.3084\,\text{m}}$

Question: 21.082P

A furnace is shaped like a long equilateral-triangular duct where the width of each side is 2 m. Heat is supplied from the base surface, whose emissivity is ε1 = 0.8, at a rate of 800 W/m2 while the side surfaces, whose emissivities are 0.4, are maintained at 600 K. Neglecting the end effects, determine the temperature of the base surface. Can you treat this geometry as a two-surface enclosure?

Step 1 of 1

The furnace duct acts like a two surface enclosure because the two surfaces of duct having similar properties. Thus consider base surface to be surface 1 and the remaining two surfaces to be surface 2.

Write the expression for heat transfer from the base surface 1 to the surface 2.

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$\frac{\dot{Q}_{12}}{A} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1}{F_{12}} + \frac{1-\varepsilon_2}{\varepsilon_2}}$$

Here, σ is Stefan Boltzmann's constant, T_1 is temperature of base surface, T_2 is temperature of surface 2, ε_1 is emissivity of surface 1, ε_2 is emissivity of surface 2, A is area of the surface, and F_{12} is shape factor.

Substitute $800\,\text{W/m}^2$ for $\dot{Q}_{12}/A$, $5.67\times10^{-8}\,\text{W/m}^2\text{K}^4$ for σ, 0.8 for ε_1, 0.4 for ε_2, 1 for F_{12}, and 600 K for T_2.

$$800 = \frac{5.67\times10^{-8}(T_1^4 - 600^4)}{\left(\frac{1-0.8}{0.8}\right) + \left(\frac{1}{1}\right) + \left(\frac{1-0.4}{0.4}\right)}$$

$$800 = \frac{5.67\times10^{-8}\left(T_1^4 - 600^4\right)}{0.25+1+1.5}$$

$$T_1 = 640.598\,\text{K}$$

Therefore, the base temperature is $\boxed{640.598\,\text{K}}$.

Question: 21.083P

Reconsider Prob. 21-82. Using EES (or other) software, investigate the effects of the rate of the heat transfer at the base surface and the temperature of the side surfaces on the temperature of the base surface. Let the rate of heat transfer vary from 500 W/m2 to 1000 W/m2 and the temperature from 300 K to 700 K. Plot the temperature of the base surface as functions of the rate of heat transfer and the temperature of the side surfaces, and discuss the results.

Step 1 of 7

Given data:

Width of the equilateral triangular duct, $w = 2\text{m}$

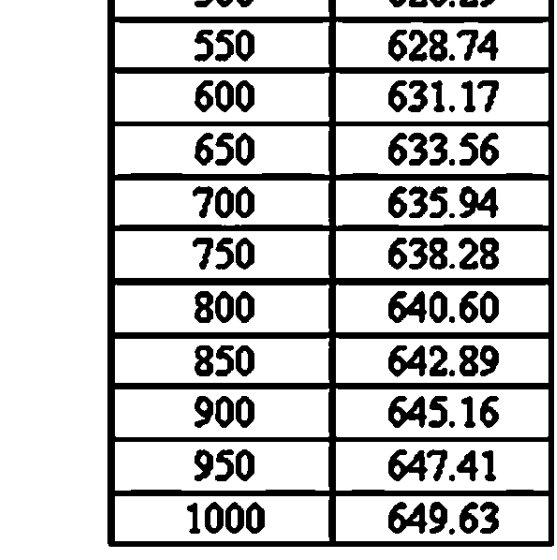
Heat supplied from the base surface, $\dot{Q}_{12} = 800\ \text{W/m}^2$

Emissivity of the base surface, $\varepsilon_1 = 0.8$

Emissivity of the side surfaces, $\varepsilon_2 = 0.4$

Temperature of the side surfaces, $T_2 = 600\text{K}$

Step 2 of 7



Step 3 of 7

This geometry can be treated as a two surface enclosure since two surfaces have identical properties. We consider base surface to be surface 1 and other two surfaces to be surface 2. Then the view factor between the two becomes $F_{12} = 1$. The temperature of the base surface is determined from

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

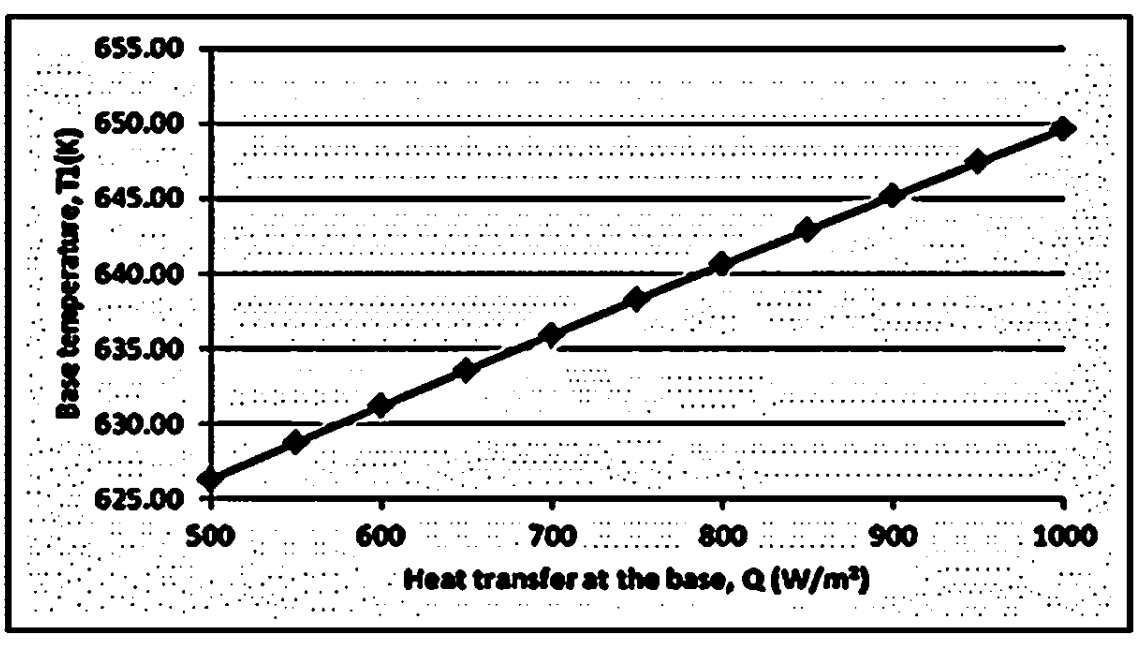
$$\dot{Q}_{12} = \frac{5.67\times10^{-8}(T_1^4 - 600^4)}{\left(\frac{1-0.8}{1\times0.8}\right) + \left(\frac{1}{1\times1}\right) + \left(\frac{1-0.4}{1\times0.4}\right)} \qquad \ldots\ldots (1)$$

Step 4 of 7

Using the Microsoft Excel sheet, from the equation(1) by varying the heat transfer at the base from 500 to the 1000 W/m^2, the variation in temperature of the base calculated and results are tabulated and plotted below.

$\dot{Q}_{12}$ (W/m^2)	T_1(K)
500	626.29
550	628.74
600	631.17
650	633.56
700	635.94
750	638.28
800	640.60
850	642.89
900	645.16
950	647.41
1000	649.63

Step 5 of 7

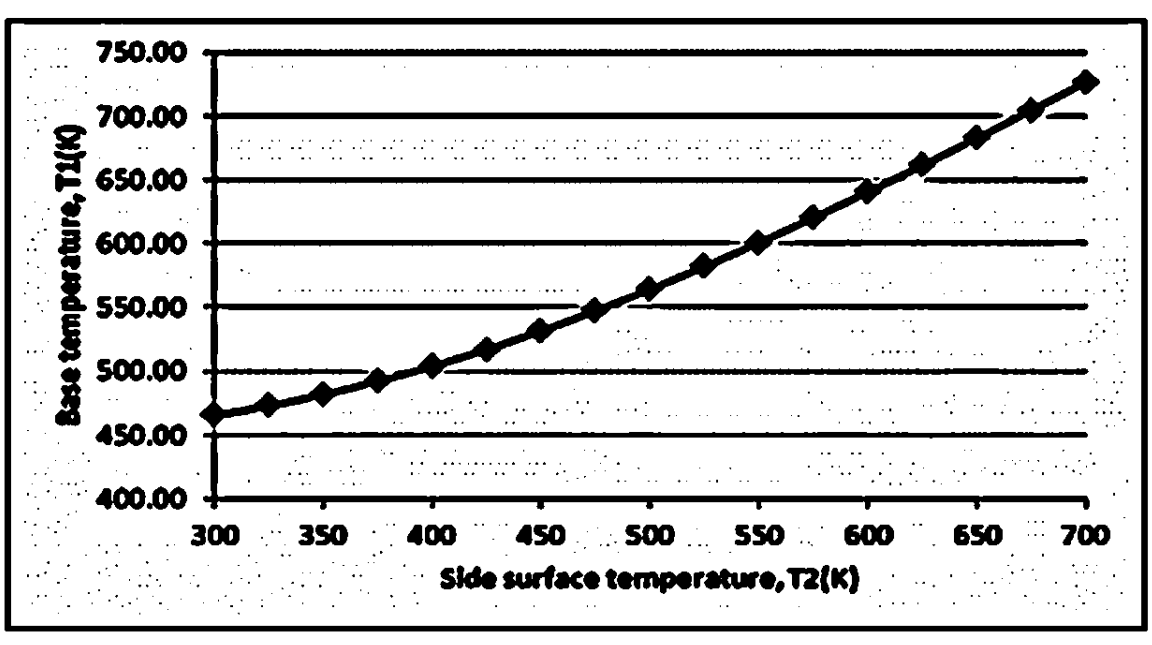


Step 6 of 7

By using equation (1) varying the side surfaces temperature from 300 K to 700 K, the variation in temperature of the base calculated and results are tabulated and plotted below.

T_2 (K)	T_1(K)
300	465.37
325	472.77
350	481.63
375	491.96
400	503.76
425	516.97
450	531.51
475	547.28
500	564.16
525	582.05
550	600.82
575	620.37
600	640.60
625	661.42
650	682.76
675	704.54
700	726.71

Step 7 of 7



Question: 21.084P

Consider a 4-m × 4-m × 4-m cubical furnace whose floor and ceiling are black and whose side surfaces are reradiating. The floor and the ceiling of the furnace are maintained at temperatures of 550 K and 1100 K, respectively. Determine the net rate of radiation heat transfer between the floor and the ceiling of the furnace.

Step 1 of 4

Given

Side of the cubical furnace, $a = 4\,\text{m}$

Temperatures of ceiling $T_1 = 1100\,\text{K}$

Temperature of floor $T_2 = 550\,\text{K}$

Step 2 of 4

Consider the ceiling surface to be surface 1, floor to be surface 2 and side surfaces to be surface 3.

From figure, view factor between two aligned rectangles of equal size

For $\frac{L_2}{D} = 1$

And $\frac{L_1}{D} = 1$

View factor from ceiling to floor, $F_{12} = 0.2$

And we know that, $F_{11} + F_{12} + F_{13} = 1$

$$\Rightarrow 0 + 0.2 + F_{13} = 1$$

$$\Rightarrow F_{13} = 0.8$$

Surface areas of the floor and ceiling are $A_1 = A_2$

$$= 4^2$$

$$= 16\,\text{m}^2$$

Step 3 of 4

Space resistances are $R_{12} = \frac{1}{A_1 F_{12}}$

$$= \frac{1}{16 \times 0.2}$$

$$= 0.3125\,\text{m}^{-2}$$

And $R_{13} = R_{23} = \frac{1}{A_1 F_{13}}$

$$= \frac{1}{16 \times .8}$$

$$= 0.07812\,\text{m}^{-2}$$

Emissive powers are $E_{b1} = \sigma T_1^4$

$$= 5.67 \times 10^{-8} \times 1100^4$$

$$= 83015\ \text{W/m}^2$$

And $E_{b2} = \sigma T_2^4$

$$= 5.67 \times 10^{-8} \times 550^4$$

$$= 5188\ \text{W/m}^2$$

Step 4 of 4

Net rate of radiation heat transfer between floor and ceiling is determined as follows

The net rate heat transfer, $\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}}\right)^{-1}}$

$$= \frac{83015 - 5188}{\left(\frac{1}{0.3125} + \frac{1}{2 \times 0.07812}\right)^{-1}}$$

$$= \frac{77827}{0.1042}$$

$$= 746900.2\,\text{W}$$

Net rate heat radiation, $\boxed{\dot{Q} = 746.9\,\text{kW}}$

Question: 21.085P

Two concentric spheres of diameters $D1$, = 0.3 m and $D2$ = 0.6 m are maintained at uniform temperatures $T1$, = 800 K and $T2$ = 500 K and have emissivities ε1 = 0.5 and ε2 = 0.7, respectively. Determine the net rate of radiation heat transfer between the two spheres. Also, determine the convection heat transfer coefficient at the outer surface if both the surrounding medium and the surrounding surfaces are at 30°C. Assume the emissivity of the outer surface is 0.35.

Step 1 of 5

Given data:

Diameter of the inner sphere, $D_1 = 0.3\,\text{m}$

Diameter of the outer sphere, $D_2 = 0.6\,\text{m}$

Surface temperature of the inner sphere, $T_1 = 800\,\text{K}$

Surface temperature of the outer sphere, $T_2 = 500\,\text{K}$

Emissivity of the inner sphere outer surface, $\varepsilon_1 = 0.5$

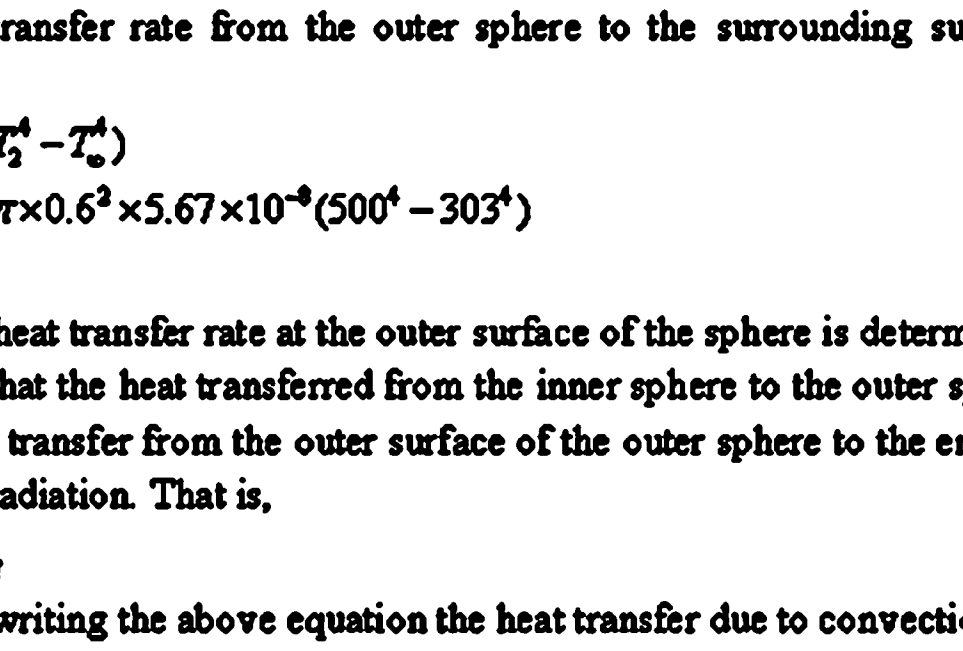
Emissivity of the outer sphere inner surface, $\varepsilon_2 = 0.7$

Emissivity of the outer sphere outer surface, $\varepsilon_3 = 0.35$

Temperature of the surrounding medium, $T_\infty = 303\,\text{K}$

Step 2 of 5

Two concentric spheres of different diameters are maintained at uniform temperatures as shown below.



Step 3 of 5

The net rate of radiation heat transfer between the two spheres can be calculated as

$$\dot{Q}_{12} = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1^2}{r_2^2}\right)}$$

$$= \frac{\pi \times 0.3^2 \times 5.67\times10^{-8}(800^4 - 500^4)}{\frac{1}{0.5} + \frac{1-0.7}{0.7}\left(\frac{0.15^2}{0.3^2}\right)}$$

$$= \frac{5564.55}{2 + 0.428(0.25)}$$

$$= 2640.8\,\text{W}$$

The net rate of heat transfer between the two spheres is $\boxed{\dot{Q}_{12} = 2640.8\,\text{W}}$.

Step 4 of 5

Radiation heat transfer rate from the outer sphere to the surrounding surfaces can be calculated as

$$\dot{Q}_{rad} = \varepsilon F_{23} A_2 \sigma(T_2^4 - T_\infty^4)$$

$$= 0.35 \times 1 \times \pi \times 0.6^2 \times 5.67\times10^{-8}(500^4 - 303^4)$$

$$= 1213.6\,\text{W}$$

The convection heat transfer rate at the outer surface of the sphere is determined by using energy balance that the heat transferred from the inner sphere to the outer sphere must be equal to the heat transfer from the outer surface of the outer sphere to the environment by convection and radiation. That is,

$$\dot{Q}_{12} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

Therefore, by rewriting the above equation the heat transfer due to convection becomes,

$$\dot{Q}_{conv} = \dot{Q}_{12} - \dot{Q}_{rad}$$

$$= 2640.8 - 1213.6$$

$$= 1427.2\,\text{W}$$

Step 5 of 5

Then the convection heat transfer coefficient can be expressed as

$$\dot{Q}_{conv} = hA_2(T_2 - T_\infty)$$

By rewriting the above equation for the convection heat transfer coefficient, the equation becomes,

$$h = \frac{\dot{Q}_{conv}}{A_2(T_2 - T_\infty)}$$

$$= \frac{1427.2}{\pi \times 0.6^2 \times (500 - 303)}$$

$$= 6.4\,\text{W/m}^2\cdot\text{K}$$

Therefore, the convection heat transfer coefficient at the outer surface is $\boxed{h = 6.4\,\text{W/m}^2\cdot\text{K}}$.

Question: 21.086P

A spherical tank of diameter $D = 2$ m that is filled with liquid nitrogen at 100 K is kept in an evacuated cubic enclosure whose sides are 3 m long. The emissivities of the spherical tank and the enclosure are $\varepsilon 1 = 0.1$ and $\varepsilon 2 = 0.8$, respectively. If the temperature of the cubic enclosure is measured to be 240 K, determine the net rate of radiation heat transfer to the liquid nitrogen.

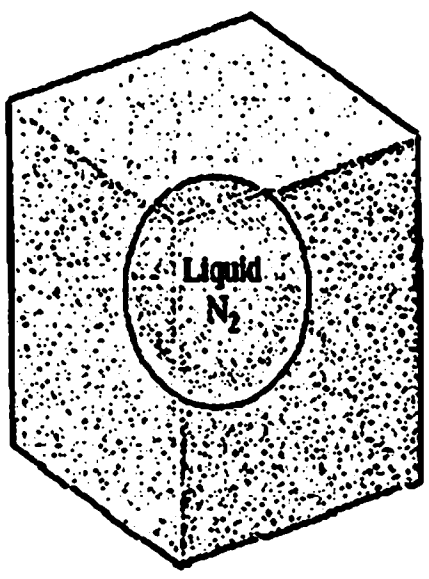


FIGURE P21–86

Step 1 of 4

Given

Diameter of spherical tank, $D = 2$ m

Emissivity of the spherical tank, $\varepsilon_1 = 0.1$

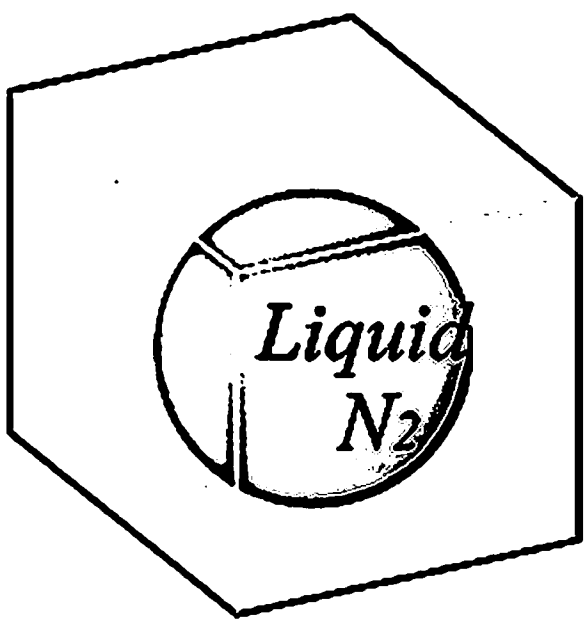
Emissivity of cube surface, $\varepsilon_2 = 0.8$

Side of the cube, $a = 3$m

Temperature of tank, $T_1 = 100\ K$

Temperature of cube surface, $T_2 = 240$ K

Step 2 of 4



Step 3 of 4

Surface area of spherical tank, $A_1 = \pi D^2$

$= \pi \times 2^2$

$= 12.57\,\text{m}^2$

Surface area of cube, $A_2 = 6a^2$

$= 6 \times 3^2$

$= 54\,\text{m}^2$

Step 4 of 4

Rate of radiation heat transfer to the liquid nitrogen is

$$\dot{Q}_{21} = -\dot{Q}_{12}$$

$$= -\frac{\sigma A_1 \left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1} + \left(\frac{1-\varepsilon_2}{\varepsilon_2}\right)\left(\frac{A_1}{A_2}\right)}$$

$$= -\frac{5.67 \times 10^{-8} \times 12.57 \times \left(100^4 - 240^4\right)}{\frac{1}{0.1} + \frac{1-0.8}{0.8}\left(\frac{12.57}{54}\right)}$$

$$= \frac{2293.36}{10.058}$$

$$= 228\,\text{W}$$

Net rate of radiation heat transfer, $\boxed{\dot{Q} = 228\,\text{W}}$

Question: 21.087P

Repeat Prob. 21-86 by replacing the cubic enclosure by a spherical enclosure whose diameter is 3 m.

Step 1 of 2

Given

Diameter of inner sphere, $D_1 = 2\ \text{m}$

Diameter of outer sphere, $D_2 = 3\ \text{m}$

Emissivity of inner sphere surface, $\varepsilon_1 = 0.1$

Emissivity of outer sphere surface, $\varepsilon_2 = 0.8$

Temperature of inner sphere, $T_1 = 100\ K$

Temperature of outer surface, $T_2 = 240\ \text{K}$

Step 2 of 2

Rate of radiation heat transfer to the liquid nitrogen is

$$\dot{Q}_{21} = -\dot{Q}_{12}$$

$$= -\frac{A_1\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1} + \left(\frac{1-\varepsilon_2}{\varepsilon_2}\right)\frac{A_1}{A_2}}$$

Surface area of the sphere, $A = \pi D^2$

$$= -\frac{\left(\pi \times 2^2\right) \times 5.67 \times 10^{-8} \times \left(100^4 - 240^4\right)}{\frac{1}{0.1} + \left(\frac{1-0.8}{0.8}\right)\left[\frac{\pi \times 2^2}{\pi \times 3^2}\right]}$$

$$= \frac{2292.7}{10.11}$$

$$= 226.8\ \text{W}$$

Net rate heat radiation, $\boxed{\dot{Q} = 226.8\ \text{W}}$

Question:

Reconsider Prob. 21-86. Using EES (or other) software, investigate the effects of the side length and the emissivity of the cubic enclosure, and the emissivity of the spherical tank on the net rate of radiation heat transfer. Let the side length vary from 2 m to 12 m and both emissivities from 0.1 to 0.9. Plot the net rate of radiation heat transfer as functions of side length and emissivities, and discuss the results.

Step 1 of 9

Given data:

Diameter of the spherical tank, $D = 2\,\text{m}$

Size of the cubical enclosure, $L = 3\,\text{m}$

Temperature of the liquid oxygen, $T_1 = 100\,\text{K}$

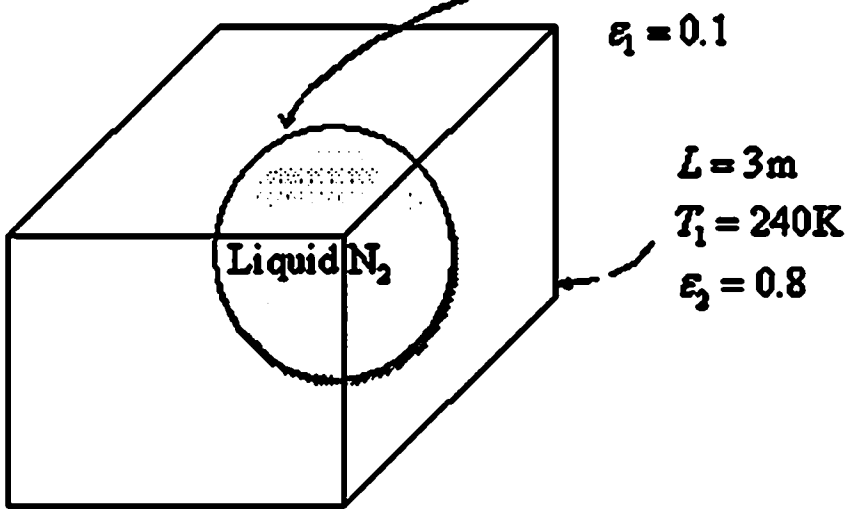
Temperature of the cubical enclosure, $T_2 = 240\,\text{K}$

Emissivity of the spherical tank, $\varepsilon_1 = 0.1$

Emissivity of the cubical enclosure, $\varepsilon_2 = 0.8$

Step 2 of 9

A spherical tank of diameter filled with liquid nitrogen is as shown below:



Step 3 of 9

The net rate of radiation heat transfer to liquid nitrogen can be expressed as

$$\dot{Q}_{12} = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{A_1}{A_2}\right)}$$

$$\dot{Q}_{12} = \frac{\pi \times 2^2 \times 5.67\times10^{-8}(100^4 - 240^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{\pi \times 2^2}{6 \times L^2}\right)} \qquad \ldots\ldots (1)$$

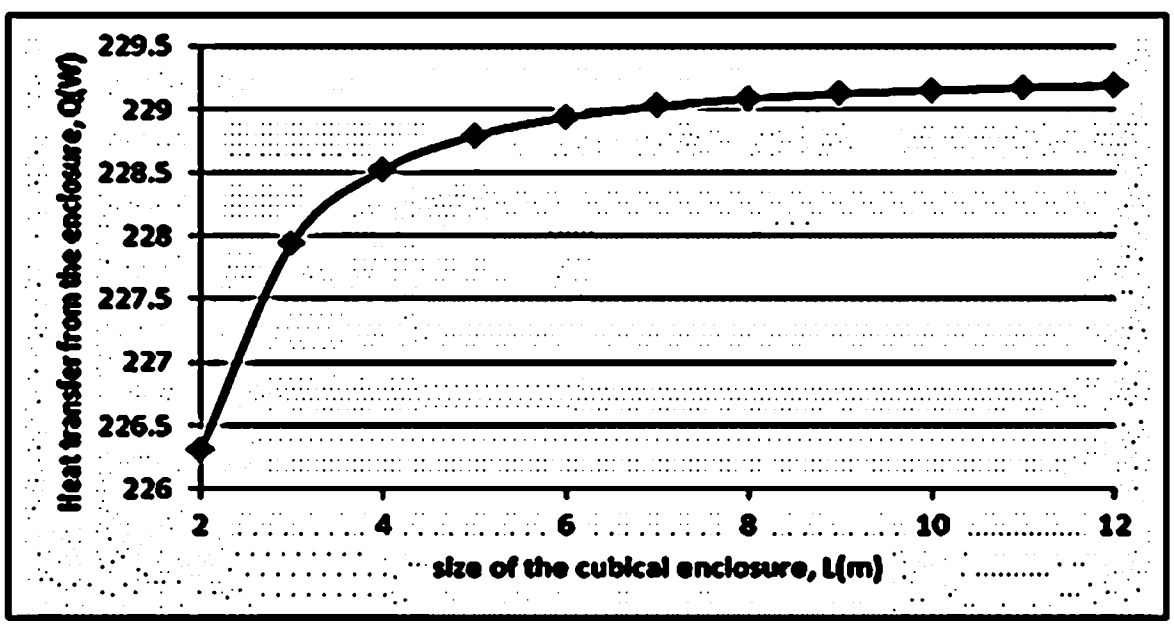
Step 4 of 9

Using equation (1) vary the size of the cubical enclosure from 2 m to 12 m, the variation in net rate of radiation heat transfer is calculated and results are tabulated.

L,m	Q,W
2	226.3073
3	227.9435
4	228.5218
5o	228.7905
6	228.9367
7	229.0249
8	229.0822
9	229.1215
10	229.1497
11	229.1705
12	229.1863

Step 5 of 9

Plotting the size of the cubical enclosure and net rate of radiation heat transfer:



Step 6 of 9

From equation (1) vary the emissivity of the spherical tank from 0.1 to 0.9, the variation in net rate of radiation heat transfer is calculated and results are tabulated.

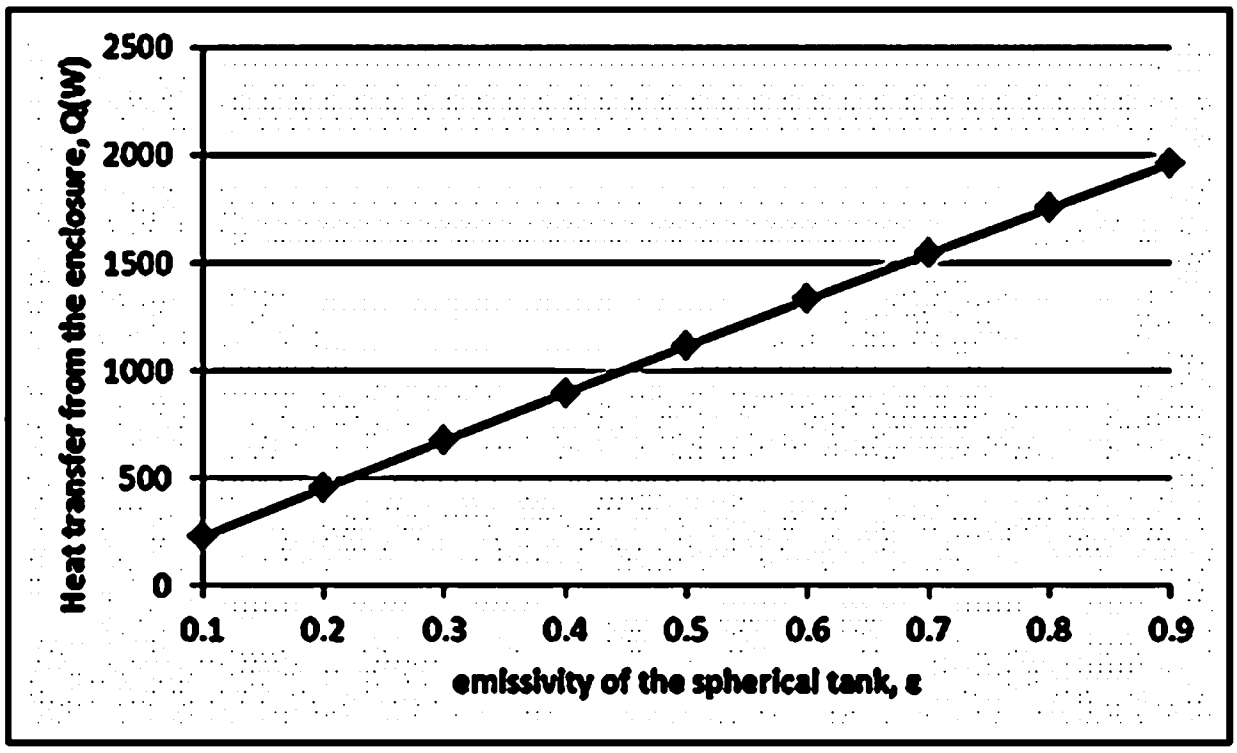
ε_1	Q(W)
0.1	227.9435
0.2	453.2653
0.3	676.0103
0.4	896.2226
0.5	1113.945
0.6	1329.219
0.7	1542.087
0.8	1752.588
0.9	1960.762

Step 7 of 9

Plotting the size of the cubical enclosure and net rate of radiation heat transfer:

Step 7 of 9

Plotting the size of the cubical enclosure and net rate of radiation heat transfer:



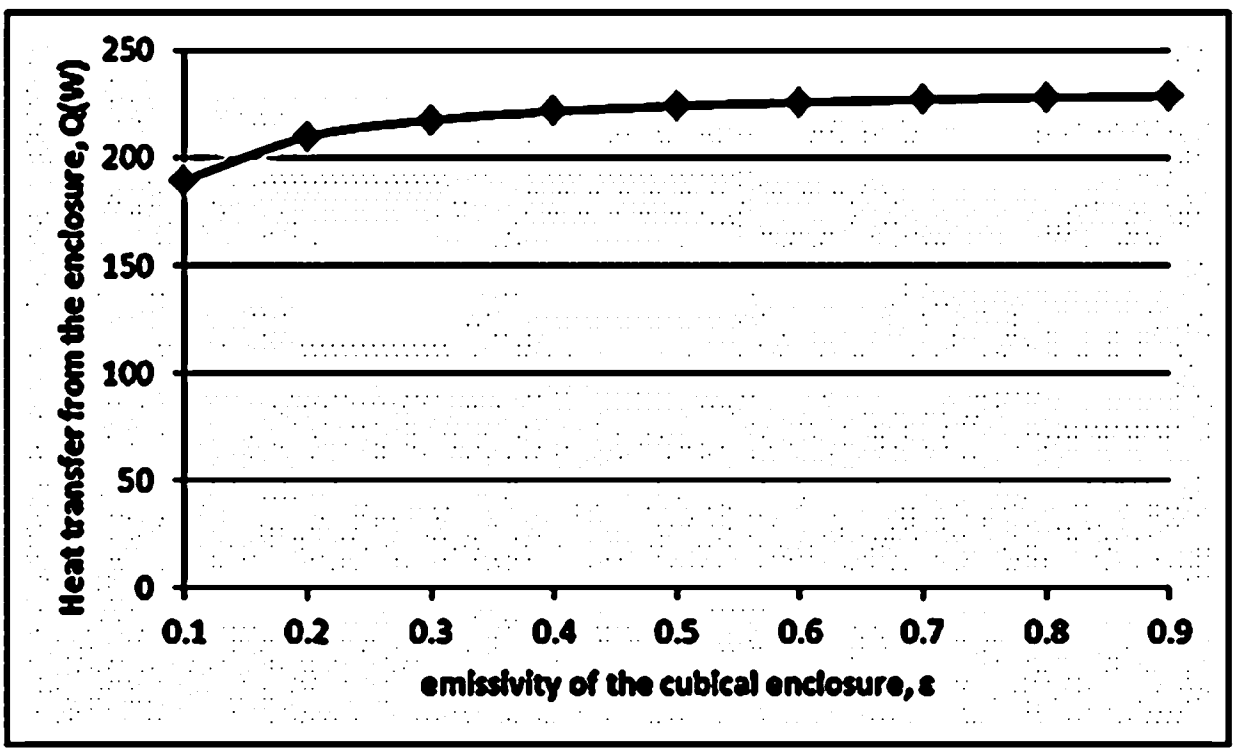
Step 8 of 9

From equation (1) vary the emissivity of the cubical enclosure from 0.1 to 0.9, the variation in net rate of radiation heat transfer is calculated and results are tabulated.

ε_2	Q(W)
0.1	189.5669
0.2	209.7456
0.3	217.4617
0.4	221.5366
0.5	224.0556
0.6	225.7671
0.7	227.0057
0.8	227.9435
0.9	228.6784

Step 9 of 9

Plotting the emissivity of the cubical enclosure and net rate of radiation heat transfer:



Question: 21.089P

Consider a circular grill whose diameter is 0.3 m. The bottom of the grill is covered with hot coal bricks at 950 K, while the wire mesh on top of the grill is covered with steaks initially at 5°C. The distance between the coal bricks and the steaks is 0.20 m. Treating both the steaks and the coal bricks as blackbodies, determine the initial rate of radiation heat transfer from the coal bricks to the steaks. Also, determine the initial rate of radiation heat transfer to the steaks if the side opening of the grill is covered by aluminum foil, which can be approximated as a reradiating surface.

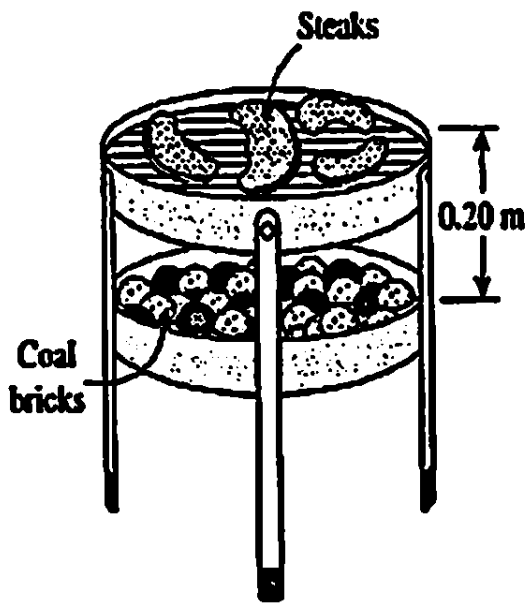


FIGURE P21–89

Step 1 of 5

Given

Diameter of circular grill, $D = 0.3\ \text{m}$

Distance between the coal bricks and the steaks, $L = 0.2\ \text{m}$

Temperatures of the hot coal bricks, $T_1 = 950\ K$

Temperatures of the steaks, $T_2 = 5°\text{c}$

Step 2 of 5

(a) First we have to find view factor F_{12}

$$R_{12} = F_{ij} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{R_j}{R_i}\right)^2\right]^{1/2}\right\}$$

Where $R_i = R_j$

$$= \frac{r_1}{L}$$
$$= \frac{0.15}{0.20}$$
$$= 0.75$$

And $S = 1 + \frac{1 + R_j^2}{R_i^2}$

$$= 1 + \frac{1 + 0.75^2}{0.75^2}$$
$$= 3.778$$

Step 3 of 5

$\therefore$ View factor, $F_{12} = \frac{1}{2}\left\{3.778 - \left[3.778^2 - 4\left(\frac{0.75}{0.75}\right)^2\right]^{1/2}\right\}$

$$= \frac{1}{2}(3.778 - 3.205)$$
$$= 0.2864$$

Initial rate of radiation heat transfer from the coal bricks to the steaks is determined as

$$\dot{Q}_{12} = F_{12}A_1\sigma\left(T_1^4 - T_2^4\right)$$
$$= 0.2864 \times \left(\pi \times 0.3^2/4\right) \times 5.67 \times 10^{-8} \times \left(950^4 - 278^4\right)$$
$$= 928\ \text{W}$$

The net rate heat transfer, $\boxed{\dot{Q} = 928\ \text{W}}$

Step 4 of 5

(b) Initial rate of radiation heat transfer to steaks if the side opening of the grill is covered by aluminum foil, which can be approximated as reradiating surface.

Emissivities of the reradiating surfaces are $\varepsilon = 1$

Surface area of the grills is $A_1 = A_2$

$$= \frac{\pi \times 0.3^2}{4}$$
$$= 0.0707\ \text{m}^2$$

The emissive powers are

$$E_{b1} = \varepsilon\sigma T_1^4$$
$$= 1 \times 5.67 \times 10^{-8} \times 950^4$$
$$= 46182.5\ \text{W/m}^2$$

And $E_{b2} = \varepsilon\sigma T_2^4$

$$= 1 \times 5.67 \times 10^{-8} \times 278^4$$
$$= 339\ \text{W/m}^2$$

Step 5 of 5

Space resistances are

$$R_{12} = \frac{1}{A_1 F_{12}}$$
$$= \frac{1}{0.0707 \times 0.2864}$$
$$= 49.39\ \text{m}^{-2}$$

And $R_{13} = R_{2B}$

$$= \frac{1}{A_1 F_{13}}$$
$$= \frac{1}{0.0707 \times (1 - 0.2864)}$$
$$= 19.81\ \text{m}^{-2}$$

Then heat transfer rate becomes

$$\dot{Q}_{12} = \frac{46182.5 - 339}{\left(\frac{1}{49.39} + \frac{1}{2 \times 19.81}\right)^{-1}}$$
$$= \frac{45843.5}{21.98}$$
$$= 2085.7\ \text{W}$$

Initial rate of radiation heat transfer to steaks, $\boxed{\dot{Q}_{12} = 2085.7\ \text{W}}$

Question: 21.090P

Consider a hemispherical furnace of diameter D = 5 m with a flat base. The dome of the furnace is black, and the base has an emissivity of 0.7. The base and the dome of the furnace are maintained at uniform temperatures of 400 and 1000 K, respectively. Determine the net rate of radiation heat transfer from the dome to the base surface during steady operation.

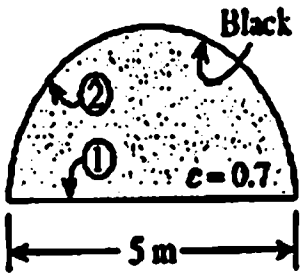


FIGURE P21–90

Step 1 of 4

Given

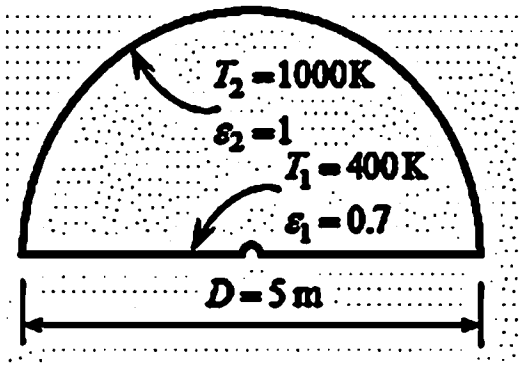
Diameter of hemispherical surface, $D = 5\ \text{m}$

Emissivity of flat base, $\varepsilon = 0.7$

Temperature of the flat base, $T_1 = 400\ K$

Temperature of dome surface, $T_2 = 1000\ \text{K}$

Step 2 of 4



Step 3 of 4

Net rate radiation heat transfer from the dome to base surface during the steady operation determined by

$$\dot{Q}_{21} = -\dot{Q}_{12}$$
$$= -\varepsilon A_1 F_{12} \sigma (T_1^4 - T_2^4)$$

Where view factor F_{12} calculated as follows

For the flat surface $F_{11} = 0$

From the summation rule

$$F_{11} + F_{12} = 1$$
$$\Rightarrow F_{12} = 1$$

Step 4 of 4

Then the heat transfer rate becomes

$$\dot{Q}_{21} = -\varepsilon A_1 F_{12} \sigma (T_1^4 - T_2^4)$$
$$= -0.7 \times \left(\pi \times \frac{5^2}{4} \right) \times 1 \times \left(5.67 \times 10^{-8}\right)\left(400^4 - 1000^4\right)$$
$$= 7.594 \times 10^5\ \text{W}$$
$$= 759.4\ \text{kW}$$

Net heat transfer, $\boxed{\dot{Q} = 759.4\ \text{kW}}$

Question: 21.091P pg 1

Consider two rectangular surfaces perpendicular to each other with a common edge which is 1.6 m long. The horizontal surface is 0.8 m wide and the vertical surface is 1.2 m high. The horizontal surface has an emissivity of 0.75 and is maintained at 450 K. The vertical surface is black and is maintained at 700 K. The back sides of the surfaces are insulated. The surrounding surfaces are at 290 K, and can be considered to have an emissivity of 0.85. Determine the net rate of radiation heat transfers between the two surfaces, and between the horizontal surface and the surroundings.

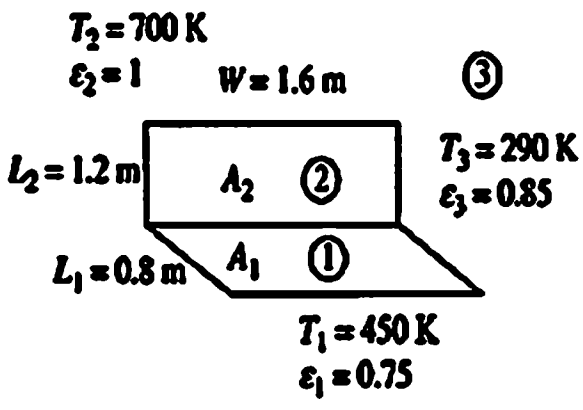


FIGURE P21-91

Step 1 of 8

Given data:

Length of the two rectangular surfaces, $W = 1.6\,\text{m}$

Width of the horizontal surface, $L_1 = 0.8\,\text{m}$

Width of the vertical surface, $L_2 = 1.2\,\text{m}$

Temperature of the horizontal surface, $T_1 = 450\,\text{K}$

Temperature of the vertical surface, $T_2 = 700\,\text{K}$

Emissivity of the horizontal surface, $\varepsilon_1 = 0.75$

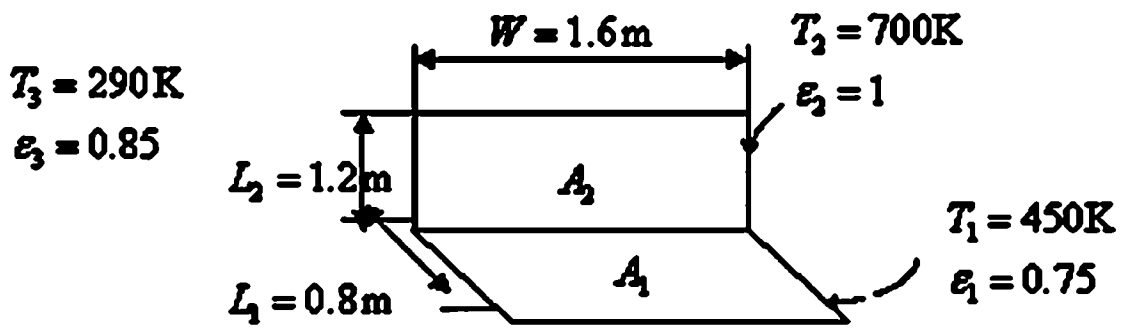
Emissivity of the vertical surface, $\varepsilon_2 = 1$

Emissivity of the surroundings, $\varepsilon_3 = 0.85$

Temperature of the surrounding medium, $T_3 = T_\infty = 290\,\text{K}$

Step 2 of 8

Two rectangular surfaces perpendicular to each other is as shown in figure:



Step 3 of 8

Consider the horizontal rectangle to be surface 1, the vertical rectangle to be surface 2 and the surroundings to be surface 3. This system can be considered to be a three-surface enclosure. The view factor from surface 1 to surface 2 is determined as

$$\frac{L_1}{W} = \frac{0.8}{1.6}$$
$$= 0.5$$
$$\frac{L_2}{W} = \frac{1.2}{1.6}$$
$$= 0.75$$

From the view factor chart

At $\frac{L_1}{W} = 0.5$ and $\frac{L_2}{W} = 0.75$ We have

$$F_{12} = 0.27$$

The surface areas can be calculated as

$$A_1 = 0.8 \times 1.6$$
$$= 1.28\,\text{m}^2$$
$$A_2 = 1.2 \times 1.6$$
$$= 1.92\,\text{m}^2$$
$$A_3 = 2 \times \left(\frac{1.2 \times 0.8}{2}\right) + \sqrt{0.8^2 + 1.2^2} \times 1.6$$
$$= 3.267\,\text{m}^2$$

Step 4 of 8

The surface area of the surroundings is determined assuming that surroundings form flat surfaces at all openings to form an enclosure. Then other view factors are determined to be

By using the reciprocity rule,

$$A_2 F_{21} = A_1 F_{12}$$
$$F_{21} = \frac{A_1}{A_2} \times F_{12}$$
$$= \frac{1.28}{1.98} \times 0.27$$
$$= 0.175$$

By using the summation rule,

$$F_{11} + F_{12} + F_{13} = 1$$
$$0 + 0.27 + F_{13} = 1$$
$$F_{13} = 1 - 0.27$$
$$F_{13} = 0.73$$

By suing the summation rule,

$$F_{21} + F_{22} + F_{23} = 1$$
$$0.175 + 0 + F_{23} = 1$$
$$F_{23} = 1 - 0.175$$
$$F_{23} = 0.825$$

Step 5 of 8

By using the reciprocity rule,

$$A_3 F_{31} = A_1 F_{13}$$
$$F_{31} = \frac{A_1}{A_3} \times F_{13}$$
$$= \frac{1.28}{} \times 0.73$$

$$A_3F_{31} = A_1F_{12}$$

$$F_{31} = \frac{A_1}{A_3} \times F_{12}$$

$$= \frac{1.28}{3.267} \times 0.73$$

$$= 0.286$$

Step 6 of 8

By using the reciprocity rule,

$$A_3F_{32} = A_2F_{23}$$

$$F_{32} = \frac{A_2}{A_3} \times F_{23}$$

$$= \frac{1.98}{3.267} \times 0.825$$

$$= 0.5$$

Step 7 of 8

To determine radiosity for each surface with specified temperature can be expressed as,
For surface 1, the equation can be written as

$$\sigma T_1^4 = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)\right]$$

$$5.67 \times 10^{-8} \times 450^4 = J_1 + \frac{1-0.75}{0.75}\left[0.27(J_1 - J_2) + 0.73(J_1 - J_3)\right]$$

For surface 2, the equation can be written as

$$\sigma T_2^4 = J_2$$

$$5.67 \times 10^{-8} \times 700^4 = J_2$$

For surface 3, the equation can be written as

$$\sigma T_3^4 = J_3 + \frac{1-\varepsilon_3}{\varepsilon_3}\left[F_{31}(J_3 - J_1) + F_{32}(J_3 - J_2)\right]$$

$$5.67 \times 10^{-8} \times 290^4 = J_3 + \frac{1-0.85}{0.85}\left[0.286(J_3 - J_1) + 0.5(J_3 - J_2)\right]$$

Solving the above three equations we will get,

$$J_1 = 2943.82\ \text{W/m}^2$$

$$J_2 = 13613.67\ \text{W/m}^2$$

$$J_3 = 1534.65\ \text{W/m}^2$$

Step 8 of 8

Therefore, the net rate of radiation heat transfers between the two surfaces and between the horizontal surface and the surroundings are determined to be

$$\dot{Q}_{21} = -\dot{Q}_{12}$$

$$= -A_1F_{12}(J_1 - J_2)$$

$$= -1.28 \times 0.27 \times (2943.82 - 13613.67)$$

$$= 3687.5\ \text{W}$$

Therefore, the net rate of radiation heat transfer between the two surfaces is

$$\boxed{\dot{Q}_{21} = 3687.5\ \text{W}}$$

The net rate of heat transfer between the horizontal surface and the surroundings is given by

$$\dot{Q}_{13} = A_1F_{13}(J_1 - J_3)$$

$$= 1.28 \times 0.73 \times (2943.82 - 1534.65)$$

$$= 1316.73\ \text{W}$$

Therefore, the net rate of heat transfer between the horizontal surface and the surroundings is $\boxed{\dot{Q}_{13} = 1316.73\ \text{W}}$

Question: 21.092P

Two long parallel 20-cm-diameter cylinders are located 30 cm apart from each other. Both cylinders are black, and are maintained at temperatures 425 K and 275 K. The surroundings can be treated as a blackbody at 300 K. For a 1-m-long section of the cylinders, determine the rates of radiation heat transfer between the cylinders and between the hot cylinder and the surroundings.

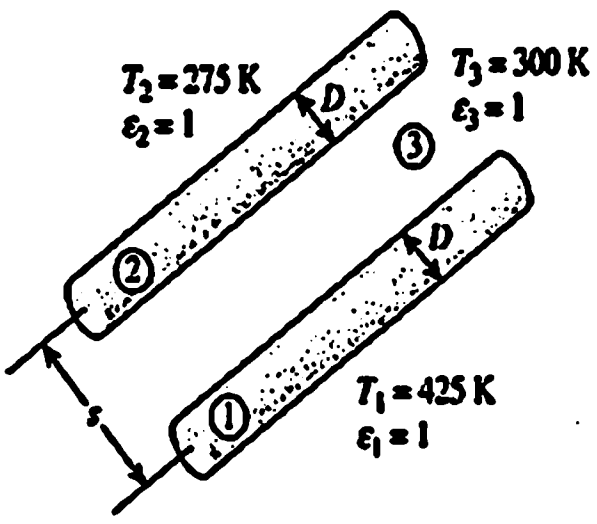


FIGURE P21–92

Step 1 of 3

Given

Diameter of each cylinder, $D = 20$ cm

Distance between both cylinders, $s = 30$ cm

Length of the each cylinder, $L = 1$m

Temperature of cylinder 1, $T_1 = 425$ K

Temperature of cylinder 2, $T_2 = 275$ K

Surrounding temperature, $T_3 = 300$ K

All are black bodies

So, emissivities are $\varepsilon_1 = 1$

$\varepsilon_2 = 1$

$\varepsilon_3 = 1$

Step 2 of 3

(a) Surface area of the cylinder which is face to the other cylinder, $A = \pi DL/2$

$$= \frac{\pi \times 0.2 \times 1}{2}$$

$$= 0.314 \text{ m}^2$$

Where the view factor between two cylinders F_{12} is given by

From crossed strings method, $F_{12} = \dfrac{\sum \text{Crossed strings} - \sum \text{Uncrossed strings}}{2 \times \text{String on surface 1}}$

$$= \frac{2\sqrt{s^2 + L^2} - 2s}{2L}$$

$$= \frac{\left(\sqrt{s^2 + L^2} - s\right)}{L}$$

$$= \frac{\left(\sqrt{0.3^2 + 1^2} - 0.3\right)}{1}$$

$$= 0.744$$

Then the heat transfer rate becomes

$$\dot{Q}_{12} = AF_{12}\sigma(T_1^4 - T_2^4)$$

$$= 0.314 \times 0.744 \times \left(5.67 \times 10^{-8}\right) \times \left(425^4 - 275^4\right)$$

$$= 356.4 \text{ W}$$

Rate of radiation heat transfer between cylinders, $\boxed{\dot{Q}_{12} = 356.4 \text{ W}}$

Step 3 of 3

(b) Rate of radiation heat transfer between the hot cylinder and the surroundings is

Surface area becomes $A_1 = \pi DL$

$$= \pi \times 0.2 \times 1$$

$$= 0.628 \text{ m}^2$$

View factor between hot cylinder and surroundings by summation rule is

$$F_{13} = 1 - F_{12}$$

$$= 1 - 0.744$$

$$= 0.256$$

Rate of heat loss becomes

$$\dot{Q}_{13} = A_1 F_{13}\sigma(T_1^4 - T_3^4)$$

$$= 0.628 \times 0.256 \times \left(5.67 \times 10^{-8}\right) \times \left(425^4 - 300^4\right)$$

$$= 223.6 \text{ W}$$

Rate of radiation heat transfer between hot cylinder and surroundings,

$\boxed{\dot{Q}_{13} = 223.6 \text{ W}}$

Question: 21.093P

Consider a long semicylindrical duct of diameter 1.0 m. Heat is supplied from the base surface, which is black, at a rate of 1200 W/m2, while the side surface with an emissivity of 0.4 is maintained at 650 K. Neglecting the end effects, determine the temperature of the base surface.

Step 1 of 5

Given

Diameter of semi cylindrical duct is $D = 1\,\text{m}$

Rate of heat supplied, $\dot{Q} = 1200\ \text{W/m}^2$

Emissivity of base surface, $\varepsilon_1 = 1$

Emissivity of side surface, $\varepsilon_2 = 0.4$

Temperature of side surface, $T_2 = 650\,\text{K}$

Step 2 of 5

Consider base surface to be surface 1 and side surface to be surface 2

Base surface area, $A_1 = 1 \times 1$

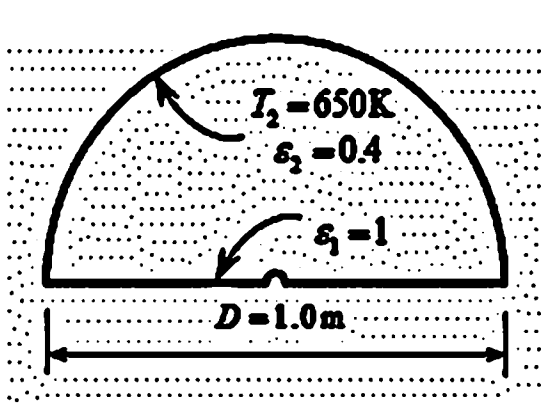
$$= 1\,\text{m}^2$$

Side surface area, $A_2 = \dfrac{\pi DL}{2}$

$$= \frac{\pi \times 1 \times 1}{2}$$

$$= 1.571\,\text{m}^2$$

Step 3 of 5



Step 4 of 5

View factor for the surface 1 with respect to itself is zero.

Then by summation rule,

$$F_{11} + F_{12} = 1$$

$$\Rightarrow 0 + F_{12} = 1$$

$$\Rightarrow F_{12} = 1$$

Step 5 of 5

Temperature of the base surface is determined by

$$\dot{Q}_{12} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\dfrac{1}{A_1 F_{12}} + \dfrac{1-\varepsilon_2}{A_2 \varepsilon_2}}$$

$$\Rightarrow 1200 = \frac{5.67 \times 10^{-8}\left(T_1^4 - 650^4\right)}{\dfrac{1}{1 \times 1} + \dfrac{1-0.4}{1.571 \times 0.4}}$$

$$\Rightarrow 2345.77 = 5.67 \times 10^{-8}\left(T_1^4 - 650^4\right)$$

$$\Rightarrow T_1^4 - 650^4 = 413.7 \times 10^8$$

$$\Rightarrow T_1 = 684.78\text{K}$$

Temperature of base surface, $\boxed{T_1 = 684.78\text{K}}$

Question: 21.094P

Consider a 30-cm-diameter hemispherical enclosure. The dome is maintained at 600 K and heat is supplied from the dome at a rate of 65 W while the base surface with an emissivity of 0.55 is maintained at 400 K. Determine the emissivity of the dome.

Step 1 of 4

Given data:

Diameter of the hemispherical enclosure, $D = 30$ cm

Temperature of the base surface, $T_1 = 400$ K

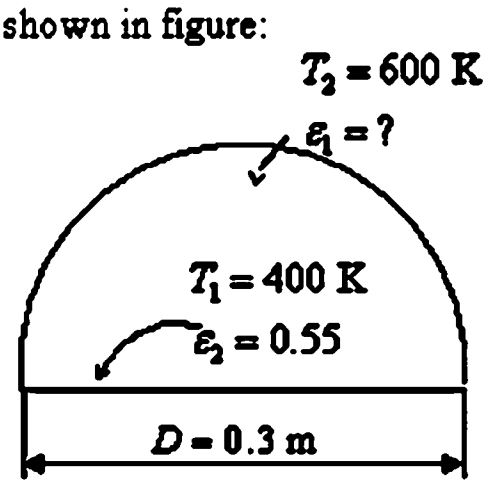
Temperature of the dome, $T_2 = 600$ K

Emissivity of the base surface, $\varepsilon_1 = 0.55$

Heat supplied from the dome, $\dot{Q}_{21} = 65$ W

Step 2 of 4

A hemispherical dome is as shown in figure:



Step 3 of 4

Consider the base surface to be surface 1, the side surface to be surface 2. This system is a two-surface enclosure. The surface areas and the view factor are determined as

$$A_1 = \frac{\pi}{4}D^2$$

$$= \frac{\pi}{4}\times 0.3^2$$

$$= 0.07069\ \text{m}^2$$

$$A_2 = \frac{\pi}{2}D^2$$

$$= \frac{\pi}{2}\times 0.3^2$$

$$= 0.1413\ \text{m}^2$$

Step 4 of 4

By using the summation rule,

$$F_{11} + F_{12} = 1$$

$$0 + F_{12} = 1$$

$$F_{12} = 1$$

The emissivity of the dome can be determined as;

$$\dot{Q}_{21} = -\dot{Q}_{12}$$

$$= -\frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$65 = -\frac{5.67\times 10^{-8}(400^4 - 600^4)}{\frac{1-0.55}{0.07069\times 0.55} + \frac{1}{0.07069} + \frac{1-\varepsilon_2}{0.1413\times\varepsilon_2}}$$

$$\varepsilon_2 = 0.0982$$

Therefore, the emissivity of the dome is $\boxed{\varepsilon_2 = 0.0982}$.

Question: 21.095P

Consider a hot cylindrical surface and a disk oriented as shown in the figure. Both the cylindrical surface and the disk behave as blackbody. The cylinder has a diameter D and a length L of 0.2 m. The disk of diameter D = 0.2 m is placed coaxially with the cylinder at a distance of L = 0.2 m apart. The cylindrical surface and the disk are maintained at uniform temperatures of 1000 K and 300 K, respectively. Determine the rate of heat transfer by radiation from cylindrical surface to the disk.

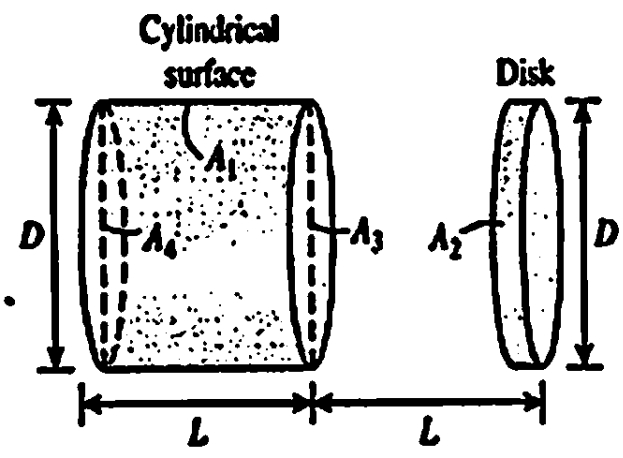


FIGURE P21–95

Step 1 of 4

Given data:

Diameter of the disk, $D = 0.2$ m

Cylinder length, $L = 0.2$ m

Distance between cylinder and disc, $L = 0.2$ m

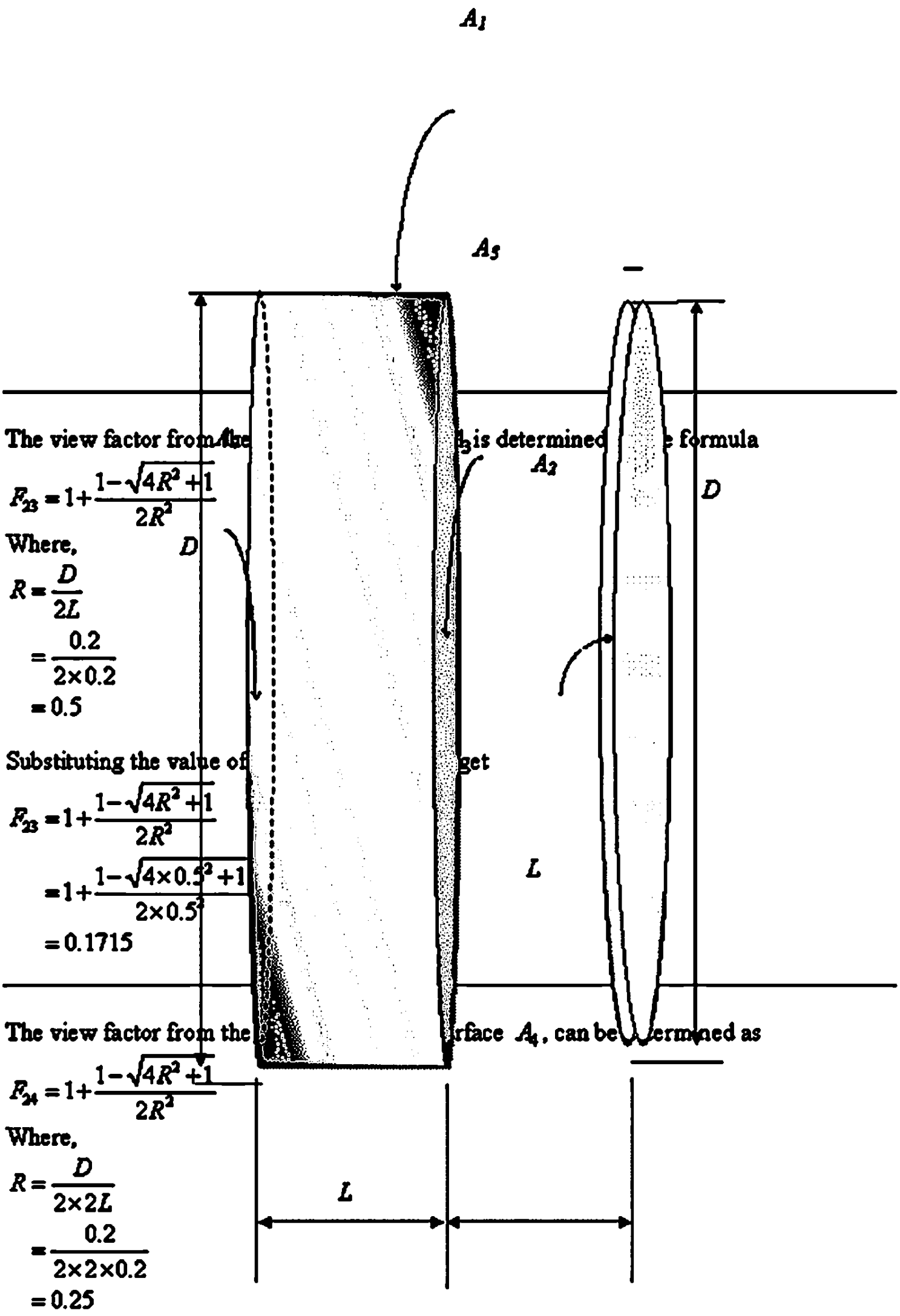
Temperature of the cylindrical surface, $T_1 = 1000$ K

Temperature of the disk, $T_2 = 300$ K

Emissivity of the cylindrical surface and the disc, $\varepsilon_1 = \varepsilon_2 = 1$

Step 2 of 4

The geometrical parameters are as shown in the figure.



The view factor from the A_3 is determined e formula

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

Where,

$$R = \frac{D}{2L}$$
$$= \frac{0.2}{2 \times 0.2}$$
$$= 0.5$$

Substituting the value of get

$$F_{23} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{4 \times 0.5^2 + 1}}{2 \times 0.5^2}$$
$$= 0.1715$$

The view factor from the rface A_4, can be ermined as

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

Where,

$$R = \frac{D}{2 \times 2L}$$
$$= \frac{0.2}{2 \times 2 \times 0.2}$$
$$= 0.25$$

Substituting the value of R in view factor, we get

$$F_{24} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$
$$= 1 + \frac{1 - \sqrt{4 \times 0.25^2 + 1}}{2 \times 0.25^2}$$
$$= 0.05573$$

Step 3 of 4

From the summation rule,

$$1 = F_{21} + F_{22} + F_{23} + F_{24}$$
$$1 = F_{21} + 0 + 0.1715 + 0.05573$$
$$F_{21} = 0.772772$$

From the reciprocity rule,

$$A_1 F_{12} = A_2 F_{21}$$
$$F_{12} = \frac{A_2}{A_1} F_{21}$$
$$= \frac{(\pi/4) D^2}{\pi D \times D} \times 0.772772$$
$$= 0.1932$$

Step 4 of 4

The net rate of radiation heat transfer from the cylindrical surface to the disc:

$$\dot{Q}_{12} = \sigma A_1 F_{12} \left(T_1^4 - T_2^4 \right)$$
$$= \left(5.67 \times 10^{-8} \right) (\pi 0.2 \times 0.2)(0.1932) \left(1000^4 - 300^4 \right)$$
$$= 1365.425 \text{ W}$$

Thus, the rate of heat transfer by radiation from the cylindrical surface to the disk is

$$\boxed{\dot{Q}_{12} = 1365.425 \text{ W}}$$

Question: 21.096P

Two parallel black disks are positioned coaxially with a distance of 0.25 m apart in a surrounding with a constant temperature of 300 K. The lower disk is 0.2 m is diameter and the upper disk is 0.4 m in diameter. If the lower disk is heated electrically at 100 W to maintain a uniform temperature of 500 K, determine the temperature of the upper disk.

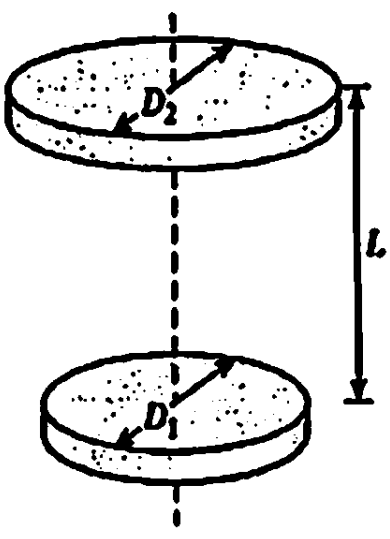


FIGURE P21–96

Step 1 of 6

Given data:

Distance between two plates, $L = 0.25$ m

Diameter of the lower disc, $D_1 = 0.2$ m

Diameter of the upper disc, $D_2 = 0.4$ m

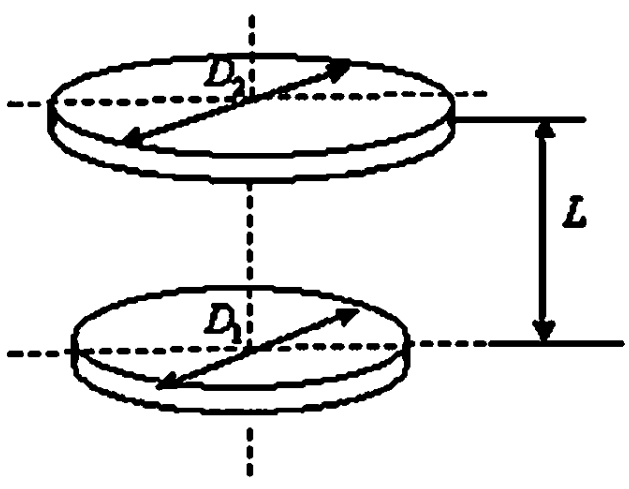
Surrounding Temperature, $T_\infty = 300$ K

Temperature of the lower disc, $T_1 = 500$ K

Heat supplied to the lower disc, $\dot{Q} = 100$ W

Step 2 of 6

The geometrical parameters are as shown in the figure:



Step 3 of 6

By considering the bottom plate as surface-1, top plate as surface-2 and the ambient as surface-3, the view factor for surface-2 from surface-1 is given by the formula

$$F_{12} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_2}{r_1}\right)^2\right]^{1/2}\right\}$$

Where

$$S = 1 + \frac{1 + R_2^2}{R_1^2}$$

$$R_1 = \frac{r_1}{L}$$
$$= \frac{0.1}{0.25}$$
$$= 0.4$$
$$R_2 = \frac{r_2}{L}$$
$$= \frac{0.2}{0.25}$$
$$= 0.8$$

Step 4 of 6

By substituting the values in equation for S, we get

$$S = 1 + \frac{1 + R_2^2}{R_1^2}$$
$$= 1 + \frac{1 + 0.8^2}{0.4^2}$$
$$= 11.25$$

The view factor can now be calculated as

$$F_{12} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_2}{r_1}\right)^2\right]^{1/2}\right\}$$
$$= \frac{1}{2}\left\{11.25 - \left[11.25^2 - 4\left(\frac{0.2}{0.1}\right)^2\right]^{1/2}\right\}$$
$$= 0.3675$$

Step 5 of 6

From the summation rule,

$$1 = F_{11} + F_{12} + F_{13}$$
$$1 = 0 + 0.3675 + F_{13}$$
$$F_{13} = 0.6325$$

The rate of radiation heat transfer from the lower plate to the ambient can be expressed as

$$\dot{Q}_{13} = A_1 F_{13} \sigma (T_1^4 - T_3^4)$$
$$= \left(\frac{\pi}{4} \times 0.2^2\right)(0.6325)\left(5.67 \times 10^{-8}\right)\left(500^4 - 300^4\right)$$
$$= 61.3\text{W}$$

The rate of radiation heat transfer from the lower plate to the upper plate is determined as

$$\dot{Q}_{12} = \dot{Q} - \dot{Q}_{13}$$
$$= 100 - 61.3$$
$$= 38.7 \text{ W}$$

Step 6 of 6

The temperature of the upper plate can be determined by using the heat transfer rate.

$$\dot{Q}_{12} = A_1 F_{12} \sigma (T_1^4 - T_2^4)$$
$$38.7 = \left(\frac{\pi}{4} \times 0.2^2\right)(0.3675)\left(5.67 \times 10^{-8}\right)\left(500^4 - T_2^4\right)$$
$$T_2^4 = \left(500^4\right) - \left(5.912 \times 10^{10}\right)$$
$$T_2 = 241.2 \text{ K}$$

Thus, the temperature of the upper plate is $\boxed{T_2 = 241.2 \text{ K}}$

Question: 21.097P

Two-phase gas-liquid oxygen is stored in a spherical tank of 1-m diameter, where it is maintained at its normal boiling point. The spherical tank is enclosed by a 1.6-m diameter concentric spherical surface at 273 K. Both spherical surfaces have an emissivity of 0.01, and the gap between the inner sphere and outer sphere is vacuumed. Assuming that the spherical tank surface has the same temperature as the oxygen, determined the heat transfer rate at the spherical tank surface.

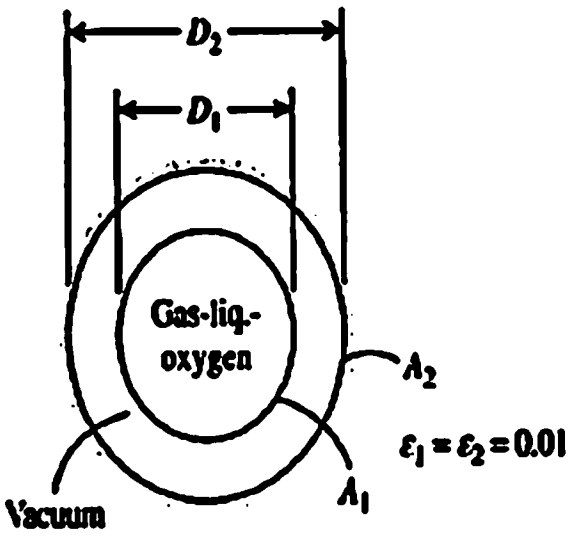


FIGURE P21-97

Step 1 of 3

Given data:

Diameter of the spherical tank, $D_1 = 1\ \text{m}$

Diameter of the concentric spherical surface, $D_2 = 1.6\ \text{m}$

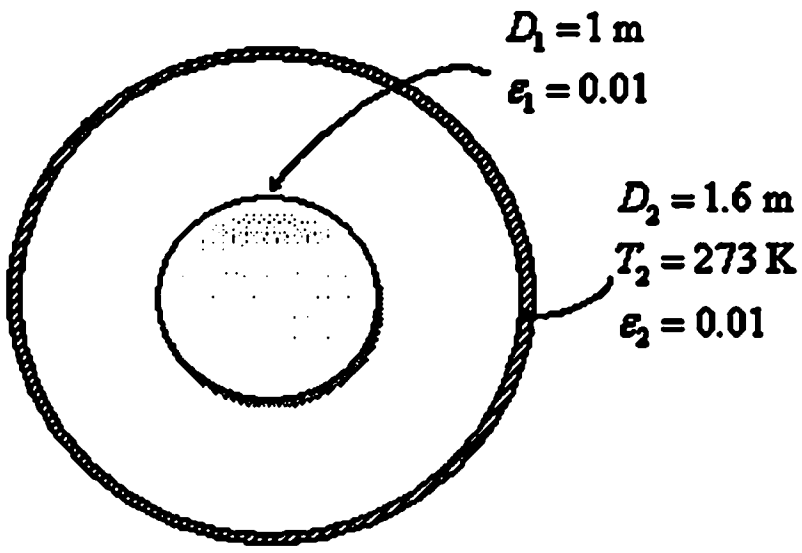
Surface temperature of the spherical tank (Boiling point), $T_1 = 241\ \text{K}$

Surface temperature of the concentric spherical surface, $T_2 = 273\ \text{K}$

Emissivity of the inner and outer spherical surfaces, $\varepsilon_1 = \varepsilon_2 = 0.01$

Step 2 of 3

The geometrical parameters are as shown in the figure:



Step 3 of 3

The radiation heat transfer rate at the spherical tank surface is given by the formula

$$\dot{Q}_{12} = \frac{A_1\sigma(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1^2}{r_2^2}\right)}$$

$$= \frac{(\pi \times 1^2)(5.67\times10^{-8})(241^4 - 273^4)}{\frac{1}{0.01} + \frac{1-0.01}{0.01}\left(\frac{0.5^2}{0.8^2}\right)}$$

$$= \frac{-388.53}{100 + 38.672}$$

$$= -2.8\ \text{W}$$

Thus the heat transfer rate at the spherical tank surface is $\boxed{\dot{Q}_{12} = -2.8\ \text{W}}$

Question: 21.098P

Daylight and incandescent light may be approximated as a blackbody at the effective surface temperatures of 5800 K and 2800 K, respectively. Determine the wavelength at maximum emission of radiation for each of the lighting sources.

Step 1 of 3

Given data:

The temperature of daylight, $T_d = 5800\ \text{K}$

Incandescent light, $T_i = 2800\ \text{K}$

Step 2 of 3

The wavelength at which the emission of radiation takes a peak value can be determined from Wien's displacement.

For a daylight

$$(\lambda_1 T_d)_{max_power} = 2897.8\ \mu\text{m.K}$$

$$(\lambda_1)_{max_power} = \frac{2897.8\ \mu\text{m.K}}{5800}$$

$$= 0.5\ \mu\text{m}$$

Thus, the maximum power in daylight is observed at $\boxed{(\lambda_1)_{max_power} = 0.5\ \mu\text{m}}$

Step 3 of 3

For an incandescent light $T_i = 2800\ \text{K}$

$$(\lambda_1 T_i)_{max_power} = 2897.8\ \mu\text{m.K}$$

$$(\lambda_1)_{max_power} = \frac{2897.8\ \mu\text{m.K}}{2800}$$

$$= 1.035\ \mu\text{m}$$

Thus, the maximum power in incandescent light is observed at $\boxed{(\lambda_1)_{max_power} = 1.035\ \mu\text{m}}$

Question: 21.099P

A horizontal opaque flat plate is well insulated on the edges and the lower surface. The top surface has an area of 5 m2, and it experiences uniform irradiation at a rate of 5000 W. The plate absorbs 4000 W of the irradiation, and the surface is losing heat at a rate of 500 W by convection. If the plate maintains a uniform temperature of 350 K, determine the absorptivity, reflectivity, and emissivity of the plate.

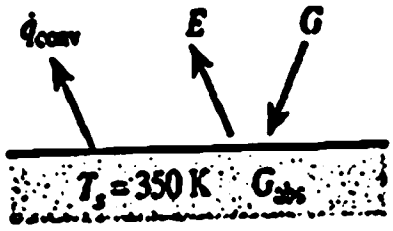


FIGURE P21-99

Step 1 of 3

Given data:

The surface area of the plate, $A = 5\,\text{m}^2$

The irradiation on the surface, $G = 5000$ W

The irradiation absorbed by the surface, $G_{abs} = 4000$ W

Heat lost by convection, $\dot{q}_{conv} = 500$ W

Surface temperature of the plate, $T_s = 350$ K

For opaque surfaces, the transmissivity is zero ($\tau = 0$)

Step 2 of 3

The absorptivity of the surface is the ratio of energy absorbed by the surface to the energy incident on the surface.

$$\alpha = \frac{G_{abs}}{G}$$
$$= \frac{4000}{5000}$$
$$= 0.8$$

Hence the absorptivity of the surface is $\boxed{\alpha = 0.8}$

The sum of the absorbed, reflected and transmitted energies is unity. Hence

$$1 = \alpha + \rho + \tau$$
$$1 = 0.8 + \rho + 0$$
$$\rho = 0.2$$

Hence the reflectivity of the surface is $\boxed{\rho = 0.2}$

Step 3 of 3

From energy balance,

$$G_{abs} = \dot{q}_{conv} + \dot{q}_{rad}$$
$$G_{abs} = \dot{q}_{conv} + \varepsilon A \sigma T_s^4$$
$$\varepsilon = \frac{G_{abs} - \dot{q}_{conv}}{A \sigma T_s^4}$$
$$= \frac{4000 - 500}{5 \times \left(5.67 \times 10^{-8}\right) \times 350^4}$$
$$= 0.8227$$

Hence the emissivity of the surface is $\boxed{\varepsilon = 0.8227}$

Question: 21.100P

The spectral emissivity of an opaque surface at 1200 K is approximated as

$\varepsilon_1 = 0 \quad \text{for} \quad \lambda < 2\,\mu\text{m}$
$\varepsilon_2 = 0.85 \quad \text{for} \quad 2 \le \lambda \le 6\,\mu\text{m}$
$\varepsilon_3 = 0 \quad \text{for} \quad \lambda > 6\,\mu\text{m}$

Determine the total emissivity and the emissive flux of the surface.

Step 1 of 4

Given
Spectral Emissivity of an opaque surface can be expressed as
$\varepsilon_1 = 0 \quad \text{for} \quad \lambda < 2\mu\text{m}$
$\varepsilon_2 = 0.85 \text{ for} \quad 2 \le \lambda \le 6\ \mu\text{m}$
$\varepsilon_3 = 0 \quad \text{for} \quad \lambda > 6\mu\text{m}$
Temperature of opaque surface, $T = 1500\,\text{K}$

Step 2 of 4

The emissivity of the surface is calculated as follows
$\lambda_1 T = 2 \times 1200$
$= 2400\,\mu\text{mK}$
$\lambda_2 T = 6 \times 1200$
$= 7200\,\mu\text{mK}$
From table, Black body radiation functions
The blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined as
$f_{\lambda_1} = 0.140256$
$f_{\lambda_2} = 0.819217$

Step 3 of 4

Total emissivity of the surface becomes
$$\varepsilon(T) = \varepsilon_1 f_{\lambda_1} + \varepsilon_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \varepsilon_3 \left(1 - f_{\lambda_2}\right)$$
$$= 0.0 \times 0.140256 + 0.85(0.819217 - 0.140256) + 0.0(1 - 0.819217)$$
$$= 0 + 0.57712 + 0$$
$$= 0.57712$$
Total emissivity of the surface, $\boxed{\varepsilon(T) = 0.57712}$

Step 4 of 4

Emissive flux of the surface is given by
$$E = \varepsilon \sigma T^4$$
$$= 0.57712 \times \left(5.67 \times 10^{-8}\right) \times 1200^4$$
$$= 67853.8\,\text{W/m}^2$$
Emissive flux of the surface, $\boxed{E = 67853.8\,\text{W/m}^2}$

Question: 21.101P

The spectral transmissivity of a glass cover used in a solar collector is given as

$\tau_1 = 0$ for $\lambda < 0.3\ \mu m$
$\tau_2 = 0.9$ for $0.3 < \lambda < 3\ \mu m$
$\tau_3 = 0$ for $\lambda > 3\ \mu m$

Solar radiation is incident at a rate of 950 W/m2, and the absorber plate, which can be considered to be black, is maintained at 340 K by the cooling water. Determine (*a*) the solar flux incident on the absorber plate, (*b*) the transmissivity of the glass cover for radiation emitted by the absorber plate, and (*c*) the rate of heat transfer to the cooling water if the glass cover temperature is also 340 K.

Step 1 of 6

Given data:

The temperature of the surface $T = 340\ K$

Incident radiation is $q = 950\ W/m^2$

The transmissivity of the cover for $\lambda < 0.3\ \mu m$, $\tau_1 = 0$

The transmissivity of the cover for $0.3\ \mu m < \lambda < 3\ \mu m$, $\tau_2 = 0.9$

The transmissivity of the cover for $\lambda > 3\ \mu m$, $\tau_3 = 0$

Step 2 of 6

a)

The temperature of solar radiation is $T = 5800\ K$

Radiation function corresponding to the temperature of 5800K,

$$\lambda_1 T = (0.3)(5800)$$
$$= 1740\ \mu m \cdot K$$
$$\lambda_2 T = (3)(5800)$$
$$= 17400\ \mu m \cdot K$$

From the black body radiation function, at $\lambda_1 T = 1740\ \mu m \cdot K$

$f_{\lambda_1} = 0.033454$

From the black body radiation function, at $\lambda_2 T = 17400\ \mu m \cdot K$

$f_{\lambda_2} = 0.97874$

Step 3 of 6

The average transmissivity of the surface is

$$\tau_s = \tau_1 f_{\lambda_1} + \tau_2 (f_{\lambda_2} - f_{\lambda_1}) + \tau_3 (1 - f_{\lambda_2})$$
$$= (0)(0.033454) + (0.9)(0.97874 - 0.033454) + (0.0)(1 - 0.97874)$$
$$= 0.851$$

Now radiation absorbed by the plate is

$$E_{abs} = 0.851 \times 950$$
$$= 808.45\ W/m^2$$

Therefore, solar flux incident on absorber plate is $\boxed{E_{abs} = 808.45\ W/m^2}$

Step 4 of 6

b)

The radiation emitted by the absorber plate is $T = 340\ K$

Radiation function corresponding to the temperature of 340K

$$\lambda_1 T = (0.3)(340)$$
$$= 102\ \mu m K$$
$$\lambda_2 T = (3)(340)$$
$$= 1020\ \mu m K$$

From the black body radiation function, at $\lambda_1 T = 102\ \mu m \cdot K$

$f_{\lambda_1} = 0.0$

From the black body radiation function, at $\lambda_2 T = 1020\ \mu m \cdot K$

$f_{\lambda_2} = 0.0005023$

Step 5 of 6

The average transmissivity of the surface is

$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 (f_{\lambda_2} - f_{\lambda_1}) + \tau_3 (1 - f_{\lambda_2})$$
$$= (0)(0.0) + (0.9)(0.0005023 - 0.0) + (0.0)(1 - 0.0005023)$$
$$= 0.0004520$$

Therefore, the transmissivity of the glass cover for the radiation emitted by the absorber plate is $\boxed{\tau_r = 0.0004520}$

Step 6 of 6

c)

The rate of heat transfer to the cooling water is the difference between the radiation absorbed by the absorber plate and radiation emitted by the absorber plate.

$$Q_{cw} = (\tau_s - \tau_r) q$$
$$= (0.851 - 0.0004520)950$$
$$= 808.02\ W/m^2$$

Therefore, rate of heat transfer to the cooling water is $\boxed{Q_{cw} = 808.02\ W/m^2}$

Question: 21.102P

The spectral absorptivity of an opaque surface is as shown on the graph. Determine the absorptivity of the surface for radiation emitted by a source at (a) 1000 K and (b) 3000 K.

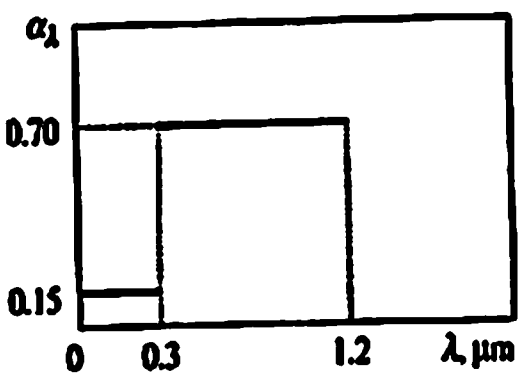


FIGURE P21-102

Step 1 of 5

Given data:

(a) T = 1000 K

Average absorptivity of the surface for radiation at calculated as follows

From Blackbody radiation function f_λ

The fraction of radiation in the range of 0.3 to 1.2 μm is

$\lambda_1 T = (0.3\ \mu m)(1000\ K)$

$=300\ \mu mK$

Step 2 of 5

From Blackbody radiation function f_λ

At $\lambda_1 T = 300\ \mu mK$

$f_{\lambda_1} = 0$

$\lambda_2 T = (1.2\ \mu m)(1000\ K)$

$=1200\ \mu mK$

From Blackbody radiation function f_λ

$\lambda_2 T = 1200\ \mu mK$

$f_{\lambda_2} = 0.002134$

Step 3 of 5

Average absorptivity of the surface is determined by

$$\alpha(T) = \alpha_1 f_{\lambda 1} + \alpha_2 (f_{\lambda 2} - f_{\lambda 1}) + \alpha_3 (1 - f_{\lambda 2})$$

$$= (0.15)(0) + (0.7)(0.002134 - 0) + (0)(1 - 0.002134)$$

$\alpha(T) = 0.0015$

Therefore, Average absorptivity of the surface is $\alpha(T) = 0.0015$

Step 4 of 5

(b) T = 3000 K

Similarly the absorptivity of the surface is determined as

From Table 12-2 Blackbody radiation function f_λ

The fraction of radiation in the range of 0.3 to 1.2 μm is

$\lambda_1 T = (0.3\ \mu m)(3000\ K)$

$=900\ \mu mK$

From Blackbody radiation function f_λ

At $\lambda_1 T = 900\ \mu mK$

$f_{\lambda_1} = 0.00017$

Step 5 of 5

The fraction of radiation is 1.2 μm

$\lambda_2 T = (1.2\ \mu m)(3000\ K)$

$=3600\ \mu mK$

From Blackbody radiation function f_λ

$\lambda_2 T = 3600\ \mu mK$

$f_{\lambda_2} = 0.403607$

Question: 21.103P

The surface in Prob. 21–102 receives solar radiation at a rate of 470 W/m2. Determine the solar absorptivity of the surface and the rate of absorption of solar radiation.

Step 1 of 3

Given

The solar radiation on the surface $I = 470\ \text{W/m}^2$

We know the surface of the sun is $T = 5800\ \text{K}$

Step 2 of 3

Average absorptivity of the surface for radiation at calculated as follows

The fraction of radiation in the range of 0.3 to 1.2 μm is

$\lambda_1 T = (0.3\ \mu\text{m})(5800\ \text{K})$

$= 1740\ \mu\text{mK}$

From Blackbody radiation function f_λ

$f_{\lambda_1} = 0.0335$

$\lambda_2 T = (1.2\ \mu\text{m})(5800\ \text{K})$

$= 6960\ \mu\text{mK}$

From Blackbody radiation function f_λ

$f_{\lambda_2} = 0.806$

Step 3 of 3

Average absorptivity of the surface is determined by

$$\alpha(T) = \alpha_1 f_{\lambda 1} + \alpha_2 (f_{\lambda 2} - f_{\lambda 1}) + \alpha_3 (1 - f_{\lambda 2})$$

$$= (0.15)(0.0335) + (0.7)(0.806 - 0.0335) + (0)(1 - 0.806)$$

$$= \alpha(T) = 0.546$$

Therefore, Average absorptivity of the surface is $\boxed{\alpha(T) = 0.546}$

Question: 21.104P

The spectral transmissivity of a 3-mm-thick regular glass can be expressed as

$\tau_1 = 0$ for $\lambda < 0.35\ \mu m$
$\tau_2 = 0.85$ for $0.35 < \lambda < 2.5\ \mu m$
$\tau_3 = 0$ for $\lambda > 2.5\ \mu m$

Determine the transmissivity of this glass for solar radiation. What is the transmissivity of this glass for light?

Step 1 of 5

Given
Thickness of glass, $t = 3mm$
Spectral transmissivity of glass can be expressed as
$\tau_1 = 0$ for $\lambda < 0.35\mu m$
$\tau_2 = 0.85$ for $0.35 \le \lambda \le 2.5\ \mu m$
$\tau_3 = 0$ for $\lambda > 2.5\mu m$

Step 2 of 5

The transmissivity of this glass for the solar radiation at $T = 5800K$ calculated as follows
$\lambda_1 T = 0.35 \times 5800$
$= 2030\mu mK$
$\lambda_2 T = 2.5 \times 5800$
$= 14500\mu mK$
From table, Black body radiation functions
The blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined as
$f_{\lambda_1} = 0.071852$
$f_{\lambda_2} = 0.966440$

Step 3 of 5

Transmissivity of this glass,
$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \tau_3 \left(1 - f_{\lambda_2}\right)$$
$$= 0.0 \times 0.071852 + 0.85(0.966440 - 0.071852) + 0.0(1 - 0.966440)$$
$$= 0 + 0.7604 + 0$$
$$= 0.7604$$
Transmissivity of this glass for solar radiation, $\boxed{\tau(T) = 0.7604}$

Step 4 of 5

The transmissivity of this glass for the light at $T = 300K$ calculated as follows
$\lambda_1 T = 0.35 \times 300$
$= 105\mu mK$
$\lambda_2 T = 2.5 \times 300$
$= 750\mu mK$
From table, Black body radiation functions
The blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$ are determined as
$f_{\lambda_1} = 0.00$
$f_{\lambda_2} = 0.000012$

Step 5 of 5

Transmissivity of this glass,
$$\tau(T) = \tau_1 f_{\lambda_1} + \tau_2 \left(f_{\lambda_2} - f_{\lambda_1}\right) + \tau_3 \left(1 - f_{\lambda_2}\right)$$
$$= 0.0 \times 0.00 + 0.85(0.000012 - 0.00) + 0.0(1 - 0.000012)$$
$$= 0 + 0.0000102 + 0$$
$$= 0.0000102$$
Transmissivity of this glass for light, $\boxed{\tau(T) = 0.0000102}$

Question: 21.105P

A 1-m-diameter spherical cavity is maintained at a uniform temperature of 600 K. Now a 5-mm-diameter hole is drilled. Determine the maximum rate of radiation energy streaming through the hole. What would your answer be if the diameter of the cavity were 3 m?

Step 1 of 3

Given

Diameter of spherical cavity, $D = 1\text{m}$

Diameter of the drilled hole, $d = 5\text{mm}$

Temperature of cavity, $T = 600\text{ K}$

Step 2 of 3

Area of the hole, $A = \dfrac{\pi d^2}{4}$

$$= \frac{\pi \times 0.005^2}{4}$$

$$= 1.96 \times 10^{-5}\text{ m}^2$$

Maximum rate of radiation energy streaming through the hole is given by

$$E = A\sigma T^4 \text{ -------(1)}$$

$$= 1.96 \times 10^{-5} \times \left(5.67 \times 10^{-8}\right) \times 600^4$$

$$= 0.144\text{ W}$$

Maximum rate of radiation energy streaming through the hole, $\boxed{E = 0.144\text{ W}}$

Step 3 of 3

Now, diameter of spherical cavity, $D = 3\text{m}$

From equation (1)

Rate of radiation energy doesn't depend up on diameter of spherical cavity but, depend on diameter of the drilled hole because of we require radiation energy through hole.

So, we get the same answer.

Maximum rate of radiation energy streaming through the hole, $\boxed{E = 0.144\text{ W}}$

Question: 21.106P

The human skin is "selective" when it comes to the absorption of the solar radiation that strikes it perpendicularly. The skin absorbs only 50 percent of the incident radiation with wavelengths between λ1 =0.517 μm and λ2 = 1.552 . The radiation with wavelengths shorter than λ1 and longer than λ2 is fully absorbed. The solar surface may be modeled as a blackbody with effective surface temperature of 5800 K. Calculate the fraction of the incident solar radiation that is absorbed by the human skin.

Step 1 of 3

Given data:

The surface of the sun is $T = 5800\ \text{K}$

The absorptivity of the skin for $\lambda < 0.517\ \mu\text{m}$, $\alpha_1 = 1$

The absorptivity of the skin for $0.517\ \mu\text{m} < \lambda < 1.552\ \mu\text{m}$, $\alpha_2 = 0.5$

The absorptivity of the skin for $\lambda > 1.552\ \mu\text{m}$, $\alpha_3 = 1$

Step 2 of 3

Radiation function corresponding to the temperature of 5800 K,

$$\lambda_1 T = (0.517)(5800)$$
$$= 2998.6\ \mu\text{m}\cdot\text{K}$$
$$\lambda_2 T = (1.552)(5800)$$
$$= 9001.6\ \mu\text{m}\cdot\text{K}$$

From the black body radiation function, at $\lambda_1 T = 2998.6\ \mu\text{m}\cdot\text{K}$

$f_{\lambda 1} = 0.273232$

From the black body radiation function, at $\lambda_2 T = 9001.6\ \mu\text{m}\cdot\text{K}$

$f_{\lambda 2} = 0.890029$

Step 3 of 3

The average absorptivity of the skin is

$$\alpha(T) = \alpha_1 f_{\lambda 1} + \alpha_2 (f_{\lambda 2} - f_{\lambda 1}) + \alpha_3 (1 - f_{\lambda 2})$$
$$= (1)(0.273232) + (0.5)(0.890029 - 0.273232) + (1)(1 - 0.890029)$$
$$= 0.6916$$

Therefore, the average absorptivity of the skin is $\boxed{\alpha(T) = 0.6916}$

Question: 21.107P

Consider a cylindrical enclosure with *A*1, *A*2 and *A*3 representing the internal base, top, and, side surfaces, respectively. Using the length to diameter ratio, *K* = L/D, determine (*a*) the expression for the view factor from the side surface to itself F33 in terms of K and (*b*) the value of the view factor *F*33 for *L* = *D*.

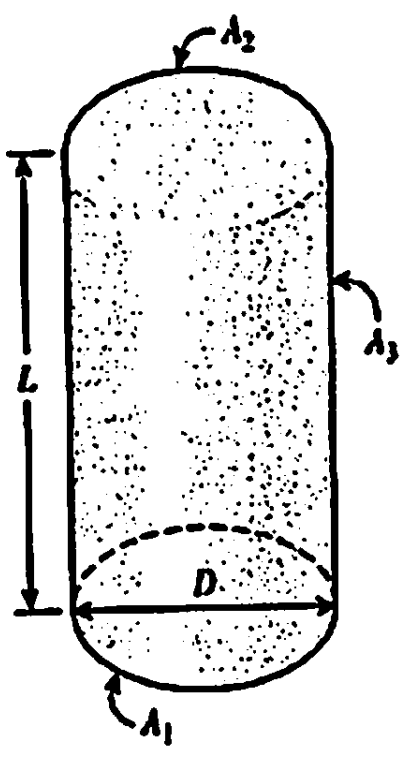


FIGURE P21–107

Step 1 of 8

Given data:

The internal base area of the cylinder is A_1

The top surface area of the cylinder is A_2

The side surface area of the cylinder is A_3

Length to diameter ratio is $K = \frac{L}{D}$

Step 2 of 8

a) For coaxial parallel disks from the table of 'view factor expressions for some common geometrics of finite size', the expression for view factor is

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

Where,

$$R = \frac{D}{2L}$$

$$= \frac{1}{2K}$$

Step 3 of 8

Therefore, the view factor from the base to the top is written as

$$F_{12} = 1 + \frac{1 - \sqrt{4R^2 + 1}}{2R^2}$$

$$= 1 + \frac{1 - \sqrt{4(1/2K)^2 + 1}}{2(1/2K)^2}$$

$$= 1 + \frac{1 - \sqrt{1/K^2 + 1}}{1/2K^2}$$

$$= 1 + 2K\left(K - \sqrt{K^2 + 1}\right)$$

Step 4 of 8

By using the reciprocity relation, the view factor from the top surface to the base surface can be expressed as

$$F_{21} = \frac{A_1}{A_2} F_{12}$$

$$= \frac{A_1}{A_1} \times \left[1 + 2K\left(K - \sqrt{K^2 + 1}\right)\right]$$

$$= 1 + 2K\left(K - \sqrt{K^2 + 1}\right)$$

As the surface-1 is flat, by consider the view factor of surface-1 with respect to itself, we get

$$F_{11} = 0$$

Step 5 of 8

The view factor from the base to the side surface is determined by applying the summation rule.

$$1 = F_{11} + F_{12} + F_{13}$$

$$F_{13} = 1 - F_{11} - F_{12}$$

$$= 1 - 0 - \left(1 + 2K\left(K - \sqrt{K^2 + 1}\right)\right)$$

$$= 2K\sqrt{K^2 + 1} - 2K^2$$

Step 6 of 8

By using the reciprocity rule, the view factor from the side surface to the base surface/top surface can be expressed as

$$F_{31} = \frac{A_3}{A_3} F_{13}$$

$$= \frac{\pi D^2/4}{\pi DL} \times \left(2K\sqrt{K^2 + 1} - 2K^2\right)$$

$$= \frac{1}{4K} \times \left(2K\sqrt{K^2 + 1} - 2K^2\right)$$

$$= \frac{1}{2}\left(\sqrt{K^2 + 1} - K\right)$$

Step 7 of 8

The view factor from the side surface to itself is determined by applying the summation.

$$1 = F_{31} + F_{32} + F_{33}$$

$$1 = F_{31} + F_{31} + F_{33}$$

$$F_{33} = 1 - 2F_{31}$$

$$= 1 - 2\left(\frac{1}{2}\right)\left(\sqrt{K^2 + 1} - K\right)$$

$$= 1 - \left(\sqrt{K^2 + 1} - K\right)$$

Thus the expression for view factor from side surface to itself is $\boxed{F_{33} = 1 - \left(\sqrt{K^2 + 1} - K\right)}$

Step 8 of 8

b)

The relation between the length and diameter is given as $L = D$

Thus the value of K becomes,

$$K = \frac{L}{D}$$

$$= \frac{D}{D}$$

$$= 1$$

The expression for view factor from side surface to itself is

$$F_{33} = 1 - \left(\sqrt{K^2 + 1} - K\right)$$

$$= 1 - \left(\sqrt{1^2 + 1} - 1\right)$$

$$= 1.414$$

Thus the view factor from side surface to itself is $\boxed{F_{33} = 1.414}$

Question: 21.108P

A long cylindrical black surface fuel rod of diameter 25 mm is shielded by a surface concentric to the rod. The shield has diameter of 50 mm and a constant surface temperature of 320 K. Both inner and outer surfaces of the shield have an emissivity of 0.05, and the gap between the fuel rod and the shield is a vacuum. If the fuel rod is generating 0.5 MW/m3 of heat per unit length, determine the surface temperature of the fuel rod.

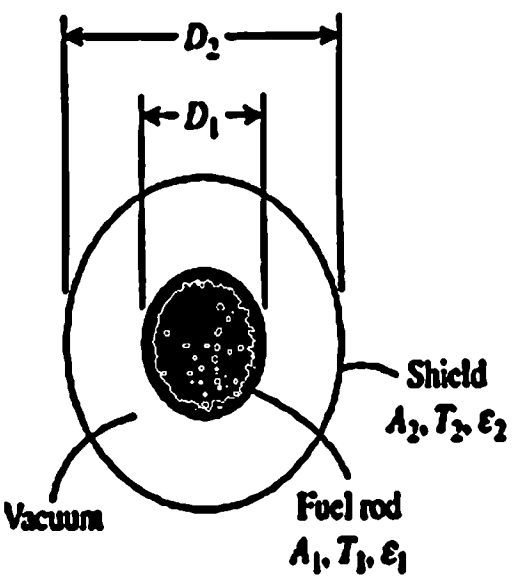


FIGURE P21–108

Suppose another (infinitely) large disk *c*, of negligible thickness and $s = 0.7$, is inserted between disks *a* and *b* such that it is parallel and equidistant to both disks. Calculate the net rate of radiation heat exchange between disks *a* and *c* and disks *c* and *b* in steady operation.

Step 1 of 3

Given data:

Diameter of the fuel rod, $D_1 = 25\,\text{mm}$

Diameter of the outer shield, $D_2 = 50\,\text{mm}$

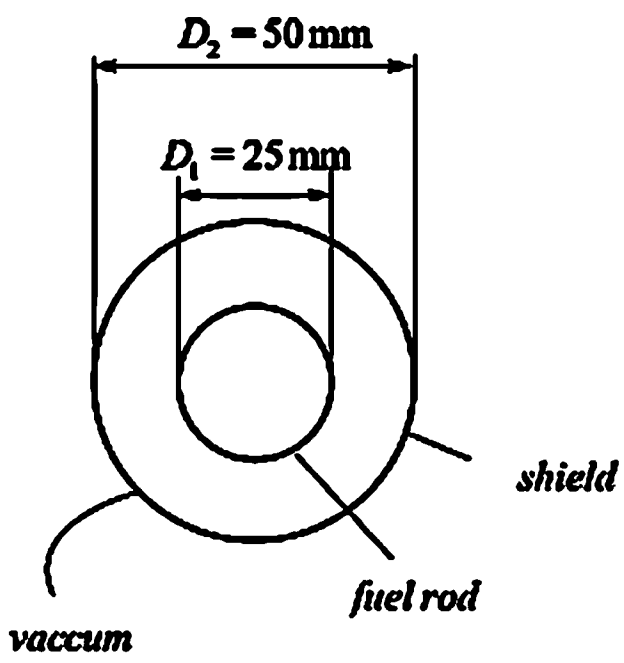
Temperature of the outer shield, $T_2 = 320\ \text{K}$

Emissivity of the radiation shield on inner and outer surfaces, $\varepsilon_2 = 0.05$

Heat generation by the fuel rod, $\dot{q}_{gen} = 0.5\ \text{MW/m}^3$

Step 2 of 3

Given figure:



Step 3 of 3

Calculation of surface temperature of fuel rod

By applying the energy balance at the surface of the fuel rod, we get,

$$\dot{E}_{in} - \dot{E}_{out} + \dot{E}_{gen} = 0$$

$$\dot{Q}_{rad\ from\ shield} - \dot{Q}_{rad\ from\ fuelrod} + \dot{Q}_{gen} = 0$$

$$\varepsilon_2 A_2 \sigma T_2^4 - A_1 \sigma T_1^4 + A_1 L \dot{q}_{gen} = 0 \qquad (\because \varepsilon_1 = 1)$$

Therefore, the surface temperature of the fuel rod can be expressed as

$$T_1^4 = \frac{\varepsilon_2 A_2 \sigma T_2^4 + A_1 L \dot{q}_{gen}}{A_1 \sigma}$$

$$T_1^4 = \frac{\left[0.05 \times (\pi \times 0.05) \times 5.67 \times 10^{-8} \times 320^4\right] + \left[\left(\frac{\pi}{4} \times 0.025^2\right) \times 0.5 \times 10^6\right]}{(\pi \times 0.025) \times 5.67 \times 10^{-8}}$$

$$T_1^4 = \frac{4.6695 + 245.43}{4.453 \times 10^{-9}}$$

$$T_1^4 = 5.616 \times 10^{10}$$

$$T_1 = 486.816\ \text{K}$$

Therefore, the surface temperature of the fuel rod is $\boxed{T_1 = 487\ \text{K}}$

Question: 21.109P

A dryer is shaped like a long semi-cylindrical duct of diameter 1.5 m. The base of the dryer is occupied with water soaked materials to be dried, and maintained at a temperature of 370 K and emissivity of 0.5. The dome of the dryer is maintained at 1000 K with emissivity of 0.8. Determine the drying rate per unit length experienced by the wet materials.

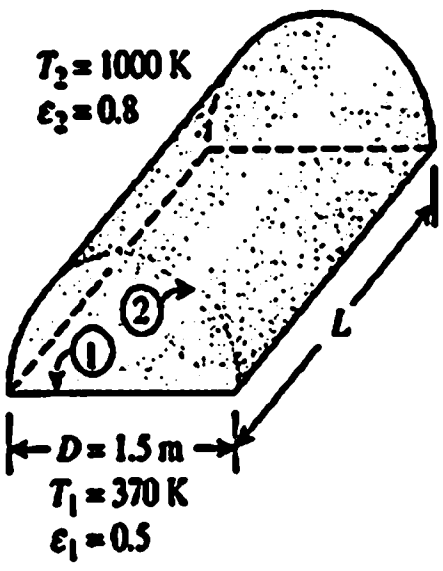


FIGURE P21–109

Step 1 of 4

Given data:

Diameter of the duct, $D = 1.5$ m

Base temperature, $T_1 = 370$ K

Dome temperature, $T_2 = 1000$ K

Emissivity of the base surface, $\varepsilon_1 = 0.5$

Emissivity of the dome surface, $\varepsilon_2 = 0.8$

Step 2 of 4

Calculation of drying rate per unit length:

The drying rate per unit length can be calculated as follows,

From the properties of saturated water,

By interpolating the values of latent heat of vaporization at $T = 95$ °C $T = 100$ °C, we get the latent heat of vaporization at $T = 97$ °C as follows,

$$h_{fg\ at\ 97^\circ C} = h_{fg\ at\ 95^\circ C} + \left(h_{fg\ at\ 100^\circ C} - h_{fg\ at\ 95^\circ C}\right)\left(\frac{97-95}{100-95}\right)$$

$$= 2269.6 + \left[(2256.4 - 2269.6) \times 0.4\right]$$

$$= 2269.6 - 5.28$$

$$= 2264.32 \text{ kJ/kg}$$

Therefore, the Latent heat of vaporization at $T_1 = 370$ K is, $h_{fg} = 2256$ kJ/kg

Since, Surface 1 is flat, $F_{11} = 0$

The view factor from the base surface A_1 to the cylindrical dome A_2 can be determined by using the summation rule as,

$$F_{11} + F_{12} = 1$$

$$0 + F_{12} = 1$$

$$F_{12} = 1$$

Step 3 of 4

The radiation heat exchange between the dome and the base surface per unit length can be determined as,

$$\dot{Q}_{12} = -\frac{\sigma\left(T_1^4 - T_2^4\right)}{\left(\frac{1-\varepsilon_1}{A_1\varepsilon_1} + \frac{1}{F_{12}A_1} + \frac{1-\varepsilon_2}{A_2\varepsilon_2}\right)}$$

$$= -\frac{5.67\times10^{-8}\times\left(370^4 - 1000^4\right)}{\left(\frac{1-0.5}{1.5\times1\times0.5} + \frac{1}{1\times1.5\times1} + \frac{1-0.8}{\frac{\pi}{2}\times1.5\times1\times0.8}\right)}$$

$$= \frac{55.63\times10^3}{1.4394}$$

$$= 38209.39 \text{ W/m}$$

$$\dot{Q}_{12} = 38209.39 \text{ W/m} \qquad \ldots\ldots (1)$$

Step 4 of 4

The amount of heat required for drying can be expressed as,

$$\dot{Q} = \dot{m}h_{fg} \qquad \ldots\ldots (2)$$

The drying rate per unit length can be determined by equating the equations (1) and (2) as,

$$\dot{Q}_{12} = \dot{Q}$$

$$38209.39 = \dot{m}h_{fg}$$

$$\dot{m} = \frac{38209.39}{h_{fg}}$$

$$= \frac{38209.39}{2256\times10^3}$$

$$\dot{m} = 0.0169 \text{ kg/s-m}$$

The drying rate per unit length experienced by the wet material is $\boxed{\dot{m} = 0.0169 \text{ kg/s-m}}$

Question: 21.110P

Two concentric spheres of diameters $D1$ = 15 cm and $D2$ = 25 cm are separated by air at 1 atm pressure. The surface temperatures of the two spheres enclosing the air are $T1$ = 350 K and $T2$ = 275 K, respectively, and their emissivities are 0.75. Determine the rate of heat transfer from the inner sphere to the outer sphere by (*a*) natural convection and (*b*) radiation.

Step 1 of 8

Given

Diameter of inner sphere, $D_1 = 15$ cm

Diameter of outer sphere, $D_2 = 25$ cm

Surface temperature of inner sphere, $T_1 = 350$ K

Surface temperature of outer sphere, $T_2 = 275$ K

Emissivity of each surface, $\varepsilon = 0.75$

Step 2 of 8

(a) The characteristic length, $L_c = \frac{1}{2}(D_2 - D_1)$

$$= \frac{1}{2}(0.25 - 0.15)$$

$$= 0.05\text{m}$$

Film temperature, $T_f = \frac{T_1 + T_2}{2}$

$$= \frac{350 + 275}{2}$$

$$= 312.5\text{K}$$

Step 3 of 8

From table, Properties of air at 1 atm pressure

At $T_f = 312.5$K

Kinematic viscosity of air, $\upsilon = 1.697 \times 10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7256$

Thermal conductivity, $k = 0.02658$ W/m.K

Now, $\beta = \frac{1}{T_f}$

$$= \frac{1}{312.5}$$

$$= 0.0032\text{K}^{-1}$$

Step 4 of 8

Rayleigh number is given by

$$Ra = \frac{g\beta(T_s - T_\infty)L_c^3}{\upsilon^2}\text{Pr}$$

$$= \frac{9.81 \times 0.0032 \times (350 - 275) \times 0.05^3 \times 0.7256}{\left(1.697 \times 10^{-5}\right)^2}$$

$$= \frac{2.13 \times 10^{-4}}{\left(1.697 \times 10^{-5}\right)^2}$$

$$= 7.41 \times 10^5$$

Step 5 of 8

The geometric factor for concentric spheres is given by

$$F_{sph} = \frac{L_c}{(D_1 D_2)^4 \left(D_1^{-3/5} + D_2^{-3/5}\right)^5}$$

$$= \frac{0.05}{[0.15 \times 0.25]^4 \left[0.15^{-3/5} + 0.25^{-3/5}\right]^5}$$

$$= \frac{0.05}{8.47}$$

$$= 5.9 \times 10^{-3}$$

Step 6 of 8

The effective thermal conductivity is given by

$$k_{eff} = 0.74k\left(\frac{\text{Pr}}{0.861 + \text{Pr}}\right)^{1/4}(F_{sph}Ra)^{1/4}$$

$$= 0.74 \times 0.02658\left(\frac{0.7256}{0.861 + 0.7256}\right)^{1/4}\left[0.0059 \times 7.41 \times 10^5\right]^{1/4}$$

$$= 0.1315\ \text{W/m.K}$$

Then the rate oh heat transfer becomes

$$\dot{Q} = k_{eff}\pi\left(\frac{D_1 D_2}{L_c}\right)(T_1 - T_2)$$

$$= 0.1315 \times \pi\left[\frac{0.15 \times 0.25}{0.05}\right](350 - 275)$$

$$= 23.24\,\text{W}$$

Rate of heat transfer by natural convection, $\boxed{\dot{Q} = 23.24\,\text{W}}$

Step 7 of 8

(b) Surface area of the inner sphere, $A_1 = \pi D_1^2$

$$= \pi \times 0.15^2$$

$$= 0.0707\ \text{m}^2$$

Step 8 of 8

Rate of heat transfer is given by

$$\dot{Q}_{12} = \frac{A_1\sigma\left(T_2^4 - T_1^4\right)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)^2}$$

$$= \frac{0.0707 \times 5.67 \times 10^{-8} \times \left(350^4 - 275^4\right)}{\frac{1}{0.75} + \frac{1 - 0.75}{0.75}\left(\frac{0.15}{0.25}\right)^2}$$

$$= \frac{37.23}{1.453}$$

$$= 25.6\,\text{W}$$

Rate of heat transfer by radiation, $\boxed{\dot{Q}_{12} = 25.6\,\text{W}}$

Question: 21.111P pg 1

Consider the two parallel coaxial disks of diameters *a* and *b*, shown in Fig. P21–111. For this geometry, the view factor from the smaller disk to the larger disk can be calculated from

$$F_{ij} = 0.5\left(\frac{B}{A}\right)^2\left\{C - \left[C^2 - 4\left(\frac{A}{B}\right)^2\right]^{0.5}\right\}$$

Where , $A = a/2L$, $B = b/2L$, and $C = 1 + [(1 + A^2)/B^2]$.. The diameter, emissivity and temperature are 20 cm , and 600°C, respectively, for disk *a*, and 40 cm, 0.80 and 200°C for disk *b*. The distance between the two disks is *L* = 10 cm.

(a) Calculate *Fab* and *Fba*.

(b) Calculate the net rate of radiation heat exchange between disks *a*and *b* in steady operation.

(c) Suppose another (infinity) large disk *c*,of negligible thickness and ε = 0.7, is inserted between disks *a* and *b* such that it is parallel and equidistant to both disks. Calculate the net rate of radiation heat exchange between disks *a*and *c* and disks *c* and *b* in steady operation.

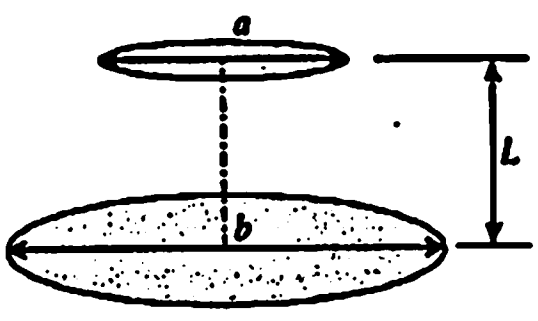


FIGURE P21–111

Step 1 of 9

Given data:

Diameter of the disk a, $a = 20\,\text{cm}$

Emissivity of the disk a, $\varepsilon_a = 0.6$

Surface temperature of the disk a, $T_a = 600\,^\circ\text{C}$

Diameter of the disk b, $b = 40\,\text{cm}$

Emissivity of the disk b , $\varepsilon_b = 0.8$

Surface temperature of the disc b, $T_b = 200\,^\circ\text{C}$

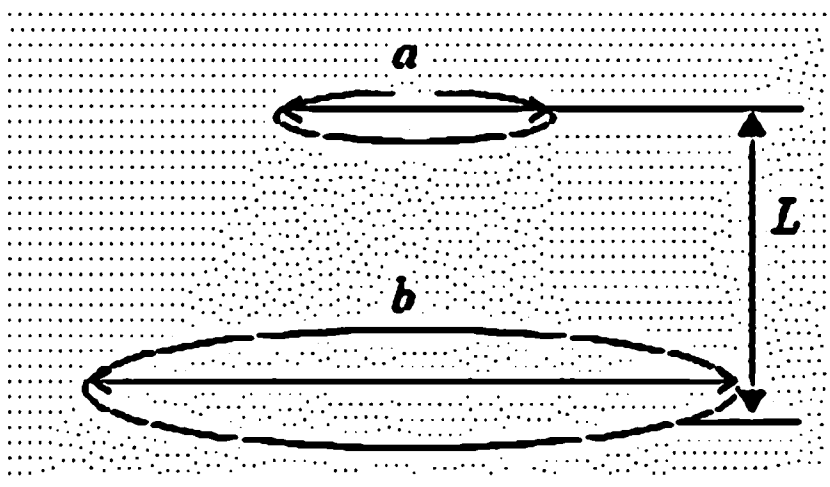
Center to center distance, $L = 10\,\text{cm}$

View factor from the smaller disk to larger disk is given by,

$$F_{ij} = 0.5\left(\frac{B}{A}\right)^2\left\{C - \left[C^2 - 4\left(\frac{A}{B}\right)^2\right]^{0.5}\right\}$$

Step 2 of 9

Given figure:



Step 3 of 9

a) Calculation of view factors from a to b and b to a:

The view factor from surface a to surface b is given as

$$F_{ab} = 0.5\left(\frac{B}{A}\right)^2\left\{C - \left[C^2 - 4\left(\frac{A}{B}\right)^2\right]^{0.5}\right\}$$

Where,

$$A = \frac{a}{2L}$$
$$= \frac{0.2}{2\times 0.1}$$
$$= 1$$
$$B = \frac{b}{2L}$$
$$= \frac{0.4}{2\times 0.1}$$
$$= 2$$
$$C = 1 + \left[\left(1 + A^2\right)/B^2\right]$$
$$= 1 + \left[\left(1 + 1^2\right)/2^2\right]$$
$$= 1.5$$

Step 4 of 9

Therefore, the view factor F_{12} can be calculated as,

$$F_{ab} = 0.5\left(\frac{B}{A}\right)^2\left\{C - \left[C^2 - 4\left(\frac{A}{B}\right)^2\right]^{0.5}\right\}$$
$$= 0.5\left(\frac{2}{1}\right)^2\left\{1.5 - \left[1.5^2 - 4\left(\frac{1}{2}\right)^2\right]^{0.5}\right\}$$
$$= 0.5\times 4\times 0.3819$$
$$= 0.7639$$

Step 5 of 9

The view factor from surface b to surface a can be calculated using the reciprocity rule.

The area of the disk 'a' and the disk 'b' can be calculated as,

$$A_a = \frac{\pi}{4}a^2$$
$$= \frac{\pi}{4}\times 0.2^2$$
$$= 0.031416\,\text{m}^2$$
$$A_b = \frac{\pi}{4}b^2$$

$$= \frac{\pi}{4} \times 0.2^2$$

$$= 0.031416 \text{m}^2$$

$$A_b = \frac{\pi}{4} b^2$$

$$= \frac{\pi}{4} \times 0.4^2$$

$$= 0.1256 \text{m}^2$$

Step 6 of 9

Therefore, the view factor F_{ba} calculated as,

$$A_b F_{ba} = A_a F_{ab}$$

$$F_{ba} = \frac{A_a}{A_b} F_{ab}$$

$$= \frac{0.031416}{0.1256} \times 0.764$$

$$= 0.191$$

Therefore, the view factor from a to b and from b to a is, $\boxed{F_{ab} = 0.764}$ and $\boxed{F_{ba} = 0.191}$

Step 7 of 9

(b)Calculation of total rate of radiation heat exchange between the disks a and b:
The net rate of radiation heat transfer between the surfaces can be determined as,

$$\dot{Q}_{ab} = \frac{\sigma(T_a^4 - T_b^4)}{\frac{1-\varepsilon_a}{A_a \varepsilon_a} + \frac{1}{A_a F_{ab}} + \frac{1-\varepsilon_b}{A_b \varepsilon_b}}$$

$$= \frac{5.67 \times 10^{-8}(873^4 - 473^4)}{\frac{1-0.6}{0.031416 \times 0.6} + \frac{1}{0.031416 \times 0.764} + \frac{1-0.8}{0.1256 \times 0.8}}$$

$$= \frac{30095.56}{64.874}$$

$$= 463.902 \text{ W}$$

$$\boxed{\dot{Q}_{ab} = 463.902 \text{W}}$$

Step 8 of 9

c) In order to calculate the net rate of radiation heat exchange, let us calculate the temperature of disk c as follows:
In this case we have an infinitely large disc; it represents that $A_c = \infty$ and the emissivity of the disk c is $\varepsilon_c = 0.7$. Since the disc is infinitely large, the view factors $F_{ac} = F_{bc} = 1$. Therefore, $\dot{Q}_{ac} = \dot{Q}_{cb} = \dot{Q}_{bc}$ by applying an Energy balance, we get,

$$\frac{\sigma(T_a^4 - T_c^4)}{\frac{1-\varepsilon_a}{A_a \varepsilon_a} + \frac{1}{A_a F_{ac}} + \frac{1-\varepsilon_c}{A_c \varepsilon_c}} = \frac{\sigma(T_b^4 - T_c^4)}{\frac{1-\varepsilon_c}{A_c \varepsilon_c} + \frac{1}{A_c F_{cb}} + \frac{1-\varepsilon_b}{A_b \varepsilon_b}}$$

By substituting the known values in the above equation (1), we get,

$$\frac{5.67 \times 10^{-8}(873^4 - T_c^4)}{\left(\frac{1-0.6}{0.031416 \times 0.6} + \frac{1}{0.031416 \times 1} + 0\right)} = \frac{5.67 \times 10^{-8}(T_c^4 - 473^4)}{\left(0 + \frac{1}{(0.1256)(1)} + \frac{1-0.8}{0.1256 \times 0.8}\right)}$$

$$\frac{(873^4 - T_c^4)}{53.0515} = \frac{(T_c^4 - 473^4)}{9.9522}$$

$$57.8 \times 10^{11} - 9.9522 T_c^4 = 53.0515 T_c^4 - 26.55 \times 10^{11}$$

$$21.303 T_c^4 = 31.25 \times 10^{11}$$

$$T_c^4 = 14.66 \times 10^{10}$$

$$T_c = 618.87 \text{K}$$

Step 9 of 9

Therefore,

$$\dot{Q}_{ac} = \dot{Q}_{cb}$$

$$= \frac{\sigma(T_a^4 - T_c^4)}{\frac{1-\varepsilon_a}{A_a \varepsilon_a} + \frac{1}{A_a F_{ac}} + \frac{1-\varepsilon_c}{A_c \varepsilon_c}}$$

$$= \frac{5.67 \times 10^{-8}(873^4 - 605^4)}{\left(\frac{1-0.6}{0.031416 \times 0.6} + \frac{1}{0.031416 \times 1} + \frac{1-0.7}{\infty \times 0.7}\right)}$$

$$= \frac{25.333 \times 10^3}{21.22 + 31.83 + 0}$$

$$= \frac{25.333 \times 10^3}{53.0509}$$

$$= 477.46 \text{ W}$$

The net rate of radiation heat exchange between disks a and c and disks c and b, $\boxed{\dot{Q}_{ac} = \dot{Q}_{cb} = 477.46 \text{ W}}$

The rate of heat transfer is higher in part (c) because the large disk c is able to collect all radiation emitted by disk a. Hence it is not acting as a radiation shield.

Question: 21.112P pg 1

A large number of long tubes, each of diameter *D*, are placed parallel to each other and at a center-to-center distance of *s*. Since all of the tubes are geometrically similar and at the same temperature, these could be treated collectively as one surface (*Aj*) for radiation heat transfer calculations. As shown in Fig. P21-112, the tube-bank (*Aj*) is placed opposite a large flat wall (*Ai*) such that the tube-bank is parallel to the wall. The radiation view factor, *Fij*, for this arrangement is given by

$$F_{ij} = 1 - \left[1 - \left(\frac{D}{s}\right)^2\right]^{0.5} + \frac{D}{s}\left\{\tan^{-1}\left[\left(\frac{s}{D}\right)^2 - 1\right]^{0.5}\right\}$$

(*a*) Calculate the view factors *Fij* and *Fji* for $s = 3.0$ cm and $D = 1.5$ cm.

(*b*) Calculate the net rate of radiation heat transfer between the wall and the tube-bank per unit area of the wall when $Ti = 900°C$, $Tj = 60°C$, $\varepsilon i = 0.8$, and $\varepsilon j = 0.9$.

(c) A fluid flows through the tubes at an average temperature of 40°C, resulting in a heat transfer coefficient of 2.0 kW/m2·K. Assuming $Ti = 900°C$, $\varepsilon i = 0.8$ and $\varepsilon j = 0.9$ (as above) and neglecting the tube wall thickness and convection from the outer surface, calculate the temperature of the tube surface in steady operation.

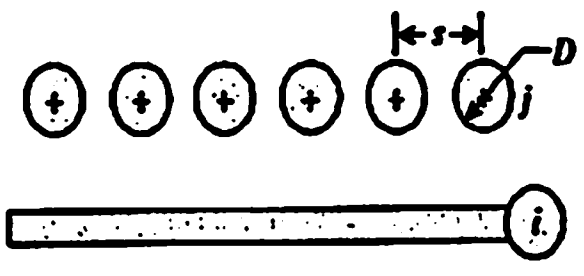


FIGURE P21-112

Step 1 of 6

Given data:

Diameter of the tubes, $D = 1.5$ cm

Center to center distance, $s = 3$ cm

Surface temperature of the tubes, $T_j = 60°C$

Emissivity of the wall, $\varepsilon_i = 0.8$

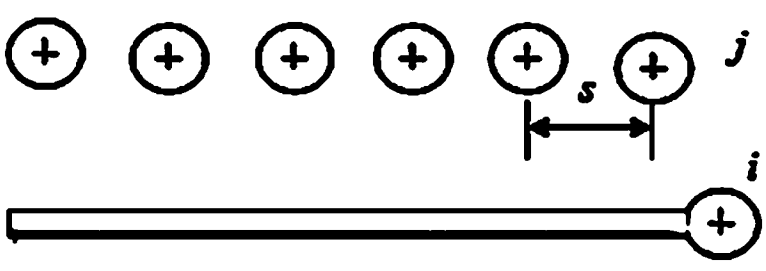
Average fluid temperature, $T_\infty = 40°C$

Heat transfer coefficient,

$$h_{conv} = 2 \text{ kW/m}^2\cdot\text{K}$$
$$= 2000 \text{ W/m}^2\cdot\text{K}$$

Step 2 of 6

Given diagram:



Step 3 of 6

(a)

Calculate the view factor from surface i to surface j is

$$F_{ij} = 1 - \left[1 - \left(\frac{D}{s}\right)^2\right]^{0.5} + \frac{D}{s}\left\{\tan^{-1}\left[\left(\frac{s}{D}\right)^2 - 1\right]^{0.5}\right\}$$

Here,

Diameter of the tubes, D

Center to center distance, s

Substitute 1.5 cm for D, 3 cm for s.

$$F_{ij} = 1 - \left[1 - \left(\frac{D}{s}\right)^2\right]^{0.5} + \frac{D}{s}\left\{\tan^{-1}\left[\left(\frac{s}{D}\right)^2 - 1\right]^{0.5}\right\}$$
$$= 1 - \left[1 - \left(\frac{1.5}{3}\right)^2\right]^{0.5} + \frac{1.5}{3}\left\{\tan^{-1}\left[\left(\frac{3}{1.5}\right)^2 - 1\right]^{0.5} \times \frac{\pi}{180}\right\}$$
$$= 0.658$$

Therefore, the view factor from surface i to j is $\boxed{0.658}$

Calculate the view factor from surface j to surface i is

By apply summation rule

$$F_{ij} + F_{ji} = 1$$

Here,

the view factor from surface j to surface i is F_{ji}

Substitute 0.658 for F_{ij}

$$F_{ij} + F_{ji} = 1$$
$$0.658 + F_{ji} = 1$$
$$F_{ji} = 0.342$$

Step 4 of 6

Therefore, the view factor from surface j to i is $\boxed{0.342}$

Step 5 of 6

(b)

Calculate the net rate of radiation heat transfer between the surfaces per unit area of the wall

$$\left[\right.$$

(b)

Calculate the net rate of radiation heat transfer between the surfaces per unit area of the wall

$$\dot{Q}_{ij} = \frac{1}{A_i}\left[\frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{A_i\varepsilon_i} + \frac{1}{A_iF_{ij}} + \frac{1-\varepsilon_j}{A_j\varepsilon_j}}\right]$$

$$= \frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \frac{1-\varepsilon_j}{\varepsilon_j}\left(\frac{A_i}{A_j}\right)}$$

Here,

Stefan Boltzmann constant is σ

Surface temperature of the tubes, T_j

Surface temperature of the wall, T_i

Emissivity of the tubes, ε_j

Emissivity of the wall, ε_i

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$1K = 1°C + 273$

Substitute $60°C$ for T_j, $900°C$ for T_i, 0.9 for ε_j, 0.8 for ε_i

$$\dot{Q}_{ij} = \frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \frac{1-\varepsilon_j}{\varepsilon_j}\left(\frac{A_i}{A_j}\right)}$$

$$= \frac{5.67\times10^{-8}\times((900+273)^4 - (60+273)^4)}{\left(\frac{1-0.8}{0.8}\right) + \left(\frac{1}{0.658}\right) + \frac{1-0.9}{0.9}\left(\frac{0.03}{\pi\times0.015}\right)}$$

$$= 57944.4\ \text{W/m}^2$$

Therefore, the net rate of radiation heat transfer between the wall and the tube-bank per unit area is

$\boxed{57944.4\ \text{W/m}^2}$

Step 6 of 6

(c)

Under steady conditions, the rate of radiation heat transfer from the wall to the tube surface is equal to the rate of convection heat transfer from the tube wall to the fluid.

Calculate the rate of heat transfer due to convention is

$$\dot{Q}_{rad} = \dot{Q}_{conv}$$

$$\frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \frac{1-\varepsilon_j}{\varepsilon_j}\left(\frac{A_i}{A_j}\right)} = h_{conv}\frac{A_j}{A_i}(T_j - T_\infty)$$

Here,

Heat transfer coefficient, h_{conv}

Average fluid temperature, T_∞

Substitute $2\ \text{kW/m}^2\cdot\text{K}$ for h_{conv}, $40°C$ for T_∞.

$$\frac{\sigma(T_i^4 - T_j^4)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{ij}} + \frac{1-\varepsilon_j}{\varepsilon_j}\left(\frac{A_i}{A_j}\right)} = h_{conv}\frac{A_j}{A_i}(T_j - T_\infty)$$

$$\frac{5.67\times10^{-8}(1173^4 - T_j^4)}{\left(\frac{1-0.8}{0.8}\right) + \left(\frac{1}{0.658}\right) + \frac{1-0.9}{0.9}\left(\frac{0.03}{\pi\times0.015}\right)} = 2000\times\frac{\pi\times0.015}{0.03}\times(T_j - 313)$$

$$T_j = 331.6\ \text{K}$$

$$= 58.6°C$$

Therefore, the temperature of the tube in steady state is $\boxed{58.6°C}$

Question: 21.113P pg 1

A simple solar collector is built by placing a 6-cm-diameter clear plastic tube around a garden hose whose outer diameter is 2 cm. The hose is painted black to maximize solar absorption, and some plastic rings are used to keep the spacing between the hose and the clear plastic cover constant. The emissivities of the hose surface and the glass cover are 0.9, and the effective sky temperature is estimated to be 15°C. The temperature of the plastic tube is measured to be 40°C, while the ambient air temperature is 25°C. Determine the rate of heat loss from the water in the hose by natural convection and radiation per meter of its length under steady conditions. *Answers:* 12.7 W, 26.1 W

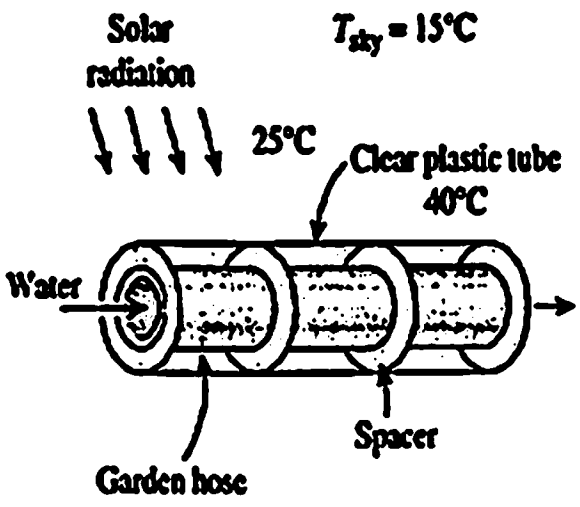


FIGURE P21–113

Step 1 of 6

Calculate the mean film temperature

$$T_f = \frac{(T_s + T_\infty)}{2}$$

Here,

Temperature of the plastic tube, T_s

Ambient air temperature, T_∞

Substitute $40°C$ for T_s, $25°C$ for T_∞

$$T_f = \frac{(T_s + T_\infty)}{2}$$
$$= \frac{(40+25)}{2}$$
$$= 32.5°C$$

Step 2 of 6

From the table the properties of air at 1 atm and the film temperature

At $T_f = 32.5°C$

$$\text{Thermal conductivity}, k = 0.02607\ \text{W/m}\cdot\text{K}$$
$$\text{kinematic viscosity}, \nu = 1.632\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Prandtle number}, \text{Pr} = 0.7275$$

Calculate the coefficient of volume expansion

$$\beta = \frac{1}{T_f + 273\text{K}}$$
$$= \frac{1}{(32.5+273)}$$
$$= 0.003273\ \text{K}^{-1}$$

The characteristic length in this case is the diameter of the plastic tube, $L_c = D_2 = 0.06\ \text{m}$

Step 3 of 6

Calculate the non-dimensional Rayleigh number

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D_2^3}{\upsilon^2}\text{Pr}$$

Here,

Acceleration due gravity is g

Substitute $9.81\ \text{m/s}^2$ for g, $1.632\times10^{-5}\ \text{m}^2/\text{s}$ for v, $0.06\ \text{m}$ for D_2, 0.7275 for Pr

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)D_2^3}{\upsilon^2}\text{Pr}$$
$$= \frac{9.81\times0.003273\times(40-25)\times0.06^3}{\left(1.632\times10^{-5}\right)^2}\times0.7275$$
$$= 284153.305$$

Step 4 of 6

Calculate the Nusselt number correlation for the Natural convention is

$$\text{Nu} = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

Substitute 284153.305 for Ra , 0.7275 for Pr

$$\text{Nu} = \left\{0.6 + \frac{0.387\text{Ra}^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$
$$= \left\{0.6 + \frac{0.387\times(284153.305)^{1/6}}{\left[1+(0.559/0.7275)^{9/16}\right]^{8/27}}\right\}^2$$
$$= 10.3035$$

Step 5 of 6

$= 10.3035$

Step 5 of 6

Calculate the heat transfer coefficient

$$h_{conv} = \mathrm{Nu}\frac{k}{L_c}$$
$$= 10.3035 \times \frac{0.02607}{0.06}$$
$$= 4.5\,\mathrm{W/m^2 \cdot K}$$

Find the Surface area of the plastic tube

$$A_{plastic} = A_2$$
$$= \pi D_2 L$$

Here Length of the tube is L

Substitute $1\,\mathrm{m}$ for L, $0.06\,\mathrm{m}$ for D_2

$$A_{plastic} = \pi D_2 L$$
$$= \pi \times 0.06 \times 1$$
$$= 0.1885\,\mathrm{m^2}$$

Calculate the rate of heat transfer from the outer surface of the plastic tube by natural convection

$$\dot{Q}_{conv} = h_{conv} A_2 (T_s - T_\infty)$$

Substitute $4.5\,\mathrm{W/m^2 \cdot K}$ for h_{conv}, $0.1885\mathrm{m^2}$ for A_2

$$\dot{Q}_{conv} = h_{conv} A_2 (T_s - T_\infty)$$
$$= 4.5(\mathrm{W/m^2 \cdot K}) \times 0.1885(\mathrm{m^2}) \times (40 - 25)$$
$$= 12.72\,\mathrm{W}$$

Therefore, the *rate* of heat transfer by natural convection is $\boxed{12.72\,\mathrm{W}}$

Step 6 of 6

Use the conversion factor to convert the temperature from $1°\mathrm{C}$ to $1\mathrm{K}$ is

$1\mathrm{K} = 1°\mathrm{C} + 273$

Calculate the rate of heat transfer by radiation from the outer surface of the plastic tube

$$\dot{Q}_{rad} = \varepsilon \sigma A_2 (T_s^4 - T_\infty^4)$$

Here,

Emissivity of the hose surface and the glass cover, ε

Substitute 0.9 for ε, $5.67 \times 10^{-8}\,\mathrm{W/m^2 \cdot K^4}$ for σ

$$\dot{Q}_{rad} = \varepsilon \sigma A_2 (T_s^4 - T_\infty^4)$$
$$= 0.9 \times 5.67 \times 10^{-8} \times 0.1885 \times \left((40 + 273\mathrm{K})^4 - (15 + 273\mathrm{K})^4\right)$$
$$= 26.15\,\mathrm{W}$$

Therefore, the rate of heat transfer by radiation is $\boxed{26.15\,\mathrm{W}}$

Question: 21.114P pg 1

Consider a 1.5-m-high and 3-m-wide solar collector that is tilted at an angle 20° from the horizontal. The distance between the glass cover and the absorber plate is 3 cm, and the back side of the absorber is heavily insulated. The absorber plate and the glass cover are maintained at temperatures of 80°C and 32°C, respectively. The emissivity of the glass surface is 0.9 and that of the absorber plate is 0.8. Determine the rate of heat loss from the absorber plate by natural convection and radiation. *Answers:* 750 W, 1289 W

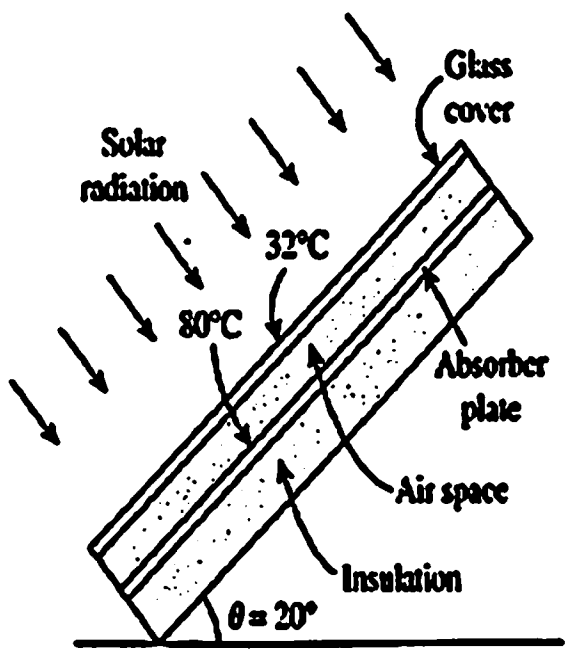


FIGURE P21–114

Step 1 of 6

Schematic diagram of solar system:

Step 2 of 6

Calculate the mean film temperature is

$$T_f = \frac{(T_1 + T_2)}{2}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Substitute $80°C$ for T_1, $32°C$ for T_2

$$T_f = \frac{(T_1 + T_2)}{2}$$
$$= \frac{(80+32)}{2}$$
$$= 56°C$$

From the table the properties of air, at 1 atm and the film temperature of are

$$\text{Thermal conductivity}, k = 0.0277\ \text{W/m}\cdot\text{K}$$
$$\text{kinematic viscosity}, \nu = 1.857\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Prandtle number}, \text{Pr} = 0.7212$$

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$$1K = 1°C + 273$$

Calculate the coefficient of volume expansion (β) is

$$\beta = \frac{1}{(56+273)}$$
$$= 0.003041\,K^{-1}$$

For $\theta = 20°$, we have horizontal rectangular enclosure. The characteristic length in this case is the distance between the two glasses $L_c = L = 0.03\,\text{m}$

Step 3 of 6

Calculate the Rayleigh number is

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2}\text{Pr}$$

Here,

Acceleration due to gravity is g

Substitute $0.003041\,K^{-1}$ for β, $80°C$ for T_1, $32°C$ for T_2 $0.03\,\text{m}$ for L, $9.81\,\text{m/s}^2$ for g, $1.857\times10^{-5}\ \text{m}^2/\text{s}$ for ν

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L^3}{\nu^2}\text{Pr}$$
$$= \frac{9.81(\text{m}^2/\text{s})\times0.003041(\text{K}^{-1})\times(80\text{-}32)(\text{K})\times0.03^3(\text{m}^3)}{\left(1.857\times10^{-5}\ \text{m}^2/\text{s}\right)^2}\times0.7212$$
$$= 80857.878$$

Step 4 of 6

Calculate the Nusselt number correlation for the Natural convention is

$$\text{Nu} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\theta}\right]\times\left[1 - \frac{1708(\sin1.8\theta)^{1.6}}{Ra\cos\theta}\right] + \left[\frac{(Ra\cos\theta)^{1/3}}{18} - 1\right]$$

Here,

Tilted angle of solar collector is θ

Substitute $20°$ for θ, 80857.878 for Ra

$$\text{Nu} = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\theta}\right]\times\left[1 - \frac{1708(\sin1.8\theta)^{1.6}}{Ra\cos\theta}\right] + \left[\frac{(Ra\cos\theta)^{1/3}}{18} - 1\right]$$
$$= 1 + 1.44\left[1 - \frac{1708}{80857.878\times\cos20}\right]\times\left[1 - \frac{1708(\sin1.8\times20)^{1.6}}{80857.878\times\cos20}\right]$$
$$\left[\frac{(80857.878\times\cos20)^{1/3}}{}\right.$$

$$\left[Ra\cos\theta\right]\left[21.114P\ \text{pg}\ 2^{18}\right]$$

$$=1+1.44\left[1-\frac{1708}{80857.878\times\cos 20}\right]\times\left[1-\frac{1708(\sin 1.8\times 20)^{1.6}}{80857.878\times\cos 20}\right]+\left[\frac{(80857.878\times\cos 20)^{1/3}}{18}-1\right]$$

$$=1+1.44\times 0.9775\times 0.9904+1.353$$

$$=3.747$$

Calculate the Surface is

$$A_s=H\times W$$

Here,

Height of the solar collector, H

Width of the solar collector, W

Substitute $1.5\,\text{m}$ for H . 3m for W

$$A_s=H\times W$$
$$=1.5(\text{m})\times 3(\text{m})$$
$$=4.5\,\text{m}^2$$

Since Nusselt number is

$$Nu=\frac{hL}{k}$$

$$h=\frac{k\,Nu}{L}$$

Here,

Heat transfer coefficient is h

Step 5 of 6

Calculate the rate of heat transfer due to natural convection

$$\dot{Q}_{conv}=hA_s(T_1-T_2)$$
$$=\frac{k\text{Nu}}{L}A_s(T_1-T_2)$$

Substitute $0.0277\ \text{W/m}\cdot\text{K}$ for k . 3.747 for Nu

$$\dot{Q}_{conv}=hA_s(T_1-T_2)$$
$$=\frac{kNu}{L}A_s(T_1-T_2)$$
$$=\frac{0.02779(\text{W/m}\cdot\text{K})\times 3.747}{0.03(\text{m})}\times 4.5(\text{m}^2)\times(80-32)$$
$$=750\,\text{W}$$

Therefore, the rate of heat transfer due to natural convection is $\boxed{750\,\text{W}}$

Step 6 of 6

Calculate the rate of heat transfer due to radiation

$$\dot{Q}_{rad}=\frac{\sigma A_2\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Emissivity of the absorber plate, ε_1

Emissivity of the glass surface, ε_2

Substitute $80°\text{C}$ for T_1 . $32°\text{C}$ for T_2 . 0.8 for ε_1 . 0.9 for ε_2

$$\dot{Q}_{rad}=\frac{\sigma A_2\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$
$$=\frac{5.67\times 10^{-8}(\text{W/m}^2\cdot\text{K}^4)\times 4.5(\text{m}^2)\times(353^4-305^4)}{\frac{1}{0.9}+\frac{1}{0.8}-1}$$
$$=1289\ \text{W}$$

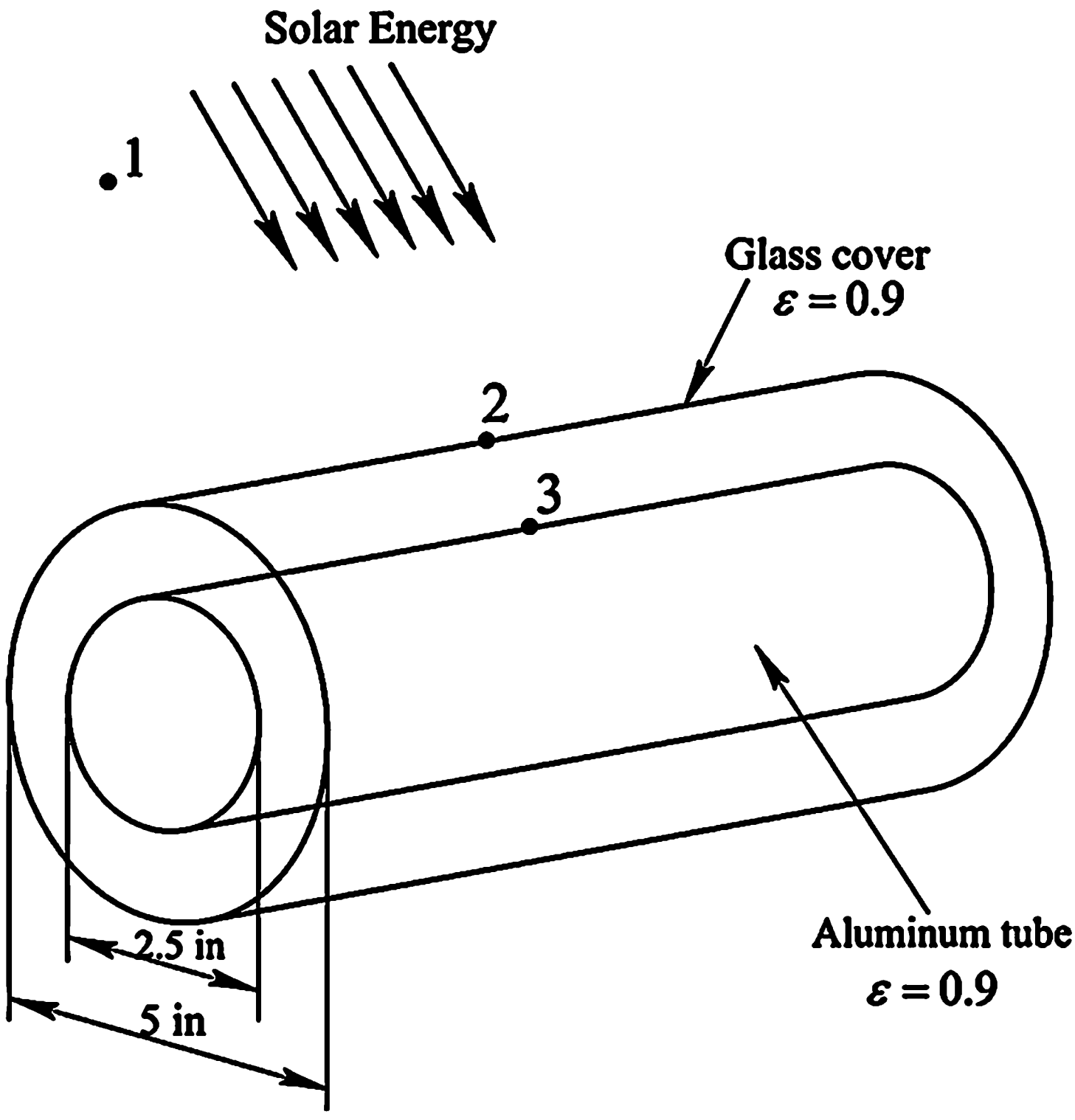
Therefore, rate of heat transfer due to radiation is $\boxed{1289\,\text{W}}$

Question: 21.115P pg 1

A solar collector consists of a horizontal aluminum tube having an outer diameter of 2.5 in enclosed in a concentric thin glass tube of diameter 5 in. Water is heated as it flows through the tube, and the annular space between the aluminum and the glass tube is filled with air at 0.5 atm pressure. The pump circulating the water fails during a clear day, and the water temperature in the tube starts rising. The aluminum tube absorbs solar radiation at a rate of 40 Btu/h per foot length, and the temperature of the ambient air outside is 75°F. The emissivities of the tube and the glass cover are 0.9. Taking the effective sky temperature to be 60°F, determine the temperature of the aluminum tube when thermal equilibrium is established (i.e., when the rate of heat loss from the tube equals the amount of solar energy gained by the tube).

Step 1 of 23

Consider the aluminum tube and the glass cover as shown in the following figure:



Step 2 of 23

Consider the heat transfer between the points 1 and 2.

Assume the exit temperature of air in the duct to be $85°F$, because the exit temperature of the air is required to evaluate the properties of air.

Calculate the average temperature.

$$T_{avg} = \frac{T_e + T_\infty}{2}$$

Here, T_{avg} is the average temperature, T_e is the exit temperature of air, and T_∞ is the temperature of the surroundings.

Substitute $85°F$ for T_e and $75°F$ for T_∞ .

$$T_{avg} = \frac{85+75}{2}$$
$$= 80°F$$

Step 3 of 23

Take the properties of air from the table A-15E 'Properties of air at 1 atm pressure'.

$$\nu = 1.697 \times 10^{-4} \ ft^2/s$$
$$Pr = 0.729$$
$$k = 0.01481 \ Btu/h \cdot ft \cdot R$$

Here, ν is the kinematic viscosity, Pr is the Prandtl number, and k is the coefficient of thermal conductivity.

Calculate the coefficient of thermal expansion.

$$\beta = \frac{1}{T_{avg}}$$

Here, β is the coefficient of thermal expansion.

Substitute $(80+460)R$ for T_{avg} .

$$\beta = \frac{1}{(80+460)}$$
$$= \frac{1}{540}$$
$$= 1.8518 \times 10^{-3}/R$$

Step 4 of 23

Calculate the surface area of the glass cover.

$$A = \pi D_o L$$

Here, A is the surface area, D_o is the outer diameter, and L is the length of the tube.

Substitute $\left(\frac{5}{12}\right)ft$ for D_o and 1 ft for L.

$$A = \pi \times \left(\frac{5}{12}\right) \times 1$$
$$= 1.30899 \ ft^2$$

$$A=\pi\times\left(\frac{5}{12}\right)\times 1$$
$$=1.30899\ \text{ft}^2$$

Step 5 of 23

Calculate the Rayleigh number as follows:

$$\text{Ra}=\frac{g\beta(T_0-T_\infty)D_0^3}{\nu^2}$$

Here, Ra is the Rayleigh number and g is the acceleration due to gravity.

Substitute $9.81\ \text{m/s}^2$ for g, $1.8518\times10^{-3}/\text{R}$ for β, $\left(5/12\right)\text{ft}$ for D_0, $1.697\times10^{-4}\ \text{ft}^2/\text{s}$ for ν, $85°\text{F}$ for T_0, and $75°\text{F}$ for T_∞.

$$\text{Ra}=\frac{9.81\times1.8518\times10^{-3}(85-75)\left(5/12\right)^3}{\left(1.697\times10^{-4}\right)^2}$$
$$=\frac{0.01314}{2.8798\times10^{-8}}$$
$$=456280.26$$

Step 6 of 23

Calculate the Nusselt number as follows:

$$\text{Nu}=\left[0.6+\frac{0.387\times(\text{Ra})^{(1/6)}}{\left(1+\left(\frac{0.559}{\text{Pr}}\right)^{(9/16)}\right)^{(8/27)}}\right]^2$$

Here, Nu is the Nusselt number.

Substitute 456280.26 for Ra and 0.7302 for Pr.

$$\text{Nu}=\left[0.6+\frac{0.387\times(456280.26)^{(1/6)}}{\left(1+\left(\frac{0.559}{0.7302}\right)^{(9/16)}\right)^{(8/27)}}\right]^2$$
$$=\left[0.6+\frac{3.3956}{1.2019}\right]^2$$
$$=11.7319$$

Step 7 of 23

Calculate the convective heat transfer coefficient as follows:

$$h=\frac{k}{D_0}\text{Nu}$$

Here, h is the convective heat transfer coefficient.

Substitute $0.01481\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$ for k, $\left(5/12\right)\text{ft}$ for D_0, and 11.7319 for Nu.

$$h=\frac{0.01481}{\left(\frac{5}{12}\right)}\times11.7319$$
$$=0.41699\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

Step 8 of 23

Calculate the heat transfer due to convection as follows:

$$Q_C=hA(T_0-T_\infty)$$

Here, Q_C is the heat transfer due to convection.

Substitute $0.41699\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$ for h, $1.30899\ \text{ft}^2$ for A, $85°\text{F}$ for T_0, and $75°\text{F}$ for T_∞.

$$Q_C=0.41699\times1.30899\times(85-75)$$
$$=5.458\ \text{Btu/h}$$

Step 9 of 23

Convert the temperatures into Rankine scale.

$$T_0(\text{R})=T_0(\text{F})+460$$
$$=85+460$$
$$=545\ \text{R}$$

Similarly, for the exit temperature,

$$T_\infty(\text{R})=T_\infty(\text{F})+460$$
$$=75+460$$
$$=535\ \text{R}$$

Similarly, for the effective temperature,

$$T_e(\text{R})=T_e(\text{F})+460$$
$$=60+460$$
$$=520\ \text{R}$$

Step 10 of 23

Calculate the heat transfer due to radiation as follows:

$$Q_R=\varepsilon\sigma A\left(T_0^4-T_e^4\right)$$

Here, Q_R is the heat transfer due to radiation, ε is the emissivity of the glass cover, and σ is the Stefan-Boltzmann constant.

$$Q_R = \varepsilon\sigma A\left(T_0^4 - T_c^4\right)$$

Here, Q_R is the heat transfer due to radiation, ε is the emissivity of the glass cover, and σ is the Stefan-Boltzmann constant.

Substitute 0.9 for ε, $0.1714\times10^{-8}\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}^4$ for σ, $1.30899\ \text{ft}^2$ for A, 545 R for T_0, and 520 R for T_c.

$$\begin{aligned} Q_R &= 0.9\times0.1714\times10^{-8}\times1.30899\times\left(545^4 - 520^4\right) \\ &= 30.506\ \text{Btu/h} \end{aligned}$$

Step 11 of 23

Calculate the total heat transferred.

$$Q_G = Q_C + Q_R$$

Here, Q_G is the total heat transferred from glass to ambient air.

Substitute $5.458\ \text{Btu/h}$ for Q_C and $30.506\ \text{Btu/h}$ for Q_R.

$$\begin{aligned} Q_G &= 5.458 + 30.506 \\ &= 35.964\ \text{Btu/h} \end{aligned}$$

The total heat transfer is equal to $40\ \text{Btu/h}$ in the equilibrium condition. So, the temperature of the exit of air is greater than the assumed temperature. Since, the value obtained is very near to the required value, consider the exit temperature of air in the duct to be approximately equal to $87°\text{F}$

$$T_0 = 87°\text{F}$$

Step 12 of 23

Consider the heat transfer between the points 2 and 3.
Assume the temperature of the aluminum tube to be $130°\text{F}$, because the exit temperature of the air is required to evaluate the properties of air.

Calculate the average temperature.

$$T_{avg} = \frac{T_i + T_0}{2}$$

Here, T_{avg} is the average temperature, T_0 is the exit temperature of air, and T_i is the temperature of the aluminum tube.

Substitute $87°\text{F}$ for T_0 and $130°\text{F}$ for T_i.

$$\begin{aligned} T_{avg} &= \frac{87 + 130}{2} \\ &= 108.5°\text{F} \end{aligned}$$

Step 13 of 23

Take the properties of air at $108.5°\text{F}$ from the table A-15E 'Properties of air at 1 atm pressure'.

$$\begin{aligned} \nu_0 &= 1.857\times10^{-4}\ \text{ft}^2/\text{s} \\ \text{Pr} &= 0.724725 \\ k &= 0.0154855\ \text{Btu/h}\cdot\text{ft}\cdot\text{R} \end{aligned}$$

Here, ν_0 is the kinematic viscosity, Pr is the Prandtl number, and k is the coefficient of thermal conductivity.

But the pressure in the enclosed region is 0.5 bar. So, the parameters k and Pr remain unchanged, but, the kinematic viscosity changes as follows:

$$\nu = \frac{\nu_0}{P}$$

Here, ν is the kinematic viscosity at 0.5 bar and P is the pressure.

Substitute $1.857\times10^{-4}\ \text{ft}^2/\text{s}$ for ν_0 and 0.5 bar for P.

$$\begin{aligned} \nu &= \frac{1.857\times10^{-4}}{0.5} \\ &= 3.714\times10^{-4}\ \text{ft}^2/\text{s} \end{aligned}$$

Step 14 of 23

Calculate the coefficient of thermal expansion.

$$\beta = \frac{1}{T_{avg}}$$

Here, β is the coefficient of thermal expansion.

Substitute $(108.5+460)\text{R}$ for T_{avg}.

$$\begin{aligned} \beta &= \frac{1}{(108.5+460)} \\ &= \frac{1}{568.5} \\ &= 1.759\times10^{-3}/\text{R} \end{aligned}$$

Step 15 of 23

Calculate the characteristic length as follows:

$$L_C = \frac{(D_0 - D_i)}{2}$$

Here, L_C is the characteristic length.

Substitute $\left(5/12\right)\text{ft}$ for D_0 and $\left(2.5/12\right)\text{ft}$ for D_i.

$$\begin{aligned} L_C &= \frac{\left(\left(5/12\right) - \left(2.5/12\right)\right)}{2} \\ &= \left(1.25/12\right)\text{ft} \end{aligned}$$

$$L_c = \frac{\left(\left(5/12\right) - \left(2.5/12\right)\right)}{2}$$
$$= \left(1.25/12\right)\text{ft}$$

Step 16 of 23

Calculate the surface area of the glass cover.

$$A = \pi D_i L$$

Here, A is the surface area, D_i is the inner diameter, and L is the length of the tube.

Substitute $\left(2.5/12\right)\text{ft}$ for D_o and 1 ft for L.

$$A = \pi \times \left(\frac{2.5}{12}\right) \times 1$$
$$= 0.65449\ \text{ft}^2$$

Step 17 of 23

Calculate the Rayleigh number as follows:

$$\text{Ra}_l = \frac{g\beta\left(T_i - T_0\right)L_C^3}{\nu^2}$$

Here, Ra_l is the Rayleigh number and g is the acceleration due to gravity.

Substitute $9.81\ \text{m/s}^2$ for g, $1.759\times10^{-3}/\text{R}$ for β, $\left(1.25/12\right)\text{ft}$ for L_c, $3.714\times10^{-4}\ \text{ft}^2/\text{s}$ for ν, $87°\text{F}$ for T_0, and $130°\text{F}$ for T_i.

$$\text{Ra}_l = \frac{9.81\times1.759\times10^{-3}\left(130-87\right)\left(1.25/12\right)^3}{\left(3.714\times10^{-4}\right)^2}$$
$$= \frac{0.00083867}{13.7938\times10^{-8}}$$
$$= 6080.05$$

Step 18 of 23

Calculate the geometric factor as follows:

$$F_{cyl} = \frac{\left(\ln\left(D_o/D_i\right)\right)^4}{L_C^3\left(D_i^{(-3/5)} + D_o^{(-3/5)}\right)^5}$$

Here, F_{cyl} is the geometric factor.

Substitute $\left(1.25/12\right)\text{ft}$ for L_c, $\left(5/12\right)\text{ft}$ for D_o, and $\left(2.5/12\right)\text{ft}$ for D_i.

$$F_{cyl} = \frac{\left(\ln\left(\left(5/12\right)/\left(2.5/12\right)\right)\right)^4}{\left(1.25/12\right)^3\left(\left(2.5/12\right)^{(-3/5)} + \left(5/12\right)^{(-3/5)}\right)^5}$$
$$= \frac{0.2308}{1.5738}$$
$$= 0.1466$$

Step 19 of 23

Calculate the effective thermal conductivity as follows:

$$k_{eff} = 0.386k\left(\frac{\text{Pr}}{0.861+\text{Pr}}\right)^{(1/4)}\left(F_{cyl}\text{Ra}_l\right)^{(1/4)}$$

Here, k_{eff} is the effective thermal conductivity.

Substitute $0.0154855\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$ for k, 6080.05 for Ra_l, 0.724725 for Pr, and 0.1466 for F_{cyl}.

$$k_{eff} = 0.386\times0.0154855\times\left(\frac{0.724725}{0.861+0.724725}\right)^{\left(\frac{1}{4}\right)}\left(0.1466\times6080.05\right)^{\left(\frac{1}{4}\right)}$$
$$= 0.005977\times0.8222\times5.46399$$
$$= 0.02685\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$$

Step 20 of 23

Calculate the heat transfer due to convection as follows:

$$Q_C = \frac{2\pi k_{eff}\left(T_i - T_0\right)}{\ln\left(D_o/D_i\right)}$$

Here, Q_C is the heat transfer due to convection.

Substitute $0.02685\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}$ for k_{eff}, $87°\text{F}$ for T_0, $130°\text{F}$ for T_i, $\left(5/12\right)\text{ft}$ for D_o, and $\left(2.5/12\right)\text{ft}$ for D_i.

$$Q_C = \frac{2\pi\times0.02685\times\left(130-87\right)}{\ln\left(\left(5/12\right)/\left(2.5/12\right)\right)}$$
$$= \frac{7.2542}{0.693}$$
$$= 10.4678\ \text{Btu/h}$$

Step 21 of 23

$$= 10.4678 \text{ Btu/h}$$

Step 21 of 23

Convert the temperatures into Rankine scale.

$$\begin{aligned} T_o(\text{R}) &= T_o(\text{F}) + 460 \\ &= 87 + 460 \\ &= 547 \text{ R} \end{aligned}$$

Similarly, for the surface temperature of the aluminum tube,

$$\begin{aligned} T_i(\text{R}) &= T_i(\text{F}) + 460 \\ &= 130 + 460 \\ &= 590 \text{ R} \end{aligned}$$

Step 22 of 23

Calculate the heat transfer due to radiation as follows:

$$Q_R = \frac{\sigma A\left(T_i^4 - T_o^4\right)}{\frac{1}{\varepsilon} + \frac{1-\varepsilon}{\varepsilon}\left(\frac{D_i}{D_o}\right)}$$

Here, Q_R is the heat transfer due to radiation, ε is the emissivity of the glass cover, and σ is the Stefan-Boltzmann constant.

Substitute 0.9 for ε, 0.1714×10^{-8} Btu/h·ft²·R⁴ for σ, 1.30899 ft² for A, 547 R for T_o, 590 R for T_i, $\left(\frac{5}{12}\right)$ft for D_o, and $\left(\frac{2.5}{12}\right)$ft for D_i.

$$Q_R = \frac{0.1714\times10^{-8}\times0.65449\times\left(590^4 - 547^4\right)}{\frac{1}{0.9} + \frac{1-0.9}{0.9}\left(\frac{\left(\frac{2.5}{12}\right)}{\left(\frac{5}{12}\right)}\right)}$$

$$\begin{aligned} &= \frac{35.5024}{1.1667} \\ &= 30.429 \text{ Btu/h} \end{aligned}$$

Step 23 of 23

Calculate the total heat transferred.

$$Q_G = Q_C + Q_R$$

Here, Q_G is the total heat transferred from glass to ambient air.

Substitute 10.4678 Btu/h for Q_C and 30.429 Btu/h for Q_R.

$$\begin{aligned} Q_G &= 10.4678 + 30.429 \\ &= 40.8968 \text{ Btu/h} \end{aligned}$$

The total heat transfer is equal to 40 Btu/h in the equilibrium condition. And the total heat transfer that is obtained is very close to the required value. So, the temperature of the aluminum tube surface is approximately equal to the assumed value.

$$T_i = 130°\text{F}$$

Hence, the temperature of the surface of the aluminum tube during the equilibrium condition is $\boxed{130°\text{F}}$.

Question: 21.116P pg 1

A vertical 2-m-high and 5-m-wide double-pane window consists of two sheets of glass separated by a 3-cm-thick air gap. In order to reduce heat transfer through the window, the air space between the two glasses is partially evacuated to 0.3 atm pressure. The emissivities of the glass surfaces are 0.9. Taking the glass surface temperatures across the air gap to be 15°C and 5°C, determine the rate of heat transfer through the window by natural convection and radiation.

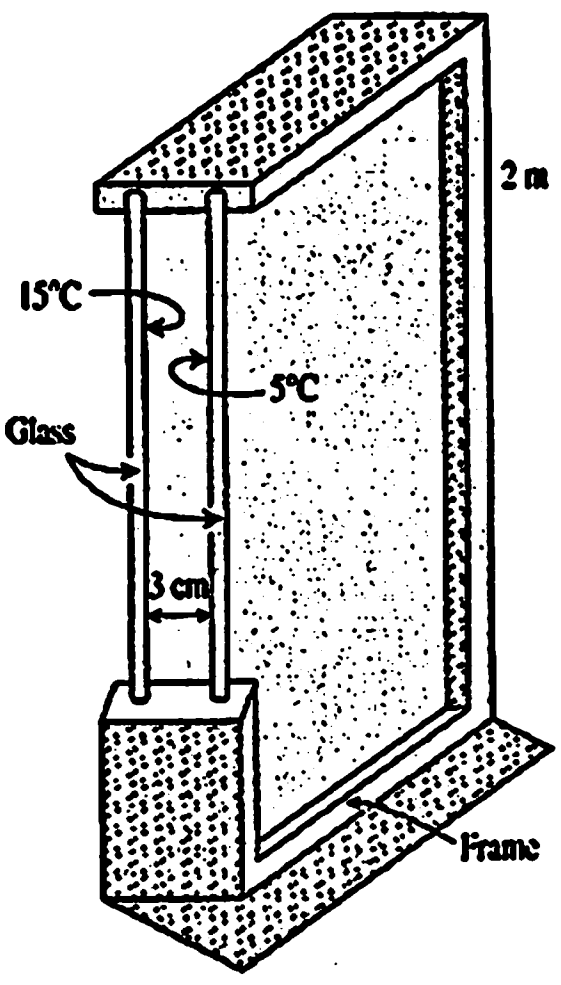
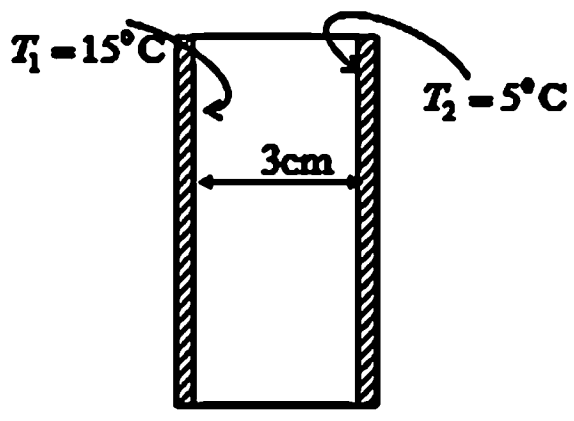


FIGURE P21–116

Step 1 of 7

The representation of vertical double pane window consists of two sheets of glass separated by an air gap is as shown below:



Step 2 of 7

Calculate the average temperature

$$T_f = \frac{(T_1 + T_2)}{2}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Substitute 15°C for T_1, 5°C for T_2

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{15+5}{2}$$
$$= 10°C$$

From the properties of air at $p = 1\text{atm}$ and average temperature $T_{avg} = 10°C$ the properties of air are

Thermal conductivity, $k = 0.02439\ \text{W/m}\cdot\text{K}$

kinematic viscosity, $\nu = 1.426\times10^{-5}\ \text{m}^2/\text{s}$

Prandtl number, $\text{Pr} = 0.7336$

Step 3 of 7

Since thermal conductivity and Prandtl number are independent of pressure these properties remain constant at $p = 1\text{atm}$ or 0.3atm

The Kinematic viscosity of air at $p = 0.3\text{atm}$ becomes

$$\nu = 1.426\times10^{-5}\ \text{m}^2/\text{s}$$
$$= \frac{1.426\times10^{-5}}{0.3}$$
$$= 4.75\times10^{-5}\ \text{m}^2/\text{s}$$

Use the conversion factor to convert the temperature from 1°C to 1K is

$$1\text{K} = 1°\text{C} + 273$$

Find the Coefficient of volume expansion

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{(10+273)}$$
$$= (1/283)\text{K}^{-1}$$

Step 4 of 7

Calculate the characteristic length in this case is the distance between the two cylinders,

$$L_c = 3\text{cm}$$
$$= \frac{3}{100}\ \text{m}$$
$$= 0.03\ \text{m}$$

Calculate the Surface area

$$A_s = H \times W$$

Here,

Height of the solar collector, H

Width of the solar collector, W

Substitute 2m for H, 5m for W

Here,

Height of the solar collector, H

Width of the solar collector, W

Substitute 2m for H, 5m for W

$$A_s = H \times W$$
$$= 2 \times 5(\text{m}^2)$$
$$= 10\,\text{m}^2$$

Step 5 of 7

Calculate the Rayleigh number,

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}\text{Pr}$$

Here,

Acceleration due to gravity is g

Substitute $1/283\,\text{K}^{-1}$ for β, 15°C for T_1, 5°C for T_2, 0.03m for L, $9.81\,\text{m/s}^2$ for g, $4.753\times10^{-5}\,\text{m}^2/\text{s}$ for ν

$$\text{Ra} = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}\text{Pr}$$
$$= \frac{9.81(\text{m/s}^2)\times(1/283)(\text{K}^{-1})\times(15-5)(\text{K})\times(0.03)^3(\text{m}^3)}{\left(4.753\times10^{-5}\,\text{m}^2/\text{s}\right)^2}\times 0.7336$$
$$= 3039.275$$

Step 6 of 7

Calculate the Nussult number,

$$Nu = 0.197 Ra^{1/4}\left(\frac{H}{L}\right)^{-1/9}$$
$$= 0.197\times(3039.275)^{1/4}\left(\frac{2}{0.03}\right)^{-1/9}$$
$$= 0.9173$$

Find the rate of heat transfer between the glass surfaces by convection

$$\dot{Q}_{conv} = kNuA_s\frac{(T_1 - T_2)}{L}$$
$$= 0.02439\times0.917\times10\frac{(15-5)}{0.03}$$
$$= 74.55\,\text{W}$$

Calculate the net rate of heat transfer due to radiation

$$\dot{Q}_{rad} = \frac{\sigma A_s\left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Emissivity of the absorber plate, ε_1

Emissivity of the glass surface, ε_2

Substitute, 288K for T_1, 278K for T_2, 0.9 for ε_1, 0.9 for ε_2

$$\dot{Q}_{rad} = \frac{\sigma A_s\left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}$$
$$= \frac{5.67\times10^{-8}(\text{W/m}^2\cdot\text{K}^4)\times10(\text{m}^2)\times(288^4 - 278^4)}{\frac{1}{0.9}+\frac{1}{0.9}-1}$$
$$= 420.71\,\text{W}$$

Step 7 of 7

Calculate the total rate of heat transfer

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad}$$
$$= 74.55(\text{W}) + 420.71(\text{W})$$
$$= 495.26\,\text{W}$$

Therefore the total rate of heat transfer through the window by natural convection and radiation is $\boxed{495.26\,\text{W}}$ and the heat transfer through the window is mostly by radiation.

Question: 21.117P

A 2-m-internal-diameter double-walled spherical tank is used to store iced water at 0°C. Each wall is 0.5 cm thick, and the 1.5-cm-thick air space between the two walls of the tank is evacuated in order to minimize heat transfer. The surfaces surrounding the evacuated space are polished so that each surface has an emissivity of 0.15. The temperature of the outer wall of the tank is measured to be 20°C. Assuming the inner wall of the steel tank to be at 0°C, determine (*a*) the rate of heat transfer to the iced water in the tank and (*b*) the amount of ice at 0°C that melts during a 24-h period.

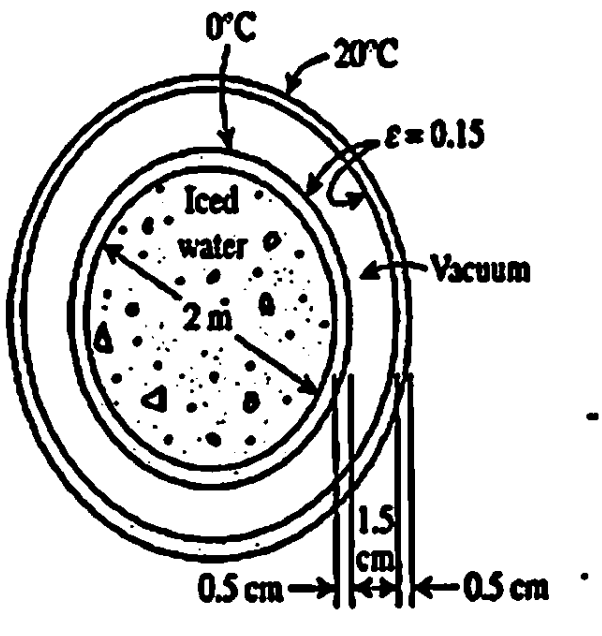


FIGURE P21–117

Step 1 of 4

Assume outer surface of the inner tank to be surface 1 and inner surface of outer tank to be surface 2.

Find the inner diameter of the spherical tank,

$$D_1 = D_i + 2t$$

Here, D_i is inner diameter of the sphere and t is thickness of the wall.

Substitute 2 m for D_i and 0.005 m for t

$$D_1 = 2 + 2\times 0.005$$
$$= 2.01\,\text{m}$$

Find the inner diameter of the outer tank,

$$D_2 = D_1 + 2l_{air}$$

Here, l_{air} is thickness of air space between two walls.

Substitute 2.01 m for D_1 and 0.015 m for l_{air}

$$D_2 = 2.01 + 2\times 0.015$$
$$= 2.04\,\text{m}$$

Find the inner tank outer surface area of the sphere.

$$A_1 = \pi D_1^2$$

Substitute 2.01 m for D_1

$$A_1 = \pi \times 2.01^2$$
$$= 12.69\,\text{m}^2$$

Step 2 of 4

(a)

Find the rate of heat transfer to the iced water in the spherical tank by using the following equation:

$$\dot{Q}_{12} = \frac{A_1\sigma\left(T_2^4 - T_1^4\right)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)^2}$$

Here, σ is Stefan Boltzmann's constant, $\varepsilon_1, \varepsilon_2$ are the emissivity's of the inner and outer surfaces, T_1 is the inner wall temperature, and T_2 is the outer wall temperature.

Substitute $12.69\,\text{m}^2$ for A_1, 0.15 for $\varepsilon_1, \varepsilon_2$, $5.67\times10^{-8}\,\text{W/m}^2.\text{K}^4$ for σ, 2.01 m for D_1, 2.04 m for D_2, $(20+273)\,\text{K}$ for T_2, and $(0+273)\,\text{K}$ for T_1.

$$\dot{Q}_{12} = \frac{12.69\times5.67\times10^{-8}\left(293^4 - 273^4\right)}{\frac{1}{0.15} + \frac{1-0.15}{0.15}\left(\frac{2.01}{2.04}\right)^2}$$
$$= \frac{1306.2788}{12.00728}$$
$$= 108.79\,\text{W}$$

Therefore, heat transfer rate to the iced water is $\boxed{108.79\,\text{W}}$.

Step 3 of 4

(b)

Find the amount of ice melts at $0°\text{C}$ during a 24-h period.

$$Q = \dot{Q}\Delta t$$

Here, t is the time period.

Substitute 108.79 W for $\dot{Q}$ and $(24\times3600)\,\text{s}$ for t.

$$Q = 108.79\times(24\times3600)$$
$$= 9399456\,\text{J}$$
$$= 9399.456\,\text{kJ}$$

Step 4 of 4

Find the amount of ice melts during the 24 hour period.

$$Q = mL_{ice}$$

Here, m is amount of ice and L_{ice} is latent heat of ice.

Substitute $9399.456\,\text{kJ}$ for Q and 334 kJ/kg for L_{ice}

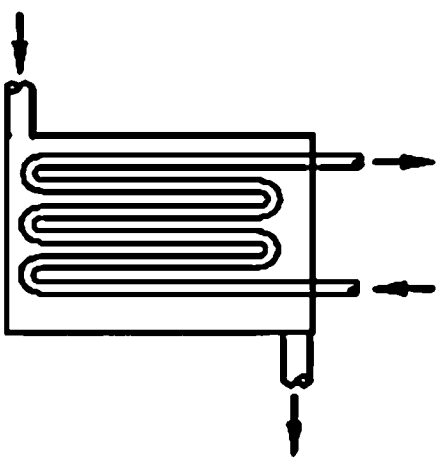
$$9399.456 = m\times334$$
$$m = 28.14\,\text{kg}$$

Therefore, the amount of ice melts in 24 h period is $\boxed{28.14\,\text{kg}}$.

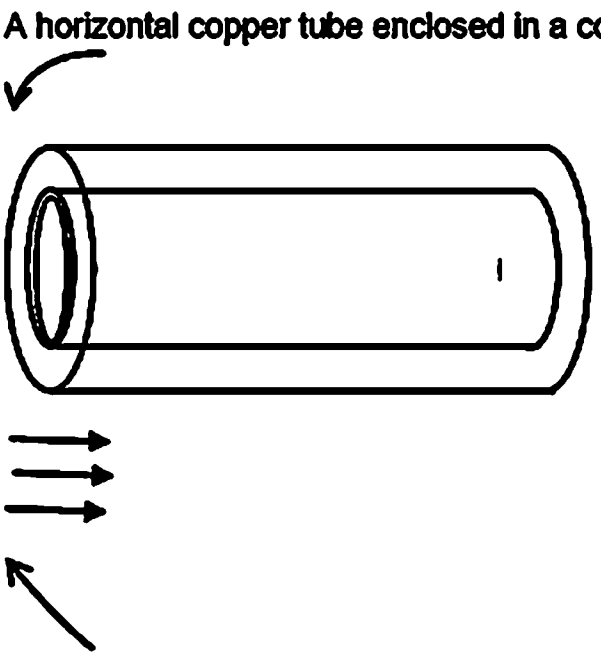
Question:

A solar collector consists of a horizontal copper tube of outer diameter 5 cm enclosed in a concentric thin glass tube of diameter 12 cm. Water is heated as it flows through the tube, and the annular space between the copper and the glass tubes is filled with air at 1 atm pressure. The emissivities of the tube surface and the glass cover are 0.85 and 0.9, respectively. During a clear day, the temperatures of the tube surface and the glass cover are measured to be 60°C and 40°C, respectively. Determine the rate of heat loss from the collector by natural convection and radiation per meter length of the tube.

Step 1 of 8



A horizontal copper tube enclosed in a concentric thin glass tube is as shown in figure,



Step 2 of 8

Calculate the average temperature

$$T_f = \frac{(T_1 + T_2)}{2}$$

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Substitute $60°C$ for T_1, $40°C$ for T_2

$$T_{avg} = \frac{T_1 + T_2}{2}$$
$$= \frac{60 + 40}{2}$$
$$= 50°C$$

From table properties of air at 1atm and the average temperature $T_{avg} = 50°C$ the properties of air are

Thermal conductivity, $k = 0.02735\ W/m \cdot K$

kinematic viscosity, $\nu = 1.798 \times 10^{-5}\ m^2/s$

Prandtle number, $Pr = 0.7228$

Step 3 of 8

Use the conversion factor to convert the temperature from $1°C$ to $1K$ is

$$1K = 1°C + 273$$

Find the Coefficient of volume expansion

$$\beta = \frac{1}{T_{avg}}$$
$$= \frac{1}{(50 + 273)}$$
$$= 3.0959 \times 10^{-3}\ K^{-1}$$

Calculate the characteristic length in this case is the distance between the two cylinders,

$$L_c = \frac{(D_2 - D_1)}{2}$$

Here,

Outer diameter of the solar collector, D_1

Diameter of the glass tube, D_2

Substitute $5\,cm$ for D_1, $12\,cm$ for D_2

$$L_c = \frac{(D_2 - D_1)}{2}$$
$$= \frac{(0.12 - 0.05)}{2}$$
$$= 0.035\,m$$

Step 4 of 8

Calculate the Rayleigh number

$$Ra = \frac{g\beta(T_1 - T_2)L_c^3}{\nu^2} Pr$$

Here,

Acceleration due to gravity is g

Substitute $3.0959\,K^{-1}$ for β, $60°C$ for T_1, $40°C$ for T_2, $0.035\,m$ for L, $9.81\,m/s^2$ for g, $1.798 \times 10^{-5}\ m^2/s$ for ν

$$\frac{g\beta(T_1 - T_2)L_c^3}{\nu^2}$$

Acceleration due to gravity is g

Substitute $3.0959\,K^{-1}$ for β, $60°C$ for T_1, $40°C$ for T_2, $0.035\,m$ for L, $9.81\,m/s^2$ for g, $1.798\times10^{-5}\ m^2/s$ for ν

$$Ra = \frac{g\beta(T_1-T_2)L_c^3}{\nu^2}Pr$$

$$= \frac{9.81(m/s^2)\times3.0959\times10^{-3}(K^{-1})\times(60-40)(K)\times(0.035)^3(m^3)}{(1.798\times10^{-5}\,m^2/s)^2}\times0.7228$$

$$= 58229$$

Calculate the effective thermal conductivity

$$F_{cyc} = \frac{\left[\ln\left(\frac{D_2}{D_1}\right)\right]^4}{L_c^3\left(D_1^{-3/5}+D_2^{-3/5}\right)^5}$$

$$= \frac{\left[\ln\left(\frac{0.12}{0.05}\right)\right]^4}{(0.035)^3\left[(0.05\,m)^{-3/5}+(0.12\,m)^{-3/5}\right]^5}$$

$$= \frac{0.5874}{3.5008}$$

$$= 0.1678$$

Step 5 of 8

Calculate the Further calculation,

$$k_{eff} = 0.386k\left(\frac{Pr}{0.861+Pr}\right)^{1/4}\left(F_{cyc}Ra_L\right)^{1/4}$$

Substitute 0.7228 for Pr, 0.1678 for F_{cyl}

$$k_{eff} = 0.386k\left(\frac{Pr}{0.861+Pr}\right)^{1/4}\left(F_{cyc}Ra_L\right)^{1/4}$$

$$= 0.386\times0.02735(W/m\cdot K)\times\left(\frac{0.7228}{0.861+0.7228}\right)^{1/4}\times(0.1678\times58229)^{1/4}$$

$$= 0.01055\times0.8219\times9.942$$

$$= 0.086207\ W/m\cdot K$$

Step 6 of 8

Calculate the rate of heat transfer between the cylinders by convection

$$\dot{Q}_{conv} = \frac{2\pi k_{eff}}{\ln\left(\frac{D_2}{D_1}\right)}(T_1-T_2)$$

$$= \frac{2\pi\times0.08620}{\ln\left(\frac{12}{5}\right)}(60-40)$$

$$= 12.374\,W$$

Step 7 of 8

Calculate the rate of heat transfer by radiation is

$$\dot{Q}_{rad} = \frac{\sigma A_1\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)}$$

Here,

Here,

Temperature of the absorber plate, T_1

Temperature of the glass cover, T_2

Emissivity of the absorber plate, ε_1

Emissivity of the glass surface, ε_2

Substitute, $333\,K$ for T_1, $313\,K$ for T_2, 0.85 for ε_1, $5.67\times10^{-8}\,W/m^2K^4$ for σ, 0.9 for ε_2

$$\dot{Q}_{rad} = \frac{\sigma A_1\left(T_1^4-T_2^4\right)}{\frac{1}{\varepsilon_1}+\frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{D_1}{D_2}\right)}$$

$$= \frac{5.67\times10^{-8}(W/m^2\text{-}K^4)\times\pi\times0.05(m)\times1(m)\times(333^4-313^4)}{\frac{1}{0.85}+\frac{1-0.9}{0.9}\left(\frac{5}{12}\right)}$$

$$= \frac{24.033}{1.2227}$$

$$= 19.655\,W$$

Step 8 of 8

Calculate the total rate of heat loss from the glass cover becomes

$$\dot{Q}_{total} = \dot{Q}_{conv}+\dot{Q}_{rad}$$

$$= 12.374(W)+19.655(W)$$

$$= 32.029\,W$$

Therefore, the rate of heat loss from the collector by natural convection and radiation per meter length of the tube is $\boxed{32.029\,W}$

Question: 21.119P pg 1

Two parallel concentric disks, 20 cm and 40 cm in diameter, are separated by a distance of 10 cm. The smaller disk ($\varepsilon = 0.80$) is at a temperature of 300°C. The larger disk ($\varepsilon = 0.60$) is at a temperature of 800°C.

(*a*) Calculate the radiation view factors.

(*b*) Determine the rate of radiation heat exchange between the two disks.

(c) Suppose that the space between the two disks is completely surrounded by a reflective surface. Estimate the rate of radiation heat exchange between the two disks.

Step 1 of 7

(a)

From table, the view factor for coaxial parallel disks is given as

$$F_{i\to j} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_j}{r_i}\right)^2\right]^{1/2}\right\}$$

$$F_{ba} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_a}{r_b}\right)^2\right]^{1/2}\right\}$$

Here,

Radius of the disk *a* is r_a

Radius of the disk *b* is r_b

Shape factor is *S*

Step 2 of 7

Find the radius of disk *a*,

$$r_a = \frac{d_a}{2}$$

Here,

Diameter of disk a, d_a

Substitute $20\,\text{cm}$ for d_a.

$$\begin{aligned} r_a &= \frac{d_a}{2} \\ &= \frac{20}{2} \\ &= 10\,\text{cm} \end{aligned}$$

Calculate the radius of disk *b*,

$$r_b = \frac{d_b}{2}$$

Here,

Diameter of the disk b, d_b

Substitute $40\,\text{cm}$ for d_b

$$\begin{aligned} r_b &= \frac{d_b}{2} \\ &= \frac{40}{2} \\ &= 20\,\text{cm} \end{aligned}$$

Step 3 of 7

Calculate the R_a by using the formula for R_a.

$$R_a = \frac{r_a}{L}$$

Here,

Center to center distance, L

Su8bstitute $10\,\text{cm}$ for L, 10 cm for r_a

$$\begin{aligned} R_a &= \frac{r_a}{L} \\ &= \frac{10}{10} \\ &= 1 \end{aligned}$$

Calculate the R_b by using the formula for R_b.

$$\begin{aligned} R_b &= \frac{r_b}{L} \\ &= \frac{20}{10} \\ &= 2 \end{aligned}$$

Step 4 of 7

Find the value of S.

$$\begin{aligned} S &= 1 + \left[\frac{1 + R_a^2}{R_b^2}\right] \\ &= 1 + \left[\frac{1 + 1^2}{2^2}\right] \\ &= 1.5 \end{aligned}$$

Calculate the view factor F_{ab}

$$F_{ba} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_a}{r_b}\right)^2\right]^{1/2}\right\}$$

Calculate the view factor F_{ab}

$$F_{ba}=\frac{1}{2}\left\{S-\left[S^2-4\left(\frac{r_a}{r_b}\right)^2\right]^{1/2}\right\}$$

Substitute 1.5 for S, $10\,\text{cm}$ for r_a, $20\,\text{cm}$ for r_b

$$\begin{aligned}F_{ba}&=\frac{1}{2}\left\{S-\left[S^2-4\left(\frac{r_a}{r_b}\right)^2\right]^{1/2}\right\}\\&=\frac{1}{2}\left\{1.5-\left[1.5^2-4\left(\frac{10}{20}\right)^2\right]^{1/2}\right\}\\&=0.191\end{aligned}$$

The view factor from surface b to surface a can be calculated using the reciprocity rule. Calculate the area of the disk 'a'

$$A_a=\frac{\pi}{4}d_a^2$$

Substitute $0.2\,\text{m}$ for d_a

$$\begin{aligned}A_a&=\frac{\pi}{4}d_a^2\\&=\frac{\pi}{4}\times 0.2^2\,(\text{m})^2\\&=0.031416\,\text{m}^2\end{aligned}$$

Calculate the area of the disk '*b*'

$$A_b=\frac{\pi}{4}d_b^2$$

Substitute $0.4\,\text{m}$ for d_b

$$\begin{aligned}A_b&=\frac{\pi}{4}d_b^2\\&=\frac{\pi}{4}\times 0.4^2\,(\text{m})^2\\&=0.1256\,\text{m}^2\end{aligned}$$

Step 5 of 7

Calculate the view factor F_{ab}

$$A_aF_{ab}=A_bF_{ba}$$

$$F_{ab}=\frac{A_b}{A_a}F_{ba}$$

Substitute 0.1256 for A_b, 0.031416 for A_a, 0.191 for F_{ba}

$$\begin{aligned}A_aF_{ab}&=A_bF_{ba}\\F_{ab}&=\frac{A_b}{A_a}F_{ba}\\&=\frac{0.1256}{0.031416}\times 0.191\\&=0.764\end{aligned}$$

Therefore, the view factor of ab is $\boxed{0.764}$.

Step 6 of 7

(b)

The net rate of radiation heat transfer between the surfaces can be determined as

$$\dot{Q}_{ab}=-\frac{\sigma\left(T_a^4-T_b^4\right)}{\frac{1-\varepsilon_a}{A_a\varepsilon_a}+\frac{1}{A_aF_{ab}}+\frac{1-\varepsilon_b}{A_b\varepsilon_b}}$$

Here,

Stefan Boltzmann constant is σ

Emissivity of the disk b, ε_b

Surface temperature of the disc b, T_b

Surface temperature of the disk a, T_a

Emissivity of the disk a, ε_a

Substitute 0.8 for ε_a, 0.6 for ε_b $800\,^\circ\text{C}$ for T_b, $300\,^\circ\text{C}$ for T_a, $5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ for σ

$$\begin{aligned}\dot{Q}_{ab}&=-\frac{\sigma\left(T_a^4-T_b^4\right)}{\frac{1-\varepsilon_a}{A_a\varepsilon_a}+\frac{1}{A_aF_{ab}}+\frac{1-\varepsilon_b}{A_b\varepsilon_b}}\\&=-\frac{5.67\times10^{-8}\times573^4-1073^4}{\frac{1-0.8}{0.031416\times0.8}+\frac{1}{0.031416\times0.764}+\frac{1-0.6}{0.1256\times0.6}}\\&=1257\,\text{W}\end{aligned}$$

Therefore, the net rate of radiation heat transfer between the surfaces is $\boxed{1257\,\text{W}}$.

Step 7 of 7

(c)

When the space between the disks is completely surrounded by a refractory surface, the net rate of radiation heat transfer can be determined as

$$\dot{Q}_{ab}=\frac{A_b\sigma\left(T_b^4-T_a^4\right)}{\left(\frac{1}{\varepsilon_b}-1\right)+\frac{A_b}{A_a}\left(\frac{1}{\varepsilon_a}-1\right)+\frac{A_a+A_b-2A_bF_{ba}}{A_a-A_bF_{ba}^2}}$$

Therefore, the net rate

radiation heat transfer can be determined as

21.119P pg 3

$$\dot{Q}_{ab} = \frac{A_b\sigma\left(T_b^4 - T_a^4\right)}{\left(\frac{1}{\varepsilon_b}-1\right)+\frac{A_b}{A_a}\left(\frac{1}{\varepsilon_a}-1\right)+\frac{A_a+A_b-2A_bF_{ba}}{A_a-A_bF_{ba}^2}}$$

$$= \frac{0.1256\times 5.67\times 10^{-8}\times 1073^4 - 573^4}{\left(\frac{1}{0.6}-1\right)+\frac{0.1256}{0.031416}\left(\frac{1}{0.8}-1\right)+\frac{0.031416+0.1256-2\times 0.1256\times 0.191}{0.031416-0.1256\times 0.191^2}}$$

$$= 1506\,\text{W}$$

Therefore, the net rate of radiation heat transfer is $\boxed{1506\text{W}}$

Question: 21.120P

Two parallel back disks are positioned coaxially with a distance of 0.25 m apart. The lower disk is 0.2 m in diameter and the upper disk is 0.4 m in diameter. If the lower disk is heated electrically at 20 W to maintain a uniform temperature of 500 K, determine the temperature of the upper disk. Answer. 423 K

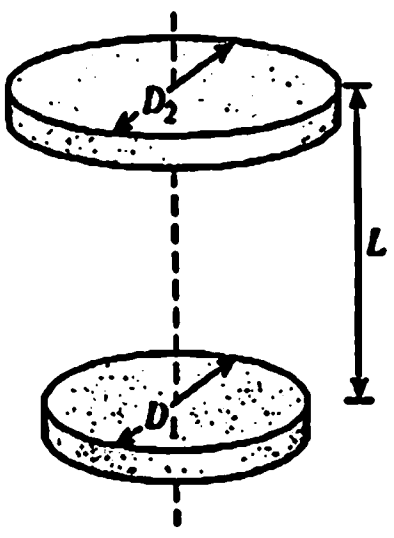


FIGURE P21–120

Step 1 of 6

Given data:

Distance between two plates, $L = 0.25\,\text{m}$

Diameter of the lower disc, $D_1 = 0.2\,\text{m}$

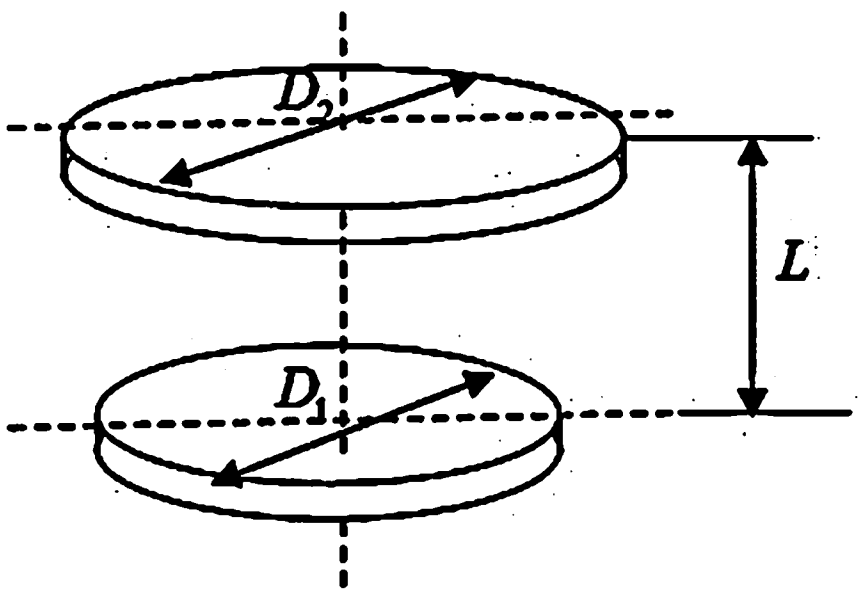
Diameter of the upper disc, $D_2 = 0.4\,\text{m}$

Temperature of the lower disc, $T_1 = 500\,\text{K}$

Heat supplied to the lower disc, $\dot{Q} = 20\,\text{W}$

Step 2 of 6

Schematic diagram of disc system:



Step 3 of 6

From the view factor expression for common geometrics of finite size

For Coaxial parallel disks

Assume, the lower plate is denoted as1 and upper plate is denoted as 2, the view factor from plate 1 to plate 2 can be expressed as

$$F_{12} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_2}{r_1}\right)^2\right]^{1/2}\right\}$$

where

$$S = 1 + \frac{1 + R_2^2}{R_1^2}$$

Step 4 of 6

The values of R_1 and R_2 can be calculated as

$$R_1 = \frac{r_1}{L}$$
$$= \frac{0.1}{0.25}$$
$$= 0.4$$
$$R_2 = \frac{r_2}{L}$$
$$= \frac{0.2}{0.25}$$
$$= 0.8$$

Step 5 of 6

Therefore,

$$S = 1 + \frac{1 + R_2^2}{R_1^2}$$
$$= 1 + \frac{1 + 0.8^2}{0.4^2}$$
$$= 11.25$$

The view factor can be calculated as

$$F_{12} = \frac{1}{2}\left\{S - \left[S^2 - 4\left(\frac{r_2}{r_1}\right)^2\right]^{1/2}\right\}$$
$$= \frac{1}{2}\left\{11.25 - \left[11.25^2 - 4\left(\frac{0.2}{0.1}\right)^2\right]^{1/2}\right\}$$
$$= 0.3675$$

Step 6 of 6

The net rate of radiation heat transfer from the lower plate to the upper plate can be expressed as

$$\dot{Q}_{12} = A_1 F_{12} \sigma (T_1^4 - T_2^4)$$

Therefore, the temperature of the upper plate can be calculated as

$$20\,(\text{W}) = \frac{\pi}{4} \times 0.2^2\,(\text{m}^2) \times 0.3675 \times 5.67 \times 10^{-8}\,(\text{W/m}^2\text{-K}^4) \times (500^4 - T_2^4)\,(\text{K}^4)$$

$$T_2 = 423\,\text{K}$$

Therefore, the temperature of the upper plate is $\boxed{T_2 = 423\,\text{K}}$.

CHAPTER 23
SOLUTIONS

Question: 22.001P

Classify heat exchangers according to flow type and explain the characteristics of each type.

Step 1 of 8

Classification of heat exchangers on the basis of flow type is as follows.

a) Parallel flow heat exchangers

b) Counter flow heat exchangers

c) Cross flow heat exchangers

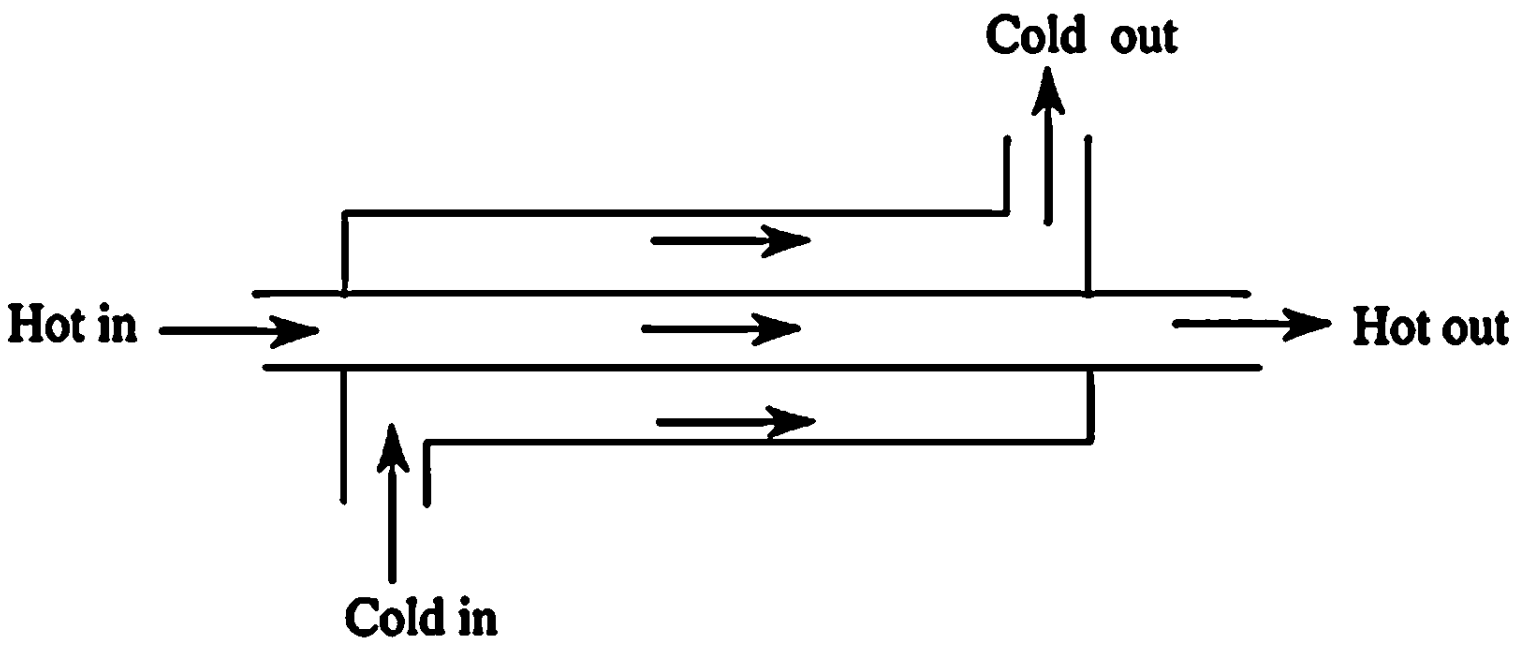
Step 2 of 8

Parallel flow:

In parallel flow arrangement, the hot and cold fluids enters at the same end, flow in the same direction, and leave at the same end.

Step 3 of 8

Draw the diagram of parallel flow heat exchanger.



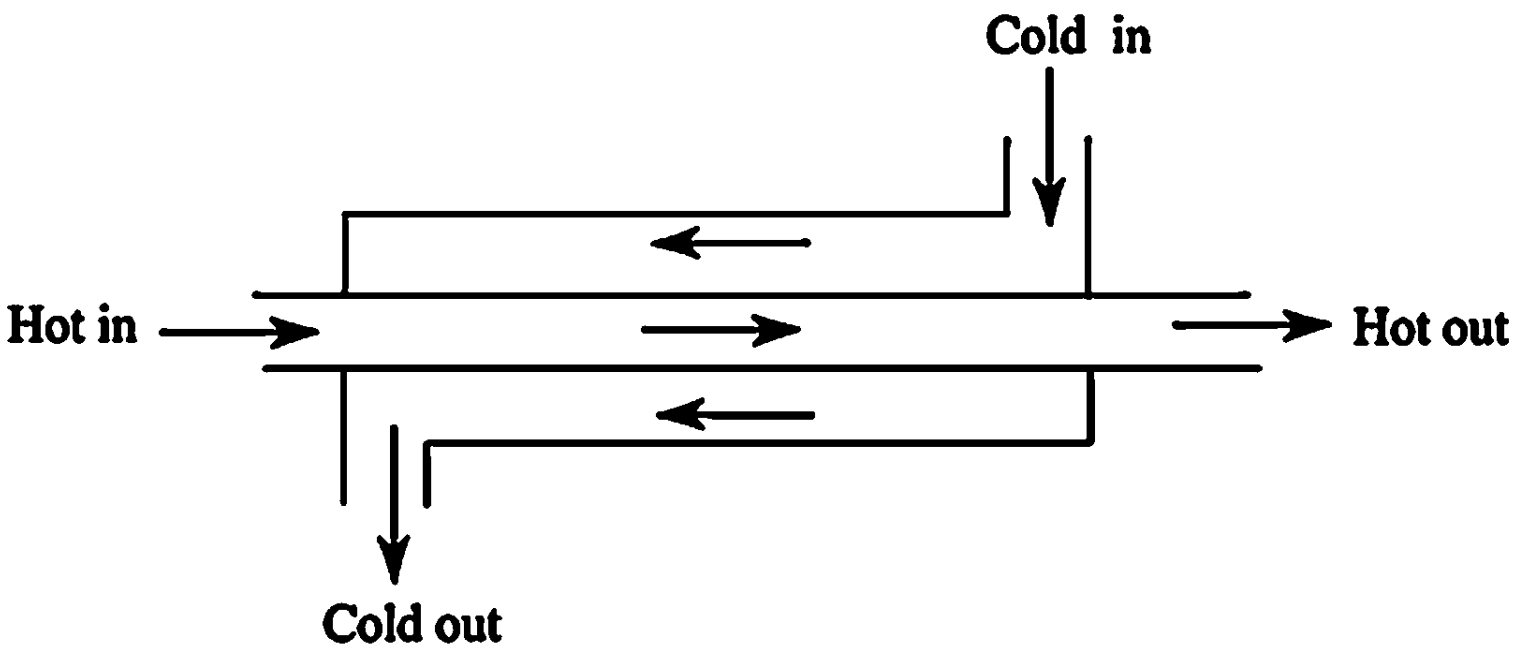
Step 4 of 8

Counter flow:

In counter flow arrangement, the hot and cold fluids enter the heat exchanger at opposite ends, flow in opposite directions.

Step 5 of 8

Draw the diagram of counter flow heat exchanger.



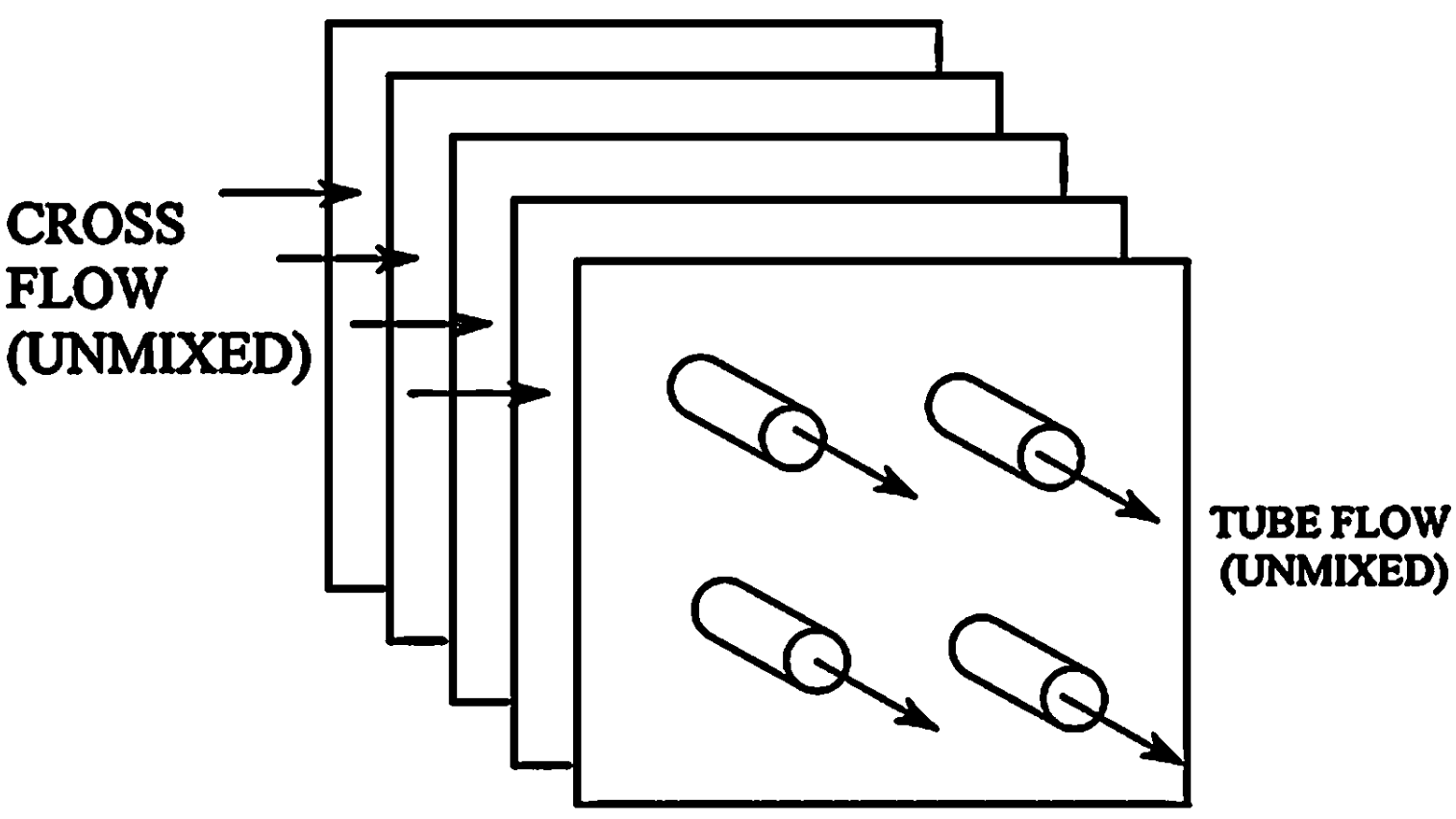
Step 6 of 8

Cross flow:

In cross flow arrangement, hot and cold fluids move in the perpendicular direction. Cross flow is further classified as unmixed and mixed flow depending on flow configuration.

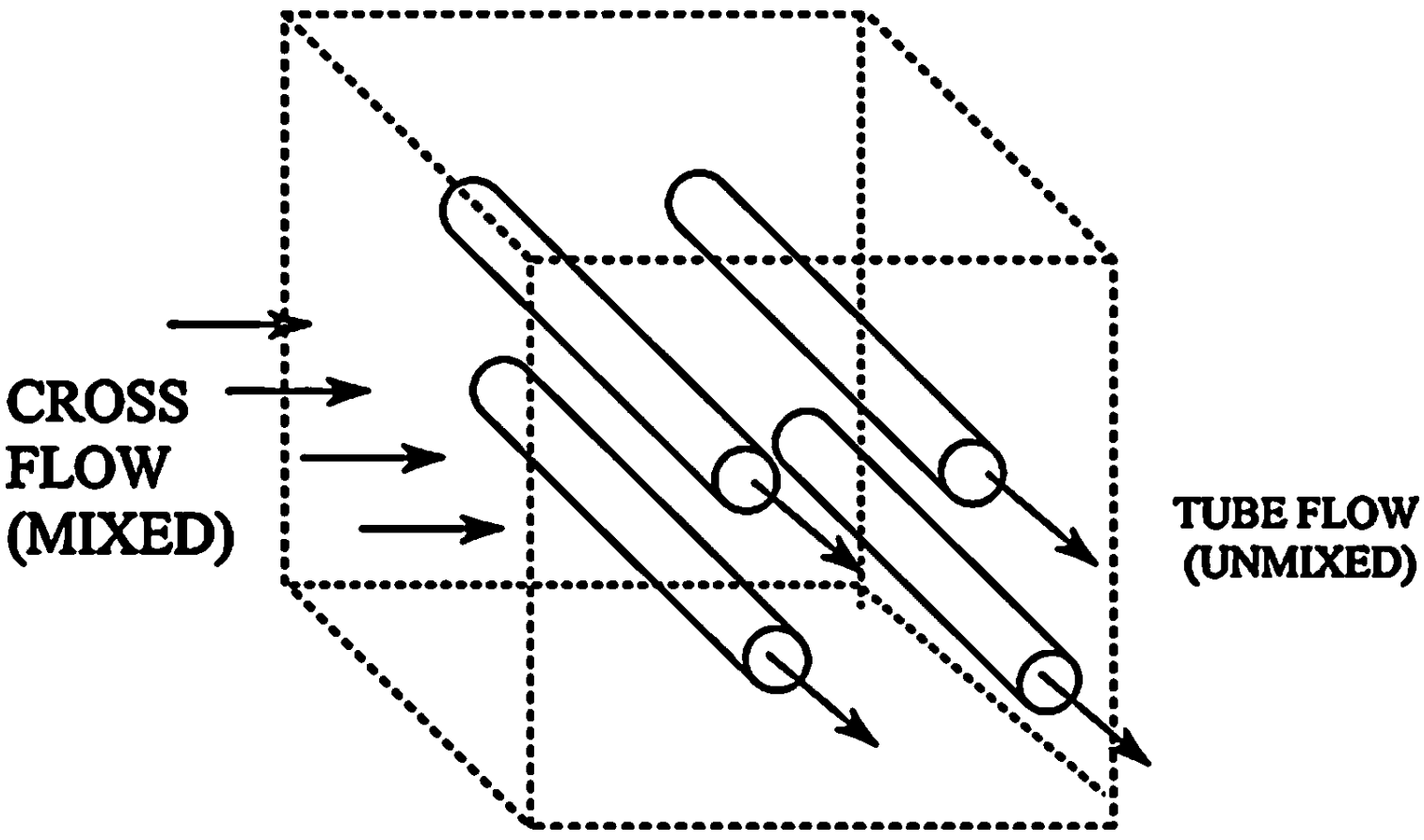
Step 7 of 8

Draw the diagram showing "both fluids unmixed".



Step 8 of 8

Draw the diagram showing "one fluid mixed and other fluid unmixed".



Question: 22.002P

When is a heat exchanger classified as being compact? Do you think a double-pipe heat exchanger can be classified as a compact heat exchanger?

Step 1 of 2

The ratio of surface area through which heat transfer takes place in a heat exchanger to its volume is called the area density (β) .Being compact of a heat exchanger depends on the area density of the heat exchanger.

Step 2 of 2

If the area density (β) of a heat exchanger is greater than $(700\ \mathrm{m^2/m^3})$ or $(200\ \mathrm{ft^2/ft^3})$ then the heat exchanger is called as compact heat exchanger.

Car radiator $(\beta \simeq 1000\mathrm{m^2/m^3})$ and glass ceramic gas turbine $(\beta \simeq 6000\mathrm{m^2/m^3})$ are best examples of compact heat exchangers.

The area density for a double pipe heat exchanger cannot be more than $700\ \mathrm{m^2/m^3}$. Therefore double pipe heat exchanger is not considered as compact heat exchanger.

Question: 22.003P

What is a regenerative heat exchanger? How does a static type of regenerative heat exchanger differ from a dynamic type?

Step 1 of 2

The ratio of surface area through which heat transfer takes place in a heat exchanger to its volume is called the area density (β) .Being compact of a heat exchanger depends on the area density of the heat exchanger.

Step 2 of 2

If the area density (β) of a heat exchanger is greater than $(700\ \mathrm{m^2/m^3})$ or $(200\ \mathrm{ft^2/ft^3})$ then the heat exchanger is called as compact heat exchanger.

Car radiator $(\beta \simeq 1000\mathrm{m^2/m^3})$ and glass ceramic gas turbine $(\beta \simeq 6000\mathrm{m^2/m^3})$ are best examples of compact heat exchangers.

The area density for a double pipe heat exchanger cannot be more than $700\ \mathrm{m^2/m^3}$. Therefore double pipe heat exchanger is not considered as compact heat exchanger.

Question: 22.004P

What is the role of the baffles in a shell-and-tube heat exchanger? How does the presence of baffles affect the heat transfer and the pumping power requirements? Explain.

Step 1 of 1

Baffles are commonly placed in the shell to force the shell-side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes.

Baffles are usually installed to increase the convection coefficient of the shell-side fluid by inducing turbulence and a cross flow velocity component.

In addition, the baffles physically support the tubes, reducing flow induced tube vibration.

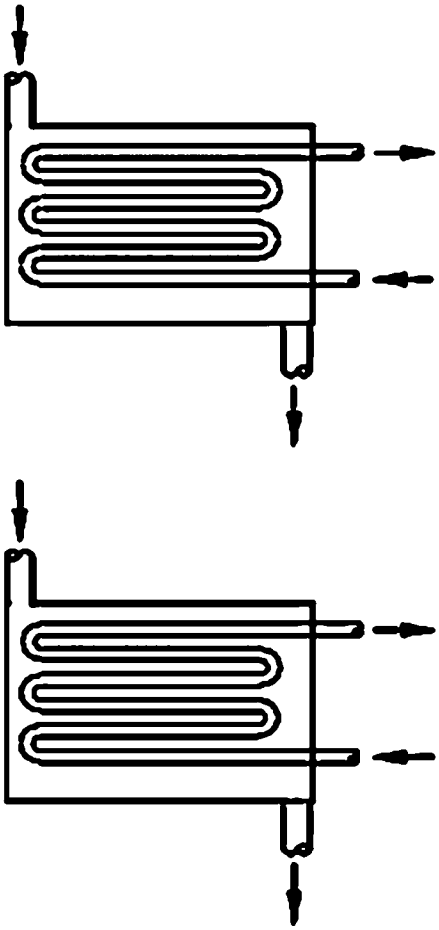
And an increased pumping power will be needed to maintain.

Question: 22.005P

How does a cross-flow heat exchanger differ from a counter-flow one? What is the difference between mixed and unmixed fluids in cross-flow?

Step 1 of 1

Draw the 1 shell and 6 tube passes arrangement as follows:



Advantages:

The heat transfer surface area is more by using 6 tube passes. If surface area increases, the heat transfer rate increases. Hence, for the same pipe diameter, the heat transfer rate is more by using six tube passes instead of two tube passes.

Disadvantages:

Despite their wide spread use, shell-and-tube heat exchangers are not suitable for use in automotive and aircraft applications because of their relatively large size and weight. The manufacturing cost increases due to increased area in case of six tube arrangement.

Question: 22.006P

What are the heat transfer mechanisms involved during heat transfer in a liquid-to-liquid heat exchanger from the hot to the cold fluid?

Step 1 of 1

The mechanism involved during heat transfer in a liquid to liquid heat exchanger from the hot fluid to the cold fluid is due to convection and conduction.
Heat is transferred from hot fluid to wall is due to convection, heat transfer through the wall is by conduction and finally from the wall to the cold fluid is due to convection

If the difference in temperature between the hot and cold fluid are too large, radiation heat transfer will come into picture.
The total thermal resistance during the heat transfer becomes

$R = R_i + R_{wall} + R_o$

Question: 22.007P

Under what conditions is the thermal resistance of the tube in a heat exchanger negligible?

Step 1 of 1

When the wall thickness of the tube is small and the thermal conductivity of the tube material high, as is usually the case, the thermal resistance of the tube is negligible.

Question: 22.008P

Consider a double-pipe parallel-flow heat exchanger of length *L*. The inner and outer diameters of the inner tube are *D*1 and *D*2, respectively, and the inner diameter of the outer tube is *D*3. Explain how you would determine the two heat transfer surface areas *Ai* and *AO*. When is it reasonable to assume *Ai* ≈ *Ao* ≈ *As*?

Step 1 of 1

A_i and A_o are surface areas of the separating wall wetted by the inner and outer fluids respectively. The surface areas can be calculated by

$$A_i = \pi D_1 L$$

$$A_o = \pi D_2 L$$

When the wall thickness of the tube is small then we can consider $A_i \cong A_o \cong A_s$

Question: 22.009P

How is the thermal resistance due to fouling in a heat exchanger accounted for? How do the fluid velocity and temperature affect fouling?

Step 1 of 1

The layer of deposits represents additional resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease. The net effect of these accumulations on heat transfer is represented by a fouling factor R_f, which is a measure of the thermal resistance introduced by fouling.

Fouling increases with increasing temperature and by decreasing fluid velocity.

Question: 22.010P

In a thin-walled double-pipe heat exchanger, when is the approximation $U = h_i$,- a reasonable one? Here U is the overall heat transfer coefficient and h_i is the convection heat transfer coefficient inside the tube.

Step 1 of 1

In a thin-walled double-pipe heat exchanger, the relation $U \cong h_i$ is valid only when the overall heat transfer coefficient U is dominated by the smaller convective coefficient, since the inverse of a large number is small. When one of the convection coefficients is smaller than the other (say $h_i << h_0$), we have $\frac{1}{h_i} >> \frac{1}{h_0}$ and thus $U \approx h_i$

Question: 22.011P

What are the common causes of fouling in a heat exchanger? How does fouling affect heat transfer and pressure drop?

Step 1 of 1

The layer of deposits represents additional resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease. The net effect of these accumulations on heat transfer is represented by a fouling factor R_f, which is a measure of the thermal resistance introduced by fouling.

Fouling increases with increasing temperature and by decreasing fluid velocity.

Question: 22.012P

Under what conditions can the overall heat transfer coefficient of a heat exchanger be determined from $U = (1/hi + VhO)-1$?

Step 1 of 1

When the wall thickness of the tube is small and the thermal conductivity of the tube material high, as is usually the case, the thermal resistance of the tube is negligible and the inner and outer surfaces of the tube are almost identical then the overall heat transfer coefficient can be calculated by

$$\frac{1}{U}=\frac{1}{h_i}+\frac{1}{h_o}$$

Question: 22.013P

Water at an average temperature of 180°F and an average velocity of 4 ft/s flows through a thin-walled $\frac{3}{4}$-in- diameter tube. The water is cooled by air that flows across the tube with a velocity of 12 ft/s at an average temperature of 80°F. Determine the overall heat transfer coefficient.

Step 1 of 6

Calculate the overall heat transfer coefficient.

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} \quad \text{..... (1)}$$

Here, h_i is the inside heat transfer coefficient and h_o is the outside heat transfer coefficient.

Obtain the properties of water from properties of saturated water table in the text book, at $180°\text{F}$.

Dynamic viscosity is $\mu = 2.317 \times 10^{-4}\ \text{lbm/ft-s}$.

Density is $\rho = 60.57\ \text{lbm/ft}^3$.

Thermal conductivity is $k = 0.388\ \text{Btu/h.ft.°F}$.

Prandtl number is $\text{Pr} = 2.15$.

Step 2 of 6

Calculate the Reynolds number of water.

$$\text{Re} = \frac{\rho V_m D_h}{\mu}$$

Here, V_m is the average velocity of water and D_h is the diameter of thin walled tube.

Substitute $60.57\ \text{lbm/ft}^3$ for ρ, $8\ \text{ft/s}$ for V_m, $0.75\ \text{in.}$ for D_h, and $2.317 \times 10^{-4}\ \text{lbm/ft-s}$ for μ.

$$\text{Re} = \frac{(60.57\ \text{lbm/ft}^3)(8\ \text{ft/s})\left(\frac{0.75}{12}\ \text{ft}\right)}{2.317 \times 10^{-4}\ \text{lbm/ft-s}}$$
$$= 130707.81$$

Since, $\text{Re} > 10000$, the flow of water is turbulent.

Step 3 of 6

Calculate the Nusselt number for the fully developed flow in tube.

$$Nu = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4}$$

Substitute 130707.81 for Re and 2.15 for Pr.

$$Nu = 0.023(130707.81)^{0.8}(2.15)^{0.4}$$
$$= 387.03$$

Calculate the inside heat transfer coefficient using the relation.

$$h_i = \frac{k}{D_h} Nu$$

Substitute $0.388\ \text{Btu/h-ft-°F}$ for k, $0.75\ \text{in.}$ for D_h, and 387.03 for Nu.

$$h_i = \frac{0.388\ \text{Btu/h-ft-°F}}{\frac{0.75}{12}\ \text{ft}}(387.03)$$
$$= 2402.7\ \text{Btu/h-ft}^2\text{-°F}$$

Step 4 of 6

Obtain the properties of air from properties of air table in the text book, at $80°\text{F}$.

Kinematic viscosity is $\nu = 1.697 \times 10^{-4}\ \text{ft}^2/\text{s}$.

Thermal conductivity of air is $k = 0.01481\ \text{Btu/h.ft.°F}$.

Prandtl number of air is $\text{Pr} = 0.729$.

Calculate the Reynolds number of air.

$$\text{Re} = \frac{V_{air} D}{\nu}$$

Substitute $12\ \text{ft/s}$ for V_{air}, $0.75\ \text{in.}$ for D_h, and $1.697 \times 10^{-4}\ \text{ft}^2/\text{s}$ for ν.

$$\text{Re} = \frac{(12\ \text{ft/s})\left(\frac{0.75}{12}\ \text{ft}\right)}{0.1697 \times 10^{-3}\ \text{ft}^2/\text{s}}$$
$$= 4419.56$$

Step 5 of 6

Calculate the Nusselt number.

$$Nu = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{[1 + (0.4)\text{Pr}^{2/3}]^{1/4}} + \left[1 + \left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

Substitute 4419.56 for Re and 0.729 for Pr.

$$Nu = 0.3 + \frac{0.62(4419.56)^{0.5}(0.729)^{1/3}}{\left[1 + (0.4)(0.729)^{2/3}\right]^{1/4}} + \left[1 + \left(\frac{4419.56}{282000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + 34.582 + 1.059$$
$$= 35.94$$

Step 6 of 6

Calculate the outside heat transfer coefficient using the relation.

$$h_o = \frac{k}{D_h} Nu$$

Substitute $0.01481\ \text{Btu/h-ft-°F}$ for k, $0.75\ \text{in.}$ for D_h, and 35.94 for Nu.

$$h_o = \frac{0.01481\ \text{Btu/h-ft-°F}}{\frac{0.75}{12}\ \text{ft}}(35.94)$$
$$= 8.516\ \text{Btu/h-ft}^2\text{-°F}$$

Substitute $8.516\ \text{Btu/h-ft}^2\text{-°F}$ for h_o and $2402.7\ \text{Btu/h-ft}^2\text{-°F}$ for h_i in equation (1).

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$
$$= \frac{1}{\frac{1}{2402.7} + \frac{1}{8.516}}$$
$$= 8.486\ \text{Btu/h-ft}^2\text{-°F}$$

Therefore, the overall heat transfer coefficient is $\boxed{8.486\ \text{Btu/h-ft}^2\text{-°F}}$.

Question: 22.014P

Water at an average temperature of 110°C and an average velocity of 3.5 m/s flows through a 7-m-long stainless steel tube (k = 14.2 W/m·K) in a boiler. The inner and outer diameters of the tube are Di = 1.0 cm and DO = 1.4 cm, respectively. If the convection heat transfer coefficient at the outer surface of the tube where boiling is taking place is $h0$ = 7200 W/m2·K, determine the overall heat transfer coefficient Ui of this boiler based on the inner surface area of the tube.

Step 1 of 4

Given

Average velocity of water is $V_m = 3.5$ m/s

Average temperature of water $T = 110°C$

Length of the stainless steel tube $L = 7$ m

Thermal conductivity of the stainless steel tube $k_{ss} = 14.2$ W/m · °C

Inner and outer diameter of the tube are $D_i = 0.01$ m and $D_o = 0.014$ m

The convection heat transfer coefficient at the outer surface of the tube is $h_o = 7200$ W/m^2 · K

Step 2 of 4

From properties of saturated water tables the properties of water at $T = 110°C$ are

$\nu = 0.268\times10^{-6}$ m^2/s

k=0.682 W/m · K

Pr=1.58

The overall heat transfer coefficient U_i of this boiler based on the inner surface area of the tube is calculated by

$$R = \frac{1}{U_i A_i}$$

$$U_i = \frac{1}{R A_i}$$

Where

$$R = R_i + R_{wall} + R_o$$

$$R = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o}$$

Step 3 of 4

The Reynolds number is

$$\mathrm{Re} = \frac{V_m D_i}{\nu}$$

$$\mathrm{Re} = \frac{(3.5\ \text{m/s})(0.01\ \text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}}$$

Re=130597.01

Re >10,000 the flow is turbulent. Assume the flow is fully developed flow.

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$$

$$= 0.023(130597.01)^{0.8}(1.58)^{0.4}$$

$$Nu = 341.93$$

And we know

$$Nu = \frac{hD_i}{k}$$

$$h_i = \frac{k}{D_i} Nu$$

$$= \frac{0.682\ \text{W/m}\cdot\text{K}}{0.01\ \text{m}}(341.93)$$

$$h_i = 23319.626\ \text{W/m}^2\cdot°\text{C}$$

Step 4 of 4

Therefore total resistance of this heat exchanger is then determined substituting all the known values

$$R = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o}$$

$$R = \frac{1}{(23319.626)[\pi(0.01)(7)]} + \frac{\ln\left(0.014/0.01\right)}{[2\pi(14.2)(7)]} + \frac{1}{(7200)[\pi(0.014)(7)]}$$

$$= 0.000194 + 0.0005387 + 0.000451$$

$$R = 0.0011837°\text{C/W}$$

Then

$$U_i = \frac{1}{R A_i}$$

$$U_i = \frac{1}{(0.0011837°\text{C/W})[\pi(0.01\ \text{m})(7\ \text{m})]}$$

$$\boxed{U_i = 3841.584\ \text{W/m}^2\cdot°\text{C}}$$

Question: 22.015P

Repeat Prob. 22-14, assuming a fouling factor Rf,i =0.0005 m2·K/W on the inner surface of the tube.

Step 1 of 4

Given

Average velocity of water is $V_m = 3.5$ m/s

Average temperature of water $T = 110°C$

Length of the stainless steel tube $L = 7$ m

Thermal conductivity of the stainless steel tube $k_{ss} = 14.2$ W/m · °C

Inner and outer diameter of the tube are $D_i = 0.01$ m and $D_o = 0.014$ m

The convection heat transfer coefficient at the outer surface of the tube is $h_o = 7200$ W/m^2 · K

Fouling factor on the inner surface of the tube is $R_{f,i} = 0.0005$ m^2.K/W

Step 2 of 4

From properties of saturated water tables the properties of water at $T = 110°C$ are

$\nu = 0.268\times10^{-6}$ m^2/s

k=0.682 W/m · K

Pr=1.58

The overall heat transfer coefficient U_i of this boiler based on the inner surface area of the tube is calculated by

$$R = \frac{1}{U_i A_i}$$

$$U_i = \frac{1}{RA_i}$$

Where

$$R = R_i + R_{wall} + R_o$$

$$R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi KL} + \frac{1}{h_o A_o}$$

Step 3 of 4

The Reynolds number is

$$\mathrm{Re} = \frac{V_m D_i}{\nu}$$

$$\mathrm{Re} = \frac{(3.5\ \text{m/s})(0.01\ \text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}}$$

Re=130597.01

Re >10,000 the flow is turbulent. Assume the flow is fully developed flow.

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$$

$$= 0.023(130597.01)^{0.8}(1.58)^{0.4}$$

$$Nu = 341.93$$

And we know

$$Nu = \frac{hD_i}{k}$$

$$h_i = \frac{k}{D_i}Nu$$

$$= \frac{0.682\ \text{W/m}\cdot\text{K}}{0.01\ \text{m}}(341.93)$$

$$h_i = 23319.626\ \text{W/m}^2\cdot°\text{C}$$

Step 4 of 4

Therefore total resistance of this heat exchanger is then determined substituting all the known values

$$R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi KL} + \frac{1}{h_o A_o}$$

$$R = \frac{1}{(23319.626)[\pi(0.01)(7)]} + \frac{(0.0005)}{\pi(0.01)(7)} + \frac{\ln\left(0.014/0.01\right)}{[2\pi(14.2)(7)]} + \frac{1}{(7200)[\pi(0.014)(7)]}$$

$$= 0.000194 + 0.00227 + 0.0005387 + 0.000451$$

$$R = 0.003453°\text{C/W}$$

Then

$$U_i = \frac{1}{RA_i}$$

$$U_i = \frac{1}{(0.003453°\text{C/W})[\pi(0.01\ \text{m})(7\ \text{m})]}$$

$$\boxed{U_i = 1316.908\ \text{W/m}^2\cdot°\text{C}}$$

Therefore the overall heat transfer coefficient of this boiler based on the inner surface area of the tube is $U_i = 1316.908$ W/m^2 · °C

Question: 22.016P

Reconsider Prob. 22-15. Using EES (or other) software, plot the overall heat transfer coefficient based on the inner surface as a function of fouling factor as it varies from 0.0001 m2·K/W to 0.0008 m2·K/W, and discuss the results.

Step 1 of 4

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"GIVEN"
T_w=110 [C]
Vel=3.5 [m/s]
L=7 [m]
k_pipe=14.2 [W/m-C]
D_i=0.010 [m]
D_o=0.014 [m]
h_o=7200 [W/m^2-C]
"R_f_i=0.0005 [m^2-C/W] "
"PROPERTIES"
k=conductivity (Water, T=T_w, P=300)
Pr=Prandtl (Water, T=T_w, P=300)
Rho=density (Water, T=T_w, P=300)
Mu=viscosity (Water, T=T_w, P=300)
Nu=mu/rho
"ANALYSIS"
Re=(Vel*D_i)/nu "Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_i=k/D_i*Nusselt
A_i=pi*D_i*L
A_o=pi*D_o*L
R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k_pipe*L)+1/(h_o*A_o)
U_i=1/(R*A_i)
```

Step 2 of 4

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as R_{fi} and U_i press 'Add' and then press 'ok'
A table will appear and enter the values of length under ' R_{fi} ' as
0.0001,0.0001778.....0 .0008in runs 1 to 10
Then from click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

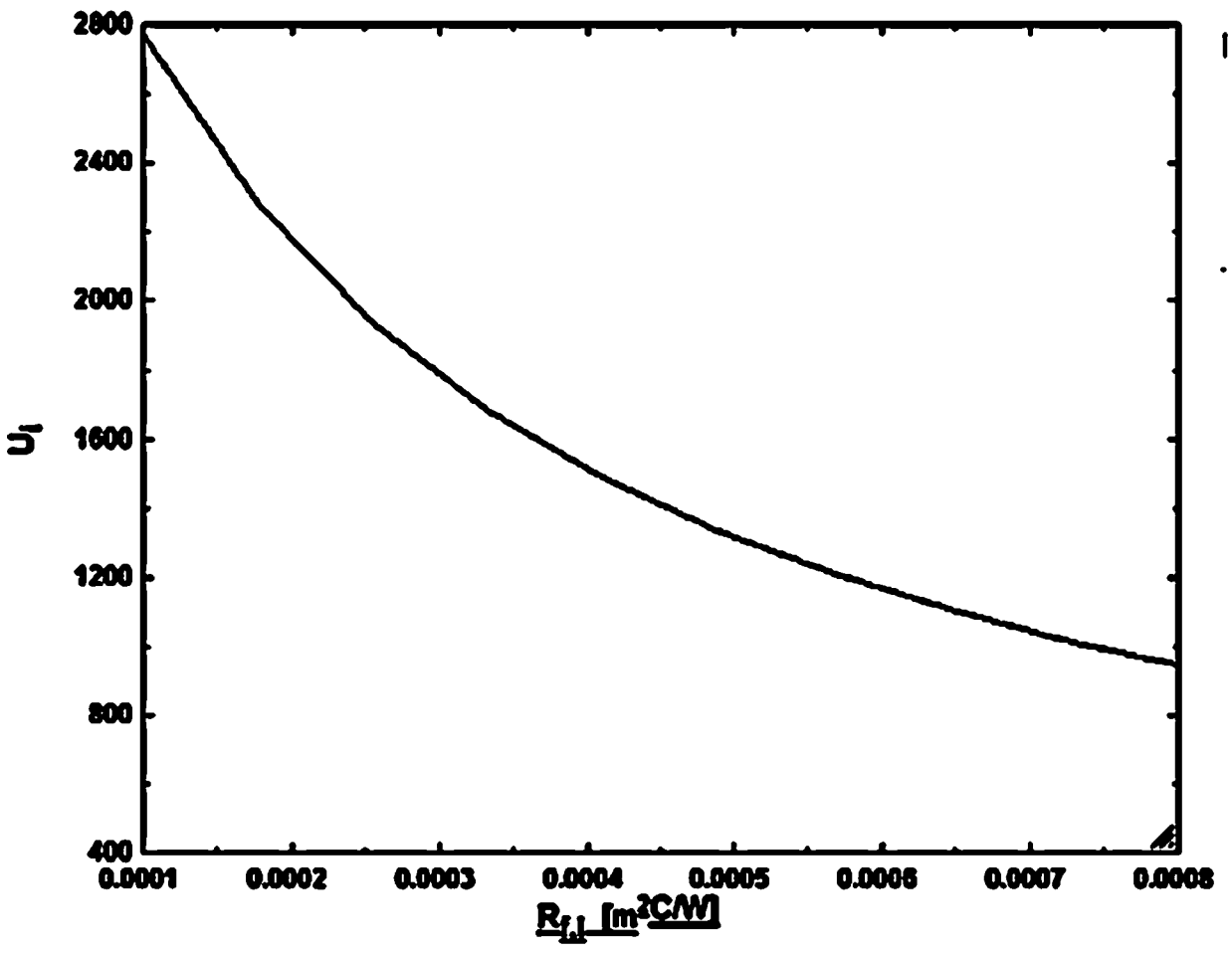
Step 3 of 4

The following results are displayed:

1..10	$R_{f,i}$ [m2-C/W]	U_i
Run 1	0.0001	2769
Run 2	0.0001778	2279
Run 3	0.0002556	1936
Run 4	0.0003333	1682
Run 5	0.0004111	1488
Run 6	0.0004889	1333
Run 7	0.0005667	1208
Run 8	0.0006444	1104
Run 9	0.0007222	1017
Run 10	0.0008	942.4

Step 4 of 4

Select the plot on menu bar, and then click x-y plot .Dialog box will be appeared.
Select X-axis as R_{fi} and Y-axis as U_i .Press OK
The Graph will be plotted as shown below.
By plotting the graph between fouling factor and heat transfer coefficient:



This graph represents that by increasing the fouling factor, the overall heat transfer coefficient decreases.

Question: 22.017P

A counter-flow heat exchanger is stated to have an overall heat transfer coefficient, based on outside tube area of 50 Btu/h·ft2·°F when operating at design and clean conditions. After a period of use, scale built-up in the heat exchanger gives a fouling factor of 0.002 h·ft2·°F/Btu. Determine (*a*) the overall heat transfer coefficient of the heat exchanger and (*b*) the percentage change in the overall heat transfer coefficient due to the scale built-up.

Step 1 of 2

a)

From the equation of overall heat transfer coefficient with fouling factor

$$\frac{1}{UA_0}=\frac{1}{U_0A_0}+\frac{R_{f,0}}{A_0} \qquad \text{..... (1)}$$

Here

Overall heat transfer coefficient based on outside tube area, U_0

Fouling factor for outside tube, $R_{f,0}$

Outer surface area, A_0

Simplify the equation (1) and substitute 50 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for U_0, 0.002 $\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}$ for R_0 in the equation (1) to determine overall heat transfer coefficient

$$\frac{1}{UA_0}=\frac{1}{U_0A_0}+\frac{R_{f,0}}{A_0}$$

$$\frac{1}{U}=\frac{1}{50}+0.002$$

$$=0.022$$

$$U=45.45\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Therefore, overall heat transfer coefficient of the heat exchanger is $\boxed{45.45\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$

Step 2 of 2

b)

Percentage change of overall heat transfer coefficient due to scale built up is

$$\%U_c=\frac{U_0-U}{U_0}$$

Substitute 45.45 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for U, 50 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for U_0 in the above equation

$$\%U_c=\frac{50-45.45}{50}$$

$$=0.091$$

$$=9.1\%$$

Therefore, percentage change of overall heat transfer coefficient U_c is $\boxed{9.1\%}$.

Question: 22.018P

The tube in a heat exchanger has a 2-in. inner diameter and a 3-in. outer diameter. The thermal conductivity of the tube material is 0.5 Btu/h·ft·°F, while the inner surface heat transfer coefficient is 50 Btu/h·ft2·°F and the outer surface heat transfer coefficient is 10 Btu/h·ft2·°F. Determine the overall heat transfer coefficients based on the outer and inner surfaces.

Step 1 of 3

Total thermal resistance of the heat exchanger is

$$R=\frac{1}{h_i A_i}+\frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL}+\frac{1}{h_0 A_o}$$

Here

Inner diameter of tube in heat exchanger, D_i

Outer diameter of tube I heat exchanger, D_o

Thermal conductivity tube material, k

Heat transfer coefficient at inner side of heat exchanger, h_i

Heat transfer coefficient at outer side of heat exchanger, h_0

Inner surface area, A_i and Outer surface area A_o

Inner surface area, $A_i=\pi D_i L$

Outer surface area, $A_o=\pi D_o L$

Substitute 2in. for D_i, 3in. for D_o, $0.5\text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$ for k, $50\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for h_i, $10\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ for h_0, 1 ft. for L in the below equation

$$R=\frac{1}{h_i A_i}+\frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL}+\frac{1}{h_0 A_o}$$

$$=\frac{1}{h_i(\pi D_i L)}+\frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL}+\frac{1}{h_0(\pi D_o L)}$$

$$=\frac{1}{50\left(\pi\left(2\text{in.}\times\frac{1\text{ft}}{12\text{in.}}\right)1\right)}+\frac{\ln\left(\frac{3}{2}\right)}{2\pi\times0.5\times1}+\frac{1}{10\left(\pi\times3\text{in.}\times\frac{1\text{ft}}{12\text{in.}}1\right)}$$

$$=0.03819+0.1290+0.127$$

$$=0.2945\ {}^\circ\text{F/Btu}$$

Step 2 of 3

Overall heat transfer coefficient based on outer surface is

$$U_o=\frac{1}{RA_0}$$

Substitute $0.2945\ {}^\circ\text{F/Btu}$ for R, 1 ft. for L in the below equation

$$U_o=\frac{1}{RA_0}$$

$$=\frac{1}{0.2945\times(\pi D_o L)}$$

$$=\frac{1}{0.2945\times\pi\times3\text{in.}\times\frac{1\text{ft}}{12\text{in.}}\times1}$$

$$=4.323\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Therefore, Overall heat transfer coefficient based on outer surface is $\boxed{4.323\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$

Step 3 of 3

Overall heat transfer coefficient based on inner surface is

$$U_i=\frac{1}{RA_i}$$

Substitute $0.2945\ {}^\circ\text{F/Btu}$ for R, 1 ft. for L in the below equation

$$U_i=\frac{1}{RA_i}$$

$$=\frac{1}{0.2945\times(\pi D_i L)}$$

$$=\frac{1}{0.2945\times\pi\times2\text{in.}\times\frac{1\text{ft}}{12\text{in.}}\times1}$$

$$=6.485\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Therefore, Overall heat transfer coefficient based on inner surface is $\boxed{6.485\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$

Question: 22.019P

A long thin-walled double-pipe heat exchanger with tube and shell diameters of 1.0 cm and 2.5 cm, respectively, is used to condense refrigerant-134a by water at 20°C. The refrigerant flows through the tube, with a convection heat transfer coefficient of *hi* = 4100 W/m2·K. Water flows through the shell at a rate of 0.3 kg/s. Determine the overall heat transfer coefficient of this heat exchanger.

Step 1 of 5

Given

Heat exchanger with tube and shell diameters are $D_o = 0.01\ \text{m}$ and $D_s = 0.025\ \text{m}$

Temperature of water $T = 20°\text{C}$

Mass flow rates of water is $\dot{m} = 0.3\ \text{kg/s}$

Convection heat transfer coefficient $h_i = 4100\ \text{W/m}^2 \cdot \text{K}$

Step 2 of 5

From saturated properties of water the properties of water at $T = 20°\text{C}$ are

$\nu = 1.004 \times 10^{-6}\ \text{m}^2/\text{s}$

$k = 0.598\ \text{W/m.°C}$

$\rho = 998\ \text{kg/m}^3$

$\text{Pr} = 7.01$

The overall heat transfer coefficient is determined by using the expression

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$

The average velocity of water in the tube

$$V_m = \frac{\dot{m}}{\rho A_c}$$

$$V_m = \frac{\dot{m}}{\rho\left(\pi\left(\frac{D_o^2 - D_i^2}{4}\right)\right)}$$

$$V_m = \frac{0.3}{998\left(\pi\left(\frac{0.025^2 - 0.01^2}{4}\right)\right)}$$

$$V_m = 0.729\ \text{m/s}$$

Step 3 of 5

Reynolds number for the flow is expressed as

$$\text{Re} = \frac{V_m D_h}{\nu}$$

Where D_h = hydraulic diameter

$$D_h = D_o - D_i$$

$$D_h = 0.025\ \text{m} - 0.01\ \text{m}$$

$$D_h = 0.015\ \text{m}$$

Then Reynolds number is

$$\text{Re} = \frac{(0.729\ \text{m/s})(0.015\ \text{m})}{1.004 \times 10^{-6}\ \text{m}^2/\text{s}}$$

$$\text{Re} = 10891.434$$

Re >10,000 the flow is turbulent. Assume the flow is fully developed flow.

Step 4 of 5

Nusselt number is

$$Nu = \frac{hD_h}{k}$$

$$Nu = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4}$$

$$= 0.023(10891.434)^{0.8}(7.01)^{0.4}$$

$$Nu = 85.0517$$

And

$$h_o = \frac{k}{D_h} Nu$$

$$= \frac{0.598}{0.015} \times 85.0517$$

$$h_o = 3390.727\ \text{W/m}^2 \cdot °\text{C}$$

Step 5 of 5

By substituting h_o and h_i overall heat transfer coefficient becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$

$$U = \frac{1}{\frac{1}{4100} + \frac{1}{3390.727}}$$

$$\boxed{U = 1855.89\ \text{W/m}^2 \cdot °\text{C}}$$

Therefore the overall heat transfer coefficient of this heat exchanger is

$$U = 1855.89\ \text{W/m}^2 \cdot °\text{C}$$

Question: 22.020P

Repeat Prob. 22–19 by assuming a 2-mm-thick layer of limestone (k = 1.3 W/m·K) forms on the outer surface of the inner tube.

Step 1 of 5

Given

Heat exchanger with tube and shell diameters are $D_o = 0.01\ \text{m}$ and $D_s = 0.025\ \text{m}$

Temperature of water $T = 20°\text{C}$

Mass flow rates of water is $\dot{m} = 0.3\ \text{kg/s}$

Convection heat transfer coefficient $h_i = 4100\ \text{W/m}^2 \cdot \text{K}$

Thickness of the layer $L = 2\,\text{mm}$

Thermal conductivity of the limestone $k = 1.3\ \text{W/m.°C}$

Step 2 of 5

From saturated properties of water the properties of water at $T = 20°\text{C}$ are

$\nu = 1.004 \times 10^{-6}\ \text{m}^2/\text{s}$

$k = 0.598\ \text{W/m.°C}$

$\rho = 998\ \text{kg/m}^3$

$\text{Pr} = 7.01$

The overall heat transfer coefficient is determined by using the expression

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$

The average velocity of water in the tube

$$V_m = \frac{\dot{m}}{\rho A_c}$$

$$V_m = \frac{\dot{m}}{\rho\left(\pi\left(\frac{D_o^2 - D_i^2}{4}\right)\right)}$$

$$V_m = \frac{0.3}{998\left(\pi\left(\frac{0.025^2 - 0.01^2}{4}\right)\right)}$$

$$V_m = 0.729\ \text{m/s}$$

Step 3 of 5

Reynolds number for the flow is expressed as

$$\text{Re} = \frac{V_m D_h}{\nu}$$

Where D_h = hydraulic diameter

$$D_h = D_o - D_i$$

$$D_h = 0.025\ \text{m} - 0.01\ \text{m}$$

$$D_h = 0.015\ \text{m}$$

Then Reynolds number is

$$\text{Re} = \frac{(0.729\ \text{m/s})(0.015\ \text{m})}{1.004 \times 10^{-6}\ \text{m}^2/\text{s}}$$

$$\text{Re} = 10891.434$$

Re >10,000 the flow is turbulent. Assume the flow is fully developed flow.

Step 4 of 5

Nusselt number is

$$Nu = \frac{hD_h}{k}$$

$$Nu = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4}$$

$$= 0.023(10891.434)^{0.8}(7.01)^{0.4}$$

$$Nu = 85.0517$$

And

$$h_o = \frac{k}{D_h} Nu$$

$$= \frac{0.598}{0.015} \times 85.0517$$

$$h_o = 3390.727\ \text{W/m}^2 \cdot °\text{C}$$

Step 5 of 5

By substituting h_o, h_i and lime stone k overall heat transfer coefficient becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o} + \left(\frac{L}{k}\right)}$$

$$U = \frac{1}{\frac{1}{4100} + \frac{1}{3390.727} + \frac{2 \times 10^{-3}}{1.3}}$$

$$\boxed{U = 481.397\ \text{W/m}^2 \cdot °\text{C}}$$

Therefore the overall heat transfer coefficient of this heat exchanger is

$U = 481.397\ \text{W/m}^2 \cdot °\text{C}$

Question: 22.021P

Reconsider Prob. 22–20. Using EES (or other) software, plot the overall heat transfer coefficient as a function of the limestone thickness as it varies from 1 mm to 3 mm, and discuss the results.

Step 1 of 3

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box.

```
"GIVEN"
D_i=0.010[m]
D_o=0.025[m]
T_w=20[C]
h_i=4100[W/m^2-C]
m_dot=0.3[kg/s]
""L_limestone=2[mm]""
k_limestone=1.3[W/m-C]
"PROPERTIES"
k=conductivity (Water, T=T_w, P=100)
Pr=Prandtl (Water, T=T_w, P=100)
Rho=density (Water, T=T_w, P=100)
Mu=viscosity (Water, T=T_w, P=100)
Nu=mu/rho
"ANALYSIS"
D_h=D_o-D_i
Vel=m_dot/(rho*A_c)
A_c=pi*(D_o^2-D_i^2)/4
Re=(Vel*D_i)/nu
"Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_o=k/D_i*Nusselt
U=1/(1/h_i+(L_limestone*Convert(mm,m))/k_limestone+1/h_o)
```

Step 2 of 3

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select number of runs = 10
Select variables in equations as R_g and U_i press 'Add' and then press 'ok'
A table will appear and enter the values of length under ' R_g ' as 1,1.222,....2.778,3in runs 1 to 10
Then from click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

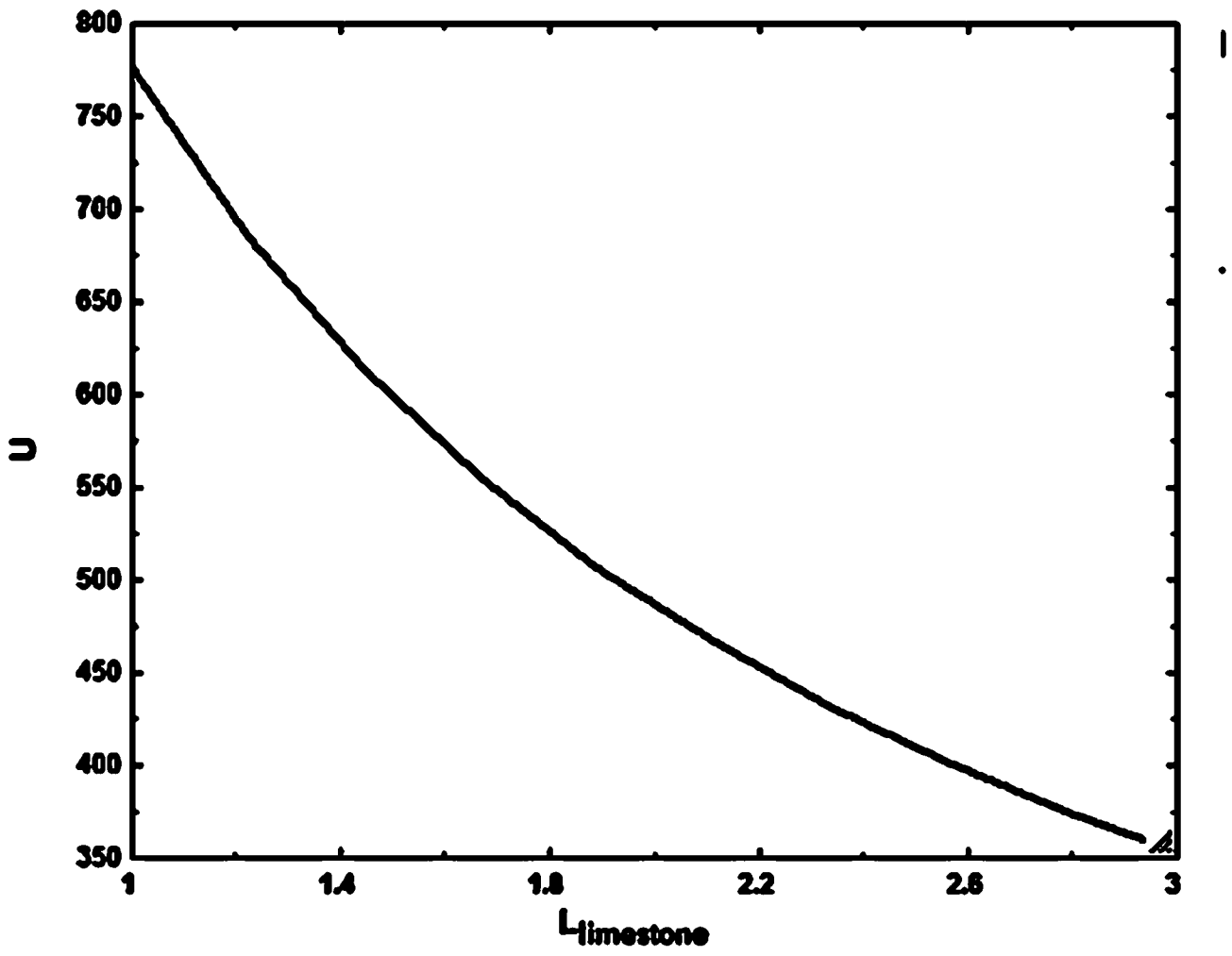
The following results are displayed:

1..10	1 $L_{limestone}$	2 U
Run 1	1	776.2
Run 2	1.222	685.2
Run 3	1.444	613.4
Run 4	1.667	555.2
Run 5	1.889	507.1
Run 6	2.111	466.6
Run 7	2.333	432.1
Run 8	2.556	402.4
Run 9	2.778	376.5
Run 10	3	353.7

Step 3 of 3

Select the plot on menu bar, and then click x-y plot .Dialog box will be appeared.
Select X-axis as $L_{limestone}$ and Y-axis as U .Press OK
The Graph will be plotted as shown below.

By plotting the graph between overall heat transfer and Length:

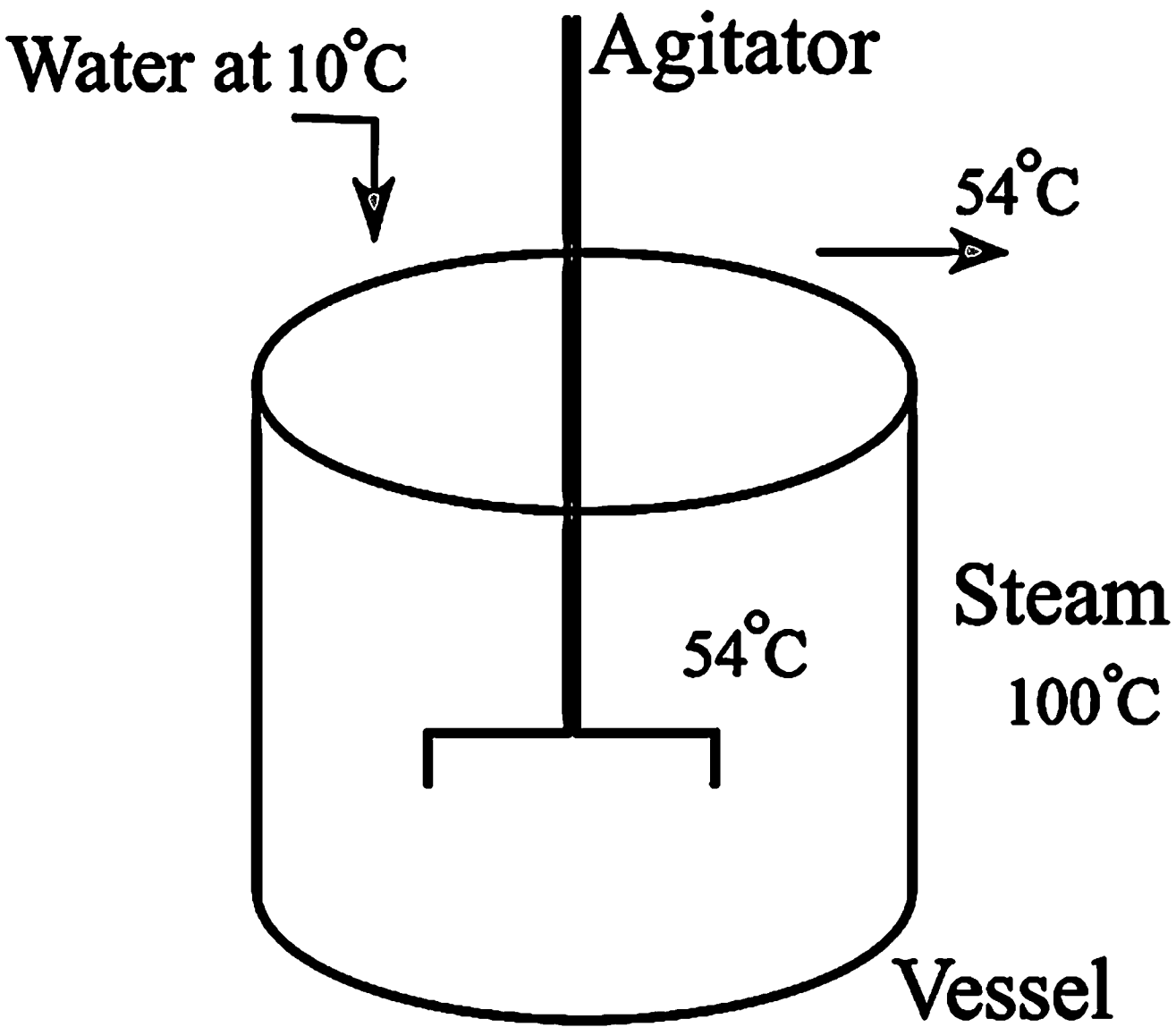


Question: 22.022P pg 1

A jacketted-agitated vessel, fitted with a turbine agitator, is used for heating a water stream from 10°C to 54°C. The average heat transfer coefficient for water at the vessel's inner-wall can be estimated from Nu = 0.76Re2/3 Re1/3. Saturated steam at 100°C condenses in the jacket, for which the average heat transfer coefficient in kW/m2·K is: *hO* = 13.1(*Tg* − *Tw*)−*0.25*. The vessel dimensions are: *Dt* = 0.6 m, *H* = 0.6 m and *Da* =0.2 m. The agitator speed is 60 rpm. Calculate the mass rate of water that can be heated in this agitated vessel steadily.

Step 1 of 9

Draw the diagram of jacketed-agitated vessel.



Step 2 of 9

Obtain the following properties of water at $54°C$ from table A-9, "properties of saturated water" in the text book.

Thermal conductivity, $k = 0.648\ W/m \cdot °C$

Density, $\rho = 985.8\ kg/m^3$

Absolute viscosity, $\mu = 0.513 \times 10^{-3}\ kg/m \cdot s$

Prandtl number, $Pr = 3.31$

Step 3 of 9

Calculate the average temperature of water.

$$T_m = \left(\frac{T_i + T_f}{2}\right)$$

Here, T_i is the initial temperature of water and T_f is the final temperature of water.

Substitute $10°C$ for T_i and $54°C$ for T_f .

$$T_m = \left(\frac{10+54}{2}\right)$$
$$= 32°C$$

Step 4 of 9

Obtain the specific heat at the average temperature of water at $32°C$ from Table A-9 "properties of saturated water" in the text book.

$$c_{pw} = 4178\ J/kg \cdot °C$$

Calculate the Reynolds number using the following relation:

$$Re = \frac{\rho n D_a^2}{\mu}$$

Substitute $985.8\ kg/m^3$ for ρ, 0.2 m for D_a, and $0.513 \times 10^{-3}\ kg/m \cdot s$ for μ .

$$Re = \frac{(985.8)\left(\frac{60}{60}\right)(0.2)^2}{0.513 \times 10^{-3}}$$
$$= 76865$$

Step 5 of 9

Calculate the Nusselts number using the following relation:

$$Nu = 0.76\, Re^{2/3}\, Pr^{1/3}$$

Substitute 76865 for Re and 3.31 for Pr .

$$Nu = 0.76\left[(76865)^{2/3} \times (3.31)^{1/3}\right]$$
$$= 2048$$

Step 6 of 9

Calculate the convective heat transfer coefficient on the inner wall of the vessel using the following relation:

$$h_i = \frac{k}{D_t} Nu$$

Substitute $0.6\ m$ for D_t, 2048 for Nu , and $0.648\ W/m \cdot °C$ for k .

$$h_i = \frac{0.648}{0.6} \times 2048$$

Substitute $0.6\,\text{m}$ for D_i, 2048 for Nu, and $0.648\,\text{W/m}^\circ\text{C}$ for k.

$$h_i = \frac{0.648}{0.6} \times 2048$$
$$= 2211\ \text{W/m}^2 \cdot ^\circ\text{C}$$

Step 7 of 9

Calculate the convective heat transfer coefficient on the outer side using the following relation:

$$h_o = 13100\left(T_s - T_w\right)^{-0.25}$$

Here, saturation steam temperature is T_s.

Substitute 100°C for T_s.

$$h_o = 13100\left(100 - T_w\right)^{-0.25} \quad \text{..... (1)}$$

Determine T_w using the following relation:

$$h_o\left(T_s - T_w\right) = h_i\left(T_w - 54\right)$$

Substitute $13100\left(100 - T_w\right)^{-0.25}$ for h_o, 100°C for T_s, and $2211\ \text{W/m}^2 \cdot ^\circ\text{C}$ for h_i.

$$\left[13100\left(100 - T_w\right)^{-0.25}\right]\left(100 - T_w\right) = 2211\left(T_w - 54\right)$$
$$13100\left(100 - T_w\right)^{0.75} = 2211\left(T_w - 54\right)$$
$$T_w = 89.2^\circ\text{C}$$

Substitute 89.2°C for T_w in equation (1).

$$h_o = 13100\left(100 - T_w\right)^{-0.25}$$
$$= 13100\left(100 - 89.2\right)$$
$$= 7226\ \text{W/m}^2 \cdot ^\circ\text{C}$$

Step 8 of 9

Calculate the overall heat transfer co-efficient using the following relation:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

Substitute $2211\ \text{W/m}^2 \cdot ^\circ\text{C}$ for h_i and $7226\ \text{W/m}^2 \cdot ^\circ\text{C}$ for h_o.

$$\frac{1}{U} = \frac{1}{2211} + \frac{1}{7226}$$
$$U = 1694\ \text{W/m}^2 \cdot ^\circ\text{C}$$

Step 9 of 9

Write the energy balance equation to calculate the mass flow rate of water.

$$\dot{m}_w c_{pw}\left(T_{out} - T_{in}\right)_{water} = UA_s\Delta T$$
$$\dot{m}_w c_{pw}\left(T_{out} - T_{in}\right)_{water} = U\left(\pi D_i H\right)\left(T_s - T_{out}\right)$$

Substitute $4178\ \text{J/kg} \cdot ^\circ\text{C}$ for c_{pw}, 54°C for T_{out}, 10°C for T_i, $1694\ \text{W/m}^2 \cdot ^\circ\text{C}$ for U,

0.6 m for D_i, 0.6 m for H, 100°C for T_s.

$$\dot{m}_w(4178)(54 - 10) = 1694 \times \pi \times 0.6 \times 0.6(100 - 54)$$
$$\dot{m}_w = 0.479\ \text{kg/s}$$
$$= 1725\ \text{kg/hr}$$

Therefore, the mass rate of water that can be heated in the agitated vessel steadily is $\boxed{1725\ \text{kg/hr}}$.

Question: 22.023P

What are the common approximations made in the analysis of heat exchangers?

Step 1 of 4

A heat exchanger to be selected should achieve a specified temperature change in a fluid stream of known mass flow rate, or to predict the outlet temperatures of the hot and cold fluid streams in a specified heat exchanger.

Step 2 of 4

Methods used in the analysis:

- LMTD Method(log mean temperature difference)
- Effectiveness-NTU Method

Step 3 of 4

Approximations made in the analysis:

- Heat exchangers usually operate for long periods of time with no change in their operating conditions, and then they can be modeled as steady-flow devices.
- As such, the mass flow rate of each fluid remains constant and the fluid properties such as temperature and velocity at any inlet and outlet remain the same.
- The fluid streams experience little or no change in their velocities and elevations, and thus the kinetic and potential energy changes are negligible.

Step 4 of 4

- The specific heat of a fluid can be treated as constant for a specified temperature range.
- Axial heat conduction along the tube can be considered negligible.
- The outer surface of the heat exchanger is assumed to be perfectly insulated, so that there is no heat loss to the surrounding medium, and any heat transfer thus occurs is between the two fluids only.

Question: 22.024P

Under what conditions will the temperature rise of the cold fluid in a heat exchanger be equal to the temperature drop of the hot fluid?

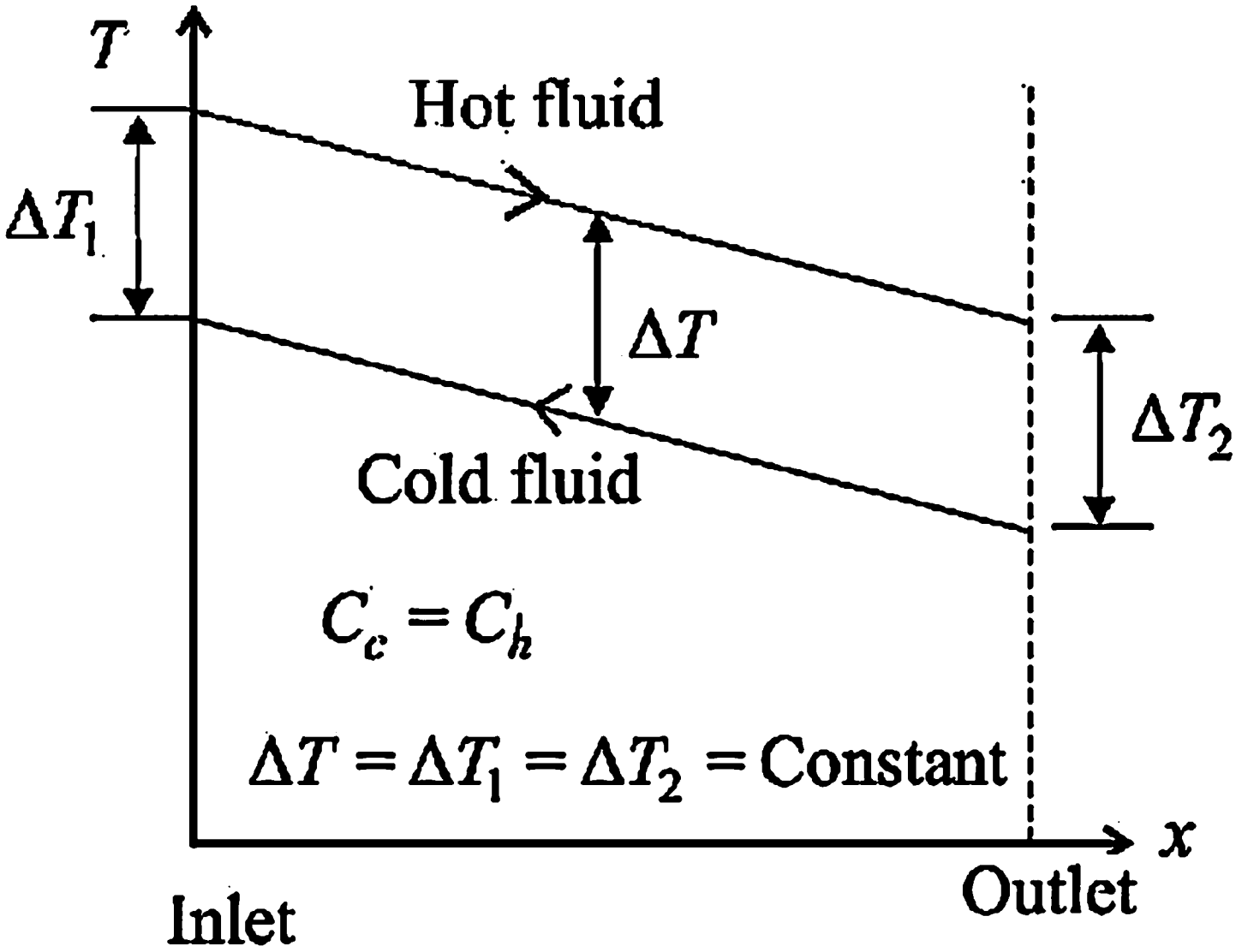
Step 1 of 1

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) = C_h(T_{h,in} - T_{h,out})$$

That is, the heat transfer rate in a heat exchanger is equal to the heat capacity rate of either fluid multiplied by the temperature change of that fluid.

The only time the temperature rise of a cold fluid is equal to the temperature drop of the hot fluid is when the heat capacity rates of the two fluids are equal to each other.

When $C_c = C_h$, $\Delta T_1 = \Delta T_1$



Question:

22.025P

What is the heat capacity rate? What can you say about the temperature changes of the hot and cold fluids in a heat exchanger if both fluids have the same capacity rate? What does a heat capacity of infinity for a fluid in a heat exchanger mean?

Step 1 of 3

The product of mass flow rate and specific heat of fluid is called as heat capacity rate.

Heat capacity rate for hot fluids is given by $C_h = \dot{m}_h c_{ph}$

Heat capacity rate for hot fluids is given by $C_c = \dot{m}_c c_{pc}$

Step 2 of 3

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out})$$

When $\dot{m}_c c_{pc} = \dot{m}_h c_{ph}$ then $\Delta T_c = \Delta T_h$

If the hot and cold fluids have the same heat capacity rate, it means that temperature difference of hot and cold fluids are equal. Then the temperature profiles of hot and cold fluids are straight lines.

Step 3 of 3

The heat capacity rate of the fluid during the phase change process must be infinity since the temperature change is practically zero.

That is $C = \dot{m}c_p \to \infty$ when $\Delta T \to 0$, so that heat transfer rate $Q = \dot{m}c_p \Delta T$ is an infinite quantity.

Question: 22.026P

Consider a condenser in which steam at a specified temperature is condensed by rejecting heat to the cooling water. If the heat transfer rate in the condenser and the temperature rise of the cooling water is known, explain how the rate of condensation of the steam and the mass flow rate of the cooling water can be determined. *Also,* explain how the total thermal resistance *R* of this condenser can be evaluated in this case.

Step 1 of 3

The mass flow rate of the cooling water is determined by

$$\dot{Q} = [\dot{m} C_p (\Delta T)]_{cooling\ water}$$

Here $(\Delta T)_{cooling\ water}$ & $\dot{Q}$ are given.

$$\dot{m}_{cooling\ water} = \frac{\dot{Q}}{c_p [T_{out} - T_{in}]}$$

Step 2 of 3

The rate of condensation of the steam is determined by

$$\dot{Q} = [\dot{m} h_{fg}]_{steam}$$

$$\dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}}$$

Step 3 of 3

The total resistance of the condenser is

$$R = \frac{\dot{Q}}{\Delta T_{cooling\ water}}$$

Question: 22.027P

Under what conditions is the heat transfer relation

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}}) = \dot{m}_h c_{ph}(T_{h,\text{in}} - T_{h,\text{out}})$$

valid for a heat exchanger?

Step 1 of 1

The heat transfer relation $\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out})$ is valid under the following conditions.

1. The heat exchanger is insulated from its surroundings in which case the only heat exchange is between the hot and cold fluids.
2. Axial conduction along the tubes is negligible.
3. Potential and kinetic energy changes are negligible.
4. The fluid specific heats are constants.
5. The overall heat transfer coefficient is constant.

Question: 22.028P

In the heat transfer relation $\dot{Q}$ = UAsΔTlm for a heat exchanger, what is ΔTlm called? How is it calculated for a parallel-flow and counter-flow heat exchanger?

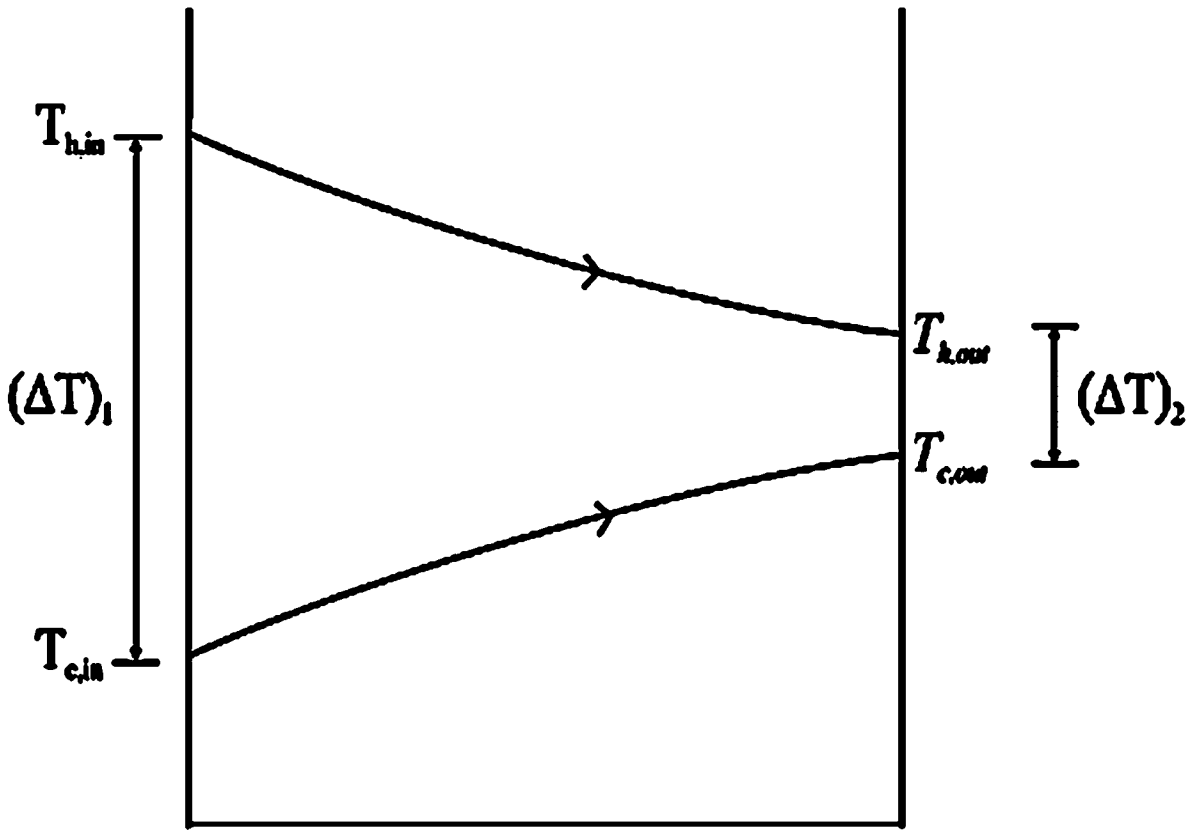
Step 1 of 2

In the heat transfer relation $\dot{Q} = UA_s\Delta T_{lm}$ where ΔT_{lm} is the log mean temperature difference.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

For parallel flow

$$\Delta T_1 = T_{h,in} - T_{c,in}$$
$$\Delta T_2 = T_{h,out} - T_{c,out}$$

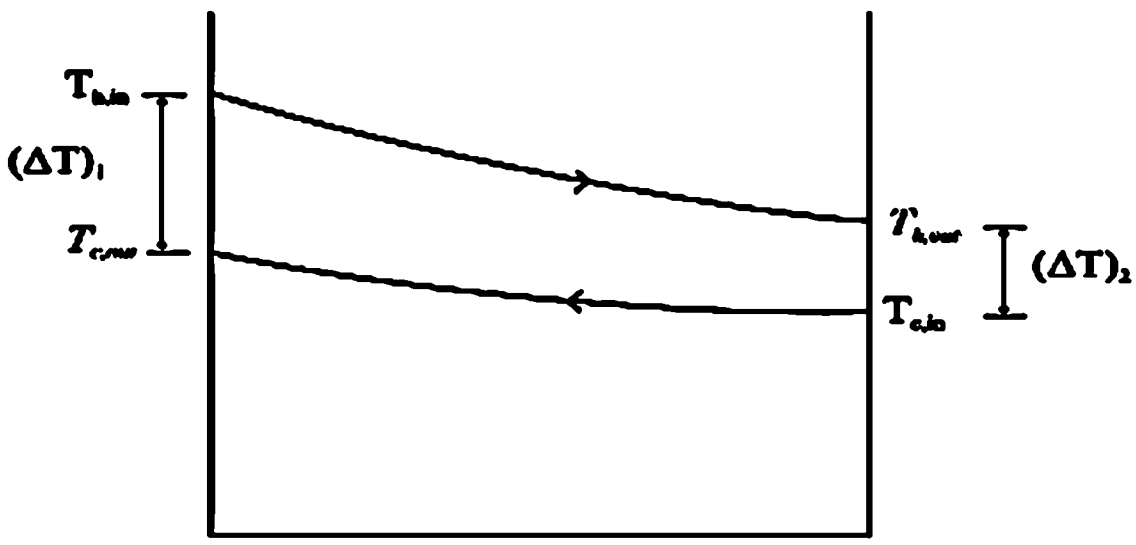


Parallel flow

Step 2 of 2

For counter flow

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$



Counter flow

Question: 22.029P

How does the log mean temperature difference for a heat exchanger differ from the arithmetic mean *temperature* difference? For specified inlet and outlet temperatures, which one of these two quantities is larger?

Step 1 of 3

The arithmetic mean temperature difference:

$$\Delta T_{am} = \frac{\Delta T_1 - \Delta T_2}{2}$$

Step 2 of 3

The logarithmic mean temperature difference ΔT_{lm} is obtained by tracing the actual temperature profile of the fluids along the heat exchanger and is an exact representation of the average temperature difference between the hot and cold fluids. It truly reflects the exponential decay of the local temperature difference.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

Step 3 of 3

Log mean temperature difference ΔT_{lm} is always less than arithmetic temperature difference ΔT_{am}.

Using ΔT_{am} in calculations instead of ΔT_{lm} will overestimate the rate of heat transfer in a heat exchanger between the two fluids.

Therefore, we should always use logarithmic mean temperature difference ΔT_{lm} when determining the rate of heat transfer in a heat exchanger.

Question: 22.030P

The temperature difference between the hot and cold fluids in a heat exchanger is given to be $\Delta T1$, at one end and $\Delta T2$, at the other end. Can the logarithmic temperature difference $\Delta T1m$ of this heat exchanger be greater than both $\Delta T1$, and $\Delta T2$? Explain.

Step 1 of 1

ΔT_{lm} Cannot be greater than both ΔT_1 and ΔT_2 because ΔT_{lm} gives the exact representation of the mean temperature difference between the hot and cold fluids.

Question: 22.031P

Can the outlet temperature of the cold fluid in a heat exchanger be higher than the outlet temperature of the hot fluid in a parallel-flow heat exchanger? How about in a counter-flow heat exchanger? Explain.

Step 1 of 1

In case of parallel flow heat exchanger hot and cold fluids are in the same direction and the temperature of the hot fluid decreases and the temperature of the cold fluid increases across the heat exchanger, but the temperature of the cold fluid never exceed that of the hot fluid no matter how long the heat exchanger is.

In case of the counter flow heat exchanger hot and cold fluids enter the heat exchanger from opposite ends, and the outlet temperature of the cold fluid in this case may exceed the outlet temperature of the hot fluid. In the limiting case, the cold fluid will be heated to the inlet temperature of the hot fluid. However, the outlet temperature of the cold fluid never exceeds the inlet temperature of the hot fluid, since this would be a violation of the 2^{nd} law of thermodynamics.

Question: 22.032P

Explain how the LMTD method can be used to determine the heat transfer surface area of a multipass shell-and-tube heat exchanger when all the necessary information, including the outlet temperatures, is given.

Step 1 of 1

Basic relation to determine the heat transfer surface area is

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

Since $\dot{Q} = UA_sF\Delta T_{lm}$

Overall heat transfer coefficient U will be given.

So in order to calculate A_s, correction factor F, ΔT_{lm} must be calculated as follows.

For F:

We have to find $P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$ and $R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$.

At this P & R, F can be found from table-correction factor F for common shell and tube and cross flow heat exchangers

For ΔT_{lm}:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

Question: 22.033P

In the heat transfer relation $\dot{Q} = UA_s F \Delta T_{lm}$ for a heat exchanger, what is the quantity *F* called? What does it represent? Can *F* be greater than one?

Step 1 of 1

In the heat transfer relation $\dot{Q} = [\dot{m} C_p (\Delta T)]_{cooling\ water}$

F is called correction factor, which depends on the geometry of the heat exchanger and the inlet and outlet temperature of the hot and cold fluid streams.

The correction factor is less than unity for a cross-flow and multipass shell-and-tube heat exchanger. That is, $F \leq 1$.

The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger.

Thus correction factor F is a measure of deviation of the ΔT_{lm} from the corresponding values for the counter flow.

Question: 22.034P

When the outlet temperatures of the fluids in a heat exchanger are not known, is it still practical to *use* the LMTD method? Explain.

Step 1 of 1

The LMTD method could still be used when outlet temperatures of the hot and cold fluids are not given, but the procedure would require tedious iterations, and thus it is not practical. In an attempt to eliminate the iterations from the solution of such problems, it is preferable to use NTU method.

Question: 22.035P

For specified inlet and outlet temperatures, for what kind of heat exchanger will the ΔT_{lm} be greatest: double-pipe parallel-flow, double-pipe counter-flow, cross-flow, or multipass shell-and-tube heat exchanger?

Step 1 of 1

For specified inlet and outlet temperature ΔT_{lm} will be more for double-pipe counter flow heat exchanger.

Question: 22.036P

A counter-flow heat exchanger is stated to have an overall heat transfer coefficient of 284 W/m2·K when operating at design and clean conditions. Hot fluid enters the tube side at 93°C and exits at 71°C, while cold fluid enters the shell side at 27°C and exits at 38°C. After a period of use, built-up scale in the heat exchanger gives a fouling factor of 0.0004 m2·K/W. If the surface area is 93 m2, determine (*a*) the rate of heat transfer in the heat exchanger and (*b*) the mass flow rates of both hot and cold fluids. Assume both hot and cold fluids have a spe*cific* heat of 4.2 kJ/kg·K.

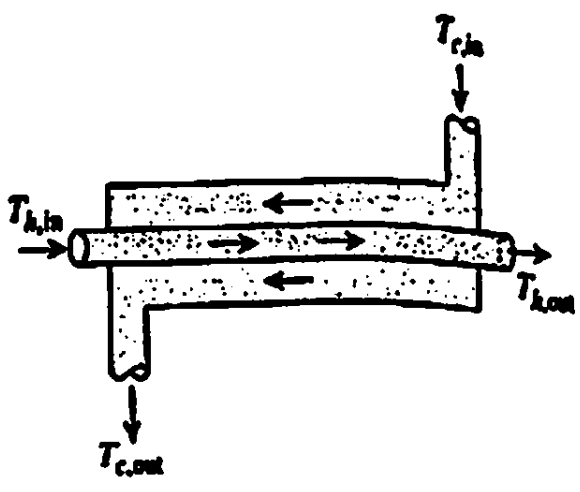


FIGURE P22–36

Step 1 of 5

(*a*)

Calculate the overall heat transfer coefficient, $U_{\text{W/scale}}$ after a period of use due to scale build up from the following relation.

$$\frac{1}{U_{\text{W/scale}}} = R_f + \frac{1}{U_{\text{W/No scale}}}$$

Here, $U_{\text{W/No scale}}$ is the overall heat transfer coefficient at design and clean conditions and

R_f is the fouling factor.

Substitute $284\ \text{W/m}^2\cdot\text{K}$ for $U_{\text{W/No scale}}$ and $0.0004\ \text{m}^2\cdot\text{K/W}$ for R_f.

$$\frac{1}{U_{\text{W/scale}}} = 0.0004\ \text{m}^2\cdot\text{K/W} + \frac{1}{284\ \text{W/m}^2\cdot\text{K}}$$

$$\frac{1}{U_{\text{W/scale}}} = 0.00392\ \text{W/m}^2\cdot\text{K}$$

$$U_{\text{W/scale}} = 255\ \text{W/m}^2\cdot\text{K}$$

Step 2 of 5

Calculate the temperature difference, ΔT_1.

$$\Delta T_1 = T_{h,i} - T_{c,o}$$

Here, $T_{h,i}$ is the inlet temperature of the hot fluid and $T_{c,o}$ is the outlet temperature of cold fluid.

Substitute $93°\text{C}$ for $T_{h,i}$ and $38°\text{C}$ for $T_{c,o}$.

$$\Delta T_1 = 93°\text{C} - 38°\text{C}$$
$$= 55°\text{C}$$

Calculate the temperature difference, ΔT_2.

$$\Delta T_2 = T_{h,o} - T_{c,i}$$

Here, $T_{h,o}$ is the outlet temperature of hot fluid and $T_{c,i}$ is the inlet temperature of the cold fluid.

Substitute $71°\text{C}$ for $T_{h,o}$ and $27°\text{C}$ for $T_{c,i}$.

$$\Delta T_2 = 71°\text{C} - 27°\text{C}$$
$$= 44°\text{C}$$

Step 3 of 5

Calculate the log mean temperature difference of the heat exchanger from the following relation.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

Substitute $55°\text{C}$ for ΔT_1 and $44°\text{C}$ for ΔT_2.

$$\Delta T_{lm} = \frac{55°\text{C} - 44°\text{C}}{\ln(55°\text{C}/44°\text{C})}$$
$$= 49.295\ \text{K}$$

Step 4 of 5

Calculate the rate of heat transfer in the heat exchanger from the following relation.

$$\dot{Q} = U_{\text{W/scale}} A_s (\Delta T_{lm})$$

Here, surface area of the heat exchanger is A_s.

Substitute $255\ \text{W/m}^2\cdot\text{K}$ for $U_{\text{W/scale}}$, $93\ \text{m}^2$ for A_s, and $49.295\ \text{K}$ for ΔT_{lm}.

$$\dot{Q} = (255\ \text{W/m}^2\cdot\text{K}) \times (93\ \text{m}^2) \times (49.295\ \text{K})$$
$$= 1.169\times10^6\ \text{W}$$

Therefore, the rate of heat transfer in the heat exchanger is $\boxed{1.169\times10^6\ \text{W}}$.

Step 5 of 5

(*b*)

Calculate the mass flow rate of hot fluid from the following heat transfer rate relation.

$$\dot{Q} = (\dot{m} c_p \Delta T)_h$$
$$= \dot{m}_h c_p (\Delta T)_h$$
$$= \dot{m}_h c_p (T_{h,i} - T_{h,o})$$

Here, c_p is the heat transfer coefficient, $\dot{m}_h$ is the mass flow rate of the hot fluid, and $(\Delta T)_h$ is the temperature difference between inlet and outlet of hot fluid.

Substitute $1.17\times10^6\ \text{W}$ for $\dot{Q}$, $4.2\times1000\ \text{J/kg·K}$ for c_p, $93°\text{C}$ for $T_{h,i}$, and $71°\text{C}$ for $T_{h,o}$.

$$1.17\times10^6\ \text{W} = m_h \times (4.2\times1000\ \text{J/kg·K}) \times (93-71)\text{K}$$

$$m_h = \frac{1.17\times10^6\ \text{W}}{(4{,}200\ \text{J/kg·K}) \times (22\text{K})}$$
$$= 12.66\ \text{kg/s}$$

Therefore, the mass flow rate of hot fluid is $\boxed{12.66\ \text{kg/s}}$.

Calculate the mass flow rate of cold fluid from the following heat transfer rate relation.

$$\dot{Q} = (\dot{m} c_p \Delta T)_c$$
$$= \dot{m}_c c_p (\Delta T)_C$$
$$= \dot{m}_c c_p (T_{c,o} - T_{c,i})$$

Here, c_p is the heat transfer coefficient, $\dot{m}_c$ is the mass flow rate of cold fluid, and $(\Delta T)_c$ is the temperature difference between outlet and inlet of cold fluid.

Substitute $1.17\times10^6\ \text{W}$ for $\dot{Q}$, $4.2\times1000\ \text{J/kg·K}$ for c_p, $38°\text{C}$ for $T_{c,o}$, and $27°\text{C}$ for $T_{c,i}$.

$$1.17\times10^6\ \text{W} = \dot{m}_c \times (4.2\times1000\ \text{J/kg·K}) \times (38-27)\text{K}$$

$$\dot{m}_c = \frac{1.17\times10^6\ \text{W}}{(4{,}200\ \text{J/kg·K}) \times (11\text{K})}$$
$$= 25.32\ \text{kg/s}$$

Therefore, the mass flow rate of cold fluid is $\boxed{25.32\ \text{kg/s}}$.

Question: 22.037P

A single-pass cross-flow heat exchanger is used to cool jacket water (cp = 1.0 Btu/lbm·°F) of a diesel engine from 190°F to 140°F, using air (cp = 0.245 Btu/lbm·°F) with inlet temperature of 90°F. Both air flow and water flow are unmixed. If the water and air mass flow rates are 92,000 lbm/h and 400,000 lbm/h, respectively, determine the log mean temperature difference for this heat exchanger.

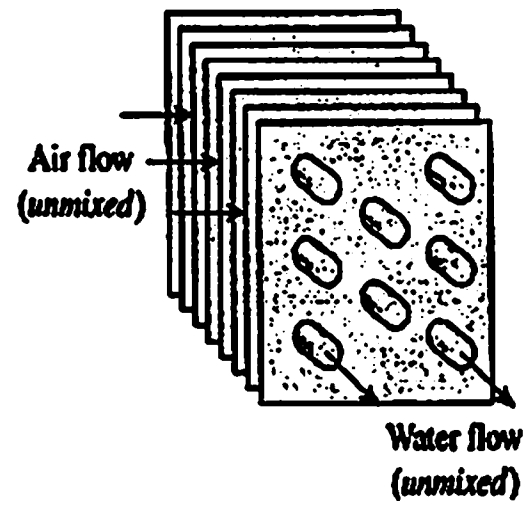


FIGURE P22–37E

Step 1 of 5

The specific heat of water, $c_{p,h} = 1 \text{ Btu/lbm} \cdot °\text{F}$

The specific heat of air, $c_{p,c} = 0.245 \text{ Btu/lbm} \cdot °\text{F}$

Inlet temperature of water, $T_{h,i} = 190°\text{F}$

Exit temperature of water, $T_{h,e} = 140°\text{F}$

Inlet temperature of air, $T_{c,i} = 90°\text{F}$

Mass flow rate of water, $\dot{m}_h = 92{,}000 \text{ lbm/h}$

Mass flow rate of air, $\dot{m}_c = 400{,}000 \text{ lbm/h}$

Step 2 of 5

Heat given out by the jacket water:

$$\begin{aligned}\dot{Q} &= \left(\dot{m}c_p\right)_h \left(T_{h,i} - T_{h,e}\right) \\ &= 92{,}000 \times 1 \times (190 - 140) \\ &= 4{,}600{,}000 \text{ Btu/h}\end{aligned}$$

Step 3 of 5

This amount of heat is absorbed by the air. Therefore, the expression for heat transfer is given by,

$$\begin{aligned}\dot{Q} &= \left(\dot{m}c_p\right)_c \left(T_{c,e} - T_{c,i}\right) \\ &= 400{,}000 \times 0.245 \times \left(T_{c,e} - 90\right) \\ T_{c,e} &= \frac{4{,}600{,}000}{400{,}000 \times 0.245} + 90 \\ &= 136.938°\text{F}\end{aligned}$$

Step 4 of 5

Determination of Log Mean Temperature Difference:

$$\begin{aligned}\Delta T_1 &= T_{h,i} - T_{c,e} \\ &= 190 - 136.938 \\ &= 53.061°\text{C}\end{aligned}$$

$$\begin{aligned}\Delta T_2 &= T_{h,e} - T_{c,i} \\ &= 140 - 90 \\ &= 50°\text{C}\end{aligned}$$

Step 5 of 5

The log mean temperature difference of the given heat exchanger is given by,

$$\begin{aligned}\Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)} \\ &= \frac{53.061 - 50}{\ln\left(53.061/50\right)} \\ &= 51.515 \text{ °F}\end{aligned}$$

Therefore, the log mean temperature difference is found to be $\boxed{\Delta T_{lm} = 51.515 \text{ °F}}$.

Question:

Ethylene glycol is heated from 25°C to 40°C at a rate of 1.5 kg/s in a horizontal copper tube (k = 386 W/m·K) with an inner diameter of 2.0 cm and an outer diameter of 2.5 cm. A saturated vapor (Tg = 110°C) condenses on the outside-tube surface with the heat transfer coefficient (in kW/m2·K) given by $9.2/(Tg - Tw)0.25$, where Tw is the average outside-tube wall temperature. What tube length must be used? Take the properties of ethylene glycol to be ρ = 1109 kg/m3, cp = 2428 kJ/kg·K, k = 0.253 W/m·K, μ = 0.01545 kg/m·s, and Pr = 148.5.

Step 1 of 10

Initial temperature of the ethylene glycol, $T_i = 25°C$

Final temperature of the ethylene glycol, $T_f = 40°C$

Mass flow rate is given by, $m = 1.5\ kg/s$

Thermal conductivity of copper, $k = 386\ W/m \cdot K$

Inner diameter of the copper tube, $D_i = 2.0cm = 0.2m$

Outer diameter of the copper tube, $D_o = 2.5cm = 0.25m$

Saturated waster temperature, $T_g = 110°C$

Let L be the length of tube.

Step 2 of 10

Heat transfer coefficient of the condenser,

$$h_o = 9.2/(T_g - T_w)^{0.25}\ kW/m^2 \cdot K$$
$$= 9200/(T_g - T_w)^{0.25}\ W/m^2 \cdot K$$

Average temperature of the ethylene glycol, $T_{b,av} = 30°C$

Properties of ethylene glycol,

Density of ethylene glycol, $\rho = 1109\ kg/m^3$

Specific heat at constant pressure, $c_p = 2428\ kJ/kg \cdot K$

Thermal conductivity of ethylene glycol, $k_g = 0.253\ W/m \cdot K$

Viscosity of ethylene glycol, $\mu = 0.01545\ kg/m \cdot s$

Prandtl number, $Pr = 148.5$

Step 3 of 10

The heat transfer coefficient,

$$\frac{1}{U_o A_o} = \frac{1}{h_0 A_o} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_i A_i}$$ (since surface area of a cylinder, $A = \pi DL$)

$$\frac{1}{U_o \pi D_o L} = \frac{1}{h_o \pi D_o L} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_i \pi D_i L}$$

$$\frac{1}{U_o} = \frac{1}{h_0} + \frac{\pi D_o L \ln(D_o/D_i)}{2\pi kL} + \frac{\pi D_o L}{h_i \pi D_i L}$$

$$= \frac{1}{h_0} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{D_o}{h_i D_i} \quad \text{..... (1)}$$

Step 4 of 10

Heat lost to the condenser must be equal to the heat transfer from the ethylene glycol to the outer tube wall. We have,

$$\dot{Q} = h_o A_o (T_g - T_w)$$
$$h_o A_o (T_g - T_w) = \dot{Q}$$

$$h_o A_o (T_g - T_w) = \frac{T_w - T_{b,av}}{\left[\frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_i A_i}\right]}$$ (but its know, $h_o = 9.2/(T_g - T_w)^{0.25}$)

$$\left[\frac{9200}{(T_g - T_w)^{0.25}}\right] \pi D_o L (T_g - T_w) = \frac{T_w - T_{b,av}}{\left[\frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_i \pi D_i L}\right]}$$

$$9200(T_g - T_w)^{0.75} = \frac{T_w - T_{b,av}}{\left[\frac{\pi D_o L \ln(D_o/D_i)}{2\pi kL} + \frac{\pi D_o L}{h_i \pi D_i L}\right]}$$

$$= \frac{T_w - T_{b,av}}{\frac{D_o \ln(D_o/D_i)}{2k} + \frac{D_o}{h_i D_i}} \quad \text{..... (2)}$$

Step 5 of 10

The velocity of flow is given by,

$$V = \frac{\dot{m}}{\rho A}$$
$$= \frac{\dot{m}}{\rho(\pi D_i^2/4)}$$
$$= \frac{1}{1109\frac{\pi}{4}(0.02)^2}$$
$$= 2.870\ m/s$$

Step 6 of 10

Reynolds number is given by,

$$Re = \frac{\rho V D}{\mu}$$

Step 6 of 10

Reynolds number is given by,

$$\begin{aligned}\mathrm{Re} &= \frac{\rho V D}{\mu}\\ &= \frac{11092.870(0.02)}{0.01545}\\ &= 4120.16\end{aligned}$$

This is a transition flow we will use the equation for fully developed turbulent flow in this case.

Nusselt number is given by,

$$\begin{aligned}Nu &= 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\\ &= 0.023(4120.16)^{0.8}(148.5)^{0.4}\\ &= 132.54\end{aligned}$$

Heat transfer coefficient of inner surface,

$$\begin{aligned}h_i &= \frac{kNu}{D_i}\\ &= \frac{132.54\times 0.253}{0.02}\\ &= 1676.64\ \mathrm{W/m^2\cdot K}\end{aligned}$$

Step 7 of 10

Substituting the value of heat transfer coefficient h_i and other parameters in equation (2) we have,

$$9200(T_g - T_w)^{0.75} = \frac{T_w - T_{b,mix}}{\frac{D_o \ln(D_o/D_i)}{2k} + \frac{D_o}{h_i D_i}}$$

$$\begin{aligned}9200(110 - T_w)^{0.75} &= \frac{T_w - 30^\circ}{\frac{0.025\ln[0.025/0.02]}{2\times 385} + \frac{0.025}{1677\times 0.02}}\\ &= \frac{T_w - 30}{7.24\times 10^{-6} + 745.4\times 10^{-6}}\\ &= \frac{T_w - 30}{752.65\times 10^{-6}}\\ T_w &= 91.58^\circ\mathrm{C}\end{aligned}$$

Step 8 of 10

The overall heat transfer coefficient from equation (1),

$$\begin{aligned}\frac{1}{U_o} &= \frac{1}{h_o} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{D_o}{h_i D_i}\\ &= \frac{(T_g - T_w)^{0.25}}{9200} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{D_o}{h_i D_i}\\ &= \frac{(110 - 91.58)^{0.25}}{9200} + \frac{0.025\ln[0.025/0.02]}{2\times 385} + \frac{0.025}{1677\times 0.02}\\ &= 9.706\times 10^{-4}\\ U_o &= 1030.3\ \mathrm{W/m^2.^\circ C}\end{aligned}$$

Step 9 of 10

Heat transfer is given by,

$$\begin{aligned}\dot{Q} &= \dot{m} c_p (T_f - T_i)\\ &= 1\times 2428(40 - 20)\\ &= 48560\ W\end{aligned}$$

The log-mean temperature difference for this heat exchanger,

$$\begin{aligned}\Delta T_{lm} &= \frac{(T_g - T_i) - (T_g - T_f)}{\ln\left(\frac{T_g - T_i}{T_g - T_f}\right)}\\ &= \frac{(110^\circ - 20^\circ) - (110^\circ - 40^\circ)}{\ln\left(\frac{110^\circ - 20^\circ}{110^\circ - 40^\circ}\right)}\\ &= 79.58^\circ\mathrm{C}\end{aligned}$$

Step 10 of 10

The length of the heat exchange can be calculated as,

$$\begin{aligned}\dot{Q} &= U_o A_o \Delta T_{lm}\\ &= U_o \pi D_o L \Delta T_{lm}\\ L &= \frac{\dot{Q}}{U_o \pi D_o \Delta T_{lm}}\\ &= \frac{48560}{1030\times \pi \times 0.025\times 79.58^\circ}\\ &= 7.54\end{aligned}$$

Therefore the length of the heat exchanger is obtained as $\boxed{7.54\,\mathrm{m}}$

Question: 22.039P

A test is conducted to determine the overall heat transfer coefficient in a shell-and-tube oil-to-water heat exchanger that has 24 tubes of internal diameter 1.2 cm and length 2 m in a single shell. Cold water (*cp* = 4180 J/kg·K) enters the tubes at 20°C at a rate of 3 kg/s and leaves at 55°C. Oil (*cp* = 2150 J/kg·K) flows through the shell and is cooled from 120°C to 45°C. Determine the overall heat transfer coefficient *Ui* of this heat exchanger based on the inner surface area of the tubes.

Step 1 of 10

Given

Number of tubes $N = 24$

Internal diameter $D_i = 1.2\text{cm}$

$$D_i = 0.012\text{m}$$

Length of shell $L = 2\text{m}$

For water:

$$T_{c,in} = 20^0\text{C}$$

$$T_{c,out} = 55^0\text{C}$$

$$\dot{m} = 3\text{kg/s}$$

The specific heat of water, $c_p = 4180\ \text{J/kg.}^\circ\text{C}$

For oil:

$$T_{h,in} = 120^0\text{C}$$

$$T_{h,out} = 45^0\text{C}$$

Step 2 of 10

The rate of heat transfer in this radiator from the cold water to the oil is determined by an energy balance.

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{water}$$

$$= (3\ \text{kg/s})(4180\ \text{J/kg.}^\circ\text{C})[55\text{-}20]^\circ\text{C}$$

$$\dot{Q} = 438.9\ \text{kW}$$

Step 3 of 10

The total surface area of tube:

$$A_i = N\pi D_i L$$

$$= (24)(\pi)(.012\ \text{m})(2\ \text{m})$$

$$A_i = 1.8096\ \text{m}^2$$

Step 4 of 10

The overall heat transfer coefficient can be found using the relation

$$\dot{Q} = U_i A_i F \Delta T_{lm}\ \text{kW}$$

$$U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm}}\ \text{W/m}^2\text{.}^\circ\text{C}$$

Where F is the correction factor and

ΔT_{lm} is the log mean temperature difference of the heat exchanger.

Step 5 of 10

$$\Delta T_1 = (T_{h,in} - T_{c,out})$$

$$= (120 - 55)$$

$$\Delta T_1 = 65^\circ\text{C}$$

Step 6 of 10

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= 45 - 20$$

$$\Delta T_2 = 25^\circ\text{C}$$

Step 7 of 10

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined by

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{65 - 25}{\ln\left(\frac{65}{25}\right)}$$

$$\Delta T_{lm} = 41.862^\circ\text{C}$$

Step 8 of 10

And

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

$$= \frac{55 - 20}{120 - 20}$$

$$P = 0.35$$

Step 9 of 10

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$

$$= \frac{120 - 45}{55 - 20}$$

$$R = 2.15$$

Step 10 of 10

At these P and R values $F = 0.7$ (From correction factor F charts for common shell and tube and cross flow heat exchangers (a))

By substituting the values overall heat transfer coefficient become

$$U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm}}\ \text{W/m}^2\text{°C}$$

$$= \frac{438.9\ \text{W}}{(1.80956\ \text{m}^2)(0.7)(41.862^\circ\text{C})}$$

$$\therefore U_i = 8.277\ \text{kW/m}^2\text{°C}$$

Question: 22.040P

A stream of hydrocarbon (*cp* = 2.2 kJ/kg·K) is cooled at a rate of 720 kg/h from 150°C to 40°C in the tube side of a double-pipe counter-flow heat exchanger. Water (*cp* =4.18 kJ/kg·K) enters the heat exchanger at 10°C at a rate of 540 kg/h. The outside diameter of the inner tube is 2.5 cm, and its length is 6.0 m. Calculate the overall heat transfer coefficient.

Step 1 of 5

Flow rate of hydrocarbon,

$$\begin{aligned}\dot{m}_h &= 720 \text{ kg/h} \\ &= \frac{720}{3600} \\ &= 0.2 \text{ kg/s}\end{aligned}$$

Flow rate of water,

$$\begin{aligned}\dot{m}_w &= 540 \text{ kg/h} \\ &= \frac{540}{3600} \\ &= 0.15 \text{ kg/s}\end{aligned}$$

Step 2 of 5

The rate of heat transfer from the hydrocarbon,

$$\dot{Q} = \dot{m} c_{p,h} (T_{out} - T_{in})$$

Here T_{in} is the inlet temperature of hydrocarbon, T_{out} is the outlet temperature of hydrocarbon, $c_{p,h}$ Specific heat of hydrocarbon, and $c_{p,w}$ Specific heat of water

Substitute 150°C for T_{in}, 40°C for T_{out}, 2.2 kJ/kg·K for $c_{p,h}$, 4.18 kJ/kg·K for $c_{p,w}$

$$\begin{aligned}\dot{Q} &= 0.2 \times (2.2) \times (150 - 40) \\ &= 48.4 \text{ kW}\end{aligned}$$

The heat transfer from the hydrocarbon to water,

$$\dot{Q} = \left(\dot{m} c_p (T_{out} - T_{in})\right)_w$$

The temperature of the heat exchanger, $T_{w_in} = 10°C$

$$\begin{aligned}48.4 &= 0.15 \times (4.18) \times (T_{w_out} - 10) \\ T_{w_out} &= 87.19°C\end{aligned}$$

Step 3 of 5

Since it is a counter flow heat exchanger the temperature difference between the hydrocarbon and the water at the two ends of the condenser is

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{w,out} \\ &= (150 - 87.19) \\ &= 62.81°C \\ \Delta T_2 &= T_{h,out} - T_{w,in} \\ &= 40 - 10 \\ &= 30°C\end{aligned}$$

Step 4 of 5

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which can be determined by using below formula,

$$\begin{aligned}\Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \\ &= \frac{62.81 - 30}{\ln\left(\frac{62.81}{30}\right)} \\ &= 44.4°C\end{aligned}$$

Step 5 of 5

The overall heat transfer coefficient can be determined as

$$\begin{aligned}\dot{Q} &= U A_s \Delta T_{lm} \\ &= U(\pi D L) \Delta T_{lm}\end{aligned}$$

Here D is the Diameter of the tube, and L is Length of the tube

Substitute 2.5 cm for D for, 6 m for L

$$\begin{aligned}48.4 &= U(\pi(0.025)6)44.4 \\ U &= 2.313 \text{ kW/m}^2\text{·K}\end{aligned}$$

Therefore, the overall heat transfer coefficient is $\boxed{U = 2.313 \text{ kW/m}^2\text{·K}}$.

Question: 22.041P

A shell-and-tube heat exchanger is used for heating 14 kg/s of oil (*cp* = 2.0 kJ/kg·K) from 20°C to 46°C. The heat exchanger has 1-shell pass and 6-tube passes. Water enters the shell side at 80°C and leaves at 60°C. The overall heat transfer coefficient is estimated to be 1000 W/m2·K. Calculate the rate of heat transfer and the heat transfer area.

Step 1 of 4

Type of heat exchanger: shell and tube

Mass flow rate of oil, $m_o = 14\text{kg/s}$

Specific heat of oil, $c_{po} = 2\text{kJ/kg-K}$

Inlet of temperature of the oil, $T_{i,o} = 20°\text{C}$

Outlet temperature of the oil, $T_{o,o} = 46°\text{C}$

Number of tubes, $n = 6$

Inlet temperature of the water, $T_{w,i} = 80°\text{C}$

Outlet temperature of the water, $T_{w,o} = 60°\text{C}$

Overall heat transfer coefficient, $U = 1000\,\text{W/m}^2\text{-K}$

Step 2 of 4

Rate of heat transfer for oil is

$$\begin{aligned} Q_0 &= m_o c_{po}\left(T_{0,o} - T_{i,o}\right) \\ &= 14 \times 2(46-20) \\ &= 728\,\text{kW} \end{aligned}$$

From the energy balance equation,

Rate of heat loss by water is equal to rate of heat gain by oil.

Hence, rate of heat transfer is $\boxed{\dot{Q} = Q_0 = 728\,\text{kW}}$

Step 3 of 4

From the equation of Log mean temperature difference, we have

$$\begin{aligned} \Delta T_{lm} &= \frac{\left(T_{w,i} - T_{i,o}\right) - \left(T_{w,o} - T_{o,o}\right)}{\ln\left(\frac{\left(T_{w,i} - T_{i,o}\right)}{\left(T_{w,o} - T_{o,o}\right)}\right)} \\ &= \frac{(80-20)-(60-40)}{\ln\left(\frac{80-20}{60-40}\right)} \\ &= \frac{40}{1.098640} \\ &= 36.408°\text{C} \end{aligned}$$

Step 4 of 4

Now, rate of heat transfer in heat exchanger can be determined by

$$\dot{Q} = Q_o = UA_s\left(\Delta T_{lm}\right)$$

$$728 \times 10^3\,\text{W} = 1000 \times A_s(36.408)$$

$$A_s = 19.99\,\text{m}^2$$

Therefore, heat transfer surface area is $\boxed{A_s = 19.99\,\text{m}^2}$.

Question: 22.042P

A double-pipe parallel-flow heat exchanger is to heat water *(cp* = 4180 J/kg·K) from 25°C to 60°C at a rate of 0.2 kg/s. The heating is to be accomplished by geothermal water *(cp* =4310 J/kg·K) available at 140°C at a mass flow rate of 0.3 kg/s. The inner tube is thin-walled and has a diameter of 0.8 cm. If the overall heat transfer coefficient of the heat exchanger is 550 W/m2·K, determine the length of the tube required to achieve the desired heating.

Step 1 of 7

Given data:

Specific heat of water, $c_p = 4180\ \text{J/kg}^\circ\text{C}$

Specific heat of geothermal water, $c_p = 4310\ \text{J/kg}^\circ\text{C}$

Mass flow rates of water, $\dot{m} = 0.2\ \text{kg/s}$

Mass flow rates of geothermal water, $\dot{m} = 0.3\ \text{kg/s}$

Overall heat transfer coefficient, $U = 550\ \text{W/m}^2\cdot^\circ\text{C}$

Step 2 of 7

The rate of heat transfer from the water is determined by an energy balance.

$$\dot{Q} = \left[\dot{m}c_p\left(T_{out} - T_{in}\right)\right]_{water}$$

$$= (0.2\ \text{kJ/s})(4180\ \text{J/kg}^\circ\text{C})[60\text{-}25]^\circ\text{C}$$

$$= 29260\ \text{kW}$$

$$= 29.26\ \text{kW}$$

Step 3 of 7

The rate of heat transfer from the geothermal water is determined by

$$\dot{Q} = \left[\dot{m}c_p\left(T_{in} - T_{out}\right)\right]_{Geothermal\ water}$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}_{max}}{\dot{m}c_p}$$

$$= 140^\circ\text{C} - \frac{29260\ \text{W}}{(0.3\ \text{kg/s})(4.31\ \text{kJ/kg}^\circ\text{C})}$$

$$\therefore T_{h,out} = 117.4^\circ\text{C}$$

Step 4 of 7

For this parallel- flow heat exchanger LMTD determined to be

$$\Delta T_1 = T_{h,in} - T_{c,in}$$

$$= (140 - 25)$$

$$= 115^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= (117.4 - 60)$$

$$= 57.4^\circ\text{C}$$

Step 5 of 7

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{115 - 57.4}{\ln\left(115/57.4\right)}$$

$$= 82.9^\circ\text{C}$$

Step 6 of 7

Heat transfer surface area on the tube side is determined to be

$$\dot{Q} = UA_s\Delta T_{lm}$$

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

By substituting all values heat transfer surface area becomes

$$A_s = \frac{29260\ \text{W}}{(550\ \text{W/m}^2.^\circ\text{C})(82.9^\circ\text{C})}$$

$$\therefore A_s = 0.6418\ \text{m}^2$$

Step 7 of 7

Length of the tube required to achieve the desired heating is calculated by

$$A_s = \pi DL$$

$$L = \frac{A_s}{\pi D}$$

$$= \frac{0.6418\ \text{m}^2}{\pi(0.008\ \text{m})}$$

$\therefore$ The length of the tube $\boxed{L = 25.5\ \text{m}}$

Question: 22.043P

Reconsider Prob. 22–42. Using EES (or other) software, investigate the effects of temperature and mass flow rate of geothermal water on the length of the tube. Let the temperature vary from 100°C to 200°C, and the mass flow rate from 0.1 kg/s to 0.5 kg/s. Plot the length of the tube as functions of temperature and mass flow rate, and discuss the results.

Step 1 of 5

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box.

```
"GIVEN"
c_pc=4180[J/kg-K]
T_c_in=25[C]
T_c_out=60[C]
"T_h_out=200[C]"
m_dot_c=0.2[kg/s]
c_ph=4310[J/kg-K]
"T_h_in is varied from 100 [C] to 200 [C]"
"m_dot_h is varied from 0.1 [kg/s] to 0.5 [kg/s]"
D=0.8[cm]
U=550[W/m^2-K]
"Analysis"
Q_dot= m_dot_c* c_pc*(T_c_out- T_c_in)
Q_dot = m_dot_h* c_ph*(T_h_in- T_h_out)
DELTAT_1=T_h_in-T_c_in
DELTAT_2=T_h_out-T_c_out
DELTAT_lm=( DELTAT_1- DELTAT_2)/ln(DELTAT_1/ DELTAT_2)
Q_dot=U*A_s* DELTAT_lm
A_s=pi*D*L
```

Step 2 of 5

Now to solve the equation for a range of geothermal water inlet temperature and mass flow rates, use parametric table by clicking on the "New parametric table" of 'table' menu.

A dialog box will open
Select number of runs=11
Select variables equations as 'L', 'm_dot_h' and 'T_h_in' and press 'Add' and then press 'OK'.
The screenshot of the "New parametric table"

First the inlet temperature of geothermal water is varied at constant flow rate of $\dot{m}_h = 0.3$ kg/s
To Enter the values in the "T_h_in" column click on the drop tab in "T_h_in" column.
New window pops up opens, fill in the details as shown and then press "OK"

Step 3 of 5

Now click on the 'solve table' of 'Calculate' menu

The solution will appear as shown below:

Parametric Table

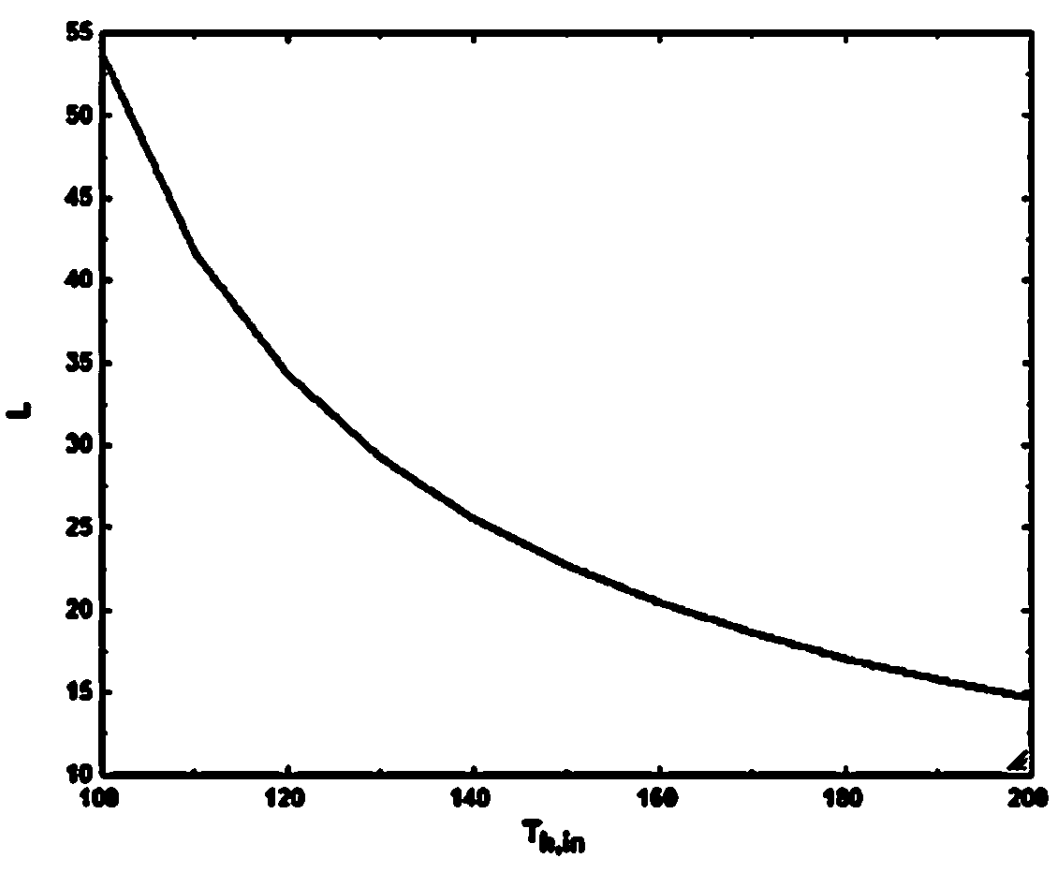
Table 2

1..11	L	$\dot{m}_h$	$T_{h,in}$
Run 1	53.73	0.3	100
Run 2	41.62	0.3	110
Run 3	34.27	0.3	120
Run 4	29.24	0.3	130
Run 5	25.54	0.3	140
Run 6	22.7	0.3	150
Run 7	20.45	0.3	160
Run 8	18.61	0.3	170
Run 9	17.08	0.3	180
Run 10	15.78	0.3	190
Run 11	14.67	0.3	200

Step 4 of 5

In order to plot length of the tube as function of inlet temperature of geothermal water, click on the 'X-Y' plot of "New Plot window" from plot menu.

Screen Shot of the plot showing the length of the tube as function of the inlet temperature of the geothermal water.



From the above plot we can interpert that as the inlet temperture of the geothermal water increases, the length of the tube decreases.

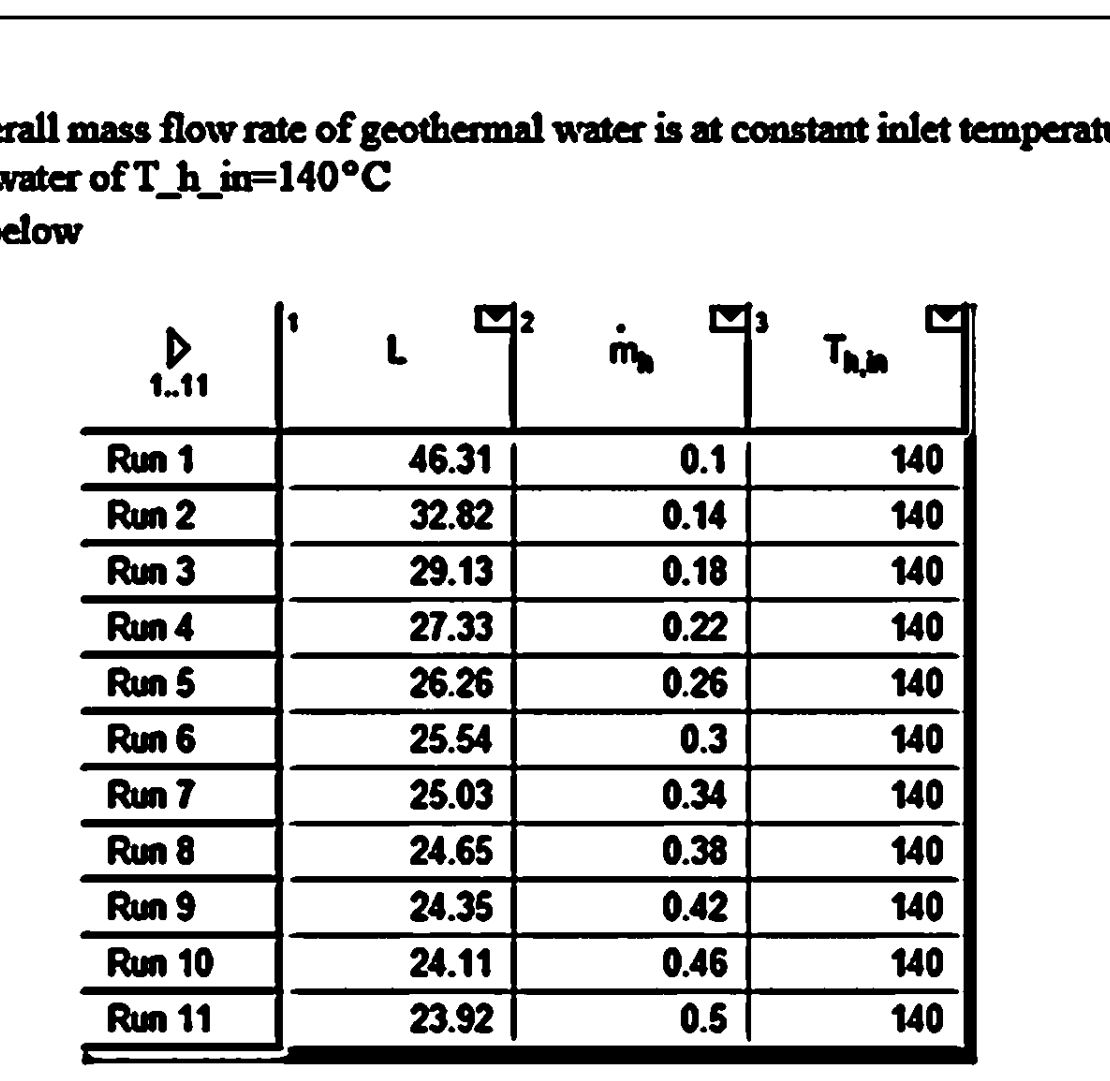
Step 5 of 5

Now the overall mass flow rate of geothermal water is at constant inlet temperature of the geothermal water of T_h_in=140°C
Proceed as below

1..11	L	$\dot{m}_h$	$T_{h,in}$
Run 1	46.31	0.1	140
Run 2	32.82	0.14	140
Run 3	29.13	0.18	140
Run 4	27.33	0.22	140
Run 5	26.26	0.26	140
Run 6	25.54	0.3	140
Run 7	25.03	0.34	140
Run 8	24.65	0.38	140
Run 9	24.35	0.42	140
Run 10	24.11	0.46	140
Run 11	23.92	0.5	140

In order to plot length of the tube as function of inlet temperature of geothermal water, click on the 'X-Y' plot of "New Plot window" from plot menu

Screen Shot of the plot showing the length of the tube as function of the mass flow rate of the geothermal water.



Question: 22.044P

A 1-shell-pass and 8-tube-passes heat exchanger is used to heat glycerin (*cp* = 0.60 Btu/lbm·°F) from 80°F to 140°F by hot water (*cp* = 1.0 Btu/lbm·°F) that enters the thin- walled 0.5-in-diameter tubes at 175°F and leaves at 120°F. The total length of the tubes in the heat exchanger is 400 ft. The convection heat transfer coefficient is 4 Btu/h·ft2·°F on the glycerin (shell) side and 50 Btu/h·ft2·°F on the water (tube) side. Determine the rate of heat transfer in the heat exchanger (*a*) before any fouling occurs and (*b*) after fouling with a fouling factor of 0.002 h·ft2·°F/Btu on the outer surfaces of the tubes.

Step 1 of 7

Number of tubes $N = 8$

Specific heat of glycerin, $c_p = 0.60\,\text{Btu/lbm}\cdot{}^\circ\text{F}$

Inlet temperature of the glycerin, $T_{in,g} = 80^\circ\text{F}$

Exit temperature of the glycerin, $T_{out,g} = 140^\circ\text{F}$

Specific heat of the hot water, $c_{pw} = 1.0\,\text{Btu/lbm}\cdot{}^\circ\text{F}$

Inlet temperature of the hot water, $T_{in,w} = 175^\circ\text{F}$

Outlet temperature of the hot water, $T_{out,w} = 120^\circ\text{F}$

Diameter of the tube $D = 0.5\,\text{in.}$

Length of the tube $L = 400\,\text{ft}$

Heat transfer coefficient for glycerin, $h_g = 4\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$

Heat transfer coefficient for water, $h_w = 50\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$

Fouling factor, $R_0 = 0.002\,\text{h}\cdot\text{ft}^2\cdot\text{F/Btu}$

Step 2 of 7

Overall heat transfer coefficient is given as

$$\frac{1}{U} = \frac{1}{h_g} + \frac{1}{h_w}$$

$$= \frac{1}{4} + \frac{1}{50}$$

$$= 0.27$$

$$U = 3.703\,\text{Btu/ft}^2\cdot{}^\circ\text{F}$$

Step 3 of 7

Surface area of the heat exchanger is

$$A_s = N\pi DL$$

$$= 8\pi\left(0.5\,\text{in.}\times\frac{1\,\text{ft}}{12\,\text{in.}}\right)400$$

$$= 418.879\,ft^2$$

Step 4 of 7

Log mean temperature difference of the counter flow heat exchanger is

$$\Delta t_{lm} = \frac{\left(T_{out,w} - T_{in,g}\right) - \left(T_{in,w} - T_{out,g}\right)}{\ln\left(\frac{T_{out,w} - T_{in,g}}{T_{in,w} - T_{out,g}}\right)}$$

$$= \frac{(120-80)-(175-140)}{\ln\left(\frac{120-80}{175-140}\right)}$$

$$= \frac{5}{0.133}$$

$$= 37.593^\circ\text{F}$$

Step 5 of 7

a)

Rate of heat transfer without fouling factor is

$$\dot{Q} = UA_s\Delta t_{lm}$$

$$= 3.703\times 418.879\times 37.593$$

$$= 58310.838\,\text{Btu}$$

Therefore, Rate of heat transfer without fouling factor is $\boxed{\dot{Q} = 58310.838\,\text{Btu}}$.

Step 6 of 7

b)

Overall heat transfer coefficient with fouling factor is given as

$$\frac{1}{U} = \frac{1}{h_g} + \frac{1}{h_w} + R_{f0}$$

$$= \frac{1}{4} + \frac{1}{50} + 0.002$$

$$= 0.27 + 0.002$$

$$= 0.272$$

$$U = 3.676\,\text{Btu/ft}^2\cdot{}^\circ\text{F}$$

Step 7 of 7

Rate of heat transfer without fouling factor is

$$\dot{Q} = UA_s\Delta t_{lm}$$

$$= 3.676\times 418.879\times 37.593$$

$$= 57893.081\,\text{Btu}$$

Therefore, Rate of heat transfer without fouling factor is $\boxed{\dot{Q} = 57893.081\,\text{Btu}}$.

Question: 22.045P

A double-pipe parallel-flow heat exchanger is used to heat cold tap water with hot water. Hot water (*cp* = 4.25 kJ/kg·K) enters the tube at 85°C at a rate of 1.4 kg/s and leaves at 50°C. The heat exchanger is not well insulated, and it is estimated that 3 percent of the heat given up by the hot fluid is lost from the heat exchanger. If the overall heat transfer coefficient and the surface area of the heat exchanger are 1150 W/m2·K and 4 m2, respectively, determine the rate of heat transfer to the cold water and the log mean temperature difference for this heat exchanger.

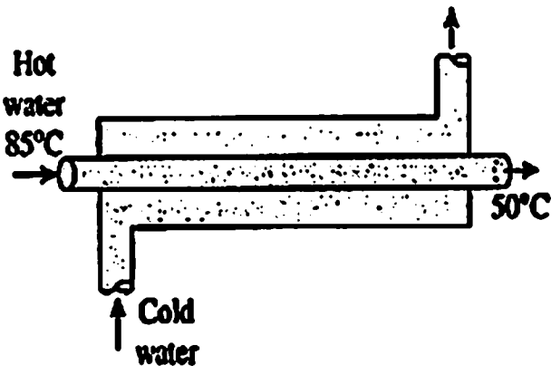


FIGURE P22–45

Step 1 of 2

The rate of heat transfer from hot water,

$$\dot{Q}_{hot_water} = [\dot{m}C_p(T_{out} - T_{in})]_{hot_water}$$

Here $\dot{m}$ is the Flow rate of hot water, c_p is The specific heat of hot water at average temperature, T_{out} is the Exit temperature of hot water and T_{in} is the Inlet temperature of hot water

Substitute 1.4 kg/s for $\dot{m}$, 4.25 kJ/kg·K for c_p, 50°C for T_{out} for and 85°C for T_{in}.

$$\begin{aligned}\dot{Q}_{hot_water} &= (1.4\ \text{kg/s})\times(4.25\ \text{kJ/kg}\cdot°\text{C})\times(85\text{-}50)°\text{C}\\ &= 208.3\ \text{kW}\end{aligned}$$

The rate of heat transfer in this radiator from the hot water to the cold water is determined by energy balance.

Since heat exchanger is not well insulated there is 3 percent of the heat given up by the hot fluid is lost from the heat exchanger then the rate of heat gain by the cold water is

$$\begin{aligned}\dot{Q}_{cold_water} &= (1-0.03)\dot{Q}_{hot_water}\\ &= (1-0.03)\times 208.3\\ &= 202.051\text{kW}\end{aligned}$$

Therefore, the rate of heat gain by the cold water is $\boxed{\dot{Q}_{cold_water} = 202.051\ \text{kW}}$.

Step 2 of 2

The logarithmic mean temperature difference for this heat exchanger,

$$\dot{Q}_{cold_water} = UA_i\Delta T_{lm}$$

Here U is the overall heat transfer coefficient, and A_i is the Area of the heat exchanger

Substitute 1150 W/m^2·K for U, and 4.25 kJ/kg·K for c_p.

$$\begin{aligned}\Delta T_{lm} &= \frac{202.051\times 10^3}{(1150)\times(4)}\\ &= 43.924°\text{C}\end{aligned}$$

Therefore, the logarithmic mean temperature difference for this heat exchanger is $\boxed{43.924°\text{C}}$.

Question: 22.046P

A thin-walled double-pipe counter-flow heat exchanger is to be used to cool oil (*cp* = 2200 J/kg·K) from 150°C to 50°C at a rate of 2.5 kg/s by water (*cp* = 4180 J/kg·K) that enters at 22°C at a rate of 1.5 kg/s. The diameter of the tube is 2.5 cm, and its length is 6 m. Determine the overall heat transfer coefficient of this heat exchanger.

Step 1 of 4

Specific heat of oil, $c_{po} = 2200\,\text{J/kg-K}$

Inlet temperature of the oil, $T_{io} = 150°\text{C}$

Exit temperature of the oil, $T_{eo} = 50°\text{C}$

Mass flow rate of oil, $m_o = 2.5\,\text{kg/s}$

Specific heat of water, $c_{pw} = 4180\,\text{J/kg-K}$

Inlet temperature of the water, $T_{wi} = 22°\text{C}$

Mass flow rate of water, $m_w = 1.5\,\text{kg/s}$

Diameter of the tube, $D = 2.5\,\text{cm} = 0.025\,\text{m}$

Length of the tube, $L = 6\,\text{m}$

Step 2 of 4

Heat transfer area of the counter flow heat exchanger is

$$A_s = \pi DL$$
$$= \pi(0.025)(6)$$
$$= 0.471\,\text{m}^2$$

Rate of heat transfer of oil is

$$Q_o = m_o c_{po}(T_{io} - T_{eo})$$
$$= 2.5 \times 2200(150 - 50)$$
$$= 550000\,\text{W}$$

From the energy balance equation

Rate of heat transfer loss by oil is equal to rate of heat gain by the water.

$$\dot{Q} = Q_0 = 550000\,\text{W} = m_w c_{pw}(T_{w0} - T_{wi})$$
$$550000 = 1.5 \times 4180(T_{wo} - 22)$$
$$T_{wo} = 109.719°\text{C}$$

Step 3 of 4

Log mean temperature difference for counter flow heat exchanger is

$$\Delta T_{lm} = \frac{(T_{io} - T_{wo}) - (T_{eo} - T_{wi})}{\ln\left(\frac{(T_{io} - T_{wo})}{(T_{eo} - T_{wi})}\right)}$$
$$= \frac{(150 - 109.719) - (50 - 22)}{\ln\left(\frac{150 - 109.719}{50 - 22}\right)}$$
$$= \frac{12.281}{0.3636}$$
$$= 33.769°\text{C}$$

Step 4 of 4

Then, overall heat transfer coefficient is

$$\dot{Q} = UA_s(\Delta T_{lm})$$
$$550000 = U \times 0.471 \times 33.769$$
$$U = 34579.88\,\text{W/m}^2\cdot\text{K}$$

Therefore, overall heat transfer coefficient is $\boxed{U = 34579.88\,\text{W/m}^2\cdot\text{K}}$

Question: 22.047P

Reconsider Prob. 22–46. Using EES (or other) software, investigate the effects of oil exit temperature and water inlet temperature on the overall heat transfer coefficient of the heat exchanger. Let the oil exit temperature vary from 30°C to 70°C and the water inlet temperature from 5°C to 25°C. Plot the overall heat transfer coefficient as functions of the two temperatures, and discuss the results.

Step 1 of 3

The given problem can be solved using EES software by following the below steps.

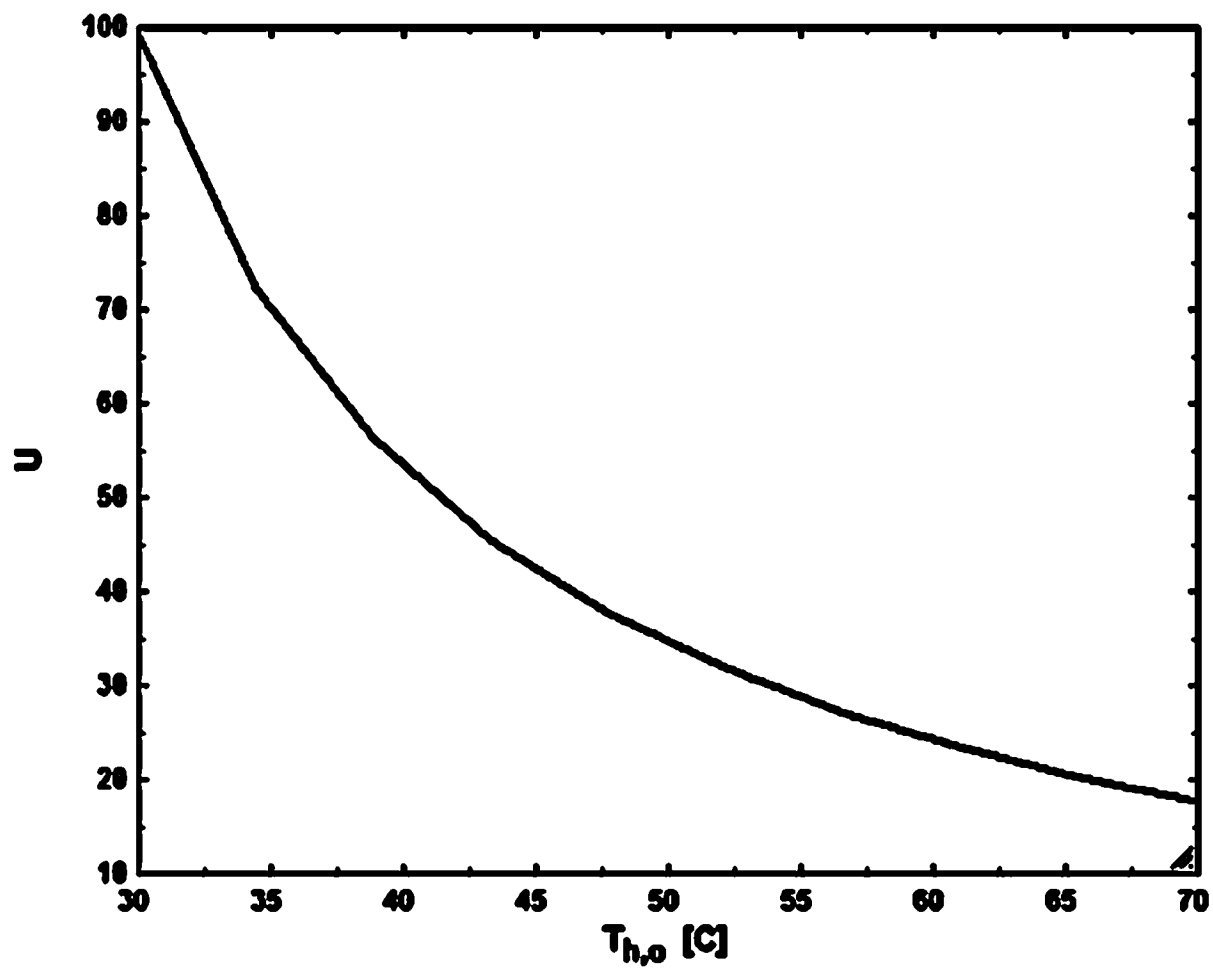
1. **Open EES software**
2. **Crease a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in dailog box.**

```
"GIVEN"
T_h_i=150[C]
T_h_o=50[C]
m_dot_h=2.5[kg/s]
c_p_h=2.2[kJ/kg-K]
T_c_i=22[C]
m_dot_c=1.5[kg/s]
c_p_c=4.18[kJ/kg-K]
D=0.025[m]
L=6[m]
"Determination for overall heat transfer coefficient"
Q_dot= m_dot_h* c_p_h*(T_h_i- T_h_o)
Q_dot = m_dot_c* c_p_c*(T_c_o- T_c_i)
DeltaT_1=T_h_i-T_c_o
DeltaT_2=T_h_o-T_c_i
DeltaT_lm=(DeltaT_1-DeltaT_2)/ln(DeltaT_1/DeltaT_2)
A=pi*D*L
Q_dot=U*A*DeltaT_lm
```

Step 2 of 3

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as $T_{h,o}$ and U press 'Add' and then press 'ok'

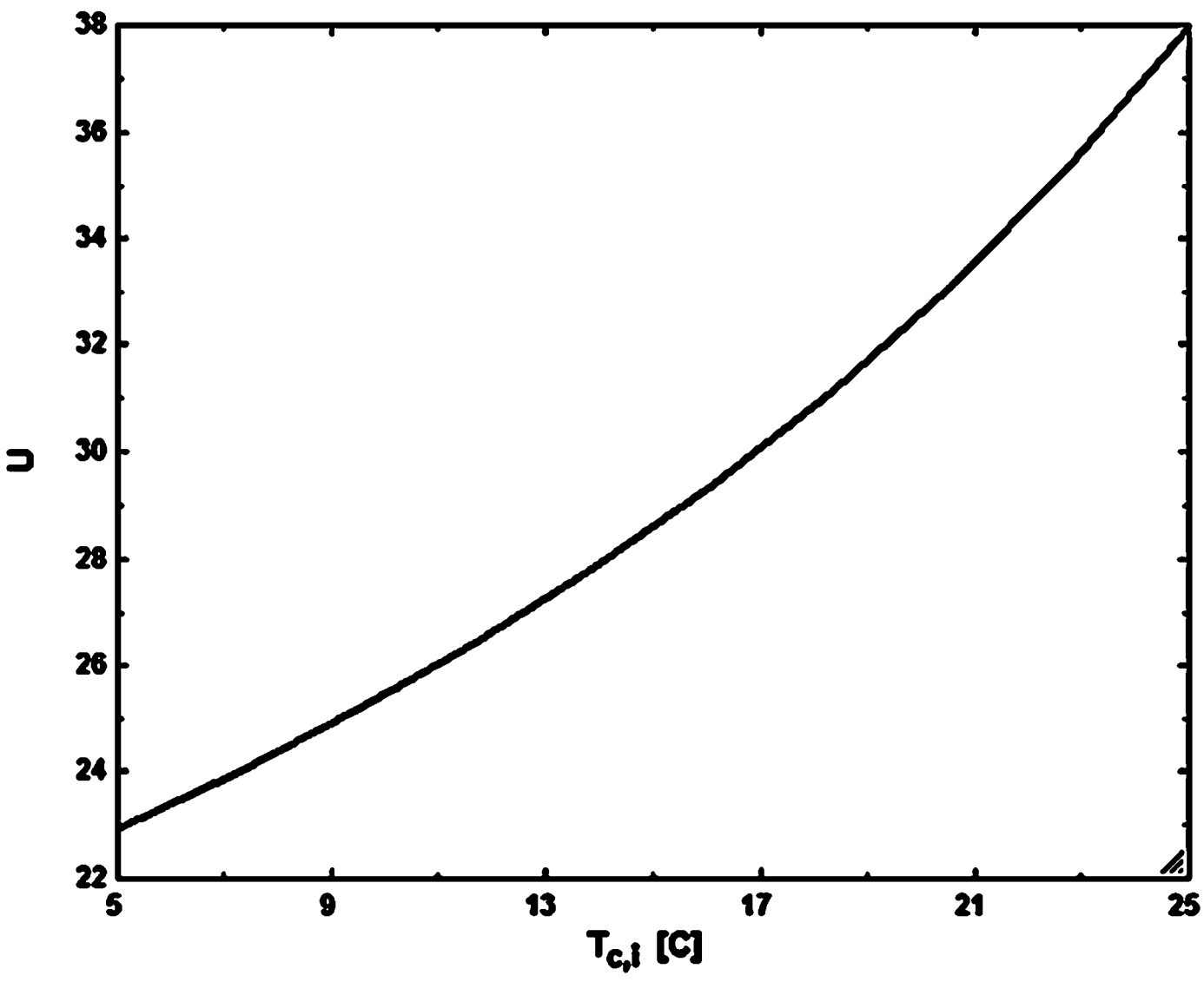
On executing the program by giving the range of the oil exit temperature, the variation in the overall heat transfer coefficient is known.



Step 3 of 3

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as $T_{c,i}$ and U press 'Add' and then press 'ok'

For the given range of water inlet temperature from 5°C to 25°C the variation in the overall heat transfer coefficient is plotted on an XY graph.



Question: 22.048P

A shell-and-tube heat exchanger with 2-shell passes and 12-tube passes is used to heat water (cp = 4180 J/kg·K) with ethylene glycol (cp = 2680 J/kg·K). Water enters the tubes at 22°C at a rate of 0.8 kg/s and leaves at 70°C. Ethylene glycol enters the shell at 110°C and leaves at 60°C. If the overall heat transfer coefficient based on the tube side is 280 W/m2·K, determine the rate of heat transfer and the heat transfer surface area on the tube side.

Step 1 of 5

Find the heat transfer rate by using the following equation:

$$\dot{Q} = \dot{m}_c c_p \left(T_{c,out} - T_{c,in}\right)$$

Here, $\dot{m}_c$ is mass flow rate of cold water, c_p is specific heat of water, $T_{c,out}$ is exit temperature of cold water, and $T_{c,in}$ is inlet temperature of cold water.

Substitute 0.8 kg/s for $\dot{m}_c$, $4180\,\text{J/kg.K}$ for c_p, $22°\text{C}$ for $T_{c,in}$, and $70°\text{C}$ for $T_{c,out}$.

$$\begin{aligned}\dot{Q} &= 0.8\times 4180(70-22)\\ &= 160512\,\text{W}\\ &= 160.512\,\text{kW}\end{aligned}$$

Therefore, heat transfer rate is $\boxed{160.512\,\text{W}}$

Step 2 of 5

Find the heat transfer surface area by using the following equation:

$$\dot{Q} = FUA_s\Delta T_{lm} \quad \text{..... (1)}$$

Here, F is correction factor, U is overall heat transfer coefficient, and ΔT_{lm} is log mean temperature difference.

Find the correction factor as follows:

Find the parameter P and R.

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

Substitute $22°\text{C}$ for $T_{c,in}$, $70°\text{C}$ for $T_{c,out}$, and $110°\text{C}$ for $T_{h,in}$

$$\begin{aligned}P &= \frac{70-22}{110-22}\\ &= 0.5454\end{aligned}$$

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$

Substitute $22°\text{C}$ for $T_{c,in}$, $70°\text{C}$ for $T_{c,out}$, $110°\text{C}$ for $T_{h,in}$, and $60°\text{C}$ for $T_{h,out}$

$$\begin{aligned}R &= \frac{110-60}{70-22}\\ &= 1.0416\end{aligned}$$

Obtain correction factor at P=0.5454 and R=1.0416 from the figure 11-9, "correction factor charts" in the text book.

$F = 0.94$

Step 3 of 5

Find log mean temperature difference by using the following equation:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

Find temperature differences $\Delta T_1, \Delta T_2$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

Substitute $70°\text{C}$ for $T_{c,out}$, and $110°\text{C}$ for $T_{h,in}$

$$\begin{aligned}\Delta T_1 &= (110-70)\\ &= 40°\text{C}\end{aligned}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

Substitute $22°\text{C}$ for $T_{c,in}$ and $60°\text{C}$ for $T_{h,out}$

$$\begin{aligned}\Delta T_2 &= (60-22)\\ &= 38°\text{C}\end{aligned}$$

Step 4 of 5

Substitute $40°\text{C}$ for ΔT_1 and $38°\text{C}$ for ΔT_2 in log mean temperature difference.

$$\begin{aligned}\Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}\\ &= \frac{40-38}{\ln\left(\frac{40}{38}\right)}\\ &= 38.99°\text{C}\end{aligned}$$

Step 5 of 5

Substitute 0.94 for F, $280\,\text{W/m}^2\text{.K}$ for U, $38.99°\text{C}$ for ΔT_{lm}, and 160512 W for $\dot{Q}$ in equation (1).

$$\begin{aligned}&\dot{Q} = FUA_s\Delta T_{lm}\\ &160512 = 0.94\times 280\times A_s\times 38.99\\ &A_s = 15.64\,\text{m}^2\end{aligned}$$

Therefore, the total surface area on the tube side is $\boxed{15.64\ \text{m}^2}$.

Question: 22.049P

Reconsider Prob. 22–48. Using EES (or other) software, investigate the effect of the mass flow rate of water on the rate of heat transfer and the tube-side surface area. Let the mass flow rate vary from 0.4 kg/s to 2.2 kg/s. Plot the rate of heat transfer and the surface area as a function of the mass flow rate, and discuss the results.

Step 1 of 3

The given problem can be solved using EES software by following the below steps.

1. **Open EES software**
2. **Crease a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in dailog box**

```
"GIVEN"
T_w_in=22 "[C]"
T_w_out=70 "[C]"
"m_dot_w=0.8 [kg/s], parameter to be varied"
C_p_w=4.18 "[kJ/kg-C]"
T_glycol_in=110 "[C]"
T_glycol_out=60 "[C]"
C_p_glycol=2.68 "[kJ/kg-C]"
U=0.28 "[kW/m^2-C]"

"ANALYSIS"
Q_dot=m_dot_w*C_p_w*(T_w_out-T_w_in)
Q_dot=m_dot_glycol*C_p_glycol*(T_glycol_in-T_glycol_out)
DELTAT_1=T_glycol_in-T_w_out
DELTAT_2=T_glycol_out-T_w_in
DELTAT_lm_CF=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
P=(T_glycol_out-T_glycol_in)/(T_w_in-T_glycol_in)
R=(T_w_in-T_w_out)/(T_glycol_out-T_glycol_in)
F=0.94 "from Fig. 13-18b of the text at the calculated P and R"
Q_dot=U*A*F*DELTAT_lm_CF
```

Step 2 of 3

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as m_w, $\dot{Q}$,and A press 'Add' and then press 'ok'
A table will appear and enter the values of length under 'm_w ' varing fro 0.4 kg/s to 2.2 kg/s
Then from click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

Parametric Table

Table 1

1..10	$\dot{m}_w$	$\dot{Q}$	A
Run 1	0.4	80.26	7.82
Run 2	0.6	120.4	11.73
Run 3	0.8	160.5	15.64
Run 4	1	200.6	19.55
Run 5	1.2	240.8	23.46
Run 6	1.4	280.9	27.37
Run 7	1.6	321	31.28
Run 8	1.8	361.2	35.19
Run 9	2	401.3	39.1
Run 10	2.2	441.4	43.01

Step 3 of 3

For graph between mass flow rate and heat transfer

Select the plot on menu bar, and then click X-Y plot .Dialog box will be appeared.

Select X-axis as m_w and Y-axis as $\dot{Q}$ .Press OK

A graph indicated in red color will be plotted as shown below.

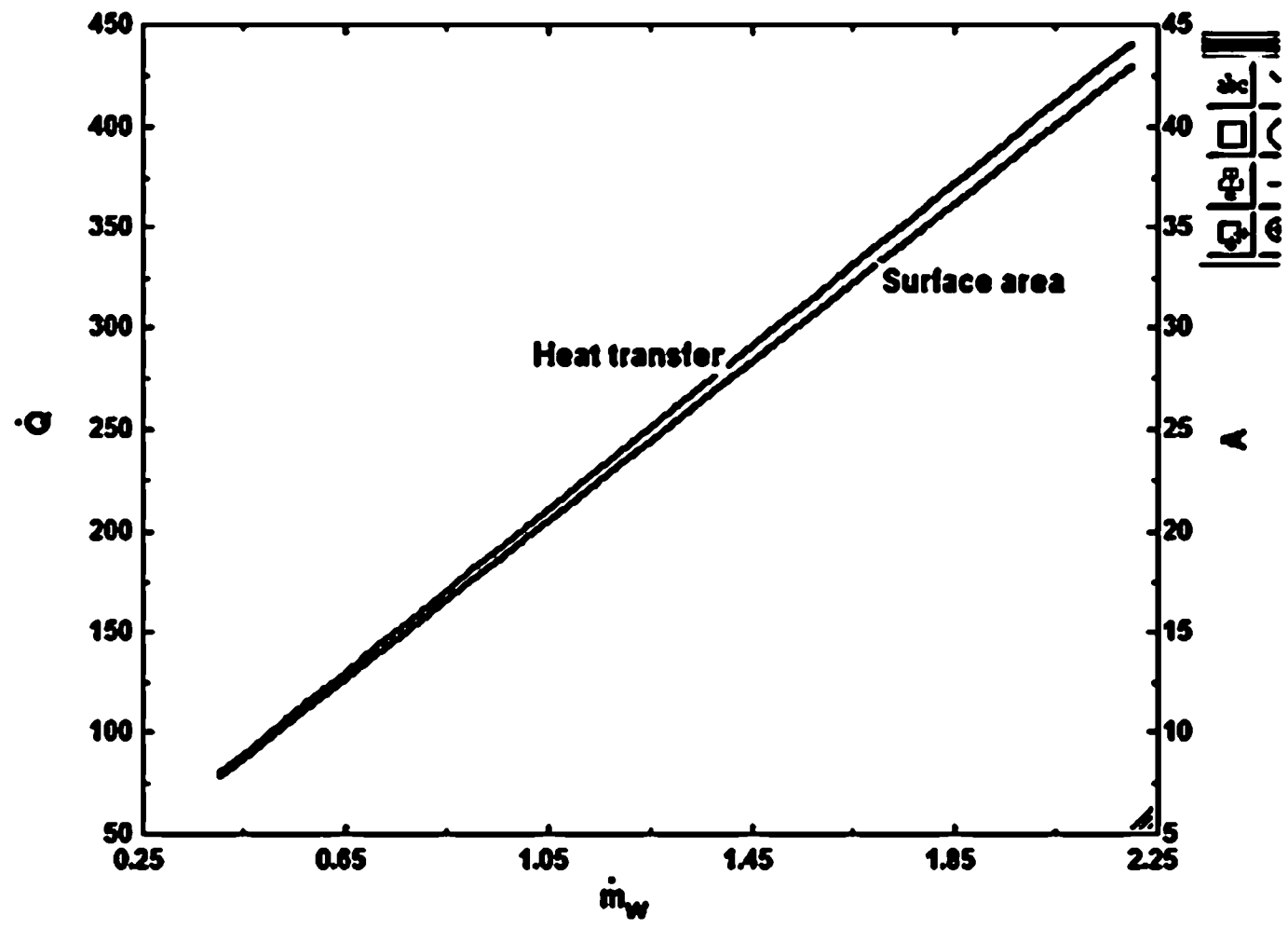
For graph between mass flow rate and surface area

Select the plot on menu bar, and then click overlay plot .Dialog box will be appeared

Select X-axis as m_w and Y-axis as A .Press OK

A graph indicated in blue color will be plotted as shown below.

Plot the graph between mass flow rate and heat transfer, surface area:



Question: 22.050P

Glycerin (*cp* = 2400 J/kg·K) at 20°C and 0.5 kg/s is to be heated by ethylene glycol (*cp* = 2500 J/kg·K) at 70°C in a thin-walled double-pipe parallel-flow heat exchanger. The temperature difference between the two fluids is 15°C at the outlet of the heat exchanger. If the overall heat transfer coefficient is 240 W/m2·K and the heat transfer surface area is 3.2 m2, determine (*a*) the rate of heat transfer, (*b*) the outlet temperature of the glycerin, and (*c*) the mass flow rate of the ethylene glycol.

Step 1 of 5

Given

The specific heat of Glycerin is $c_p = 2400\ \text{J/kg}\cdot\text{K}$

The specific heat of Ethylene glycol is $c_p = 2500\ \text{J/kg}\cdot\text{K}$

Inlet temperature of glycerin is $T_{c,in} = 20°\text{C}$

Inlet temperature of ethylene glycol is $T_{h,in} = 70°\text{C}$

Overall heat transfer coefficient $U = 240\ \text{W/m}^2\cdot\text{K}$

Heat transfer surface area is $A = 3.2\ \text{m}^2$

Mass flow rate of Glycerin $\dot{m}_{Glycerin} = 0.5\ \text{kg/s}$

The temperature difference between the two fluids at the outlet of the heat exchanger is $\Delta T_2 = 15°\text{C}$

Step 2 of 5

LMTD of the parallel flow heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,in}$$
$$= 70 - 20$$
$$\Delta T_1 = 50°\text{C}$$

Then

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \Delta T_1 / \Delta T_2}$$
$$\Delta T_{lm} = \frac{50 - 15}{\ln 50/15}$$
$$\Delta T_{lm} = 29.07°\text{C}$$

Step 3 of 5

(a) The rate of heat transfer can be calculated as

$$\dot{Q} = UA_s\Delta T_{lm}$$
$$\dot{Q} = (240\ \text{W/m}^2\cdot°\text{C})(3.2\ \text{m}^2)(29.07°\text{C})$$
$$\boxed{\dot{Q} = 22325.76\ \text{W}}$$
$$\boxed{\dot{Q} = 22.32576\text{kW}}$$

Step 4 of 5

(b)The outlet temperature of the Glycerin is calculated by the relation

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{Glycerin}$$
$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}c_p}$$
$$= 20°\text{C} + \frac{22325.76\ \text{W}}{(0.5\ \text{kg/s})(2400\ \text{kJ/kg}\cdot°\text{C})}$$
$$\boxed{T_{c,out} = 38.6°\text{C}}$$

Step 5 of 5

(c)

The mass flow rate of ethylene glycol can be calculated as

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{Ethylene\ glycol}$$

Where exit temperature of ethylene glycol is

$$\Delta T_2 = T_{h_out} - T_{c_out} = 15$$
$$T_{h_out} = 15 + T_{c_out}$$
$$= 15 + 38.6$$
$$T_{h_out} = 53.6°\text{C}$$
$$\dot{m}_{Ethylene\ glycol} = \left(\frac{\dot{Q}}{c_p[T_{out} - T_{in}]}\right)_{Ethylene\ glycol}$$
$$\dot{m}_{Ethylene\ glycol} = \frac{22.32576}{(2.5)(70 - 53.6)}$$
$$\boxed{\dot{m}_{Ethylene\ glycol} = 0.5445\ \text{kg/s}}$$

Question: 22.051P

Air (cp = 1005 J/kg·K) is to be preheated by hot exhaust gases in a cross-flow heat exchanger before it enters the furnace. Air enters the heat exchanger at 95 kPa and 20°C at a rate of 0.8 m3/s. The combustion gases (cp = 1100 J/kg·K) enter at 180°C at a rate of 1.1 kg/s and leave at 95°C. The product of the overall heat transfer coefficient and the heat transfer surface area is UAS = 1200 W/K. Assuming both fluids to be unmixed, determine the rate of heat transfer and the outlet temperature of the air.

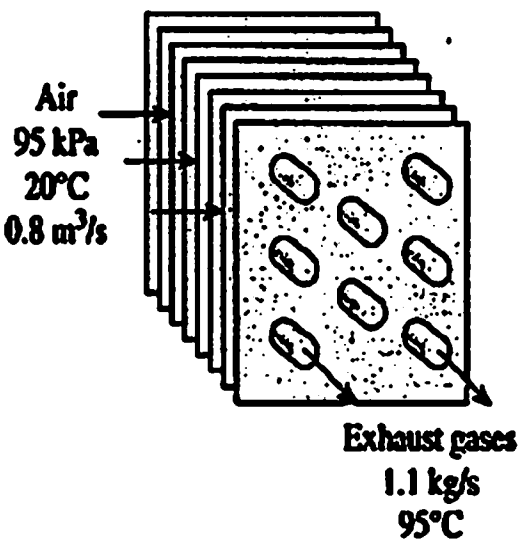


FIGURE P22–51

Step 1 of 3

Given

The specific heat of exhaust gases is $c_{p_gas} = 1100\ \text{J/kg}\cdot\text{K}$

The specific heat of air is $c_{p_air} = 1005\ \text{J/kg}\cdot\text{K}$

Mass flow rate of exhaust gases is $\dot{m}_g = 1.1\ \text{kg/s}$

The product of the overall heat transfer coefficient and the heat transfer surface area is $UA_s = 1200\ \text{W/°C}$

Air enters at pressure $P = 95\text{kPa}$

Temperature of air at inlet $T = 20°\text{C}$

Volumetric flow rate of air at inlet is $\dot{V}_1 = 0.8\,\text{m}^3/\text{s}$

Step 2 of 3

Mass flow rate of air can be calculated by using the ideal gas equation

$$\dot{m}_{air} = \frac{P_1 \dot{V}_1}{RT_1}$$

$$= \frac{(95\ \text{kPa})(0.8\ \text{m}^3/\text{s})}{(0.287\ \text{kPa.m}^3/\text{kg.K})(293\ \text{K})}$$

$$\dot{m}_{air} = 0.9038\ \text{kg/s}$$

The rate of heat transfer can be calculated by using the relation

$$\dot{Q} = [\dot{m}_h c_{ph}(T_{out} - T_{in})]_{\text{combustion gases}}$$

$$\dot{Q} = (1.1\ \text{kg/s})(1100\ \text{J/kg}\cdot\text{K})(180 - 95)°\text{C}$$

$$\dot{Q} = 102.85\ \text{kJ/s}$$

Step 3 of 3

Therefore the outlet temperature of the air is

$$\dot{Q} = \left[\dot{m}_c c_{pc}(T_{out} - T_{in})\right]_{air}$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}}$$

$$T_{out_air} = 20 + \frac{102.85}{0.9038 \times 1.005}$$

$$\boxed{T_{out_air} = 133.23°\text{C}}$$

Question: 22.052P

A shell-and-tube heat exchanger with 2-shell passes and 12-tube passes is used to heat water *(cp* = 4180 J/kg·K) in the tubes from 20°C to 70°C at a rate of 4.5 kg/s. Heat is supplied by hot oil *(cp* = 2300 J/kg·K) that enters the shell side at 170°C at a rate of 10 kg/s. For a tube-side overall heat transfer coefficient of 350 W/m2·K, determine the heat transfer surface area on the tube side.

Step 1 of 9

Given

Type: 2shell and 12 tube passes

Specific heats of water $c_{pc} = 4180$ J/kg.°C

Specific heats of oil $c_{ph} = 2300$ J/kg.°C

Mass flow rates of water $\dot{m}_c = 4.5$ kg/s

Mass flow rates of oil $\dot{m}_h = 10$ kg/s

$T_{c,in} = 20^0$C

$T_{c,out} = 70^0$C

$T_{h,in} = 170^0$C

Tube side overall heat transfer coefficient $U_{c,w} = 350$ W/m^2°C

Step 2 of 9

Heat transfer surface area on the tube side:

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

$\dot{Q}$, F &ΔT_{lm} are calculated by the following relations

Step 3 of 9

Heat transfer rate is $\dot{Q} = [\dot{m}_c c_{pc}(T_{c,out} - T_{c,in})]_{water}$

$$= (4.5 \text{ kg/s})(4180 \text{ J/kg.°C})(65\text{-}25)\text{°C}$$

$$\dot{Q} = 940.5 \text{ kW}$$

Step 4 of 9

Then the outlet temperature of the oil is

$$\dot{Q} = [\dot{m}_h c_{ph}(T_{h,out} - T_{h,in})]_{oil}$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{\dot{m}_h c_{ph}}$$

$$= 170\text{°C} - \frac{940500 \text{ W}}{(10 \text{ kg/s})(2300 \text{ J/kg.°C})}$$

$$T_{h,out} = 129.108\text{°C}$$

Step 5 of 9

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= (170 - 70)\text{°C}$$

$$\Delta T_1 = 100\text{°C}$$

Step 6 of 9

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= (129.108 - 20)\text{°C}$$

$$\Delta T_2 = 109.108\text{°C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{100 - 109.108}{\ln\left(\frac{100}{109.108}\right)}$$

$$\therefore \Delta T_{lm} = 104.4878\text{°C}$$

Step 7 of 9

And $$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

$$= \frac{129.108 - 170}{20 - 170}$$

$$P = 0.27$$

Step 8 of 9

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$

$$= \frac{20 - 70}{129.108 - 170}$$

$$R = 1.2$$

At these P and R values $F = 1$

(From correction factor F charts for common shell and tube and cross flow heat exchangers (b))

Step 9 of 9

By substituting all values heat transfer surface area becomes

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

$$A_s = \frac{940500}{(350)(1)(104.435)}$$

$$\therefore A_s = 25.73 \text{ m}^2$$

Question: 22.053P

Repeat Prob. 22–52 for a mass flow rate of 3 kg/s for water.

Step 1 of 6

Given

Type: 2shell and 12 tube passes

Specific heats of water $c_{pc} = 4180$ J/kg.°C

Specific heats of oil $c_{ph} = 2300$ J/kg.°C

Mass flow rates of water $\dot{m}_c = 3$ kg/s

Mass flow rates of oil $\dot{m}_h = 10$ kg/s

Inlet temperature of water in the tube $T_{c,in} = 20^0$C

Exit temperature of water in the tube $T_{c,out} = 70^0$C

Inlet temperature of hot oil in the shell side $T_{h,in} = 170^0$C

Tube side overall heat transfer coefficient $U_{c,w} = 350$ W/m$^2 \cdot$K

Step 2 of 6

Heat transfer surface area on the tube side:

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

Heat transfer rate is

$$\dot{Q} = [\dot{m}_c c_{pc}(T_{c,out} - T_{c,in})]_{water}$$

$$= (3 \text{ kg/s})(4180 \text{ J/kg} \cdot \text{K})(70 - 20)^\circ\text{C}$$

$$\dot{Q} = 627000 \text{ W}$$

Step 3 of 6

Then the outlet temperature of the oil is

$$\dot{Q} = [\dot{m}_h c_{ph}(T_{h,out} - T_{h,in})]_{oil}$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{\dot{m}_h c_{ph}}$$

$$= 170^\circ\text{C} - \frac{627000 \text{ W}}{(10 \text{ kg/s})(2300 \text{ J/kg} \cdot ^\circ\text{C})}$$

$$T_{h,out} = 142.739^\circ\text{C}$$

Step 4 of 6

The log mean temperature difference of the counter flow heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= 170 - 70$$

$$\Delta T_1 = 100^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= 142.739 - 20$$

$$\Delta T_2 = 122.739^\circ\text{C}$$

Then LMTD is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{100 - 122.739}{\ln\left(\frac{100}{122.739}\right)}$$

$$\Delta T_{lm} = 110.981^\circ\text{C}$$

Step 5 of 6

The correction factor for two shell and 12 tube pass can be determined as follows

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

$$= \frac{70 - 20}{170 - 20}$$

$$P = 0.333$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$

$$= \frac{170 - 142.739}{70 - 20}$$

$$R = 0.545$$

At these P and R values from correction factor F charts for two shell and 12tube passes and cross flow heat exchangers (b)

$$F = 1$$

Step 6 of 6

By substituting all values heat transfer surface area becomes

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

$$= \frac{627000}{(350)(1)(110.981)}$$

$$\boxed{A_s = 16.14 \text{ m}^2}$$

Therefore the heat transfer surface area on the tube side is $A_s = 16.14$ m^2

Question: 22.054P

A shell-and-tube heat exchanger with 2-shell passes and 8-tube passes is used to heat ethyl alcohol *(cp* = 2670 J/kg·K) in the tubes from 25°C to 70°C at a rate of 2.1 kg/s. The heating is to be done by water *(cp* = 4190 J/kg·K) that enters the shell side at 95°C and leaves at 45°C. If the overall heat transfer coefficient is 950 W/m2·K, determine the heat transfer surface area of the heat exchanger.

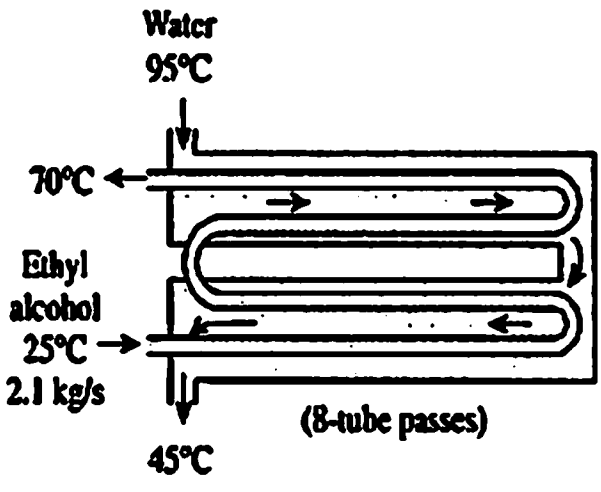


FIGURE P22–54

Step 1 of 6

Given data:

Specific heats of water $c_{ph} = 4190$ J/kg.°C

Specific heats of ethyl alcohol $c_{pc} = 2670$ J/kg.°C

Overall heat transfer coefficient $U = 950$ W/m^2.°C

Step 2 of 6

Heat transfer rate is

$$\dot{Q} = [\dot{m}_c c_{pc}(T_{out} - T_{in})]_{\text{Ethyl alcohol}}$$
$$= (2.1)(2670)(70-25)$$
$$= 252.315 \text{ kW}$$

Step 3 of 6

And

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$
$$= \frac{45-95}{25-95}$$
$$= 0.7$$
$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$
$$= \frac{25-70}{45-95}$$
$$= 0.9$$

Step 4 of 6

From the correction chart
At these P and R values $F = 0.77$
For this counter flow heat exchanger LMTD determined to be

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= (95-70)$$
$$= 25°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= (45-25)$$
$$= 20°\text{C}$$

Step 5 of 6

Log mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$
$$= \frac{25-20}{\ln\left(\frac{25}{20}\right)}$$
$$= 22.407°\text{C}$$

Step 6 of 6

Heat transfer surface area is determined to be

$$\dot{Q} = UA_s\Delta T_{lm}$$
$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$
$$A_s = \frac{252315}{(950)(0.77)(22.407)}$$
$$= 15.3937 \text{ m}^2$$

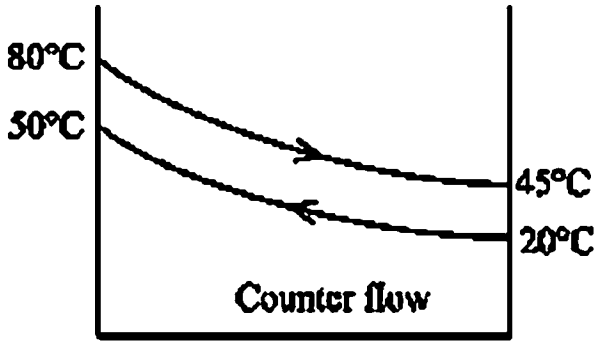
$$\therefore A_s = 15.3937 \text{ m}^2$$

Question: 22.055P

Consider a water-to-water double-pipe heat exchanger whose flow arrangement is not known. The temperature measurements indicate that the cold water enters at 20°C and leaves at 50°C, while the hot water enters at 80°C and leaves at 45°C.Do you think this is a parallel-flow or counter-flow heat exchanger? Explain.

Step 1 of 1

In parallel-flow heat exchangers, the temperature of the cold water can never exceed that of the hot fluid. In this case $T_{\text{cold out}} = 50°C$ which is greater than $T_{\text{hot out}} = 45°C$. Therefore this must be a counter-flow heat exchanger



Question: 22.056P

Cold water *(cp* = 4180 J/kg·K) leading to a shower enters a thin-walled double-pipe counter-flow heat exchanger at 15°C at a rate of 1.25 kg/s and is heated to 60°C by hot water *(cp* = 4190 J/kg·K) that enters at 100°C at a rate of 4 kg/s. If the overall heat transfer coefficient is 880 W/m2·K, determine the rate of heat transfer and the heat transfer surface area of the heat exchanger.

Step 1 of 7

Using NTU method heat transfer surface area is determined as follows

The heat capacity rate of cold fluid is,

$$C_c = \dot{m}_c C_{pc}$$

Here

Specific heat of cold water, $c_{pc} = 4180$ J/kg-K

Mass flow rate of cold water, $\dot{m}_c = 1.25$ kg/s

Substitute 4180 J/kg-K for c_{pc}, 1.25 kg/s for $\dot{m}_c$ in the above equation

$$\begin{aligned} C_c &= \dot{m}_c C_{pc} \\ &= (1.25 \text{ kg/s})(4180 \text{ J/kg-K}) \\ &= 5225 \text{ W/K} \end{aligned}$$

Step 2 of 7

The heat capacity rate of hot fluid is,

$$C_h = \dot{m}_h C_{ph}$$

Here

Specific heat of hot water, $c_{ph} = 4190$ J/kg-K

Mass flow rate of hot water, $\dot{m}_h = 4$ kg/s

Substitute 4190 J/kg-K for c_{ph}, 4 kg/s for $\dot{m}_h$ in the above equation

$$\begin{aligned} C_h &= \dot{m}_h C_{ph} \\ &= (4 \text{ kg/s})(4190 \text{ J/kg-K}) \\ &= 16760 \text{ W/K} \end{aligned}$$

Step 3 of 7

Therefore the minimum heat capacity is

$$C_{min} = C_c = 5225 \text{ W/K}$$

And

$$\begin{aligned} C &= \frac{C_{min}}{C_{max}} \\ &= \frac{5225}{16760} \\ &= 0.31 \end{aligned}$$

Step 4 of 7

The maximum heat transfer rate is,

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

Temperature at inlet of cold fluid, $T_{c,in}$

Temperature at inlet of hot fluid, $T_{h,in}$

Substitute 15°C for $T_{c,in}$, 100°C for $T_{h,in}$, 5225 W/K for C_{min} in the above equation

$$\begin{aligned} \dot{Q}_{max} &= C_{min}\left(T_{h,in} - T_{c,in}\right) \\ &= 5225 \text{ W/K}(100°\text{C} - 15°\text{C}) \\ &= 444125 \text{ W} \\ &= 444.125 \text{ kW} \end{aligned}$$

Step 5 of 7

Actual heat transfer is

$$\dot{Q} = C_c(T_{c,out} - T_{c,in})$$

Here

Temperature at outlet of cold fluid, $T_{h,in}$

Substitute 15°C for $T_{c,in}$, 60°C for $T_{h,in}$, 5225 W/K for C_c in the above equation

$$\begin{aligned} \dot{Q} &= C_c(T_{c,out} - T_{c,in}) \\ &= (5225 \text{ W/K})(60°\text{C} - 15°\text{C}) \\ &= 235125 \text{ W} \\ &= 235.125 \text{ kW} \end{aligned}$$

Therefore, the actual rate of heat transfer $\dot{Q}$ is $\boxed{235.125 \text{ kW}}$

Step 6 of 7

Then effectiveness of the heat exchanger is

$$\begin{aligned} \varepsilon &= \frac{\dot{Q}}{\dot{Q}_{max}} \\ &= \frac{235.125}{444.125} \\ &= 0.52 \end{aligned}$$

NTU of the heat exchanger is,

$$\begin{aligned} NTU &= \frac{1}{C-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C - 1}\right) \\ &= \frac{1}{0.31-1}\ln\left(\frac{0.52-1}{0.52\times 0.31-1}\right) \\ &= 0.808 \end{aligned}$$

Step 7 of 7

Heat transfer surface area is,

$$NTU = \frac{UA_s}{C_{min}}$$

Overall heat transfer coefficient, U

Substitute 950 W/m^2·K for U, 0.808 for NTU, 5225 W/K for C_{min} in the above equation.

$$\begin{aligned} NTU &= \frac{UA_s}{C_{min}} \\ A_s &= \frac{NTU(C_{min})}{U} \\ &= \frac{(0.808)(5225 \text{ W/K})}{880 \text{ W/m}^2\text{·K}} \\ &= 4.79 \text{ m}^2 \end{aligned}$$

Therefore, the heat transfer surface area A_s is $\boxed{4.79 \text{ m}^2}$

Question: 22.057P

Steam is to be condensed on the shell side of a 1 -shell- pass and 8-tube-passes condenser, with 60 tubes in each pass at 90°F *(hfg* = 1043 Btu/lbm). Cooling water *(cp* = 1.0 Btu/lbm·°F) enters the tubes at 55°F and leaves at 70°F. The tubes are thin walled and have a diameter of 3/4 in and length of 5 ft per pass. If the overall heat transfer coefficient is 600 Btu/h·ft2·°F, determine (*a*) the rate of heat transfer, *(b)* the rate of condensation of steam, and (*c*) the mass flow rate of cold water.

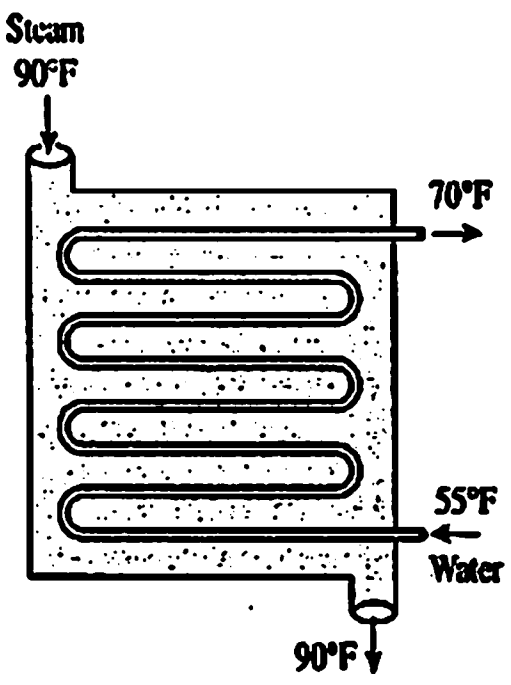


FIGURE P22–57E

Step 1 of 5

Given

The heat of vaporization of water at 90°F is $h_{fg} = 1043$ Btu/lbm

Specific heat of cold water is $c_p = 1$ Btu/lbm · °F

Overall heat transfer coefficient is $U = 600$ Btu/h.ft^2 · °F

The diameter of the tube

$$D = \frac{3}{4}\text{in}$$

$$D = 0.0625 \text{ ft}$$

Length of the tube is $L = 5$ ft

Cooling water enters the tubes at temperature $T_{c,in} = 55°F$

Cooling water exits the tubes at temperature $T_{c,out} = 70°F$

Condenser temperature $T_{h,in} = T_{h,out} = 90°F$

Step 2 of 5

Heat transfer surface area is

$$A_s = n\pi DL$$
$$= 8(50)(\pi)(0.0625 \text{ ft})(5 \text{ ft})$$
$$A_s = 471.238 \text{ ft}^2$$

The log mean temperature difference of the heat exchanger is calculated as

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$\Delta T_1 = (90 - 70)°F$$
$$\Delta T_1 = 20°F$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= (90 - 55)$$
$$\Delta T_2 = 35°F$$

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \Delta T_1 / \Delta T_2}$$
$$= \frac{20 - 35}{\ln 20/35}$$
$$\Delta T_{lm} = 26.8°F$$

Step 3 of 5

(a) The rate of heat transfer becomes

$$\dot{Q} = UA_s \Delta T_{lm}$$

$$\dot{Q} = (600 \text{ Btu/h.ft}^2\text{.°F})(471.238 \text{ ft}^2)(26.8°F)$$

$$\boxed{\dot{Q} = 7.577 \times 10^6 \text{ Btu/h}}$$

Step 4 of 5

b) The rate of condensation of the steam

$$\dot{Q} = (\dot{m} h_{fg})_{steam}$$

$$\dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}}$$
$$= \frac{7.577 \times 10^6 \text{ Btu/h}}{1043 \text{ Btu/lbm}}$$
$$= 7264.62 \text{lbm/h}$$

$$\boxed{\dot{m}_{steam} = 2.0179 \text{lbm/s}}$$

Step 5 of 5

c) The mass flow rate of the cold water is

$$\dot{Q} = [\dot{m} c_p (T_{out} - T_{in})]_{cold\ water}$$

$$\dot{m}_{cold\ water} = \frac{\dot{Q}}{c_p (T_{out} - T_{in})}$$
$$= \frac{7.577 \times 10^6 \text{ Btu/h}}{(1.0 \text{ Btu/lbm} \cdot °F)(70°F - 55°F)}$$
$$= 0.5051 \times 10^6 \text{ lbm/h}$$

$$\boxed{\dot{m}_{cold\ water} = 140.305 \text{ lbm/s}}$$

Question:

Reconsider Prob. 22–57E. Using EES (or other) software, investigate the effect of the condensing steam temperature on the rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water. Let the steam temperature vary from 80°F to 120°F. Plot the rate of heat transfer, the condensation rate of steam, and the mass flow rate of cold water as a function of steam temperature, and discuss the results.

Step 1 of 8

This given problem is solved in EES "Engineering Equation Solver" in the following method.
Open a new window in the EES software by ceasing " File > New " and type the program as shown below.

Equations Window

```
"Given data"
n=8*60
h_fg=1043 [Btu/lbm]
cp=1 [Btu/lbm-F]
D=(3/4) [in]*convert(in,ft)
L=5 [ft]
T_c_in=55 [F]
T_c_out=70 [F]
"T_h_in=90 [F]"
T_h_in=T_h_out
U=600 [Btu/h-ft^2]

A_s=n*pi*D*L                              "Surface area "
"Log mean temperature difference of the heat exchanger"
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
DELTAT_1=T_h_in-T_c_out
DELTAT_2=T_h_out-T_c_in

"The rate of heat transfer "
Q_dot=U*A_s*DELTAT_lm

"The rate of condensation of the steam"
Q_dot=m_dot_steam*h_fg

"Mass flow rate of the cold water "
Q_dot=m_dot_water*cp*(T_c_out-T_c_in)*convert(lb_m/hr,lb_m/s)
```

Step 2 of 8

To check weather the program typed is in correct mathematical format or not go to "Options > Function info" a window opens as show below.

Formatted Equations

Given data

$n = 8 \cdot 60$

$h_{fg} = 1043 \quad [Btu/lbm]$

$cp = 1 \quad [Btu/lbm\text{-}F]$

$D = 3 / 4 \cdot 1 \quad [in] \cdot \left| 0.083333 \cdot \frac{ft}{in} \right|$

$L = 5 \quad [ft]$

$T_{c,in} = 55 \quad [F]$

$T_{c,out} = 70 \quad [F]$

$T_{h,in} = 90 \quad [F]$

$T_{h,in} = T_{h,out}$

$U = 600 \quad [Btu/h\text{-}ft^2]$

$A_s = n \cdot \pi \cdot D \cdot L$ Surface area

Log mean temperature difference of the heat exchanger

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left[\frac{\Delta T_1}{\Delta T_2}\right]}$$

$\Delta T_1 = T_{h,in} - T_{c,out}$

$\Delta T_2 = T_{h,out} - T_{c,in}$

The rate of heat transfer

$\dot{Q} = U \cdot A_s \cdot \Delta T_{lm}$

The rate of condensation of the steam

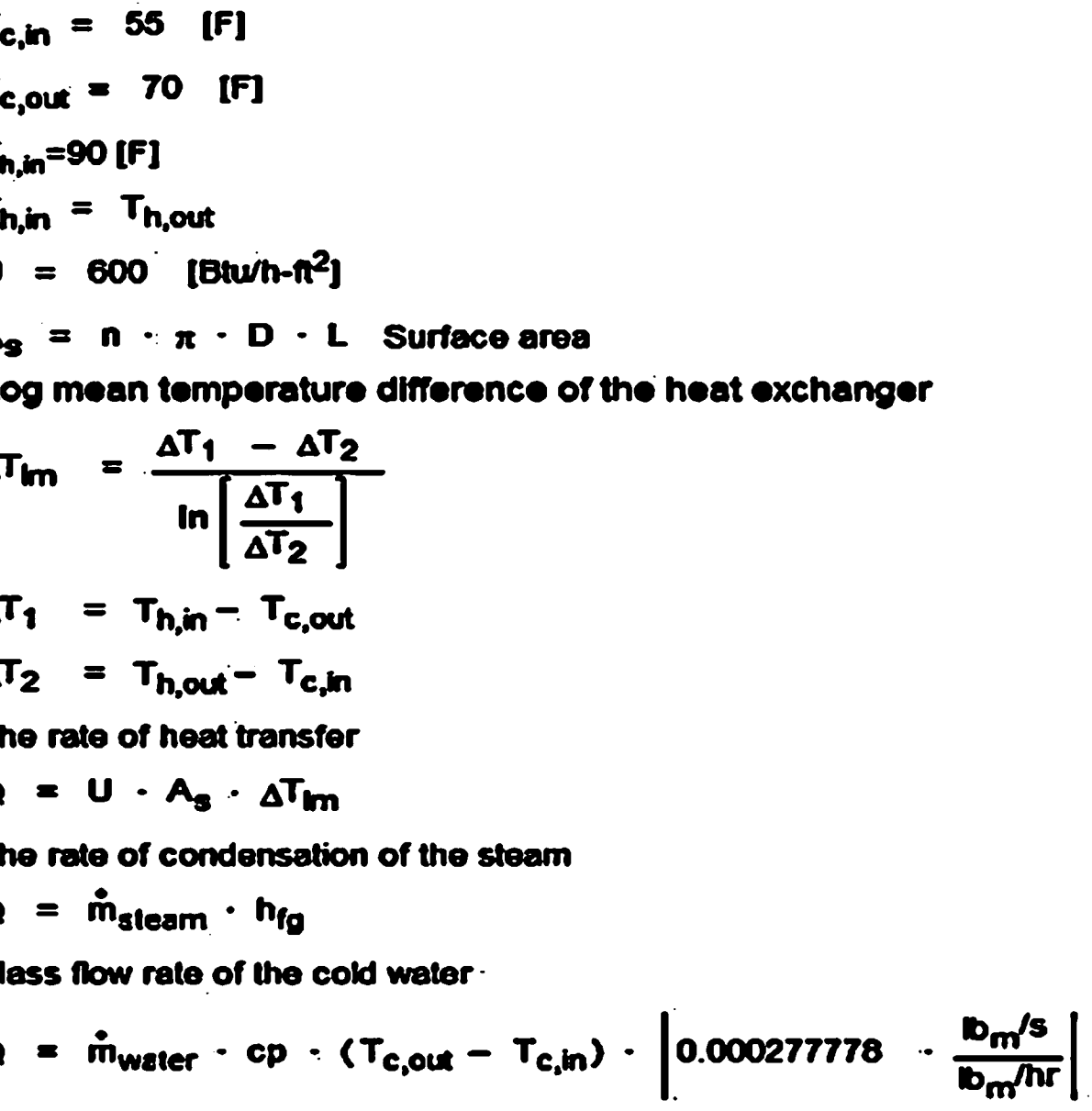
$\dot{Q} = \dot{m}_{steam} \cdot h_{fg}$

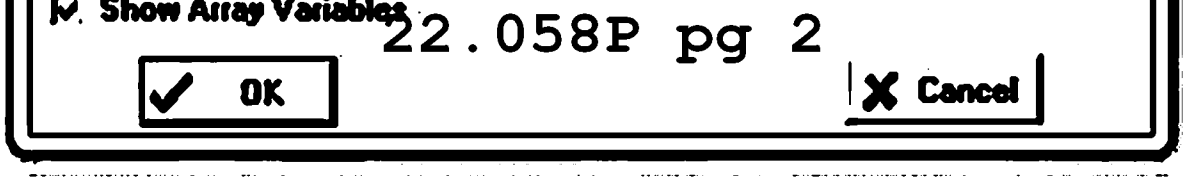
Mass flow rate of the cold water

$$\dot{Q} = \dot{m}_{water} \cdot cp \cdot (T_{c,out} - T_{c,in}) \cdot \left| 0.000277778 \cdot \frac{lb_m/s}{lb_m/hr} \right|$$

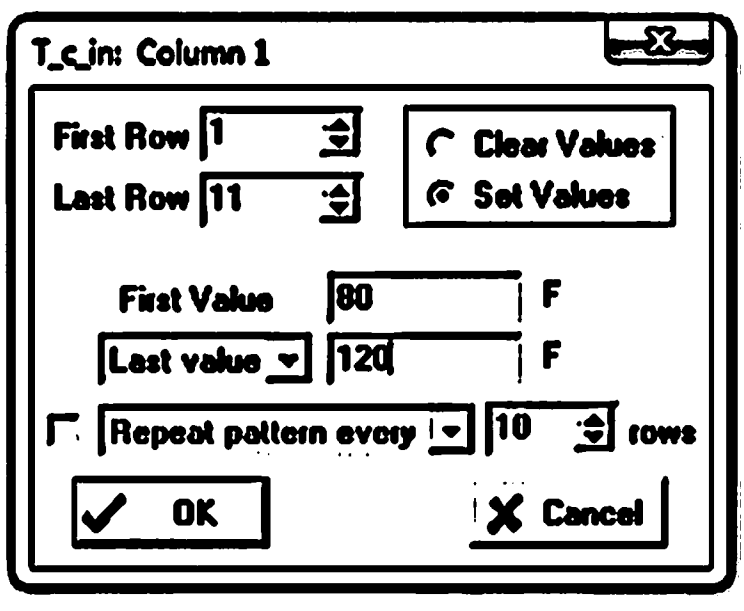
Step 3 of 8

Now to create a tabulated result go to "Tables > New Parametric Tables". Fill in the details as shown below,





To enter the values in the velocity column click on the drop tab in velocity column. A new windows opens fill in the details as shown below,



Step 4 of 8

Click on the run button which is symbolized as a green icon on the left side your results will be displayed as shown,

Parametric Table

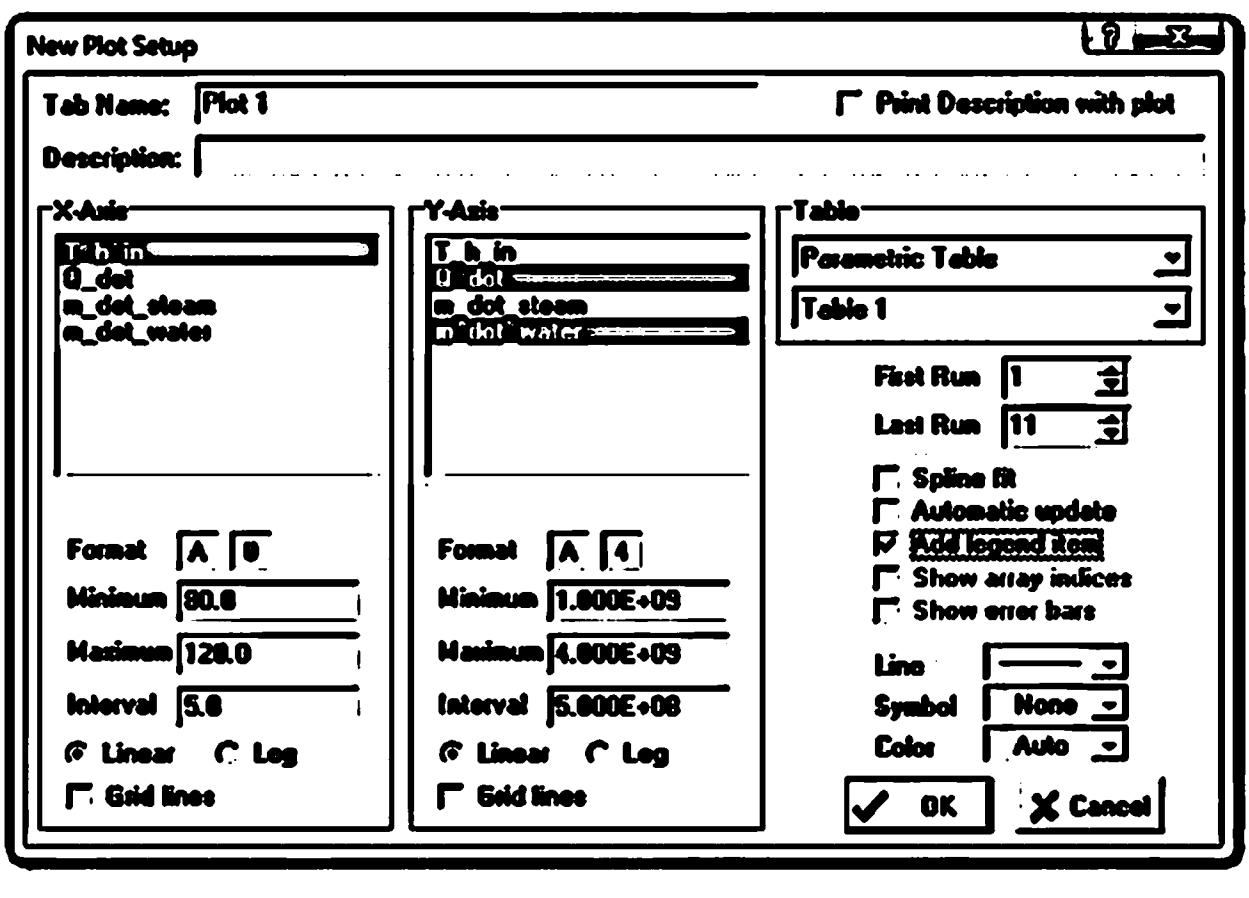
Table 1

	1 $T_{h,in}$ [F]	2 $\dot{Q}$ [Btu/hr]	3 $\dot{m}_{steam}$ [lbm/hr]	4 $\dot{m}_{water}$ [lbm/hr]
Run 1	80	4.629E+06	4438	1.111E+09
Run 2	84	5.824E+06	5584	1.398E+09
Run 3	88	6.997E+06	6709	1.679E+09
Run 4	92	8.158E+06	7822	1.958E+09
Run 5	96	9.311E+06	8928	2.235E+09
Run 6	100	1.046E+07	10029	2.510E+09
Run 7	104	1.160E+07	11126	2.785E+09
Run 8	108	1.274E+07	12222	3.059E+09
Run 9	112	1.388E+07	13315	3.333E+09
Run 10	116	1.502E+07	14408	3.606E+09
Run 11	120	1.617E+07	15499	3.880E+09

Step 5 of 8

Due to the difference in range of values of the variables we draw the plot twice. Fist for $T_{h,in}$ Vs $\dot{Q}, \dot{m}_{water}$.

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 6 of 8

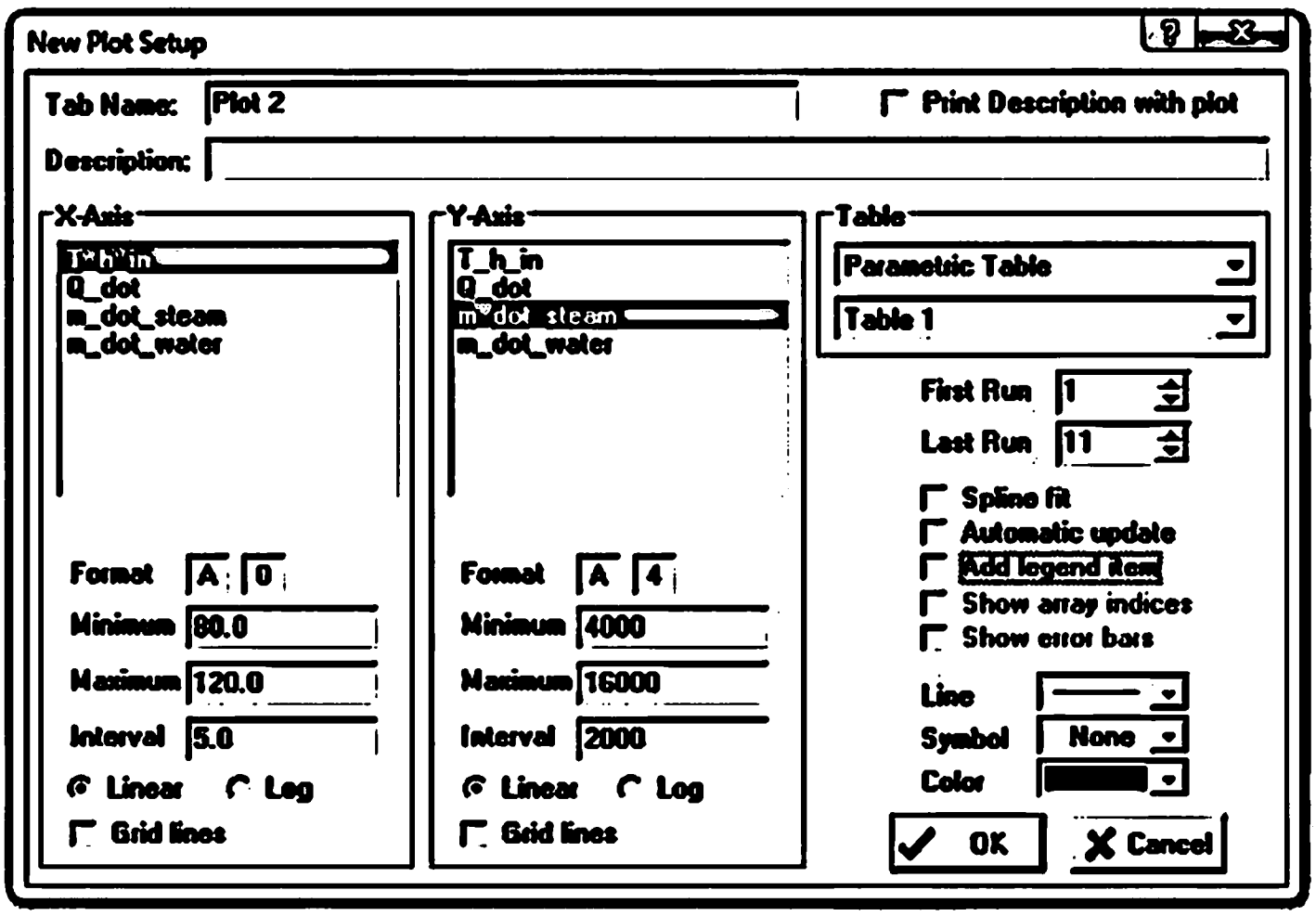
By clicking on OK the graph will be generated as shown,

Step 7 of 8

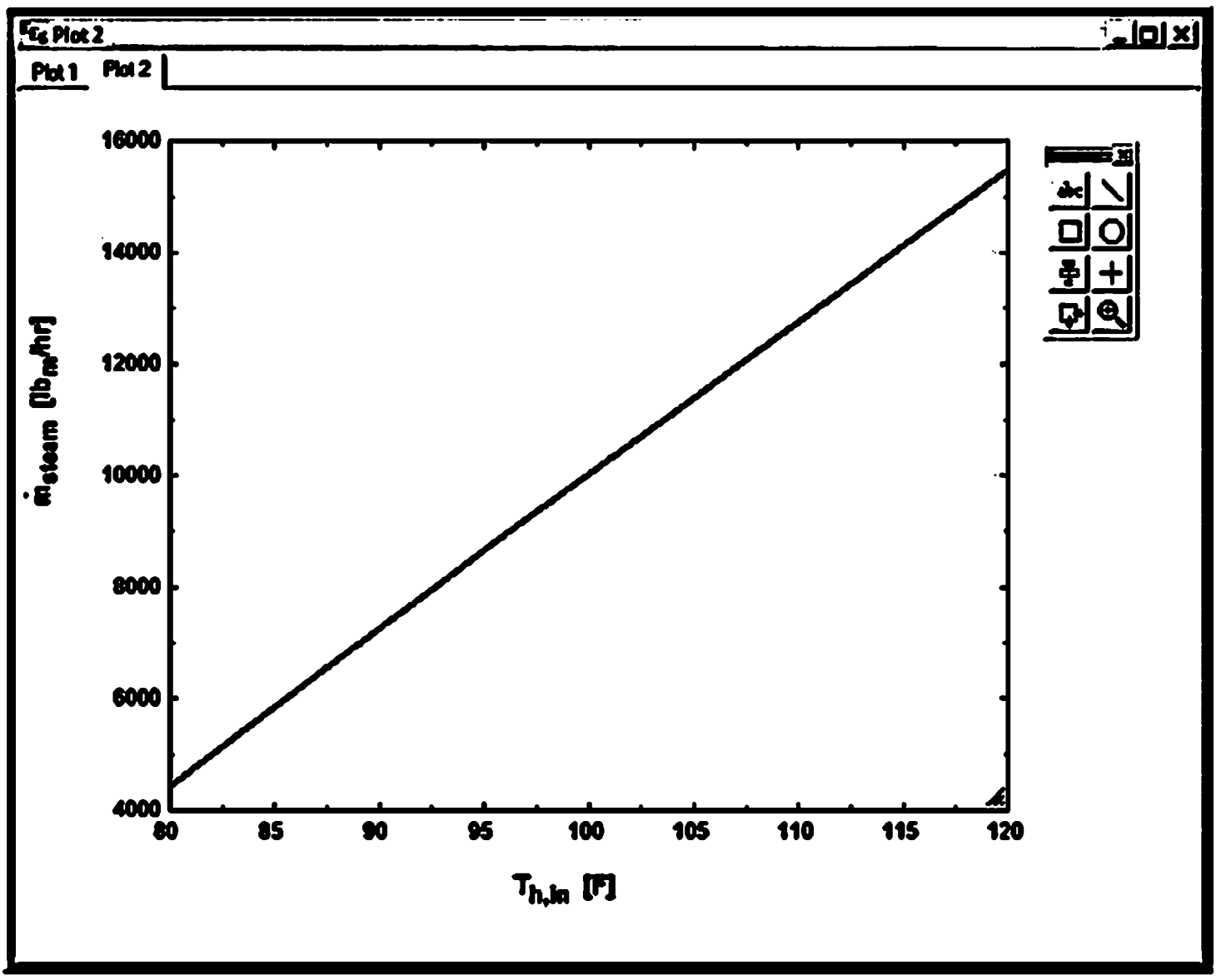
Step 7 of 8

The plot for $T_{h,in}$ Vs $\dot{m}_{steam}$

Now to generate a graph go to " Plots > New Plot window > X-Y Plot ", verify the settings as shown below,



Step 8 of 8



Question: 22.059P

Hot exhaust gases of a stationary diesel engine are to be used to generate steam in an evaporator. Exhaust gases (*cp* = 1051 J/kg·K) enter the heat exchanger at 550°C at a rate of 0.25 kg/s while water enters as saturated liquid and evaporates at 200°C (*hfg* = 1941 kJ/kg). The heat transfer surface area of the heat exchanger based on water side is 0.5 m2 and overall heat transfer coefficient is 1780 W/m2·K. Determine the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water.

Step 1 of 5

Given

The specific heat of exhaust gases, $c_{ph} = 1051\ \text{J/kg}^\circ\text{C}$

The heat of vaporization of exhaust gases is $h_{fg} = 1941\ \text{kJ/kg}$

Mass flow rate of exhaust gases $\dot{m}_h = 0.25\ \text{kg/s}$

Heat transfer surface area based on water side, $A_s = 0.5\ \text{m}^2$

Overall heat transfer coefficient $U = 1780\ \text{W/m}^2{}^\circ\text{C}$

Inlet temperature of exhaust gases, $T_{h,in} = 550^\circ\text{C}$

Temperature of evaporator liquid, $T_{c,in} = T_{c,out} = 200^\circ\text{C}$

Step 2 of 5

The LMTD between the steam and the exhaust gases:

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= (550 - 200)^\circ\text{C}$$
$$\Delta T = 350^\circ\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$\Delta T_2 = (T_{h,out} - 200)^\circ\text{C}$$
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$\Delta T_{lm} = \frac{350 - (T_{h,out} - 200)}{\ln\left[\frac{350}{(T_{h,out} - 200)}\right]}$$

Step 3 of 5

The rate of heat transfer is expressed as

$$\dot{Q} = UA_s \Delta T_{lm}$$

And $\dot{Q} = [\dot{m}_h C_{hp} (T_{h,out} - T_{h,in})]_{\text{exhaust gases}}$

Therefore

$$UA_s \Delta T_{lm} = [\dot{m}_h C_{ph} (T_{h,in} - T_{h,out})]_{\text{exhaust gases}}$$

$$1780 \times 0.5 \times \left\{\frac{350 - (T_{h,out} - 200)}{\ln\left[\frac{350}{(T_{h,out} - 200)}\right]}\right\} = 0.25 \times 1051 \times \left(550 - T_{h,out}\right)$$

$$\therefore T_{out} = 211.8^\circ\text{C}$$

Step 4 of 5

Heat transfer rate is determined by

$$\dot{Q} = [\dot{m}_h C_{hp} (T_{h,out} - T_{h,in})]_{\text{exhaust gases}}$$
$$= (0.25)(1051) \times (550\text{-}211.8)$$

$$\therefore \dot{Q} = 88.9\ \text{kW}$$

Step 5 of 5

Rate of vaporization of water can be calculated by

$$\dot{m}_{water} = \frac{\dot{Q}}{h_{fg}}$$
$$= \frac{88.9\ \text{kJ/s}}{1941\ \text{kJ/kg}}$$

$$\therefore \dot{m}_{water} = 0.0458\ \text{kg/s}$$

Question: 22.060P

Reconsider Prob. 22–59. Using EES (or other) software, investigate the effect of the exhaust gas inlet temperature on the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water. Let the temperature of exhaust gases vary from 300°C to 600°C. Plot the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water as a function of the temperature of the exhaust gases, and discuss the results.

Step 1 of 4

The given problem can be solved using EES software by following the below steps.

1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
"GIVEN"
"T_exhaust_in=550 [C]"
C_p_exhaust=1.051 [kJ/kg-C]
m_dot_exhaust=0.25 [kg/s]
T_w=200 [C]
h_fg_w=1941 [kJ/kg]
A=0.5 [m^2]
U=1.780 [kW/m^2-C]
"ANALYSIS"
DELTAT_1=T_exhaust_in-T_w
DELTAT_2=T_exhaust_out-T_w
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_exhaust*C_p_exhaust*(T_exhaust_in-T_exhaust_out)
Q_dot=m_dot_w*h_fg_w
```

Step 2 of 4

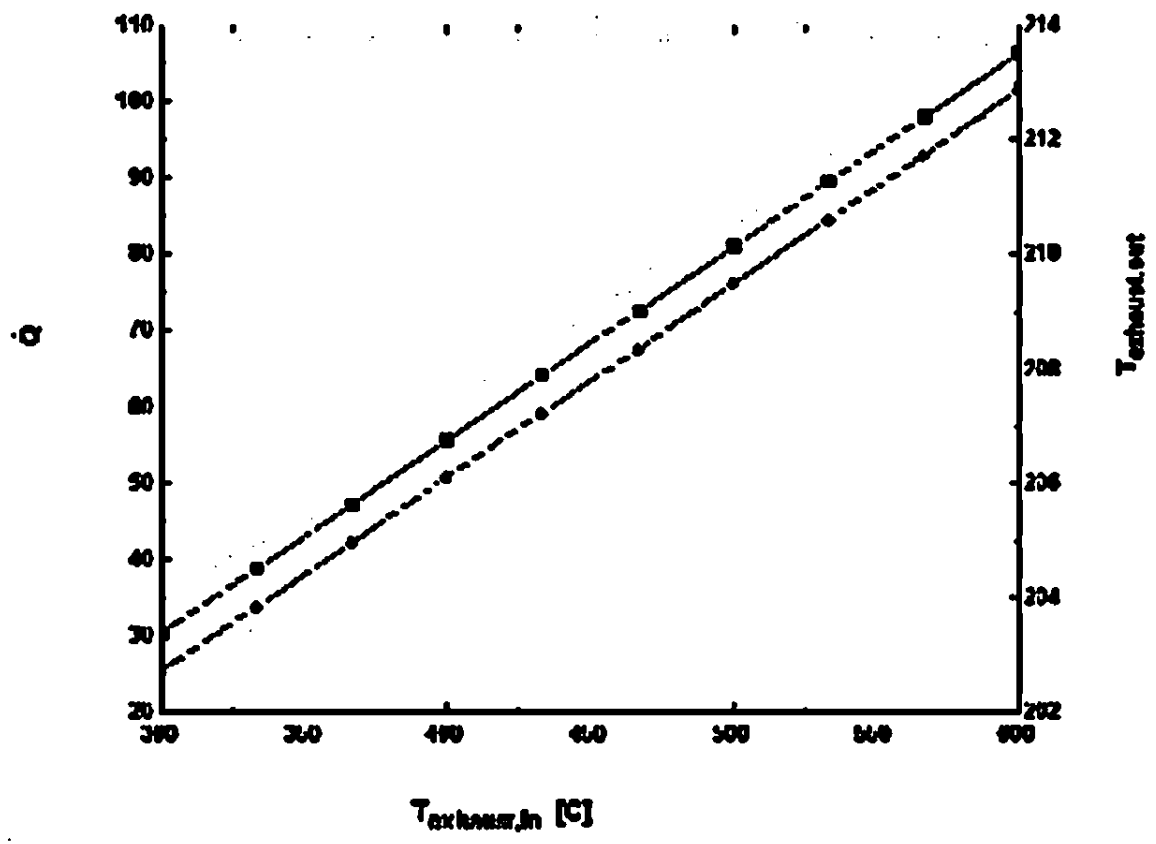
After writing the equations in the Equation window then click on table in menu bar. Then select new parameter table. The temperature of exhaust gases the varying from the value $T_{exhaust,in} = 300°C$ to $600°C$ we get

Table 1

	$\dot{m}_w$	$\dot{Q}$	$T_{exhaust,in}$ [C]	$T_{exhaust,out}$
Run 1	0.01308	25.39	300	203.4
Run 2	0.01744	33.85	333.3	204.5
Run 3	0.0218	42.31	366.7	205.6
Run 4	0.02615	50.77	400	206.8
Run 5	0.03052	59.24	433.3	207.9
Run 6	0.03488	67.7	466.7	209
Run 7	0.03921	76.16	500	210.1
Run 8	0.0435	84.62	533.3	211.3
Run 9	0.04795	93.09	566.7	212.4
Run 10	0.05232	101.5	600	213.5

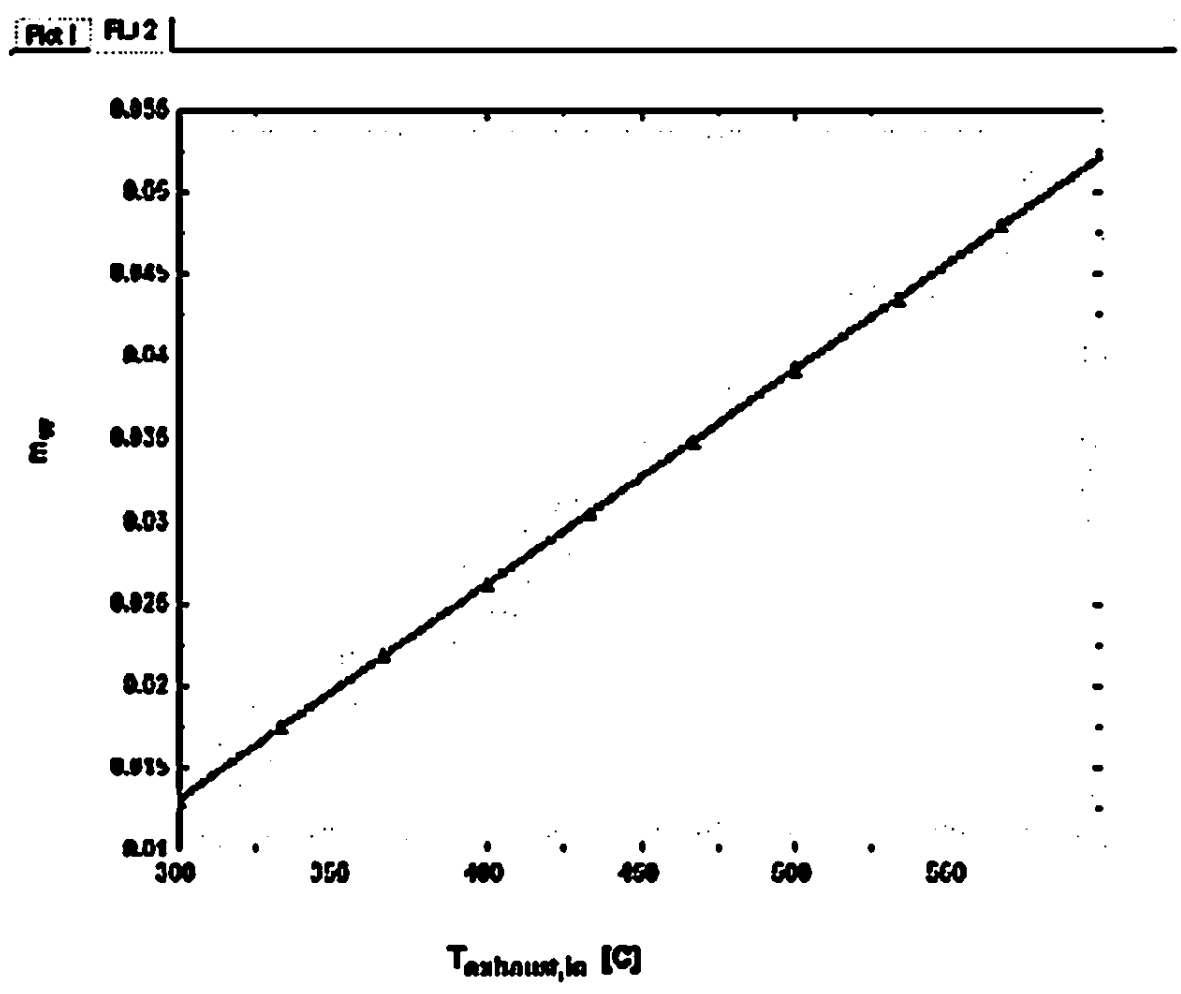
Step 3 of 4

To plot the rate of heat transfer, the exist temperature of the exhaust gases and the rate of evaporation of water as a function of the exhaust gases, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu. Now specify the X and Y parameters as $T_{exhaust,in}$ and $\dot{Q}$ and click OK. Now from the 'New Plot window' select overlay and specify the second parameters on the X-Y as $T_{exhaust,in}$ and $T_{exhaust,out}$.



Step 4 of 4

To plot the the rate of evaporation of water as a function of the exhaust gases, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu. Now specify the X and Y parameters as $T_{exhaust,in}$ and $\dot{m}_w$ and click OK.



Question: 22.061P

In a textile manufacturing plant, the waste dyeing water (cp = 4295 J/kg·K) at 80°C is to be used to preheat fresh water (cp = 4180 J/kg·K) at 10°C at the same flow rate in a double-pipe counter-flow heat exchanger. The heat transfer surface area of the heat exchanger is 1.65 m2 and the overall heat transfer coefficient is 625 W/m2·K. If the rate of heat transfer in the heat exchanger is 35 kW, determine the outlet temperature and the mass flow rate of each fluid stream.

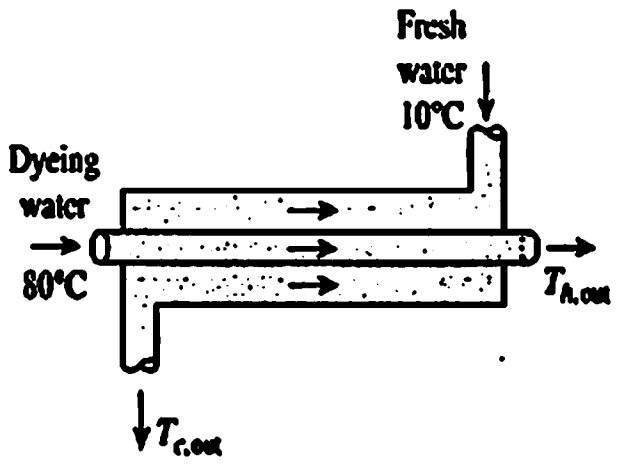


FIGURE P22–61

Step 1 of 5

Specific heat of waste dyeing water, $c_p = 4295\ \text{J/kg}\cdot\text{K}$

Specific heat of fresh water, $c_{pw} = 4180\ \text{J/kg}\cdot\text{K}$

Inlet temperature of waste dyeing water, $T_{h_in} = 80°\text{C}$

Inlet temperature of fresh water, $T_{c_in} = 10°\text{C}$

Overall heat transfer coefficient, $U = 625\ \text{W/m}^2\cdot\text{K}$

Surface area of the heat exchanger, $A_s = 1.65\ \text{m}^2$

The rate of heat transfer in the heat exchanger, $\dot{Q} = 35\ \text{kW}$

Step 2 of 5

The mass flow rate of waste dyeing water and fresh water are same and let it be $\dot{m}$

The temperature differences between the dyeing water and the fresh water at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$\Delta T_1 = 80 - T_{c,out}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$\Delta T_2 = T_{h,out} - 10$$

Step 3 of 5

Therefore the LMTD of the heat exchanger is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{\left(80 - T_{c,out}\right) - \left(T_{h,out} - 10\right)}{\ln\left[\frac{\left(80 - T_{c,out}\right)}{\left(T_{h,out} - 10\right)}\right]}$$

Step 4 of 5

The rate of heat transfer can be expressed as

$$\dot{Q} = UA_s\Delta T_{lm}$$

$$35 = 0.625 \times 1.65 \times \frac{\left(80 - T_{c,out}\right) - \left(T_{h,out} - 10\right)}{\ln\left[\frac{\left(80 - T_{c,out}\right)}{\left(T_{h,out} - 10\right)}\right]} \qquad \ldots\ldots (1)$$

The rate of heat transfer can also be expressed for dyeing water

$$\dot{Q} = \dot{m}c_p\left(T_{h,in} - T_{h,out}\right)$$

$$35 = \dot{m} \times 4.295\left(80 - T_{h,out}\right) \qquad \ldots\ldots (2)$$

Step 5 of 5

And the rate of heat transfer for fresh cooling water,

$$\dot{Q} = \dot{m}c_{pw}\left(T_{c,out} - T_{c,in}\right)$$

$$35 = \dot{m} \times 4.18\left(T_{c,out} - 10\right) \qquad \ldots\ldots (3)$$

Solving equations (1), (2) and (3) by using an equation solver such as EES, the unknowns are determined to be

Outlet temperature of fresh water is $\boxed{T_{c,out} = 40.6°\text{C}}$

Outlet temperature of waste dyeing water is $\boxed{T_{h,out} = 50.2°\text{C}}$

Mass flow rate is $\boxed{\dot{m} = 0.278\ \text{kg/s}}$.

Question: 22.062P

A heat exchanger contains 400 tubes with inner diameter of 23 mm and outer diameter of 25 mm. The length of each tube is 3.7 m. The corrected log mean temperature difference is 23°C, while the inner surface convection heat transfer coefficient is 3410 W/m2·K and the outer surface convection heat transfer coefficient is 6820 W/m2·K. If the thermal resistance of the tubes is negligible, determine the heat transfer rate.

Step 1 of 4

Perimeter of the inner surface,

$$p_i = \pi D_i$$

Here, inner diameter of tube, D_i

Substitute $23\,\text{mm}$ for D_i in the above equation

$$\begin{aligned} p_i &= \pi D_i \\ &= \pi \times 0.023 \\ &= 0.07226\,\text{m} \end{aligned}$$

Total inner surface area is,

$$A_i = p_i \times L \times n$$

Here, Length of the tube, L,

Number of tubes in the heat exchanger, n

Substitute $3.7\,\text{m}$ for L, 400 for n, $0.07226\,\text{m}$ for p_i in the below equation

$$\begin{aligned} A_i &= p_i \times L \times n \\ &= 0.07226 \times 3.7 \times 400 \\ &= 106.94\,\text{m} \end{aligned}$$

Step 2 of 4

Perimeter of the outer surface,

$$p_o = \pi D_o$$

Here, Outer diameter of tube, D_o

Substitute $25\,\text{mm}$ for D_o in the above equation

$$\begin{aligned} p_o &= \pi D_o \\ &= \pi \times 0.025 \\ &= 0.07854\,\text{m} \end{aligned}$$

Total outer surface area is;

$$A_o = p_o \times L \times n$$

Here, Length of the tube, L,

Number of tubes in the heat exchanger, n

Substitute $3.7\,\text{m}$ for L, 400 for n, $0.07854\,\text{m}$ for p_o in the below equation

$$\begin{aligned} A_o &= p_o \times L \times n \\ &= 0.07854 \times 3.7 \times 400 \\ &= 116.24\,\text{m} \end{aligned}$$

Step 3 of 4

Overall heat transfer coefficient is,

$$\frac{1}{UA_s} = \frac{1}{h_i A_i} + R_{wall} + \frac{1}{h_o A_o}$$

Here

Inner surface heat transfer coefficient, h_i

Outer surface heat transfer coefficient, h_o

Substitute $3410\ \text{W/m}^2\cdot\text{K}$ for h_i, $6820\ \text{W/m}^2\cdot\text{K}$ for h_o, $116.24\,\text{m}$ for A_o, $106.94\,\text{m}$ for A_i in the above equation

$$\begin{aligned} \frac{1}{UA_s} &= \frac{1}{h_i A_i} + R_{wall} + \frac{1}{h_o A_o} \\ &= \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \qquad \text{(Negleting wall thickness, } R_{wall} = 0) \\ &= \frac{1}{3410 \times 106.94} + \frac{1}{6820 \times 116.24} \\ &= 249771.41\,\text{W/m}^2\cdot\text{K} \end{aligned}$$

Step 4 of 4

Heat transfer rate is,

$$\dot{Q} = UA_s \Delta T_{lm}$$

Here, Corrected log mean temperature difference, ΔT_{lm}

Substitute $23°\text{C}$ for ΔT_{lm}, $249771.41\,\text{W/m}^2\cdot\text{K}$ for (UA_s) in the above equation

$$\begin{aligned} \dot{Q} &= UA_s \Delta T_{lm} \\ &= 249771.41 \times 23 \\ &= 5744742\,\text{W/m}^2 \\ &= 5744.71\ \text{kW/m}^2 \end{aligned}$$

The total heat transfer rate is obtained as $\boxed{5744.71\text{kW/m}^2}$

Question: 22.063P

A single pass heat exchanger is to be designed to heat 100,000 lbm of water in an hour from 60°F to 100°F by condensation of water vapor at 230°F on the shell side. Each tube has an inner diameter of 1.2 in. and a wall thickness of 0.12 in. The inner surface convection heat transfer coefficient is 480 Btu/h·ft2·°F and the outer surface convection heat transfer coefficient is 2000 Btu/h·ft2·°F. If the inlet velocity of water (*cpc* = 1 Btu/lbm·°F and ρ = 62.3 lbm/ft3) on the tube side is 4 ft/s, determine the required number of tubes and length of tubes. Assume the thermal resistance of the tubes is negligible.

Step 1 of 6

Mass flow rate of cold water,

$$\dot{m}_c = A \times V \times \rho \times N$$

$$= \frac{\pi}{4} \times d_i^2 \times V \times \rho \times N$$

Here, Number of tubes in the heat exchanger, N, Length of the tube, L, Velocity of the water, V ,Density of water, ρ , Mass flow rate of the water to be heated, $\dot{m}_c$, Inner diameter of the tube, d_i

$$\dot{m}_c = 10000 \text{ lbm/hr}$$

$$= 27.78 \text{ lbm/s}$$

Substitute 27.78 lbm/s for $\dot{m}_c$, 62.3 lbm/ft^3 for ρ, 4 ft/s for V, 1.2 in. for d_i in the above equation.

$$\dot{m}_c = \frac{\pi}{4} \times d_i^2 \times V \times \rho \times N$$

$$27.78 = \frac{\pi}{4} \times \left(\frac{1.2}{12}\right)^2 \times 4 \times 62.3 \times N$$

$$N = 14.2$$

The number of tubes required is $\boxed{N = 14.2}$

Step 2 of 6

Logarithmic mean temperature is,

$$\Delta T_{lm} = \frac{(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})}{\ln\left(\frac{(T_{h,i} - T_{c,i})}{(T_{h,o} - T_{c,o})}\right)}$$

Initial temperature of the water to be heated, $T_{c,i}$

Final temperature of the water after heated, $T_{c,o}$

Condensation water temperature, $T_{h,i} = T_{h,o}$

$$= T_{con}$$

Substitute 60°F for $T_{c,i}$, 100°F for $T_{c,o}$, 230°F for $T_{h,i}$ and $T_{h,o}$ in the above equation

$$\Delta T_{lm} = \frac{(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})}{\ln\left(\frac{(T_{h,i} - T_{c,i})}{(T_{h,o} - T_{c,o})}\right)}$$

$$= \frac{(230 - 60) - (230 - 100)}{\ln\left(\frac{(230 - 60)}{(230 - 100)}\right)}$$

$$= 149.1°\text{F}$$

Step 3 of 6

Perimeter of the inner surface,

$$p_i = \pi d_i$$

Here, inner diameter of tube, d_i

Substitute 1.2 in. for d_i in the above equation

$$p_i = \pi d_i$$

$$= \pi \times 1.2$$

$$= 3.77 \text{ in.}$$

$$= 0.314 \text{ ft}$$

Total inner surface area is,

$$A_i = p_i \times L \times n$$

Here, Length of the tube, L,
Number of tubes in the heat exchanger, N

Substitute 14.2 for N, 0.314 ft for p_i in the below equation

$$A_i = p_i \times L \times N$$

$$= 0.314 \times L \times 14.2$$

$$= 4.46L$$

Step 4 of 6

Perimeter of the outer surface,

$$p_o = \pi d_o$$

$$= \pi(d_i + t)$$

Here, Thickness of the pipe, t

Substitute 1.2 in. for d_i, 0.12 in. for t in the above equation;

$$p_o = \pi(d_i + t)$$

$$= \pi(1.2 + 0.12)$$

$$= 4.146 \text{ in.}$$

$$= 0.3455 \text{ ft}$$

Total outer surface area is,

$$A_o = p_o \times L \times N$$

Substitute 14.2 for N, 0.3455 ft for p_o in the above equation

$$A_o = p_o \times L \times N$$

$$= 0.3455 \times L \times 14.2$$

$$= 4.91L$$

Step 5 of 6

Overall heat transfer coefficient is,

$$\frac{1}{UA_s} = \frac{1}{h_i A_i} + R_{wall} + \frac{1}{h_o A_o}$$

Here

Inner surface heat transfer coefficient, h_i

Outer surface heat transfer coefficient,

(Negleting wall thickness, $R_{wall} = 0$)

$$\frac{1}{UA_s} = \frac{1}{h_i A_i} + R_{wall} + \frac{1}{h_o A_o}$$

$$= \frac{1}{h_i A_i} + \frac{1}{h_o A_o}$$

Substitute 480 Btu/h·ft^2·°F for h_i, 2000 Btu/h·ft^2·°F for h_o, $4.91L$ for A_o, $4.46L$ for A_i in the above equation

$$\frac{1}{UA_s} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o}$$

$$= \frac{1}{480 \times [4.46L]} + \frac{1}{2000 \times [4.91L]}$$

$$UA_s = 1757.63L \text{ Btu/h·ft}^2\text{·°F}$$

Step 6 of 6

Heat transfer rate is,

$$\dot{Q} = UA_s \Delta T_{lm}$$

Here, Corrected log mean temperature difference, ΔT_{lm}

Substitute 149.1 for ΔT_{lm}, $1757.63L$ Btu/h·ft^2·°F for (UA_s) in the above equation

$$\dot{Q} = UA_s \Delta T_{lm}$$

$$\dot{m} c_p (T_{c,o} - T_{c,i}) = UA_s \Delta T_{lm}$$

$$100000 \times 1 \times (100 - 60) = (1757.63L) \times 149.1$$

$$L = 1.53 \text{ ft}$$

$$= 18.32 \text{ in.}$$

There for the each of length is $\boxed{L = 18.32 \text{ in}}$.

Question: 22.064P

What does the effectiveness of a heat exchanger represent? Can effectiveness be greater than one? On what factors does the effectiveness of a heat exchanger depend?

Step 1 of 1

Effectiveness of heat exchanger is defined as the ratio of actual heat transfer rate to the maximum possible heat transfer rate.

$$\varepsilon = \frac{\dot{Q}}{Q_{max}}$$

Since the actual heat transfer is always less than maximum possible heat transfer effectiveness is always less than one.

Effectiveness of heat exchanger depends on the **geometry** of the heat exchanger as well as **flow arrangement**.

Question: 22.065P

For a specified fluid pair, inlet temperatures, and mass flow rates, what kind of heat exchanger will have the highest effectiveness: double-pipe parallel-flow, double-pipe counter-flow, cross-flow, or multipass shell-and-tube heat exchanger?

Step 1 of 1

Double-pipe counter-flow heat exchanger has high effectiveness for a specified fluid pair, inlet temperature and mass flow rate.

Question: 22.066P

Explain how you can evaluate the outlet temperatures of the cold and hot fluids in a heat exchanger after its effectiveness is determined.

Step 1 of 1

Effectiveness can be found by $\varepsilon = \dfrac{\dot{Q}_{act}}{\dot{Q}_{max}}$

$$\dot{Q}_{act} = \varepsilon \dot{Q}_{max}$$

Outlet temperatures of hot fluid:

$$\dot{m}_h C_{p,h}(T_{h,in} - T_{h,out}) = \varepsilon C_{min}(T_{h,in} - T_{c,in})$$

$$\therefore T_{h,out} = T_{h,in} + \frac{(\varepsilon)(C_{min})(T_{h,in} - T_{c,in})}{\dot{m}_h C_{p,h}}$$

On substituting all values on right hand side we can find $T_{h,out}$

Outlet temperatures of cold fluid:

$$\dot{m}_c C_{p,c}(T_{c,in} - T_{c,out}) = \varepsilon C_{min}(T_{h,in} - T_{c,in})$$

$$\therefore T_{c,out} = T_{c,in} - \frac{(\varepsilon)(C_{min})(T_{h,in} - T_{c,in})}{\dot{m}_c C_{p,c}}$$

On substituting all values on right hand side we can find $T_{c,out}$

Question: 22.067P

Can the temperature of the cold fluid rise above the inlet temperature of the hot fluid at any location in a heat exchanger? Explain.

Step 1 of 1

While exchanging the heat from cold fluid to the hot fluid, temperature of the cold fluid rises and may reach to the maximum temperature i.e. inlet temperature of the hot fluid.

The temperature of the cold fluid will not rise above the inlet temperature of hot fluid at any location.

Question: 22.068P

Consider a heat exchanger in which both fluids have the same specific heats but different mass flow rates. Which fluid will experience a larger temperature change: the one with the lower or higher mass flow rate?

Step 1 of 1

By energy balance is given by

$$\dot{Q} = \dot{m}_c c_{p,c} \Delta T_{cold} = \dot{m}_h c_{p,h} \Delta T_{hot}$$

Since both fluids have same specific heats

i.e., $c_{p,c} = c_{p,h}$

$$\dot{m}_c \Delta T_{cold} = \dot{m}_h \Delta T_{hot}$$

From this relation it is clearly understood that the fluid with lower mass flow rate will experience a larger temperature change.

Question: 22.069P

Explain how the maximum possible heat transfer rate $\dot{Q}$max in a heat exchanger can be determined when the mass flow rates, specific heats, and the inlet temperatures of the two fluids are specified. Does the value of $\dot{Q}$max depend on the type of the heat exchanger?

Step 1 of 1

The rate of heat transfer for a given mass flow rates, specific heats and inlet temperatures of the two fluids can be determined as follows

$$C_h = \dot{m} c_{ph}$$

$$C_c = \dot{m}_c c_{pc}$$

$$C_{min} = \text{min of } (C_h, C_c)$$

The fluid with smaller heat capacity rate will experience a larger temperature change, and thus it will be the first to experience the maximum temperature

Therefore, maximum possible heat transfer rate in a heat exchanger is

$$\therefore \dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$

$\dot{Q}_{max}$ is same for all heat exchangers. It is independent of type of heat exchanger.

Question: 22.070P

Under what conditions can a counter-flow heat exchanger have an effectiveness of one? What would your answer be for a parallel-flow heat exchanger?

Step 1 of 1

Effectiveness for a counter flow heat exchanger is given by

$$\varepsilon = \frac{1-\exp\left[-NTU(1-c)\right]}{1-c\exp\left[-NTU(1-c)\right]}$$

For $\varepsilon = 1$

$$1-\exp\left[-NTU(1-c)\right] = 1-c\exp\left[-NTU(1-c)\right]$$

$$c = 1$$

When capacity ratio c is equal to one and NTU value is very large then the counter flow heat exchanger will have an effectiveness of one i.e. 100% effectiveness.

In this case parallel flow heat exchanger will have 50% effectiveness.

Question: 22.071P

Consider a double-pipe counter-flow heat exchanger. In order to enhance heat transfer, the length of the heat exchanger is now doubled. Do you think its effectiveness will also double?

Step 1 of 1

NTU is expressed by $NTU = \frac{UA_s}{C_{min}}$

Where A_s : heat transfer surface area

$A_s \propto$ Length of heat exchanger

By increasing length of the heat exchanger NTU of heat exchanger will be increased.

Effectiveness for double pipe counter flow heat exchanger is given by

$$\varepsilon = \frac{1-\exp\left[-NTU(1-c)\right]}{1-c\exp\left[-NTU(1-c)\right]}$$

Effectiveness increases rapidly with NTU for small values but rather slowly for larger values.

Therefore effectiveness of the heat exchanger will not be double when the length of the heat exchanger is doubled.

Question: 22.072P

Consider two double-pipe counter-flow heat exchangers that are identical except that one is twice as long as the other one. Which heat exchanger is more likely to have a higher effectiveness?

Step 1 of 1

NTU is expressed by $NTU = \frac{UA_s}{C_{min}}$

Where A_s : heat transfer surface area

$A_s \propto$ Length of heat exchanger

By increasing length of the heat exchanger NTU of heat exchanger will be increased.

Effectiveness for double pipe counter flow heat exchanger is given by

$$\varepsilon = \frac{1-\exp[-NTU(1-c)]}{1-c\exp[-NTU(1-c)]}$$

Effectiveness increases rapidly with NTU for small values (up to about NTU=1.5) but rather slowly for larger values.

Therefore double-pipe counter-flow heat exchanger with double length will have the higher effectiveness.

Question: 22.073P

How is the NTU of a heat exchanger defined? What does it represent? Is a heat exchanger with a very large NTU (say, 10) necessarily a good one to buy?

Step 1 of 1

NTU of the heat exchanger is a dimensionless group and is represented as

$$NTU = \frac{UA_s}{C_{min}}$$

Where U is the overall heat transfer coefficient and A_s is heat transfer surface area of heat exchanger.

For specified values of U and C_{min}, the value of NTU is a measure of the heat transfer surface area A_s. Thus, the longer the NTU, the larger the heat exchanger.

Effectiveness of heat exchanger increase slowly for large value of NTU. So large value of NTU in a heat exchanger is not economically justified. Therefore it is not necessarily a good one to buy.

Question: 22.074P

Consider a heat exchanger that has an NTU of 4. Someone proposes to double the size of the heat exchanger and thus double the NTU to 8 in order to increase the effectiveness of the heat exchanger and thus save energy. Would you support this proposal?

Step 1 of 1

Effectiveness increases rapidly with NTU for small values (up to about NTU=1.5) but rather slowly for larger values. Therefore the use of heat exchanger with a large NTU (usually larger than 3) and thus a large size cannot be justified.

Large increase in NTU corresponds to small increase in effectiveness.

So it is not desirable to double the size of the heat exchanger and thus double the NTU in order to increase the effectiveness of heat exchanger and thus save the energy.

Question: 22.075P

Consider a heat exchanger that has an NTU of 0.1. Someone proposes to triple the size of the heat exchanger and thus triple the NTU to 0.3 in order to increase the effectiveness of the heat exchanger and thus save energy. Would you support this proposal?

Step 1 of 1

Effectiveness increases rapidly with NTU for small values (up to about NTU=1.5) but rather slowly for larger values. Therefore the use of heat exchanger with a large NTU (usually larger than 3) and thus a large size cannot be justified.

Small increase in NTU corresponds to large increase in effectiveness.

Therefore the proposal of small increase in NTU (from 0.1 to 0.3) in order to increase the effectiveness can be supported.

Question: 22.076P

A 1-shell and 2-tube type heat exchanger has an overall heat transfer coefficient of 300 Btu/h-ft2·°F. The shell side fluid has a heat capacity rate of 20,000 Btu/h·°F, while the tube side fluid has a heat capacity rate of 40,000 Btu/h·°F. The inlet temperatures on the shell side and tube side are 200°F and 90°F, respectively. If the total heat transfer area is 100 ft2, determine (*a*) the heat transfer effectiveness and (*b*) the actual heat transfer rate in the heat exchanger.

Step 1 of 4

Given data:

The overall heat transfer coefficient, $U = 300\ \text{Btu/h.ft}^2\text{.°F}$

Heat capacity on shell and tube side,

$C_{shell} = 20,000\ \text{Btu/h.°F}$

$C_{tube} = 40,000\ \text{Btu/h.°F}$

Inlet temperatures of shell and tube side are

$T_{shell,in} = 200°\text{F}$

$T_{tube,in} = 90°\text{F}$

The heat transfer surface area, $A_s = 100\ \text{ft}^2$

Step 2 of 4

The smaller of two heat capacities are

$C_{min} = C_{shell} = 20,000\ \text{Btu/h.°F}$

The maximum rate of heat transfer in the shell and tube heat exchanger is

$$\dot{Q} = C_{min}\left(T_{shell,in} - T_{tube,in}\right)$$
$$= 20000(200-90)$$
$$= 2200000\ \text{Btu/h}$$

$$\dot{Q}_{max} = 2200000 \times \frac{1}{3600}$$
$$= 611.1\ \text{Btu/s}$$

Step 3 of 4

The NTU of heat exchanger is expressed as

$$NTU = \frac{UA_s}{C_{min}}$$
$$= \frac{(300)100}{20000}$$
$$= 1.5$$

Also

$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{20000}{40000}$$
$$= 0.5$$

From graph Effectiveness of heat exchangers for 1 shell pass and two tube passes at $NTU = 1.5$ and $c = 0.5$ the value of effectiveness is determined as

$\boxed{\varepsilon = 0.64}$

Therefore heat transfer effectiveness is $\varepsilon = 0.64$

Step 4 of 4

(b)

The actual heat transfer rate of the heat exchanger is

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$
$$= (0.64)(611.1)$$
$$= 391.104\ \text{Btu/s}$$

$\boxed{\dot{Q} = 391.104\ \text{Btu/s}}$

Therefore, the actual heat transfer rate of the heat exchanger is $\dot{Q} = 391.104\ \text{Btu/s}$

Question: 22.077P

A cross-flow heat exchanger with both fluids unmixed has an overall heat transfer coefficient of 200 W/m2·K, and a heat transfer surface area of 400 m2. The hot fluid has a heat capacity of 40,000 W/K, while the cold fluid has a heat capacity of 80,000 W/K. If the inlet temperatures of both hot and cold fluids are 80°C and 20°C, respectively, determine the exit temperature of the cold fluid.

Step 1 of 4

Given data:

The overall heat transfer coefficient $U = 200\,\mathrm{W/m^2\text{-}K}$

Heat capacity on hot and cold fluids is

$C_h = 40,000\,\mathrm{W/K}$

$C_c = 80,000\,\mathrm{W/K}$

Inlet temperatures of on hot and cold fluids are

$T_{h,in} = 80°\mathrm{C}$

$T_{c,in} = 20°\mathrm{C}$

The surface area $A_s = 400\,\mathrm{m^2}$

Step 2 of 4

The smaller of two heat capacities are

$C_{min} = C_h = 40,000\,\mathrm{W/K}$

The maximum rate of heat transfer in a cross flow heat exchanger is

$$\dot{Q} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

$$\dot{Q} = 40000(80-20)$$

$$\dot{Q}_{max} = 2400\ \mathrm{kW}$$

The NTU of heat exchanger is expressed as

$$NTU = \frac{UA_s}{C_{min}}$$

$$= \frac{(200)400}{40000}$$

$$NTU = 2$$

Step 3 of 4

Also

$$c = \frac{C_{min}}{C_{max}}$$

$$= \frac{40000}{80000}$$

$$c = 0.5$$

From graph effectiveness for heat exchangers at $c = 0.5$ and $NTU = 2$ the value of effectiveness is determined as

$\varepsilon = 0.74$

The actual heat transfer rate of the heat exchanger is

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$

$$\dot{Q} = (0.74)(2400)$$

$$\dot{Q} = 1776\ \mathrm{kW}$$

Step 4 of 4

The exit temperature of the cold side is determined as

$$\dot{Q} = C_c\left(T_{c,out} - T_{c,in}\right)$$

$$1776\times10^3 = 80000\left(T_{c,out} - 20\right)$$

$$T_{c,out} = 42.2°\mathrm{C}$$

The exit temperature of the cold side $\boxed{T_{c,out} = 42.2°\mathrm{C}}$

Question: 22.078P

In a 1 -shell and 2-tube heat exchanger, cold water with inlet temperature of 20°C is heated by hot water supplied at the inlet at 80°C. The cold and hot water flow rates are 5000 kg/h and 10,000 kg/h, respectively. If the shell-and-tube heat exchanger has a *UAS* value of 11,600 W/K, determine the cold water and hot water outlet temperatures. Assume *cpc* = 4178 J/kg·K and *cph* = 4188 J/kg·K.

Step 1 of 6

The ratio of the heat capacities is,

$$c = \frac{C_{min}}{C_{max}} = \frac{(\dot{m}c_p)_c}{(\dot{m}c_p)_h}$$

Here,

Mass flow rate of cold fluid, $\dot{m}_c$

Mass flow rate of hot fluid, $\dot{m}_h$

Specific heat of cold fluid, $c_{p,c}$

Specific heat of hot fluid, $c_{p,h}$

Substitute 5,000 kg/hr for $\dot{m}_h$, 10,000 kg/hr for $\dot{m}_c$, 4,178 J/kg·K for $c_{p,c}$ 4,188 J/kg-K for $c_{p,h}$ in the above equation

$$c = \frac{(\dot{m}c_p)_c}{(\dot{m}c_p)_h} = \frac{5,000 \times 4,178}{10,000 \times 4,188} = 0.498$$

Step 2 of 6

The smaller of the two heat capacities,

$$C_{min} = C_c = (\dot{m}c_p)_c = \frac{5,000}{60 \times 60}\text{kg/s} \times 4,178 = 5,802.778 \text{ W/K}$$

The NTU of heat exchanger is;

$$\text{NTU} = \frac{UA_s}{C_{min}}$$

Here,

The product of the overall heat transfer coefficient and the area of the 1-shell, 2-tube heat exchanger, UA_s

Substitute 11,600 W/K for UA_s, 5,802.778 W/K for C_{min}, in the above equation

$$\text{NTU} = \frac{UA_s}{C_{min}} = \frac{11,600}{5,802.778} = 1.99$$

Step 3 of 6

The maximum heat transfer rate is,

$$\dot{Q}_{max} = C_{min}(T_{h,i} - T_{c,i})$$

Here,

Inlet temperature of cold fluid, $T_{c,i}$

Inlet temperature of hot fluid, $T_{h,i}$

Substitute 20°C for $T_{c,i}$, 80°C for $T_{h,i}$, 5,802.778 W/K for C_{min}, in the above equation

$$\dot{Q}_{max} = C_{min}(T_{h,i} - T_{c,i}) = 5,802.778 \times (80 - 20) = 5,802.778 \times 60 = 348,166.667 \text{ W}$$

Step 4 of 6

From the graph containing effectiveness and NTU, at $c = 0.498$ and $\text{NTU} = 1.99$ the value of effectiveness is determined to be

$$\varepsilon = 0.68$$

The actual rate is,

$$\dot{Q} = \varepsilon \dot{Q}_{max} = 0.68 \times 348,166.667 = 236,753.334 \text{ W}$$

Step 5 of 6

Determine the exit temperatures of the cold and hot fluids as follows.
Consider the expression for the actual heat transfer rate in terms of cold fluid temperatures.

$$\dot{Q} = C_c(T_{c,o} - T_{c,i}) = (\dot{m}c_p)_c(T_{c,o} - T_{c,i})$$

Here, exit temperature of cold fluid, $T_{c,o}$

Substitute, 10,000 kg/hr for $\dot{m}_c$, 4,178 J/kg-K for $c_{p,c}$, 20°C for $T_{c,i}$, 236,753.334 W for $\dot{Q}$ in the above equation

$$\dot{Q} = (\dot{m}c_p)_c(T_{c,o} - T_{c,i})$$

$$236,753.334 = \left(\frac{5,000}{60 \times 60} \times 4,178\right)(T_{c,o} - 20)$$

$$T_{c,o} = \frac{236,753.334 \times 60 \times 60}{5,000 \times 4,178} + 20 = 60.8°\text{C}$$

Therefore the exit temperatures of the cold fluid is **60.8°C**

Step 6 of 6

Similarly, the heat transfer rate in terms of hot fluid temperatures is,

$$\dot{Q} = C_h(T_{h,i} - T_{h,o}) = (\dot{m}c_p)_h(T_{h,i} - T_{h,o})$$

Here, Exit temperature of hot fluid, $T_{h,o}$

Substitute, 10,000 kg/hr for $\dot{m}_c$, 4,188 J/kg·K for $c_{p,h}$, 20°C for $T_{c,i}$, 236,753.334 W for $\dot{Q}$ in the above equation

$$\dot{Q} = (\dot{m}c_p)_h(T_{h,i} - T_{h,o})$$

$$236,753.334 = \left(\frac{10,000}{60 \times 60} \times 4,188\right)(80 - T_{h,o})$$

$$T_{h,o} = 80 - \frac{236,753.334 \times 60 \times 60}{10,000 \times 4,188} = 59.64°\text{C}$$

Therefore the exit temperatures of hot fluid is **59.64°C**.

Question: 22.079P

The radiator in an automobile is a cross-flow heat exchanger (*UAS* = 10 kW/K) that uses air (*cp* = 1.00 kJ/kg·K) to cool the engine-coolant fluid (*cp* = 4.00 kJ/kg-K). The engine fan draws 22°C air through this radiator at a rate of 8 kg/s while the coolant pump circulates the engine coolant at a rate of 5 kg/s. The coolant enters this radiator at 80°C. Under these conditions, the effectiveness of the radiator is 0.4. Determine (*a*) the outlet temperature of the air and (*b*) the rate of heat transfer between the two fluids.

Step 1 of 3

The ratio of the heat capacities is,

$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{(\dot{m}c_p)_c}{(\dot{m}c_p)_h}$$

Here,

Mass flow rate of cold fluid, $\dot{m}_c$

Mass flow rate of hot fluid, $\dot{m}_h$

Specific heat of cold fluid, $c_{p,c}$

Specific heat of hot fluid, $c_{p,h}$

Substitute 5 kg/s for $\dot{m}_h$, 8 kg/s for $\dot{m}_c$, 1,000 J/kg·K for $c_{p,c}$ 4,000 J/kg·K for $c_{p,h}$ in the above equation

$$c = \frac{(\dot{m}c_p)_c}{(\dot{m}c_p)_h}$$
$$= \frac{8\times1{,}000}{5\times4{,}000}$$
$$= 0.4$$

The smaller of the two heat capacities,

$$C_{min} = C_c$$
$$= (\dot{m}c_p)_c$$
$$= 8\times1{,}000$$
$$= 8{,}000 \text{ W/K}$$

Step 2 of 3

b)

For the ease of calculation, first determine the heat transfer rate between the fluids is.

The maximum heat transfer rate is,

$$\dot{Q}_{max} = C_{min}(T_{h,i} - T_{c,i})$$

Here,

Inlet temperature of cold fluid, $T_{c,i}$

Inlet temperature of hot fluid, $T_{h,i}$

Substitute 22°C for $T_{c,i}$, 80°C for $T_{h,i}$, 8,000 W/K for C_{min}, in the above equation

$$\dot{Q}_{max} = C_{min}(T_{h,i} - T_{c,i})$$
$$= 8{,}000\times(80-22)$$
$$= 8{,}000\times58$$
$$= 464{,}000 \text{ W}$$

The actual heat transfer rate between the two fluids is,

$$\dot{Q} = \varepsilon\dot{Q}_{max}$$

Here, The effectiveness of the of the radiator, $\varepsilon = 0.4$

$$\dot{Q} = \varepsilon\dot{Q}_{max}$$
$$= 0.4\times464{,}000$$
$$= 185{,}600 \text{ W}$$

Therefore, the amount of heat transferred from the

Step 3 of 3

a)

Determine the exit temperature of the cold fluid is as follows.

The product of the overall heat transfer coefficient and the area of the cross flow heat exchanger, $UA_s = 10{,}000$ W/K

Consider the expression for the actual heat transfer rate in terms of cold fluid temperatures.

$$\dot{Q} = C_c(T_{c,o} - T_{c,i})$$
$$= (\dot{m}c_p)_c(T_{c,o} - T_{c,i})$$

Here, exit temperature of cold fluid, $T_{c,o}$

$$\dot{Q} = C_c(T_{c,o} - T_{c,i})$$
$$= (\dot{m}c_p)_c(T_{c,o} - T_{c,i})$$
$$185{,}600 = (8\times1{,}000)(T_{c,o} - 22)$$
$$T_{c,o} = \frac{185{,}600}{8{,}000} + 22$$
$$= 45.2°\text{C}$$

Therefore the exit temperature of the air or the cold fluid is $\boxed{45.2°\text{C}}$.

Question: 22.080P

Water (*cp* = 4180 J/kg·K) enters the 2.5-cm-internal diameter tube of a double-pipe counter-flow heat exchanger at 20°C at a rate of 2.2 kg/s. Water is heated by steam condensing at 120°C (*hfg* = 2203 kJ/kg) in the shell. If the overall heat transfer coefficient of the heat exchanger is 700 W/m2·K, determine the length of the tube required in order to heat the water to 80°C using (*a*) the LMTD method and (*b*) the ε-NTU method.

Step 1 of 6

Given data:

Diameter of the tube, $D = 2.5\,\text{cm}$

Specific heat water, $c_p = 4180\ \text{J/kg}\cdot\text{K}$

Mass flow rate of water, $\dot{m}_w = 2.2\ \text{kg/s}$

Initial temperature of water, $T_{in} = 20\,^\circ\text{C}$

Exit temperature of water, $T_{out} = 80\,^\circ\text{C}$

Steam temperature $T = 120\,^\circ\text{C}$

Overall heat transfer coefficient, $U = 700\ \text{W/m}^2\cdot\text{K}$

Enthalpy of the steam, $h_{fg} = 2203\ \text{kJ/kg}$

Step 2 of 6

a) Determination of length of the tube by Log Mean Temperature Difference method:

The heat transfer rate is given by,

$$\dot{Q} = [\dot{m}_c c_{pc}(T_{out} - T_{in})]_{water}$$
$$= 2.2\times4180\times(80-20)$$
$$= 551.76\ \text{kW}$$

Calculation of LMTD for the given counter flow heat exchanger:

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= (120-80)$$
$$= 40^\circ\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= (120-20)$$
$$\Delta T_2 = 100^\circ\text{C}$$

Now, LMTD is,

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$
$$= \frac{40-100}{\ln\left(40/100\right)}$$
$$= 65.48\,^\circ\text{C}$$

Step 3 of 6

The heat transfer surface area is calculated using the formula,

$$\dot{Q} = UA_s\Delta T_{lm}$$
$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}}$$
$$= \frac{551.76\times10^3}{(700)(65.48)}$$
$$= 12.04\ \text{m}^2$$

Length of the tube is calculated using the formula,

$$A_s = \pi DL$$
$$L = \frac{A_s}{\pi D}$$
$$= \frac{12.04\ \text{m}^2}{\pi(0.025\ \text{m})}$$
$$\boxed{L = 153.298\ \text{m}}$$

Step 4 of 6

b) Calculation of the length of the tube by Effectiveness-Number of Transfer Units method:

The maximum rate of heat transfer is given by,

$$\dot{Q} = \dot{m}_c c_{pc}\left(T_{h,out} - T_{w,in}\right)$$
$$= 2.2\times4180\times(120-20)$$
$$= 919.6\ \text{kW}$$

Effectiveness of the heat exchanger is calculated using the formula,

$$\varepsilon = \frac{Q}{Q_{max}}$$
$$= \frac{551.76}{919.6}$$
$$= 0.6$$

The NTU of this heat exchanger is determined using the relation in the table containing NTU relations for heat exchangers.

$$\text{NTU} = -\ln(1-\varepsilon)$$
$$= -\ln(1-0.6)$$
$$= 0.9163$$

Step 5 of 6

Using the value of NTU, the area of the heat exchanger is calculated.

$$\text{NTU} = \frac{UA_s}{C_{min}}$$
$$A_s = \frac{\text{NTU}\,C_{min}}{U}$$
$$= \frac{(0.9163)(2.2\times4.18)}{0.7}$$
$$= 12.037\ \text{m}^2$$

Step 6 of 6

The length of the tube is therefore,

$$A_s = \pi DL$$
$$L = \frac{A_s}{\pi D}$$
$$= \frac{12.037}{\pi\times0.025}$$
$$\boxed{L = 153.26\ \text{m}}$$

Hence, the length of the tube required in order to heat the water from 20 ℃ to 80 ℃ is $L = 153.26\ \text{m}$.

Question:

Ethanol is vaporized at 78°C (h_{fg} = 846 kJ/kg) in a double-pipe parallel-flow heat exchanger at a rate of 0.04 kg/s by hot oil (cp = 2200 J/kg·K) that enters at 115°C. If the heat transfer surface area and the overall heat transfer coefficients are 6.2 m2 and 320 W/m2·K, respectively, determine the outlet temperature and the mass flow rate of oil using (*a*) the LMTD method and (*b*) the ε-NTU method.

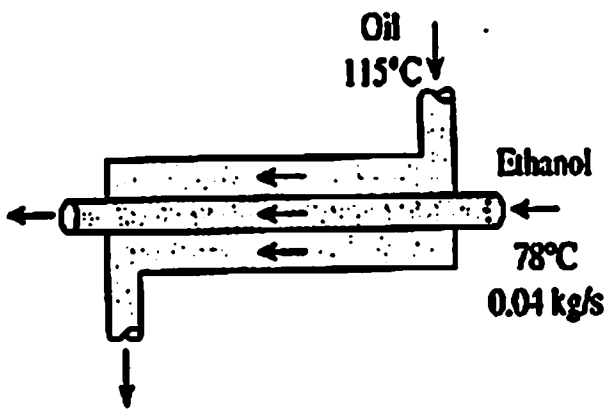


FIGURE P22–81

Step 1 of 5

Given

Specific heats of hot oil is $c_p = 2200\ \text{J/kg.°C}$

Mass flow rates of hot oil is $\dot{m}_{hot\,oil} = 0.04\ \text{kg/s}$

Temperature of Ethanol $T_{Ethanol} = 78°\text{C}$

Enthalpy of vaporization $h_{fg} = 846\ \text{kJ/kg}$

Inlet temperature of hot oil $T_{h,in} = 115°\text{C}$

Heat transfer surface area is $A_s = 6.2\ \text{m}^2$

Overall heat transfer coefficient $U = 320\ \text{W/m}^2.°\text{C}$

Step 2 of 5

a) Using LMTD method

Mass flow rate of the oil is determined as

$$\dot{Q} = \dot{m}_h C_p (T_{h,in} - T_{h,out})$$

$$\dot{m}_h = \frac{\dot{Q}}{C_p (T_{h,in} - T_{h,out})} \quad \text{..... (1)}$$

Where heat transfer rate and hot oil outlet temperature can be calculated as follows

The rate of heat transfer is

$$\dot{Q} = \left(\dot{m} h_{fg}\right)_{Ethanol}$$

$$\dot{Q} = 0.04(846)$$

$$\dot{Q} = 33.84\ \text{kW}$$

Step 3 of 5

LMTD for this heat exchanger is

$$\dot{Q} = U A_s \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{\dot{Q}}{U A_s}$$

$$\Delta T_{lm} = \frac{33.84 \times 10^3}{320 \times 6.2}$$

$$\Delta T_{lm} = 17.056°\text{C}$$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= 115 - 78$$

$$\Delta T = 37°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$\Delta T_2 = T_{h,out} - 78$$

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$17.06 = \frac{37 - \left(T_{h,out} - 78\right)}{\ln\left(37 / \left(T_{h,out} - 78\right)\right)}$$

by solving the above equation

$$\boxed{T_{h,out} = 84°\text{C}}$$

By substituting the values in equation (1) the mass flow rate of hot oil becomes

$$\dot{m}_h = \frac{\dot{Q}}{C_p (T_{h,in} - T_{h,out})}$$

$$\dot{m}_h = \frac{33.84}{2.2(115 - 84)}$$

$$\boxed{\dot{m}_h = 0.496\ \text{kg/s}}$$

Step 4 of 5

b) Using NTU method

For the condensing and evaporating type of heat exchangers

$$C = C_{min} / C_{max}$$

$$C = 0$$

Effectiveness of the heat exchanger is

For the condensing and evaporating type of heat exchangers

$$C = C_{min} / C_{max}$$

$$C = 0$$

Effectiveness of the heat exchanger is

$$\varepsilon = \frac{Q}{Q_{max}}$$

$$\varepsilon = \frac{C_{min}(T_{h,in} - T_{c,in})}{C_{min}(T_{h,in} - T_{c,in})}$$

$$\varepsilon = \frac{115 - T_{h,out}}{115 - 78}$$

But in case of condenser or evaporator type of heat exchangers effectiveness is

$$\varepsilon = 1 - e^{-NTU}$$

Therefore

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{-NTU} \quad \text{..... (2)}$$

Where NTU is

$$NTU = \frac{UA_s}{C_{min}}$$

$$= \frac{320 \times 6.2}{\dot{m}_h \times 2200}$$

$$NTU = \frac{0.9018}{\dot{m}_h}$$

Then substituting NTU in equation (2)

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{-NTU}$$

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{\frac{0.9018}{\dot{m}_h}} \quad \text{..... (3)}$$

Step 5 of 5

And heat transfer rate is

$$\dot{Q} = C_h(T_{h,in} - T_{h,out})$$

$$\dot{Q} = \dot{m}_h \times 2200(115 - T_{h,out})$$

$$33840 = \dot{m}_h \times 2200(115 - T_{h,out})$$

$$\dot{m}_h = \frac{33480}{2200(115 - T_{h,out})}$$

$$\dot{m}_h = \frac{15.218}{(115 - T_{h,out})}$$

By substituting $\dot{m}_h$ in equation (3)

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{\frac{-0.9018}{\dot{m}_h}}$$

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{\frac{-0.9018}{\frac{15.218}{115 - T_{h,out}}}}$$

$$\frac{115 - T_{h,out}}{115 - 78} = 1 - e^{-0.0592(115 - T_{h,out})}$$

By solving the above expression

$$\boxed{T_{h,out} = 83.8°C}$$

And the mass flow rate can be obtained by using the above relation

$$\dot{m}_h = \frac{15.218}{(115 - T_{h,out})}$$

$$= \frac{15.218}{115 - 83.8}$$

$$\boxed{\dot{m}_h = 0.487 \text{ kg/s}}$$

Question: 22.082P

Air (*cp* = 1005 J/kg-K) enters a cross-flow heat exchanger at 20°C at a rate of 3 kg/s, where it is heated by a hot water stream (*cp* = 4190 J/kg·K) that enters the heal exchanger at 70°C at a rate of 1 kg/s. Determine the maximum heat transfer rate and the outlet temperatures of both fluids for that case.

Step 1 of 5

Given data:

Specific heat of air, $c_{pc} = 1005\ \text{J/kg} \cdot {}^\circ\text{C}$

Specific heat of water, $c_{ph} = 4190\ \text{J/kg} \cdot {}^\circ\text{C}$

Mass flow rate of water, $\dot{m}_h = 1\ \text{kg/s}$

Mass flow rate of air, $\dot{m}_c = 3\ \text{kg/s}$

Inlet temperature of the air, $T_{c,in} = 20°\text{C}$

Inlet temperature of the water, $T_{h,in} = 70°\text{C}$

Step 2 of 5

Maximum possible heat transfer rate is determined as

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

Where C_{min} is calculated as follows

The heat capacity rates of hot and cold fluids are calculated as

$$C_h = \dot{m}_h C_{ph}$$
$$= (1\ \text{kg/s})(4190\ \text{J/kg} \cdot °\text{C})$$
$$= 4190\ \text{W/°C}$$
$$C_c = \dot{m}_c C_{pc}$$
$$= (3\ \text{kg/s})(1005\ \text{J/kg} \cdot °\text{C})$$
$$= 3015\ \text{W/°C}$$
$$\therefore C_{min} = C_c = 3015\ \text{W/°C}$$

Step 3 of 5

By substituting the value of C_{min} heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (3015\ \text{W/°C})(70°\text{C} - 20°\text{C})$$
$$= 150750\ \text{W}$$
$$= 150.75\text{kW}$$

$\therefore$ Maximum heat transfer rate is $\boxed{\dot{Q}_{max} = 150.75\text{kW}}$

Step 4 of 5

Then the outlet temperature of the air calculated by the relation

$$\dot{Q} = \left[C_c\left(T_{out} - T_{in}\right)\right]_{air}$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}_{max}}{C_c}$$
$$= 20°\text{C} + \frac{150750\ \text{W}}{3015\ \text{W/°C}}$$
$$= 70°\text{C}$$

$\therefore$ The outlet temperature of the air is $\boxed{T_{c,out} = 70°\text{C}}$

Step 5 of 5

Then the outlet temperature of the water calculated by the relation

$$\dot{Q} = \left[C_h\left(\left(T_{out} - T_{in}\right)\right]_{water}\right.$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}_{max}}{C_h}$$
$$= 70°\text{C} - \frac{150750\ \text{W}}{4190\ \text{W/°C}}$$
$$= 34°\text{C}$$

$\therefore$ The outlet temperature of the water is $\boxed{T_{h,out} = 34°\text{C}}$

Question: 22.083P

Hot oil (cp = 2200 J/kg·K) is to be cooled by water (cp = 4180 J/kg·K) in a 2-shell-passes and 12-tube-passes heat exchanger. The tubes are thin-walled and are made of copper with a diameter of 1.8 cm. The length of each tube pass in the heat exchanger is 3 m, and the overall heat transfer coefficient is 340 W/m2-K. Water flows through the tubes at a total rate of 0.1 kg/s, and the oil through the shell at a rate of 0.2 kg/s. The water and the oil enter at temperatures 18°C and 160°C, respectively. Determine the rate of heat transfer in the heat exchanger and the outlet temperatures of the water and the oil.

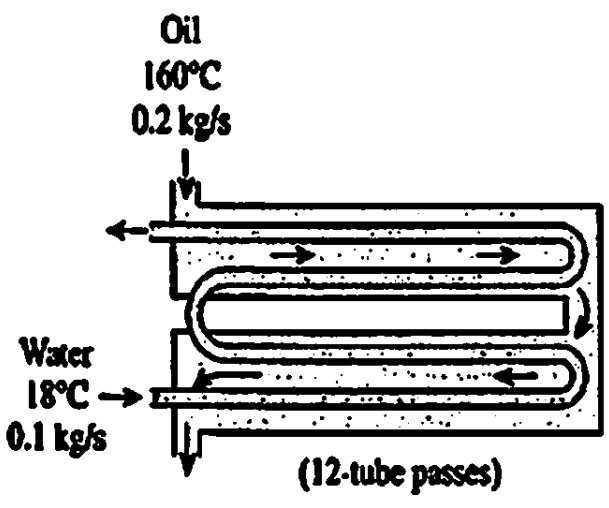


FIGURE P22–83

Step 1 of 11

Given

Type: 2shell passes and 12 tube passes

Specific heats of hot oil $c_{ph} = 2200\ \text{J/kg}^\circ\text{C}$

Specific heats of water $c_{pc} = 4810\ \text{J/kg}^\circ\text{C}$

Diameter of thin walled tubes: $D = 1.8\text{cm}$

$D = 0.018\ \text{m}$

Length of thin walled tubes $L = 3\ \text{m}$

Overall heat transfer coefficient $U = 340\ \text{W/m}^2{}^\circ\text{C}$

Mass flow rates of hot oil $\dot{m}_h = 0.2\ \text{kg/s}$

Mass flow rates of water $\dot{m}_c = 0.1\ \text{kg/s}$

Water inlet temperature, $T_{c,in} = 18^\circ\text{C}$

Hot oil inlet temperature $T_{h,in} = 160^\circ\text{C}$

Step 2 of 11

Heat capacity rate of the hot fluid:

$$C_h = \dot{m}_h c_{ph}$$
$$= (0.2\ \text{kg/s})(2200\ \text{J/kg}^\circ\text{C})$$
$$C_h = 440\ \text{W/}^\circ\text{C}$$

Step 3 of 11

Heat capacity rate of the cold fluid:

$$C_c = \dot{m}_c c_{pc}$$
$$= (0.1\ \text{kg/s})(4180\ \text{J/kg}^\circ\text{C})$$
$$C_c = 418\ \text{W/}^\circ\text{C}$$

Step 4 of 11

$$C_c(418) < C_h(440)$$
$$\therefore C_{min} = C_c$$
$$C_{min} = 418\ \text{W/}^\circ\text{C}$$

Step 5 of 11

Ratio of capacity rates:

$$C = \frac{C_{min}}{C_{max}}$$
$$= \frac{418\ \text{W/}^\circ\text{C}}{440\ \text{W/}^\circ\text{C}}$$
$$C = 0.95$$

Step 6 of 11

Fluid with smaller value of capacity rate experience the greater change in temperature so that the maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (418\ \text{W/}^\circ\text{C})(160\text{-}18)^\circ\text{C}$$
$$\dot{Q}_{max} = 59.356\ \text{kW}$$

Step 7 of 11

Heat exchange surface area is

$$A = n\pi DL$$
$$= (12)(\pi)(.018\ \text{m})(3\ \text{m})$$
$$A = 2.0357\ \text{m}^2$$

Step 8 of 11

NTU of this heat exchanger:

$$NTU = \frac{UA_i}{C_{min}}$$
$$= \frac{(340\ \text{W/m}^2{}^\circ\text{C})(2.0357\ \text{m}^2)}{418\ \text{W/}^\circ\text{C}}$$
$$NTU = 1.656$$

Step 9 of 11

Corresponding to $C = 0.95$ and NTU=1.656 Effectiveness $\varepsilon = 0.61$ (from Fig. (d) Effectiveness of the heat exchanger)

Actual heat transfer rate:

$$\dot{Q}_{act} = \varepsilon \dot{Q}_{max}$$
$$= (0.61)(59.356)$$
$$\boxed{\dot{Q}_{act} = 36.2\ \text{kW}}$$

Step 10 of 11

Outlet temperatures of the cold fluid stream:

$$\dot{Q}_{act} = C_c(T_{c,out} - T_{c,in})$$
$$T_{c,out} = T_{c,in} + \frac{\dot{Q}_{act}}{C_c}$$
$$= 18 + \frac{36200}{418}$$
$$\boxed{\therefore T_{c,out} = 104.602^\circ\text{C}}$$

Step 11 of 11

Outlet temperatures of the hot fluid stream:

$$\dot{Q}_{act} = C_h(T_{h,in} - T_{h,out})$$
$$T_{h,out} = T_{h,in} - \frac{\dot{Q}_{act}}{C_c}$$
$$= 160 - \frac{36200}{440}$$
$$\boxed{\therefore T_{h,out} = 77.727^\circ\text{C}}$$

Question: 22.084P

Consider an oil-to-oil double-pipe heat exchanger whose flow arrangement is not known. The temperature measurements indicate that the cold oil enters at 15°C and leaves at 55°C, while the hot oil enters at 80°C and leaves at 40°C. Do you think this is a parallel-flow or counter-flow heat exchanger? Why? Assuming the mass flow rates of both fluids to be identical, determine the effectiveness of this heat exchanger.

Step 1 of 2

Given data:

Inlet temperature of cold oil $T_{c,in} = 15°C$

Exit temperature of cold oil $T_{c,out} = 55°C$

Inlet temperature of hot oil $T_{h,in} = 80°C$

Exit temperature of hot oil $T_{h,out} = 40°C$

Step 2 of 2

In case of the parallel-flow heat exchangers the outlet temperature of the cold fluid $(T_{c,out})$ cannot exceed the outlet temperature of the hot fluid $(T_{h,out})$

But in this present problem $(T_{c,out} = 55°C) > (T_{h,out} = 40°C)$

Therefore the given heat exchanger is counter - flow heat exchanger.

Effectiveness of the heat exchanger is determined as follows

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$

$$= \frac{C_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,in})}$$

But mass flow rates are same $C_h = C_c = C_{min}$

$$\varepsilon = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})}$$

$$= \frac{80-40}{80-15}$$

$$\boxed{\varepsilon = 0.6154}$$

Question: 22.085P

Hot water enters a double-pipe counter-flow water- to-oil heat exchanger at 190°F and leaves at 100°F. Oil enters at 70°F and leaves at 130°F. Determine which fluid has the smaller heat capacity rate and calculate the effectiveness of this heat exchanger.

Step 1 of 2

Given data:

Cold oil inlet temperature, $T_{c,in} = 70°F$

Cold oil outlet temperature, $T_{c,out} = 130°F$

Hot water inlet temperature, $T_{h,in} = 190°F$

Hot water outlet temperature, $T_{h,out} = 100°F$

Step 2 of 2

Hot water has the smaller heat capacity rate since it experiences a more temperature difference when compared to cold water.

Effectiveness of the heat exchanger is determined as follows

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$

$$= \frac{C_h\left(T_{h,in} - T_{h,out}\right)}{C_{min}\left(T_{h,in} - T_{c,in}\right)}$$

But mass flow rates are same $\therefore C_h = C_{min}$

$$\varepsilon = \frac{C_h\left(T_{h,in} - T_{h,out}\right)}{C_h\left(T_{h,in} - T_{c,in}\right)}$$

$$= \frac{190°F - 100°F}{190°F - 70°F}$$

$$= 0.75$$

$\therefore$ The effectiveness of the heat exchanger is $\boxed{\varepsilon = 0.75}$

Question: 22.086P

Saturated water vapor at 100°C condenses in a 1 -shell and 2-tube heat exchanger with a surface area of 0.5 m2 and an overall heat transfer coefficient of 2000 W/m2-K. Cold water (*cpc* = 4179 J/kg·K) flowing at 0.5 kg/s enters the tube side at 15°C, determine (*a*) the heat transfer effectiveness, (*b*) the outlet temperature of the cold water, and (*c*) the heat transfer rate for the heat exchanger.

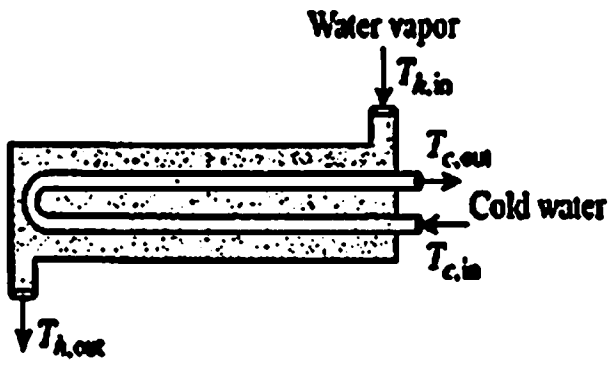


FIGURE P22–86

Step 1 of 5

Given data:

Temperature of saturated water $T_{h,in} = 100°C$

Mass flow rates of water is $\dot{m}_{water} = 0.5 kg/s$

Heat transfer surface area is $A_s = 0.5 m^2$

Overall heat transfer coefficient $U = 2000 W/m^2\text{-}K$

Temperature of cold water $T_{c,in} = 15°C$

Step 2 of 5

Heat capacity rate of the cold water is

$$C_c = \dot{m} c_{pc}$$

$$C_c = 0.5 \times 4179$$

$$C_{min} = C_c = 2089.5 \text{ W/K}$$

And $C = 0$

The maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$

$$= (2.0895 \text{ kW/°C})(100°C - 15°C)$$

$$\dot{Q}_{max} = 177.6 \text{ kW}$$

NTU of the heat exchanger is

$$NTU = \frac{UA_s}{C_{min}}$$

$$= \frac{(2 kW/m^2.°C)(0.5 m^2)}{2.0895 \text{ kW/°C}}$$

$$NTU = 0.478$$

Step 3 of 5

(a) Effectiveness of this heat exchanger

$$\varepsilon = 1 - \exp(-NTU)$$

$$= 1 - \exp(-0.478)$$

$$\boxed{\varepsilon = 0.38}$$

Step 4 of 5

(b) The outlet temperature of cold water

The actual heat transfer rate calculated as

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$

$$\dot{m} c_p (T_{c,out} - T_{c,in}) = \varepsilon \dot{Q}_{max}$$

$$0.5(4.179)(T_{c,out} - 15) = 0.38(177.6)$$

$$\boxed{T_{c,out} = 47.3°C}$$

Step 5 of 5

(c) The actual heat transfer rate calculated as

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$

$$\dot{Q} = 0.38(177.6)$$

$$\boxed{\dot{Q} = 67.49 kW}$$

Question:

Hot water (*cph* = 4188 J/kg·K) with mass flow rate of 2.5 kg/s at 100°C enters a thin-walled concentric tube counter- flow heat exchanger with a surface area of 23 m2 and an overall heat transfer coefficient of 1000 W/m2-K. Cold water (*cpc* =4178 J/kg·K) with mass flow rate of 5 kg/s enters the heat exchanger at 20°C, determine (*a*) the heat transfer rate for the heat exchanger and (*b*) the outlet temperatures of the cold and hot fluids. After a period of operation, the overall heat transfer coefficient is reduced to 500 W/m2-K, determine (*c*) the fouling factor that caused the reduction in the overall heat transfer coefficient.

Step 1 of 7

(a)

Heat transfer rate for the heat exchanger:

Calculate the ratio of the heat capacities.

$$c = \frac{C_{min}}{C_{max}} = \frac{(\dot{m}c_p)_c}{(\dot{m}c_p)_h}$$

Here, $\dot{m}_c$ is the mass flow rate of cold fluid, $\dot{m}_h$ is the mass flow rate of hot fluid, $c_{p,c}$ is the specific heat of cold fluid, and $c_{p,h}$ is the specific heat of hot fluid.

Substitute $2.5\,\text{kg/hr}$ for $\dot{m}_h$. $5\ \text{kg/s}$ for $\dot{m}_c$. $4{,}178\ \text{J/kg·K}$ for $c_{p,c}$, and $4{,}188\ \text{J/kg·K}$ for $c_{p,h}$.

$$c = \frac{2.5\times4{,}188}{5\times4{,}178} = 0.501$$

Calculate the smaller of the two heat capacities.

$$C_{min} = C_h = (\dot{m}c_p)_h = 2.5\times4{,}188 = 10{,}470\ \text{W/K}$$

Step 2 of 7

Calculate the NTU of heat exchanger.

$$\text{NTU} = \frac{UA_s}{C_{min}}$$

Here, U is the overall heat transfer coefficient of the counter flow heat exchanger and A_s is the surface area of the heat exchanger.

Substitute $1{,}000\ \text{W/m}^2\text{·K}$ for U . $23\ \text{m}^2$ for A_s , and $10{,}470\ \text{W/K}$ for C_{min} .

$$\text{NTU} = \frac{1{,}000\times23}{10{,}470} = 2.196$$

Determine, from the graph containing effectiveness and NTU, at $c = 0.501$ and $\text{NTU} = 2.196$ the value of effectiveness is $\varepsilon = 0.78$.

Step 3 of 7

Calculate the heat transfer rate in terms of hot fluid temperatures.

$$\dot{Q} = C_{min}\left(T_{h,i} - T_{h,o}\right)$$

Here, $T_{h,o}$ is the exit temperature of hot fluid.

Substitute $10{,}470\ \text{W/K}$ for C_{min} , 100 for $T_{h,i}$, and 20 for $T_{c,i}$.

$$\dot{Q}_{max} = 10{,}470\times(100-20) = 10{,}470\times80 = 837{,}600\ \text{W}$$

Calculate the actual heat transfer rate between the two fluids.

$$\dot{Q} = \varepsilon\dot{Q}_{max} = 0.78\times837{,}600 = 653{,}328\ \text{W}$$

Therefore, the actual heat transfer rate between the two fluids is $\boxed{653{,}328\ \text{W}}$.

Step 4 of 7

(b)

The exit temperatures of the cold and hot fluids are determined as follows.

Consider the expression for the actual heat transfer rate in terms of cold fluid temperatures.

$$\dot{Q} = C_c\left(T_{c,o} - T_{c,i}\right) = (\dot{m}c_p)_c\left(T_{c,o} - T_{c,i}\right)$$

Substitute $5\ \text{kg/s}$ for $\dot{m}_c$. $4{,}178\ \text{J/kg·K}$ for $c_{p,c}$.$(653{,}328\ \text{W})$ for $\dot{Q}$, and $20°\text{C}$ for $T_{c,i}$.

$$653{,}328 = (5\times4{,}178)\left(T_{c,o} - 20\right)$$

$$T_{c,o} = \frac{653{,}328}{20{,}890} + 20 = 31.27°\text{C}$$

Therefore, the exit temperatures of the cold is $\boxed{31.27°\text{C}}$.

Step 5 of 7

Calculate the actual heat transfer rate in terms of hot fluid temperatures.

$$\dot{Q} = C_h\left(T_{h,i} - T_{h,o}\right)$$

Step 5 of 7

Calculate the actual heat transfer rate in terms of hot fluid temperatures.

$$\dot{Q} = C_h\left(T_{h,i} - T_{h,o}\right)$$
$$= \left(\dot{m}c_p\right)_h\left(T_{h,i} - T_{h,o}\right)$$

Substitute $2.5\,\text{kg/hr}$ for $\dot{m}_h$, $4{,}188\ \text{J/kg}\cdot\text{K}$ for $c_{p,h}$, $(653{,}328\ \text{W})$ for $\dot{Q}$, and $100°\text{C}$ for $T_{h,i}$.

$$653{,}328 = (2.5\times 4{,}188)(100 - T_{h,o})$$
$$T_{h,o} = 100 - \frac{653{,}328}{10{,}470}$$
$$= 37.6°\text{C}$$

Therefore, the exit temperature of hot fluid is $\boxed{37.6°\text{C}}$.

Step 6 of 7

(c)

Determination of fouling factor F :

The overall heat transfer coefficient of the heat exchanger falls to $U' = 500\ \text{W/m}^2\cdot\text{K}$.

$$\text{NTU}' = \frac{U'A_s}{C_{\min}}$$
$$= \frac{500\times 23}{10{,}470}$$
$$\text{NTU}' = 1.098$$

From the graph containing effectiveness and NTU, at $c = 0.501$ and $\text{NTU}' = 1.098$ the value of effectiveness is determined to be $\varepsilon' = 0.57$.

Calculate the maximum heat transfer rate using the relation,

$$\dot{Q}'_{\max} = C_{\min}\left(T_{h,i} - T_{c,i}\right)$$

Substitute 10470 for $C_{\min}$, $100°\text{C}$ for $T_{h,i}$, and $20°\text{C}$ for $T_{c,i}$.

$$\dot{Q}'_{\max} = 10{,}470\times(100 - 20)$$
$$= 10{,}470\times 80$$
$$= 837{,}600\ \text{W}$$

Step 7 of 7

Calculate the actual heat transfer rate between the two fluids.

$$\dot{Q}' = \varepsilon'\dot{Q}'_{\max}$$
$$= 0.57\times 837{,}600$$
$$= 477{,}432\ \text{W}$$

Calculate the Fouling factor which refers to the ratio of the heat transfer rate due to fouling, to the actual rate of heat transfer.

$$F = \frac{\dot{Q}'}{\dot{Q}}$$
$$= \frac{477{,}432}{653{,}328}$$
$$= 0.73$$

Therefore, the fouling factor that caused reduction in the overall heat transfer coefficient is found to be $\boxed{0.73}$.

Question: 22.088P

A cross-flow air-to-water heat exchanger with an effectiveness of 0.65 is used to heat water (cp = 4180 J/kg·K) with hot air (cp = 1010 J/kg-K). Water enters the heat exchanger at 20°C at a rate of 4 kg/s, while air enters at 100°C at a rate of 9 kg/s. If the overall heat transfer coefficient based on the water side is 260 W/m2-K, determine the heat transfer surface area of the heat exchanger on the water side. Assume both fluids are unmixed.

Step 1 of 4

Given data:

Effectiveness of the cross-flow heat exchanger, $\varepsilon = 0.65$

Specific heat of hot air, $c_{ph} = 1010\ \text{J/kg}\cdot\text{°C}$

Specific heat of water, $c_{pc} = 4180\ \text{J/kg}\cdot\text{°C}$

Mass flow rate of hot air, $\dot{m}_h = 9\ \text{kg/s}$

Mass flow rate water, $\dot{m}_c = 4\ \text{kg/s}$

Overall heat transfer coefficient $U = 260\ \text{W/m}^2\cdot\text{°C}$

Inlet temperature of the water, $T_{c,in} = 20\text{°C}$

Inlet temperature of the hot air, $T_{h,in} = 100\text{°C}$

Step 2 of 4

Heat capacity rate of the hot and cold fluids are

$$C_c = \dot{m}_c c_{pc}$$
$$= (4\ \text{kg/s})(4180\ \text{J/kg}\cdot\text{°C})$$
$$= 16720\ \text{W/°C}$$
$$C_h = \dot{m}_h c_{ph}$$
$$= (9\ \text{kg/s})(1010\ \text{J/kg}\cdot\text{°C})$$
$$= 9090\ \text{W/°C}$$
$$C_{min} = c_h$$
$$\therefore C_{min} = 9090\ \text{W/°C}$$

Step 3 of 4

Ratio of capacity rates become

$$C = \frac{C_{min}}{C_{max}}$$
$$= \frac{9090\ \text{W/°C}}{16720\ \text{W/°C}}$$
$$= 0.544$$

Step 4 of 4

Then the NTU of this heat exchanger corresponding to $C = 0.544$ and $\varepsilon = 0.65$ is determined from the effectiveness for heat exchangers of cross-flow with both fluids unmixed is $NTU = 1.5$

Then the surface area of this heat exchanger becomes

$$NTU = \frac{UA_s}{C_{min}}$$
$$A_s = \frac{NTU\ C_{min}}{U}$$
$$= \frac{(1.5)(9090\ \text{W/°C})}{260\ \text{W/m}^2\cdot\text{°C}}$$
$$= 52.4\ \text{m}^2$$

$\therefore$ The heat transfer surface are of the heat exchanger is $\boxed{A_s = 52.4\ \text{m}^2}$

Question: 22.089P

Cold water (cp = 4.18 kJ/kg·K) enters a cross-flow heat exchanger at 14°C at a rate of 0.35 kg/s where it is heated by hot air (cp = 1.0 kJ/kg·K) that enters the heat exchanger at 65°C at a rate of 0.8 kg/s and leaves at 25°C. Determine the maximum outlet temperature of the cold water and the effectiveness of this heat exchanger.

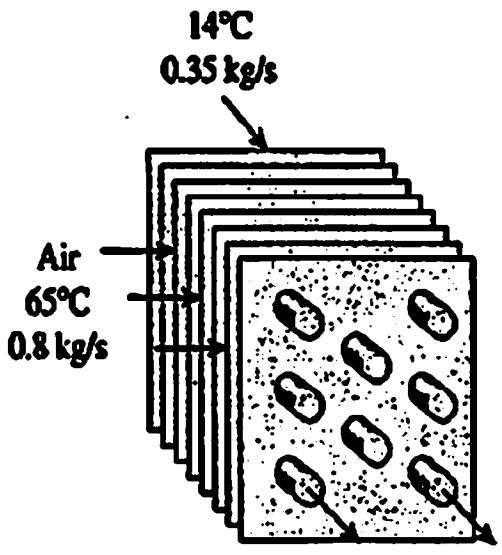


FIGURE P22–89

Step 1 of 5

Given

Type: cross flow heat exchanger

The specific heat of cold water, $c_{pc} = 4.180\ \text{kJ/kg}^\circ\text{C}$

Inlet temperature of coldwater $T_{c,in} = 14^\circ\text{C}$

Mass flow rate of cold water $\dot{m}_c = 0.35\ \text{kg/s}$

The specific heat of hot air, $c_{ph} = 1.0\ \text{kJ/kg}^\circ\text{C}$

Inlet temperature of hot air $T_{h,in} = 65^\circ\text{C}$

Outlet temperature of hot air $T_{h,out} = 25^\circ\text{C}$

Mass flow rate of hot air $\dot{m}_h = 0.8\ \text{kg/s}$

Step 2 of 5

Heat capacity rate of the hot fluid:

$$C_h = \dot{m}_h c_{ph}$$
$$= (0.8\ \text{kg/s})(1.0\ \text{kJ/kg}^\circ\text{C})$$
$$C_h = 0.8\ \text{kW/}^\circ\text{C}$$

Heat capacity rate of the cold fluid:

$$C_c = \dot{m}_c c_{pc}$$
$$= (0.35\ \text{kg/s})(4.18\ \text{kJ/kg}^\circ\text{C})$$
$$C_c = 1.463\ \text{kW/}^\circ\text{C}$$
$$C_h > C_c$$
$$\therefore C_{min} = C_c$$
$$C_{min} = 0.8\ \text{kW/}^\circ\text{C}$$

Step 3 of 5

The maximum heat transfer rate:

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (0.8)(65-14)$$
$$\dot{Q}_{max} = 40.8\ \text{kW}$$

Outlet temperature of the cold water:

$$\dot{Q}_{max} = C_c(T_{c,out} - T_{c,in})]$$
$$T_{c,out} = T_{c,in} + \frac{\dot{Q}_{max}}{C_c}$$
$$= 14 + \frac{40.8}{1.463}$$
$$\boxed{T_{c,out} = 41.89\ ^\circ\text{C}}$$

Step 4 of 5

Effectiveness of this heat exchanger:

$$\varepsilon = \frac{\dot{Q}_{act}}{\dot{Q}_{max}}$$

Actual rate of heat transfer is

$$\dot{Q}_{act} = [C_{ph}(T_{h,out} - T_{h,in})]$$
$$= (0.8\ \text{kW/}^\circ\text{C})(65\text{-}25)^\circ\text{C}$$
$$\dot{Q}_{act} = 32\ \text{kW}$$

Step 5 of 5

Effectiveness becomes $\varepsilon = \frac{32}{40.8}$

$$\boxed{\therefore\ \varepsilon = 0.78}$$

Question: 22.090P

Water from a lake is used as the cooling agent in a power plant. To achieve condensation of 3.2 kg/s of steam exiting the turbine, a shell-and-tube heat exchanger is used, which has a single shell and 220 thin-walled, 25-mm-diameter tubes, each tube making two passes. Steam flows through the shell, while cooling water flows through the tubes. Steam enters as saturated vapor at 60°C and leaves as saturated liquid. Cooling water at 20°C is available at a rate of 200 kg/s. The convection coefficient at the outer surface of the tubes is 8500 W/m2·K. Determine (*a*) the temperature of the cooling water leaving the condenser and (*b*) the required tube length per pass. (Use the following average properties for water: *cp* =4180 J/kg·K, μ = 8 × 10−4 N·s/m2, *k* = 0.6 W/m·K, Pr = 6).

Step 1 of 7

Given data:

Mass flow rates of water is $\dot{m} = 3.2\text{kg/s}$

Number of tubes $n = 220$

Diameter of the tube $D_i = 25\text{ mm}$

Steam enters as saturated vapor and leaves as saturated liquid at temperature $T_{steam} = 60°\text{C}$

Inlet temperature of cooling water $T_{c,in} = 20°\text{C}$

The convection coefficient at the outer surface is $h_o = 8500\text{ W/m}^2\text{-K}$

The properties of water are

Dynamic viscosity $\mu = 8\times10^{-4}\text{ N.s/m}^2$

Thermal conductivity $k = 0.6\text{ W/m.°C}$

Specific heat $c_p = 4180\text{J/kgK}$

Prandtl number $\text{Pr} = 6$

Enthalpy of vaporization at $T_{steam} = 60°\text{C}$ is $h_{fg} = 2359\text{kJ/kg}$

Step 2 of 7

(a) The temperature of the cooling water leaving the condenser is

The amount of heat transfer rate is

$$\dot{Q} = \dot{m}h_{fg}$$
$$= 3.2(2359)$$
$$= 7548.8\text{ kW}$$

And we know

$$\dot{Q} = \dot{m}_c c_c\left(T_{c,out} - T_{c,in}\right)$$
$$7548.8 = 200(4.18)\left(T_{c,out} - 20\right)$$
$$\boxed{T_{c,out} = 29.03°\text{C}}$$

Step 3 of 7

(b) The length of the tube

Reynolds number for the flow is expressed as

$$\text{Re} = \frac{4\dot{m}}{n\pi D\mu}$$
$$= \frac{4\times200}{220\times\pi\times(0.025)8\times10^{-4}}$$
$$\text{Re} = 57874.5$$

Re >10,000 the flow is turbulent and assuming fully developed flow.

Step 4 of 7

Nusselt number

$$\text{Nu} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4}$$
$$= 0.023(57874.5)^{0.8}(6)^{0.4}$$
$$\text{Nu} = 304.07$$

And

$$\text{Nu} = \frac{h_i D}{k}$$
$$h_i = \frac{k}{D}\text{Nu}$$
$$= \frac{0.6}{0.025}\times304.07$$
$$h_i = 7297.68\text{ W/m}^2\cdot°\text{C}$$

Step 5 of 7

Then the overall heat transfer coefficient is

$$U = \frac{1}{\frac{1}{h_i}+\frac{1}{h_o}}$$
$$= \frac{1}{\frac{1}{7297.68\text{ W/m}^2\cdot°\text{C}}+\frac{1}{8500\text{ W/m}^2\cdot°\text{C}}}$$
$$= 3926.54\text{ W/m}^2\cdot°\text{C}$$

Temperature differences in counter flow heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= 60 - 29.03$$
$$\Delta T_1 = 30.97°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= 60 - 20$$
$$\Delta T_2 = 40°\text{C}$$

Step 6 of 7

Therefore the LMTD is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)}$$
$$= \frac{30.97 - 40}{\ln[30.97/40]}$$
$$\Delta T_{lm} = 35.3°\text{C}$$

Step 7 of 7

Since the phase change on the shell side the correction factor for this case is

$$F = 1$$

Then the tube length per pass is determined to be

$$\dot{Q} = UA_s F\Delta T_{lm}$$
$$\dot{Q} = U(\pi DL)F\Delta T_{lm}$$
$$7548.8 = (3926.54)(\pi\times(0.025)L)(1)(35.3)$$
$$\boxed{L = 0.693\text{ m}}$$

Question: 22.091P

Water (cp = 4180 J/kg·K) is to be heated by solar- heated hot air (cp = 1010 J/kg·K) in a double-pipe counter- flow heat exchanger. Air enters the heat exchanger at 90°C at a rate of 0.3 kg/s, while water enters at 22°C at a rate of 0.1 kg/s. The overall heat transfer coefficient based on the inner side of the tube is given to be 80 W/m2·K. The length of the tube is 12 m and the internal diameter of the tube is 1.2 cm. Determine the outlet temperatures of the water and the air.

Step 1 of 8

Given data:

Specific heat of hot air $c_{ph} = 1010$ J/kg°C

Specific heat of water $c_{pc} = 4180$ J/kg°C

Mass flow rate of hot air, $\dot{m}_h = 0.3$ kg/s

Mass flow rate of water, $\dot{m}_c = 0.1$ kg/s

Overall heat transfer coefficient $U = 80$ W/m^2°C

The diameter of the tube, $D = 0.012$ m

The length of the tube, $L = 12$ m

Step 2 of 8

Outlet temperatures of the hot and cold fluids are determined as

Heat transfer rate of each fluid is given by

$$\dot{Q} = C_h\left(T_{h,in} - T_{h,out}\right)$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h}$$

The heat capacity rate of the hot fluid is

$$C_h = \dot{m}_h C_{ph}$$
$$= (0.3\text{kg/s})(1010\ \text{J/kg.°C})$$
$$= 303\ \text{W/°C}$$

Step 3 of 8

$$\dot{Q} = C_c\left(T_{c,out} - T_{c,in}\right)$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$

The heat capacity rate of the cold fluid is

$$C_c = \dot{m}_c C_{pc}$$
$$= (0.1\ \text{kg/s})(4180\ \text{J/kg.°C})$$
$$= 418\ \text{W/°C}$$

Step 4 of 8

Therefore $C_{min} = C_c = 303$ W/°C

And $C = \frac{C_{min}}{C_{max}} = \frac{303}{418} = 0.725$

The maximum heat transfer rate is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (303\ \text{W/°C})(90°\text{C} - 22°\text{C})$$
$$= 20,604\ \text{kW}$$

Step 5 of 8

Hear transfer surface area is

$$A_s = \pi DL$$
$$= (\pi)(0.012\ \text{m})(12\ \text{m})$$
$$= 0.45\ \text{m}^2$$

NTU for this heat exchanger is

$$NTU = \frac{UA_s}{C_{min}}$$
$$= \frac{(80\ \text{W/m}^2\text{.°C})\ (0.45\ \text{m}^2)}{303\ \text{W/°C}}$$
$$= 0.1188$$

Step 6 of 8

Effectiveness for this heat exchanger is calculated as

$$\varepsilon = \frac{1 - \exp[-NTU(1-C)]}{1 - C\exp[-NTU(1-C)]}$$
$$= \frac{1 - \exp[-0.1188(1-0.725)]}{1 - 0.725\exp[-0.1188(1-0.725)]}$$
$$= \frac{0.03214}{0.2983}$$
$$= 0.1077$$

Step 7 of 8

Then the actual heat transfer rate is

$$\dot{Q} = \varepsilon\dot{Q}_{max}$$
$$= (0.108)(20{,}604\ \text{W})$$
$$= 2219.93\ \text{W}$$

By substituting the heat transfer rate outlet temperatures of the hot and cold fluids are

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h}$$
$$= 90°\text{C} - \frac{2219.93\ \text{W}}{303\ \text{W/°C}}$$
$$= 82.67°\text{C}$$

∴ The outlet temperature of the water, $\boxed{T_{h,out} = 82.67°\text{C}}$

Step 8 of 8

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$
$$= 22°\text{C} + \frac{2219.93\ \text{W}}{418\ \text{W/°C}}$$
$$= 27.31°\text{C}$$

∴ The outlet temperature of the hot air, $\boxed{T_{c,out} = 27.31°\text{C}}$

Question: 22.092P

Reconsider Prob. 22-91. Using EES (or other) software, investigate the effects of the mass flow rate of water and the tube length on the outlet temperatures of water and air. Let the mass flow rate vary from 0.05 kg/s to 1.0 kg/s and the tube length from 5 m to 25 m. Plot the outlet temperatures of the water and the air as func-tions of the mass flow rate and the tube length, and discuss the results.

Step 1 of 4

Procedure:

Double click on the EES icon then click new command on menu bar.

Now new window will be appeared, then type equation in a new window.

Then Click table on menu bar, select the new parameter table.

And run the table for now values.

Then click ok.

Now again select the plot command on menu bar .then select the required variables from the list, then click okay.

Step 2 of 4

Equation window:

```
"GIVEN"
T_air_in=90 "[C]"
m_dot_air=0.3 "[kg/s]"
C_p_air=1.01 "[kJ/kg-C]"
T_w_in=22 "[C]"
m_dot_w=0.1
C_p_w=4.18 "[kJ/kg-C]"
U=0.080 "[kW/m^2-C]"
"L=12 [m]"
D=0.012 "[m]"

"ANALYSIS"
DELTAT_1=T_air_in-T_w_out
DELTAT_2=T_air_out-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
A=pi*D*L
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_air*C_p_air*(T_air_in-T_air_out)
Q_dot=m_dot_w*C_p_w*(T_w_out-T_w_in)
```

Step 3 of 4

Table of Temperature of water and air as a function of mass flow rate ::

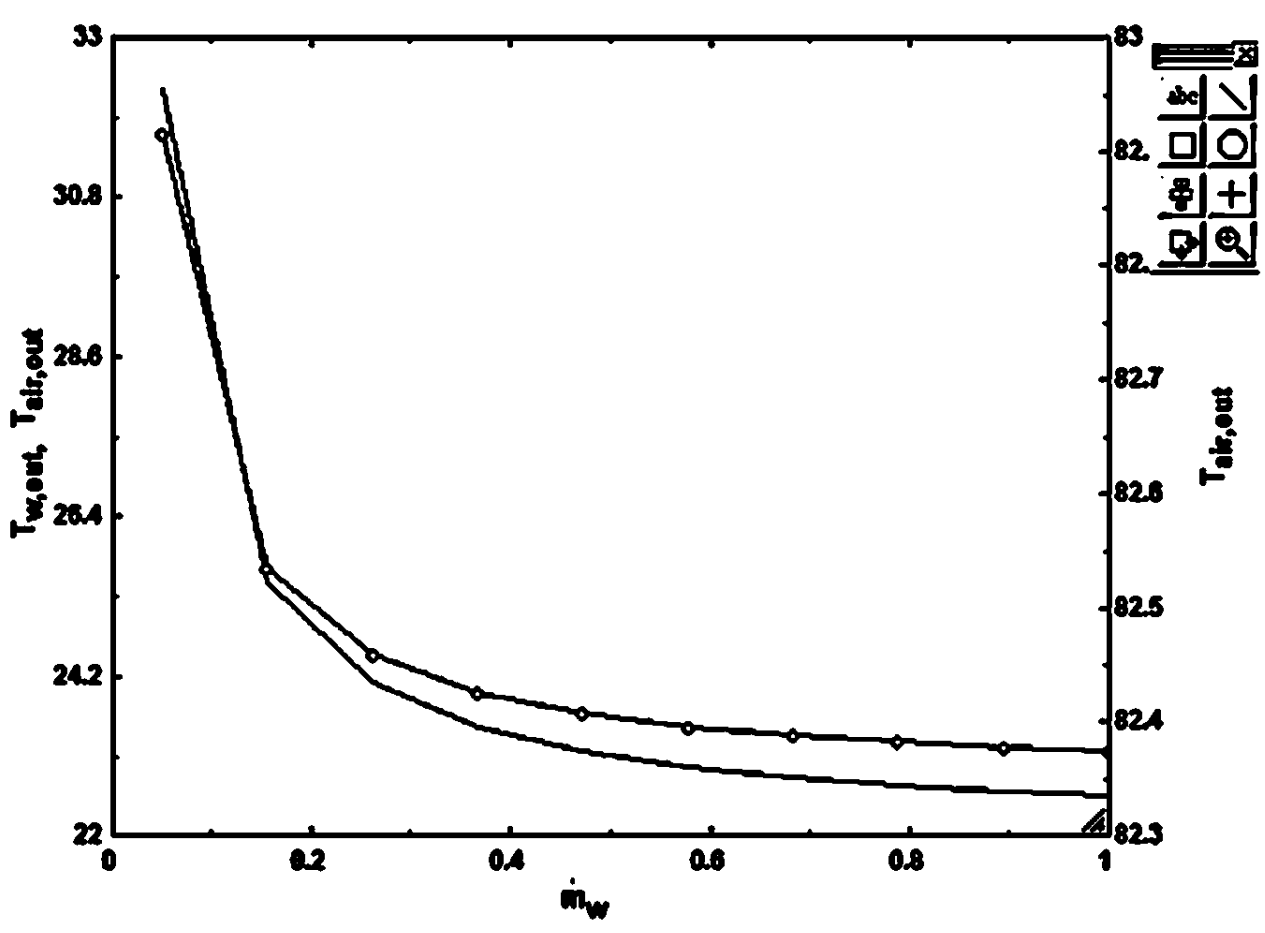
1..10	1 $\dot{m}_w$	2 $T_{w,out}$	3 $T_{air,out}$
Run 1	0.05	32.27	82.92
Run 2	0.1556	25.48	82.53
Run 3	0.2611	24.09	82.46
Run 4	0.3667	23.5	82.43
Run 5	0.4722	23.17	82.41
Run 6	0.5778	22.95	82.4
Run 7	0.6833	22.81	82.39
Run 8	0.7889	22.7	82.38
Run 9	0.8944	22.62	82.38
Run 10	1	22.55	82.37

Table of Temperature of water and air as a function of Length:

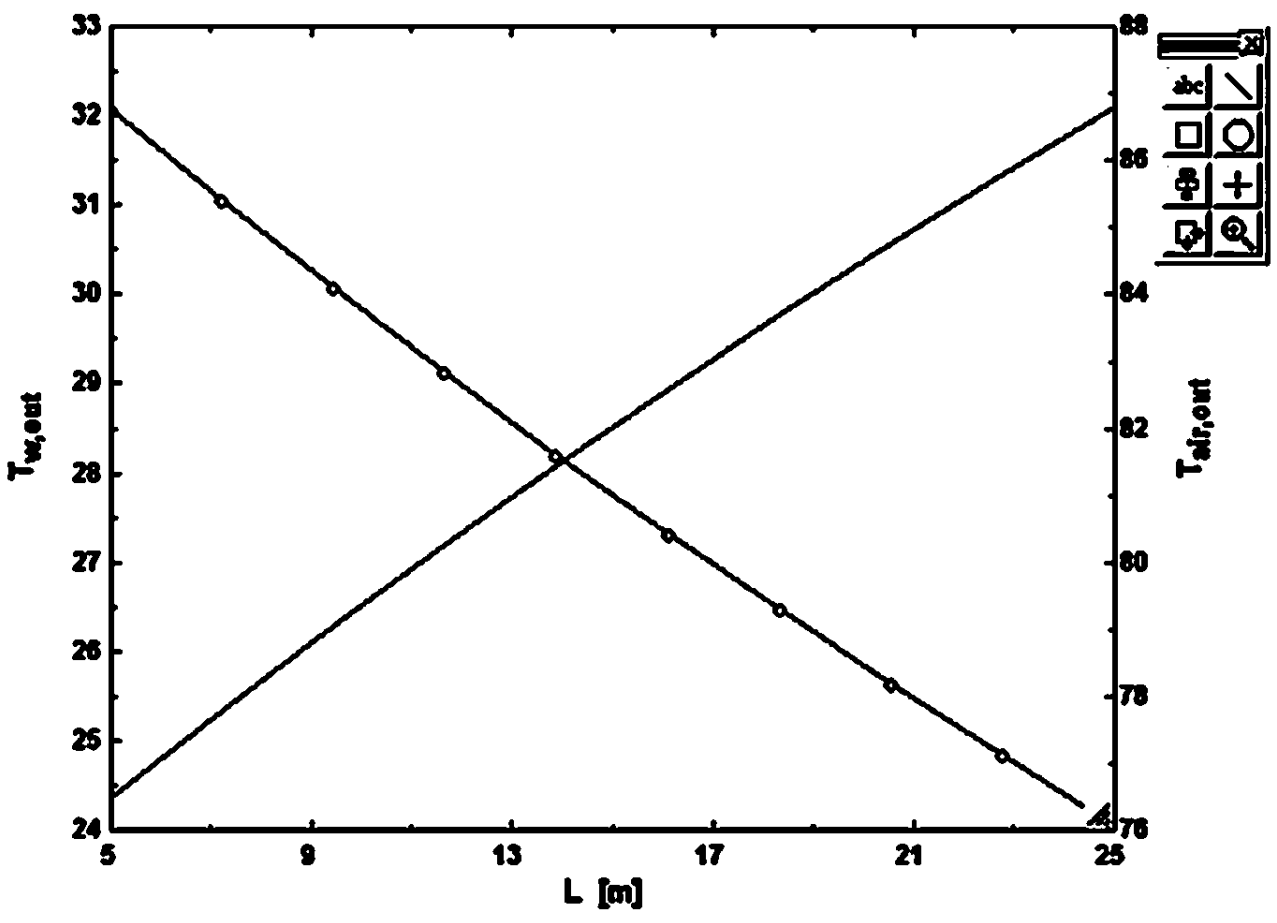
1..10	1 L [m]	2 $T_{air,out}$	3 $T_{w,out}$
Run 1	5	86.76	24.35
Run 2	7.222	85.4	25.34
Run 3	9.444	84.09	26.29
Run 4	11.67	82.82	27.2
Run 5	13.89	81.6	28.09
Run 6	16.11	80.42	28.94
Run 7	18.33	79.28	29.77
Run 8	20.56	78.18	30.57
Run 9	22.78	77.11	31.35
Run 10	25	76.07	32.1

Step 4 of 4

Graph between Temperature of water and air out Vs. mass flow rate :



Graph between Temperature of water and air out Vs. Length:



Question: 22.093P

A thin-walled double-pipe, counter-flow heat exchanger is to be used to cool oil (cp = 0.525 Btu/lbm·°F) from 300°F to 105°F at a rate of 5 lbm/s by water (cp = 1.0 Btu/lbm·°F) that enters at 70°F at a rate of 3 lbm/s. The diameter of the tube is 5 in and its length is 200 ft. Determine the overall heat transfer coefficient of this heat exchanger using (*a*) the LMTD method and (*b*) the ε-NTU method.

Step 1 of 13

Given

Specific heat of water, $c_{pc} = 1$ Btu/lbm °F

Specific heat of oil, $c_{ph} = 0.525$ Btu/lbm °F

Mass flow rate of oil $m_h = 5$ lbm/s

Mass flow rate of water, $m_c = 3$ lbm/s

The diameter of the tube is $D = 5$ in

The total length of the tube is $L = 200$ ft

Step 2 of 13

a) Using LMTD method overall heat transfer coefficient is determined as follows

Heat transfer rate determined to be

$$\dot{Q} = [\dot{m}_h c_{hc}(T_{out} - T_{in})]_{oil}$$

$$= (5 \text{ lbm/s})(0.525 \text{ Btu/lbm °F})(300-105)°\text{F}$$

$$= 511.9 \text{ Btu/s}$$

Step 3 of 13

The mass flow rate of glycerin is determined to be

$$\dot{Q} = [\dot{m} C_p (T_{out} - T_{in})]_{water}$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$

$$= 70°\text{F} + \frac{511.9 \text{ Btu/s}}{(3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})}$$

$$= 240.6°\text{F}$$

Step 4 of 13

For this counter flow heat exchanger LMTD determined to be

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= (300-240.6)°\text{F}$$

$$= 59.4°\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= (105-70)°\text{F}$$

$$= 35°\text{F}$$

Step 5 of 13

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{59.4-35}{\ln(59.4/35)}$$

$$= 46.1°\text{F}$$

Step 6 of 13

Overall heat transfer coefficient of the heat exchanger is determined to be

$$U_i = \frac{\dot{Q}}{A_i \Delta T_{lm}}$$

$$= \frac{511.9 \text{ Btu/s}}{\pi(5/12 \text{ ft})(200 \text{ ft})(46.1°\text{F})}$$

$$= 0.0424 \text{Btu/s} \cdot \text{ft}^2 \cdot °\text{F}$$

∴ The Overall heat transfer coefficient $\boxed{U_i = 0.0424 \text{Btu/s} \cdot \text{ft}^2 \cdot °\text{F}}$

Step 7 of 13

b) Using NTU method overall heat transfer coefficient is determined as follows

Heat transfer rate of each fluid is given by

$$\dot{Q} = C_h (T_{h,in} - T_{h,out})$$

The heat capacity rate of the hot fluid is

$$C_h = \dot{m}_h C_{ph}$$

$$= (5 \text{ lbm/s})(0.525 \text{ Btu/lbm.°F})$$

$$= 2.625 \text{ Btu/s.°F}$$

Step 8 of 13

$$\dot{Q} = C_c (T_{c,out} - T_{c,in})$$

The heat capacity rate of the cold fluid is

$$C_c = \dot{m}_c C_{pc}$$

$$= (3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})$$

$$= 3.0 \text{ Btu/s.°F}$$

Step 9 of 13

Therefore $C_{min} = C_h = 2.625$ Btu/s.°F

And $C = \frac{C_{min}}{C_{max}} = \frac{2.625}{3.0} = 0.875$

The maximum heat transfer rate is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$

$$= (2.625 \text{ Btu/s.°F})(300°\text{F} - 70°\text{F})$$

$$= 603.75 \text{ Btu/s}$$

Step 10 of 13

Actual heat transfer is calculated by

$$\dot{Q} = C_h (T_{h,in} - T_{h,out})$$

$$= (2.625 \text{ Btu/s.°F})(300°\text{F-}105°\text{F})$$

$$= 511.9 \text{ Btu/s}$$

Step 11 of 13

Then effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$

$$= \frac{511.9}{603.75}$$

$$= 0.85$$

Step 12 of 13

NTU of the heat exchanger is

$$NTU = \frac{1}{C-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C - 1}\right)$$

$$= \frac{1}{0.875-1}\ln\left(\frac{0.85-1}{0.85 \times 0.875-1}\right)$$

$$= 4.28$$

Step 13 of 13

Surface area of the heat exchanger is

$$A_s = \pi DL$$

$$= \pi(5/12 \text{ ft})(200 \text{ ft})$$

$$= 261.799 \text{ ft}^2$$

Overall heat transfer coefficient becomes

$$NTU = \frac{UA_s}{C_{min}}$$

$$U = \frac{NTU \, C_{min}}{A_s}$$

$$= \frac{(4.28)(2.625 \text{ Btu/s.°F})}{261.799 \text{ ft}^2}$$

$$= 0.0429 \text{Btu/s} \cdot \text{ft}^2 \cdot °\text{F}$$

∴ The Overall heat transfer coefficient $\boxed{U_i = 0.0429 \text{Btu/s} \cdot \text{ft}^2 \cdot °\text{F}}$

Question: 22.094P

A shell-and-tube heat exchanger with 2-shell passes and 8-tube passes is used to heat ethyl alcohol (cp = 2670 J/kg·K) in the tubes from 25°C to 70°C at a rate of 2.1 kg/s. The heating is to be done by water (cp = 4190 J/kg·K) that enters the shell at 95°C and leaves at 60°C. If the overall heat transfer coefficient is 800 W/m2·K, determine the heat transfer surface area of the heat exchanger using (*a*) the LMTD method and (*b*) the ε-NTU method.

Step 1 of 13

Given data:

Specific heat of water, $c_{ph} = 4190\ \text{J/kg°C}$

Specific heat of ethyl alcohol, $c_{pc} = 2670\ \text{J/kg°C}$

Mass flow rates of ethyl alcohol is $\dot{m}_c = 2.1\ \text{kg/s}$

Overall heat transfer coefficient $U = 800\ \text{W/m}^2\text{°C}$

Step 2 of 13

a) Using LMTD method

Temperature difference between two fluids at two ends of the heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= 95°C - 70°C$$
$$= 25°C$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= 60°C - 25°C$$
$$= 35°C$$

Step 3 of 13

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$
$$= \frac{25-35}{\ln(25/35)}$$
$$= 29.7°C$$

Step 4 of 13

For shell and tube heat exchanger computing the parameters P & R

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$
$$= \frac{70-25}{95-25}$$
$$= 0.64$$
$$R = \frac{T_1 - T_2}{t_2 - t_1}$$
$$= \frac{95-60}{70-25}$$
$$= 0.78$$

Step 5 of 13

At these P and R values F= 0.93 (Fig. 16-18c)

Heat transfer rate determined to be

$$\dot{Q} = \left[\dot{m}_c c_{pc}(T_{out} - T_{in})\right]_{\text{ethyl alcohol}}$$
$$= (2.1\ \text{kg/s})(2.67\ \text{kJ/kg.°C})(70°C - 25°C)$$
$$= 252.3\ \text{kW}$$

Step 6 of 13

Heat transfer surface area is

$$\dot{Q} = UA_s\Delta T_{lm}$$
$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$
$$= \frac{252.3\ \text{kW}}{(0.8\ \text{kW/m}^2\cdot°C)(0.93)(29.7°C)}$$
$$= 11.4\ \text{m}^2$$

∴ The heat transfer surface area, $\boxed{A_s = 11.4\ \text{m}^2}$

Step 7 of 13

b) Using NTU method

Heat transfer rate determined to be

$$\dot{Q} = \left[\dot{m}_c c_{pc}(T_{out} - T_{in})\right]_{\text{ethyl alcohol}}$$
$$= (2.1\ \text{kg/s})(2.67\ \text{kJ/kg}\cdot°C)(70°C - 25°C)$$
$$= 252.3\ \text{kW}$$

Step 8 of 13

Mass flow rate of the hot fluid is

$$\dot{Q} = \dot{m}_h C_{ph}(T_{h,in} - T_{h,out})$$
$$\dot{m}_h = \frac{\dot{Q}}{C_{ph}(T_{h,in} - T_{h,out})}$$
$$= \frac{252.3\ \text{kW}}{(4.19\ \text{kJ/kg.°C})(95°C - 60°C)}$$
$$= 1.72\ \text{kg/s}$$

Step 9 of 13

The heat capacity rates of the hot and cold fluids are

$$C_c = \dot{m}_c C_{pc}$$
$$= (2.1\ \text{kg/s})(2.67\ \text{kJ/kg}\cdot°C)$$
$$= 5.61\ \text{kW/°C}$$
$$C_h = \dot{m}_h C_{ph}$$
$$= (1.72\ \text{kg/s})(4.19\ \text{kJ/kg}\cdot°C)$$
$$= 7.21\ \text{kW/°C}$$

Step 10 of 13

Therefore the minimum heat capacity

$$C_{min} = C_c = 5.61\ \text{kW/°C}$$

And $$C = \frac{C_{min}}{C_{max}}$$
$$= \frac{5.61}{7.21}$$
$$= 0.78$$

Step 11 of 13

Maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (5.61\ \text{W/°C})(95°C - 25°C)$$
$$= 392.7\ \text{kW}$$

Step 12 of 13

Effectiveness of the heat exchanger is

$$\varepsilon = \frac{Q}{Q_{max}}$$
$$= \frac{252.3}{392.7}$$
$$= 0.64$$

Step 13 of 13

Heat transfer surface area of the heat exchanger determined as

$$NTU = \frac{UA_s}{C_{min}}$$
$$A_s = \frac{NTU\ C_{min}}{U}$$
$$= \frac{(1.7)(5.61\ \text{kW/°C})}{0.8\ \text{kW/m}^2\text{.°C}}$$
$$= 11.9\ \text{m}^2$$

∴ The heat transfer surface area, $\boxed{A_s = 11.9\ \text{m}^2}$

Question: 22.095P

Steam is to be condensed on the shell side of a 1-shell- pass and 8-tube-passes condenser, with 50 tubes in each pass, at 30°C (*hfg* = 2431 kJ/kg). Cooling water (*cp* = 4180 J/kg·K) enters the tubes at 18°C at a rate of 2200 kg/h. The tubes are thin-walled, and have a diameter of 1.5 cm and length of 2 m per pass. If the overall heat transfer coefficient is 3000 W/m2·K, determine (*a*) the rate of heat transfer and (*b*) the rate of condensation of steam.

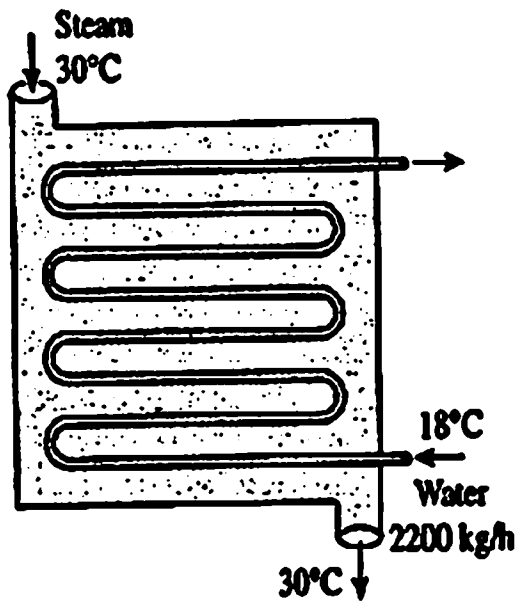


FIGURE P22–95

Step 1 of 4

Given data:

The specific heat of water is $c_p = 4180 \text{J/kg.°C}$

Inlet temperature of saturated vapor $T_{steam} = 30°\text{C}$

The heat of vaporization of steam is $h_{fg} = 2431 \text{kJ/kg}$

Mass flow rate of water

$\dot{m}_{water} = 2200 \text{ kg/h}$

$= \frac{2200}{3600}$

$\dot{m}_{water} = 0.611 \text{ kg/s}$

Overall heat transfer coefficient $U = 3000 \text{ W/m}^2\text{.°C}$

Step 2 of 4

The heat capacity rate for the condensing type heat exchanger is infinity

So

$C_{min} = C_c$

$= \dot{m}_c C_{pc}$

$= 0.611(4.18)$

$C_{min} = 2.55 \text{ kW/°C}$

And $C = 0$

The maximum heat transfer rate becomes

$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$

$= (2.55 \text{ kW/°C})(30°\text{C} - 18°\text{C})$

$\dot{Q}_{max} = 30.654 \text{ kW}$

Hat transfer surface area calculated as

$A_s = 8n\pi DL$

$= 8 \times 50\pi(0.015 \text{ m})(2 \text{ m})$

$A_s = 37.7 \text{ m}^2$

Step 3 of 4

NTU of the heat exchanger is

$\text{NTU} = \frac{UA_s}{C_{min}}$

$= \frac{(3\text{kW/m}^2\text{.°C})(37.7\text{m}^2)}{2.55 \text{ kW/°C}}$

$\text{NTU} = 44.353$

Effectiveness for this heat exchanger becomes

$\varepsilon = 1 - \exp(-\text{NTU})$

$= 1 - \exp(-44.353)$

$= 1$

Then the actual heat transfer rate calculated as

$\dot{Q} = \varepsilon \dot{Q}_{max}$

$= (1)(30.654 \text{ kW})$

$\boxed{\dot{Q} = 30.654 \text{ kW}}$

Step 4 of 4

b) Condensation steam rate is calculated as

$\dot{Q} = \dot{m} h_{fg}$

$\dot{m} = \frac{\dot{Q}}{h_{fg}}$

$= \frac{30.654 \text{ kJ/s}}{2431 \text{ kJ/kg}}$

$\boxed{\dot{m} = 0.0126 \text{kg/s}}$

Question: 22.096P pg 1

Reconsider Prob. 22-95.Using EES (or other) software, investigate the effects of the condensing steam temprature and the tube diameter on the rate of heat transfer and the rate of condensation of steam. Let the steam temprature vary from 20°C to 70°C the tube diameter from 1.0 cm to 2.0 cm. Plot the rate of heat transfer and the rate of condensation as functions of steam temperature and diameter, and discuss the results.

Step 1 of 8

The given problem can be solved using EES software by following the below steps.

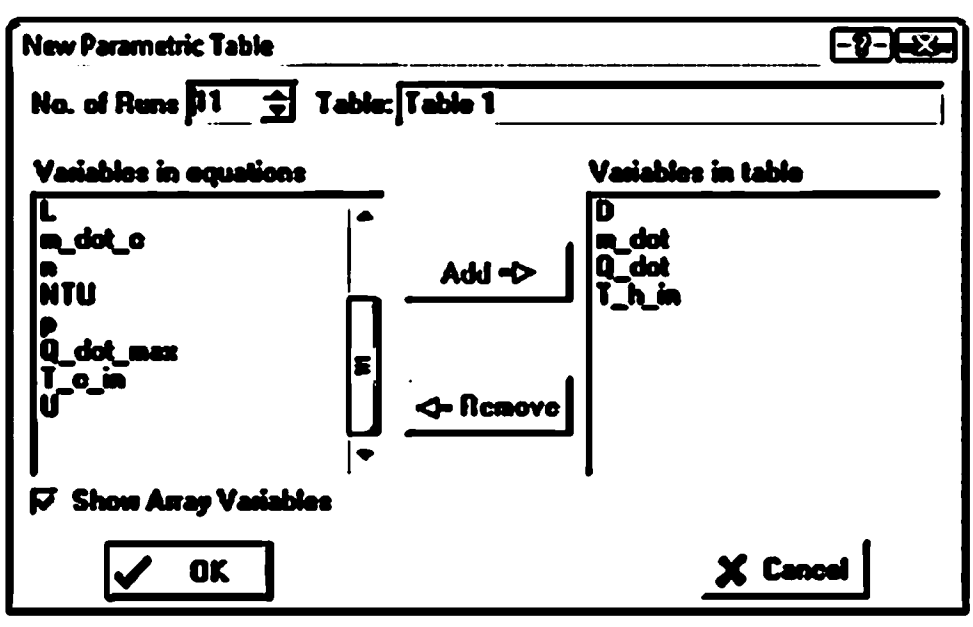
1. Open EES software
2. Create a new file by clicking on the 'New' of the 'File menu'
3. Now type the following code in dailog box

```
p=8                                          "Number of passes"
n=50                                         "Number of tubes"
"T_h_in is varied from 20[C] to 70[C]"       "Temperature of steam"
h_fg=2431[kJ/kg K]                           "Heat of vapourization of steam"
c_pc=4180[J/ kg K]                           "Specific heat of cooling water"
T_c_in=18[C]                                 "Inlet temperature of cooling water"
m_dot_c=2200[kg/h]*convert(kg/h,kg/s)        "Mass flow rate of cooling water"
"D is varied from 1.0[cm] to 2.0[cm]"        "Diameter of tube"
L=2[m]                                       "Lenght of tube"
U=3000[W/m^2 K]                              "Overall heat transfer coefficient"

C_min=m_dot_c*c_pc                           "Specific heat of cold water"
Q_dot_max=C_min*(T_h_in-T_c_in)              "Maximum rate of heat transfer"
A_s=p*n*pi*D*L                               "Heat transfer surface area"
NTU=U*A_s/C_min
epsilon=1-exp(-NTU)
Q_dot=epsilon*Q_dot_max*convert(J/s,KJ/s)
Q_dot=m_dot*h_fg
```

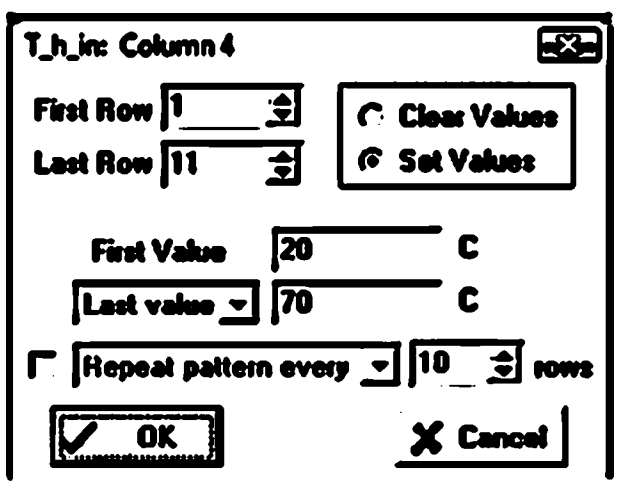
Step 2 of 8

Now to solve the equation for a range of condensing steam temperature and tube diameters, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open.
Select no of runs = 11.
Select variables in equations as "D", "m_dot", "Q_dot" and "T_h_in" press 'Add' and then press 'OK'. The screenshot of the "New Parametric Table" is shown:



Step 3 of 8

First the condensing steam temperature is varied at constant diameter of the tube of $D = 0.015\,m$.
To enter the values in the "T_h_in" column, click on the drop tab in "T_h_in" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 4 of 8

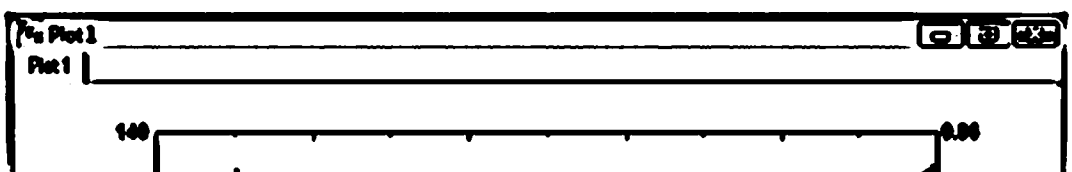
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

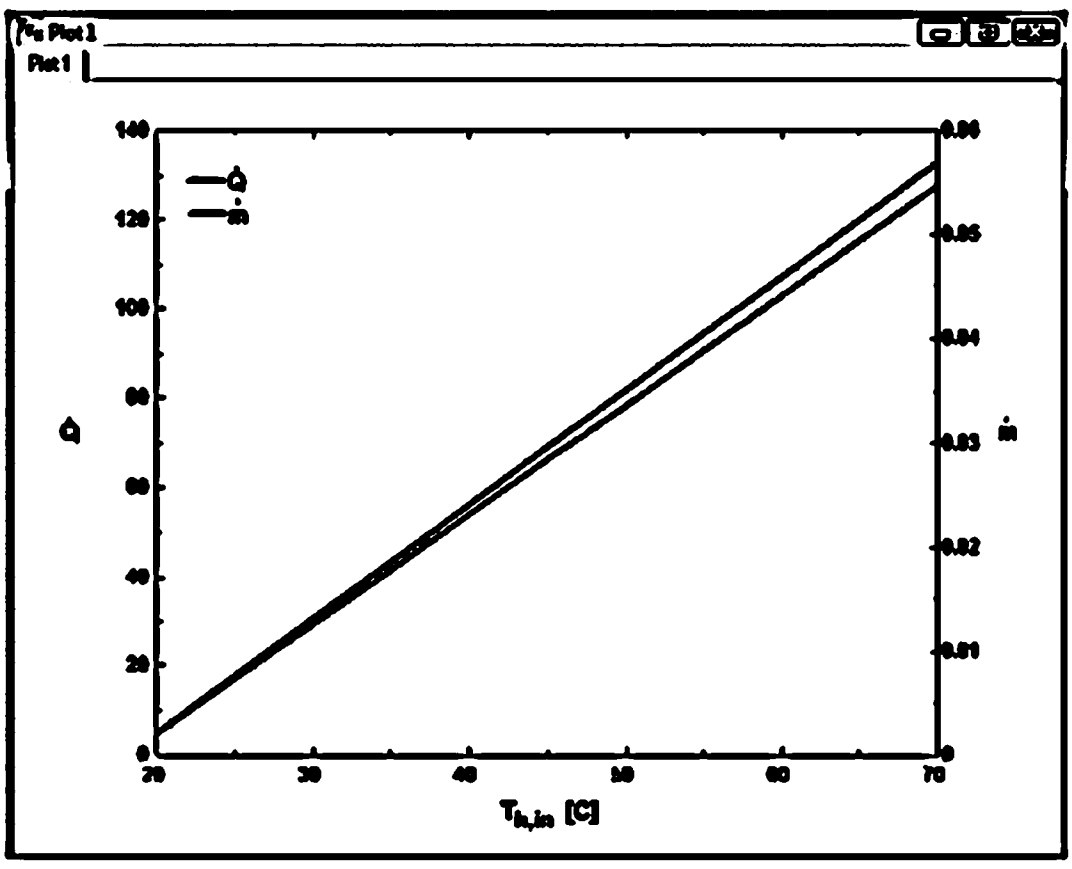
$\triangleright_{1..11}$	1 D	2 $\dot{m}$	3 $\dot{Q}$	4 $T_{h,in}$ [C]
Run 1	0.015	0.002102	5.109	20
Run 2	0.015	0.007355	17.88	25
Run 3	0.015	0.01261	30.65	30
Run 4	0.015	0.01786	43.43	35
Run 5	0.015	0.02312	56.2	40
Run 6	0.015	0.02837	68.97	45
Run 7	0.015	0.03362	81.74	50
Run 8	0.015	0.03888	94.51	55
Run 9	0.015	0.04413	107.3	60
Run 10	0.015	0.04939	120.1	65
Run 11	0.015	0.05464	132.8	70

Step 5 of 8

Inorder to plot the heat transfer and rate of condensation as function of steam temperature, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot showing the heat transfer and rate of condensation as function of steam temperature is shown:



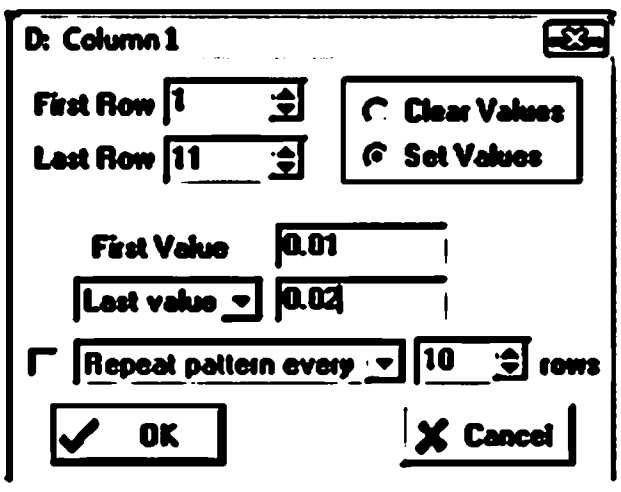
steam temperature is shown: 22.096P pg 2



From the above plot we can interpret as the steam temperature increases, the heat transfer and rate of condensation also increases.

Step 6 of 8

Now the diameter of the tube is varied at constant condensing steam temperature of T = 30°C.
To enter the values in the "D" column, click on the drop tab in "D" column. A new windows pops up, fill in the details as shown below and then press 'OK'



Step 7 of 8

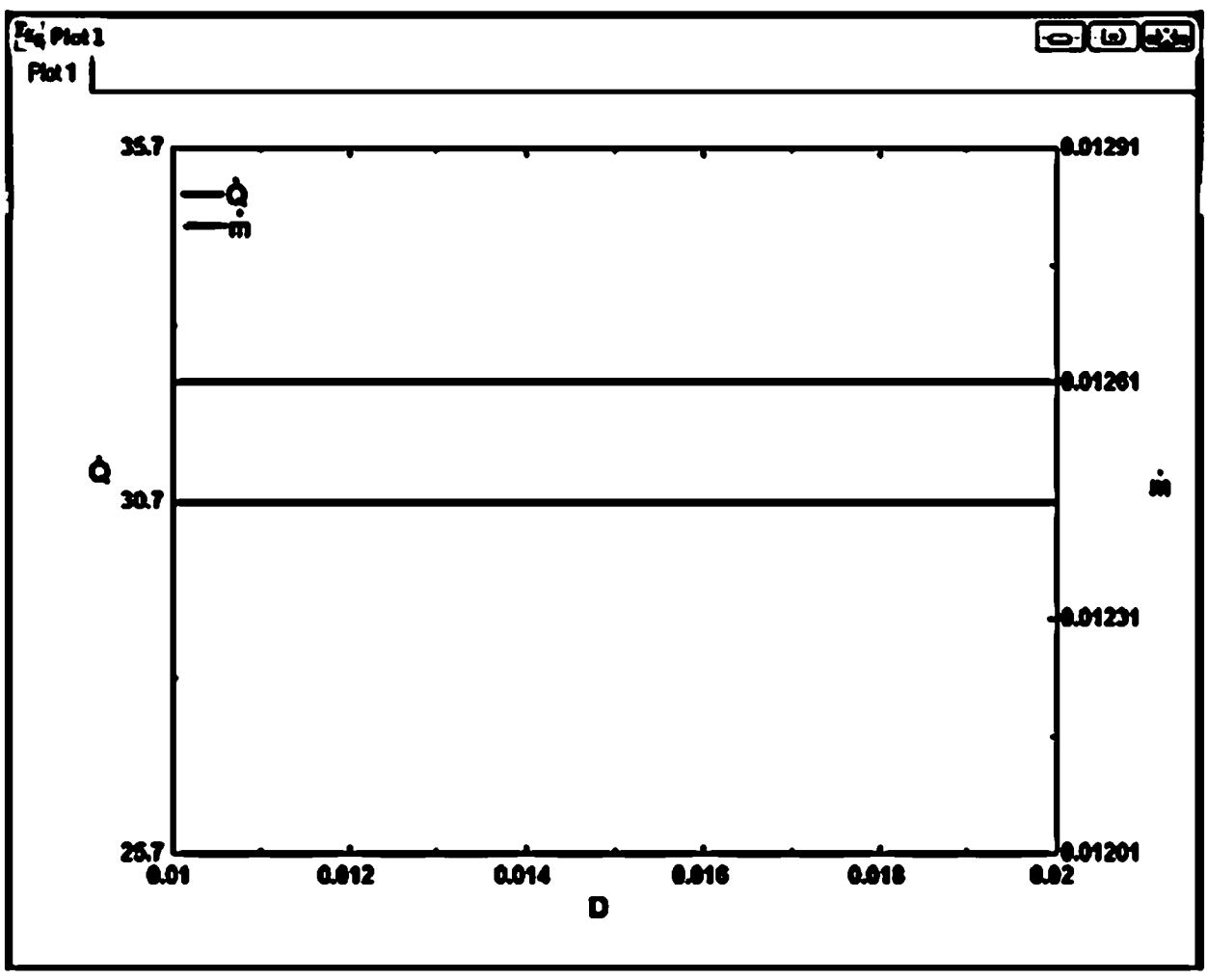
Now click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below:

Table 1

1..11	1 D	2 ṁ	3 Q̇	4 $T_{h,in}$
Run 1	0.01	0.01261	30.65	30
Run 2	0.011	0.01261	30.65	30
Run 3	0.012	0.01261	30.65	30
Run 4	0.013	0.01261	30.65	30
Run 5	0.014	0.01261	30.65	30
Run 6	0.015	0.01261	30.65	30
Run 7	0.016	0.01261	30.65	30
Run 8	0.017	0.01261	30.65	30
Run 9	0.018	0.01261	30.65	30
Run 10	0.019	0.01261	30.65	30
Run 11	0.02	0.01261	30.65	30

Step 8 of 8

Inorder to plot the heat transfer and rate of condensation as function of diameter of the tube, click on the 'X-Y Plot' of 'New Plot window' from 'Plots' menu.
Screenshot of the plot showing the heat transfer and rate of condensation as function of diameter of the tube is shown:



From the above plot we can interpret as the diameter of the tube increases, there will be no effect on the heat transfer and rate of condensation.

Question: 22.097P

Cold water (cp = 4180 J/kg·K) leading to a shower enters a thin-walled double-pipe counter-flow heat exchanger at 15°C at a rate of 0.25 kg/s and is heated to 45°C by hot water (cp = 4190 J/kg·K) that enters at 100°C at a rate of 3 kg/s. If the overall heat transfer coefficient is 950 W/m2·K, determine the rate of heat transfer and the heat transfer surface area of the heat exchanger using the ε-NTU method.

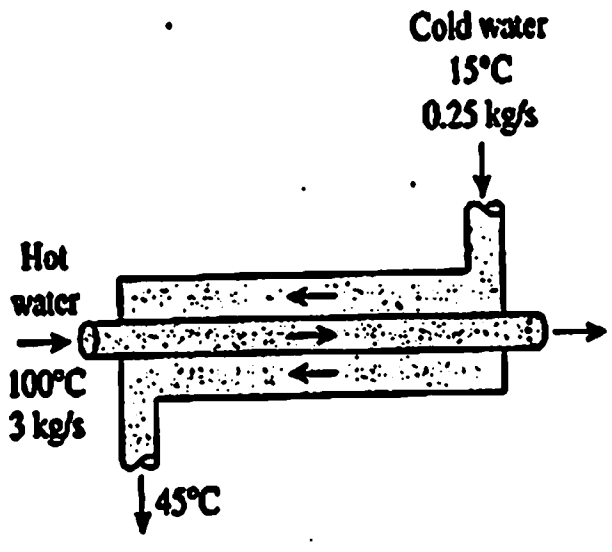


FIGURE P22–97

Step 1 of 8

Given data:

Specific heat of hot water, $c_{ph} = 4190$ J/kg°C

Specific heat of cold water, $c_{pc} = 4180$ J/kg°C

Mass flow rate of cold water, $\dot{m}_h = 0.25$ kg/s

Mass flow rate of hot water, $\dot{m}_c = 3$ kg/s

Overall heat transfer coefficient $U = 950$ W/m^2°C

Step 2 of 8

Using NTU method heat transfer surface area is determined as follows

The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h C_{ph}$$
$$= (0.25 \text{ kg/s})(4180 \text{ J/kg°C})$$
$$= 1045 \text{ W/°C}$$
$$C_c = \dot{m}_c C_{pc}$$
$$= (3 \text{ kg/s})(4190 \text{ J/kg°C})$$
$$= 12,570 \text{ W/°C}$$

Step 3 of 8

Therefore the minimum heat capacity is

$$C_{min} = C_c = 1045 \text{ W/°C}$$

$$\text{And } C = \frac{C_{min}}{C_{max}}$$
$$= \frac{1045}{12,570}$$
$$= 0.083$$

Step 4 of 8

The maximum heat transfer rate is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (1045 \text{ W/°C})(100°C\text{-}15°C)$$
$$= 88,825 \text{ W}$$

Step 5 of 8

Actual heat transfer is calculated by

$$\dot{Q} = C_h(T_{h,in} - T_{h,out})$$
$$= (1045 \text{ W/°C})(45°C - 15°C)$$
$$= 31350 \text{ W}$$

∴ The Actual rate of heat transfer is $\boxed{\dot{Q} = 31.35 \text{kW}}$

Step 6 of 8

Then effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$
$$= \frac{31,350}{88,825}$$
$$= 0.35$$

Step 7 of 8

NTU of the heat exchanger is

$$NTU = \frac{1}{C-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C - 1}\right)$$
$$= \frac{1}{0.083-1}\ln\left(\frac{0.35-1}{0.35\times 0.083-1}\right)$$
$$= 0.438$$

Step 8 of 8

Heat transfer surface area is calculated as

$$NTU = \frac{UA_s}{C_{min}}$$
$$A_s = \frac{NTU(C_{min})}{U}$$
$$= \frac{(0.438)(1045 \text{ W/°C})}{950 \text{ W/m}^2\text{.°C}}$$
$$= 0.4818 \text{m}^2$$

∴ The heat transfer surface area $\boxed{A_s = 0.4818 \text{m}^2}$

Question: 22.098P pg 1

Reconsider Prob. 22-97. Using EES (or other) software, investigate the effects of the inlet temperature of hot water and the heat transfer coeffi-cient on the rate of heat transfer and the surface area. Let the inlet temperature vary from 60°C to 120°C and the overall heat transfer coefficient from 750 W/m2·K to 1250 W/m2-K. Plot the rate of heat transfer and surface area as functions of the inlet temperature and the heat transfer coefficient, and discuss the results.

Step 1 of 7

The given problem can be solved using EES software by following the below steps.

1. **Open EES software**
2. **Crease a new file by clicking on the 'New' of the 'File menu'**
3. **Now type the following code in dailog box**

```
"GIVEN"
T_cw_in=15[C]
T_cw_out=45[C]
m_dot_cw=0.25 [kg/s]
C_p_cw=4.18 [kJ/kg-C]
T_hw_in=100[C], "Vary this parameter "
m_dot_hw=3[kg/s]
C_p_hw=4.19 [kJ/kg-C]
"U=0.95 [kW/m^2-C], "Vary this parameter "

"ANALYSIS"
" Solve this problem using LMTD method than NTU method its would be easier by
EES. Below, Use LMTD method. Both methods give the same results."
DELTAT_1=T_hw_in-T_cw_out
DELTAT_2=T_hw_out-T_cw_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_hw*C_p_hw*(T_hw_in-T_hw_out)
Q_dot=m_dot_cw*C_p_cw*(T_cw_out-T_cw_in)
```

Step 2 of 7

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 10
Select variables in equations as $T_{hw,in}$ and Q ,A press 'Add' and then press 'ok'
A table will appear and enter the values of length under '$T_{hw,in}$ ' vary from 60°C to 120°C
Then from click on 'Solve table' of 'Calculate' menu.
The Solution will appear as shown below

Step 3 of 7

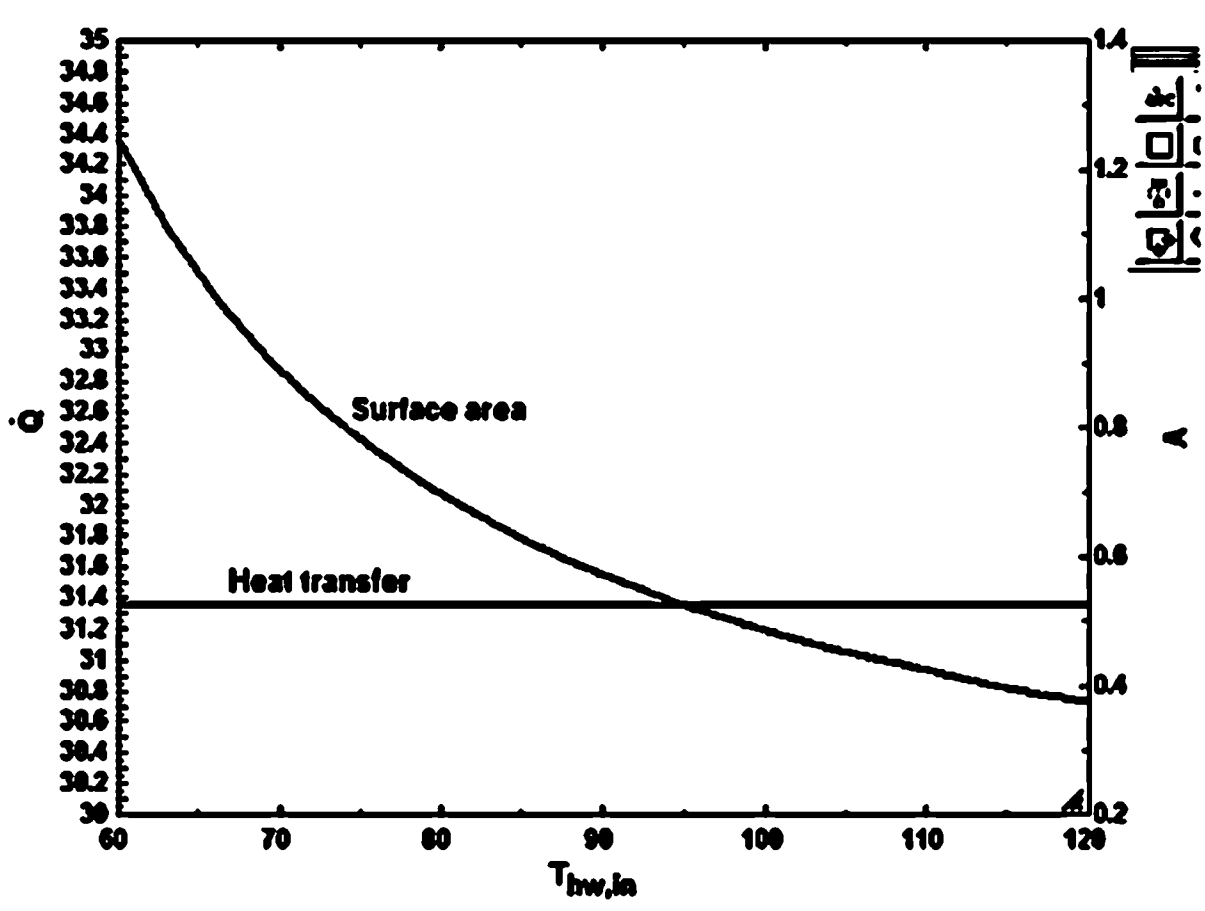
The following results are displayed:

Table 1

1..10	$T_{hw,in}$	$\dot{Q}$	A
Run 1	60	31.35	1.25
Run 2	66.67	31.35	0.9833
Run 3	73.33	31.35	0.814
Run 4	80	31.35	0.6957
Run 5	86.67	31.35	0.6082
Run 6	93.33	31.35	0.5405
Run 7	100	31.35	0.4865
Run 8	106.7	31.35	0.4425
Run 9	113.3	31.35	0.4058
Run 10	120	31.35	0.3748

Step 4 of 7

Select the plot on menu bar, and then click x-y plot .Dialog box will be appeared.
Select X-axis as $T_{hw,in}$ and Y-axis as $\dot{Q}$, A .Press OK
The Graph will be plotted as shown below.
By plotting the graph of heat transfer, surface area as function of inlet temperature:



This graph represents that by increasing the inlet temperature; the heat transfer remains constant and surface decreases.

Step 5 of 7

Now to solve the equation for a range of length values, use parametric table by clicking on the 'New parametric table' of 'Tables' menu.
A dailog box will open. Now
Select no of runs = 20
Select variables in equations as U and Q ,A press 'Add' and then press 'ok'
A table will appear and enter the values of length under 'U ' vary from 0.75 kW/m^2·K to 1250 kW/m^2·K
Then from click on 'Solve table' of 'Calculate' menu.

Select no of runs = 20 22.098P pg 2

Select variables in equations as U and Q, A press 'Add' and then press 'ok'

A table will appear and enter the values of length under 'U' vary from

$0.75\ kW/m^2 \cdot K$ to $1250\ kW/m^2 \cdot K$

Then from click on 'Solve table' of 'Calculate' menu.

The Solution will appear as shown below

Step 6 of 7

The following results are displayed:

1..20	U	Q̇	A
Run 1	0.75	31.35	0.6163
Run 2	0.7171	31.35	0.6446
Run 3	0.6842	31.35	0.6755
Run 4	0.6513	31.35	0.7097
Run 5	0.6184	31.35	0.7474
Run 6	0.5855	31.35	0.7894
Run 7	0.5526	31.35	0.8364
Run 8	0.5197	31.35	0.8893
Run 9	0.4868	31.35	0.9494
Run 10	0.4539	31.35	1.018
Run 11	0.4211	31.35	1.098
Run 12	0.3882	31.35	1.191
Run 13	0.3553	31.35	1.301
Run 14	0.3224	31.35	1.434
Run 15	0.2895	31.35	1.597
Run 16	0.2566	31.35	1.801
Run 17	0.2237	31.35	2.066
Run 18	0.1908	31.35	2.423
Run 19	0.1579	31.35	2.927
Run 20	0.125	31.35	3.698

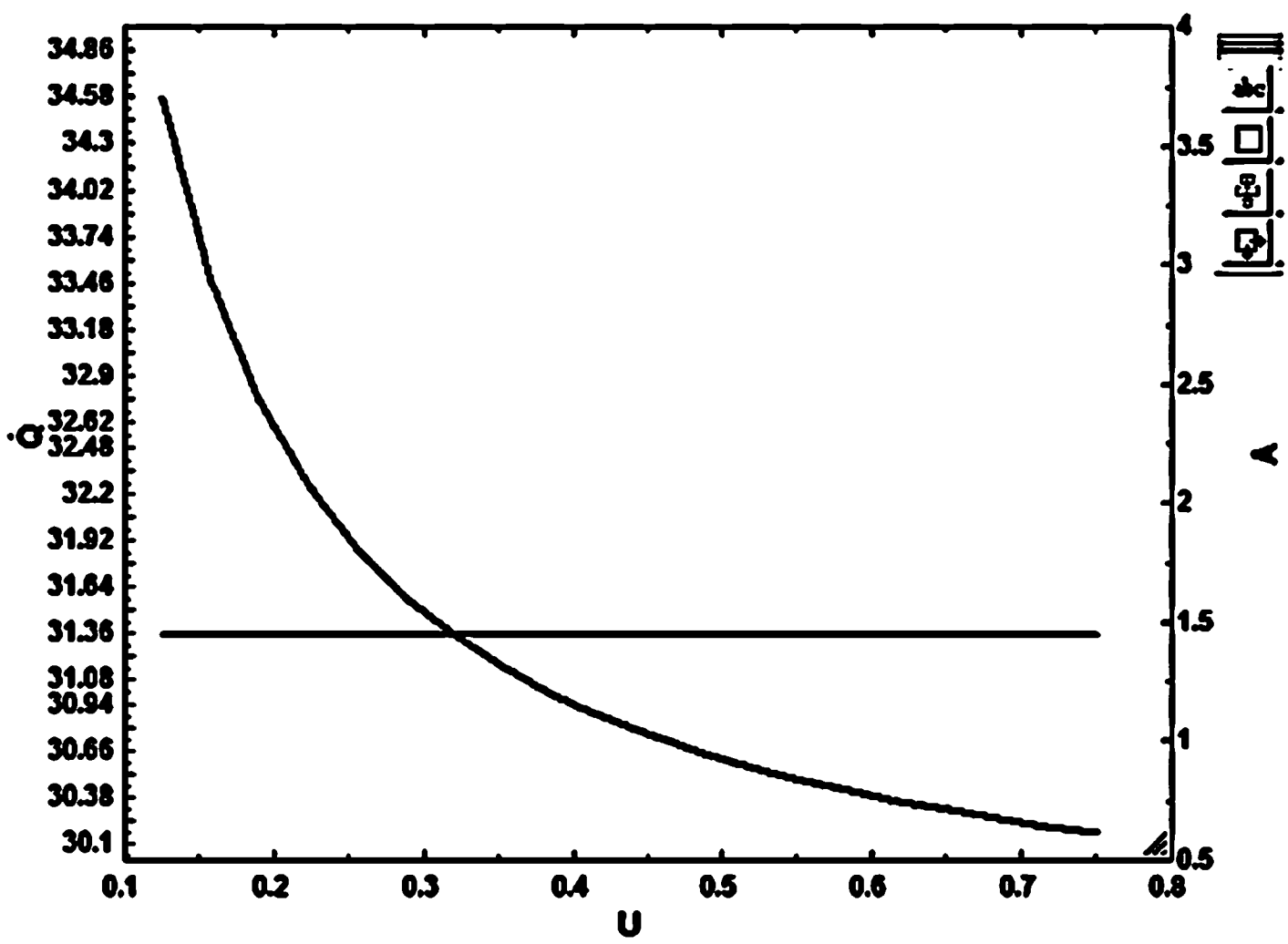
Step 7 of 7

Select the plot on menu bar, and then click x-y plot .Dialog box will be appeared.

Select X-axis as $T_{hot,in}$ and Y-axis as $\dot{Q}, A$ .Press OK

The Graph will be plotted as shown below.

By plotting the graph of heat transfer, surface area as function of overall heat transfer coefficient:



This graph represents that by increasing overall heat transfer coefficient; the heat transfer remains constant and surface decreases.

Question: 22.099P

Oil is being cooled from 180°F to 120°F in a 1-shell and 2-tube heat exchanger with an overall heat transfer coefficient of 40 Btu/h·ft2·°F. Water (*cpc* = 1.0 Btu/lbm·°F) enters at 80°F and exits at 100°F with a mass flow rate of 20,000 lbm/h, determine (*a*) the NTU value and (*b*) the surface area of the heat exchanger.

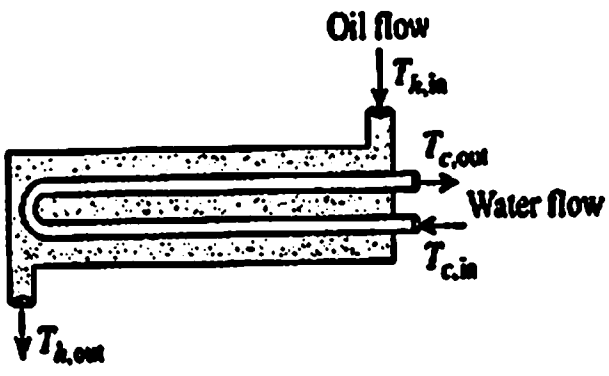


FIGURE P22–99E

Step 1 of 4

a)

NTU method:

Calculate the rate of heat transfer

$$\dot{Q} = \dot{m} c_p (T_{out} - T_{in})$$

Here,

Mass flow rate of Ethylene glycol is $\dot{m}$

Specific heat is c_p

Outlet temperature is T_{out}

Inlet temperature is T_{in}

Substitute 5.56 lbm/s for $\dot{m}$, 1 kJ/kg.K for c_p, 100°F for T_{out}, 80°F for T_{in}

$$\dot{Q} = 5.56(1)(100-80)$$

$$\dot{Q} = 111.2 \text{ Btu/s}$$

Step 2 of 4

Calculate the C_{min} for heat exchanger is

$$\dot{Q} = mc_p (T_o - T_{io})$$

$$111.2 = mc_p (180-120)$$

$$mc_p = 1.853 \text{ Btu/°F}$$

Here,

Oil inlet temperature is T_o

Oil exit temperature is T_{io}

Here,

$$C_{max} = mc_{pc}$$
$$= 5.56 \times 1$$
$$= 5.56 \text{ Btu/s·°F}$$
$$C_{min} = mc_p$$
$$= 1.853 \text{ Btu/°F}$$

Calculate the maximum rate of heat transfer is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$

Substitute 1.853 Btu/°F for C_{min}, 180°F for T_o, 120°F for T_{io}

$$\dot{Q} = 1.852(180-80)$$

$$\dot{Q} = 185.2 \text{ Btu/s}$$

Step 3 of 4

Calculate the Effectiveness of the heat exchanger is

$$\varepsilon = \frac{Q}{Q_{max}}$$

Substitute 111.2 Btu/s for $\dot{Q}$, 185.2 Btu/s for $\dot{Q}_{max}$

$$\varepsilon = \frac{111.2}{185.2}$$
$$= 0.67$$

Calculate The NTU of this heat exchanger

$$NTU = -\ln(1-\varepsilon)$$

$$NTU = -\ln(1-0.67)$$

$$NTU = 1.11$$

Hence, number of tubes is $\boxed{1.11}$

Step 4 of 4

(b)

Calculate the surface area of this heat exchanger becomes

$$NTU = \frac{UA_s}{C_{min}}$$

$$A_s = \frac{NTU\, C_{min}}{U}$$

Here,

Overall heat transfer coefficient is U

Substitute 1.11 for NTU, 6670.8 Btu/hr.°F for C_{min} 40 Btu/h·ft^2·F for U

$$A_s = \frac{(1.11)(6670.8)}{40}$$
$$= 185.11 \text{ ft}^2$$

Hence, surface area is $\boxed{185 \text{ ft}^2}$

Question: 22.100P

Oil in an engine is being cooled by air in a cross-flow heat exchanger, where both fluids are unmixed. Oil (*cph* = 2047 J/kg·K) flowing with a flow rate of 0.026 kg/s enters the heat exchanger at 75°C, while air (*cpc* = 1007 J/kg·K) enters at 30°C with a flow rate of 0.21 kg/s. The overall heat transfer coefficient of the heat exchanger is 53 W/m2·K and the total surface area is 1 m2. Determine (*a*) the heat transfer effectiveness and (*b*) the outlet temperature of the oil.

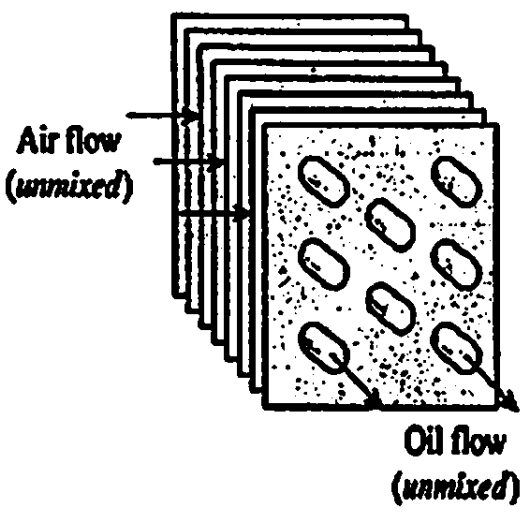


FIGURE P22–100

Step 1 of 5

Given

The specific heat of oil $c_{ph} = 2047\ \text{J/kg.°C}$

The specific heat of air $c_{pc} = 1007\ \text{J/kg.°C}$

Mass flow rate of oil is $\dot{m}_{oil} = 0.026\ \text{kg/s}$

Mass flow rate of air is $\dot{m}_{air} = 0.21\ \text{kg/s}$

Inlet temperature of oil $T_{h,in} = 75°\text{C}$

Inlet temperature of air $T_{c,in} = 30°\text{C}$

Overall heat transfer coefficient is $U = 53\ \text{W/m}^2\text{.K}$

Step 2 of 5

Heat capacity rate of the hot and cold fluids are

$$C_h = \dot{m} c_{ph}$$
$$= 0.026 \times 2047$$
$$C_h = 53.222\ \text{W/°C}$$

$$C_c = \dot{m}_c c_{pc}$$
$$= 0.21 \times 1007$$
$$C_c = 211.47\ \text{W/°C}$$

Step 3 of 5

Then

$$C_{min} = C_h = 53.22\ \text{W/°C}$$

And capacity ratio

$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{53.22}{211.47}$$
$$c = 0.25166$$

Step 4 of 5

(a) Heat transfer effectiveness

The maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

$$\dot{Q}_{max} = 53.22(75-30)$$

$$\dot{Q}_{max} = 2394.9\ \text{W}$$

NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{min}}$$
$$= \frac{53 \times 1}{53.22\ \text{W/°C}}$$
$$\text{NTU} = 0.9958$$

Then the effectiveness of the heat exchanger for unmixed cross flow heat exchanger is

$$\varepsilon = 1 - \exp\left[-\frac{1}{c}\left(1 - \exp(-c\,\text{NTU})\right)\right]$$

$$\varepsilon = 1 - \exp\left[-\frac{1}{0.25166}\left(1 - \exp(-0.25166 \times 0.9958)\right)\right]$$

$$\boxed{\varepsilon = 0.585}$$

Step 5 of 5

(b)

Outlet temperature of the cold water determined to be

$$\dot{Q}_{max} = C_h\left(T_{h,in} - T_{h,out}\right)$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}_{max}}{C_h}$$

$$T_{h,out} = 75 - \frac{2394.9}{53.222}$$

$$\boxed{T_{h,out} = 30°\text{C}}$$

Question: 22.101P

A cross-flow heat exchanger with both fluids unmixed has an overall heat transfer coefficient of 200 W/m2·K, and a heat transfer surface area of 400 m2. The hot fluid has a heat capacity of 40,000 W/K, while the cold fluid has a heat capacity of 80,000 W/K. If the inlet temperatures of both hot and cold fluids are 80°C and 20°C, respectively, determine (*a*) the exit temperature of the hot fluid and (*b*) the rate of heat transfer in the heat exchanger.

Step 1 of 5

Given data:

The overall heat transfer coefficient $U = 200\ \text{W/m}^2\text{K}$

Heat capacity of hot and cold fluids is

$C_h = 40,000\ \text{W/K}$

$C_c = 80,000\ \text{W/K}$

Inlet temperatures of hot and cold fluids are

$T_{h,in} = 80°\text{C}$

$T_{c,in} = 20°\text{C}$

The heat transfer surface area $A_s = 400\ \text{m}^2$

Step 2 of 5

The smaller of two heat capacities is

$C_{min} = C_h = 40,000\ \text{W/K}$

The maximum rate of heat transfer rate in a cross flow exchanger is

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

$$\dot{Q}_{max} = 40000(80-20)$$

$$= 2400000\ \text{W}$$

$$\dot{Q}_{max} = 2400\ \text{kW}$$

Step 3 of 5

The NTU of heat exchanger is expressed as

$$\text{NTU} = \frac{UA_s}{C_{min}}$$

$$= \frac{(200)400}{40000}$$

$$\text{NTU} = 2$$

Also

$$c = \frac{C_{min}}{C_{max}}$$

$$= \frac{40000}{80000}$$

$$c = 0.5$$

From table Effectiveness for heat exchangers, the value of effectiveness is determined as

At $\text{NTU} = 2$ and $c = 0.5$

The heat exchanger effectiveness is

$\varepsilon = 0.74$

Step 4 of 5

(a) The exit temperature of hot fluid

$$\dot{Q}_h = \varepsilon\dot{Q}_{max} = C_h\left(T_{h,in} - T_{h,out}\right)$$

$$0.74\times2400\times10^3 = 80000\left(80 - T_{h,out}\right)$$

$$\boxed{T_{h,out} = 57.8°\text{C}}$$

Therefore the exit temperature of the hot fluid, $T_{h,out} = 57.8°\text{C}$

Step 5 of 5

(b)The actual heat transfer rate of the heat exchanger is

$$\dot{Q} = \varepsilon\dot{Q}_{max}$$

$$\dot{Q} = (0.74)(2400)$$

$$\boxed{\dot{Q} = 1776\ \text{kW}}$$

Therefore, the actual heat transfer rate of heat exchanger, $\dot{Q} = 1776\ \text{kW}$

Question: 22.102P

A single-pass cross-flow heat exchanger uses hot air (mixed) to heat water (unmixed), flowing with a mass flow rate of 3 kg/s, from 30°C to 80°C. The hot air enters and exits the heat exchanger at 220°C and 100°C, respectively. If the overall heat transfer coefficient is 200 W/m2·K, determine the required surface area.

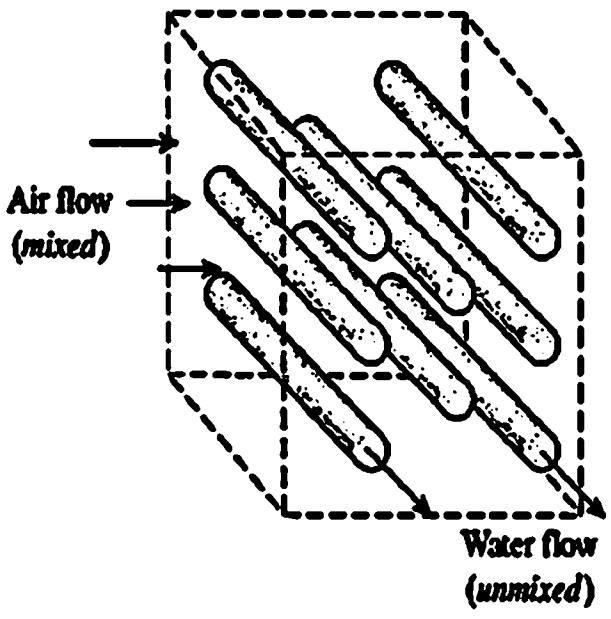


FIGURE P22–102

Step 1 of 4

Given data:

Mass flow rate of water $\dot{m}_{water} = 3\ \text{kg/s}$

Inlet and outlet temperature of water

$T_{c,in} = 30°\text{C}$

$T_{c,out} = 80°\text{C}$

Inlet and outlet temperature of hot air

$T_{h,in} = 220°\text{C}$

$T_{h,out} = 100°\text{C}$

Overall heat transfer coefficient $U = 200\ \text{W/m}^2\cdot\text{K}$

Step 2 of 4

Temperature difference between two fluids at two ends of the heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= 220 - 80$$
$$= 140°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= 100 - 30$$
$$= 70°\text{C}$$

Therefore the LMTD is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$
$$= \frac{140 - 70}{\ln[140/70]}$$
$$\Delta T_{lm} = 101°\text{C}$$

The specific heat of water is $c_{pc} = 4.18\ \text{kJ/kg.K}$

Heat transfer rate determined to be

$$\dot{Q} = \left[\dot{m}_c c_{pc}(T_{out} - T_{in})\right]_{water}$$
$$= 3(4.18)(80 - 30)$$
$$= 627\ \text{kW}$$

Step 3 of 4

The correction factor can be obtained as

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$
$$= \frac{T_{h,out} - T_{h,in}}{T_{c,in} - T_{h,in}}$$
$$= \frac{100 - 220}{30 - 220}$$
$$P = 0.63$$
$$R = \frac{T_1 - T_2}{t_2 - t_1}$$
$$= \frac{T_{c,in} - T_{c,out}}{T_{h,out} - T_{h,in}}$$
$$= \frac{30 - 80}{100 - 220}$$
$$R = 0.416$$

At $P = 0.63$ and $R = 0.416$ from correction factor F charts for single pass cross flow with one fluid mixed and other unmixed the correction factor is

$F = 0.9$

Step 4 of 4

Heat transfer surface area is

$$\dot{Q} = UA_s F \Delta T_{lm}$$
$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$
$$A_s = \frac{627 \times 10^3\ \text{W}}{(200\ \text{W/m}^2\cdot°\text{C}) \times 0.9 \times (101°\text{C})}$$

$$\boxed{A_s = 34.48\ \text{m}^2}$$

Therefore, the surface area, $A_s = 34.48\ \text{m}^2$

Question:

The mass flow rate, specific heat, and inlet temperature of the tube-side stream in a double-pipe, parallel-flow heat exchanger are 3200 kg/h, 2.0 kJ/kg·K, and 120°C, respectively. The mass flow rate, specific heat, and inlet temperature of the other stream are 2000 kg/h, 4.2 kJ/kg·K, and 20°C, respectively. The heat transfer area and overall heat transfer coefficient are 0.50 m2 and 2.0 kW/m2·K, respectively. Find the outlet temperatures of both streams in steady operation using (*a*) the LMTD method and (*b*) the effectiveness-NTU method.

Step 1 of 9

(a)

Using LMTD method:

The rate of heat transfer of hot fluid can be calculated as:

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h1} - T_{h2})$$

Here,

Mass flow rate of tube side stream, $\dot{m}_h$

Specific heat tube side stream, c_{ph}

Inlet temperature of tube side (hot) stream, T_{h1}

Hot fluid exit temperature, T_{h2}

Substitute 3200 kg/h for $\dot{m}_h$, 2.0 kJ/kg·K for c_{ph}, 120°C for T_{h1}, in the above equation

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h1} - T_{h2})$$

$$= \frac{3200}{3600} \times 2.0 \times (120 - T_{h2})$$

$$= 1.77 \times (120 - T_{h2}) \quad \text{..... (1)}$$

The rate of heat transfer of cold fluid is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c2} - T_{c1})$$

Here,

Mass flow rate of second stream, $\dot{m}_c$

Specific heat of second stream, c_{pc}

Inlet temperature of second stream, T_{c_1}

Cold fluid exit temperature, T_{c2}

Substitute 2000 kg/h for $\dot{m}_c$, 4.2 kJ/kg.K for c_{pc}, 20°C for T_{c_1} in the above equation

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c2} - T_{c1})$$

$$= \frac{2000}{3600} \times 4.2 \times (T_{c2} - 20)$$

$$= 2.333 (T_{c2} - 20) \quad \text{..... (2)}$$

Step 2 of 9

Log mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad \text{..... (3)}$$

Where temperature differences at the inlet and exit of the heat exchanger is

$$\Delta T_1 = T_{h1} - T_{c1}$$

Substitute the values of 120°C for T_{h1}, 20°C for T_{c1} in the above equation

$$\Delta T_1 = T_{h1} - T_{c1}$$

$$= 120 - 20$$

$$= 100°C$$

$$\Delta T_2 = T_{h2} - T_{c2}$$

Substitute the values of 120°C for T_{h1}, 20°C for T_{c_1} in the equation (3)

$$\Delta T_{lm} = \frac{100 - (T_{h2} - T_{c2})}{\ln\left(\frac{100}{(T_{h2} - T_{c2})}\right)}$$

Step 3 of 9

Heat transfer can also be determined by using the equation

$$\dot{Q} = UA \times \Delta T_{lm}$$

Here,

Heat transfer area, A

Overall heat transfer coefficient, U

Substitute $0.50\,m^2$ for A, $2.0\,kW/m^2 \cdot K$ for U, ΔT_{lm} values in the above equation

$$\dot{Q} = UA \times \Delta T_{lm}$$

$$= 2.0 \times 0.5 \times \left(\frac{100 - (T_{h2} - T_{c2})}{\ln\left(\frac{100}{(T_{h2} - T_{c2})}\right)} \right)$$

$$= \left(\frac{100 - (T_{h2} - T_{c2})}{\ln\left(\frac{100}{(T_{h2} - T_{c2})}\right)} \right) \quad \text{..... (4)}$$

Step 4 of 9

Solving the equations (1), (2) and (4) by using equation solver

$$\dot{Q} = 63.88\,kW$$

$$T_{h2} = 84.19°C$$

$$T_{c2} = 47.17°C$$

Therefore, the exit temperature of hot fluid T_{h2} is $\boxed{84.19°C}$

Exit temperature of cold fluid T_{c2} is $\boxed{47.17°C}$

Step 5 of 9

(b)

Using effectiveness-NTU method:

The heat capacity of hot fluid is

$$C_h = \dot{m}_h c_{ph}$$

Substitute 3200 kg/h for $\dot{m}_h$, 2.0 kJ/kg·K for c_{ph}, in the above equation.

$$C_h = \dot{m}_h c_{ph}$$

The heat capacity of hot fluid is

$$C_h = \dot{m}_h c_{ph}$$

Substitute 3200 kg/h for $\dot{m}_h$, 2.0 kJ/kg·K for c_{ph}, in the above equation.

$$\begin{aligned} C_h &= \dot{m}_h c_{ph} \\ &= \frac{3200}{3600} \times 2.0 \\ &= 1.77 \text{ kW/K} \end{aligned}$$

The heat capacity of cold fluid is

$$C_c = \dot{m}_c c_{pc}$$

Substitute 2000 kg/h for $\dot{m}_c$, 4.2 kJ/kg.K for c_{pc}, in the above equation

$$\begin{aligned} C_c &= \dot{m}_c c_{pc} \\ &= \frac{2000}{3600} \times 4.2 \\ &= 2.333 \text{ kW/K} \end{aligned}$$

Step 6 of 9

From the above heat capacity ratios the minimum heat capacity rate is

$$C_{min} = C_h = 1.77 \text{ kW/K}$$

Heat capacity ratios

$$\begin{aligned} c &= \frac{C_{min}}{C_{max}} \\ &= \frac{1.77}{2.333} \\ &= 0.758 \end{aligned}$$

Step 7 of 9

Number of transfer units, NTU of this heat exchanger is

$$\text{NTU} = \frac{UA}{C_{min}}$$

Substitute 0.50 m^2 for A, $2.0 \text{ kW/m}^2 \cdot \text{K}$ for U, 1.77 kW/K for C_{min} values in the above equation

$$\begin{aligned} \text{NTU} &= \frac{UA}{C_{min}} \\ &= \frac{2 \times 0.5}{1.77} \\ &= 0.564 \end{aligned}$$

Effectiveness of the parallel flow heat exchanger is

$$\varepsilon = \frac{1 - \exp\left(-\text{NTU}(1+c)\right)}{1+c}$$

Substitute 0.564 for NTU, 0.758 for c in the above equation

$$\begin{aligned} \varepsilon &= \frac{1 - \exp\left(-\text{NTU}(1+c)\right)}{1+c} \\ &= \frac{1 - \exp\left(-0.564 \times (1+0.758)\right)}{1+0.758} \\ &= 0.3577 \end{aligned}$$

Step 8 of 9

Maximum heat transfer rate is

Substitute the values of 120°C for T_{h1}, 20°C for T_{c1} in the above equation

$$\begin{aligned} \dot{Q}_{max} &= C_{min}(T_{h1} - T_{c1}) \\ &= 1.77(120 - 20) \\ &= 177 \text{ kW} \end{aligned}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$

$$\begin{aligned} \dot{Q} &= \varepsilon \dot{Q}_{max} \\ &= 0.3577 \times 177 \\ &= 63.312 \text{ kW} \end{aligned}$$

Step 9 of 9

The rate of heat transfer of hot fluid can be calculated as:

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h1} - T_{h2})$$

Substitute the values of 3200 kg/h for $\dot{m}_h$, 2.0 kJ/kg·K for c_{ph}, 120°C for T_{h1}, 20°C for T_{c1}, in the equation

$$\begin{aligned} \dot{Q} &= \dot{m}_h c_{ph}(T_{h1} - T_{h2}) \\ 63.312 &= \frac{3200}{3600} \times 2.0 \times (120 - T_{h2}) \\ T_{h2} &= 84.387°\text{C} \end{aligned}$$

Therefore, exit temperature of the hot fluid T_{h2} is $\boxed{84.387°\text{C}}$

The rate of heat transfer of cold fluid is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c2} - T_{c1})$$

Substitute 2000 kg/h for $\dot{m}_c$, 4.2 kJ/kg.K for c_{pc}, 20°C for T_{c_1} in the above equation

$$\begin{aligned} \dot{Q} &= \dot{m}_c c_{pc}(T_{c2} - T_{c1}) \\ 63.312 &= \frac{2000}{3600} \times 4.2 \times (T_{c2} - 20) \\ T_{c2} &= 47.13°\text{C} \end{aligned}$$

Therefore, exit temperature of the cold fluid T_{c2} is $\boxed{47.13°\text{C}}$

Question: 22.104P

A shell-and-tube heat exchanger with 1-shell pass and 14-tube passes is used to heat water in the tubes with geothermal steam condensing at 120°C (hfg = 2203 kJ/kg) on the shell side. The tubes are thin-walled and have a diameter of 2.4 cm and length of 3.2 m per pass. Water (cp = 4180 J/kg·K) enters the tubes at 18°C at a rate of 6.2 kg/s. If the temperature difference between the two fluids at the exit is 46°C, determine (*a*) the rate of heat transfer, (*b*) the rate of condensation of steam, and (*c*) the overall heat transfer coefficient.

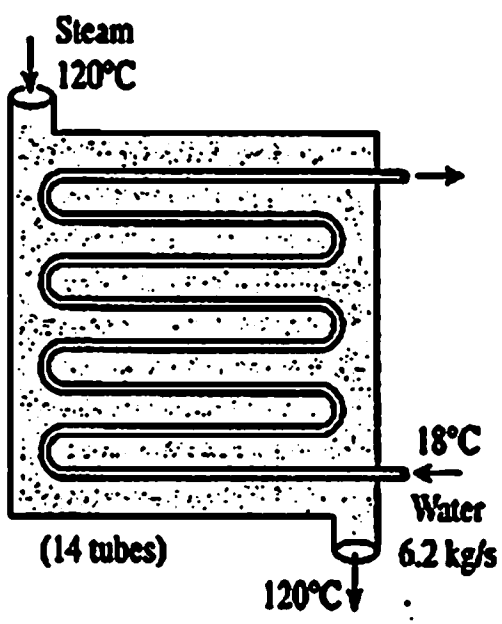


FIGURE P22–104

Step 1 of 5

Given data:

The specific heat of water is $c_p = 4180\ \text{J/kg}\cdot^\circ\text{C}$

Temperature of steam $T_{h,i} = T_{h,e} = 120^\circ\text{C}$

Inlet temperature of water $T_{c,i} = 18^\circ\text{C}$

The temperature difference between the two fluids at the exit is $T_{h,e} - T_{c,e} = 46^\circ\text{C}$

The heat of vaporization of steam is $h_{fg} = 2203\ \text{kJ/kg}$

Mass flow rate of water $\dot{m}_{water} = 6.2\ \text{kg/s}$

The diameter and length of the tube $D = 2.4\ \text{cm}, L = 3.2\ \text{m}$

Step 2 of 5

a) Heat transfer rate is calculated as follows

The outlet temperature of the water is

$T_{h,e} - T_{c,e} = 46^\circ\text{C}$

$T_{c,e} = T_{h,e} - 46^\circ\text{C}$

$T_{c,out} = 120 - 46$

$T_{c,out} = 74^\circ\text{C}$

Then the rate of heat transfer becomes

$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{water}$

$\dot{Q} = (6.2\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(74^\circ\text{C} - 18^\circ\text{C})$

$\boxed{\dot{Q} = 1451.296\ \text{kW}}$

Therefore, the heat transfer $\dot{Q} = 1451.296\ \text{kW}$

Step 3 of 5

b) The rate of condensation steam is

$\dot{Q} = (\dot{m}h_{fg})$

$1415.296 = \dot{m}(2203)$

$\boxed{\dot{m} = 0.658\ \text{kg/s}}$

Therefore, the rate of mass $\dot{m} = 0.658\ \text{kg/s}$

Step 4 of 5

b) The heat transfer surface area is

$A_i = n\pi D_i L$

$= 14\pi(0.024\ \text{m})(3.2\ \text{m})$

$A_i = 3.3778\ \text{m}^2$

For this counter flow heat exchanger LMTD can be calculated as

$\Delta T_1 = T_{h,i} - T_{c,e}$

$= 120^\circ\text{C} - 74^\circ\text{C}$

$\Delta T_1 = 46^\circ\text{C}$

$\Delta T_2 = T_{h,out} - T_{c,in}$

$= 120^\circ\text{C} - 18^\circ\text{C}$

$\Delta T_2 = 102^\circ\text{C}$

Then LMTD of this heat exchanger is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{46 - 102}{\ln(46/102)}$$

$\Delta T_{lm} = 70.32^\circ\text{C}$

Step 5 of 5

We know the correction factor in case of condenser is $F = 1$

Therefore the Overall heat transfer coefficient of the heat exchanger is determined to be

$\dot{Q} = U_i A_i F \Delta T_{lm}$

$$U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm}}$$

$$U_i = \frac{1451.296 \times 10^3\ \text{W}}{3.3778\ \text{m}^2 \times 1 \times 70.32^\circ\text{C}}$$

$\boxed{U_i = 6110.028\ \text{W/m}^2\cdot^\circ\text{C}}$

Therefore the Overall heat transfer coefficient, $U_i = 6110.028\ \text{W/m}^2\cdot^\circ\text{C}$

Question: 22.105P

Geothermal water (*cp* = 4250 J/kg·K) at 75°C is to be used to heat fresh water (*cp* = 4180 J/kg·K) at 17°C at a rate of 1.2 kg/s in a double-pipe counter-flow heat exchanger. The heat transfer surface area is 25 m2, the overall heat transfer coefficient is 480 W/m2·K, and the mass flow rate of geothermal water is larger than that of fresh water. If the effectiveness of the heat exchanger is desired to be 0.823, determine the mass flow rate of geothermal water and the outlet temperatures of both fluids.

Step 1 of 10

Given data:

Specific heat of geothermal water, $c_{ph} = 4250\ \text{J/kg°C}$

Specific heat of freshwater, $c_{pc} = 4180\ \text{J/kg°C}$

Mass flow rate of fresh water is $\dot{m}_c = 1.2\ \text{kg/s}$

Overall heat transfer coefficient $U = 480\ \text{W/m}^2\text{°C}$

Effectiveness of the heat exchanger is $\varepsilon = 0.823$

Heat transfer surface area is $A_s = 25\ \text{m}^2$

Step 2 of 10

Outlet temperatures of the geothermal water and fresh water are determined as

$$\dot{Q} = C_h\left(T_{h,in} - T_{h,out}\right)$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h}$$

$$\dot{Q} = C_c\left(T_{c,out} - T_{c,in}\right)$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$

Step 3 of 10

The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h C_{ph}$$
$$= \dot{m}_h\,(4250\ \text{J/kg}\cdot\text{°C})$$
$$C_c = \dot{m}_c C_{pc}$$
$$= (1.2\ \text{kg/s})(4180\ \text{J/kg}\cdot\text{°C})$$
$$= 5016\ \text{W/°C}$$

Step 4 of 10

Therefore $C_{min} = C_c = 5016\ \text{W/°C}$

$$C = \frac{C_{min}}{C_{max}}$$
$$= \frac{5016}{\dot{m}_h\,(4250\ \text{J/kg}\cdot\text{°C})}$$

NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{min}}$$
$$= \frac{(0.480\ \text{kW/m}^2\cdot\text{°C})(25\ \text{m}^2)}{5.016\ \text{kW/°C}}$$
$$= 2.392$$

Step 5 of 10

Heat capacity ratio can be calculated as

$$\varepsilon = \frac{1-\exp\left[-\text{NTU}(1-C)\right]}{1-C\exp\left[-\text{NTU}(1-C)\right]}$$

$$0.823 = \frac{1-\exp\left[-2.392(1-C)\right]}{1-C\exp\left[-2.392(1-C)\right]}$$

$$0.823\left[1-C\exp\left(-2.392\left(1-C\right)\right)\right] = 1-\exp\left[-2.392(1-C)\right]$$

By trail and error method

Substituting different values of the C which will satisfy the equation

If $C = 0.2$

$$0.823\left[1-0.2\exp\left(-2.392\left(1-0.2\right)\right)\right] = 1-\exp\left[-2.392(1-0.2)\right]$$

$$0.798 \neq 0.852$$

It is not satisfying the equation so go for next trail

If $C = 0.3$

$$0.823\left[1-0.3\exp\left(-2.392\left(1-0.3\right)\right)\right] = 1-\exp\left[-2.392(1-0.3)\right]$$

$$0.7767 \neq 0.8125$$

It is not satisfying the equation so go for next trail

If $C = 0.4$

$$0.823\left[1-0.4\exp\left(-2.392\left(1-0.4\right)\right)\right] = 1-\exp\left[-2.392(1-0.4)\right]$$

$$0.7018 = 0.7019$$

If $C = 0.494$

$$0.823\left[1-0.494\exp\left(-2.392\left(1-0.494\right)\right)\right] = 1-\exp\left[-2.392(1-0.494)\right]$$

$$0.7018 = 0.7019$$

It is satisfying the equation

$\therefore C = 0.494$

Step 6 of 10

Mass flow rate of the geothermal water is calculated as

$$C = \frac{1.1802}{\dot{m}_h}$$

$$0.494 = \frac{1.1802}{\dot{m}_h}$$

$$\dot{m}_h = \frac{1.1802}{0.494}$$

$$\dot{m}_h = 2.389\ \text{kg/s}$$

$\therefore$ The mass flow rate of geothermal water, $\boxed{\dot{m}_h = 2.389\ \text{kg/s}}$

Step 7 of 10

The maximum heat transfer rate is

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$
$$= (5016\ \text{W/°C})(75\text{°C} - 17\text{°C})$$
$$= 290.928\ \text{kW}$$

The actual heat transfer rate is

$$\dot{Q} = \varepsilon\dot{Q}_{max}$$
$$= (0.823)(290.928\ \text{kW})$$
$$= 239.433\ \text{kW}$$

Step 8 of 10

Heat capacity of the hot fluid is

$$C_h = \dot{m}_h C_{ph}$$
$$= (2.389\ \text{kg/s})(4250\ \text{J/kg}\cdot\text{°C})$$
$$= 10.15325\ \text{kW/°C}$$

Step 9 of 10

By substituting the heat transfer rate outlet temperatures of the geothermal water and fresh water are

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h}$$
$$= 75\text{°C} - \frac{239.433\ \text{kW}}{10.15325\ \text{kW/°C}}$$
$$= 51.4\text{°C}$$

$\therefore$ The outlet temperature of the hot fluid, $\boxed{T_{h,out} = 51.4\text{°C}}$

Step 10 of 10

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$
$$= 17\text{°C} + \frac{239433\ \text{W}}{5016\ \text{W/°C}}$$
$$= 64.7\text{°C}$$

$\therefore$ The outlet temperature of the cold fluid, $\boxed{T_{c,out} = 64.7\text{°C}}$

Question: 22.106P

Hot water at 60°C is cooled to 36°C through the tube side of a 1-shell pass and 2-tube passes heat exchanger. The coolant is also a water stream, for which the inlet and outlet temperatures are 7°C and 31°C, respectively. The overall heat transfer coefficient and the heat transfer area are 950 W/m2·K and 15 m2, respectively. Calculate the mass flow rates of hot and cold water streams in steady operation.

Step 1 of 5

Calculation of mass flow rates of hot and cold water streams:

In order to calculate the mass flow rate of hot and cold water streams, let us calculate the rate of heat exchange

Temperature difference between two fluids at two ends of the heat exchanger is,

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

Here $T_{h,in}$ is the Inlet temperature of hot water, $T_{h,out}$ is the outlet temperature of hot water $T_{c,in}$ is the Inlet temperature of cold water and $T_{c,out}$ is the outlet temperature of cold water

Substitute 60 °C for $T_{h,in}$, 36 °C for $T_{h,out}$, 7 °C for $T_{c,in}$, 31°C for $T_{c,out}$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= 60 - 31$$
$$\Delta T_1 = 29°C$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= 36 - 7$$
$$\Delta T_2 = 29°C$$

Step 2 of 5

Since $\Delta T_2 = \Delta T_1$, the LMTD is $\Delta T_{lm} = 29$ °C

The correction factor for 1- shell pass and 2- tube passes heat exchanger can be calculated as follows,

We have

Heat transfer area, $A = 15$ m^2

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$
$$P = \frac{31 - 7}{60 - 7}$$
$$P = 0.45$$
$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$
$$R = \frac{60 - 36}{31 - 7}$$
$$R = 1$$

From Correction factor charts for common shell and tube and cross flow heat exchangers at above P and R values, the correction factor

$F = 0.88$

Step 3 of 5

Overall heat transfer coefficient, $U = 950$ W/m^2·K

Now, the rate of heat exchange is expressed as,

$$\dot{Q} = UA_s F(\Delta T_{lm})$$
$$= (950)(15)(0.88)(29)$$
$$= 363660 \text{ W}$$
$$\dot{Q} = 363.66 \text{ kW}$$

Step 4 of 5

Therefore, the mass flow rates of cold stream is,

$$\dot{Q} = \dot{m}_c C_{pc}(T_{c,out} - T_{c,in})$$
$$\dot{m}_c = \frac{363.66}{4.18(31 - 7)}$$
$$\boxed{\dot{m}_c = 3.625 \text{ kg/s}}$$

Step 5 of 5

Similarly the mass flow rate of hot stream is,

$$\dot{Q} = \dot{m}_h C_{pc}(T_{h,in} - T_{h,out})$$
$$\dot{m}_h = \frac{363.66}{4.18(60 - 36)}$$
$$\boxed{\dot{m}_h = 3.625 \text{ kg/s}}$$

Therefore, the mass flow rates of hot and cold water streams, $\boxed{\dot{m}_h = 3.625 \text{ kg/s}}$

And $\boxed{\dot{m}_c = 3.625 \text{ kg/s}}$

Question: 22.107P

Cold water (*cp* = 4180 J/kg·K) enters the tubes of a heat exchanger with 2-shell passes and 20-tube passes at 15°Cat a rate of 4 kg/s, while hot oil (*cp* = 2200 J/kg-K) enters the shell at 130°C at the same mass flow rate and leaves at 60°C. If the overall heat transfer coefficient based on the outer surface of the tube is 220 W/m2-K, determine (*a*) the rate of heat transfer and (*b*) the heat transfer surface area on the outer side of the tube.

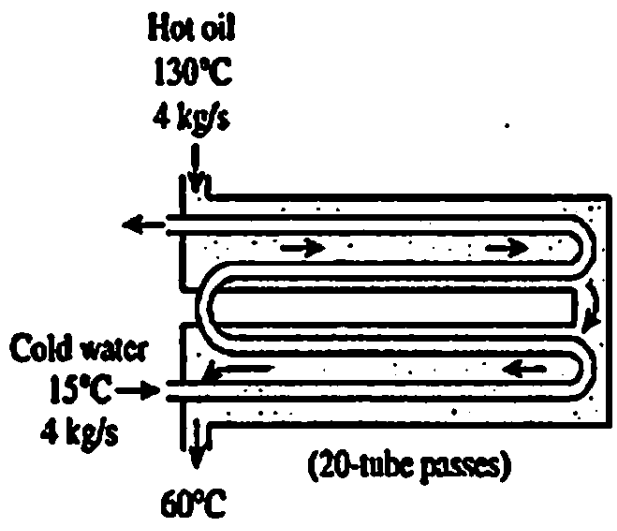


FIGURE P22–107

Step 1 of 6

Given data:

Specific heat of hot oil $c_{ph} = 2200\ \text{J/kg}\cdot\text{K}$

Specific heat of cold water $c_{pc} = 4180\ \text{J/kg}\cdot\text{K}$

Mass flow rate of cold water is $\dot{m}_c = 4\ \text{kg/s}$

Inlet temperature of cold water $T_{c,in} = 15°\text{C}$

Inlet temperature of hot oil $T_{h,in} = 130°\text{C}$

Exit temperature of hot oil $T_{h,out} = 60°\text{C}$

Mass flow rate of hot oil is $\dot{m}_h = 4\ \text{kg/s}$

Overall heat transfer coefficient $U = 220\ \text{W/m}^2\cdot\text{K}$

Step 2 of 6

(a) The rate of heat transfer is

$$\dot{Q} = [\dot{m}_h c_{ph}(T_{h,out} - T_{h,in})]_{heat}$$

$$\dot{Q} = 4\left((2.2\ \text{kJ/kg.°C}) \times (130°\text{C} - 60°\text{C})\right)$$

$$\boxed{\dot{Q} = 616\ \text{kW}}$$

Therefore, the rate of heat transfer is $\dot{Q} = 616\ \text{kW}$

Step 3 of 6

(b) The surface area of the tube can be calculated as follows

Heat transfer surface area on the tube side is determined to be

$$\dot{Q} = UA_s\Delta T_{lm}$$

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

The outlet temperature of the cold water can be calculated as

$$\dot{Q} = \dot{m}_h c_{pc}\left(T_{c,out} - T_{c,in}\right)$$

$$616 = 4 \times 4.18\left(T_{c,out} - 15\right)$$

$$T_{c,out} = 51.84°\text{C}$$

Step 4 of 6

For this counter flow heat exchanger LMTD can be determined as

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= 130°\text{C} - 51.84°\text{C}$$

$$\Delta T_1 = 78.16°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= 60°\text{C} - 15°\text{C}$$

$$\Delta T_2 = 45°\text{C}$$

Then LMTD is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{78.16 - 45}{\ln\left(\frac{78.16}{45}\right)}$$

$$\Delta T_{lm} = 60.062$$

Step 5 of 6

And correction factor for 2 shell pass and two tube pass heat exchanger can be calculated as follows

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

$$= \frac{51.84 - 15}{130 - 15}$$

$$P = 0.32$$

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}$$

$$= \frac{130 - 60}{51.84 - 15}$$

$$R = 1.90$$

From Correction factor *F* charts for common shell and tube and cross flow heat exchangers at *P* and *R* values

$$F = 0.92$$

Step 6 of 6

By substituting all values in heat transfer surface area

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}}$$

$$A_s = \frac{616\ \text{kW}}{(0.22\ \text{kW/m}^2\cdot°\text{C})(0.92)(60.062°\text{C})}$$

$$\boxed{A_s = 50.672\ \text{m}^2}$$

Therefore, the surface area, $A_s = 50.672\ \text{m}^2$

Question: 22.108P

Water (*cp* = 1.0 Btu/lbm-°F) is to be heated by solar- heated hot air (*cp* = 0.24 Btu/lbm-°F) in a double-pipe counter- flow heat exchanger. Air enters the heat exchanger at 190°F at a rate of 0.7 lbm/s and leaves at 135°F. Water enters at 70°F at a rate of 0.35 lbm/s. The overall heat transfer coefficient based on the inner side of the tube is given to be 20 Btu/h-ft2-°F. Determine the length of the tube required for a tube internal diameter of 0.5 in.

Step 1 of 7

Given data:

Specific heat of water $c_{ph} = 1\ \text{Btu/lbm} \cdot {}^\circ\text{F}$

Specific heat of hot air $c_{pc} = 0.24\ \text{Btu/lbm} \cdot {}^\circ\text{F}$

Mass flow rate of hot air $\dot{m}_h = 0.7\ \text{lbm/s}$

Mass flow rate of water $\dot{m}_c = 0.35\ \text{lbm/s}$

Overall heat transfer coefficient $U = 20\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$

Internal diameter of the tube, $D_i = 0.5\text{in}$

Step 2 of 7

Heat transfer surface area on the tube side is determined to be

$$\dot{Q} = UA_s\Delta T_{lm}$$

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}}$$

$\dot{Q}$, F & ΔT_{lm} are calculated by the following relations

Heat transfer rate is

$$\dot{Q} = [\dot{m}_c c_{pc}(T_{out} - T_{in})]_{water}$$

$$= (0.7\ \text{lbm/s})(0.24\ \text{Btu/lbm.°F})(190°\text{F} - 135°\text{F})$$

$$= 9.24\ \text{Btu/s}$$

Step 3 of 7

Then the outlet temperature of the oil is

$$\dot{Q} = \left[\dot{m}_h c_{ph}(T_{out} - T_{in})\right]_{oil}$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}}$$

$$= 70°\text{F} + \frac{9.24\ \text{Btu/s}}{(0.35\ \text{lbm/s})(1.0\ \text{Btu/lbm.°F})}$$

$$= 96.4°\text{F}$$

Step 4 of 7

For this counter flow heat exchanger LMTD determined to be

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= 190°\text{F} - 96.4°\text{F}$$

$$= 93.6°\text{F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= 135°\text{F} - 70°\text{F}$$

$$= 65°\text{F}$$

Step 5 of 7

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$= \frac{93.6 - 65}{\ln(93.6/65)}$$

$$= 78.43°\text{F}$$

Step 6 of 7

By substituting all values heat transfer surface area becomes

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}}$$

$$A_s = \frac{9.24\ \text{Btu/s}}{(20/3600\ \text{Btu/s} \cdot \text{ft}^2 \cdot °\text{F})(78.43°\text{F})}$$

$$A_s = 21.2\ \text{ft}^2$$

Step 7 of 7

Therefore the length of the tube required becomes

$$A_s = \pi DL$$

$$L = \frac{A_s}{\pi D}$$

$$= \frac{21.2\ \text{ft}^2}{\pi(0.5/12\ \text{ft})}$$

$$= 162.0\text{ft}$$

∴ **The length of the tube required,** $\boxed{L = 162.0\text{ft}}$

Question: 22.109P

By taking the limit as $\Delta T2 \rightarrow \Delta T1$ show that when $\Delta T1 = \Delta T2$ for a heat exchanger, the Tlm relation reduces to $\Delta Tlm = \Delta T1 = \Delta T2$.

Step 1 of 1

In counter flow heat exchanger the temperature difference between the hot and the cold fluids remains constant along the heat exchanger when the heat capacity rates of the two fluids are equal (that is, $\Delta T = \text{constant}$ when $C_h = C_c$).

If $\Delta T_1 = \Delta T_2$ then the log mean temperature difference relation

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

$$= \frac{0}{0}$$

By applying L'Hospital's rule (taking derivatives of nominator and denominator separately with respect to ΔT_1 or ΔT_2) LMTD becomes

$$\Delta T_{lm} = \frac{d(\Delta T_1 - \Delta T_2)/d\Delta T_1}{d[\ln(\Delta T_1 / \Delta T_2)]/d\Delta T_1}$$

$$= \frac{1}{1/\Delta T_1}$$

$$= \Delta T_1$$

$$= \Delta T_2$$

Question: 22.110P

The condenser of a room air conditioner is designed to reject heat at a rate of 22,500 kJ/h from refrigerant-134a as the refrigerant is condensed at a temperature of 40°C. Air (*cp* = 1005 J/kg·K) flows across the finned condenser coils, entering at 25°C and leaving at 32°C. If the overall heat transfer coefficient based on the refrigerant side is 150 W/m2·K, determine the heat transfer area on the refrigerant side.

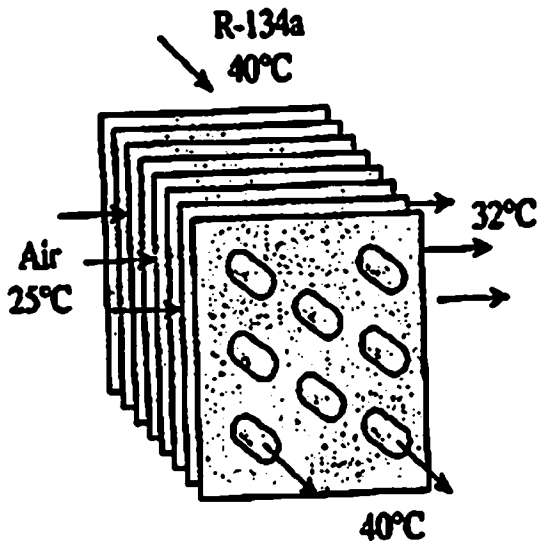


FIGURE P22–110

Step 1 of 4

Given data:

Heat transfer rate is, $\dot{Q} = 22500\,\text{kJ/h}$

$= 6.25\,\text{kW}$

The specific heat of air is $c_p = 1005\,\text{J/kg°}\cdot\text{K}$

Inlet temperature of the R-134a, $T_{in,\text{R-134a}} = 40°\text{C}$

Inlet temperature of the air, $T_{in,air} = 25°\text{C}$

Outlet temperature of the air, $T_{out,air} = 32°\text{C}$

Overall heat transfer coefficient $U = 150\ \text{W/m}^2\cdot\text{K}$

Step 2 of 4

Temperature difference between two fluids at two ends of the heat exchanger is

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$= 40°\text{C} - 32°\text{C}$$
$$= 8°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$
$$= 40°\text{C} - 25°\text{C}$$
$$= 15°\text{C}$$

Step 3 of 4

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$
$$= \frac{8-15}{\ln(8/15)}$$
$$= 11.136°\text{C}$$

Step 4 of 4

Heat transfer surface area is

$$\dot{Q} = UA_s\Delta T_{lm}$$
$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}}$$
$$= \frac{6250\ \text{W}}{(150\ \text{W/m}^2.°\text{C})(11.136°\text{C})}$$
$$= 3.74172\,\text{m}^2$$

∴ The heat transfer area is $\boxed{A_s = 3.74\ \text{m}^2}$

Question: 22.111P

Air (*cp* = 1005 J/kg·K) is to be preheated by hot exhaust gases in a cross-flow heat exchanger before it enters the furnace. Air enters the heat exchanger at 95 kPa and 20°C at a rate of 0.4 m3/s. The combustion gases (*cp* = 1100 J/kg·K) enter at 180°C at a rate of 0.65 kg/s and leave at 95°C. The product of the overall heat transfer coefficient and the heat transfer surface area is *UAS* = 1620 W/K. Assuming both fluids to be unmixed, determine the rate of heat transfer.

Step 1 of 2

Given data:

The specific heat of combustion gasses is $c_p = 1100\ \text{J/kg} \cdot ^\circ\text{C}$

Mass flow rate of combustion gasses $\dot{m}_{gas} = 0.65\ \text{kg/s}$

Inlet temperature of the gases $T_{in,gas} = 180^\circ\text{C}$

Outlet temperature of the gases $T_{out,gas} = 95^\circ\text{C}$

Step 2 of 2

Heat transfer rate is determined as

$$\dot{Q} = \left[\dot{m}c_p\left(T_{in} - T_{out}\right)\right]_{gas}$$

$$= (0.65\ \text{kg/s})(1.1\ \text{kJ/kg} \cdot ^\circ\text{C})(180^\circ\text{C} - 95^\circ\text{C})$$

$$= 60.775\ \text{kW}$$

∴ The rate of heat transfer, $\boxed{\dot{Q} = 60.775\ \text{kW}}$

Question: 22.112P

In a chemical plant, a certain chemical is heated by hot water supplied by a natural gas furnace. The hot water (cp= 4180 J/kg·K) is then discharged at 60°C at a rate of 8 kg/min. The plant operates 8 h a day, 5 days a week, 52 weeks a year. The furnace has an efficiency of 78 percent, and the cost of the natural gas is $1.00 per therm (1 therm = 105,500 kJ). The average temperature of the cold water entering the furnace throughout the year is 14°C. In order to save energy, it is proposed to install a water-to-water heat exchanger to preheat the incoming cold water by the drained hot water. Assuming that the heat exchanger will recover 72 percent of the available heat in the hot water, determine the heat transfer rating of the heat exchanger that needs to be purchased and suggest a suitable type. Also, determine the amount of money this heat exchanger will save the company per year from natural gas savings.

Step 1 of 6

Given data:

Specific heat of the hot water is $c_{ph} = 4180\ \text{J/kg°C}$

Effectiveness of the heat exchanger is $\varepsilon = 0.72$

Inlet temperature of the hot water, $T_{in,hotwater} = 60°\text{C}$

Inlet temperature of the cold water, $T_{in,coldwater} = 14°\text{C}$

Mass of hot water discharging, $\dot{m}_h = 8\ \text{kg/min}$

Step 2 of 6

The maximum heat transfer rate is determined by

$$\dot{Q}_{max} = \dot{m}_h c_{ph}\left(T_{h,in} - T_{c,in}\right)$$
$$= (8/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(60°\text{C} - 14°\text{C})$$
$$= 25.63\ \text{kW}$$

Step 3 of 6

Actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$
$$= (0.72)(25.63\ \text{kJ/s})$$
$$= 18.45\ \text{kW (or)}$$
$$= 18.45\ \text{kJ/s}$$

Step 4 of 6

Then calculating the annual operating hours

$$\text{hours} = (8\ \text{h/day})(5\ \text{days/week})(52\ \text{week/year})$$
$$= 2080\ \text{h/year}$$

The energy saved during the entire year will be

$$\text{Energy saved} = (\text{heat transfer rate})(\text{operating time})$$
$$= (18.45\ \text{kJ/s})(2080\ \text{h/year})(3600\ \text{s/h})$$
$$= 1.38\times10^{8}\ \text{kJ/year}$$

Step 5 of 6

Then amount of fuel and money saved will be

$$\text{Fuel saved} = \frac{\text{Energy saved}}{\text{Furnace efficiency}}$$
$$= \frac{1.38\times10^{8}\ \text{kJ/year}}{0.78}\left(\frac{1\ \text{therm}}{105,500\ \text{kJ}}\right)$$
$$= 1677\ \text{therms/year}$$

Step 6 of 6

Therefore the amount of money this heat exchanger will save the company per year from natural gas savings

$$\text{Money saved} = (\text{fuel saved})(\text{the price of fuel})$$
$$= (1677\ \text{therms/year})(\$\ 1.0/\text{therm})$$
$$= (1677\ \text{therms/year})(\$\ 1.0/\text{therm})$$

∴ The amount of money saved, $\boxed{\text{Money saved} = \$1677/\text{year}}$

Question:

A shell-and-tube heat exchanger is used for cooling 47 kg/s of a process stream flowing through the tubes from 160°C to 100°C. This heat exchanger has a total of 100 identical tubes, each with an inside diameter of 2.5 cm and negligible wall thickness. The average properties of the process stream are: p = 950 kg/m3, k = 0.50 W/mK, cp = 3.5 kJ/kg·K and μ = 2.0 mPa·s. The coolant stream is water (cp = 4.18 kJ/kg·K) at a flow rate of 66 kg/s and an inlet temperature of 10°c, which yields an average shell-side heat transfer coefficient of 4.0 kW/m2·K. Calculate the tube length if the heat exchanger has (a) a 1-shell pass and a 1-tube pass and (b) a 1-shell pass and 4-tube passes.

Step 1 of 12

Heat transfer rate of the hot fluid is

$$\dot{Q}=\dot{m}_h c_p\left(T_{h,in}-T_{h,out}\right)$$

Here, $\dot{m}_h$ is the flow rate of hot fluid stream, is the specific heat hot water, $T_{h,in}$ is the inlet temperature of the hot fluid, and $T_{h,out}$ is the exit temperature of the hot fluid.

Substitute 47 kg/s for $\dot{m}_h$, 3.5 kj/kg.K for c_{ph}, 160° C for $T_{h,in}$, 100° C for $T_{h,out}$

$$Q=47\times3.5\times(160-100)$$
$$=9870\text{ kW}$$

The exit temperature of the cooling water is

$$\dot{Q}=\dot{m}_c c_{pc}\left(T_{c,out}-T_{c,in}\right)$$

Here $\dot{m}_c$ is the Mass flow rate of cold water, c_{pc} is the specific heat cool water, and $T_{c,in}$ the inlet of temperature of cold water.

Substitute 66 kg/s for $\dot{m}_c$, 4.18 kJ/kgK for c_{ph}, 10°C for $T_{c,in}$

$$\dot{Q}=\dot{m}_c c_{pc}\left(T_{c,out}-T_{c,in}\right)$$
$$9870=66\times4.18\times\left(T_{c,out}-10\right)$$
$$T_{c,out}=45.776\text{ °C}$$

Step 2 of 12

Assume the arrangement is counter flow heat exchanger then temperature difference at The inlet and exit of the heat exchanger is

$$\Delta T_1=T_{h,in}-T_{c,out}$$
$$=160-45.776$$
$$=114.224°C$$
$$\Delta T_2=T_{h,out}-T_{c,in}$$
$$=100-10$$
$$=90°C$$

Step 3 of 12

Then logarithmic temperature difference of the counter flow heat exchanger is

$$\Delta T_{lm}=\frac{\Delta T_1-\Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$
$$=\frac{114.224-90}{In\left(\frac{114.224}{90}\right)}$$
$$=101.63\text{ °C}$$

Step 4 of 12

Velocity of the fluid flowing in the tube is

$$V=\frac{\dot{m}_h}{\rho A}$$

Here $\dot{m}_h$ is the flow rate of hot fluid stream, ρ is the density, D is the inside diameter of the tube, and μ is the Dynamic viscosity, and n is total number of identical tubes

Substitute 47 kg/s for $\dot{m}_h$, 950 kg/m^3 for ρ, 2.5cm for D, 2.0mPa .s for μ.

$$V=\frac{\dot{m}_h}{\rho\times\left(n\times\frac{\pi}{4}D^2\right)}$$
$$=\frac{47}{950\times\left(100\times\frac{\pi}{4}\times0.025^2\right)}$$
$$=1.0078\text{ m/s}$$

Reynolds number of the flow is

$$\text{Re}=\frac{\rho VD}{\mu}$$
$$=\frac{950\times1.0078\times0.025}{2.0\times10^{-3}}$$
$$=11967.62$$

Step 5 of 12

Prandtl number can be calculated by using the formula
Thermal conductivity $k=0.50$ W/m.K

$$\text{Pr}=\frac{\mu c_{pc}}{k}$$
$$=\frac{2.0\times10^{-3}\times3.5\times10^3}{0.5}$$
$$=14$$

Since Reynolds number is greater than 10000,
Assume flow is fully is turbulent flow and
Nusselt number in this case is

$$Nu=0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3}$$
$$=0.023\times11967.62^{0.8}\times14^{0.3}$$
$$=92.89$$

Step 6 of 12

Tube side heat transfer coefficient can be calculate by using Nusselt number

$$Nu=\frac{h_i D}{}$$

Step 6 of 12

Tube side heat transfer coefficient can be calculate by using Nusselt number

$$Nu = \frac{h_i D}{k}$$

$$92.89 = \frac{h_i \times 0.025}{0.5}$$

$$h_i = 1857.8 \text{ W/m}^2 \cdot \text{K}$$

Conduction resistance is negligible, since negligible wall thickness. Then the overall heat Transfer coefficient can be calculated as

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_0}}$$

$$= \frac{1}{\frac{1}{1857.8} + \frac{1}{4000}}$$

$$= 1268.59 \text{ W/m}^2 \cdot \text{K}$$

Step 7 of 12

For one shell and a one tube pass, the correction factor is 1

$F = 1$

The rate of heat transfer for this arrangement is

$$Q = F \times U \times A \times \Delta T_{lm}$$

$$= F \times U \times (n \times DL) \times \Delta T_{lm}$$

$$9870 \times 10^3 = 1 \times 1286.59 \times (100 \times \pi \times 0.025 \times L) \times 101.63$$

$$L = 9.747 \text{ m}$$

Therefore, the length, L of the tube for one shell pass and a one tube pass is $\boxed{9.747\text{m}}$

Step 8 of 12

(b)

The tube length for 1 shell pass and 4tube passes:

Since total number of tubes is 100, number of tubes per pass is

$$\frac{100}{4} = 25$$

Number of tubes $n = 100$

Therefore, 25 tubes per pass in this heat exchanger and it increases the velocity in 4 tube pass is 4 times the velocity of one tube pass

Then

$$V_4 = 4 \times V$$

$$= 4 \times 1.0078$$

$$= 4.0312 m/s$$

Reynolds number in 4 tube pass is

$$R_e = 4 \times R_e$$

$$= 4 \times 11967.62$$

$$= 47870.48$$

Step 9 of 12

Nusselt number in case of fully turbulent flow is

$$Nu = 0.023 R_e^{0.8} \text{Pr}^{0.3}$$

$$= 0.023 \times 47870.48^{0.8} \times 14^{0.3}$$

$$= 281.59$$

Then tube side heat transfer coefficient can be calculated by using Nusselt number

$$Nu = \frac{h_2 D}{k}$$

$$281.59 = \frac{h_2 \times 0.025}{0.5}$$

$$h_2 = 5631.8 \text{ W/m}^2\text{-K}$$

Step 10 of 12

Conduction resistance is negligible, since negligible wall thickness, and then the overall heat transfer coefficient can be calculated as

Average shell side heat transfer coefficient $h_o = 4.0 W/m^2.K$

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_0}}$$

$$= \frac{1}{\frac{1}{5631.8} + \frac{1}{4000}}$$

$$= 2338.83 \text{ W/m}^2\text{-K}$$

Step 11 of 12

Correction factory for 1 shell and 4 tube passes can be determined as follows

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$= \frac{T_{c,in} - T_{c,out}}{T_{h,out} - T_{h,in}}$$

$$= \frac{10 - 45.779}{100 - 160}$$

$$= 0.596$$

Step 12 of 12

$$p = \frac{t_2 - t_1}{T_1 - t_1}$$

$$= \frac{T_{h,out} - T_{h,in}}{T_{c,in} - T_{h,in}}$$

$$= \frac{100 - 160}{10 - 160}$$

$$= 0.4$$

From correction factor char for 1 shell and 4 tube passes at $P = 0.4 \text{ and } R = 0.596$, the

$$= \frac{T_{c,in} - T_{h,in}}{t_{h,in} - t_{c,in}}$$

$$= \frac{100-160}{10-160}$$

$$= 0.4$$

From correction factor char for 1 shell and 4 tube passes at $P = 0.4$ *and* $R = 0.596$, the correction factor is F = 0.96

Then rate of heat transfer for one shell and 4 tube passes is

$$Q = F \times U \times A \times \Delta T_{lm}$$

$$= F \times U \times (n \times \pi DL) \times \Delta T_{lm}$$

$$9870 \times 10^3 = 0.96 \times 2338.83 \times (100 \times \pi \times 0.025 \times L) \times 101.63$$

$$L = 5.507m$$

Therefore, the length, L of the tube for one shell and a four tube pass is **5.507 m**

Question: 22.114P pg 1

A 2-shell passes and 4-tube passes heat exchanger is used for heating a hydrocarbon stream *(cp* = 2.0 kJ/kg·K) steadily from 20°C to 50°C. A water stream enters the shell- side at 80°C and leaves at 40°C. There are 160 thin- walled tubes, each with a diameter of 2.0 cm and length of 1.5m. The tube-side and shell-side heat transfer coefficients are 1.6 and 2.5 kW/m2·K, respectively, (a) Calculate the rate of heat transfer and the mass rates of water and hydrocarbon streams. *(b)* With usage, the outlet hydrocarbon-stream temperature was found to decrease by 5°C due to the deposition of solids on the tube surface. Estimate the magnitude of fouling factor.

Step 1 of 12

Assume the arrangement is counter flow heat exchanger then temperature difference at The inlet and exit of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out}.$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

Here, $T_{h,in}$ is the inlet temperature of the water on the shell side and $T_{h,out}$ is the exit temperature of the water on the shell side, $T_{c,in}$ the inlet of temperature of hydrocarbon stream, $T_{c,out}$ is exit temperature of the hydrocarbon stream

Substitute 20°C for $T_{c,in}$, 50°C for $T_{c,out}$, 80°C for $T_{h,in}$, 40°C for $T_{h,out}$ in the below equation

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out}\\ &= 80 - 50\\ &= 30°C\end{aligned}$$

$$\begin{aligned}\Delta T_2 &= T_{h,out} - T_{c,in}\\ &= 40 - 20\\ &= 20°C\end{aligned}$$

Step 2 of 12

Then logarithmic temperature difference of the counter flow heat exchanger is

$$\begin{aligned}\Delta T_{lm} &= \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)}\\ &= \frac{30-20}{In\left(\frac{30}{20}\right)}\\ &= 24.66\ °C\end{aligned}$$

Step 3 of 12

Conduction resistance is negligible, since negligible wall thickness.
Then the calculate overall heat Transfer coefficient as below

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$

Here, h_i, h_o is the tube and shell side heat transfer co efficient respectively

Substitute 1.6 kW/m^2 for h_i, 2.5 kW/m^2 for h_o

$$\begin{aligned}U &= \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}\\ &= \frac{1}{\frac{1}{1.6} + \frac{1}{2.5}}\\ &= 0.975\ \text{kW/m}^2\text{·K}\end{aligned}$$

Step 4 of 12

Determine, correction factor for 2 shell and 4 tube passes as follows

$$\begin{aligned}R &= \frac{T_1 - T_2}{t_2 - t_1}\\ &= \frac{T_{c,in} - T_{c,out}}{T_{h,out} - T_{h,in}}\\ &= \frac{20-50}{40-80}\\ &= 0.75\end{aligned}$$

$$\begin{aligned}P &= \frac{t_2 - t_1}{T_1 - t_1}\\ &= \frac{T_{h,out} - T_{h,in}}{T_{c,in} - T_{h,in}}\\ &= \frac{40-80}{20-80}\\ &= 0.666\end{aligned}$$

From above the correction factor is for 2 shells and 4 tubes passes at $P = 0.666, R = 0.75$, is $F = 0.91$.
Number of tubes n is160

Step 5 of 12

The rate of heat transfer for this arrangement is

$$\begin{aligned}Q &= F \times U \times A \times \Delta T_{lm}\\ &= F \times U \times (n \times DL) \times \Delta T_{lm}\end{aligned}$$

Here, L is the length of a tube; D is the diameter of the thin tube

Substitute 1.5 m for L, 2 cm for D, 0.91 for F, 0.975 kW/m^2·K for U, 24.66 for ΔT_{lm} in the above equation

$$\begin{aligned}Q &= F \times U \times (n \times DL) \times \Delta T_{lm}\\ &= 0.91 \times 0.975 \times 10^3 (160 \times \pi \times 0.02 \times 1.5) \times 24.66\\ &= 329.936\ \text{kW}\end{aligned}$$

Therefore, the rate of heat transfer **329.936 kW**

Step 6 of 12

The specific heat of hydrocarbon stream and water are

$$c_{pc} = 2.0\ \text{kJ/kg·K}$$

From the table of properties of water, the specific heat of water is

$$c_{ph} = 4.18\ \text{kJ/kg·K}$$

$c_{pc} = 2.0\ \text{kJ/kg·K}$ 22.114P pg 2

From the table of properties of water, the specific heat of water is

$c_{ph} = 4.18\ \text{kJ/kg·K}$

Step 7 of 12

Calculate mass flow rate of the water stream

$$\dot{Q} = \dot{m}_h c_{p,h}\left(T_{h,i} - T_{h,o}\right)$$

Substitute $329.936\ \text{kW}$ for $\dot{Q}$, $4.18\ \text{kJ/kg·K}$ for c_{ph}, $80°\text{C}$ for $T_{h,in}$, $40°\text{C}$ for $T_{h,out}$ in the above equation

$$329.936\ \text{kW} = \dot{m}_h \times 4.18 \times (80-40)$$

$$\dot{m}_h = 1.97\ \text{kg/s}$$

Therefore, the mass flow rate water $\boxed{1.97\ \text{kg/s}}$

Step 8 of 12

Calculate mass flow rate of the hydro carbon stream

$$\dot{Q} = \dot{m}_c c_{p,c}\left(T_{c,i} - T_{c,o}\right)$$

Substitute $329.936\ \text{kW}$ for $\dot{Q}$, $2\ \text{kJ/kg·K}$ for $c_{p,c}$, $20°\text{C}$ for $T_{c,in}$, $50°\text{C}$ for $T_{c,out}$ in the above equation

$$329.936\ \text{kW} = \dot{m}_h \times 2 \times (50-20)$$

$$\dot{m}_c = 5.49\ \text{kg/s}$$

Therefore, the mass flow rate water $\boxed{5.49\ \text{kg/s}}$

Step 9 of 12

(b)

Outlet temperature of hydrocarbon stream is decreased by $T_{c,out,old} = 45°\text{C}$

Calculate the rate of heat transfer

$$\dot{Q}_{old} = \dot{m}_c c_{p,c}\left(T_{c,out,old} - T_{c,in}\right)$$
$$= 5.498 \times 2.0 \times (45-20)$$
$$= 274.9\ \text{kW}$$

Calculate the exit temperature of hot fluid

$$\dot{Q}_{old} = \dot{m}_h c_{p,h}\left(T_{h,in} - T_{h,out,old}\right)$$
$$274.9 = 1.973 \times 4.18 \times \left(80 - T_{h,out,old}\right)$$
$$T_{h,out,old} = 46.66°\text{C}$$

Calculate the temperature difference at the inlet and exit of the heat exchanger are

Substitute $20°\text{C}$ for $T_{c,in}$, $45°\text{C}$ for $T_{c,out,old}$, $80°\text{C}$ for $T_{h,in}$, $46.66°\text{C}$ for $T_{h,out,old}$ in the below equation

$$\Delta T_1 = T_{h,in} - T_{c,out,old}$$
$$= 80 - 45$$
$$= 35°\text{C}$$
$$\Delta T_2 = T_{h,out,old} - T_{c,in}$$
$$= 46.66 - 20$$
$$= 26.66\ °\text{C}$$

Step 10 of 12

Then logarithmic temperature difference of the counter flow heat exchanger is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{In\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$
$$= \frac{35 - 26.66}{In\left(\frac{35}{26.66}\right)}$$
$$= 30.641\ °\text{C}$$

Determine, correction factor for 2 shell and 4 tube passes as follows

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$
$$= \frac{T_{c,in} - T_{c,out,old}}{T_{h,out,old} - T_{h,in}}$$
$$= \frac{20-45}{46.66-80}$$
$$= 0.75$$

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$
$$= \frac{T_{h,out,old} - T_{h,in}}{T_{c,in} - T_{h,in}}$$
$$= \frac{46.66-80}{20-80}$$
$$= 0.555$$

From above the correction factor is for 2 shells and 4 tubes passes at $P = 0.666, R = 0.75$, is $F = 0.97$.

Step 11 of 12

Find the overall heat transfer rate

$$\dot{Q}_{old} = F \times U_{old} \times A \times \Delta T_{lm}$$
$$= F \times U_{old} \times (n \times DL) \times \Delta T_{lm}$$

Here, L is the length of a tube; D is the diameter of the thin tube

Substitute $1.5\ \text{m}$ for L, $2\ \text{cm}$ for D, 0.91 for F, $0.975\ \text{kW/m}^2\text{·K}$ for U, 24.66 for ΔT_{lm} in the above equation

$$274.9 = F \times U_{old} \times (n \times DL) \times \Delta T_{lm}$$
$$= 0.97 \times U_{old} \times (160 \times \pi \times 0.02 \times 1.5) \times 30.16$$
$$U_{old} = 0.6133\ \text{kW/m}^2$$

Step 12 of 12

Calculate magnitude of the fouling factor

$$R_f = \frac{1}{U_{old}} - \frac{1}{U}$$

Step 12 of 12

Calculate magnitude of the fouling factor

$$R_f = \frac{1}{U_{old}} - \frac{1}{U}$$

Substitute $0.6133\,\text{kW/m}^2$ for U_{old}, $0.975\,\text{kW/m}^2$ for U in the above equation

$$\begin{aligned} R_f &= \frac{1}{U_{old}} - \frac{1}{U} \\ &= \frac{1}{0.6133} - \frac{1}{0.975} \\ &= 0.0006048\,\text{K}\cdot\text{m}^2/\text{W} \end{aligned}$$

Therefore, the magnitude of the fouling factor is $\boxed{0.0006048\,\text{K}\cdot\text{m}^2/\text{W}}$

Question: 22.115P

Air at 18°C (*cp* = 1006 J/kg·K) is to be heated to 58°C by hot oil at 80°C (*cp* = 2150 J/kg·K) in a cross-flow heat exchanger with air mixed and oil unmixed. The product of heat transfer surface area and the overall heat transfer coefficient is 750 W/K and the mass flow rate of air is twice that of oil. Determine (*a*) the effectiveness of the heat exchanger, (*b*) the mass flow rate of air, and (*c*) the rate of heat transfer.

Step 1 of 5

Specific heats of hot oil and air are $c_{ph} = 2150\ \text{J/kg}\cdot{}^\circ\text{C}$ and $c_{pc} = 1006\ \text{J/kg}\cdot{}^\circ\text{C}$

Product of heat transfer surface area and overall heat transfer coefficient is

$UA = 750\ \text{W/K}$

Temperature of cold air at inlet $T_{c,in} = 18^\circ\text{C}$

Temperature of air at exit $T_{c,out} = 58^\circ\text{C}$

Hot oil inlet temperature $T_{h,in} = 80^\circ\text{C}$

Mass flow rate of air is twice that of oil

$\dot{m}_c = 2\dot{m}_h$

$\dot{m}_h = 0.5\dot{m}_c$

Type of heat exchanger is cross flow heat exchanger with air mixed and oil unmixed

Step 2 of 5

The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph}$$
$$= \dot{m}_h (2150\ \text{J/kg.°C})$$
$$= 0.5\dot{m}_c \times 2150$$
$$C_h = 1075\ \text{J/kg.°C}$$

$$C_c = \dot{m}_c c_{pc}$$
$$C_c = \dot{m}_c (1006\ \text{J/kg.°C})$$
$$C_c = 1006\dot{m}_c\ \text{J/kg.°C}$$

Therefore

$$C_{min} = C_c = 1.006\dot{m}_c$$
$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{1.006\dot{m}_c}{1.075\dot{m}_c}$$
$$c = 0.936$$

Step 3 of 5

a)

Effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$
$$= \frac{C_c(T_{c,out} - T_{c,in})}{C_c(T_{h,in} - T_{c,in})}$$
$$= \frac{58 - 18}{80 - 18}$$
$$= 0.645$$

Effectiveness of the heat exchanger is $\boxed{\varepsilon = 0.645}$

Step 4 of 5

b)

Mass flow rate of the air is calculated as follows

From NTU relations for heat exchanger for hot oil unmixed $(C_h = C_{max})$ and air mixed $(C_c = C_{min})$ the NTU expression is

$$NTU = -\frac{\ln[c\ln(1-\varepsilon)+1]}{c}$$
$$= -\frac{\ln[0.936\times\ln(1-0.645)+1]}{0.936}$$
$$NTU = 3.7236$$

NTU for this heat exchanger can also be calculated as

$$NTU = \frac{UA_s}{C_{min}}$$
$$3.7236 = \frac{750}{1006\dot{m}_c}$$
$$3.7236 = \frac{0.7455}{\dot{m}_c}$$
$$\dot{m}_c = 0.2002\ \text{kg/s}$$

Mass flow rate of the air is $\boxed{\dot{m}_c = 0.2002\ \text{kg/s}}$

Step 5 of 5

c)

Heat transfer rate is calculated as

$$\dot{Q} = \dot{m}_c c_p (T_{c,out} - T_{c,in})$$
$$= 0.2002\times 1006(58-18)$$
$$\dot{Q} = 8056.048\ \text{W}$$

Heat transfer rate is $\boxed{\dot{Q} = 8.056\ \text{kW}}$

Question: 22.116P pg 1

During an experiment, a plate heat exchanger that is used to transfer heat from a hot-water stream to a cold-water stream is tested, and the following measurements are taken:

	Hot Water Stream	Cold Water Stream
Inlet temperature, °C	38.9	14.3
Outlet temperature, °C	27.0	19.8
Volume flow rate,L/min	2.5	4.5

The heat transfer area is calculated to be 0.056 m2.

(*a*)Calculate the rate of heat transfer to the cold water.

(*b*)Calculate the overall heat transfer coefficient.

(*c*) Determine if the heat exchanger is truly adiabatic. If not, determine the fraction of heat loss and calculate the heat transfer efficiency.

(d) Determine the effectiveness and the NTU values of the heat exchanger.

Also, discuss if the measured values are reasonable.

Step 1 of 8

Measurements obtained from experiment are:

	Hot water stream	Cold water stream
Inlet temperature, °C(T_{in})	38.9	14.3
Outlet temperature, °C(T_{out})	27.0	19.8
Volume flow rate, L/min	2.5	4.5

Step 2 of 8

(a)

Determine the rate of heat transfer to the cold water

$$\dot{Q}=\dot{m}_c c_{pc}\left(T_{c,out}-T_{c,in}\right)$$

Here $\dot{m}_c$ is the Mass flow rate of cold water, c_{pc} is the specific heat cold water, and $T_{c,in}$ the inlet of temperature of cold water, $T_{c,out}$ is the exit temperature of the cold water.

Substitute 14.3 for $T_{c,in}$, 19.0 for $T_{c,out}$, 0.075 kg/s for $\dot{m}_c$, 4180 for c_{pc}, in the above equation

$$\begin{aligned}\dot{Q}_{cold}&=\dot{m}_c c_{pc}\left(T_{c,out}-T_{c,in}\right)\\&=0.075\times4180\times(19.8-14.3)\\&=1724.25\text{ J/s}\end{aligned}$$

Therefore, the rate of heat transferred to cold water is 1724.25 J/s

Step 3 of 8

Calculate LMTD of the parallel flow heat exchanger is

$$\begin{aligned}\Delta T_1&=T_{h,in}-T_{c,in}\\&=38.9-14.3\\&=24.6^\circ\text{C}\\\Delta T_2&=T_{h,out}-T_{c,out}\\&=27-19.8\\&=7.2\end{aligned}$$

Substitute above obtained values in the following relation

$$\Delta T_{lm}=\frac{\Delta T_1-\Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)}$$

$$\Delta T_{lm}=\frac{24.6-7.2}{\ln\left(24.6/7.2\right)}$$

$$\Delta T_{lm}=14.16^\circ\text{C}$$

Step 4 of 8

(b)

The rate of heat transfer in the heat exchanger,

$$\dot{Q}_{cold}=UA_s\Delta T_{lm}$$

Here, the overall heat transfer coefficient of the given counter flow heat exchanger, U

Surface area of the heat exchanger, A_s

Substitute 0.056 m^2 for A_s, 14.16°C for ΔT_{lm}, 1724.25 J/s for $\dot{Q}$ in the above equation

$$\begin{aligned}U&=\frac{\dot{Q}_{cold}}{A_s\Delta T_{lm}}\\&=\frac{1724.25}{0.056\times14.16}\\&=2174.44\text{ W/m}^2\cdot\text{K}\end{aligned}$$

Therefore, the overall heat transfer coefficient is $\boxed{2174.44\text{ W/m}^2\cdot\text{K}}$

Step 5 of 8

(c)

Determine the rate of heat transfer from hot water.

$$\dot{Q}_{hot}=\dot{m}_h c_{p,h}\left(T_{h,in}-T_{h,out}\right)$$

Here $\dot{m}_c$ is the Mass flow rate of cold water, c_{pc} is the specific heat cold water, and $T_{c,in}$ the inlet of temperature of cold water, $T_{c,out}$ is the exit temperature of the cold water.

Substitute 38.9 for $T_{h,in}$, 27.0 for $T_{h,out}$, 0.041 kg/s for $\dot{m}_h$, 4180 for c_{ph}, in the above equation

$$\begin{aligned}\dot{Q}_{hot}&=\dot{m}_h c_{p,h}\left(T_{h,in}-T_{h,out}\right)\\&=\frac{2.5\times4180\times(38.9-27)}{60}\\&=2075.58\text{ J/kg}\end{aligned}$$

Therefore, the rate of heat transferred from hot water is 2075.58 J/kg

Since heat lost by hot fluid is not equal to the heat gained by the cold fluid

Therefore, the heat exchanger is $\boxed{\text{not truly adiabatic}}$

$= 2075.58 \text{ J/kg}$ 22.116P pg 2

Therefore, the rate of heat transferred from hot water is 2075.58 J/kg

Since heat lost by hot fluid is not equal to the heat gained by the cold fluid

Therefore, the heat exchanger is **not truly adiabatic**

Calculate the fraction of heat loss by the heat exchanger

$$Q_{loss} = \frac{\dot{Q}_{hot} - Q_{cold}}{\dot{Q}_{hot}}$$
$$= \frac{2075.58 - 1724.25}{2075.58}$$
$$= 0.169$$

Therefore, the fraction of heat loss is **0.169**

Step 6 of 8

Calculate the heat transfer efficiency

$$\eta_{heat\,transfer} = \frac{\text{heat transfered to the cold fluid}}{\text{heat produced}} \times 100\%$$
$$= \frac{\dot{Q}_{cold}}{\dot{Q}_{hot}} \times 100\%$$
$$= \frac{1724.25}{2075.58} \times 100\%$$
$$= 0.831 \times 100\%$$
$$= 83.1\ \%$$

Therefore, heat transfer efficiency is **83.1 %**

Step 7 of 8

(d)

The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph}$$
$$= (0.041)(4180 \text{ J/kg.°C})$$
$$C_h = 171.38 \text{ W/°C}$$
$$C_c = \dot{m}_c C_{pc}$$
$$= (0.075 \text{ kg/s})(4180 \text{ J/kg.°C})$$
$$= 313.5 \text{ W/°C}$$

Therefore

$C_{min} = C_h = 171.38 \text{ W/°C}$

And find the capacity ratio

$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{171.38}{313.5}$$
$$= 0.54$$

Step 8 of 8

Find maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$
$$= (171.38)(38.9 - 14.3)$$
$$= 4215.94 \text{ J/s}$$

Find NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{min}}$$
$$= \frac{2174.44 \times 6.4}{171.38}$$
$$\text{NTU} = 7.105$$

Therefore, the NTU of heat exchanger is **7.105**

From table NTU relations for heat exchanger

Find effectiveness of parallel flow heat exchanger is

$$\varepsilon_p = \frac{1 - \exp\left[-\text{NTU}(1+c)\right]}{1+c}$$
$$= \frac{1 - \exp\left[-7.105(1+0.54)\right]}{1+0.54}$$
$$= 0.649$$

Therefore, the effectiveness of heat exchanger is **0.649**

Question: 22.117P

A shell-and-tube heat exchanger with 2-shell passes and 4-tube passes is used for cooling oil (cp = 2.0 kJ/kg·K) from 125°C to 55°C. The coolant is water, which enters the shell side at 25°C and leaves at 46°C. The overall heat transfer coefficient is 900 W/m2·K. For an oil flow rate of 10 kg/s, calculate the cooling water flow rate and the heat transfer area.

Step 1 of 5

Given data:

Specific heat of hot oil, $c_{ph} = 2000\ \text{J/kg°C}$

Specific heat of water, $c_{pc} = 4180\ \text{J/kg°C}$

Mass flow rates of oil is $\dot{m}_h = 10\ \text{kg/s}$

Overall heat transfer coefficient $U = 900\ \text{W/m}^2\text{°C}$

Step 2 of 5

Heat transfer rate is calculated by

$$\dot{Q} = \left[\dot{m}_h c_{ph}\left(T_{out} - T_{in}\right)\right]_{oil}$$

$$= (10\ \text{kg/s})(2000\ \text{J/kg}\cdot\text{°C})(125-55)\text{°C}$$

$$= 1400\ \text{kW}$$

The mass flow rate of water is determined to be

$$\dot{Q} = \left[\dot{m}c_{pc}\left(T_{out} - T_{in}\right)\right]_{water}$$

$$\dot{m}_{water} = \frac{\dot{Q}}{c_{pc}\left(T_{out} - T_{in}\right)}$$

$$= \frac{1400\ \text{kJ/s}}{(4.180\ \text{kJ/kg}\cdot\text{°C})(46-25)\text{°C}}$$

$$= 15.94\ \text{kg/s}$$

∴ The mass flow rate of the water, $\boxed{\dot{m}_{water} = 15.94\ \text{kg/s}}$

Step 3 of 5

Heat transfer surface area is determined to be

$$\dot{Q} = UA_s\Delta T_{lm}$$

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}}$$

Where the LMTD for this heat exchanger is determined by

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$= (125-46)\text{°C}$$

$$= 79\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$= (55-25)\text{°C}$$

$$= 30\text{°C}$$

Step 4 of 5

The proper average temperature difference between the two fluids is the logarithmic mean temperature difference which is determined from

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{79-30}{\ln(79/30)}$$

$$= 50.61\text{°C}$$

Step 5 of 5

By substituting LMTD heat transfer area is

$$A_s = \frac{1400000\ \text{W}}{(900\ \text{W/m}^2\cdot\text{°C})(50.61\text{°C})}$$

$$A_s = 30.738\ \text{m}^2$$

∴ the heat transfer surface area is $\boxed{A_s = 30.738\ \text{m}^2}$

Question: 22.118P

Consider a water-to-water counter-flow heat exchanger with these specifications. Hot water enters at 90°C while cold water enters at 20°C. The exit temperature of hot water is 15°C greater than that of cold water, and the mass flow rate of hot water is 50 percent greater than that of cold water. The product of heat transfer surface area and the overall heat transfer coefficient is 2200 W/K. Taking the specific heat of both cold and hot water to be *cp* = 4180 J/kg·K, determine (*a*) the outlet temperature of the cold water, (*b*) the effectiveness of the heat exchanger, (*c*) the mass flow rate of the cold water, and (*d*) the heat transfer rate.

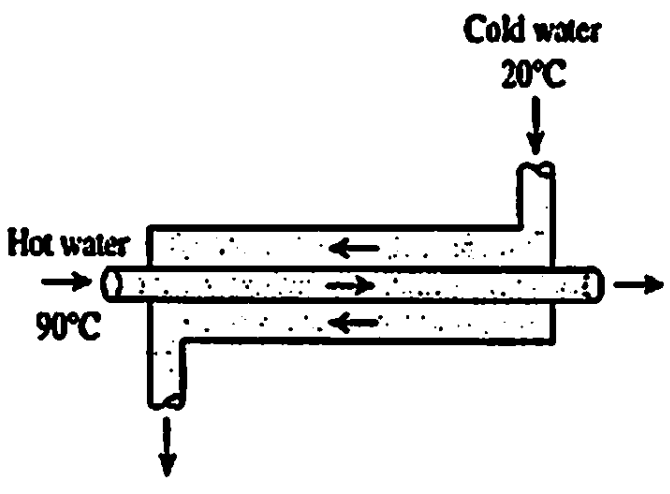


FIGURE P22–118

Step 1 of 8

Specific heat hot water, $c_{ph} = 4180$ J/kg°C

Specific heat cold water, $c_{pc} = 4180$ J/kg°C

Overall heat transfer coefficient $U = 2200$ W/K

And given condition for mass flow rats as $\dot{m}_h = 1.5\dot{m}_c$

Condition for exit temperatures of hot and cold fluids is $T_{h,out} = T_{c,out} + 15°C$

Step 2 of 8

The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h C_{ph}$$
$$= \dot{m}_h (4180 \text{ J/kg} \cdot °C)$$
$$= 1.5\dot{m}_c (4180 \text{ J/kg} \cdot °C)$$
$$= \dot{m}_c (6270 \text{ J/kg} \cdot °C)$$
$$= 6.27\dot{m}_c \text{ kJ/s} \cdot °C$$
$$C_c = \dot{m}_c C_{pc}$$
$$= \dot{m}_c (4180 \text{ J/kg} \cdot °C)$$
$$= 4.18\dot{m}_c \text{ kJ/s} \cdot °C$$

Step 3 of 8

Therefore $C_{min} = C_c$
$$= 4.18\dot{m}_c$$

And $$C = \frac{C_{min}}{C_{max}}$$
$$= \frac{4.18\dot{m}_c}{6.27\dot{m}_c}$$
$$= 0.667$$

Step 4 of 8

a) Cold fluid outlet temperature is calculated by equating the heat transfer rates of hot and cold fluids as follows

Heat transfer rate of cold fluid is

$$\dot{Q} = C_c (T_{c,out} - T_{c,in})$$
$$= (4.18\dot{m}_c)(T_{c,out} - 20) \quad \ldots\ldots (1)$$

Heat transfer rate of hot fluid is

$$\dot{Q} = C_h (T_{h,in} - T_{h,out})$$
$$= (6.27\dot{m}_c)\left[90 - (T_{c,out} + 15)\right]$$
$$= (6.27\dot{m}_c)(75 - T_{c,out}) \quad \ldots\ldots (2)$$

Step 5 of 8

By equating equations (1) & (2) heat transfer rates of hot and cold fluids

$$4.18\dot{m}_c (T_{c,out} - 20) = 6.27\dot{m}_c (75 - T_{c,out})$$
$$4.18\dot{m}_c T_{c,out} - 83.6\dot{m}_c = 470.25\dot{m}_c - 6.27\dot{m}_c T_{c,out}$$
$$4.18\dot{m}_c T_{c,out} + 6.27\dot{m}_c T_{c,out} = 553.85\dot{m}_c$$
$$10.45\dot{m}_c T_{c,out} = 553.85\dot{m}_c$$
$$T_{c,out} = 53°C$$

∴ The outlet temperature of the cold fluid, $\boxed{T_{c,out} = 53°C}$

Step 6 of 8

b)

Effectiveness of the heat exchanger is determined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$
$$= \frac{C_c (T_{c,out} - T_{c,in})}{C_c (T_{h,in} - T_{c,in})}$$
$$= \frac{4.18\dot{m}_c (53 - 20)}{4.18\dot{m}_c (90 - 20)}$$
$$= 0.47143$$

∴ The effectiveness of the heat exchanger, $\boxed{\varepsilon = 0.47}$

Step 7 of 8

c)

Mass flow of the cold water is calculated as follows

NTU of the heat exchanger is

$$NTU = \frac{1}{C-1} \ln\left(\frac{\varepsilon - 1}{\varepsilon C - 1}\right)$$
$$= \frac{1}{0.667 - 1} \ln\left(\frac{0.47143 - 1}{0.47143 \times 0.667 - 1}\right)$$
$$= 0.78094$$

NTU of the heat exchanger can also determined as

$$NTU = \frac{UA_s}{C_{min}}$$
$$0.7809 = \frac{2.2 \text{ kW/°C}}{4.18\dot{m}_c}$$
$$\dot{m}_c = 0.6739 \text{ kg/s}$$

∴ The mass flow rate of cold water, $\boxed{\dot{m}_c = 0.674 \text{ kg/s}}$

Step 8 of 8

d)

Heat transfer rate is calculated as

$$\dot{Q} = C_{min} (T_{c,out} - T_{c,in})$$
$$= (0.674 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(53 - 20)°C$$
$$= 92.97156 \text{ kW}$$

∴ The rate of heat transfer, $\boxed{\dot{Q}_{max} = 92.97 \text{ kW}}$

Question: 22.119P

A single-pass cross-flow heat exchanger with both fluids unmixed has water entering at 16°C and exiting at 33°C, while oil (cp = 1.93 kJ/kg·K and ρ = 870 kg/m3) flowing at 0.19 m3/min enters at 38°C and exits at 29°C. If the surface area of the heat exchanger is 20 m2, determine (*a*) the NTU value and (*b*) the value of the overall heat transfer coefficient.

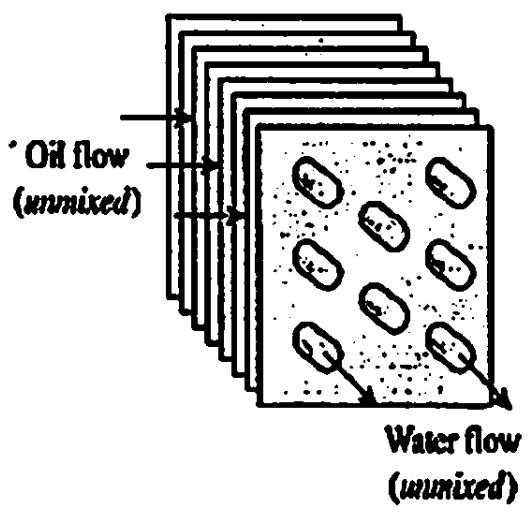


FIGURE P22–119

Step 1 of 8

A single pass cross flow heat exchanger

Find heat transfer rate for oil;

$$\dot{Q} = \dot{m}_{oil} c_{p,oil} \left(T_{oil,in} - T_{oil,out}\right)$$

$$= \frac{\dot{m}_{oil}}{\rho} c_{p,oil} \left(T_{oil,in} - T_{oil,out}\right)$$

Here, Mass flow rate of oil, $\dot{m}_{oil}$, the specific heat of oil, $c_{p,oil}$, Inlet temperature of oil, $T_{oil,in}$, Exit temperature of oil, $T_{oil,out}$

Substitute $0.19\,m^3/min$ for m, $870\ kg/m^3$, $1.93\,kJ/kg\cdot K$ for c_p, $(38-29)°C$ for ΔT in the above equation.

$$\dot{Q} = \frac{\dot{m}_{oil}}{\rho} c_{p,oil} \left(T_{oil,in} - T_{oil,out}\right)$$

$$= \frac{0.19\times 870}{60} \times 1.93 \times (38-29)$$

$$= 47854\ W$$

Determine the mass flow rate of the water from by using above heat transfer rate

$$\dot{Q} = \dot{m}_{water} c_{p,oil} \left(T_{water,out} - T_{water,in}\right)$$

$$47854\ W = \dot{m}_{water} \times 4180 \times (33-16)$$

$$\dot{m}_{water} = \frac{47854}{4180\times 17}$$

$$= 0.673\ kg/s$$

Step 2 of 8

Determine heat capacity rate of water fluid is,

$$C_c = \dot{m}_{water} c_{p,water}$$

Here, Specific heat of cold water, $c_{p,water}$

Mass flow rate of cold water, $\dot{m}_{water}$

Substitute $4180\ J/kg\cdot K$ for c_{pc}, $3.16\ kg/s$ for $\dot{m}_{water}$ in the above equation

$$C_c = \dot{m}_{water} c_{p,water}$$

$$= (0.673\ kg/s)(4180\ J/kg\cdot K)$$

$$= 2814.94\ W/K$$

Step 3 of 8

Determine heat capacity rate of oil,

$$C_h = \dot{m}_{oil} c_{p,oil}$$

Substitute $1.93\,kJ/kg\cdot K$ for c_p, $2.755\ kg/s$ for $\dot{m}_h$ in the above equation

$$C_h = \dot{m}_h C_{ph}$$

$$= (2.755\ kg/s)(1930\ J/kg\cdot K)$$

$$= 5317.5\ W/K$$

Step 4 of 8

Then

$$C_{min} = C_c = 2814.94\ W/K$$

And capacity ratio

$$c = \frac{C_{min}}{C_{max}}$$

$$= \frac{2814.94}{5317.5}$$

$$c = 0.529$$

Step 5 of 8

Determine the maximum heat transfer rate

$$\dot{Q}_{max} = C_{min} \times (\Delta T)_{max}$$

$$= C_{min} \times \left(T_{oil,in} - T_{water,in}\right)$$

$$= 2814.94 \times (38-16)$$

$$= 61928.68\ W$$

Find effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$

$$= \frac{47854\ W}{61928.68\ W}$$

$$= 0.772$$

$$= 77.27\%$$

Step 6 of 8

(a)

Determine the number of heat transfer units (NTU) using below relation for single pass cross flow heat exchanger with both fluids unmixed

Use the cross flow with both fluids unmixed chart

For effectiveness $\varepsilon = 77.27\,\%$ and capacity ratio $c = 0.529$

$$NTU = 2.39$$

Therefore, NTU value is $\boxed{2.39}$

Step 7 of 8

(b)

Now determine the overall heat transfer units using the below relation of NTU

$$NTU = \frac{A_s U}{C_{min}}$$

Here, A_s is the surface area of heat exchanger

Step 8 of 8

Substitute $20\,m^2$ for A_s, 2.39 for NTU, $2814.94\ W/K$ for C_{min} in the below relation

$$U = \frac{NTU \times C_{min}}{A_s}$$

$$= \frac{2.39\times 2814.94}{20}$$

$$= 336.38\,W/m^2\cdot K$$

Therefore, the overall heat transfer rate is $\boxed{336.38\,W/m^2\cdot K}$

Question: 22.120P

Saturated water vapor at 100°C condenses in the shell side of a 1-shell and 2-tube heat exchanger with a surface area of 0.5 m2 and an overall heat transfer coefficient of 2000 W/m2-K. Cold water (*cpc* = 4179 J/kg·K) flowing at 0.5 kg/s enters the tube side at 15°C, determine the outlet temperature of the cold water and the heat transfer rate for the heat exchanger.

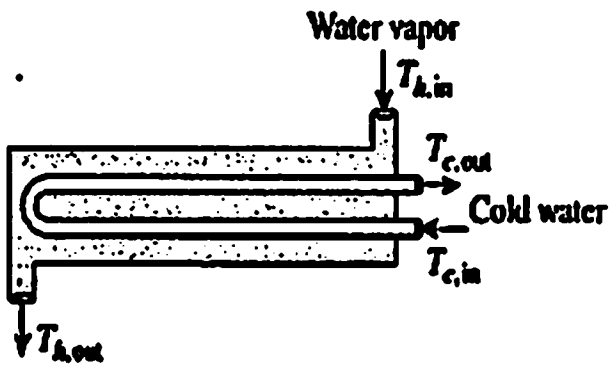


FIGURE P22–120

Step 1 of 4

Temperature of saturated water $T_{h,in} = 100°C$

Mass flow rates of water is $\dot{m}_{water} = 0.5\ kg/s$

Heat transfer surface area is $A_s = 0.5\ m^2$

Overall heat transfer coefficient $U = 2000\ W/m^2 \cdot K$

Temperature of cold water $T_{c,in} = 15°C$

Step 2 of 4

Heat capacity rate of the cold water is

$$C_c = \dot{m} c_{pc}$$

$$C_c = 0.5 \times 4179$$

$$C_{min} = C_c = 2089.5\ W/K$$

And $C = 0$

The maximum heat transfer rate becomes

$$\dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in})$$

$$= (2.0895\ kW/°C)(100°C - 15°C)$$

$$\dot{Q}_{max} = 177.6\ kW$$

NTU of the heat exchanger is

$$NTU = \frac{UA_s}{C_{min}}$$

$$= \frac{(2\ kW/m^2.°C)\ (0.5\ m^2)}{2.0895\ kW/°C}$$

$$NTU = 0.478$$

(a) Effectiveness of this heat exchanger

$$\varepsilon = 1 - \exp(-NTU)$$

$$= 1 - \exp(-0.478)$$

$$\boxed{\varepsilon = 0.38}$$

Step 3 of 4

(b) The outlet temperature of cold water

The actual heat transfer rate calculated as

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$

$$\dot{m} c_p (T_{c,out} - T_{c,in}) = \varepsilon \dot{Q}_{max}$$

$$0.5(4.179)(T_{c,out} - 15) = 0.38(177.6)$$

$$T_{c,out} = 47.3°C$$

The outlet temperature of cold water is $\boxed{T_{c,out} = 47.3°C}$

Step 4 of 4

(c) The actual heat transfer rate calculated as

$$\dot{Q} = \varepsilon \dot{Q}_{max}$$

$$\dot{Q} = 0.38(177.6)$$

$$= 67.49\ kW$$

The actual heat transfer rate $\boxed{\dot{Q} = 67.49\ kW}$

Question: 22.121P

Oil in an engine is being cooled by air in a cross-flow heat exchanger, where both fluids are unmixed. Oil (*cph* = 2047 J/kg·K) flowing with a flow rate of 0.026 kg/s enters the tube side at 75°C, while air (*cpc* = 1007 J/kg·K) enters the shell side at 30°C with a flow rate of 0.21 kg/s. The overall heat transfer coefficient of the heat exchanger is 53 W/m2·K and the total surface area is 1 m2. If the correction factor is *F* = 0.96, determine the outlet temperatures of the oil and air.

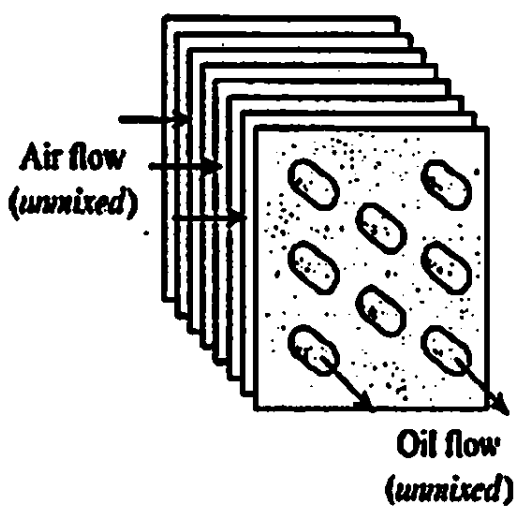


FIGURE P22–121

Step 1 of 6

Find the heat capacity rate of hot fluid by using the expression,

$$C_c = \dot{m}_c c_{pc}$$

Here, $\dot{m}_c$ is mass flow rate of cold fluid (air) and c_{pc} is specific heat of cold fluid.

Substitute 0.21kg/s for $\dot{m}_c$ and 1007 J/kg·K for c_{pc}

$$C_c = (0.21)(1007)$$
$$= 211.47 \text{ W/K}$$

Similarly, find the heat capacity rate of cold fluid by using the expression,

$$C_h = \dot{m}_h c_{ph}$$

Here, $\dot{m}_h$ is mass flow rate of hot fluid (oil) and c_{pc} is specific heat of hot fluid.

Substitute 0.026kg/s for $\dot{m}_h$ and 2047 J/kg·K for c_{ph}

$$C_h = (0.026)(2047)$$
$$= 53.22 \text{ W/K}$$

Therefore, minimum capacity heat rate is

$$C_{min} = C_h = 53.22 \text{ W/K}$$

Step 2 of 6

Find the capacity heat ratio by using the ratio,

$$c = \frac{C_{min}}{C_{max}}$$
$$= \frac{C_h}{C_c}$$
$$= \frac{53.22}{211.47}$$
$$= 0.2517$$

Step 3 of 6

Find the maximum heat transfer rate by using the equation,

$$\dot{Q}_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right)$$

Here, $T_{c,in}$ is inlet temperature of cold fluid and $T_{h,in}$ is inlet temperature of hot fluid.

Substitute 75°C for $T_{h,in}$ and 30°C for $T_{c,in}$

$$\dot{Q}_{max} = 53.22 \times (75 - 30)$$
$$= 2394.9 \text{ W}$$

Find the NTU of the heat exchanger by using the expression,

$$\text{NTU} = \frac{UA_s}{C_{min}}$$

Here, U is overall heat transfer coefficient and A_s is total surface area of the heat exchanger.

Substitute 53 W/m^2-K for U and 1m^2 for A_s

$$\text{NTU} = \frac{53 \times 1}{53.22}$$
$$= 0.9958$$

Step 4 of 6

Then find the effectiveness of the heat exchanger when both fluids unmixed by using the expression,

$$\varepsilon = 1 - \exp\left\{\frac{\text{NTU}^{0.22}}{c}\left[\exp\left(-c\text{NTU}^{0.78}\right) - 1\right]\right\}$$
$$= 1 - \exp\left\{\frac{0.9958^{0.22}}{0.2517}\left[\exp\left(-0.2517 \times 0.9958^{0.78}\right) - 1\right]\right\}$$
$$= 0.585$$

Find the actual heat transfer by using the effectiveness relation,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}$$
$$\dot{Q} = \varepsilon \dot{Q}_{max}$$
$$= 0.585 \times 2394.9$$
$$= 1401.016 \text{ W}$$
$$= 1.401 \text{ kW}$$

Step 5 of 6

Therefore, determine the outlet temperatures of the cold and hot fluids using the equation,

$$\dot{Q} = C_c\left(T_{c,out} - T_{c,in}\right)$$

Here, $T_{c,out}$ is outlet temperature of cold fluid.

Substitute the known values in the above equation,

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c}$$
$$= 30 + \frac{1.401 \times 10^3}{211.47}$$
$$= 36.62°\text{C}$$

Therefore, temperature of the hot fluid is $\boxed{36.62°\text{C}}$

Step 6 of 6

Similarly temperature of the cold fluid by using the expression,

$$\dot{Q} = C_h(T_{h,in} - T_{h,out})$$

Here, $T_{h,out}$ is temperature of hot fluid

Substitute the known values in the above equation,

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h}$$
$$= 75 - \frac{1.402 \times 10^3}{53.22}$$
$$= 48.656°\text{C}$$

Therefore, outlet temperature of hot fluid is $\boxed{48.656°\text{C}}$