University of Jordan
Faculty of Engineering & Technology
Mechanical Engineering Dept.

Thermal and Fluid Sciences Lab.

Dr. Adnan Jaradat

Experiment 1: *The Liquid-Vapor Saturation Curve*

Report ☐ *Full* ☐ *Short*

Student Name:--

Student No. :--

The liquid-Vapor Saturation Curve

* *Objectives*

1. To show that the pressure and temperature are dependent properties in the saturation region, and to draw the relation between them in that region.
2. To compare the experimental results with theoretical values taken from the steam tables.

* *Theory*

It can be shown that:

$$\boxed{\frac{(dT)}{(dP)_{sat}} = \frac{Tv_{fg}}{h_{fg}}}$$

Where

$V_{fg} = V_g - V_f$

V_g = Specific volume of saturated vapor (m^3/kg)

v_f = Specific volume of saturated liquid (m^3/kg)

$h_{fg} = h_g - h_f$ (kJ/kg)

h_g = Specific enthalpy of saturated vapor (kJ/kg)

h_f = Specific enthalpy of saturated liquid (kJ/kg)

T = Absolute temperature (K) $\rightarrow$ $\boxed{K = 273.15 + C^{\circ}}$

And $\frac{(dT)}{(dP)_{sat}}$ is the slope of the pressure – temperature curve for saturated steam in equilibrium with saturated liquid.

The above equation is known as the Clausius-Claperon Equation. The P-T curve can be drawn from experimental data and the above theoretical relationship can be investigated using the steam tables.

* ## *Apparatus*

Marcet boiler: It consists of a small cylindrical boiler fitted with a thermometer and a pressure gauge.

* ## *Procedure*

(1) Open the level tap at the side of the boiler, and remove the filler plug.

(2) Fill the boiler with clean water through the filler plug until water reaches the level of the tap.

(3) Replace the filler plug and tighten (leave tap open so that the air can escape during heating).

(4) Insert the thermometer into the hole in the filler plug.

(5) Place the beaker directly under the level tap.

(6) Plug the unit into wall socket.

(7) Wait until the water is boiling then close level tap (this should take about twenty minutes).

Note : ON NO ACCOUNT should this tap be reopened during the experiment.

(8) As the pressure rises take simultaneous readings of pressure and temperature at pressure intervals of one bar, to a maximum pressure of 2 bars.

(9) Unplug the unit from the wall socket, and as the pressure drops take simultaneous readings of pressure and temperature at intervals of one bar until the pressure-gauge readings reduce to zero.

Note : The pressure will fall very quickly for the first four or five bars.

(10) Note the barometric pressure and complete the table below.

* Data

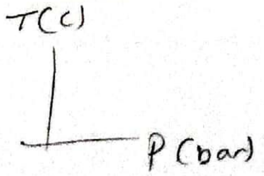
Barometric pressure = --

Sample no	steam Gauge pres. Bar	Absolute Pressure (Absolute = gauge + Bar) Bar	Saturation Temperature: Increasing pressure Temp. °C	Saturation Temperature: Decreasing pressure Temp. °C	Mean Temp. °C → K
1	0				
2	1				
3	2				
4	3				
5	4				
6	5				
7	6				
8	7				
9	8				
10	9				
11	10				
12	11				
13	12				

* Results and Discussions

(1) From the experimental results, draw the relation between the temperature and the pressure in the saturation region during increasing and decreasing the pressures.

(2) Compare the experimental results with theoretical values taken from the steam table.



(3) Make a neat skitch of the apparatus.

* Conclusions

exp. 3 slope

(1) Measure the slope of the pressure-temperature curve at three different pressure.

(2) Calculate corresponding value of the slope from the steam tables and compare the two.

theo

(3) Suggest reasons for any differences.

Thermodynamic Properties from the Steam Table

Pressure P bar	Temperature T °C	V_{fg} m^3/kg	h_{fg} kJ/kg
1.0	99.0	1.694	2258
2.0	120.2	0.8856	2202
3.0	133.5	0.6057	2164
4.0	143.6	0.4623	2134
5.0	151.8	0.3748	2109
6.0	158.8	0.3156	2087
7.0	165.0	0.2728	2067
8.0	170.4	0.2403	2048
9.0	175.1	0.2149	2031
10.0	179.9	0.1944	2015
11.0	184.1	0.1774	2000
12.0	188.0	0.1632	1986
13.0	191.6	0.1512	1972
14.0	195.0	0.1408	1960
15.0	198.3	0.1317	1947
16.0	201.4	0.1237	1935
17.0	204.3	0.1167	1923
18.0	207.1	0.1104	1912
19.0	209.8	0.1047	1901
20.0	212.4	0.09957	1890

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Experiment 2: *Flow Through A Nozzle*

Report ☐ *Full* ☐ *Short*

Student Name:--

Student No. :--

Flow Through A Nozzle

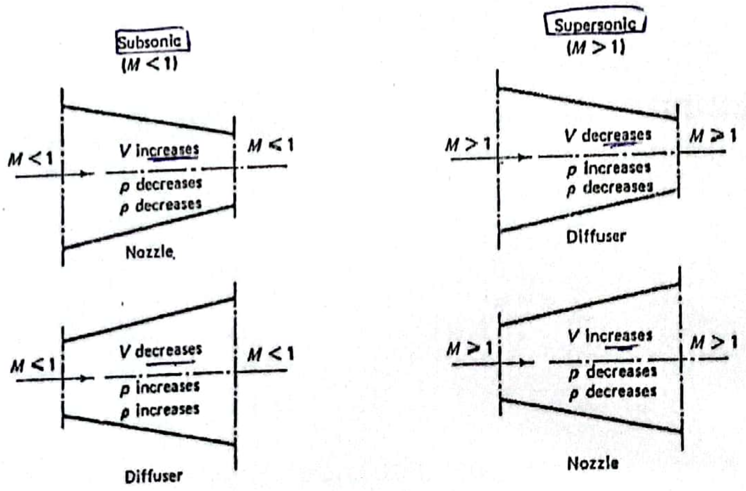
* *Objectives*

To study the pressure distribution through a nozzle for different exit pressures and different flow rates.

* *Introduction*

A nozzle is a steady state-flow device, whose purpose is to create a high-velocity fluid stream at the expense of its pressure. Nozzles are commonly utilized in jet engines, rockets, spacecrafts, and even garden hoses. The cross-sectional area of a nozzle decreases in the flow direction for subsonic flows, and increases for supersonic flows, see Figure 1.

The rate of heat transfer between the fluid flowing through a nozzle and the surroundings is usually very small ($Q^{\bullet} \cong 0$). There is little or no change in potential energy ($\Delta pe \cong 0$), and the process involves no work ($W^{\bullet} = 0$).



$v \propto \frac{1}{\rho}$

Figure 1 Variation of velocity, pressure, and density due to area change for subsonic and supersonic flows.

Theory

The flow of ideal gas through three different nozzles is shown in Figure 2. The nozzle discharges into a plenum chamber, in which the pressure is P_b can be regulated. Let P_e be the exit pressure just at the exit cross-section of the nozzle. When P_b is reduced, gas is drawn through the nozzle. As P_b is reduced more, the mass flow rate of gas increases and the velocity increases. The value of the velocity is the highest at the minimum area, and this can't be higher than the critical value i.e when the velocity reaches the velocity of sound. At this state the exit pressure (P_e) is at it's critical value, P^*, which is given by:

$$\frac{P^*}{P_0} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

Where

P^*	is the critical pressure
P_0	is the stagnation pressure
γ	is the specific heat ratio

for air $\gamma = 1.4$ thus

$P^* = 0.528\ P_0$

For conditions other than the critical condition, the velocity at the throat is given by:

$$\upsilon_t = \sqrt{\frac{2\gamma RT_0}{\gamma-1}\left[1-\left(\frac{P_t}{P_0}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

Where T_0 and P_0 are the temperature and pressure in the nozzle chest, and P_t is the pressure at the throat.

While the mass flow rate at the throat is given by:

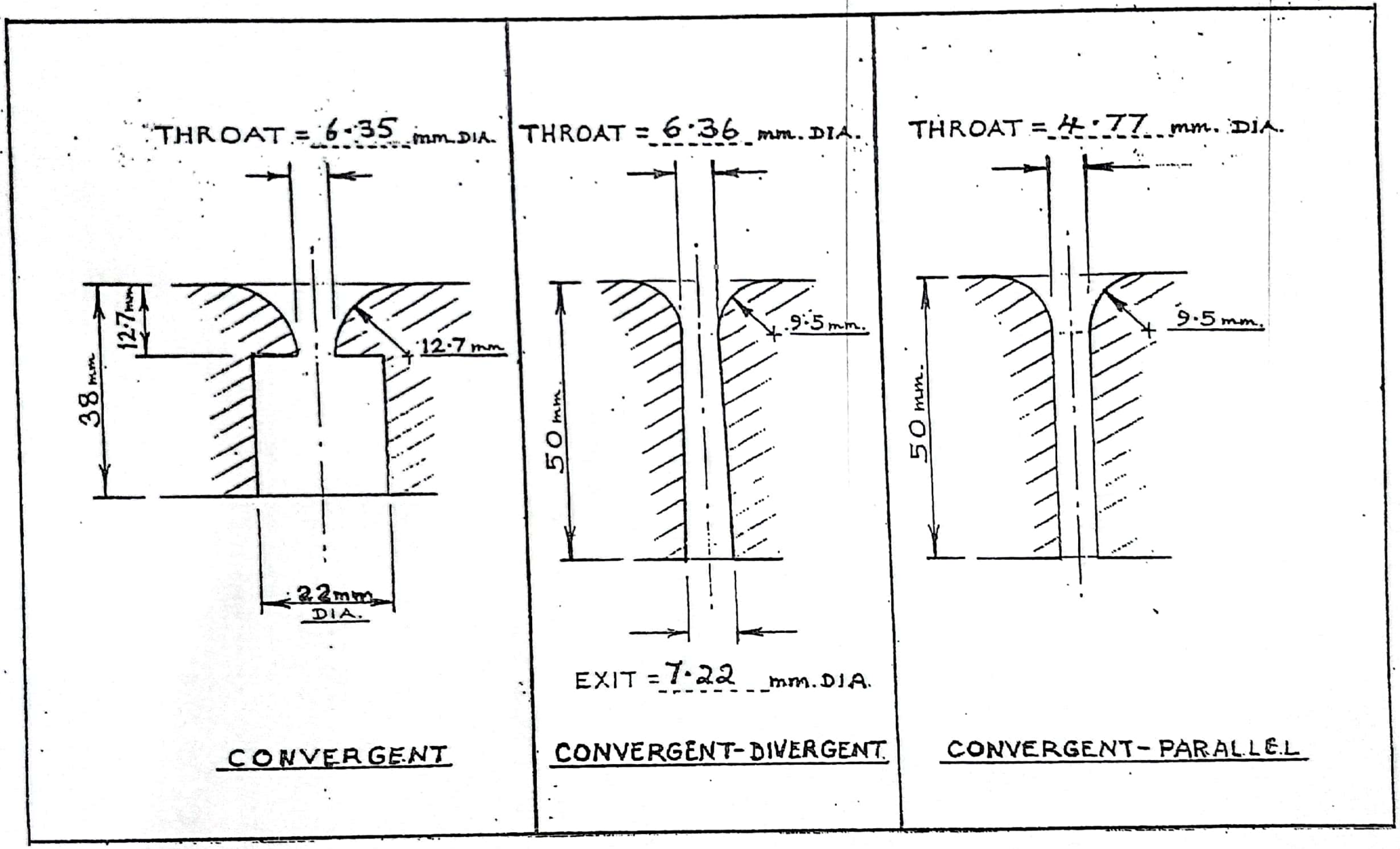


Figure 2 Nozzle profiles.

$$\dot{m}_t = A_t P_0 \left(\frac{P_t}{P_0}\right)^{1/\gamma} \sqrt{\frac{2\gamma}{(\gamma-1)RT_0}\left[1-\left(\frac{P_t}{P_0}\right)^{\frac{\gamma-1}{\gamma}}\right]} \quad kg/s$$

In this relation the pressure is in kPa and R = 0.287 kJ/kgK

T = 293 K

* Apparatus

A schematic layout of the apparatus is shown in Figure 3. Air is admitted to a cast iron pressure chest by way of adjustable vales. A nozzle of highly finished brass is screwed into a seating in the base of the chest and the air or steam expands through the nozzle.

To enable the pressure distribution through the nozzle to be plotted, a search tube or probe of stainless steel may be traversed along the axis of the nozzle. A small cross hole in the search tube connects with a high grade pressure gauge which registers the pressure at any point in the nozzle. The search tube is traversed by rotating a calibrated dial and pressures are usually recorded at intervals of 2.5 mm. A pointer moves with the search tube past a replica of the nozzle profile in order to indicate the point in the nozzle at which the pressure is being measured. The nozzle discharges into a vertical pipe of large bore fitted with a throttling valve for controlling the downstream pressure. Other instruments include a second pressure gauge for recording the pressure in the chest (P_0) and a thermometer for indicating the temperature of air or steam in the chest.

Technical Data

The throat diameter is as follows:

d_N	For the convergent nozzle	= 6.35 mm	
	For the convergent→divergent nozzle	= 6.36 mm	
d_N	For the convergent→parallel nozzle	= 4.77 mm	→ 0.00477 m
d_p	Probe diameter	= 3.33 mm	→ 0.00333 m

$$A_t = \frac{\pi}{4}\left(d_n^2 - d_p^2\right) =$$

throught area

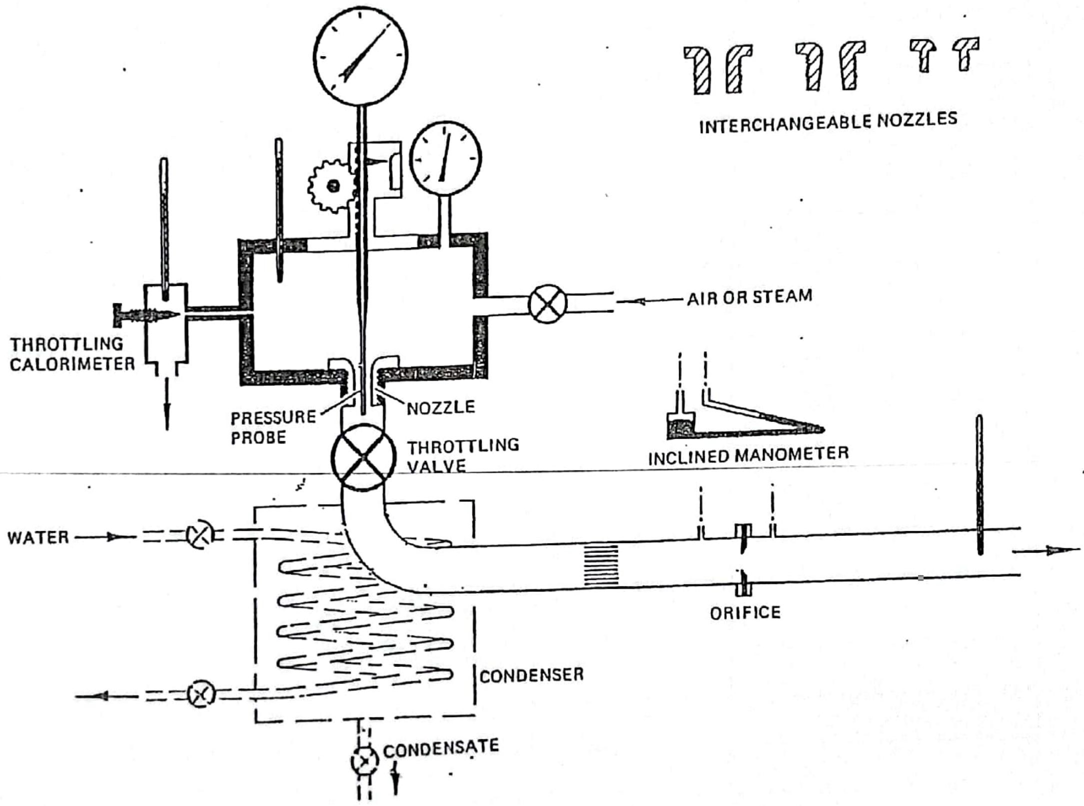


Figure 3 Schematic layout of the nozzle flow apparatus.

* *Procedure*

(1) Open back pressure valve. Raise probe to no. 1 position.

(2) Set inlet prèssure to 400 kPa. Inlet pressure and chest pressure are to be equal. Chest pressure to be observed throughout experiment and re-adjusted to initial setting if necessary.

(3) Record probe pressure at each of the stations shown on the nozzle replica.

(4) Repeat for other values of back pressure (pressure in the exit pipe). Check that the pressure at the throat is not be lower than that for the condition of choking (critical condition).

* ***Data collected***

Atmospheric pressure = 90 kPa Atmospheric temperature = 20 °C

Position No.	X/L	P_0 = 200 kPa Position Pressure kPa	P_0= 300 kPa Position Pressure kPa	P_0= 400 kPa Position Pressure kPa
7	0.0			
8				
9				
10				
11	1.0			
12				
13				
14				
15	2			
16				
17				
18				
19	3			
20				
21				
22				
23	4			
24				
25				
26				
27	5			
28				
29				
30				
31	6			
32				

Note that

position no. (7) is the entrance to the nozzle

position no. (11) is the location of the throat

position no. (27) is the end of the parallel section

* *Results and discussions*

Calculate the throat area after deducting that of the probe.

For different values of pressure ratio across the nozzle, plot:

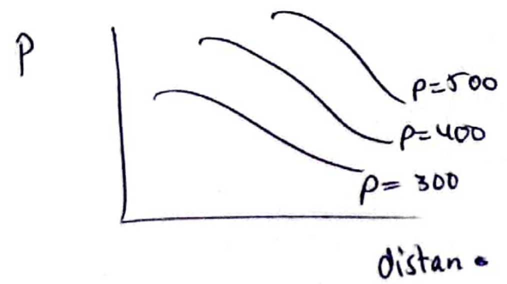
(1) the pressure variation with the distance along the nozzle (absolute values).

(2) The mass flow rate with the pressure ratio.

Note: $\boxed{\text{the pressure ratio = back pressure / chest pressure}}$

$$= \frac{atm\,p}{atmp \pm P_0}$$

1



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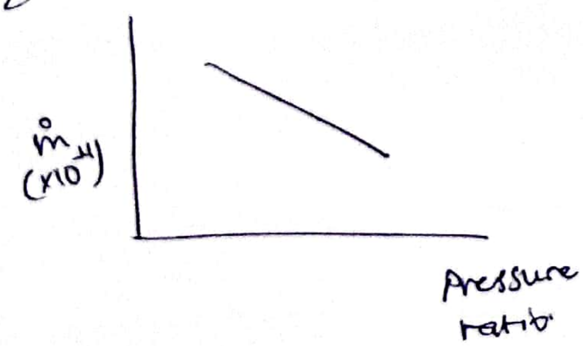

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Experiment 3: *Losses In Pipes*

Report ☐ *Full* ☐ *Short*

Student Name:--

Student No. :--

Losses In Pipes

* *Objectives*

1. To determine the variation of friction factor with Reynolds Number for a pipe.
2. To determine the loss coefficient for several fittings.

* *Theory*

The energy loss in internal flows (laminar or turbulent) is due to :

h_f (a) Friction losses due to the shear stress on the wall of the duct. These losses are often called the "*Major Losses*". The head loss occurs from fiction effects is designated "h_f".

h_m (b) *Minor Losses*, which are the losses that the flow experiences at the pipe inlet and exit in addition to the losses caused by valves, bends, elbows, sudden and gradual contractions and expansions. The head loss occurs from each of the above is h_m and Σh_m is designated for all minor losses.

Both losses, major and minor, are expressed in a head loss h_L in the energy equation as shown below

$$\frac{P_1}{\gamma}+\frac{V_1^2}{2g}+Z_1+h_p=\frac{P_2}{\gamma}+\frac{V_2^2}{2g}+Z_2+h_T+h_L \rightarrow \Sigma h_f+\Sigma h_m$$

Where

head loss $\leftarrow$ $\boxed{h_L = h_f + \Sigma h_m}$

$\boxed{h_p = \text{Pump head}}$

$\boxed{h_T = \text{Turbine head}}$

Major Losses

It is found convenient to express the head loss due to friction effects for internal flows, circular or non circular pipes, smooth or rough surfaces as laminar → Re ≤ 2300 turbulent → Re > 4000

2300 ≤ Re ≤ 4000
transitional

$$\boxed{h_f = f\frac{L}{D}\frac{V_m^2}{2g}}$$

Major losses

Where

L and D are the length and diameter of the pipe

V_m is the mean velocity of the fluid.

f is the friction factor (also called the Darcy friction factor)

$Re = \frac{\text{internal forces}}{\text{Viscous Force}} = \frac{V_{avg} D}{\nu} = V_{avg} \cdot D \cdot \rho / \mu$

For laminar flow (i.e Reynolds number, Re < 2300)

$$f = \frac{64}{Re}$$ ← Dete → obj.

For turbulent flow the friction factor depends on Reynolds number and the relative roughness ε/D, which is the ratio of the mean height of roughness of the pipe to the pipe diameter.

Ks ε

Experimental results of this dependence are presented in the relation known as the "Colebrook equation":

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right)$$

$$f = \frac{0.25}{\left[\log_{10}\left(\frac{\varepsilon_s}{3.7D} + \frac{5.74}{Re^{0.9}}\right)\right]^2}$$

This form was plotted into the famous "Moody chart".

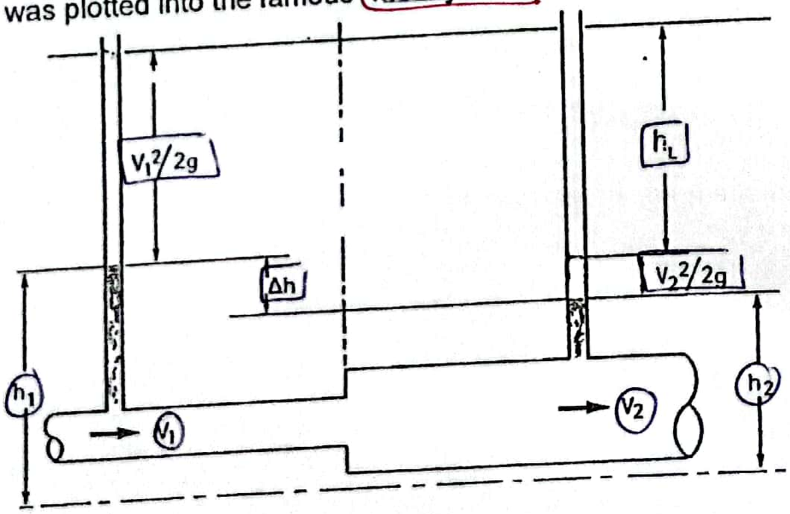

Figure 1 Pressure tapping upstream and downstream of a sudden expansion.

Minor Losses

LBC

In general it is not possible to use simple theory to predict the head loss due to fittings such as bends and valves. The loss for any particular fitting therefore has to be determined by experiment. In order to avoid errors due to the disturbance in the flow close to the fitting, the head loss is measured using pressure tapping placed several pipe diameters upstream and downstream. Considering a general case as shown in Figure 1, in which the upstream and downstream pipe diameters are different, the Bernoulli's equation can be written as

$$\frac{P_1}{\gamma}+Z_1+\frac{V_1^2}{2g}=\frac{P_2}{\gamma}+Z_2+\frac{V_1^2}{2g}+h_L$$

if $\quad h=\frac{P}{\gamma}+Z \quad$ then

$$\boxed{h_1+\frac{V_1^2}{2g}=h_2+\frac{V_2^2}{2g}+h_L}$$

Where h_L is the total head loss between the pressure tapping.

The total head loss has two components:

a- Friction loss h_f in the upstream and downstream pipes.

b- Minor head loss h_m due to the fitting alone.

Re-writing Bernoulli's equation and noting that $(h_1 - h_2)$ is the measured head loss Δh recorded by the manometer, we have

zero

$$h_m=\Delta h+\left[\frac{V_1^2-\cancel{V_2^2}}{2g}\right]-h_f$$

$V_1 \simeq V_2$

In order to obtain the head loss due to fittings we therefore have to correct the measured head loss for the change in velocity head and also subtract the head loss due to friction. If the upstream and downstream diameters are the same, then $V_1 = V_2$ and we have

$$h_m = \Delta h - h_f$$

The minor head loss is usually expressed in terms of the loss coefficient K defined

$$K = \frac{h_m}{V^2 / 2g}$$

Where V is the velocity in the smaller pipe (in this case V_1)

* *Apparatus*

The apparatus shown diagrammatically in Figure 2, consists of two separate hydraulic circuits each one containing a number of pipe system components. Both circuits are supplied with water from the same hydraulic bench. The components in each of the circuits are as follows

Dark Blue Circuit (DBC)

1. Gate valve (D)
2. Standard Elbow Bend (C)
3. 90° Miter Bend (B)
4. Straight pipe

R

radius = 12.7 mm

length = 914.4 mm

Small diameter = 13.6 mm d

Large diameter = 26.2 mm D

Pipe material is copper

$\varepsilon = 0.0015$ mm

Light Blue Circuit (LBC)

1. Globe Valve (K)
2. Sudden Expansion (E)
3. Sudden Contraction (F)

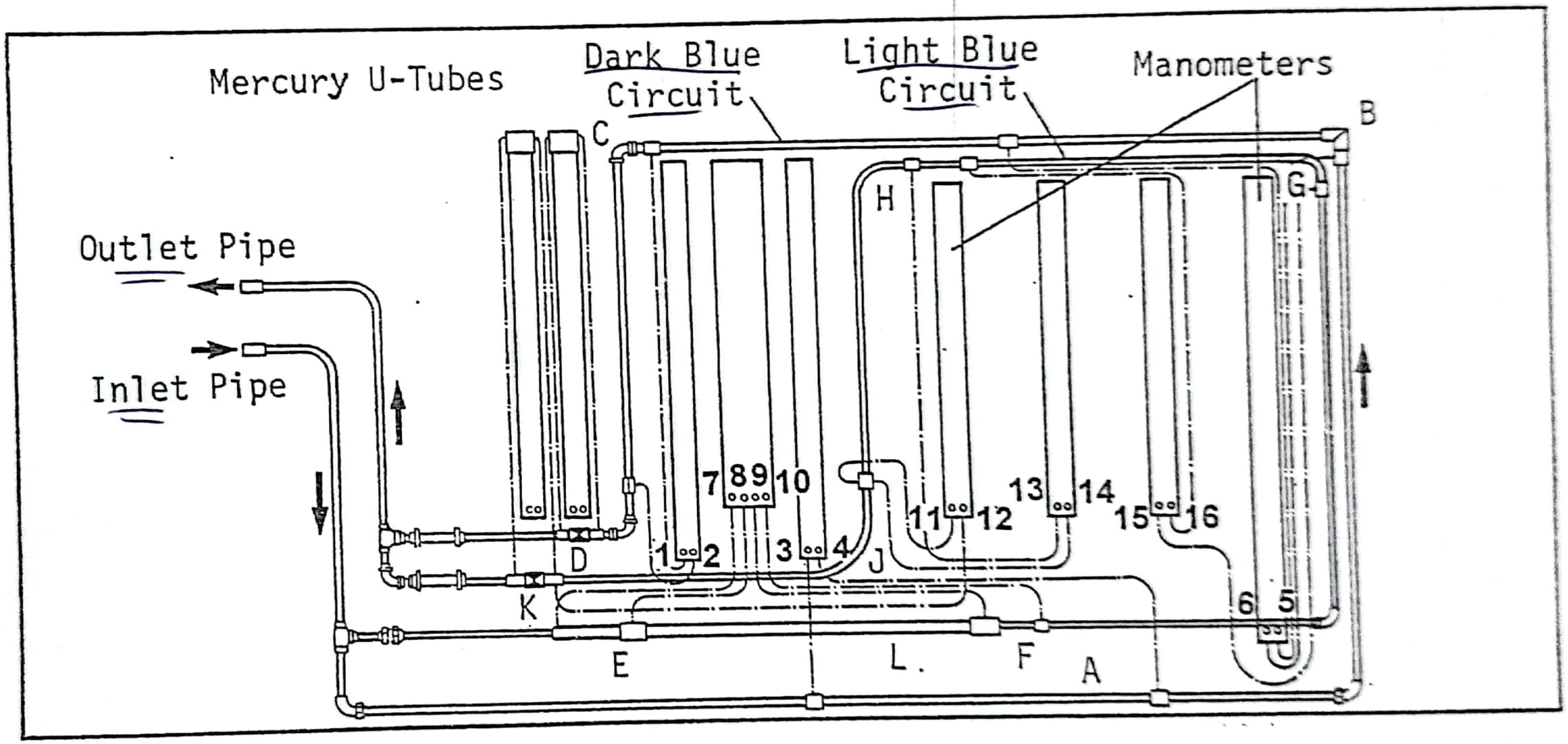

Figure 2 The Apparatus

4. 150 mm radius 90° bend (J) , R/d = 11.1

5. 100 mm radius 90° bend (H) , R/d = 7.4

6. 60 mm radius 90° bend (G) , R/d = 3.7

In all cases (except the gate and globe valves) the pressure change across each of the components is measured by a pair of pressurized piezometer tubes. In the case of the valves pressure measurement is made by U-tube manometer containing mercury.

* *Procedure*

(1) Close the globe valve K and open the gate valve D, see Figure 2. Switch on the bench pump and open the bench supply valve to admit water to the dark blue circuit. Allow water to flow for 2 to 3 minutes.

(2) Close the gate valve D and bleed all of the trapped air into the top of the manometers tubes. Check that all the manometers show zero pressure difference.

(3) Open the gate valve and then, by carefully opening the bled screws at the top of the mercury U tube, fill each limb with water. Make sure that all air bubbles have been expelled, then close the bleed screws.

(4) Close the gate valve, open the globe valve, and repeat the procedure ~~for the light blue circuit.~~

(5) Open fully the bench supply valve. Then close the globe valve and open fully the gate valve to obtain the maximum flow rate through the dark blue circuit.

(6) If necessary, adjust the water levels in the manometers by pumping air into, or releasing air from the bleed valves at the tops of the manometers.

(7) Record the readings of each of the manometer across the straight pipe in the dark blue circuit. Note the reference number of that manometer.

(8) Measure the flow rate by timing the collection of water in the bench weighing tank.

(9) Measure the water temperature by holding a mercury thermometer in the flow at exit-from the outlet hose.

(10) Close the gate valve to reduce the differential manometer reading by about 10% . Again read the manometer as in step (7).

(11) Repeat this procedure until you have about 10 sets of readings over the whole range of flow.

$h_f \rightarrow$ friction losses $\rightarrow$ major losses

$h_m \rightarrow$ minor losses

* **Data**

(DBC)

Table 1,a Readings from the dark blue circuit

$\rightarrow h_4 - h_3$

No.	m kg	Time S	$\Delta h = h_f$ ($m\,H_2O$)	h_3 ($m\,H_2O$)	h_4 ($m\,H_2O$)
1					
2					
3					
4					
5					

Table 1,b Calculation of the friction factor f small diameter (13.6 mm)

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

No.	m° kg/s (m/t)	V ($\dot{m}/\rho A$) m/s (~~$\Delta h/t$~~)	R_e $= \frac{\rho V D}{\mu} = \frac{\dot{m} D}{A \mu}$	Flow type	f 0.02993 ~~0.080~~
1	0.224	1.54	11635.6	Turbulent	
2					
3					
4					
5					

$V = \frac{\dot{m}}{\rho A} \rightarrow \rho = \frac{\dot{m}}{VA} = \frac{kg \cdot s}{m^3}$

DBC $\leftarrow$ تغير نقطة ، Value $\leftarrow$ 5 trials

small diameter

$A = \frac{\pi}{4} d^2 = 145.27 \times 10^{-6}\ m$

~~$116508\,36.9 \times 10^{-3}$~~

~~11680.3~~

11635.6

$\mu = 1.8 \times 10^{-3}\ kg/(m \cdot s)$

3.9

$\boxed{\varepsilon = 0.0015\ \ mm}$

$l = 0.9144$ (given)

* ***Data***

LBC

Table 2 Readings from the light blue circuit

Mean temperature = ----------------°C

No	m	t	Manometer readings and differential heads (mm water)															U-tube (mm Hg)		
			Expansion			Contraction			Bend J			Bend H			Bend G			Globe Valve		
	kg	S	7	8	Δh	9	10	Δh	11	12	Δh	13	14	Δh	15	16	Δh	h_1	h_2	Δh
1	7.5	39.6	318	360	-42	345	131	214	517	304	213	512	268	244	552	333	219	195	305	-110
2	7.5	43.6	355	388	-33	375	197	178	510	342	168	515	310	205	555	364	191	175	325	-150
3	7.5	50.7	411	440	-29	411	320	91	516	375	140	514	355	159	550	396	154	155	345	-190
4	7.5	57.1	440	400	40	455	445	10	512	400	112	516	385	131	546	428	118	135	365	-230
5	7.5	67	470	485	-15	485	465	20	510	427	83	513	420	93	539	445	94	115	385	-270

$A = \frac{\pi}{4} d^2 = \frac{\pi}{4} (13.6)^2 \times 10^{-6} = 145.27 \times 10^{-6}\ m$

$D = d = 13.6 \times 10^{-3}\ m$

$\mu = 1.8 \times 10^{-3}\ Kg(m.s)$

$\varepsilon = 0.0015\ mm$

$V = \frac{\dot{m}}{\rho A}$ $\quad Re = \frac{\rho V D}{\mu}$

Table 3 Results for light blue circuit

No.	V (m/s)	Re	f	Head losses for different fittings (mm water)																	
				Expansion			Contraction			Bend J			Bend H			Bend G			Globe Valve		
				Δh	h_f	h_m	Δh	h_f	h_m	Δh	h_f	h_m	Δh	h_f	h_m	Δh	h_f	h_m	Δh	h_f	h_m
1																					
2																					
3																					
4																					
5																					

Bend J: $r = 150\ mm$, $\frac{R}{d} = 11.1$

Bend H: $r = 100\ mm$, $\frac{R}{d} = 7.4$

Bend G: $r = 60\ mm$, $\frac{R}{d} = 3.7$

$$f = \frac{0.25}{\left(\log\left(\frac{\varepsilon}{3.7D} + \frac{5.74}{Re^{0.9}}\right)\right)^2}$$

$$\Delta h = h_m + h_f$$

$$K = \frac{h_m}{V^2/2g}$$

in the smaller pipe

Scanned with CamScanner

Table 4 The minor loss coefficient K

Fitting type	Test No.						Standard Value
	1	2	3	4	5	Average	
Expansion							
Contraction							
Bend J							
Bend H							
Bend G							
Globe Value							

* *Results and discussions*

(1) Complete all of the attached tables

(2) Plot the relationship between the friction factor and Reynolds Number and, the graph should also show the theoretical relationships. (DBC only)

(3) Does the loss coefficients K vary with the flow rate? yes

(4) How do your values of K compare with standard data?

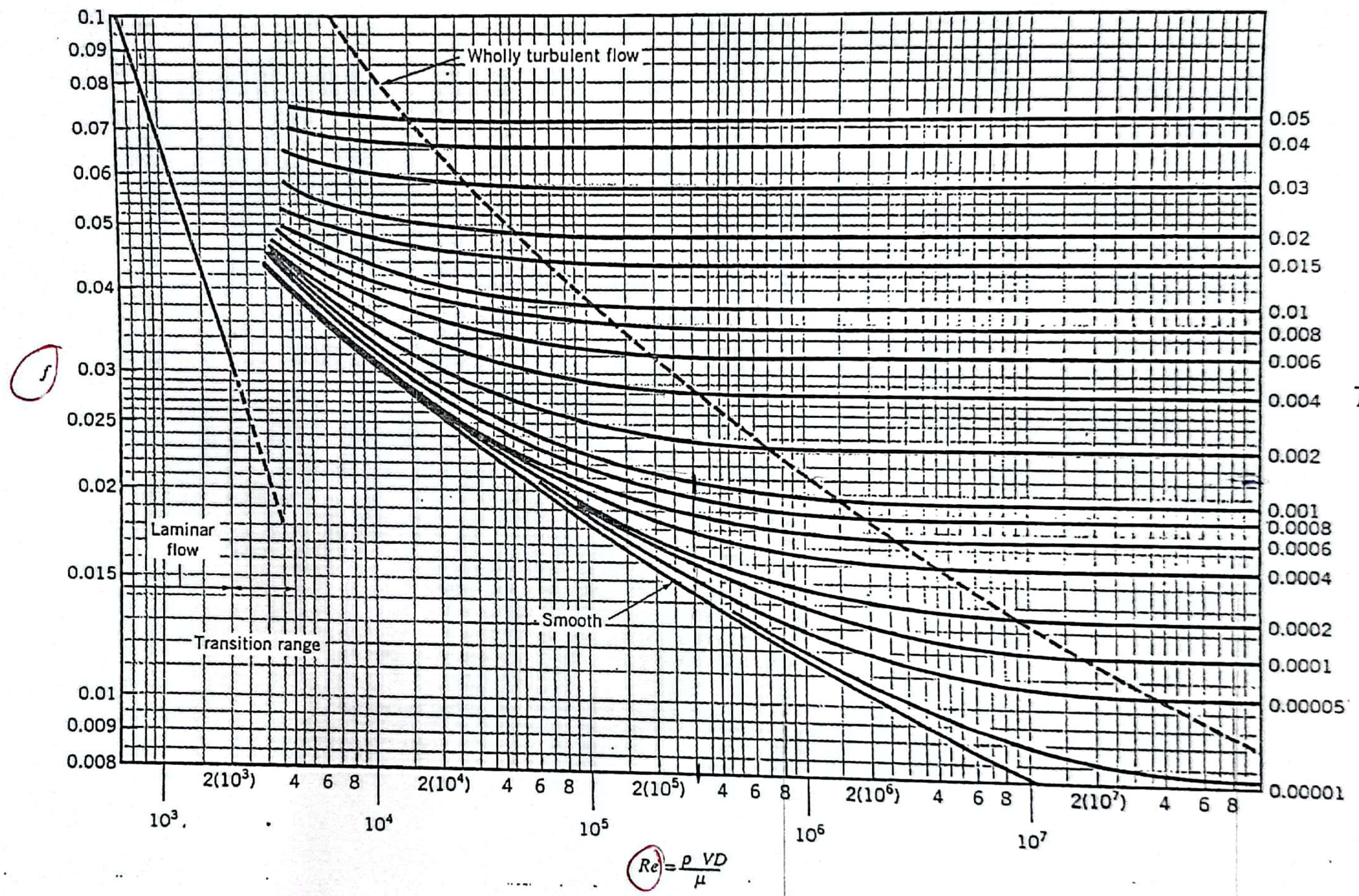

Figure 3 The Moody chart for the friction factor for fully developed flow in circular tubes.

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Experiment 4: *Hydrostatic Pressure Force on a Plane Surface*

Report ☐ *Full* ☑ *Short*

Student Name:-----------------------------------

Student No. :------------------------------------

Hydrostatic Pressure Force on a Plane Surface

$P_c = (P_o + \rho g y_c \sin\alpha)$

* Objectives

1. To determine the position of the center of pressure on the rectangular face of the toroid.
2. To compare the measured value with that predicted from the theoretical analysis.

Experimental value

* Theory

Figure 1 shows a plane surface inclined at an angle α to the free surface of the liquid 00. Since there can be no shear stress in a static fluid medium, the force on the plane is due to pressure only and must act normal to the surface. This pressure force is found to be $\boxed{F = \rho g A y_c \sin\alpha}$

$P_c = \rho g A y_c \sin\alpha$

plane surface ... of Force ... fluid

$F = (\rho g y_c \sin\alpha) A$

where

A	is the area of the surface	(m^2)
ρ	is the density of the liquid	(kg/m^3)
g	is the gravitational acceleration	(m/s^2)
y_c	is the coordinate of the centroid	(m)

$h_c = y_c \sin\alpha \rightarrow$ ... centroid of Area

The force ,F, may be taken as acting at the center of pressure CP. Now to determine the position of the center of pressure we take moments about O After some arrangements and eliminations we can prove that the position of the center of pressure is given by

$I_{xx} = \frac{dh^3}{12}$

Area

$$\boxed{y_{cp} = y_c + \frac{I_{xx,c}}{y_c A}}$$

$A = d \times h$

Where $I_{xx,c}$ is the second moment of area (also called the moment of inertia) about the axis parallel to the x-axis and passing through the centroid C.

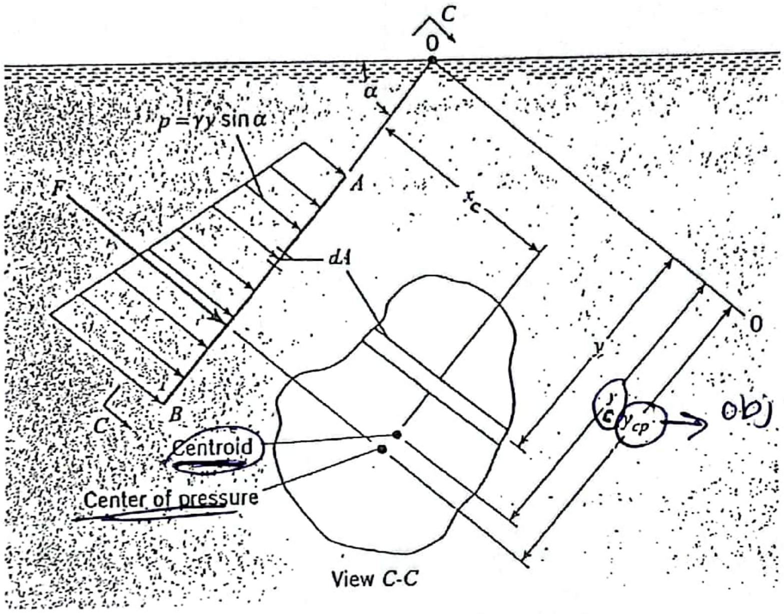
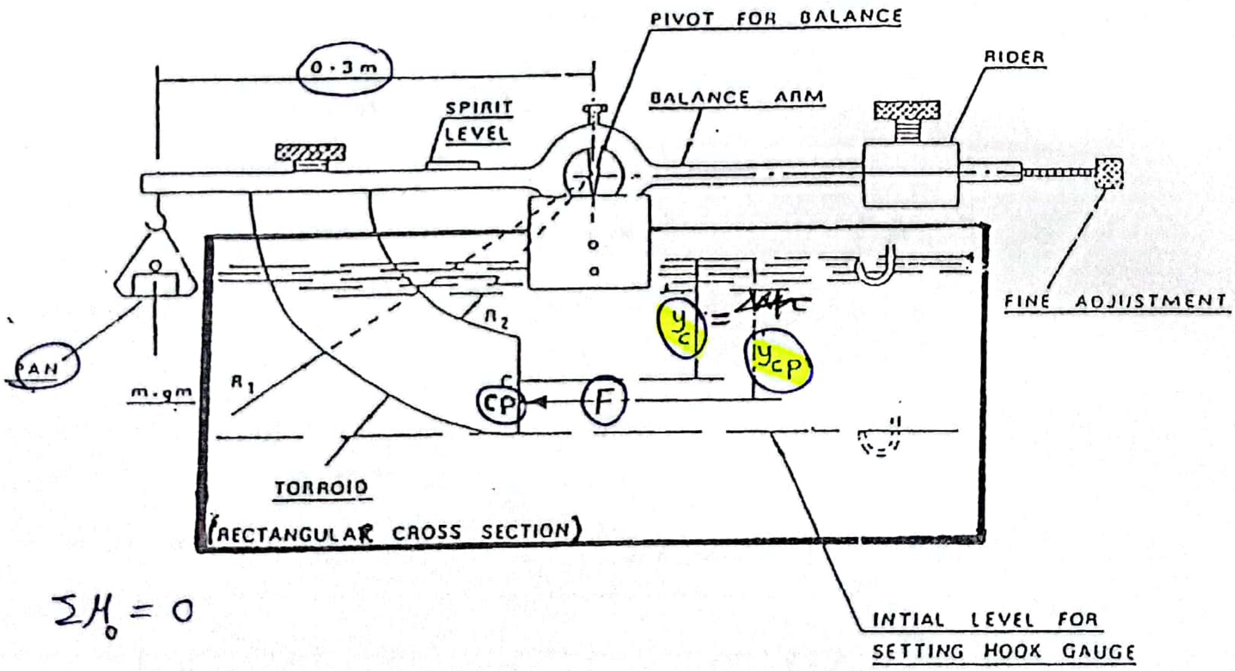


Figure 1 A plane surface immersed in liquid.

water purple



$\Sigma M_0 = 0$

Figure 2 The apparatus.

Bandicut

pressure → 1 single

Force

* Apparatus

The apparatus is shown in Figure 2. A toroid is mounted on a balance and pivoted about the center of curvature of the toroid. Thus only the vertical face along GC of the toroid has any moment about the balancing point. A rider weight balances the weight of the toroid in the dry, so that the moment of the hydrostatic force on C is measured by the weight at the pan. The toroid is immersed in a tank containing water and the depth of immersion is measured by a hook gauge.

* Procedure

1. Measure the dimensions a, b, c, and d , see Figure 3.
2. Level the tank under the toroid and adjust the weight W_1 to level the balance arm. Carefully, admit water to the tank until it just touches the bottom of the quadrant. Take the vernier reading.
3. Raise the water level in increments of about 10 mm and add weight to the balance pan to level the balance arm for each depth of water. Note the mass M and the depth of immersion h for each reading.

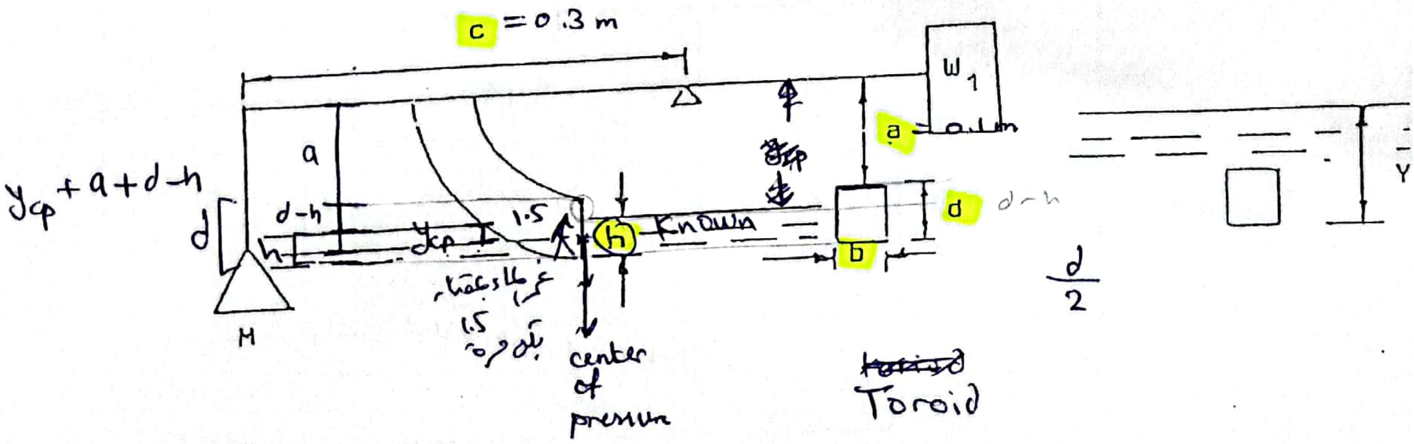
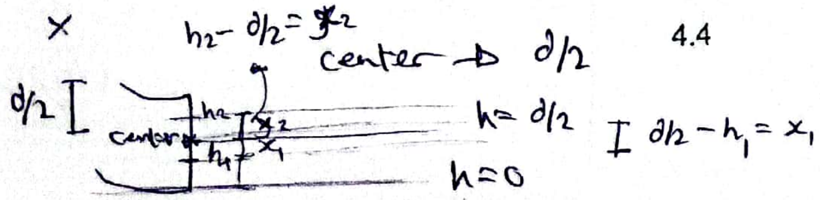


Figure 3 A schematic diagram of the toroid and the balancing weight.

4.4



$I_{xxc} = \frac{bh^3}{12}$

$y_c = h/2$

totally immersed

* ***Data collected***

a = 0.1 m b = 0.075 m c = 0.3 m d = 0.1 m

Partial Immersion

$A = b \times h$ $y_{cp} = y_c + \frac{I_{xxc}}{y_{cp}}$

h (cm) $\times 10^{-2}$	M (g)	Immersed Area (m^2)	Theoretical y_{cp}(cm)	Experimental y_{cp}(cm)	M/h^2
1.5					
3					
4.5					
6					
7.5					
9					

$h \propto M$

Total Immersion

h (cm)	M (g)	Theoretical y_{cp}(cm)	Experimental y_{cp}(cm)	y_c $\left(h-\frac{d}{2}\right)$
10				
11				
12				
13				
14				

* **_Results and discussions_**

Partial immersion (h < d)

(1) Derive an expression for the theoretical depth of the center of pressure (y_{cp}) below the surface and, for each level, compare it with the experimental depth of center of pressure. (You may assume the force **F** to be that calculated from the formula).

(2) Derive the equation

$$\frac{M}{h^2} = -\frac{\rho b(a+b)}{2c} + \frac{\rho bh}{6c}$$

$y = ax + b$

$M/h^2 = \left(\frac{\rho b}{6c}\right)h - \frac{\rho b\,(a+b)}{2c}$

(3) Plot (M/h^2) versus h and obtain the slope and the intercept, and compare their values with those given in the above equation.

Total Immersion (h > d)

(1) As in (1) above.

(2) Derive the equation

$$M = \frac{\rho bd^3}{12c} + (h - d/2)\left[\frac{\rho bd}{c}(a + 1/2(d))\right]$$

(3) Plot (M) versus (h – d/2) and obtain the intercept and the slope, and compare their values with those given in the above equation.

* **_Conclusions_**

1. How do the experimental values of the position of center of pressure correspond with the calculated theoretical values from physical dimensions? No
2. How do the measured values of slope and intercept obtained in 3 above correspond with theoretical values?
3. The pressure force act on the four surfaces of the rectangular toroid which are submerged were ignored in the experiment. Are they zero, negligible or is there a mistake in the experiment method? they cancle each other
4. You ignored the buoyancy effect of the submerged part of the toroid comment on the implications.
5. Would the location of the center of pressure change if a different fluid were used in the tank? Explain.

ofcourse

ρ will differ $\rightarrow$ pressure will differ

University of Jordan
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Dr. Adnan Jaradat

Experiment 5: *Impact Of A Water Jet*

Report ☐ *Full* ☐ *Short*

Student Name:-----------------------------------

Student No. :-----------------------------------

Impact of A Water Jet

* *Objectives*

1. To determine the force produced by a water jet when it strikes a flat vane and a hemispherical cup.
2. To compare the results measured with the theoretical values calculated from the momentum flux in the jet.

* *Introduction*

Over the years, engineers have found many ways to utilize the force that can be imparted by a jet of fluid on a surface diverting the flow. For example, the pelton wheel has been used to make flour. Furthermore, the impulse turbine is still used in the first and sometimes the second stage of a steam turbine. Firemen make use of the kinetic energy stored in a jet to deliver water above the level of the nozzle to extinguish fires in high-rise buildings. Fluid jets are also used in industry for cutting metals and debarring. Many other applications of fluid jets can be cited which reveals their technological importance.

This experiment aims at assessing the different forces exerted by the same water jet on a variety of geometrically different plates. The results obtained experimentally are to be compared with the ones inferred from theory through utilizing the applicable versions of the Bernoulli and momentum equations.

* *Theory*

For the general case shown in Figure 1, the momentum flux in the jet is $\dot{m} u_0$ where $\dot{m}$ is the mass flow rate and u_0 is the jet velocity just upstream of the vane. After being deflected through an angle β the momentum flux is $\dot{m} u_1 \cos\beta$ in the x-direction.

The force on the fluid is therefore $\dot{m} u_1 \times \cos\beta - \dot{m} u_0$ in the x-direction. Thus the force **F** in the x-direction on the vane is therefore.

$$\boxed{F = \dot{m}(u_0 - u_1 \cos\beta)} \quad \text{(1)}$$

in ~~general~~ general

theo

Flat
$\beta = 90$
$F = \dot{m} u_0$

hemispherical
$\beta = 180$
$F = 2\dot{m} u_0$

$$F = \dot{m} u_1 \cos\beta - u_0 \dot{m}$$

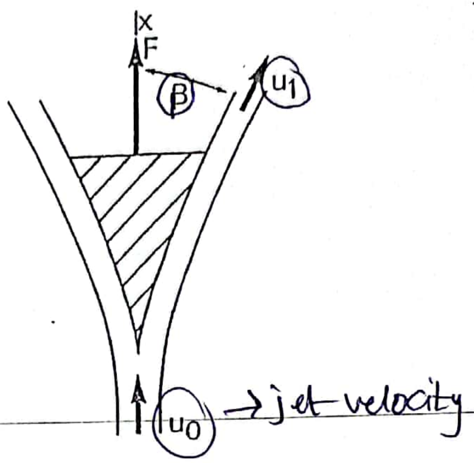
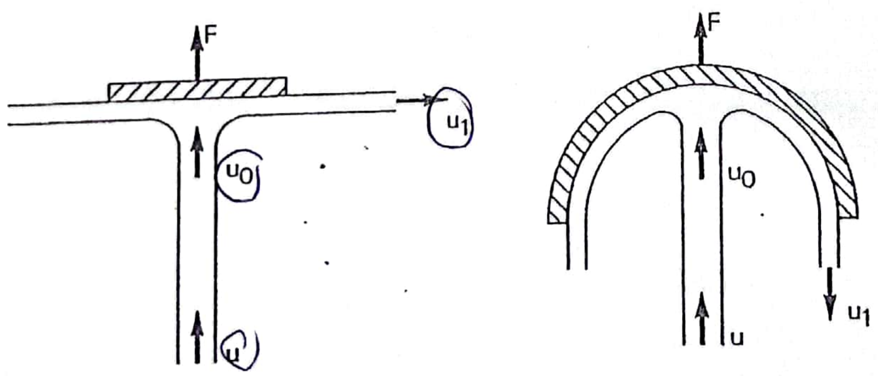


Figure 1 Flow of a jet over a curved vane.



(a) Flat Plate (b) Hemispherical Cup

Figure 2 Flow of a jet over a flat plate and hemispherical cup.

theoritical F

Now in the case of a flat plate, Figure 2a , $\boxed{\beta = 90^\circ}$ So $\cos\beta = 0$ and equation (1) reduces to (2)

$$\boxed{F = \dot{m} u_0}$$

For a hemispherical cup, Figure 2b, $\boxed{\beta = 180^\circ}$ so $\cos\beta = -1$ and equation (1) reduces to (3)

$$F = \dot{m}(u_0 - u_1)$$

Furthermore, if there is a negligible reduction of speed so that $u_1 = -u_0$. Then (4)

$$\boxed{F = 2\dot{m} u_0}$$

In the experiment it is not possible to measure directly the velocity just upstream of the vane. However, the velocity u at the exit of the nozzle can be experimentally determined. The velocity u_0 is somewhat less than this due to declamation caused by gravity and can be calculated from the Bernoulli equation, that is

~~exp.~~ $\frac{\dot{m}}{\rho A}$

$$\frac{u^2}{2g} + z + \frac{P}{\rho g} = \frac{u_0^2}{2g} + z_0 + \frac{P_0}{\rho g} \qquad (5)$$

Now from figure 2a, $\boxed{Z = 0}$, $P_0 = P$, $\boxed{Z_0 = s}$ and this yields (6)

$$\boxed{u^2 = u_0^2 + 2gs}$$

which can be written as (7)

$$\boxed{u_0^2 = u^2 - 2gs}$$

Where s is the distance between the nozzle exit and the surface of the vane.

u can be determined from the continuity equation $\dot{m} = \rho u A$

Where A is the cross-sectional area of the nozzle πd^2

Hence $u = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho \pi d^2}$

Where d is the diameter of the nozzle

In order to calculate the force on the vane due to the jet we take moment about the pivot of the weighing beam, Figure 3, and substitute known values into the equation we get.

$$F \times 0.1525 = 0.610 \times g \times \Delta x \qquad (8)$$

or $\boxed{F = 4g\,\Delta x}$

which is considered the $\boxed{\text{experimental}}$ value of F

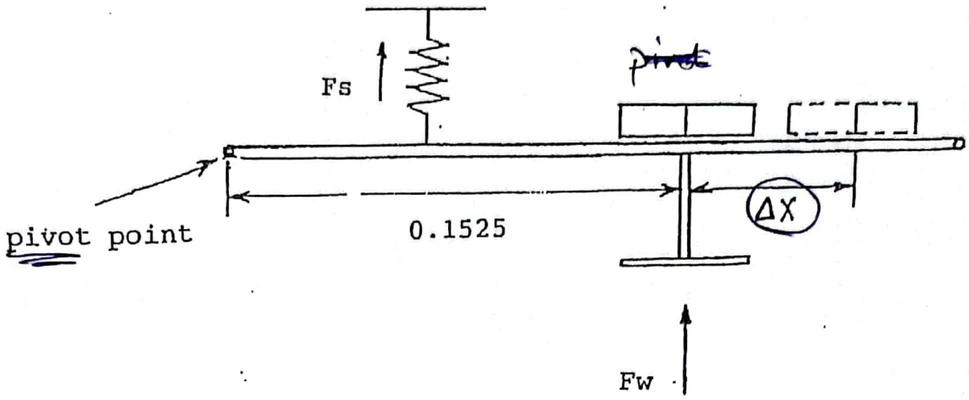


Figure 3 The balancing arm attached to the vane.

* *Apparatus*

Hydraulic Bench, Water Jet Apparatus, Stop Watch

The water is supplied to the jet apparatus in a closed loop by a pump. The flow rate is determined with the use of a weighing tank and a stopwatch. The water issues vertically upwards into the air, through a nozzle. Two objects are available: a flat plate and a hemispherical cup. Each object can be mounted on a horizontal lever above the water jet and receive its impact. The force on the object can be determined with the use of weights that can be hung at different positions on the lever, see Figure 4.

Technical Data

Mass of jockey weight,	m	= 0.610 kg
Distance from center-line of vane to weigh-beam pivot		= 0.1525 m
Diameter of nozzle,	d	= 0.01 m
Height of vane above nozzle outlet,	s	= 0.04 m
Diameter of hemispherical cup		= 0.06 m

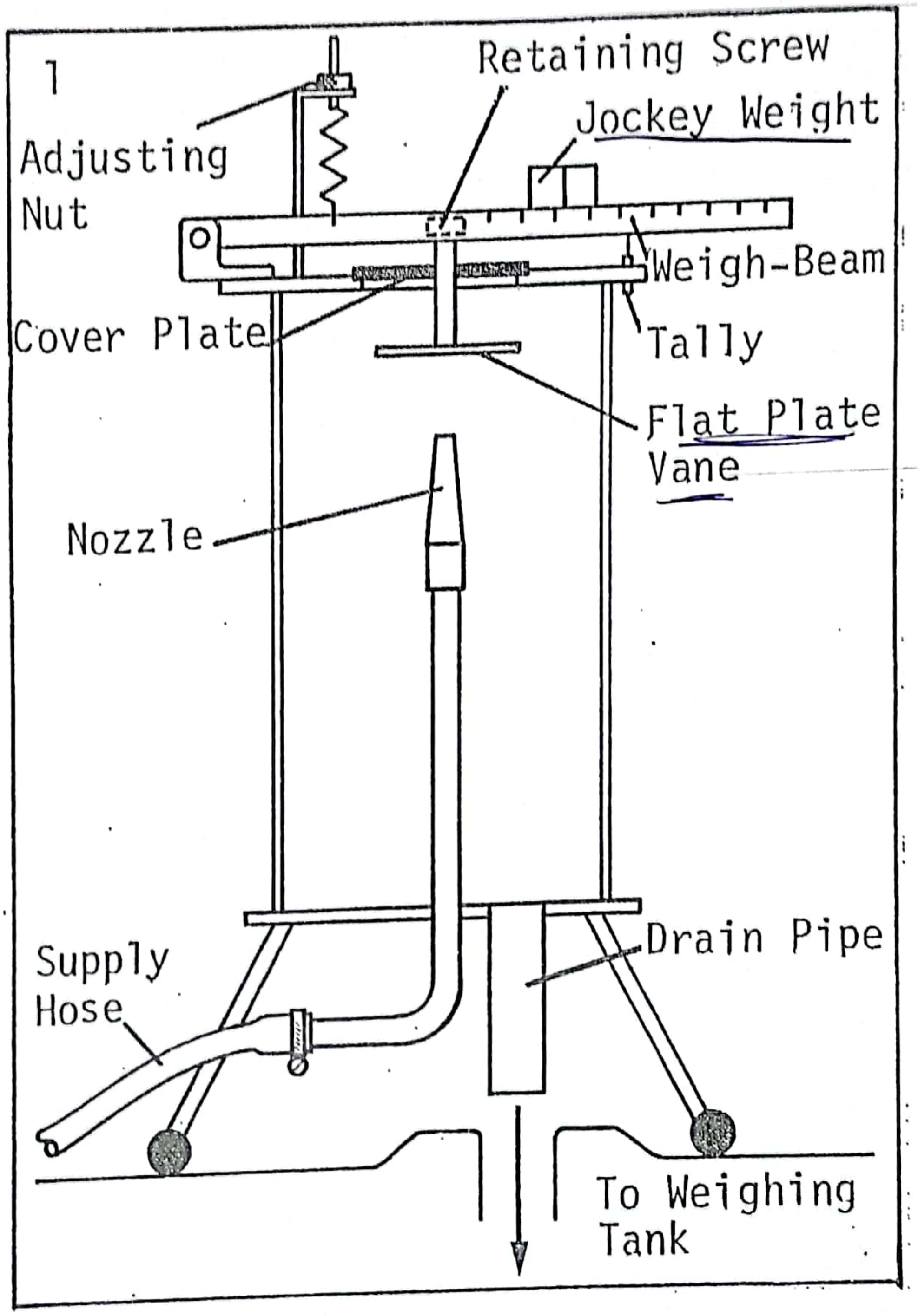


Figure 4 The Apparatus.

* *Procedure*

(1) Stand the apparatus on the hydraulic bench, with the drain pipe immediately above the hole leading to the weighing tank, see Figure 4. Connect the bench supply hose to the inlet pipe on the apparatus, using a hose-clip to secure the connection.

(2) Fit the flat plate to the apparatus. If the cup is fitted, remove it by undoing the retaining screw and lifting it out, complete with the loose cover plate. Take care not to drop the cup in the plastic cylinder.

(3) Fit the cover plate over the stem of the flat plate fitting and hold it in position below the beam. Screw in the retaining screw and tighten it.

(4) Set the weigh-beam to its datum position. First, set the jockey weight on the beam so that the datum groove is at zero on the scale, Figure 5. Turn the adjusting nut, above the spring, until the grooves on the tally are in line with the top plate as shown in Figure 6. This indicates the datum position to which the beam must be returned, during the experiment, to measure the force produced by the jet.

(5) Switch on the bench pump and open the bench supply valve to admit water to the apparatus. Check that the drain pipe is over the hole leading to the weighing tank.

(6) Fully open the supply valve and slide the jockey weight along the beam until the tally returns to the record the reading on the scale corresponding to the groove on the jockey weight.

(7) Measure the flow rate by timing the collection of, say, 30 kg of water in the bench weighing tank.

(8) Move the jockey weight inwards by 10 to 15 mm and reduce the flow rate until the beam is approximately level. Set the beam to exactly the correct position (as indicated by the tally) by moving the jockey weight, and record the scale reading. Measure the flow rate.

(9) Repeat step 8 until you have about 6 sets of readings over the range flow. For the last set, the jockey should be set at about 10 mm from the zero position. At the lower flow rates you can reduce the mass of water collected in the weighing tank to 15 kg.

(10) Switch off the bench pump and fit the hemispherical cup to the apparatus using the method in steps 2 and 3. Repeat step 4 to check the datum setting.

(11) Repeat steps 5 to 9, but this time move the jockey in steps of about 25 mm and take the last set of readings at about 20 mm.

(12) Switch off the bench pump and record the mass m of the jockey weight, the diameter d of the nozzle, and the distance s of the vanes from the outlet of the nozzle.

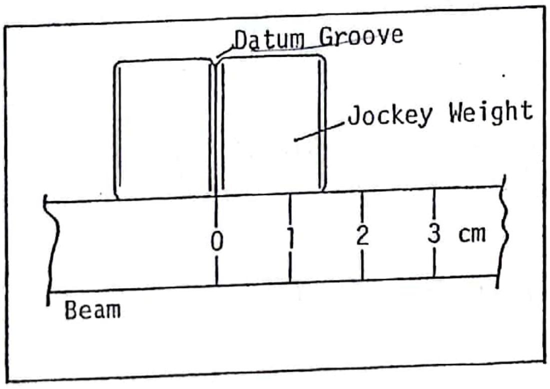


Figure 5 Jockey in datum position

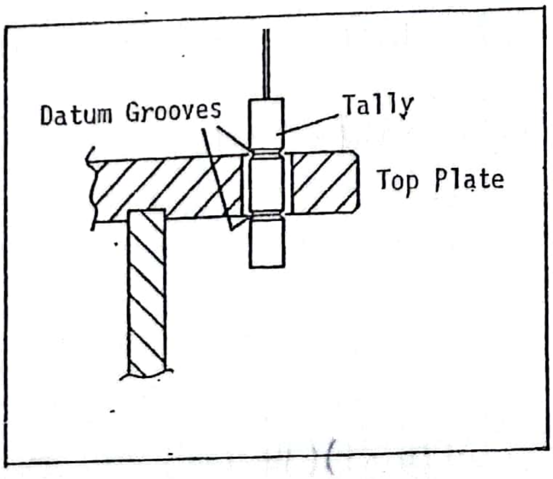


Figure 6 Tally in datum position

$u = \frac{\dot{m}}{\rho A}$; $u_0^2 = u^2 - 2gs$

$s = 0.04$ m

$d_{nozzle} = 0.01$ m

$d_{semi} = 0.06$ m

* ***Data collected*** and

Results for flat plate: $\frac{m}{t}$

Mass of water m(kg)	t (s)	Δx (mm)	$m^{\bullet}$ (kg/s)	u' (m/s)	u_0 (m/s)	$m^{\bullet}u_0$ (N)	F Theo. N	F Exp. N	Error
7.5									
7.5									
7.5									
7.5									
7.5									
7.5									
7.5									

Results for hemispherical cup:

Mass of water m(kg)	t (s)	Δx (mm)	$m^{\bullet}$ (kg/s)	u (m/s)	u_0 (m/s)	$2m^{\bullet}u_0$ (N)	F Theo. N	F Exp. N	Error
7.5									
7.5									
7.5									
7.5									
7.5									
7.5									
7.5									

* **Results and discussions**

(1) Plot F against $\dot{m} u_0$ for the two vanes, and, for the two values of F, the theoretical and the experimental , comment on their linearity.

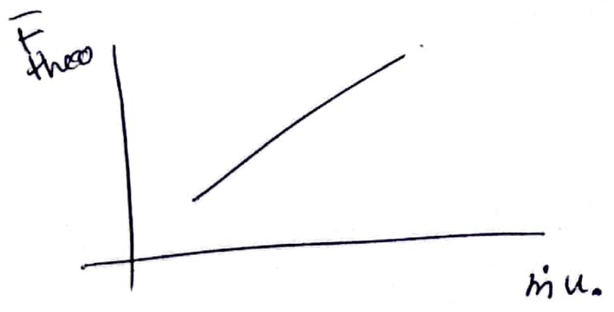
(2) If the lines on your graph do not pass through the origin, what reason might there be for this? $\rightarrow \dot{m} \neq 0$

(3) Why is the force on the hemispherical cap somewhat less than twice that on the flat plate? β is twice β in Flat

* **Conclusions**

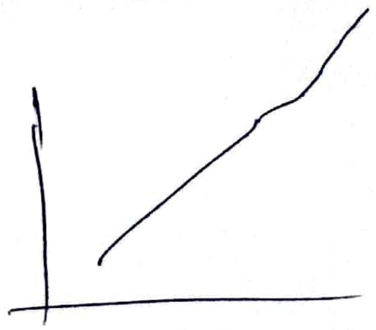
(1) How well do your results compare with the theory?

(2) What accuracy have you achieved in measuring the force on each of the vanes (calculated from the displacement)? flow rate in hemispherical is the least → require longer time to raise the tank



Flat

Semispherical



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Dr. Adnan Jaradat

Experiment 6: *Thermal Conductivity*

Report ☐ *Full* ☑ *Short*

Student Name:---

Student No. :---

Thermal Conductivity

* Objectives

To determine;

1. The thermal conductivity of a good conductor.
2. Rate of heat transfer.

* Theory

When a temperature gradient exists in a solid body, an energy transfer from high-temperature region to low-temperature region takes place. In this case it is said that the energy is transferred by conduction and that the $\boxed{\text{heat transfer rate}}$ per unit area is proportional to the $\boxed{\text{temperature gradient:}}$

$$\boxed{Q^{\bullet} = -kA\frac{dT}{dx}}$$

exp. $k = -\dfrac{\dot{Q}}{A\,\frac{dT}{dx}}$

where

$Q^{\bullet}$ = The heat transfer rate (W)

$(25\cancel{8}\times10^{-3})^2$

A = The cross-sectional area (m^2) $= \frac{\pi}{4}D^2 = \frac{\pi}{4} \times \cancel{205}$

dT/dx = The temperature gradient

? k = The thermal conductivity of the material, (W/mK)

To determine the thermal conductivity "k" of each specimen, both $Q^{\bullet}$ and $\frac{dT}{dx}$ are to be evaluated. In this experiment $Q^{\bullet}$ is the rate of heat transferred to the cooling water and is given by

$\dot{V} = 67\times10^{-6}\ m^3$

$$\boxed{Q^{\bullet} = m^{\bullet}C\,(T_{out} - T_{in})}$$

where

$m^{\bullet}$ = The mass flow rate of water (kg/s) $= \dot{V}\rho = \dfrac{67\times10^{-6}\times1000}{25(5)}$

C = Specific heat for water

$= \boxed{4.18\ \text{(kJ/kg °C)}}$

T_{in} = Water inlet temperature (°C) = 13 °C

T_{out} = Water exit temperature (°C) = 25 °C

The temperature gradient within the specimen can be approximated to be linear between the two thermocouple positions and is given by

$$\frac{dT}{dx} = \frac{T_3 - T_4}{L}$$

hotter

T_4

T_3

T_2

T_1

colder

where

T_4 = The thermocouple temperature (hot end) (°C)

T_3 = The thermocouple temperature (cold end) (°C)

L = The distance between the two thermocouples (m)

The text book values for thermal conductivity for the specimens provided are:

Aluminum	210 W/mK	theoritically
Copper	385 W/mK	
Mild Steel	42 W/mK	
Stainless steel	30 W/mK	

However, the actual values will vary slightly from the textbook values as any variation in material composition will affect this.

* Apparatus

The apparatus, Figure 1, consists of a self clamping specimen stack assembly with electrically heated source, calorimeter base, Dewar vessel enclosure to ensure negligible loss of heat and constant head cooling water supply tank. A multipoint thermocouple switch is mounted on the steel cabinet base and two mercury and glass thermometers are provided for water inlet and outlet temperature readings. Four NiCr/NiAl thermocouples are fitted and connections are provided for a suitable potentiometer instrument to give accurate metal temperature readings.

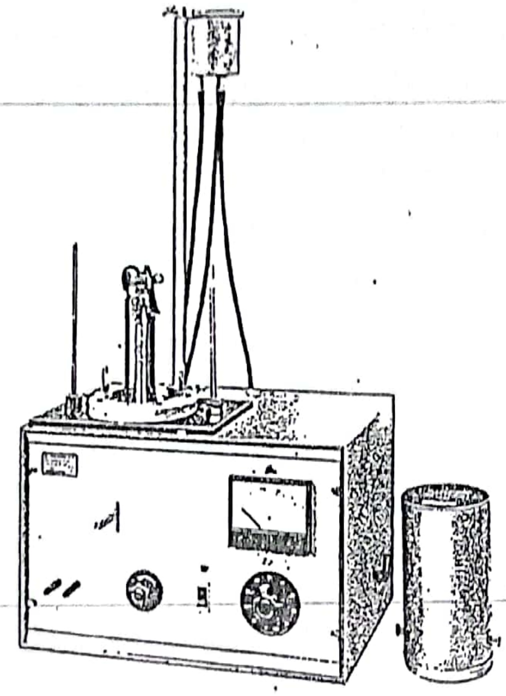
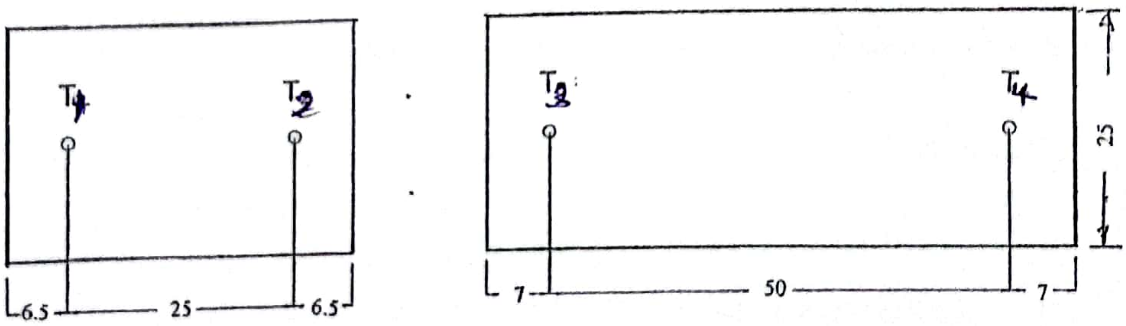


Figure 1 Thermal conductivity apparatus.

Four metal specimens are provided. Two holes are provided in each specimen for insertion of the thermocouples. A sketch of the specimen is shown in Figure 2.



short specimen

long specimen

Stainless Steel or Mild Steel

Copper or Aluminum

All dimensions in mm

Figure 2 A sketch of the specimen.

low cond. material

high cond. material

* *Procedure*

(1) The apparatus is assembled with one short specimen (mild steel or stainless steel, i.e. low conductivity material) in lower position, and one long specimen (copper or aluminum, i.e. high conductivity material) in upper position.

After selecting specimens to be used in the experiment, ensure that they are completely free from dirt especially at the ends where contact is to be made. Apply a light smear of Silicone Grease at the ends of the specimens before assembly to ensure good thermal contact.

(2) Operate the clamp by moving the protruding lever positioned on the front of the apparatus to a downward position and place specimens between heating element and clamp. Ensure that the holes for the thermocouples are accessible. Release the lever, thereby clamping specimens in position. Insert thermocouples into holes provided.

(3) Ensure that the thermostat adjustment control which is situated on the front of heating element is turned fully clockwise. This sets the cutout temperature to approximately 210 °C. The normal maximum working temperature is 200°C.

(4) Place the Dewar vessel in position over specimens.

(5) Fit the thermometers into the special leak proof connections provided on top of calorimeter base.

a) Connect water pipes from water supply to header tank, header tank to inlet on apparatus, and header tank overflow to drain.

b) Turn on water supply. Adjust flow rate through the apparatus by means of the inlet flow valve positioned at inlet pipe. Note that the actual flow rate is not critical, however , a temperature difference of about 8°C should be sought.

(6) Connect the potentiometer instrument to the two terminals provided on the front of the apparatus.

(7) Connect the control box to the socket on the right hand side of the conductivity apparatus, and connect the control box to a single phase AC mains . Check that the supply voltage is correct.

(8) Switch on the electrical supply and check that indicating lights on both control box and calorimeter base are operative.

(9) Before readings can be obtained from the apparatus, the heat flow must reach a steady state condition. This can be done in either of two ways as follows

a) Set current input to a maximum, this being about 0.55 amps. Maintain this until a temperature of 200 °C is obtained from the thermocouple nearest the element (T_4). This will take 15-20 minutes. Reduce the current to 0.3 amps until the temperatures have become steady. This will take 20-25 minutes.

b) Set current to 0.3 amps. Leave for a period of approximately 2 hours.

<u>Note</u>: In both of the above methods the water must be flowing continuously.

* *Data collected*

Test Material:

Current:

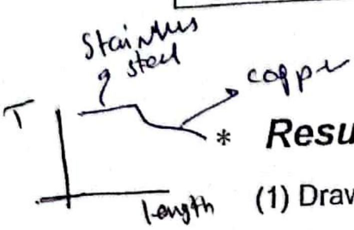
Water flow rate:

Time to reach steady state:

Total time for test:

Temperature readings obtained

Test No.	Time	T_1(°C)	T_2(°C)	T_3(°C)	T_4(°C)
1					
2					
3					
4					
5					
Total					
Average					



* *Results and discussions*

(1) Draw the temperature vs the length for the specimens used in the experiment.

(2) Determine the thermal conductivities for the specimens used in the experiment.

Experimentally

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Thermal and Fluid Sciences Lab.

Dr. Adnan Jaradat

Experiment 7: *Heat Pump and Air Cooler*

Report ☐ *Full* ☐ *Short*

Student Name:--

Student No. :--

Heat Pump and Air Cooler

يعني مسألة
Heat pump

* *Objectives*

The objectives of this experiment are

1. To determine the coefficient of performance (COP) of a heat pump.
2. To determine the coefficient of performance (COP) of an air cooler

* *Introduction*

An air cooler and a heat pump essentially comprise the same cycle component, see Figure 1. It is their objectives that differentiate between them. In an air cooler, or a refrigerator, the heat extracted from the air, $Q^{\bullet}_{L}$ (absorbed by the refrigerant in the evaporator) is the required cooling effect. However, the heat rejected to the circulating water in the condenser, $Q^{\bullet}_{H}$, although necessary, is not the objective. On the other hand, the heat pump utilizes the heat absorbed from a low-temperature source to maintain a heated space at a higher temperature, thus $Q^{\bullet}_{H}$ is the required heating effect. The theoretical model with which such a device is compared in order to evaluate its performance is the reversible simple refrigeration cycle shown in Figure (1). Such a cycle takes in heat isothermally from a reservoir at temperature T_L and rejects heat isothermally to another reservoir at temperature T_H. The intervening processes are adiabatic reversible expansion and compression processes. The coefficient of performance of the machine when operating as a heat pump, is defined as:

condenser من عنده

$$COP_{pump} = \beta' = \frac{Q^{\bullet}_{H}}{W^{\bullet}} = \frac{Q^{\bullet}_{H}}{Q^{\bullet}_{H} - Q^{\bullet}_{L}} \quad \frac{\text{سعر الحرارة المسؤول}}{\text{power of the compression}} = \frac{T_H}{T_H - T_L}$$

And when operating as an air cooler or refrigerator the COP is

$$COP_{cooler} = \beta = \frac{Q^{\bullet}_{L}}{W^{\bullet}} = \frac{Q^{\bullet}_{L}}{Q^{\bullet}_{H} - Q^{\bullet}_{L}} = \frac{\text{evaporator, كمية الحرارة من}}{\text{work of compressor}} = \frac{T_L}{T_H - T_L}$$

These equations apply also to a real machine which is operating between the same temperature limits. However, the numerical values will be less than those corresponding to the ideal reversible engine. For an ideal reversible engine the heat transfer ratio $Q^{\bullet}_L/Q^{\bullet}_H$ can be replaced by the ratio of the absolute temperature of the two reservoirs as follows

$$\frac{Q^{\bullet}_H}{Q^{\bullet}_L} = \frac{T_H}{T_L}$$

max. COP that can be obtained from the cooler/~~heat pump~~

$$\frac{T_L}{T_H - T_L}$$

Thus for a reversible refrigeration cycle, the COP is

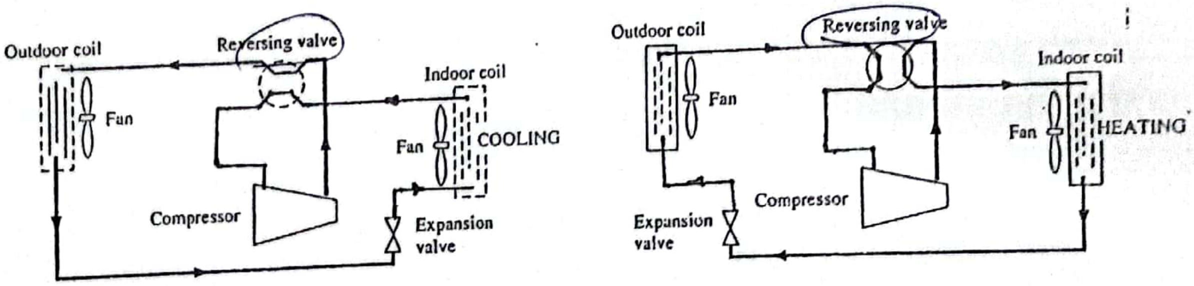
$$\beta = \frac{T_L}{T_H - T_L}$$

And for a reversible heat pump the COP is

$$\beta^1 = \frac{T_H}{T_H - T_L}$$

→ max. cop that can be obtained from the heat pump

Note that in all cases, $\boxed{\beta^1 = \beta + 1}$



a. An air-cooler cycle

b. A heat pump cycle

Figure 1 A simple refrigeration cycle.

* *Apparatus*

The apparatus consists of two separate units, the air conditioner and the control console. The two units are connected, by electrical cables, thermocouple wires and nylon water pipes. They are completely detachable formability and flexibility of installation, see Figure 2.

The air conditioner is completely self contained and consists of a hermetically sealed refrigeration system driven by a 1.1 kW motor, a refrigerant-to-water heat exchanger, a refrigerant-to-air heat exchanger, reversing valve, fan and motor, condensate collector and electrical controls.

The air to be conditioned enters by way of the finned, refrigerant-to-air heat exchanger, passes through the centrifugal fan which is driven by a motor immersed in the air flow. The air is then discharged to a duct of circular cross-section carrying a pitot tube and a thermocouple. When the air is being cooled, and the relative humidity is high enough, moisture is deposited on the heat exchanger and drained off to a measuring vessel.

The control console carries all the electrical switches and fuses, a wattmeter and a multi point digital temperature indicator. It also carries a flowmeter for measurement of the quantity of water passing through the conditioner, an inclined manometer for use with the pitot-static tube for air flow measurement and a vessel containing a 2 kW immersion heater. The latter is sometimes necessary when the conditioner is operating as a heat pump extracting heat from the circulating water since, if the temperature of the water on entry to the conditioner falls below about 10°C, there is a likelihood of freezing taking place.

The following instruments are provided:

a) Wattmeter for measurement of electrical power input to refrigerator compressor and to the fan.
b) Multipoint digital temperature indicator.
c) Whirling psychrometer for measurement of relative humidity of air entering and leaving the conditioner.
d) Pitot-static tube and inclined manometer for measurement of air flow.
e) Cooling water flowmeter.
f) Graduated collecting vessel for condensate.
g) Thermocouples for temperature measurement

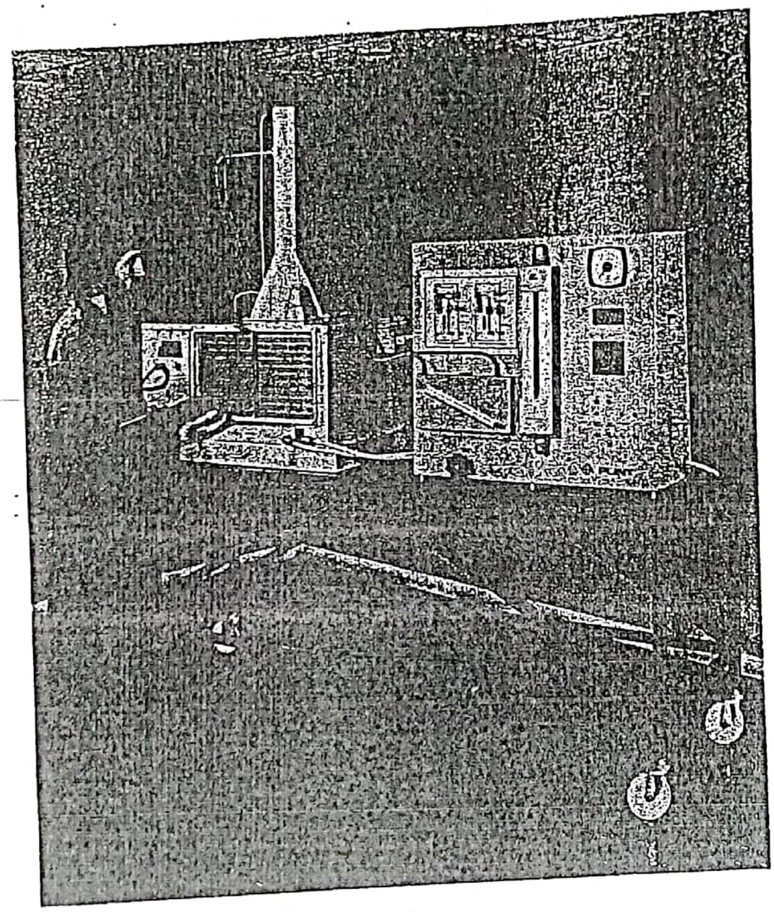


Figure 2 The Apparatus.

Technical Data

Versatemp high level air conditioner type VH250

Maximum power consumption, air conditioner	1.5 kW
Power consumption, immersion heater	2 kW
Maximum cooling water flow rate	5 l/min
Nominal air flow rate	9 m^3/min.
معدل التبريد والتسخين Nominal heating/cooling capacity	1.8 to 3.5 kW

In the calculations that follow the International System of Units (S.I.) has been used.

* Procedure

(1) Before starting the machine ensure that the cooling water is turned on and regulated to give a flow of about 4 1/min. If the temperature of the cooling water entering the apparatus is less than 10°C and the machine is to operate as a heat pump, the water must also be switched on to ensure that freezing does not take place in the refrigerant-to-water heat exchanger.

(2) Select cooling or heating as desired and switch on the compressor and fan. It takes between thirty minutes and one hour for temperature conditions in the apparatus to stabilise. It is suggested that a set of readings should be taken every ten minutes and continued until two successive readings show a change in air and water temperature of not more than 0.3°C.

(3) The wattmeter shows the total electrical input to both the fan and the refrigerator compressor. The input to the fan alone may be measured by momentarily switching off the fan to measure the power to the compressor motor, then subtracting this reading from the reading for both the fan and the compressor.

(4) The relative humidity of the air entering the conditioner should be measured by means of the sling hygrometer provided. Ensure that the reservoir in the hygrometer is filled with water and that it is whirled for a sufficiently long period (about thirty seconds) to give steady readings.
The relative humidity can be then used to find the humidity ratio ω, with the help of the phsychrometric chart or the relative humidity table, given at the end of the experiment.

When operating as a heater there will be no moisture deposited in the conditioner, since the relative humidity falls as the air passes through the machine. When operating as a cooler there may or may not be deposition of moisture depending on the relative humidity of the air entering the machine. It may take a considerable time, as much as two to three hours, for the rate of flow of condensate to reach a stable value and, for this reason, it may be desirable when a cooling test is to be made to start up the apparatus several hours in advance of the laboratory period. There is some intrinsic variation in condensate flow rate and the measuring period should be as long as possible.

(5) Record the following temperatures

T_1 Air at inlet

T_2 Air at discharge

T_3 Circulating water at inlet

T_4 Circulating water at discharge

	Heat Pump	Cooler
T_5 Compressor:	Discharge	Inlet
T_6 Compressor:	Inlet	Discharge
T_7 Refrigerant-to-water heat exchanger:	Discharge	Inlet
T_8 Refrigerant-to-water heat exchanger:	Inlet	Discharge
T_9 Refrigerant-to-air heat exchanger:	Inlet	Discharge
T_{10} Refrigerant-to-air heat exchanger:	Discharge	Inlet

(6) The air flow is measured by means of a Pitot tube mounted in the center of the discharge duct. The pressure of air at this point is effectively equal to that of the atmosphere, P_0. If H_1mm H_2O is the velocity head measured by the Pitot tube, the mass rate of flow is given by:

سواء ال heating or cooling

$$\dot{m}_1 = 0.00105\sqrt{\frac{H_1 P_0}{T_2}} \quad \text{Kg/s}$$

P_0 is in Pa and T_2 is in K

The Air Cooler

* *Theory*

The machine arrangement when operating as an air cooler is shown schematically in Figure 3. The steady state steady flow (SSSF) equation for the system may be derived from the energy flow diagram Figure 4, as

$$\boxed{\dot{Q}_H - \dot{W}_c - \dot{W}_F = \dot{Q}_L}$$

Where

$\dot{Q}_H$ and $\dot{Q}_L$ can be found from the energy balance for both the condenser (using circulating water) and the evaporator which is cooling the air, thus

$$\dot{Q}_H = \dot{m}_w(h_{f4} - h_{f3})$$

And

air

$$\dot{Q}_L = \dot{m}_a(h_2 - h_1) + \dot{m}_a(w_2 h_{v2} - w_1 h_{v1}) - \cancel{\dot{m}_c h_c}$$ zero → no condensate

air

Where

$\dot{m}_a$ = Mass flow rate of dry air (kg/s)

$\dot{m}_w$ = Mass flow rate of cooling water in the condenser (kg/s)

$\dot{m}_c$ = Mass flow rate of condensate water from the air stream (kg/s)

h_1 and h_2 = Enthalpy of dry air at inlet and exit conditions (kJ/kg)

w_1 and w_2 = Humidity ratios of air at air inlet and exit condition (kg/kg of dry air)

h_{v1} and h_{v2} = Enthalpy of water vapor carried by the air at inlet and exit h_{g2} conditions (kJ/kg). They are taken as

$h_{v1} = h_{g1}$ and $h_{v2} = h_{g2}$

h_c = Enthalpy of condensate = h_{f2} (kJ/kg)

$\dot{W}_c$ = Electrical input to the compressor (kW)

$\dot{W}_F$ = Electrical input to the compressor (kW)

The coefficient of performance is then defined as:

$$\beta = \frac{\dot{Q}_L}{\dot{Q}_H - \dot{Q}_L} = \frac{\dot{Q}_L}{W}$$

This may be compared with an ideal performance based upon the temperature difference across the refrigerator circuit

$$(COP_R)_{max} = \frac{T_{10}}{T_8 - T_{10}}$$ ideal

In a real machine such as the present one the coefficient of performance falls short of the ideal for a number of reasons, of which the most important are:

a) Electrical and mechanical losses in the motors of both the fan and the compressor.

b) The imperfection (irreversibility) of the refrigeration cycle itself.

c) The necessity for temperature differences between refrigerant and air, and between refrigerant and water. As a result of which the refrigerant cycle operates between substantially wider temperature limits than those applicable to the water and air forming the source and sink.

d) Electrical and mechanical losses the in circulating fan.

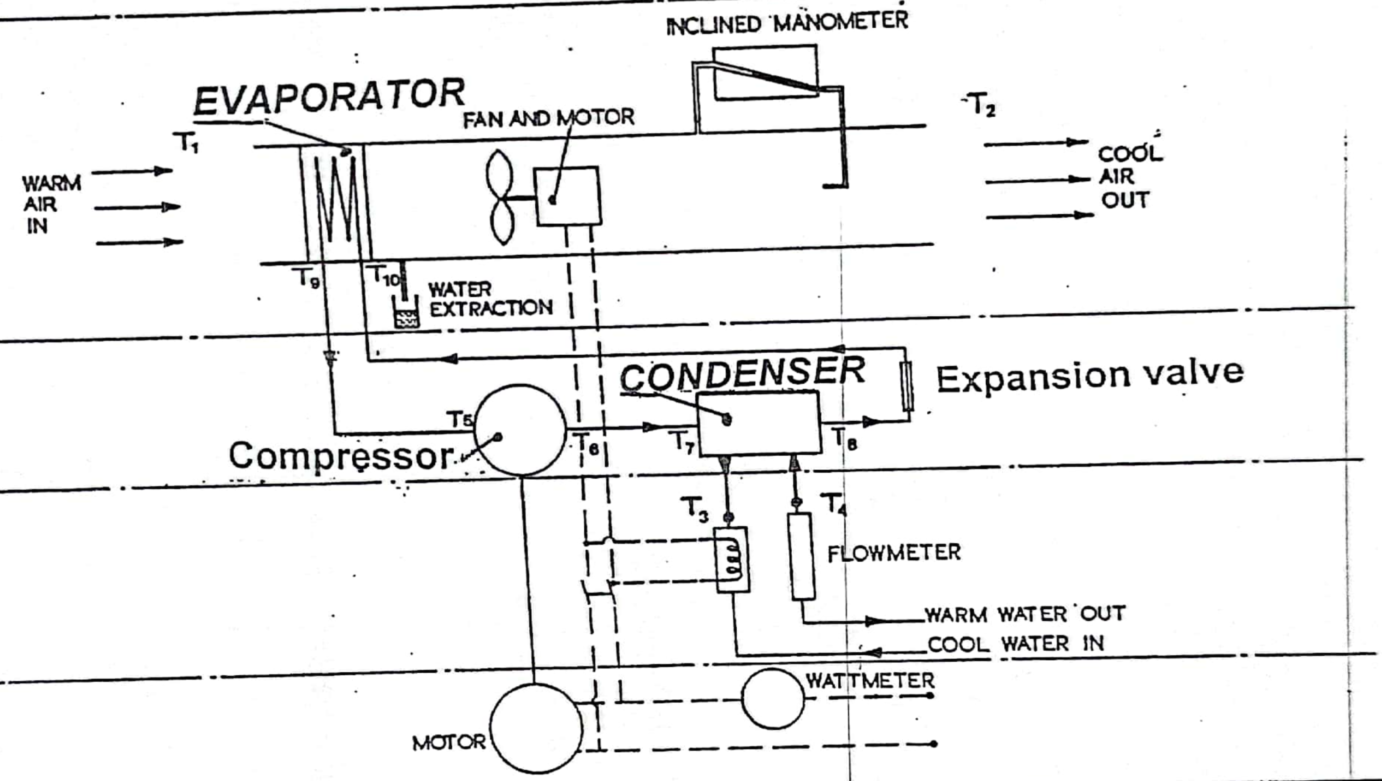


Figure 3 Flow diagram for an air-cooler.

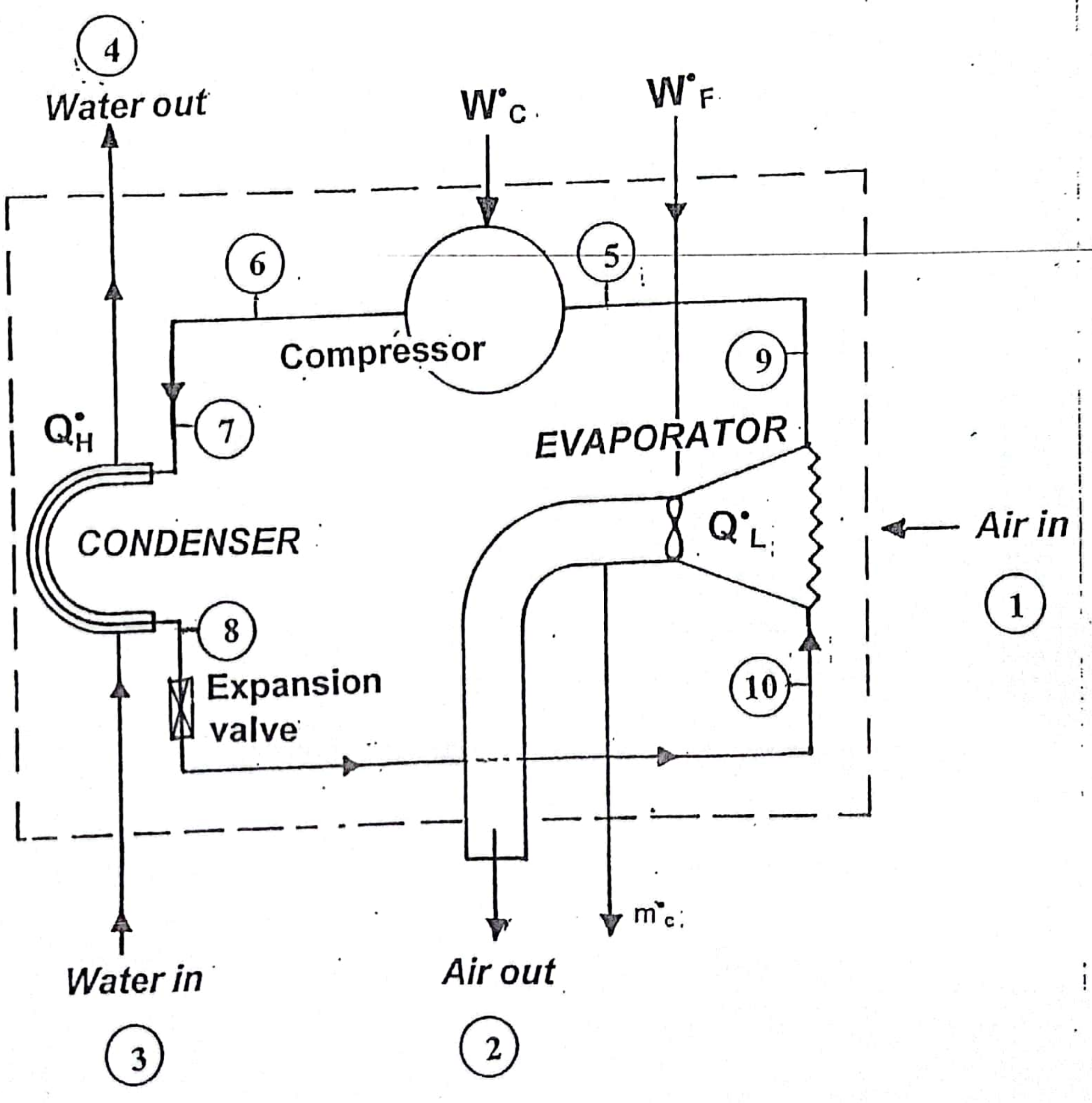


Figure 4 Energy flow diagram for an air-cooler.

* *Data collected*

The experimental temperatures

Location	Temp (°C)	
Air in	T_1	
Air out	T_2	
Water inlet	T_3	
Water outlet	T_4	
Compressor inlet	T_5	
Compressor outlet	T_6	
Heat Ex. Water (comp. End)	T_7	
Heat Ex. Water (valve end)	T_8	
Heat Ex. Air (comp.End)	T_9	
Heat Ex. Air (valve End)	T_{10}	

Inlet dry bulb temp.	$T_{I,db}$	
Inlet wet bulb temp.	$T_{I,wb}$	
Exit dry bulb temp.	$T_{e,db}$	
Exit wet bulb temp.	$T_{e,wb}$	
Condensate temp.	T_c	

Manometer reading H_I = ---------------------- mm H_2O

Condensate rate $Q^\bullet$ = ---------------------- ml/min

Compressor power $W^\bullet_c$ = ---------------------- kW

compressor & fan ← Total electric power $W^\bullet_T$ = ---------------------- kW

<u>Mass Flow Rates</u>

Dry air $m^\bullet_a$ = ---------------------- kg/s

Condensate at discharge $m^\bullet_c$ = ---------------------- kg/s

Circulating water $m^\bullet_w$ = ---------------------- kg/s

$$COP_{heat} = \frac{Q_H}{W_{con}}$$

sycometric chart

$$COP_{cooler} = \frac{Q_L}{W_{compressor}}$$

Enthalpys

Dry air entering conditioner	h_1 = ------------------------kJ/kg
Water vapour entering conditioner	h_{v1} = ------------------------kJ/kg
Dry air leaving conditioner	h_2 = ------------------------kJ/kg
Water vapour leaving conditioner	h_{v2} = ------------------------kJ/kg
Condensate	h_c = ------------------------ kJ/kg
Circulating water at inlet	h_{f3} = ------------------------ kJ/kg
Circulating water at outlet	h_{f4} = ------------------------ kJ/kg

* ***Results and discussions***

Calculate the COP for the Air Cooler and compare it with the ideal theoretical value.

The Heat Pump

* *Theory*

The machine arrangement when operating as an air cooler is shown schematically in Figure 5. The steady state steady flow (SSSF) equation for the system , see Figure 6 is the same as for the air cooler and may be written as:

$$\boxed{Q^{\bullet}_{H} - W^{\bullet}_{c} - W^{\bullet}_{F} = Q^{\bullet}_{L}}$$

Where $Q^{\bullet}_{H}$ and $Q^{\bullet}_{L}$ can be found from the energy balance for both the condenser using circulating air, and the evaporator which is cooling the water, thus

$$Q^{\bullet}_{H} = m^{\bullet}_{a}(h_2 - h_1) + m^{\bullet}_{a}\, w(h_{v2} - h_{v1}) \quad \text{and}$$

heat

$$Q^{\bullet}_{L} = m^{\bullet}_{w}(h_{f4} - h_{f3})$$ → tables

Where

$m^{\bullet}_{a}$ = Mass flow rate of dry air (kg/s).

$m^{\bullet}_{w}$ = Mass flow rate of cooling water in the condenser (kg/s).

h_1 and h_2 =Enthalpies of dry air at inlet and exit conditions (kJ/kg)

w = Humidity ratio of air at air inlet and exit conditions (kg/kg of dry air) → Psychromtric chart

h_{v1} and h_{v2} = Enthalpies of water vapor carried by the air at inlet condition ($=h_{g1}$), and at exit condition ($=h_{g2}$) kJ/kg

$W^{\bullet}_{c}$ = Electrical input to the compressor (kW)

$W^{\bullet}_{F}$ = Electrical input to the compressor (kW)

The coefficient of performance is then defined as:

$$\beta' = \frac{Q^{\bullet}_{H}}{Q^{\bullet}_{H} - Q^{\bullet}_{L}} = \frac{Q^{\bullet}_{H}}{W^{\bullet}_{c} + W^{\bullet}_{F}}$$

This may be compared with an ideal performance based upon the temperature difference across the refrigerator circuit

$$(COP_{HP})_{max} = \frac{T_{10}}{T_{10} - T_{8}}$$

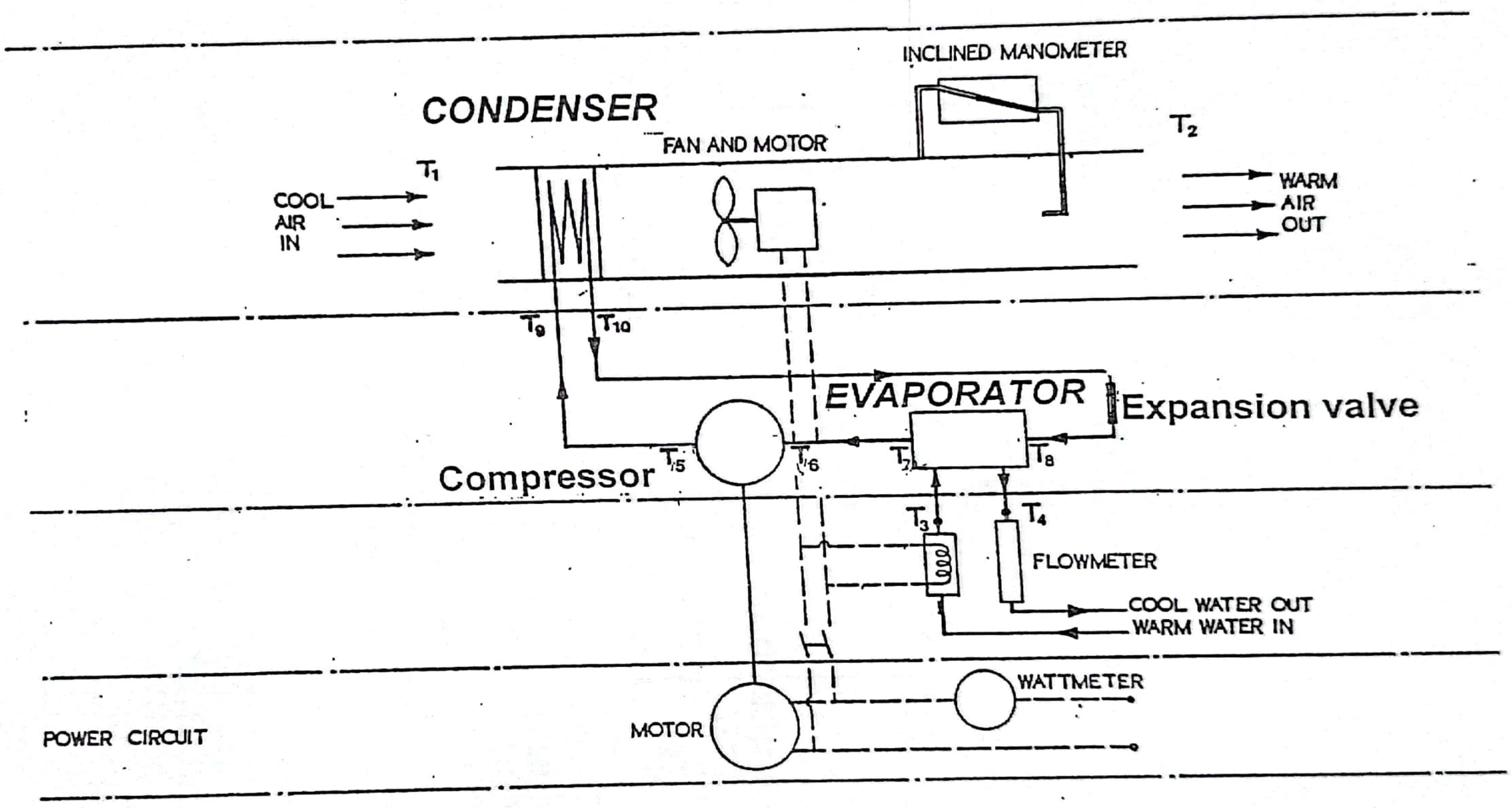


Figure 5 Flow diagram for a heat pump.

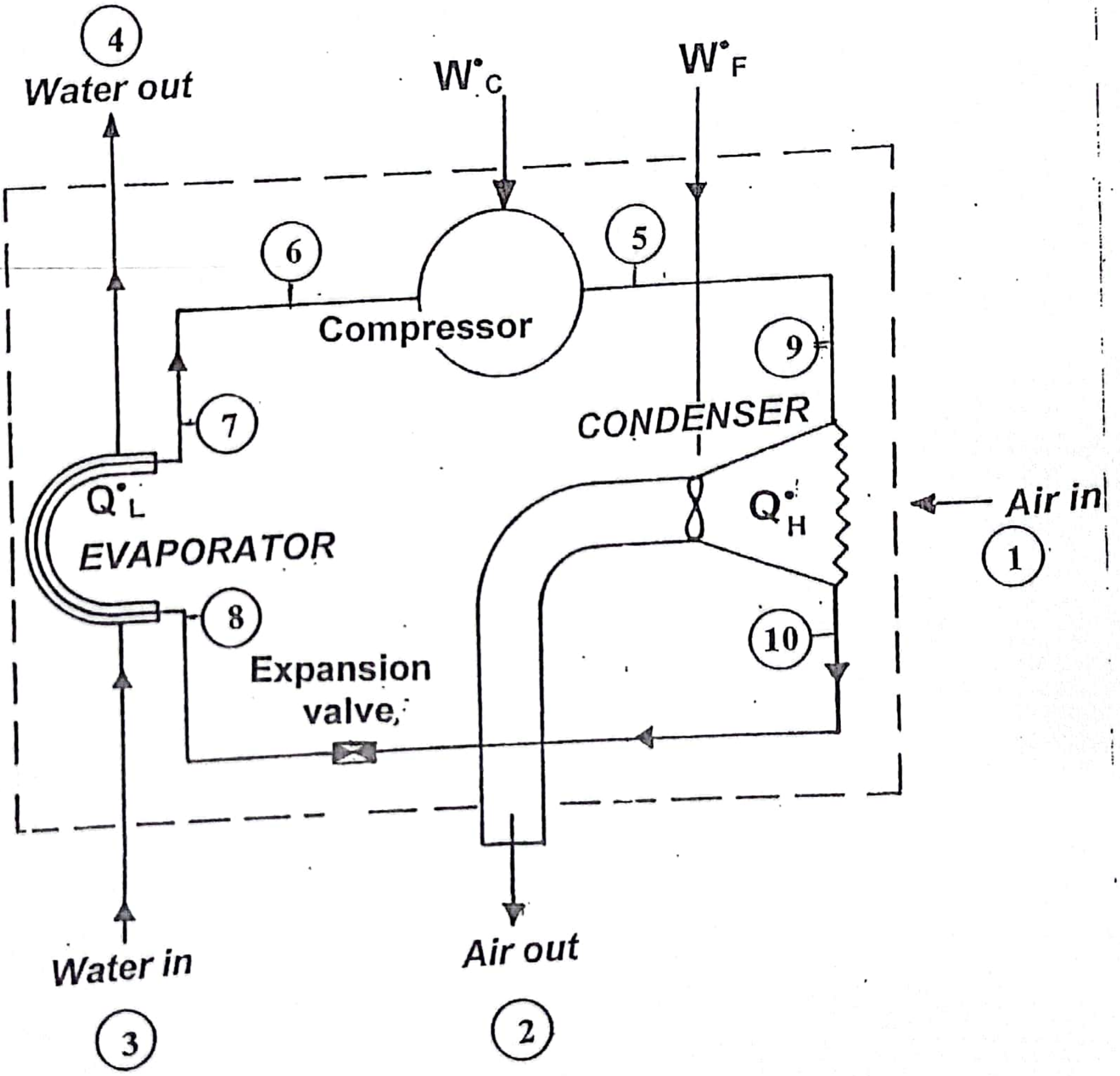


Figure 6 Energy flow diagram for a heat pump

In a real machine such as the present one the coefficient of performance falls short of the ideal for a number of reasons, of which the most important are:

a) Electrical and mechanical losses in the motors of both the fan and the compressor.

b) The imperfection (irreversibility) of the refrigeration cycle itself.

c) The necessity for temperature differences between refrigerant and air, and between refrigerant and water, as a result of which the refrigerant cycle operates between substantially wider temperature limits than those applicable to the water ad air forming the source and sink.

d) Electrical and mechanical losses in the circulating fan.

* ***Data collected***

The experimental temperatures

Location	Temp (°C)	
Air in	T_1	
Air out	T_2	
Water inlet	T_3	
Water outlet	T_4	
Compressor inlet	T_6	
Compressor outlet	T_5	
Heat Ex. Water (comp. End)	T_7	
Heat Ex. Water (valve end)	T_8	
Heat Ex. Air (comp.End)	T_9	
Heat Ex. Air (valve End)	T_{10}	

Inlet dry bulb temp.	$T_{I,db}$	
Inlet wet bulb temp.	$T_{I,wb}$	
Exit dry bulb temp,	$T_{e,db}$	
Exit wet bulb temp.	$T_{e,wb}$	
Condensate temp.	T_c	

Manometer reading H_I = ------------------------ mm H_2O

Compressor power $W^\bullet_c$ = ------------------------ kW

Total electric power $W^\bullet_T$ = ------------------------ kW

Mass Flow Rates

Dry air $m^\bullet_a$ = ------------------------kg/s

Condensate at discharge $m^\bullet_c$ = ------------------------ kg/s

Circulating water $m^\bullet_w$ = ------------------------ kg/s

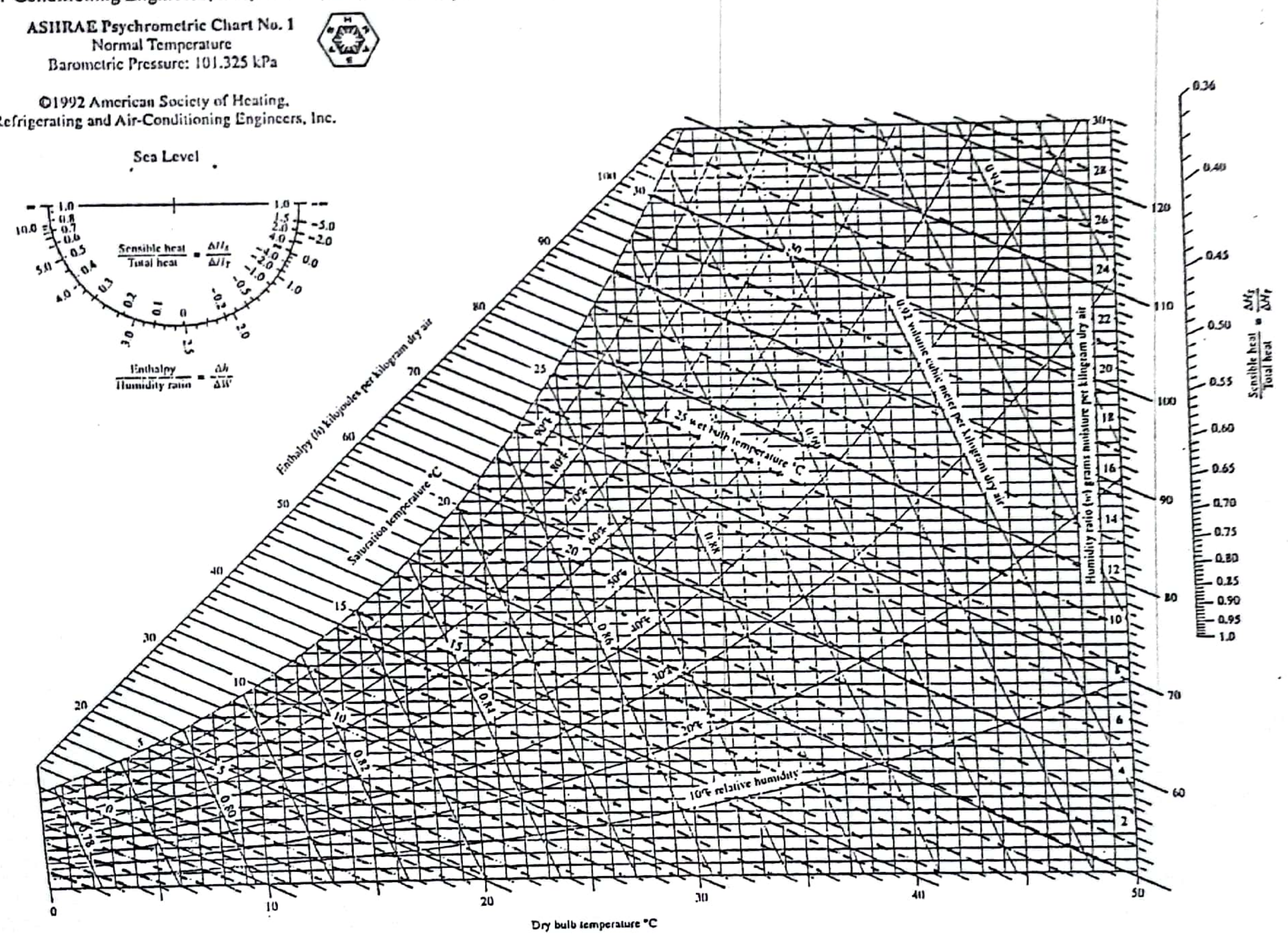
Enthalpies

Dry air entering conditioner	h_1 = ----------------------kJ/kg
Water vapour entering conditioner	h_{v1} = ----------------------kJ/kg
Dry air leaving conditiner	h_2 = ----------------------kJ/kg
Water vapour leaving conditoner	h_{v2} = ----------------------kJ/kg
Circulating water at inlet	h_{f3} = ----------------------kJ/kg
Circulating water at outlet	h_{f4} = ----------------------kJ/kg

* ***Results***

Calculate the COP for the Heat Pump and compare it with the ideal theoretical value.

Psychrometric chart at 1 atm total pressure. (Reprinted by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA; used with permission.)



University of Jordan
Faculty of Engineering & Technology
Mechanical Engineering Dept.

Thermal and Fluid Sciences Lab.

Dr. Adnan Jaradat

Experiment 8: ***Comparison of Pumps Characteristics***

Report ☐ ***Full*** ☐ ***Short***

Student Name:---

Student No. :---

Comparison of Pumps Characteristics

* Objectives

1. To establish a set of pump characteristics, at constant and variable speeds, for the following pumps :
   a) Positive (single cylinder double acting pump) displacement reciprocating pump.
   b) Centrifugal pump (horizontal)
2. To compare the performance characteristics of each pump under identical speed conditions.

* Theory

The primary function of pumps is not only transportation of fluids, but to add energy to the transported fluids. Different types of pumps are designed to process fluids under variable engineering conditions.

1. Calculation of pump characteristic parameters:

   a) $\boxed{\text{Water power} = \rho g Q^{\bullet} h_p 10^{-3}}$ (kW) , also called the pump power

   experimental relationship

   Where

   ρ = Density of water (kg/m^3). $= 1000$

   g = Acceleration due to gravity (m/s^2). $= 9.81$

   $Q^{\bullet}$ = Volumetric flow rate (m^3/s). $= \frac{m}{\rho \Delta t}$

   hp = Head gain in meter of water (the pump head). In terms of pressure difference ΔP, through the pump, the pump head is given by:

   head pump ←

   $\boxed{hp = \Delta P / g\rho 10^{-5}}$

   $\Delta P = (\text{delivery pressure} - \text{suction pressure})$

   P_d P_s

   Where

   ΔP = Delivery pressure (P_d) – suction pressure (P_s), bar

   قدرة الطاقة (الي المستهلكة)

   b) $\boxed{\text{Brake power} = 2\pi\, \omega F R 10^{-3} \text{ (kW).}}$

   Where

   ω = Motor speed (rev/s).

   ~~F = Brake load (N)~~ = Spring load (kg) × 9.81.

   R = Torque arm radius (m) = 0.15 m.

   c) $\boxed{\text{Overall efficiency } \eta_o = \frac{\text{Water power}}{\text{Brake power}}}$ < 1

$\frac{\dot{m}}{\rho} = 1000\ kg/m^3$

d) Volumetric efficiency $\eta_v = \frac{\text{measured volumetric flow rate } Q^{\bullet}}{\text{Calculated volumetric flow rate } Q_c^{\bullet}}$ $= \frac{\dot{Q}_{measured}}{\dot{Q}_{calculated}}$

e) The calculated volumetric flow rate $Q_c^{\bullet}$ is obtained as follows:-

a. **For Positive displacement reciprocating pump:-**

For single acting pump

$$Q_c^{\bullet} = A_p L \omega \quad (m^3/s)$$

For a double acting pump

$$Q_c^{\bullet} = 2 A_p L \omega \quad (m^3/s)$$

Where

A_p = Total cross sectional area of cylinder (m^2) $\boxed{= 15.55 \times 10^{-4}\ m^2}$

L = Stroke of piston (m) = $\boxed{0.0413\ m}$

ω = Pump speed (rev/s)

For the present arrangement $\boxed{\omega = \frac{\text{motor speed}}{5}}$

b. **For centrifugal pump:-**

$$Q_c^{\bullet} = \frac{0.75}{12.5} \times 10^{-3}\ \omega \quad (m^3/s)$$

For the present arrangement $\boxed{\omega = 2(\text{motor speed})}$

* *Apparatus*

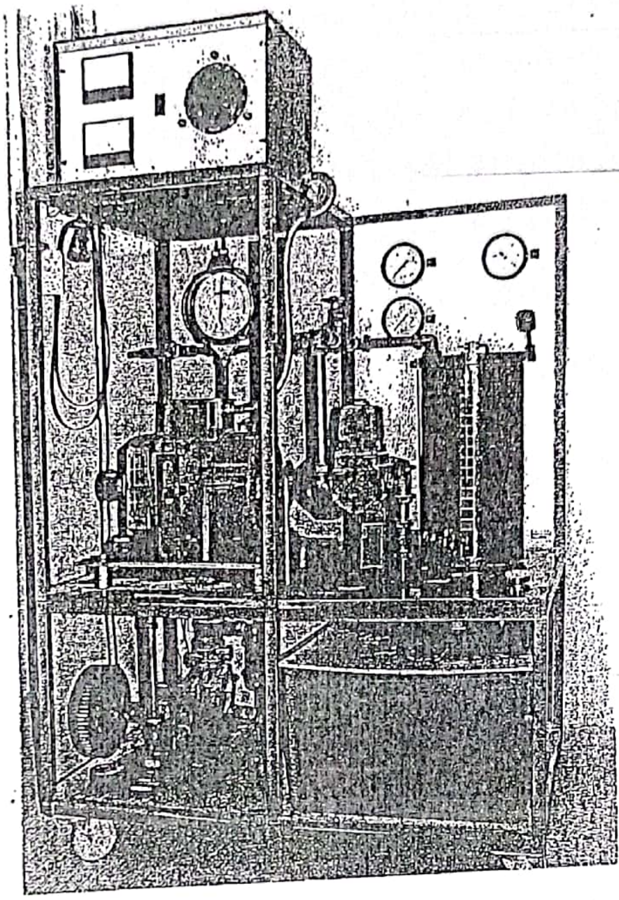
The water circuit incorporates a reservoir, a volumetric measuring tank with over flow, and bottom release valve mounted above the reservoir. In addition the necessary pipe work and valve are provided. The driving motor is a D.C. machine coupled to a spring dynamometer, Figure 1, The system is instrumented for the measurement of speed, torque, flow rate and the necessary pressures. The Universal Pump Test Bed comprises three pumps.

max current

a. **Positive displacement reciprocating Pump.**

A reciprocating pump driven by a motor (power pump), where the pump piston or plunger is connected to a crank-shaft that is geared to counter-shaft and connected to the driving motor by toothed belt. The operation of power driven reciprocating pump is such that the outward stroke of plunger creates a suction behind it enabling the liquid to flow into the cylinder through the suction valve.

vertical



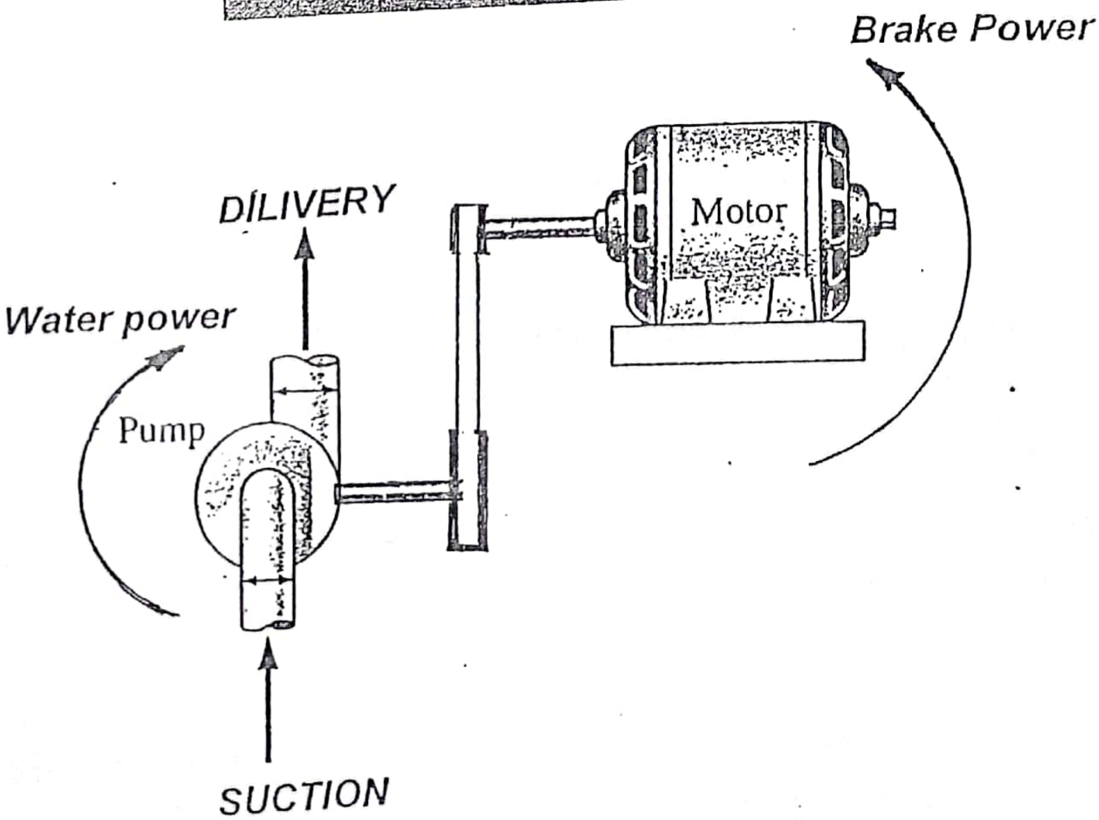


Figure 1 The Apparatus.

On the reverse stroke the liquid is forced to flow out of discharge valve.

This is a Stuart Turner single cylinder double acting pump having a nominal output of 0.45 liters/s at a head of 53 meters when running at the maximum speed of 4 rev/s. Bore and stroke are 44.5 and 41.3 mm respectively. The Reciprocating pump is connected to the D.C. motor using a toothed belt and pulley giving a motor/pump speed ratio of 5 to 1.

b. Centrifugal Pump.

This is a horizontal centrifugal pump which has a nominal output of 0.75 liters/s at a head of 1.06 meters when running at 12.5 rev/sec. Connection to the D.C. motor by toothed belt and pulleys gives a motor/pump speed ratio of 1 to 2.

c. Gear Pump.

This is an inviolute gear type with a nominal output of 0.2 liters/sec when running at 5 rev/sec. It is connected to the D.C. dynamometer by a toothed belt drive and pulley system giving a motor/pump speed ratio of 3 to 1.

Pressure angle 20°	No. of teeth 12
Diametrical pitch 0.236 teeth/mm	Face width 41.3 mm
Root diameter 41.5 mm	Clearance 0.7 mm

Note: This pump is not working in the present set-up.

* *Procedure*

Starting

1. Fully open the suction and delivery valves of the pump to be tested and isolate any other pumps in the system by closing their corresponding valves.
2. Open the pressure gauge valves for the pump to be tested.
3. Position the motor variable speed control on the control at zero volts. (fully anti-clockwise).
4. Switch on the power supply to the unit .
5. With the motor torque arm in the horizontal position adjust the spring dynamometer to give a zero reading. The unit is now ready to run tests.

Precaution

Do not run the motor for periods of more than10 minutes at an armature current of 5 amps which is the maximum permitted current for continuous running.

Stopping

1. Return the motor variable speed control to the zero position. (fully anti-clockwise)
2. Close the pump inlet valve (s). When using the centrifugal pump first close the discharge valve before the inlet valve if the pump is to remain primed.

Constant speed test

1. **Positive displacement reciprocating Pump**
   - Gradually increase the voltage applied to the motor until the pump is running at the operation speed selected.
   - Adjust the delivery pressure to the maximum value.
   - Record the brake load.
   - Record the shaft speed.
   - Measure the water flow rate by recording the time taken to collect a measured volume of water. (15 Liters)
   - Read suction and delivery pressures. Repeat this procedure for lower values of delivery pressure.

2. **Centrifugal pump:-**
   - Bring the pump gradually up to the selected operating speed and open inlet and outlet valves to allow maximum delivery.
   - Note suction and delivery pressures.
   - Note brake load.
   - Note pump speed.
   - Note flow rate.
   - Repeat for increments of delivery pressure closing the delivery valve in several steps.

Variable speed tests

1. **Positive displacement reciprocating pump**
   - Bring the pump gradually to maximum speed and delivery pressure.
   - Note delivery pressure, suction pressure, brake load, pump speed and volumetric flow rate.
   - Repeat this procedure for lower values of pump speed.
   - For each pump speed selected, maintain a constant pressure rise by adjusting the delivery valve.

* *Data collected*

* Centrifugal Pump:

Table 1 Data for the centrifugal pump

w = 15 rev/s **Current ....(A)**

P_s (bar	P_d (bar)	Spring load (kg)	Mass of water (kg)	Time (s)
0	0.4		5	
0	0.5		5	
0	0.6		5	
0	0.7		5	
0	0.8		5	

For a constant P_d = 0.4 (bar) **Current ....(A)**

P_s (bar)	w (rev/s)	Spring load (kg)	Mass of water (kg)	Time (s)
0	10		5	
0	12		5	
0	15		5	
0	17		5	
0	20		5	

* Reciprocating Pump

Table 2 Data for the reciprocating pump

* w = 15 rev.sec.

P_s (bar)	P_d (bar)	Spring load (kg)	Mass of water (kg)	Time (s)	Current (A)
0	0.5		5		
0	1.0		5		
0	2.0		5		
0	3.0		5		
0	4.0		5		

* P_d = 1.5 (bar)

Current ....(A)

w (rev/s)	P_s (bar)	Spring load (kg)	Mass of water (kg)	Time (s)	Current (A)
10	0		5		
12	0		5		
14	0		5		
16	0		5		
18	0		5		
20	0				

Results

Table 3 Results for the centrifugal pump

Centrifugal Pump

* w = 10 rev/s

P_d (bar)	$\dot{Q}$ (m^3/s)	Brake Load F (N)	Water Power (kW)	Brake Power (kW)	Overall Efficiency η_o	Pump Head h_p (m)	Volumetric Efficiency η_v
0.1							
0.2							
0.3							
0.4							
0.5							

$\frac{\dot{Q}}{\dot{Q}_c}$

* w = 15 rev/s

0.4							
0.5							
0.6							
0.7							
0.8							

* P_d = 0.4 (bar)

W = 10							
12							
15							
17							
20							

$\dot{Q}_c \Rightarrow$ centri $\frac{0.75}{12.5} \times 10^{-3}$ (w) (m^3/s)

$\rightarrow$ reci: single $\rightarrow A_p L w$; double $\rightarrow 2 A_p L w$

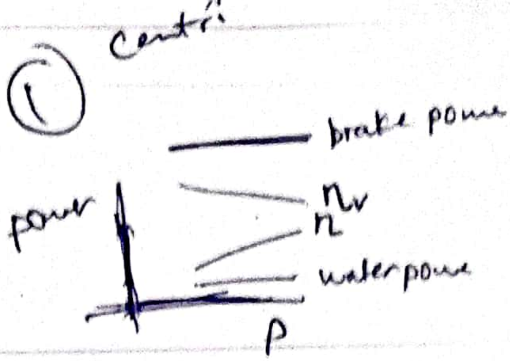
$A_p = 15.55$

Reciprocating Pump

Table 4 Results for the reciprocating pump

P_d (bar)	$Q^{\cdot}$ (m^3/s)	Brake Load F (N)	Water Power (kW)	Brake Power (kW)	Overal Efficiency η_0	Pump Head h_p (m)	Volumetric Efficiency η_v
0.5							
1.0							
2.0							
3.0							
4							

ω Rev/s	P_d (bar)	$Q^{\cdot}$ (m^3/s)	Brake Load F (N)	Water Power (kW)	Brake Power (kW)	Overal Efficiency η_0	Pump Head h_p (m)	Volumetric Efficiency η_v
10	1.5							
12	1.5							
14	1.5							
16	1.5							
18	1.5							
20	1.5							



* *Results and discussions*

Constant speed tests

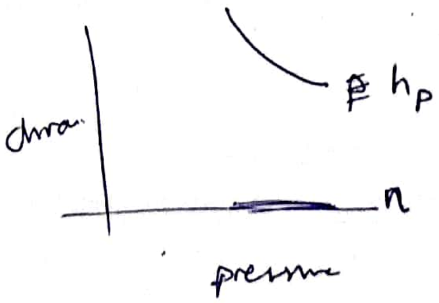
1. For both pumps
   On the same figure, but using different scales, show the relationship between water power, brake power, overall efficiency and volumetric efficiency with output pressure.

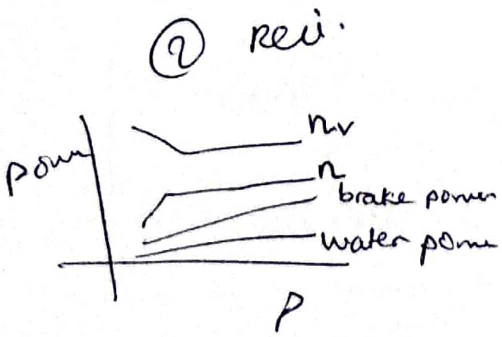
2. For the centrifugal pump
   On the same figure, but using different scales, show the relationship of pump head, water power, overall efficiency with volumetric flow rate.

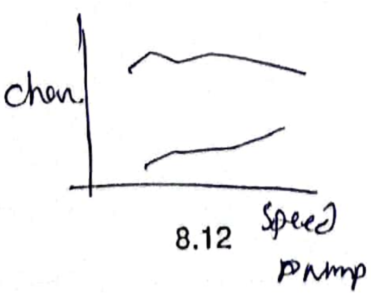
3. Comment on the most suitable applications for these two types of pumps and on their limitations.

Variable speed tests

For the positive displacement pump only, and on the same figure, show the relationship between brake power, input torque and volumetric flow rate with pump speeds .







University of Jordan
Faculty of Engineering & Technology
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Thermal and Fluid Sciences Lab.

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Experiment 9: *Flow Visualization*

Report ☐ *Full* ☐ *Short*

Student Name:--

Student No. :---

Flow Visualization

* Objectives

To make the student familiar with the flow patterns and to see the stream lines when different shapes of objects are inserted in the flow path.

* Apparatus

a - Flow Visualization equipment

The flow visualization equipment is powered by a synchronous A.C. motor driving a centrifugal blower, which produces a maximum wind speed of about 12 m/s in the test section. A particularly uniform low-turbulence flow is achieved by the use of 3 gauze screens, a small-cell honeycomb and a 9:1 light hill contraction. The test section is effectively 2-dimensional, being 25.4 mm wide, 254 mm deep and 267 mm long. A viewing area of 267 mm X 229 mm is illuminated by an integral strip light and fitted with a window incorporating the model attachment and means of rotation. Provision is made for the fitting of a smoke generator.

b- Integrated smoke generator. (Figure 1)

In this equipment, an oil mist is formed by the atomization of a heated mineral oil in an air stream. The resultant "vapor" being largely a suspension of fine liquid droplets, from a highly visible "smoke" with contrast characteristics particularly suitable for photography.

Oil is gravity-fed from an oil reservoir to a pyrex vaporizing tube incorporating an electrical heater. The vapor, entrained in an air stream provided by a twin diaphragm pump, is then passed to a flexible smoke reservoir which serves to smooth the flow and damp out variations in density. (It also facilitates clearance of condensed oil from the outlet tubes). Fine control over the generation of smoke is provided by a valve adjacent to the vaporizing tube. The contents of the smoke reservoir are discharged through on outlet pipe to the smoke rake provided with the equipment. A throttle clamp enables smoke circuit oscillations to be damped out.