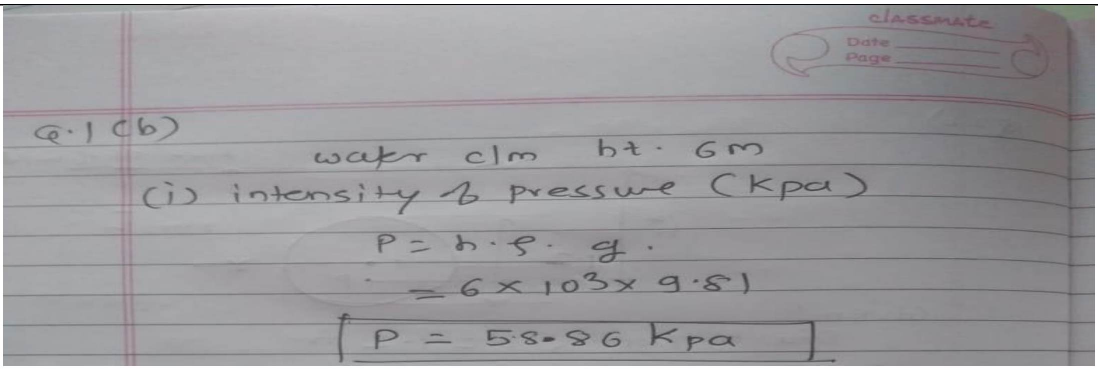
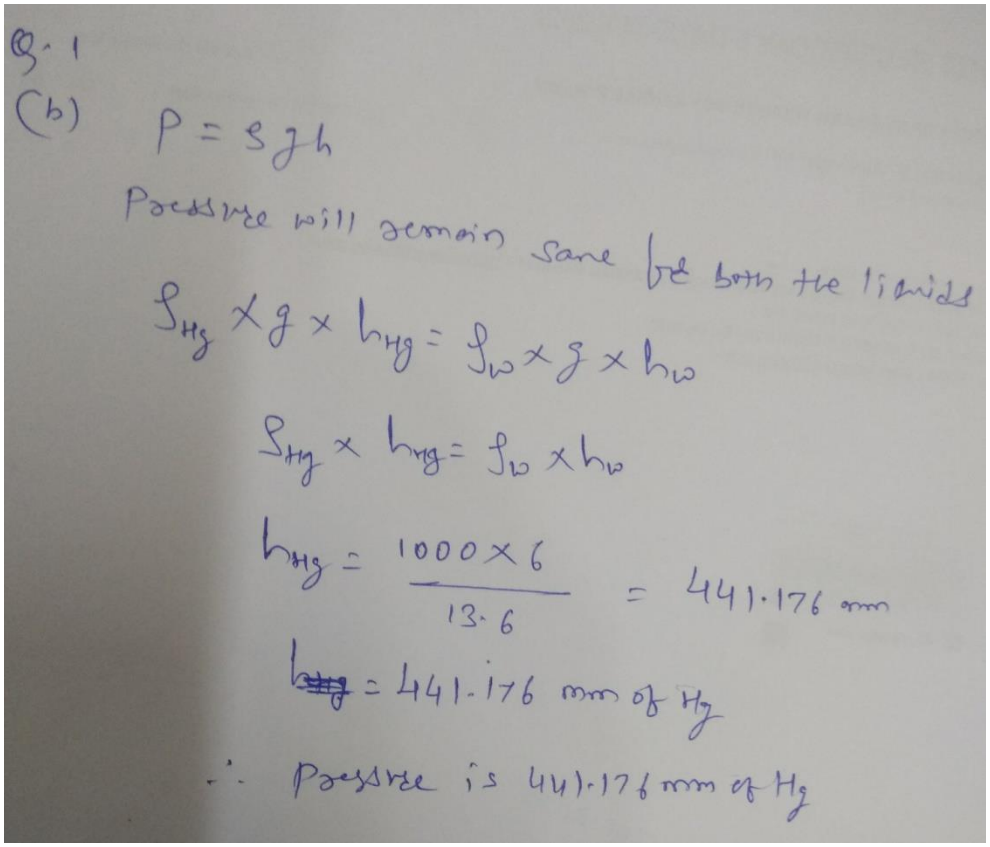
SUMMER – 2022 EXAMINATION

Subject Name: **Model AnswerSubject Code:** 22445

Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more Importance (Not applicable for subject English and Communication Skills.
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgement on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.
8) As per the policy decision of Maharashtra State Government, teaching in English/Marathi and Bilingual (English + Marathi) medium is introduced at first year of AICTE diploma Programme from academic year 2021-2022. Hence if the students in first year (first and second semesters) write answers in Marathi or bilingual language (English +Marathi), the Examiner shall consider the same and assess the answer based on matching of concepts with model answer.

Q.1.		Attempt any FIVE of the following:	10 Marks
a)	i)	**Surface tension:** The property of the fluid which enables it to resist tensile stressis called surface tension.	01 M
	ii)	**Dynamic viscosity**: Dynamic viscosity μ, may be defined as the shear stress required to produce unit rate of angular deformation. Mathematically, $\mu = \frac{\tau}{dv/dy}$	01 M (either definition or methematical relation)
b)		 Q.1 (b) water c/m ht. 6m (i) intensity of pressure (kpa) P = h.ρ. g. = 6×10³× 9.81 $\boxed{P = 58.86 \text{ kpa}}$	1 M for P

		10. 33 m ----- 760 mm og Hg 6 m = ? $\frac{6}{10.33} * 760 = \mathbf{441.41\ mm\ of\ Hg}$ **OR**  Q. 1 (b) P = ρgh Pressure will remain same for both the liquids $\rho_{Hg} \times g \times h_{Hg} = \rho_w \times g \times h_w$ $\rho_{Hg} \times h_{Hg} = \rho_w \times h_w$ $h_{Hg} = \frac{1000 \times 6}{13.6} = 441.176$ mm h_{Hg} = 441.176 mm of Hg ∴ Pressure is 441.176 mm of Hg	1M for pressure in mm of hg
c)	i) ii)	**Steady Flow:**Fluid flow is said to be steady if at any point in the flowing fluid various characteristics such as velocity, pressure, density, temperature etc., do not change with time. **Unsteady or non-steady flow**: -Fluid flow is said to be unsteady if at any point in the flowing fluid any one or all the characteristics such as velocity, pressure, density, temperature etc., changes with time.	1M 1M
d)		**Laws of fluid friction for Turbulent Flow** Frictional resistance is proportional to square of velocity of flow.	2M

Frictional resistance is independent of pressure.

Frictional resistance slightly varies with change in temperature of fluid.

Frictional resistance is proportional to density of fluid flow.

e) **Minor Losses:-**

Loss of head at Entry.

Loss of head at Exit.

Loss of head due to sudden enlargement.

Loss of head due to sudden contraction

Loss of head due to sudden obstruction.

Loss of head due to bend or Elbow.

Any four Losses 2M

f)

Sr. No.	Specific Speed	Type of turbine
i)	8.5 to 30(10 to 35)	Pelton wheel with single jet
ii)	50 to 340(60 to 400)	Francis turbine
iii)	300 to 1000	Kaplan turbine or propeller turbine

2M

g) **The main components of reciprocating pump are as follows:**

Suction Pipe

Suction Valve

Delivery Pipe

Delivery Valve

Cylinder

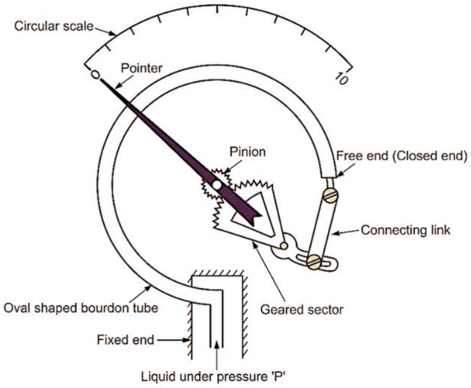
Piston and Piston Rod

Crank and Connecting Rod

Strainer

Any four

2 M

2		**Attempt any THREE of the following:**	12 M
a)		**Bourdon tube pressure gauges** are used for the measurement of relative pressures from 0.6 ... 7,000 bar. They are classified as mechanical pressure measuring instruments, and thus operate without any electrical power. **Bourdon tube pressure gauge** Bourdon tubes are radially formed tubes with an oval cross-section. The pressure of the measuring medium acts on the inside of the tube and produces a motion in the non-clamped end of the tube. This motion is the measure of the pressure and is indicated via the movement. The C-shaped Bourdon tubes, formed into an angle of approx. 250°, can be used for pressures up to 60 bar. For higher pressures, Bourdon tubes with several superimposed windings of the same angular diameter (helical tubes) or with a spiral coil in the one plane (spiral tubes) are used.  	02 M 02 M For fig.

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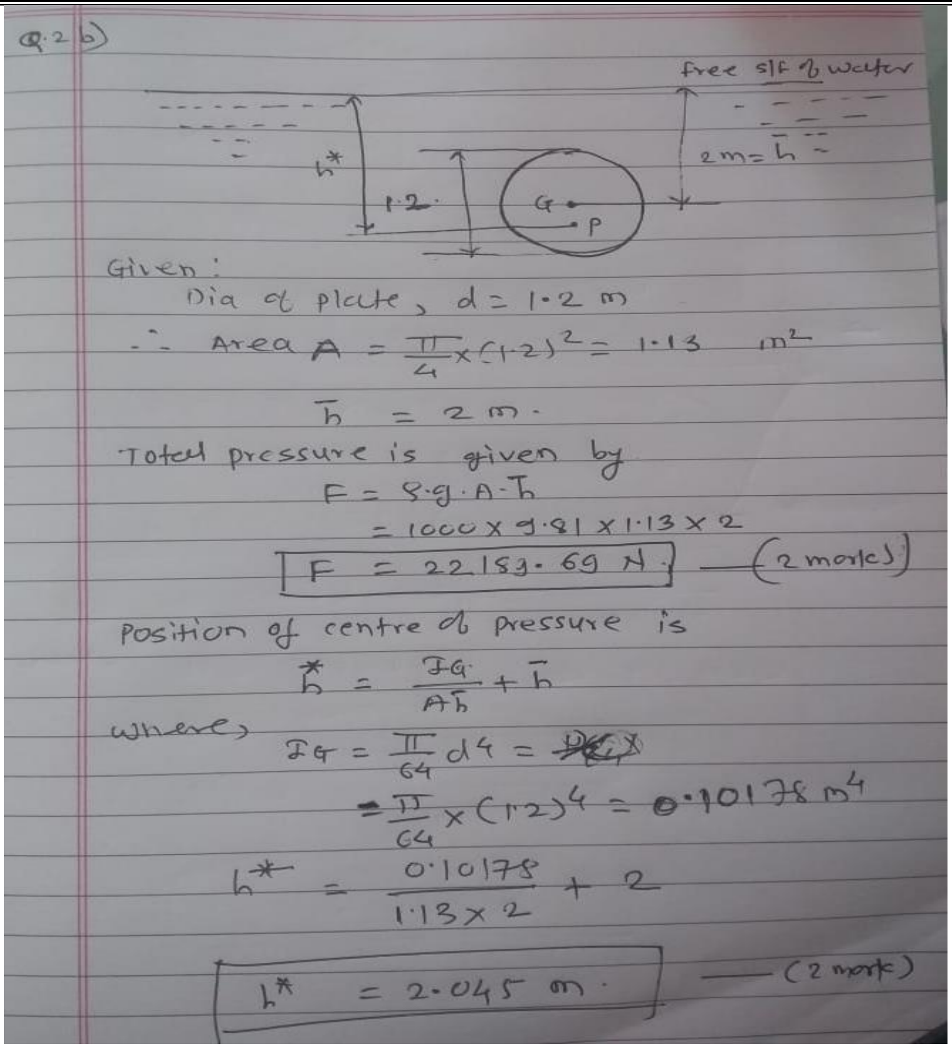

Q·2 b)

Given :

Dia of plate, d = 1·2 m

$\therefore$ Area A = $\frac{\pi}{4} \times (1·2)^2 = 1·13$ m^2

$\bar{h}$ = 2 m.

Total pressure is given by

$F = \rho \cdot g \cdot A \cdot \bar{h}$

$= 1000 \times 9·81 \times 1·13 \times 2$

$\boxed{F = 22159·69 \ N}$ — (2 marks)

Position of centre of pressure is

$h^* = \frac{I_G}{A\bar{h}} + \bar{h}$

where,

$I_G = \frac{\pi}{64} d^4 =$ [illegible]

$= \frac{\pi}{64} \times (1·2)^4 = 0·10178 \ m^4$

$h^* = \frac{0·10178}{1·13 \times 2} + 2$

$\boxed{h^* = 2·045 \ m.}$ — (2 mark)
</td>
<td>01 m to find out area (A), 01 m for F,
01 m for IG,
01 m for h</td>
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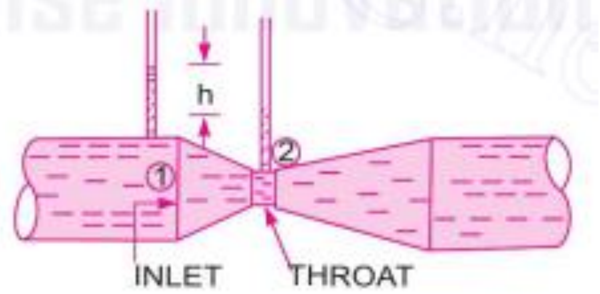
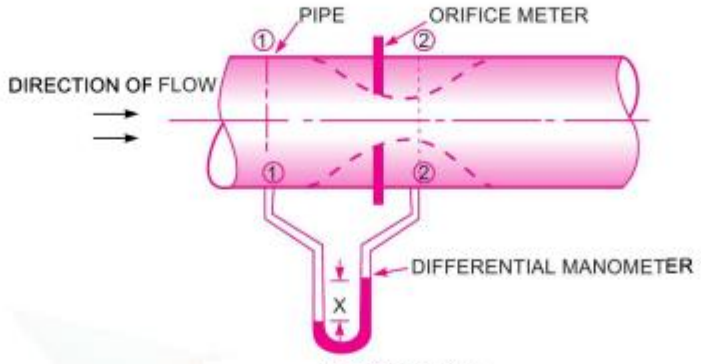
c)		 *Venturimeter.*	Fig.
		Construction : As stated above it has three parts converging part, throat and diverging part. These three parts are arranged in systematic order. First one is inlet section or converging section. It is the region where the cross section emerges into conical shape for the connectivity with the throat region. .In this part cross section area decreases from beginning to ending. This section is connected to inlet pipe on one end and cylindrical throat on the other end. The angle of convergence is generally 20-22 degrees . Second one is cylindrical throat .It is the middle part of the venturimeter. It is the cylindrical pipe in venturimeter through which the fluid passes after converging in the convergent section. Throat has generally a diameter of throat is half the diameter of pipe. The diameter of the throat remains same through out its length. Last one is diverging section . It is the end of the venturimeter. On one side it is attached to throat of venturimeter and on the other side it is attached to the pipe. The divergent section has an angle 5 to 15 degrees . The diverging angle is less than the converging angle because the length of the diverging cone is larger than converging cone. The main reason of the small diverging angle is to avoid flow seperation from the walls.	1& 1/2 M for construction
		Working :- Venturimeter works on the principle of Bernoulli's equation i.e when velocity increases pressure decreases . Cross section of throat is less than cross section of inlet pipe. Since the cross -section decreases from inlet pipe to throat, the velocity of the fluid increases and hence the pressure decreases. Due to decrease in pressure, a pressure difference is created between the inlet pipe and throat of the venturimeter . This pressure difference can be measured by placing a differential manometer between the inlet section and throat section or by using two guages at inlet section and throat. After getting the pressure difference flow rate through pipe is calculated.	1& 1/2 M working

d)

1M for Fig



Orifice meter.

d1 = Diameter at section 1 (Inlet section)
P1 = Pressure at section 1 (Inlet section)
v1 = Velocity of fluid at section 1 (Inlet section)
A1 = Area of pipe at section 1 (Inlet section)
d2 = Diameter at section 2
P2 = Pressure at section 2
v2 = Velocity of fluid at section 2
A2 = Area at section 2

3 M for Equation

We will have following equation after applying Bernoulli's equation at section 1 and section 2.

$$\frac{p_1}{\rho g}+\frac{v_1^{\,2}}{2g}+z_1=\frac{p_2}{\rho g}+\frac{v_2^{\,2}}{2g}+z_2$$

$$\Rightarrow\left(\frac{p_1}{\rho g}+z_1\right)-\left(\frac{p_2}{\rho g}+z_2\right)=\frac{v_2^{\,2}-v_1^{\,2}}{2g}$$

$$\Rightarrow h=\frac{v_2^{\,2}-v_1^{\,2}}{2g}$$

$$\Rightarrow v_2=\sqrt{2gh+v_1^2}$$

where h is the differential head.

Let A0 is the area of the orifice
Co-efficient of contraction, CC = A2/A0
Let us recall the continuity equation and we will have following equation

$$A_1 v_1 = A_2 v_2$$

$$\Rightarrow v_1 = \frac{A_0 C_c}{A_1} v_2$$

$$v_2 = \sqrt{2gh + \frac{A_0^2 C_c^2 v_2^2}{A_1^2}}$$

$$\Rightarrow v_2 = \frac{\sqrt{2gh}}{\sqrt{1 - \frac{A_0^2}{A_1^2} C_c^2}}$$

Thus, discharge,

$$Q = A_2 v_2 = v_2 A_0 C_c = \frac{A_0 C_c \sqrt{2gh}}{\sqrt{1 - \frac{A_0^2}{A_1^2} C_c^2}}$$

If C_d is the co-efficient of discharge for orifice meter, which is defined as

$$C_d = C_c \frac{\sqrt{1 - \frac{A_0^2}{A_1^2}}}{\sqrt{1 - \frac{A_0^2}{A_1^2} C_c^2}}$$

$$\Rightarrow C_c = C_d \frac{\sqrt{1 - \frac{A_0^2}{A_1^2} C_c^2}}{\sqrt{1 - \frac{A_0^2}{A_1^2}}}$$

Thus we will use the value of CC in above equation of discharge Q and we will have following result for rate of flow or discharge through orifice meter.

$$Q = C_d \frac{A_0 A_1 \sqrt{2gh}}{\sqrt{A_1^2 - A_0^2}}$$

Co-efficient of discharge of the orifice meter will be quite small as compared to the co-efficient of discharge of the venturimeter.

3	a	**Attempt any THREE of the following:** **A venturi meter having throat diameter 6.3 cm is provided on a pipe of 15 cm diameter. If oil of specific gravity 0.88 is flowing in the upward direction, determine the Ventury head and the discharge if the manometer shows 12.80 cm of mercury deflection. If the vertical**	01 m to find h(ventury head), 02 m for

distance between inlet and throat is 22 cm. Determine the actual head of the venturi meter. Assume Cd = 0.65.

Throat dia, d2= 6.3cm ,$a2 = \frac{\pi}{4}d2^2$= 31.17 cm2

Pipe dia , d1= 15 cm,$a1 = \frac{\pi}{4}d1^2$= 176.71 cm2

Sp Gravity of oil Soil= 0.88

Manometer Reading , x= 12.8 cm of Hg

Cd =0.65

1.Ventury Head (h)

$h = x\left(\frac{Shg}{Soil} - 1\right) = 12.8\left(\frac{13.6}{0.88} - 1\right) = 185.01 cm\ of\ oil$----------------------------01 mark

2.Discharge (Q) ---02 mark

$$Q = Cd * a1 * a2 * \frac{\sqrt{2gh}}{\sqrt{a1^2 - a2^2}}$$

$$= 0.65 * 176.71 * 31.17 * \frac{\sqrt{2 * 981 * 18501}}{\sqrt{176.71^2 - 31.17^2}}$$

Q= 12400.99 cm3/sec

Q= 12.4 litres / sec

3.Actual Ventury Head if Z2-Z1= 22cm ----------------------------------01 mark

$$\left(\frac{p1}{\rho g} + Z1\right) - \left(\frac{p2}{\rho g} + Z2\right) = h$$

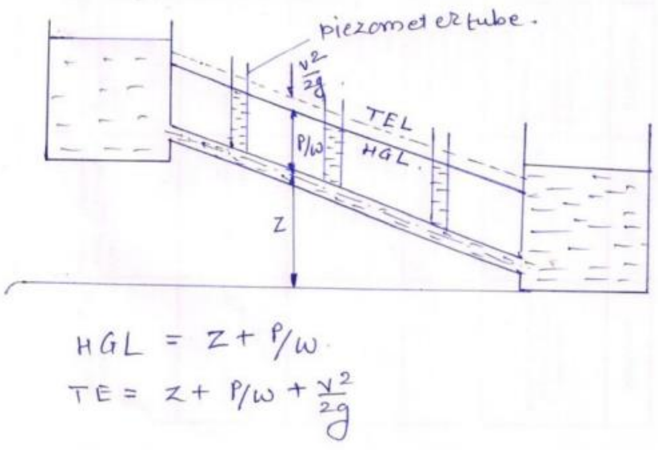
$$\left(\frac{p1}{\rho g} - \frac{p2}{\rho g}\right) + Z1 - Z2 = h$$

Z2-Z1= 22 cm , h = 185.01

Therefore ,

$$\left(\frac{p1}{\rho g} - \frac{p2}{\rho g}\right) = 185.01 + 22 = 207.01\ cm\ of\ oil$$

discharge, 01 for actual head

	b	**Explain the terms hydraulic gradient and total energy lines with diagram.** Hydraulic Gradient Line It is defined as the line which gives the sum of pressure head (p/w) and datum head (z) of a flowing fluid in a pipe with respect to some reference line. OR It is the line which is obtained by joining the top of all vertical ordinates, showing the pressure head of a flowing fluid in a pipe from center of the pipe. Total Energy Line (TEL) (Explain with diagram 2m) It is defined as the line which gives the sum of pressure head, datum head and kinetic head of a flowing fluid in a pipe with respect to some reference line. OR It is defined as the line which is obtained by joining the tops of all the tops of all vertical ordinates showing the sum of pressure head and kinetic head from center of the pipe.  	02 marks sketch, 02 marks explanation
	c	**Find the diameter of a pipe of length 9 km, when rate of flow of water through the pipe is 255 litre/sec. and head loss due to friction is 6.5 m. Take C = 55 for Chezy's formula.** Given Data: L=9 km = 9000 m, Q= 255 lit /sec = 0.255 m3/sec, hf= 6.5 m---------(1 mark) $Q = A * V \ ,V = \frac{Q}{A} = \frac{0.255}{\left(\frac{\pi}{4}\right)*d^2}$ ---(1 mark) Chezy's formula , V= $C\sqrt{mi} = 55\sqrt{\frac{d}{4}*\frac{hf}{L}} = 55\sqrt{\frac{d}{4}*\frac{6.5}{9000}}$-----------(2 mark)	01 m for unit conversion , 01 m for Q, 01 for chezy'sformula ,01 mark for finding d.

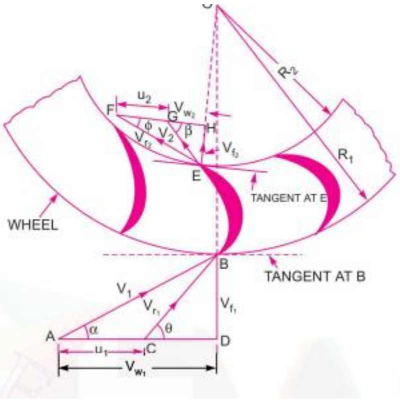
$$\frac{0.255}{\left(\frac{\pi}{4}\right)*d^2} = 55\sqrt{\frac{d}{4} * \frac{6.5}{9000}}$$

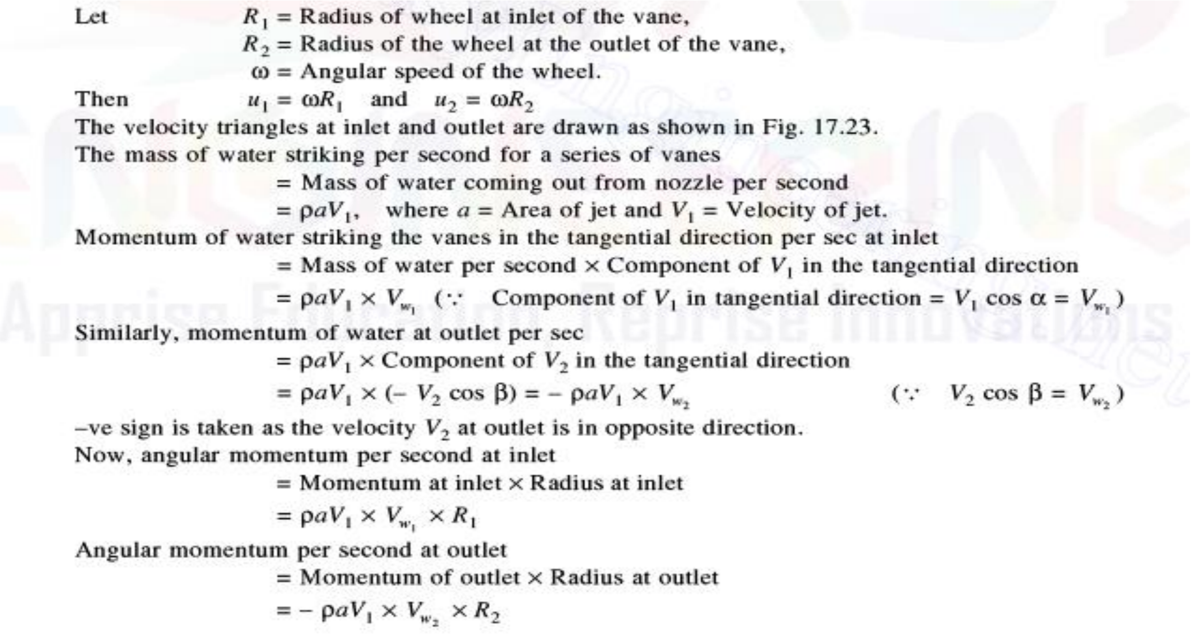
Solving above equation

d = 0.7196 m

Diameter of a pipe is d= 719.6 mm

d Find equation for force and work done for the impact of jet on a series of moving radial vanes (As applied to turbines). 04 marks




Let R_1 = Radius of wheel at inlet of the vane,
R_2 = Radius of the wheel at the outlet of the vane,
ω = Angular speed of the wheel.

Then $u_1 = \omega R_1$ and $u_2 = \omega R_2$

The velocity triangles at inlet and outlet are drawn as shown in Fig. 17.23.

The mass of water striking per second for a series of vanes

= Mass of water coming out from nozzle per second

= $\rho a V_1$, where a = Area of jet and V_1 = Velocity of jet.

Momentum of water striking the vanes in the tangential direction per sec at inlet

= Mass of water per second × Component of V_1 in the tangential direction

= $\rho a V_1 \times V_{w_1}$ ($\because$ Component of V_1 in tangential direction = $V_1 \cos\alpha = V_{w_1}$)

Similarly, momentum of water at outlet per sec

= $\rho a V_1$ × Component of V_2 in the tangential direction

= $\rho a V_1 \times (-V_2 \cos\beta) = -\rho a V_1 \times V_{w_2}$ ($\because$ $V_2 \cos\beta = V_{w_2}$)

–ve sign is taken as the velocity V_2 at outlet is in opposite direction.

Now, angular momentum per second at inlet

= Momentum at inlet × Radius at inlet

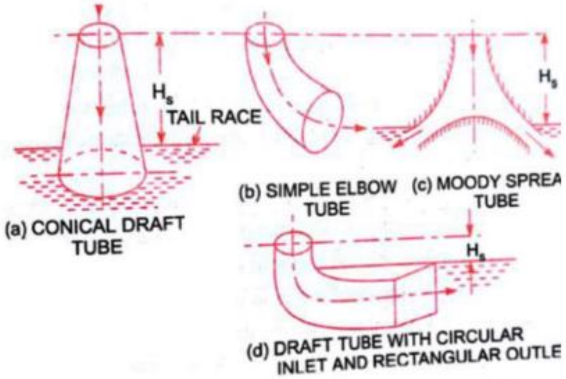
= $\rho a V_1 \times V_{w_1} \times R_1$

Angular momentum per second at outlet

= Momentum of outlet × Radius at outlet

= $-\rho a V_1 \times V_{w_2} \times R_2$

		Torque exerted by the water on the wheel, $T =$ Rate of change of angular momentum $=$ [Initial angular momentum per second – Final angular momentum per second] $= \rho a V_1 \times V_{w_1} \times R_1 - (-\rho a V_1 \times V_{w_2} \times R_2) = \rho a V_1 [V_{w_1} \times R_1 + V_{w_2} R_2]$ Work done per second on the wheel $=$ Torque $\times$ Angular velocity $= T \times \omega$ $= \rho a V_1 [V_{w_1} \times R_1 + V_{w_2} R_2] \times \omega = \rho a V_1 [V_{w_1} \times R_1 \times \omega + V_{w_2} R_2 \times \omega]$ $= \rho a V_1 [V_{w_1} u_1 + V_{w_2} \times u_2]$ $(\because \quad u_1 = \omega R_1 \text{ and } u_2 = \omega R_2)$ F_x is written as $F_x = \rho a V_{r_1} [V_{w_1} \pm V_{w_2}]$	
	e	A jet of water 10 cm diameter strikes on a flat plate with a velocity of 20 m/s. The plate is moving with a velocity of 10m/s in the direction of jet and away from the jet. Find the efficiency of the jet. Given data : Dia. Of pipe , d = 10 cm Velocity of the Jet ,V= 20 m/s Velocity of the plate ,u = 10 m/s $Densityofthewater , þ = 1000\ kg\ /m3$ $F = þ * a * (V - u)2$ $F = 1000 * \frac{\pi}{4} d^2 * (V - u)2$---(1 mark) $KEatinlet = (\frac{1}{2} * \ þ * a * V3)$--(1 mark) Efficiency of the jet. $\varkappa = \frac{\text{work done per}}{Energyatinlet} = \frac{F*u}{KE} = \frac{þ*a*(V-u)2*u}{(\frac{1}{2}* þ*a*V3)} = \frac{2*(20-10)^2*10}{20^3}$ =0.25-------(2 mark) Efficiency of the jet. $\varkappa$ =25 %	01 m to find out F, 01 m for KE, 02 m for efficiency.
4	a	Attempt any THREE of the following: Describe with neat sketches different types of draft tubes with use.	2 marks for fig. and 2 marks for explanation



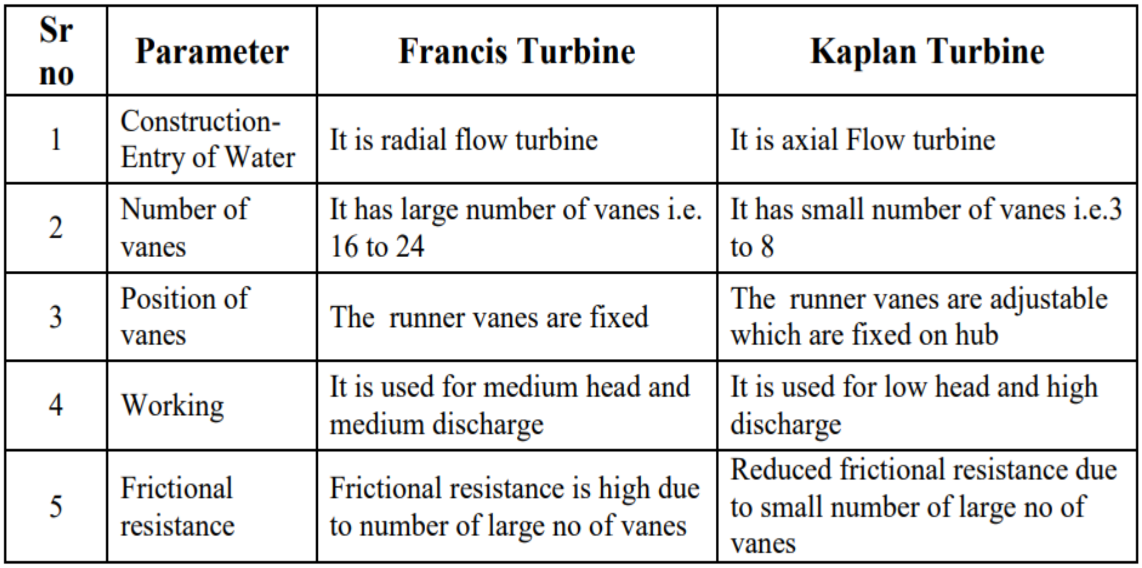
Conical Draft Tube: (Straight Divergent Tube) The shape of this tube is that of frustum of a cone. It is usually employed for low specific speed, vertical shaft Francis turbine. The cone angle is restricted to 8 to avoid the losses due to separation. The tube must discharge sufficiently low under tail water level. The maximum efficiency of this type of draft tube is 90%.

Simple Elbow Type The draft tube is bent to keep its definite length. Simple elbow type draft tube will serve such a purpose. Its efficiency is, however, low (about 60%). This type of draft tube turns the water from the vertical to the horizontal direction with a minimum depth of excavation. The horizontal portion of the draft tube is generally inclined upwards to lead the water gradually to the level of the tail race and to prevent entry of air from the exit end.

Elbow Draft Tube It is circular in cross section at inlet in its vertical leg which turns into rectangular cross section in horizontal portion of tube at outlet. The horizontal portion of tube is gradually inclined upwards so that water leaves tube almost at tail race level. Efficiency of this tube is in range of 60 to 80%

Moody's Spreading Draft Tube: This is a modification of conical tube and a solid conical cone is provided in the center of the tube with a flare at the bottom end. Such an arrangement allows a large exit area without excessive length. The solid core at the center enables full flow and reduces the eddy losses. The efficiency of the tube is about 85%.

b



Sr no	Parameter	Francis Turbine	Kaplan Turbine
1	Construction-Entry of Water	It is radial flow turbine	It is axial Flow turbine
2	Number of vanes	It has large number of vanes i.e. 16 to 24	It has small number of vanes i.e.3 to 8
3	Position of vanes	The runner vanes are fixed	The runner vanes are adjustable which are fixed on hub
4	Working	It is used for medium head and medium discharge	It is used for low head and high discharge
5	Frictional resistance	Frictional resistance is high due to number of large no of vanes	Reduced frictional resistance due to small number of large no of vanes

04 marks 01 m for each point

c

A pelton wheel 2.5m diameter operates under the following conditions.

i) Net available head (H) = 400m

ii) Speed (N) = 250rpm

iii) Coefficient of velocity of the jet (Cv) = 0.98

iv) Friction coefficient for vanes (K) = 0.95

v) Blade Angle (6) = 15°

vi) Diameter of jet (d) = 25 cm

vii) Mechanical efficiency (nm) = 0.90

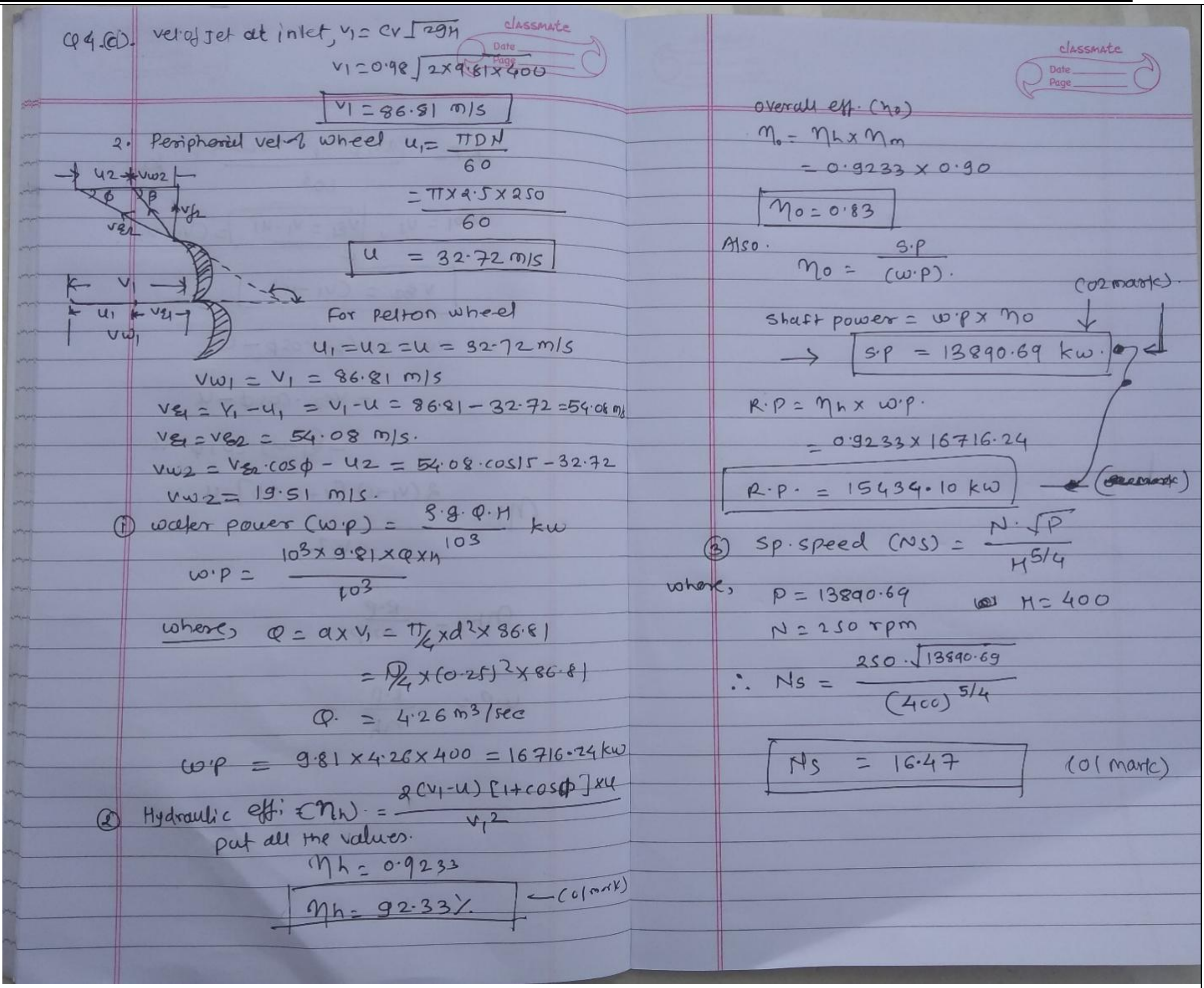
Determine:

1) The power developed

2) Hydraulic efficiency

3) Specific speed

01 m to find out u, 01 m for water power, 01 m for hydraulic efficiency, 01 m for sp. speed

Q4.(c). vel of Jet at inlet, $V_1 = C_v\sqrt{2gH}$

$V_1 = 0.98\sqrt{2 \times 9.81 \times 400}$

$\boxed{V_1 = 86.81 \text{ m/s}}$

2. Peripheral vel of wheel $u_1 = \frac{\pi D N}{60}$

$= \frac{\pi \times 2.5 \times 250}{60}$

$\boxed{u = 32.72 \text{ m/s}}$

For pelton wheel

$u_1 = u_2 = u = 32.72 \text{ m/s}$

$V_{w1} = V_1 = 86.81 \text{ m/s}$

$V_{r1} = V_1 - u_1 = V_1 - u = 86.81 - 32.72 = 54.08 \text{ m/s}$

$V_{r1} = V_{r2} = 54.08 \text{ m/s}.$

$V_{w2} = V_{r2} \cdot \cos\phi - u_2 = 54.08 \cdot \cos 15 - 32.72$

$V_{w2} = 19.51 \text{ m/s}.$

① water power (w.p) $= \frac{\rho \cdot g \cdot Q \cdot H}{103}$ kw

$w.p = \frac{10^3 \times 9.81 \times Q \times H}{10^3}$

where, $Q = a \times V_1 = \frac{\pi}{4} \times d^2 \times 86.81$

$= \frac{\pi}{4} \times (0.25)^2 \times 86.81$

$Q = 4.26 \text{ m}^3/\text{sec}$

$w.p = 9.81 \times 4.26 \times 400 = 16716.24 \text{ kw}$

② Hydraulic effi (η_h) $= \frac{2(V_1 - u)[1 + \cos\phi] \times u}{V_1^2}$

put all the values.

$\eta_h = 0.9233$

$\boxed{\eta_h = 92.33\%}$ ← (01 mark)

overall eff. (η_o)

$\eta_o = \eta_h \times \eta_m$

$= 0.9233 \times 0.90$

$\boxed{\eta_o = 0.83}$

Also. $\eta_o = \frac{S.P}{(w.P)}.$

(02 mark).

shaft power = w.p × η_o

→ $\boxed{S.P = 13890.69 \text{ kw}.}$

$R.P = \eta_h \times w.p.$

$= 0.9233 \times 16716.24$

$\boxed{R.P. = 15434.10 \text{ kw}}$

③ Sp. speed (Ns) $= \frac{N \cdot \sqrt{P}}{H^{5/4}}$

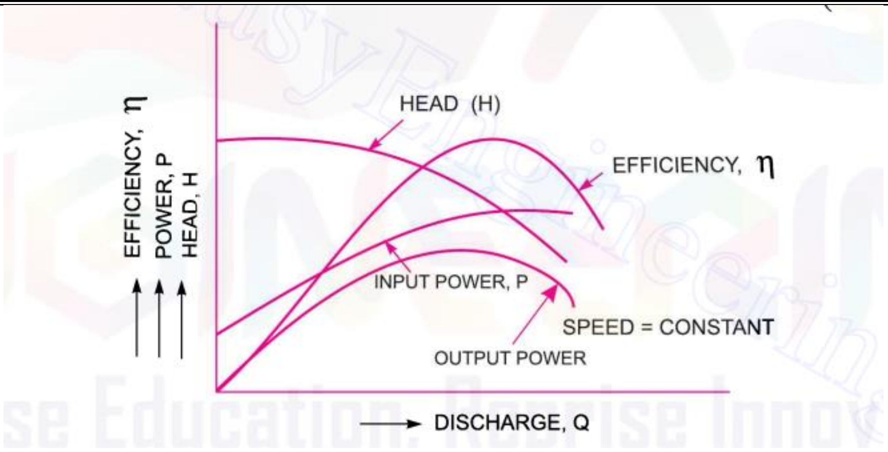
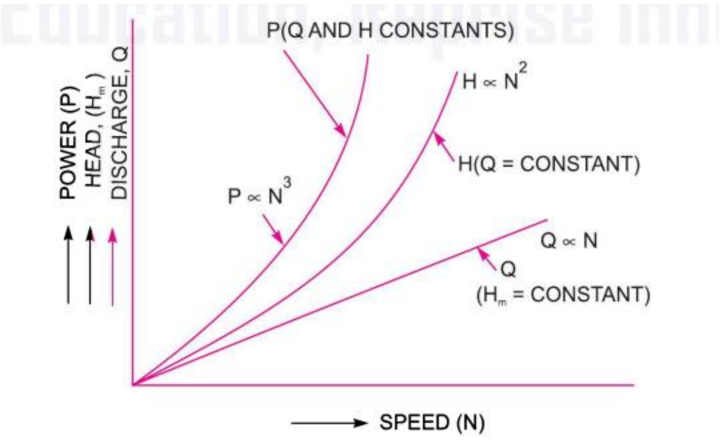
where, $P = 13890.69$ $H = 400$

$N = 250$ rpm

$\therefore Ns = \frac{250 \cdot \sqrt{13890.69}}{(400)^{5/4}}$

$\boxed{Ns = 16.47}$ (01 mark)

	d	Draw and explain the main characteristics curves of centrifugal pump in discharge Vs overall efficiency.	figure for 2 marks and explaination for 2 marks

		 Main characteristics curves The main characteristics curves are obtained by keeping the pump at constant speed and varying the discharge over desired range. The discharge is varied by means of deliver valve. For different values of discharge the measurements are taken or calculated for manometric head, shaft power and efficiency These curve are useful in evaluating the performance of pump at different speeds. 	
	e	Write **any four** operational difficulties commonly experienced in centrifugal pump and their remedies. 1. Pump Running Dry If you are not getting any flow after starting your centrifugal pump, there could be several different causes and remedies. Air in pump — Ensure the pipework and pump are completely filled with liquid. Suction lift is too high — Check for any obstructions in the inlet and verify that static lift is correct. Clogged parts — Check and clean the valve, impeller, and strainer.	04 m 1 m for each

2. Reversed Impeller Rotation

Impellers rotating in the wrong direction is a common problem with centrifugal pumps. If the impellers turn the wrong way, they could cause severe damage to the pump. When wiring power to the pump's motor, it's critical to verify which way the motor turns. You can "bump start" the motor to do this.

3. Pump Leakage

Another common problem with these types of centrifugal pumps is leakage. When materials escape the pump and create a mess, this is a serious issue. Excessive temperature, corrosion, or pressure can loosen the joints and seals, allowing fluid and debris to escape.

But there may be a simple fix. Stopping your leaky pump could be as easy as tightening the fasteners surrounding the joints. In other cases, however, may need to replace a gasket or mechanical seal.

4. Slow Pump Re-Priming

There is probably something wrong with pump if it takes too long to re-prime. The most common cause of a slow re-priming pump is excessive clearance, leading to inefficiency and overheating. But other possible reasons exist as well, such as a leaking gasket, a clogged recirculation port, or a worn-out volute.

5. Pump Seizure

Pump seizure can happen for several reasons, including foreign objects entering the pump, low flow operation, and off-design conditions. Inspect the pump for foreign objects and debris first and then check the impellers and power source.

6. Pump Vibration

When the pump vibrating too much or notice usual noises coming from the device, this could signify a serious issue. Often, vibrations and noises tell that failed bearings or a foreign object stuck inside the pump.

Start with the most straightforward thing first and look for debris or foreign objects. When noises and vibrations occur together, the pump could be experiencing cavitation and may need to be examined by a professional.

7. Debris in Pump

Debris in your pump can create havoc with many of its parts and systems. If pump isn't pumping or is less efficient , check for a clogged suction pipe or debris in the impeller.

8. Pump Driver Overloaded

In centrifugal pumps, overloading occurs when the driving motor draws excess current, which results in greater than normal power consumption. Pumps should start with a minimum load with discharge valves open. If the power drawn by the pump increases too much, it may ultimately lead to tripping or overloading of the motor. Some of the most common causes of pump driver overload include:

- The speed of the pump is too fast
- An oversized impeller was installed
- Worn or damaged bearings
- Processing liquids of higher viscosity
- Bent shaft
- Misalignment between driver and pump
- Mechanical seal putting too much pressure on the seat
- Stationary parts coming into contact with rotating parts
- Pump operating too far out of optimum range

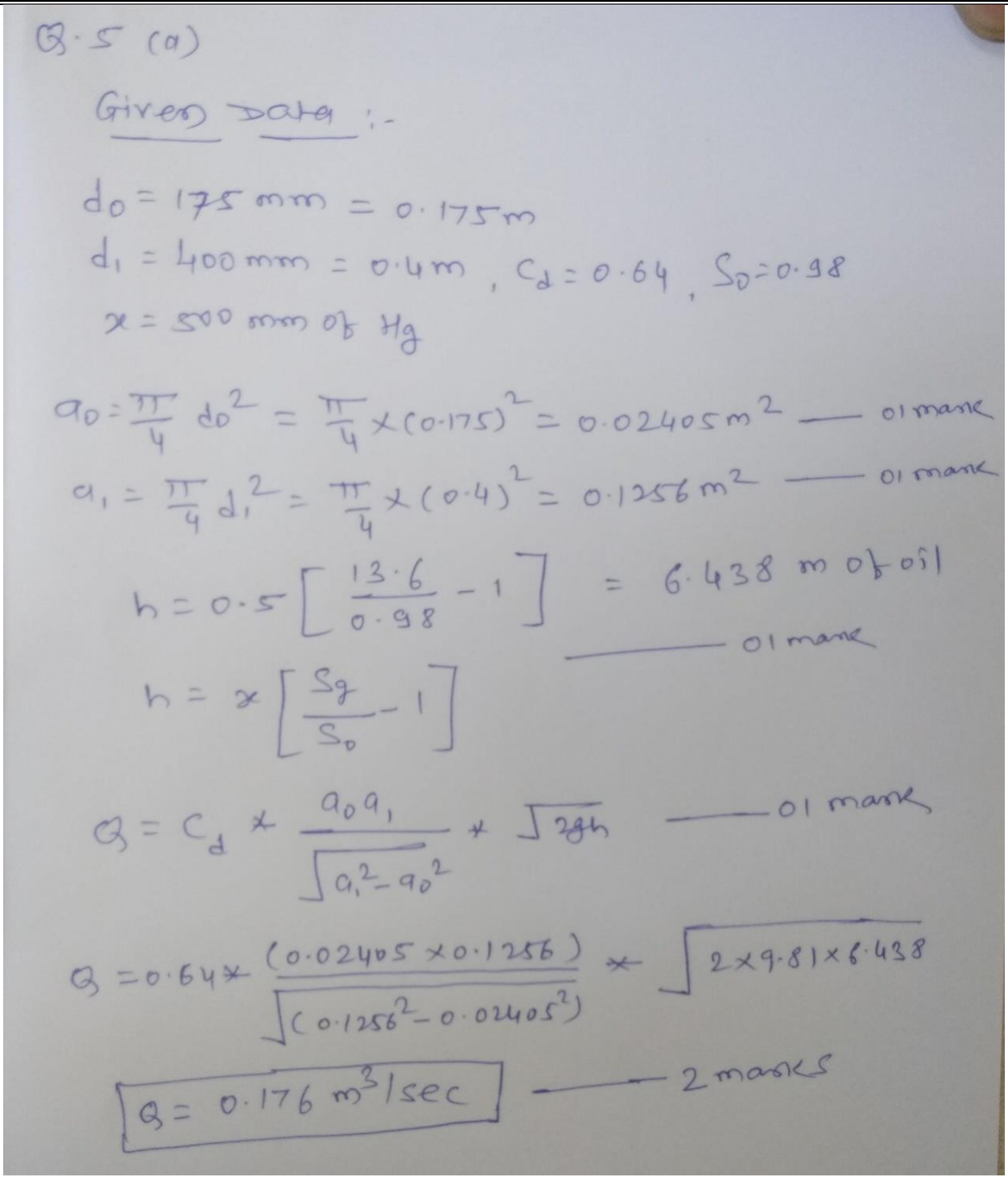
9. Poor Efficiency

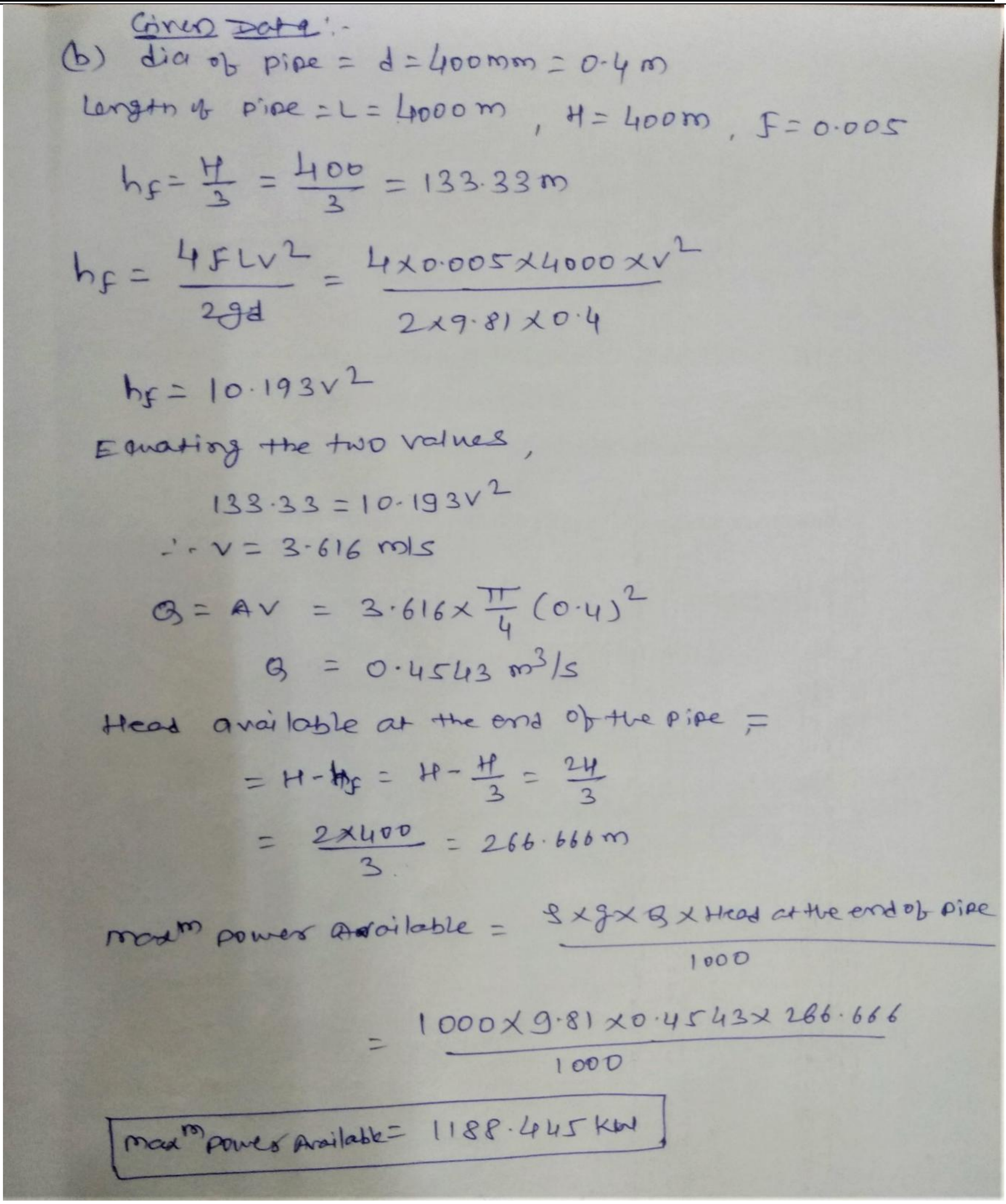
If the pump isn't operating efficiently anymore, meaning it's taking too long for it to pump out fluid, some of the most common causes of this problem include the following.

- A leaky gasket
- Incorrect impeller rotation
- Damaged or worn impeller, worn-out ring, or damaged wear plate
- An open bypass valve
- Blockage in pump inlet, discharge line, or impeller

10. Bearing Overheating

Centrifugal pumps should not feel hot to the touch. When they do, this is a sign of trouble . There may be a blockage in the suction strainer, the recirculation port, the valve, or the open-ended discharge line.

5	a	 Q.5 (a) Given Data :- $d_0 = 175$ mm $= 0.175$ m $d_1 = 400$ mm $= 0.4$ m , $C_d = 0.64$, $S_0 = 0.98$ $x = 500$ mm of Hg $a_0 = \frac{\pi}{4} d_0^2 = \frac{\pi}{4} \times (0.175)^2 = 0.02405 m^2$ — 01 mark $a_1 = \frac{\pi}{4} d_1^2 = \frac{\pi}{4} \times (0.4)^2 = 0.1256 m^2$ — 01 mark $h = 0.5\left[\frac{13.6}{0.98} - 1\right] = 6.438$ m of oil — 01 mark $h = x\left[\frac{S_g}{S_0} - 1\right]$ $Q = C_d \times \frac{a_0 a_1}{\sqrt{a_1^2 - a_0^2}} \times \sqrt{2gh}$ — 01 mark $Q = 0.64 \times \frac{(0.02405 \times 0.1256)}{\sqrt{(0.1256^2 - 0.02405^2)}} \times \sqrt{2 \times 9.81 \times 6.438}$ $\boxed{Q = 0.176 \ m^3/sec}$ — 2 marks	To find ao---1m a1---1m h---1m Q formula 1m Q----2 m

<table>
<tr>
<td></td>
<td>b</td>
<td>


Given Data:-

(b) dia of pipe = d = 400mm = 0.4 m

Length of pipe = L = 4000 m , H = 400m , f = 0.005

$$h_f = \frac{H}{3} = \frac{400}{3} = 133.33\,m$$

$$h_f = \frac{4fLv^2}{2gd} = \frac{4\times0.005\times4000\times v^2}{2\times9.81\times0.4}$$

$$h_f = 10.193v^2$$

Equating the two values,

$$133.33 = 10.193v^2$$

$$\therefore v = 3.616\,m/s$$

$$Q = AV = 3.616\times\frac{\pi}{4}(0.4)^2$$

$$Q = 0.4543\,m^3/s$$

Head available at the end of the pipe =

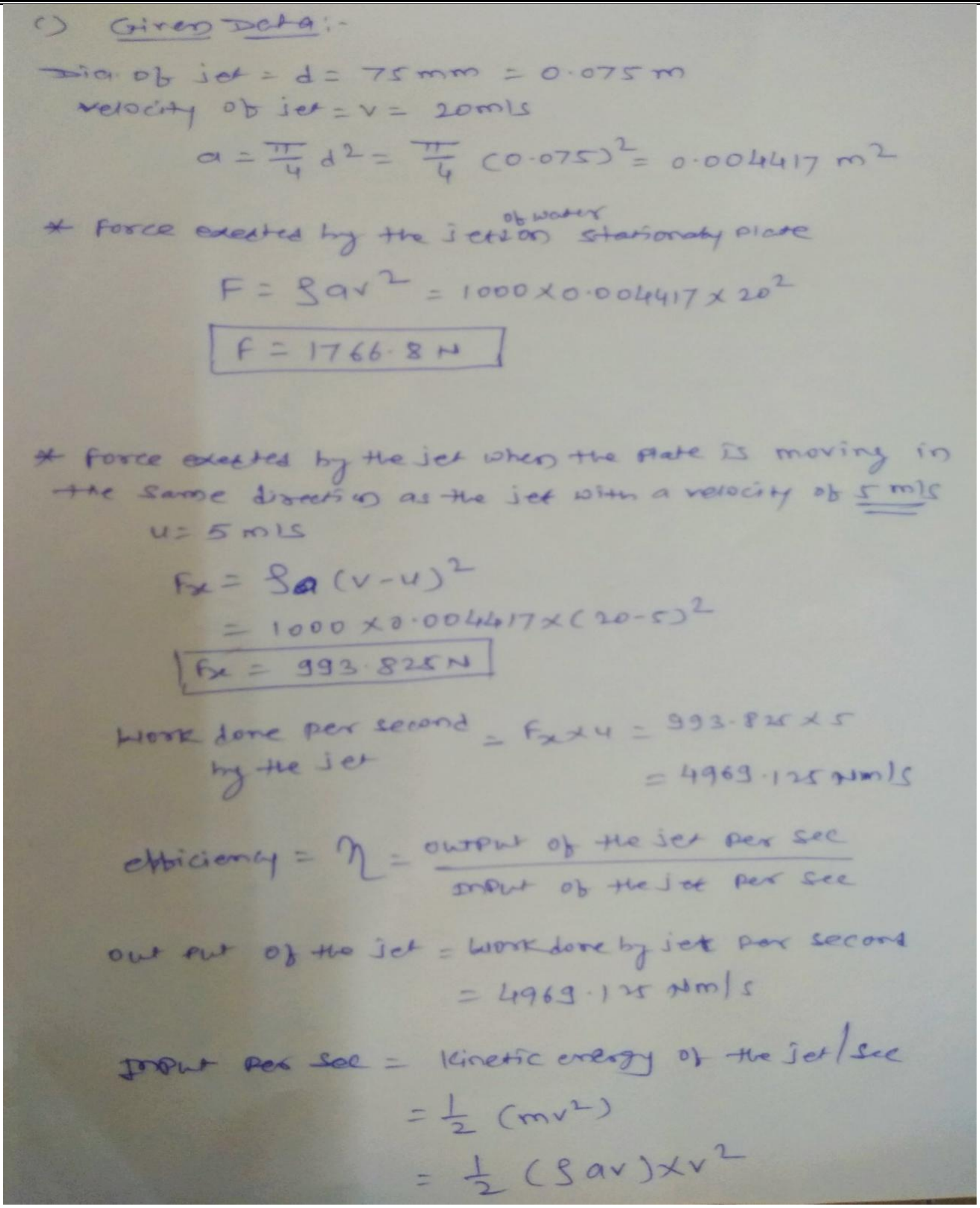
$$= H - h_f = H - \frac{H}{3} = \frac{2H}{3}$$

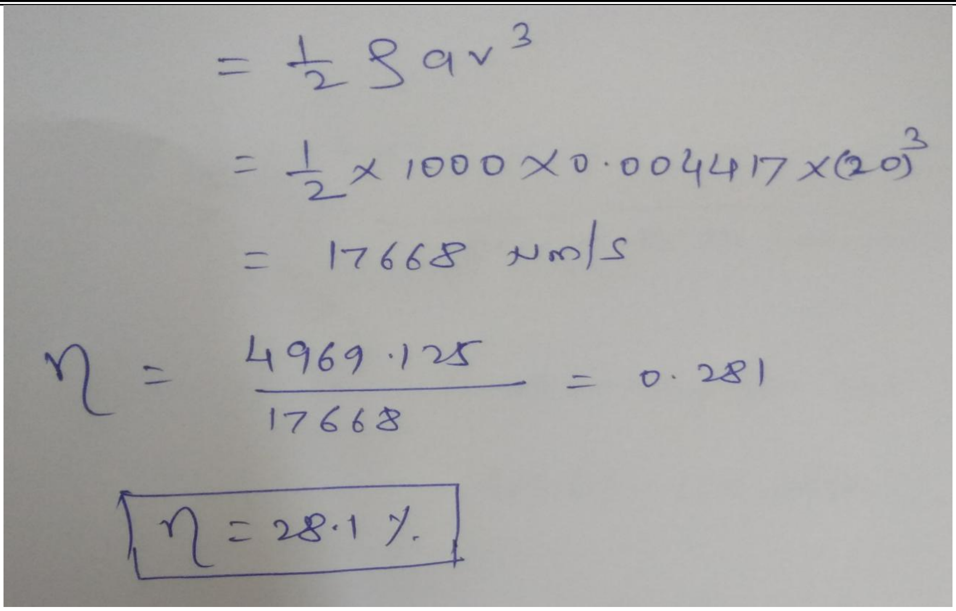
$$= \frac{2\times400}{3} = 266.666\,m$$

$$\text{max}^m \text{ power Available} = \frac{\rho\times g\times Q\times \text{Head at the end of pipe}}{1000}$$

$$= \frac{1000\times9.81\times0.4543\times266.666}{1000}$$

$$\boxed{\text{Max}^m \text{ power Available} = 1188.445\,kW}$$
</td>
<td>Hf---2m
Q---1m
Head ---1m
Power –2m</td>
</tr>
</table>

	c	 (c) Given Data:- Dia. of jet = d = 75 mm = 0.075 m velocity of jet = v = 20 m/s $a = \frac{\pi}{4} d^2 = \frac{\pi}{4} (0.075)^2 = 0.004417 \ m^2$ * Force exected by the jet of water on stationary plate $F = \rho a v^2 = 1000 \times 0.004417 \times 20^2$ $\boxed{F = 1766.8 \ N}$ * Force exected by the jet when the plate is moving in the same direction as the jet with a velocity of 5 m/s u = 5 m/s $F_x = \rho a (v-u)^2$ $= 1000 \times 0.004417 \times (20-5)^2$ $\boxed{F_x = 993.825 \ N}$ Work done per second by the jet $= F_x \times u = 993.825 \times 5$ $= 4969.125 \ Nm/s$ efficiency $= \eta = \frac{\text{output of the jet per sec}}{\text{Input of the jet per sec}}$ out put of the jet = work done by jet per second $= 4969.125 \ Nm/s$ Input per sec = Kinetic energy of the jet/sec $= \frac{1}{2} (mv^2)$ $= \frac{1}{2} (\rho a v) \times v^2$	F---1m Fx--- 1m Work done--- 1m Efficiency --- 3m



$= \frac{1}{2} \rho a v^3$

$= \frac{1}{2} \times 1000 \times 0.004417 \times (20)^3$

$= 17668$ Nm/s

$\eta = \frac{4969.125}{17668} = 0.281$

$\boxed{\eta = 28.1\ \%}$

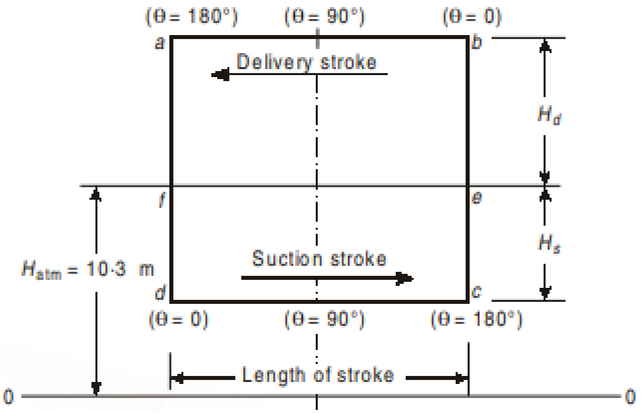
6 a

Impulse Turbine	Reaction Turbine
1. The entire available energy of the water is converted into kinetic energy.	1. Only a portion of the fluid energy is converted into kinetic energy before the fluid enters the turbine runner.
2. The work is done only by the change in the kinetic energy of the jet	2. The work is done partly by the change in the velocity head, but almost entirely by the change in pressure head.
3. Flow regulation is possible without loss.	3. It is not possible to regulate the flow without loss.
4.. Unit is installed above the tailrace.	4. Unit is entirely submerged in water below the tailrace.
5. Casing has no hydraulic function to perform, because the jet is unconfined and is at atmospheric pressure. Thus, casing serves only to prevent splashing of water.	5. Casing is absolutely necessary, because the pressure at inlet to the turbine is much higher than the pressure at outlet. Unit has to be sealed from atmospheric pressure.
6. It is not essential that the wheel should run full and air has free access to the buckets	6. Water completely fills the vane passage.
7. Pelton wheel Turbine	7.Frances Turbine, Kaplan Turbine
8.No need of draft tube	8. Draft tube required
9.High head	9. Low or medium head

b **Indicator Diagram: -** The indicator diagram for a reciprocating pump is defined as the graph between the pressure head in cylinder and the distance travelled by the piston from inner dead centre for one complete revolution of the crank.

02 m for theoretical indicator dig. 2m for

Theoretical Indicator Diagram for Single acting reciprocating pump:-



Effect of Acceleration and Friction in Suction and Delivery Pipes on Indicator Diagram.

Fig. shows the combined effect of acceleration and friction in suction and delivery pipes. The pressure head in the cylinder during suction and delivery strokes will change as given below :

(*i*) At the beginning of the suction stroke, $\theta = 0°$ and hence h_{as} is equal to $\frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r$. But $h_{fs} = 0$. Thus, the pressure head in the cylinder will be $(h_s + h_{as})$ below the atmospheric pressure head.

(*ii*) At the middle of the suction stroke, $h_{as} = 0$ but $h_{fs} = \frac{4 \times f \times l_s}{d_s \times 2g} \times \left(\frac{A}{a_s} \omega r\right)^2$. Thus, the pressure head in the cylinder will be $(h_s + h_{fs})$ below the atmospheric pressure head.

(*iii*) At the end of the suction stroke, $h_{as} = -\frac{l_s}{g} \times \frac{A}{a_s} \omega^2 r$ but $h_{fs} = 0$. Thus, the pressure head in the cylinder will be $(h_s - h_{as})$ below the atmospheric pressure head.

(*iv*) At the beginning of the delivery stroke, $h_{ad} = -\frac{l_d}{g} \times \frac{A}{a_d} \times \omega^2 r$ but $h_{fd} = 0$. Thus, the pressure head in the cylinder will be $(h_d + h_{ad})$ above the atmospheric pressure head.

explanation of effect,2m for fig of effect (*v*) In the middle of the delivery stroke, $h_{ad} = 0$ and $h_{fd} = \frac{4fl_d}{d_d \times 2g} \times \left(\frac{A}{a_d}\omega r\right)^2$. Thus the pressure head in the cylinder will be $(h_d + h_{fd})$ above the atmospheric pressure head.

(*vi*) At the end of the delivery stroke, $h_{ad} = -\frac{l_d}{g} \times \frac{A}{a_d} \times \omega^2 r$ and $h_{fd} = 0$. Thus, the pressure head in the cylinder will be $(h_d - h_{ad})$ above the atmospheric pressure head.

Thus, the indicator diagram with acceleration and friction in suction and delivery pipes will become as shown in Fig.

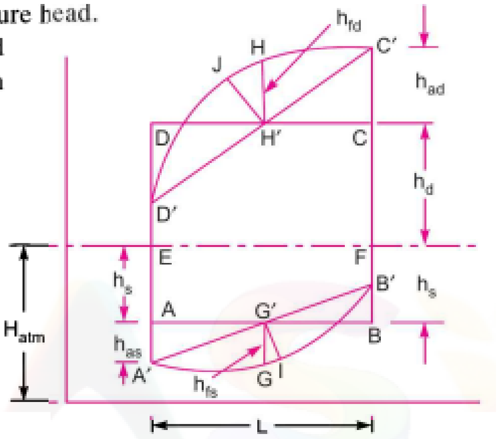
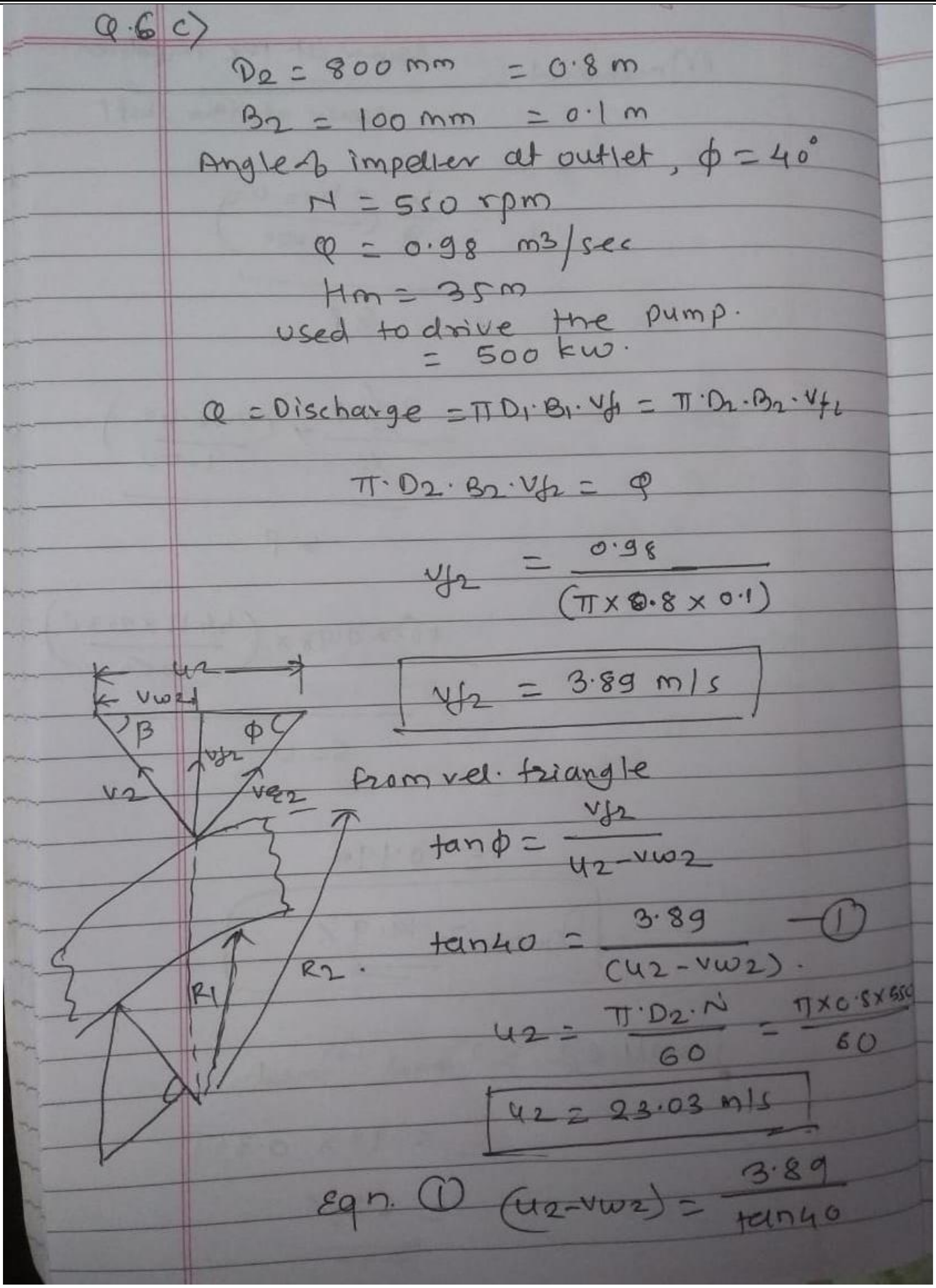


Fig. *Effect of acceleration and friction on indicator diagram.*

c



Q.6 c)

$D_2 = 800\ mm = 0.8\ m$

$B_2 = 100\ mm = 0.1\ m$

Angle of impeller at outlet, $\phi = 40^\circ$

$N = 550\ rpm$

$Q = 0.98\ m^3/sec$

$H_m = 35m$

used to drive the pump. $= 500\ kw$.

Q = Discharge $= \pi D_1 \cdot B_1 \cdot V_{f1} = \pi \cdot D_2 \cdot B_2 \cdot V_{f2}$

$\pi \cdot D_2 \cdot B_2 \cdot V_{f2} = Q$

$$V_{f2} = \frac{0.98}{(\pi \times 0.8 \times 0.1)}$$

$$\boxed{V_{f2} = 3.89\ m/s}$$

from vel. triangle

$$\tan\phi = \frac{V_{f2}}{u_2 - V_{w2}}$$

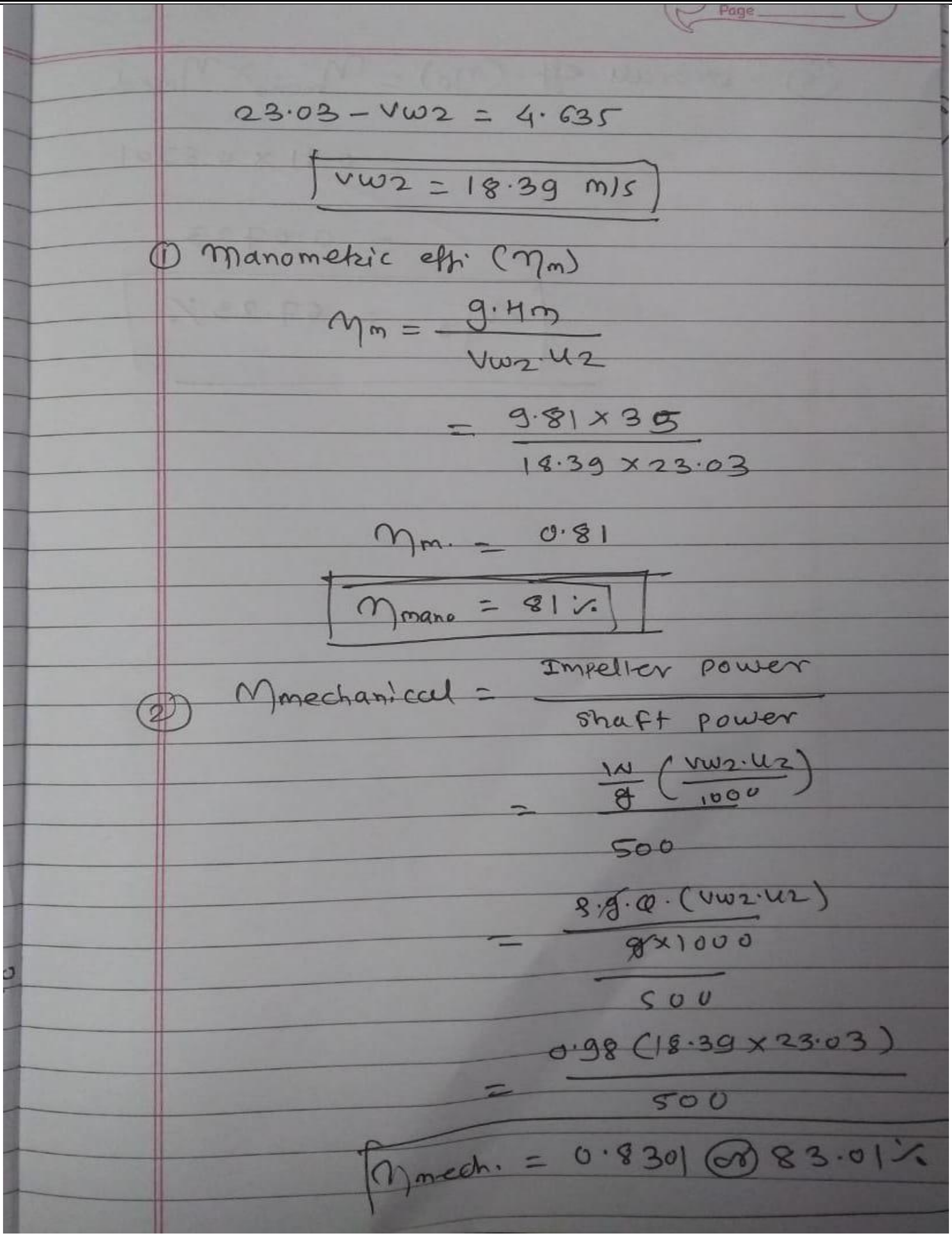
$$\tan 40 = \frac{3.89}{(u_2 - V_{w2})} \quad \text{—①}$$

$$u_2 = \frac{\pi \cdot D_2 \cdot N}{60} = \frac{\pi \times 0.8 \times 550}{60}$$

$$\boxed{u_2 = 23.03\ m/s}$$

Eqn. ① $(u_2 - V_{w2}) = \dfrac{3.89}{\tan 40}$

01 m to find out Vf2, 01m u2, 01 m Vw2, 01 m manometric efficiency, 1 m mechnaical efficiency, 1 m overall efficiency



$23.03 - V_{w2} = 4.635$

$$\boxed{V_{w2} = 18.39 \text{ m/s}}$$

① Manometric effi (η_m)

$$\eta_m = \frac{g \cdot H_m}{V_{w2} \cdot U_2}$$

$$= \frac{9.81 \times 35}{18.39 \times 23.03}$$

$$\eta_m = 0.81$$

$$\boxed{\eta_{mano} = 81\%}$$

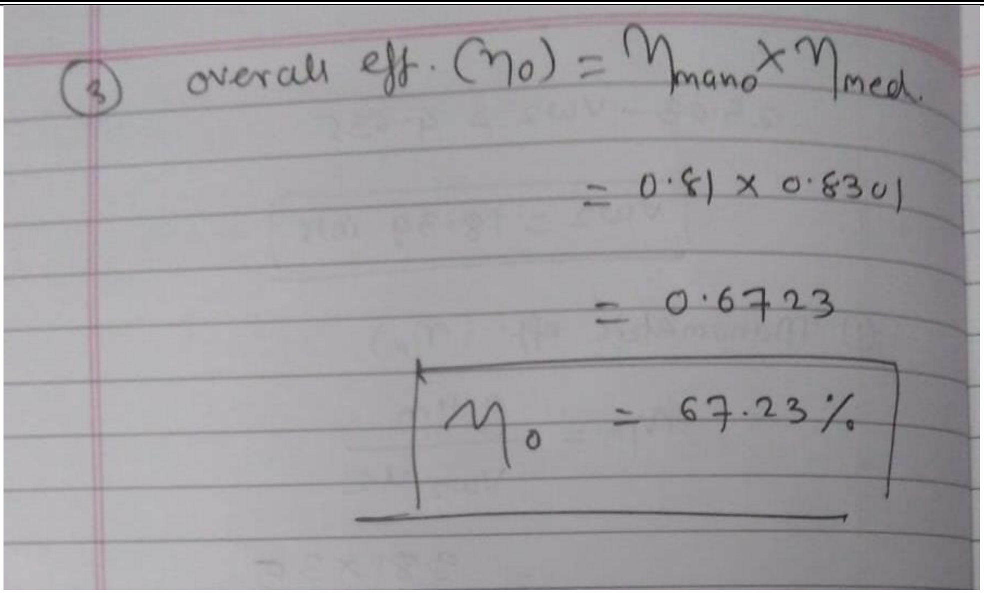
② $\eta_{mechanical} = \dfrac{\text{Impeller power}}{\text{shaft power}}$

$$= \frac{\frac{W}{\cancel{g}}\left(\frac{V_{w2} \cdot U_2}{1000}\right)}{500}$$

$$= \frac{\frac{\rho \cdot \cancel{g} \cdot Q \cdot (V_{w2} \cdot U_2)}{\cancel{g} \times 1000}}{500}$$

$$= \frac{0.98\,(18.39 \times 23.03)}{500}$$

$$\boxed{\eta_{mech.} = 0.8301 \text{ or } 83.01\%}$$



③ overall eff. $(\eta_0) = \eta_{mano} \times \eta_{med.}$

$= 0.81 \times 0.8301$

$= 0.6723$

$$\boxed{\eta_0 = 67.23\%}$$