

Fluid Mechanics and Machinery (313309)

Model Question Paper — Set C · Model Answers

Q.1

a) Define vapour pressure and kinematic viscosity.

2 M

(i) **Vapour pressure:** It is the pressure exerted by the vapour of a liquid on the liquid surface when the vapour and liquid are in equilibrium (rate of evaporation = rate of condensation) in a closed space at a given temperature. It increases with temperature; if the pressure in a flowing liquid falls below its vapour pressure, the liquid boils and cavitation occurs.

(ii) **Kinematic viscosity (v):** It is the ratio of dynamic viscosity to mass density, $v = \mu/\rho$. Unit: m^2/s ; 1 stoke = $1 \text{ cm}^2/\text{s} = 10^{-4} \text{ m}^2/\text{s}$.

(1 M each)

b) Specific weight, specific gravity and specific volume of glycerine.

2 M

Given: Density of glycerine, $\rho = 1260 \text{ kg/m}^3$; $g = 9.81 \text{ m/s}^2$; $\rho_{\text{water}} = 1000 \text{ kg/m}^3$

$$\text{Specific weight, } w = \rho g = 1260 \times 9.81 = \boxed{12\,360.6 \text{ N/m}^3}$$

$$\text{Specific gravity, } S = \frac{\rho}{\rho_{\text{water}}} = \frac{1260}{1000} = \boxed{1.26}$$

$$\text{Specific volume, } v = \frac{1}{\rho} = \frac{1}{1260} = \boxed{7.937 \times 10^{-4} \text{ m}^3/\text{kg}}$$

(Each value $\frac{2}{3}$ M)

c) HGL vs TEL.

2 M

Hydraulic gradient line (HGL)	Total energy line (TEL)
Joins points of piezometric head ($p/\rho g + Z$).	Joins points of total head ($p/\rho g + V^2/2g + Z$).
Lies below TEL by $V^2/2g$.	Lies above HGL by $V^2/2g$.
May rise in the direction of flow (e.g. at enlargement).	Always falls in the direction of flow (unless a pump adds energy).

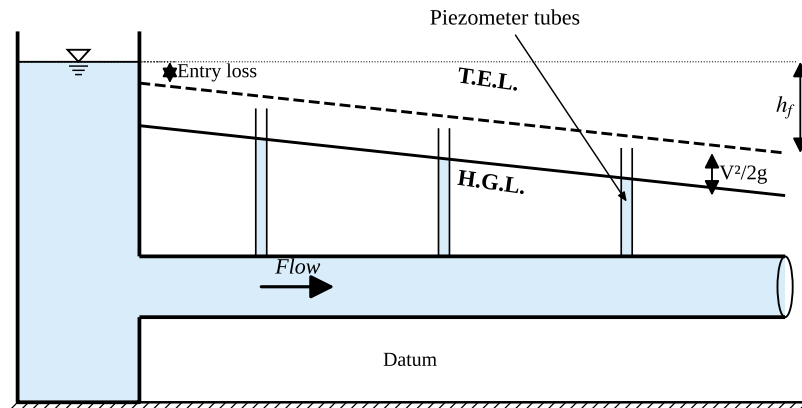


Fig.: Hydraulic gradient line (HGL) and total energy line (TEL)

(Any two points 1 M, sketch 1 M)

d) Use of draft tube; conical draft tube.

2 M

- It permits the turbine to be installed above the tail race level without loss of head (creates a negative / suction head at runner outlet), so the turbine can be inspected easily.
- Being a diverging tube, it converts a large part of the kinetic energy of water leaving the runner into useful pressure energy, so the net head and hence the **output and efficiency** of the turbine increase.

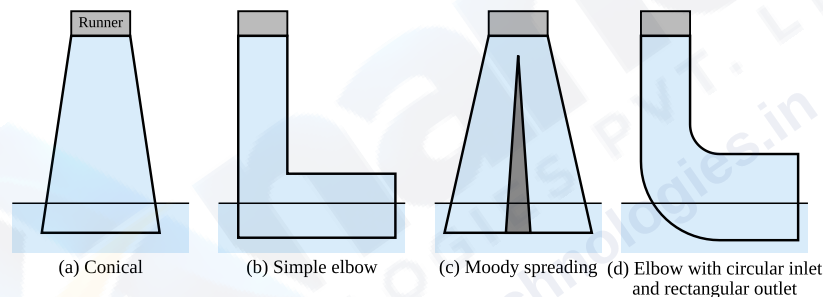


Fig.: Draft tubes — (a) is the conical draft tube

(Reasons 1 M, sketch 1 M)

e) Hydraulic coefficients.

2 M

- **Coefficient of contraction** C_c = area of jet at vena contracta / area of orifice (≈ 0.61 to 0.69).
- **Coefficient of velocity** C_v = actual velocity of jet at vena contracta / theoretical velocity $\sqrt{2gH}$ (≈ 0.95 to 0.99).
- **Coefficient of discharge** C_d = actual discharge / theoretical discharge (≈ 0.61 to 0.65).

$$\text{Relation: } C_d = C_c \times C_v$$

(Definitions 1.5 M, relation $\frac{1}{2}$ M)

f) Laws of fluid friction for laminar flow.

2 M

1. Frictional resistance is proportional to the velocity of flow.
2. It is independent of the pressure.
3. It is proportional to the area of surface in contact.

4. It is independent of the nature (roughness) of the surface in contact.
5. It is greatly affected by temperature (viscosity).

(Any four, $\frac{1}{2}$ M each)

g) Uses of air vessel.

2 M

1. It gives a continuous and nearly uniform discharge in the delivery pipe.
2. It reduces the friction head, so a large amount of work against friction is saved (about 84.8 % in a single acting pump).
3. It reduces the acceleration head, so the pump can run at a higher speed without separation (cavitation).
4. Longer suction lift and longer delivery pipes become possible.

(Any two, 1 M each)

Q.2

a) Total pressure and centre of pressure — plate with greatest and least depths.

4 M

Given: $d = 1.8$ m; greatest depth 3.0 m, least depth 1.2 m. Since $3.0 - 1.2 = 1.8$ m = diameter, the plate is **vertical** with its top edge 1.2 m below the free surface.

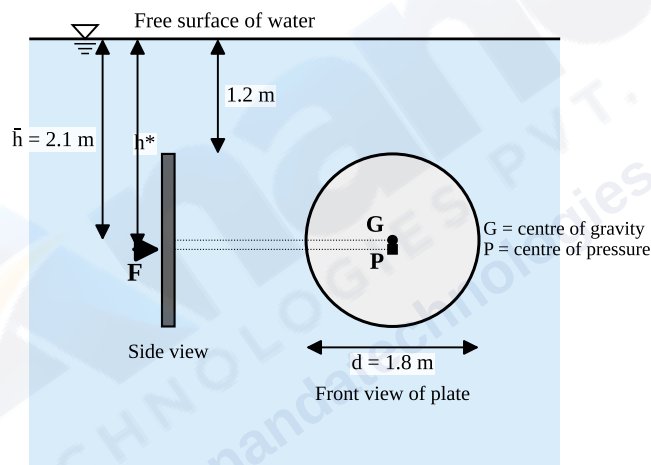


Fig.: Vertical circular plate

$$\text{Area, } A = \frac{\pi}{4} \times 1.8^2 = 2.545 \text{ m}^2$$

$$\text{Depth of centroid below free surface, } \bar{h} = 1.2 + \frac{1.8}{2} = 2.1 \text{ m}$$

$$\text{Total pressure, } F = \rho g A \bar{h} = 1000 \times 9.81 \times 2.545 \times 2.1 = 52\,423.2 \text{ N} = \boxed{52.423 \text{ kN}}$$

$$\text{M.I. about centroidal axis, } I_G = \frac{\pi d^4}{64} = \frac{\pi \times 1.8^4}{64} = 0.5153 \text{ m}^4$$

$$\text{Centre of pressure, } h^* = \frac{I_G}{A \bar{h}} + \bar{h} = \frac{0.5153}{2.545 \times 2.1} + 2.1 = 0.0964 + 2.1 =$$

2.196 m below free surface

(Total pressure 2 M, centre of pressure 2 M)

b) Differential U-tube manometer — explanation and expression.

4 M

A **differential manometer** measures the difference of pressure between two points A and B in a pipe or in two different pipes. It is a U-tube containing a heavy manometric liquid (usually mercury, density ρ_g); its two limbs are connected to A and B.

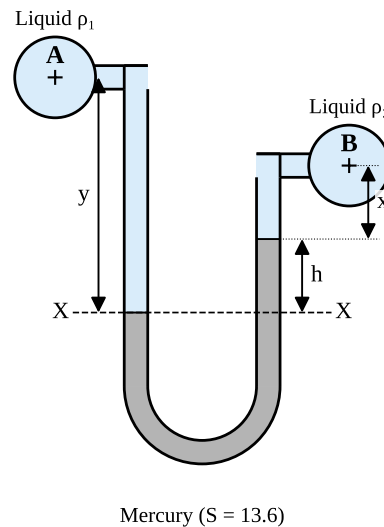


Fig.: Differential U-tube manometer (A and B at different levels)

Let h = difference of mercury levels, y = height of A above the lower mercury surface X-X, x = height of B above the mercury surface in the right limb. Pressures at X-X in both limbs are equal:

Left limb: $p_A + \rho_1 g y$; Right limb: $p_B + \rho_2 g x + \rho_g g h$

$$p_A + \rho_1 g y = p_B + \rho_2 g x + \rho_g g h$$

$$p_A - p_B = \rho_g g h + \rho_2 g x - \rho_1 g y$$

Special case — A and B at the same level and containing the same liquid ($\rho_1 = \rho_2$, $y = x + h$):

$$p_A - p_B = g h (\rho_g - \rho_1)$$

(Sketch 1 M, explanation 1 M, derivation 2 M)

c) Absolute pressure in a composite liquid column.

4 M

Given: Oil ($S = 0.8$) 60 cm at top, water 40 cm, mercury ($S = 13.6$) 25 cm at bottom; $p_{\text{atm}} = 1.01325$

bar

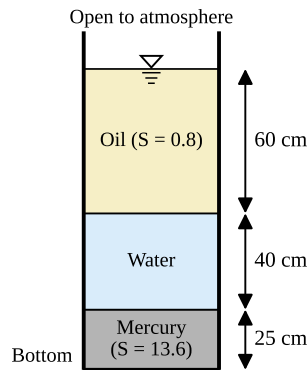


Fig.: Composite liquid column

Atmospheric pressure at top, $p_{\text{atm}} = 1.01325 \text{ bar} = 101\,325 \text{ N/m}^2$

Below oil ($S = 0.8$, height 0.6 m): $p = \text{previous} + S \rho_w g h = \text{previous} + 0.8 \times 1000 \times 9.81 \times 0.6$
 $= \text{previous} + 4\,708.8 = 106\,033.8 \text{ N/m}^2 = \mathbf{1.0603 \text{ bar}}$

Below water ($S = 1.0$, height 0.4 m): $p = \text{previous} + S \rho_w g h = \text{previous} + 1.0 \times 1000 \times 9.81 \times 0.4$
 $= \text{previous} + 3\,924 = 109\,957.8 \text{ N/m}^2 = \mathbf{1.0996 \text{ bar}}$

Below mercury ($S = 13.6$, height 0.25 m): $p = \text{previous} + S \rho_w g h = \text{previous} + 13.6 \times 1000 \times 9.81 \times 0.25$
 $= \text{previous} + 33\,354 = 143\,311.8 \text{ N/m}^2 = \mathbf{1.4331 \text{ bar}}$

(i) Absolute pressure at the oil–water interface = **1.0603 bar**

(ii) Absolute pressure at the bottom of the tank = **1.4331 bar**

(Interface 2 M, bottom 2 M)

d) Bourdon tube gauge sketch and comparison with manometers.

4 M

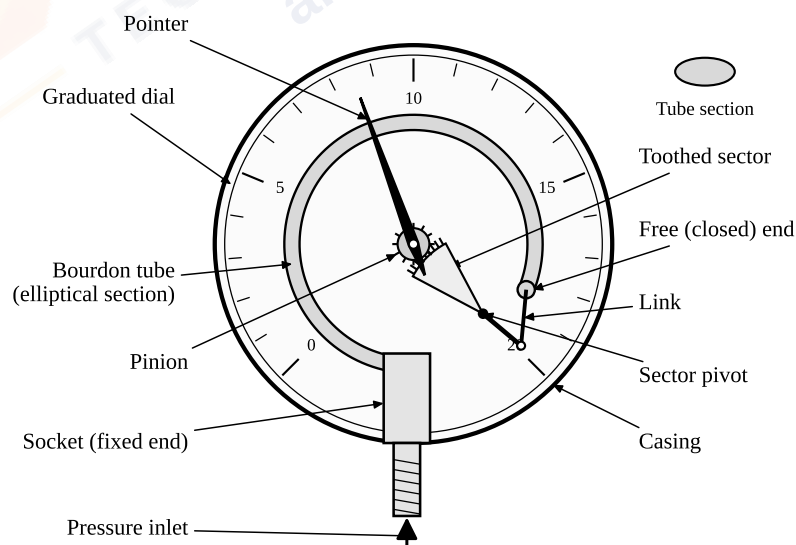


Fig.: Bourdon tube pressure gauge

Sr.	Bourdon tube pressure gauge	Manometer
1	Works on elastic deformation of a curved tube.	Works on hydrostatic balance of a liquid column ($p = \rho gh$).
2	Suitable for high pressures (up to several hundred bar).	Suitable for low pressures and small pressure differences.
3	Direct reading on a dial; quick and convenient.	Height of liquid column has to be measured and converted.
4	Compact and portable; can be fitted in any position.	Bulky; must be kept vertical; manometric liquid may spill.
5	Needs periodic calibration; accuracy may change due to fatigue.	Accurate; does not need calibration (primary instrument).

(Sketch 2 M, any four points 2 M)

Q.3

a) Throat diameter of venturimeter.

4 M

Given: $d_1 = 0.25$ m, $Q = 0.06$ m³/s, $x = 0.10$ m of mercury, $C_d = 0.97$

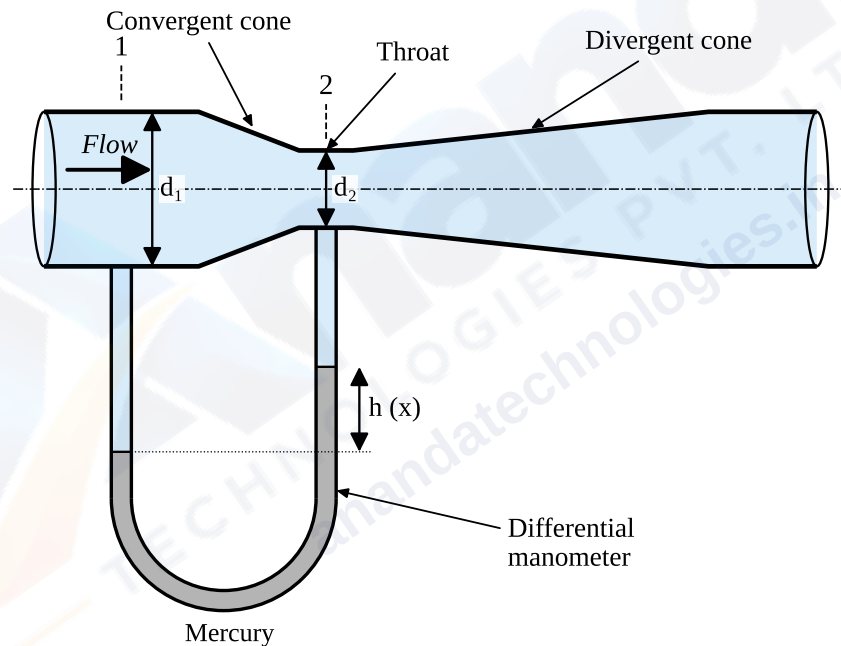


Fig.: Venturimeter with differential U-tube manometer

$$a_1 = \frac{\pi}{4} \times 0.25^2 = 0.0491 \text{ m}^2$$

$$h = x (S_m/S_w - 1) = 0.1 \times (13.6 - 1) = 1.26 \text{ m of water; } \sqrt{2gh} = \sqrt{2 \times 9.81 \times 1.26} = 4.972 \text{ m/s}$$

$$Q = \frac{C_d a_1 a_2 \sqrt{2gh}}{\sqrt{a_1^2 - a_2^2}} \Rightarrow Q^2(a_1^2 - a_2^2) = C_d^2 a_1^2 a_2^2 (2gh)$$

$$\Rightarrow a_2 = \frac{Q a_1}{\sqrt{[(C_d a_1 \sqrt{(2gh)})^2 + Q^2]}} = \frac{0.06 \times 0.0491}{\sqrt{[(0.97 \times 0.0491 \times 4.972)^2 + 0.06^2]}} = 0.0121 \text{ m}^2$$

$$d_2 = \sqrt{(4a_2/\pi)} = \sqrt{(4 \times 0.0121 / \pi)} = 0.1239 \text{ m} = \boxed{123.91 \text{ mm}}$$

(Head 1 M, equation 1 M, throat diameter 2 M)

b) Water hammer — phenomenon, effects and remedies.

4 M

Water hammer: When water flowing in a long pipe is suddenly brought to rest by closing a valve at the end of the pipe, the momentum of the moving water is destroyed. A pressure wave of very high intensity is set up which travels back and forth along the pipe with the velocity of sound in water (C). The sudden rise of pressure produces a hammering noise and vibration — this is called **water hammer**.

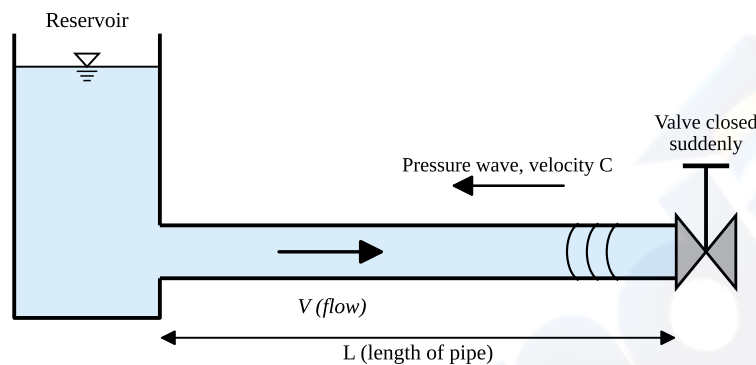


Fig.: Water hammer — sudden closure of valve at the end of a long pipe

Effects:

- Very high pressure rise which may burst the pipe or damage joints.
- Noise, vibration and hammering sound in the pipe line.
- Damage to valves, fittings and pipe supports (fatigue due to repeated pressure waves).
- Pressure may fall below vapour pressure during the negative wave, causing separation and cavitation.

Methods to reduce water hammer:

- Close the valve slowly (time of closure greater than $2L/C$).
- Provide a **surge tank** near the valve / turbine to absorb the pressure rise.
- Provide relief valves / air vessels (air chambers) and pressure release valves.
- Use pipes of sufficient strength (thickness) and keep velocity of flow low.

(Explanation with sketch 2 M, effects 1 M, remedies 1 M)

c) Pressure at section 2 by Bernoulli's equation.

4 M

Given: $Q = 0.12 \text{ m}^3/\text{s}$; $d_1 = 0.3 \text{ m}$, $p_1 = 180 \text{ kN/m}^2$, $z_1 = 5 \text{ m}$; $d_2 = 0.15 \text{ m}$, $z_2 = 8 \text{ m}$

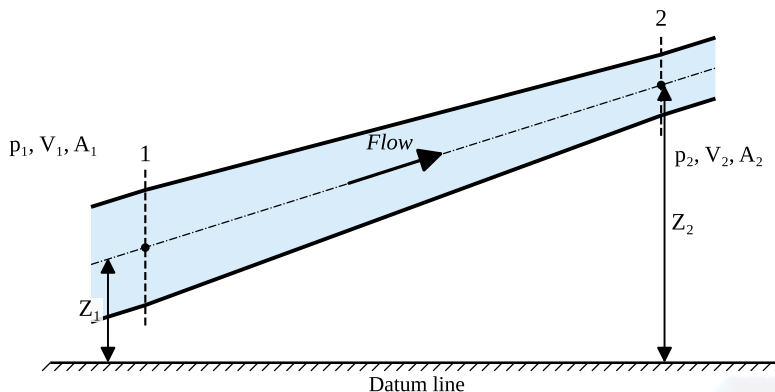


Fig.: Pipe between sections 1 and 2

$$a_1 = 0.0707 \text{ m}^2, a_2 = 0.0177 \text{ m}^2$$

$$V_1 = Q/a_1 = 0.12/0.0707 = 1.698 \text{ m/s}; V_2 = Q/a_2 = 0.12/0.0177 = 6.791 \text{ m/s}$$

$$p_1/\rho g = \frac{180\,000}{9810} = 18.349 \text{ m}; V_1^2/2g = 0.1469 \text{ m}; V_2^2/2g = 2.35 \text{ m}$$

$$\text{Bernoulli between 1 and 2: } p_1/\rho g + V_1^2/2g + z_1 = p_2/\rho g + V_2^2/2g + z_2$$

$$18.349 + 0.1469 + 5 = p_2/\rho g + 2.35 + 8$$

$$p_2/\rho g = 13.145 \text{ m of water}$$

$$p_2 = 13.145 \times 9810 = 128\,954.8 \text{ N/m}^2 = \boxed{128.95 \text{ kN/m}^2}$$

(Velocities 1 M, Bernoulli equation 1 M, answer 2 M)

d) Laminar flow vs turbulent flow.

4 M

Sr.	Laminar flow	Turbulent flow
1	Fluid particles move in smooth parallel layers along straight paths.	Fluid particles move in an irregular, zig-zag manner with mixing.
2	Reynolds number in pipe $Re < 2000$.	Reynolds number in pipe $Re > 4000$.
3	Occurs at low velocity / high viscosity.	Occurs at high velocity / low viscosity.
4	Velocity distribution in a pipe is parabolic; $V_{\max} = 2 V_{\text{mean}}$.	Velocity distribution is flatter (logarithmic); $V_{\max} \approx 1.2 V_{\text{mean}}$.
5	Head loss is proportional to V ($h_f = 32\mu VL/\rho g D^2$).	Head loss is proportional to about V^2 ($h_f = 4fLV^2/2gD$).
6	Shear stress is due to viscosity only.	Shear stress is mainly due to turbulent mixing (eddies).

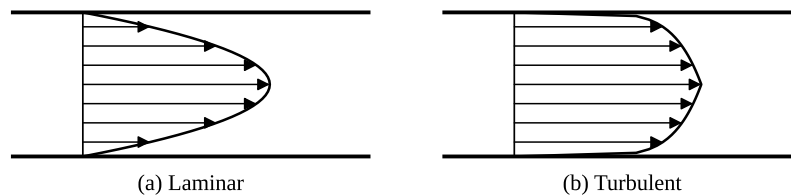


Fig.: Velocity distribution — (a) laminar flow (parabolic) (b) turbulent flow (flatter)

(Any four points, 1 M each)

Q.4

a) Jet on fixed inclined plate.

4 M

Given: $d = 0.075 \text{ m}$, $V = 22 \text{ m/s}$, $\theta = 35^\circ$

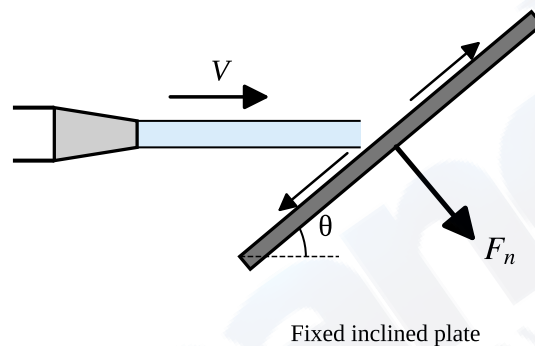


Fig.: Jet striking a fixed inclined plate

$$\text{Area of jet } a = \frac{\pi}{4} \times 0.075^2 = 0.00442 \text{ m}^2; \rho a V^2 = 1000 \times 0.00442 \times 22^2 = 2138.2 \text{ N}$$

(i) Plate normal to jet: $F = \rho a V^2 = \boxed{2138.2 \text{ N}}$

(ii) Plate inclined at $\theta = 35^\circ$ to the jet:

$$\text{Normal force } F_n = \rho a V^2 \sin \theta = 2138.2 \times \sin 35^\circ = 2138.2 \times 0.5736 = \boxed{1226.4 \text{ N}}$$

$$\text{Component along jet } F_x = F_n \sin \theta = 1226.4 \times 0.5736 = \boxed{703.46 \text{ N}}$$

$$\text{Component normal to jet } F_y = F_n \cos \theta = 1226.4 \times 0.8192 = \boxed{1004.6 \text{ N}}$$

(Normal force 2 M, components 1 M, normal plate 1 M)

b) Faults in centrifugal pump — causes and remedies.

4 M

Fault	Causes	Remedies
Pump does not deliver water	Pump not primed; air leakage in suction pipe; suction lift too high; impeller choked; wrong direction of rotation	Prime the pump properly; tighten joints / gland; reduce suction lift; clean impeller and strainer; correct the direction of rotation
Insufficient discharge	Air leakage in suction line or gland; speed too low; partially choked impeller / foot valve; worn wearing rings	Make joints air-tight; check driver speed; clean impeller, strainer and foot valve; replace wearing rings
Insufficient pressure (head)	Speed too low; air or gas in water; impeller damaged; wrong rotation	Increase speed; remove air leakage; replace impeller; correct rotation
Pump loses prime after starting	Air leakage in suction line; foot valve leaking; suction lift high; water level in sump falls	Tighten joints; repair foot valve; reduce suction lift; maintain sump level

(Any four faults, 1 M each)

c) Pelton wheel — construction and working.

4 M

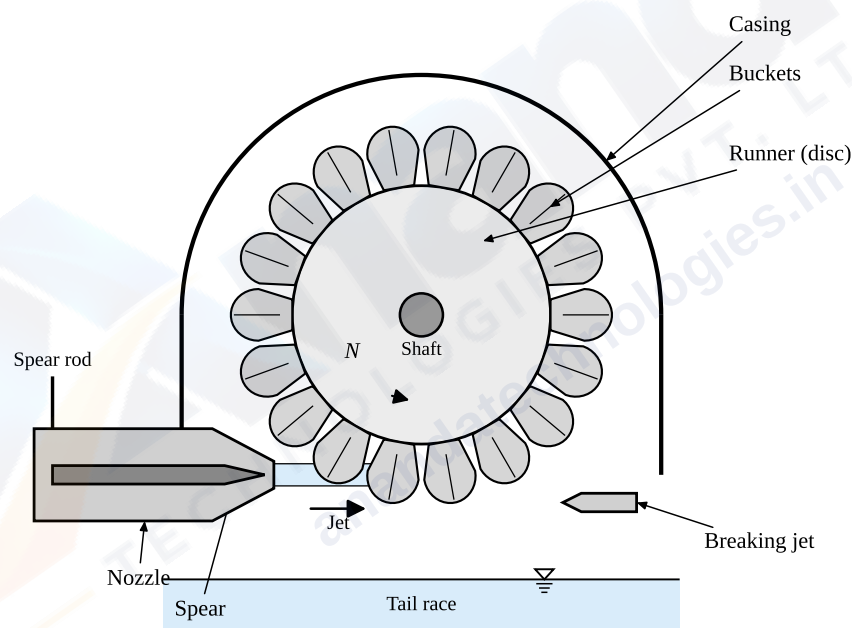


Fig.: Pelton wheel (impulse turbine) — main parts

- **Nozzle and spear:** converts the whole pressure energy of water into kinetic energy (high-velocity jet). The spear (needle) moving inside the nozzle regulates the discharge according to the load.
- **Runner with buckets:** a circular disc with double-hemispherical (cup) buckets fixed on its periphery. Each bucket has a **splitter** at the centre which divides the jet into two equal parts and deflects it through about 160° – 165° .
- **Casing:** prevents splashing and guides water to the tail race; it has no hydraulic function.

- **Breaking jet:** a small nozzle directing a jet on the back of the buckets to stop the runner quickly when the main jet is closed.

Working: Water from the penstock flows through the nozzle and comes out as a high-velocity jet at atmospheric pressure. The jet strikes the buckets tangentially; its momentum changes as it is deflected by the buckets, producing a force and torque on the runner, so the runner rotates. Water then falls freely into the tail race. It is an **impulse, tangential flow turbine** for high head and low discharge.

(Sketch 2 M, construction & working 2 M)

d) Single acting reciprocating pump; difference from double acting.

4 M

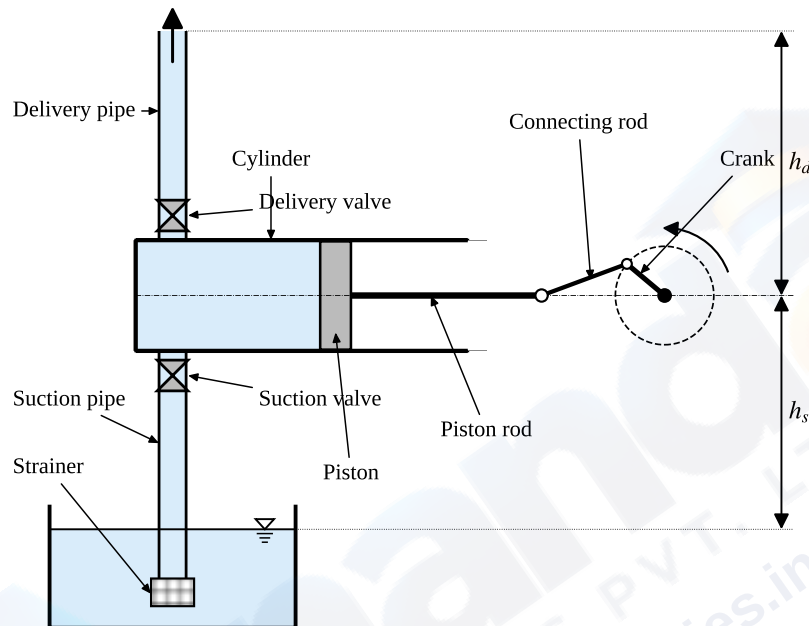


Fig.: Single acting reciprocating pump

- **Suction stroke:** the piston moves away from the valve end; vacuum is created in the cylinder, the suction valve opens and water enters from the sump.
- **Delivery stroke:** the piston moves towards the valve end; pressure rises, the suction valve closes, the delivery valve opens and water is pushed into the delivery pipe.

Difference in double acting pump: it has suction and delivery valves on both sides of the piston, so suction and delivery take place in each stroke; water is delivered twice per revolution, the discharge is about double ($Q = 2ALN/60$) and more uniform.

(Sketch 2 M, working 1 M, difference 1 M)

e) Factors for selection of a hydraulic turbine.

4 M

- **Available head:** high head (above 250 m) — Pelton; medium head (60–250 m) — Francis; low head (below 60 m) — Kaplan / propeller.
- **Quantity of water (discharge) available:** small discharge — Pelton; medium — Francis; large — Kaplan.
- **Specific speed / speed of runner:** required speed of generator decides specific speed and hence the type (Pelton 8.5–51, Francis 51–255, Kaplan 255–860).

- **Part-load efficiency:** where load varies widely, Kaplan or Pelton (flat efficiency curves) is preferred to Francis or propeller.
- **Cavitation and setting:** reaction turbines must be set at a safe height above tail race (Thoma's cavitation factor).
- **Cost, overall size and maintenance:** initial cost, size of power house and ease of maintenance.

(Any four factors, 1 M each)

Q.5

a) Maximum power, discharge and efficiency of transmission.

6 M

Given: $L = 3000$ m, $D = 0.2$ m, $p = 60$ bar $= 60 \times 10^5$ N/m², $f = 0.0075$ ($h_f = 4fLV^2/2gD$)

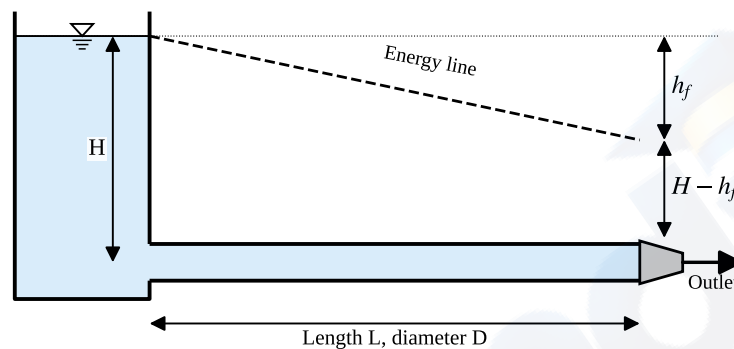


Fig.: Transmission of power through a pipe

Power at outlet $P = \rho g Q (H - h_f)$; since $h_f \propto Q^2$, $dP/dQ = 0$ gives $h_f = H/3$ for maximum power.

$$\text{Head at inlet } H = \frac{p}{\rho g} = \frac{6\,000\,000}{9810} = 611.62 \text{ m}$$

$$\text{Condition for maximum power: } h_f = H/3 = 611.62/3 = 203.87 \text{ m}$$

$$\text{Darcy: } h_f = \frac{4 f L V^2}{2 g D} \Rightarrow 203.87 = \frac{4 \times 0.0075 \times 3000 \times V^2}{2 \times 9.81 \times 0.2}$$

$$V^2 = \frac{203.87 \times 2 \times 9.81 \times 0.2}{4 \times 0.0075 \times 3000} = 8.889 \Rightarrow V = 2.981 \text{ m/s}$$

$$Q = A V = \frac{\pi}{4} \times 0.2^2 \times 2.981 = 0.0314 \times 2.981 = 0.0937 \text{ m}^3/\text{s}$$

$$\text{Maximum power transmitted } P_{\max} = \rho g Q (H - h_f) = 9810 \times 0.0937 \times (611.62 - 203.87)$$

$$P_{\max} = 374\,656.8 \text{ W} = \boxed{374.66 \text{ kW}}$$

$$\text{Efficiency of transmission at maximum power } \eta = (H - h_f)/H = \boxed{66.67 \%}$$

(Head 1 M, condition 1 M, velocity & discharge 2 M, power 1 M, efficiency 1 M)

b) Pelton wheel — power, hydraulic efficiency and specific speed.

6 M

Given: $D = 2 \text{ m}$, $H = 350 \text{ m}$, $N = 360 \text{ rpm}$, $d = 0.2 \text{ m}$, $C_v = 0.98$, $\phi = 15^\circ$, $V_{r2} = 0.95 V_{r1}$, $\eta_m = 0.90$

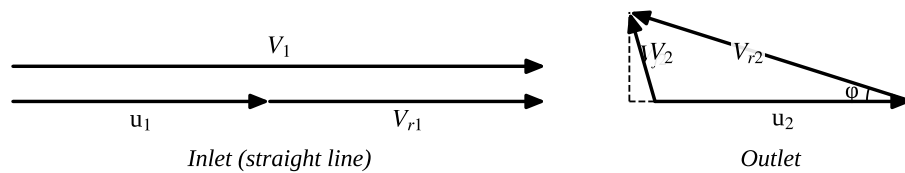


Fig.: Velocity triangles of Pelton bucket (not to scale)

$$\text{Jet velocity } V_1 = C_v \sqrt{2gH} = 0.98 \times \sqrt{2 \times 9.81 \times 350} = 81.21 \text{ m/s}$$

$$\text{Bucket speed } u = u_1 = u_2 = \frac{\pi D N}{60} = \frac{\pi \times 2 \times 360}{60} = 37.699 \text{ m/s}$$

$$\text{Inlet triangle (straight line): } V_{w1} = V_1 = 81.21 \text{ m/s}; V_{r1} = V_1 - u = 43.511 \text{ m/s}$$

$$\text{Outlet: angle } \phi = 180^\circ - 165^\circ = 15^\circ; V_{r2} = 0.95 \times V_{r1} = 41.335 \text{ m/s}$$

$$V_{w2} = V_{r2} \cos \phi - u = 41.335 \times \cos 15^\circ - 37.699 = 2.228 \text{ m/s (positive: opposite to } u, \text{ so it is added)}$$

$$\text{Discharge } Q = a V_1 = \frac{\pi}{4} \times 0.2^2 \times 81.21 = 0.0314 \times 81.21 = 2.551 \text{ m}^3/\text{s}$$

$$\text{Power developed by runner} = \rho Q (V_{w1} + V_{w2}) u = 1000 \times 2.551 \times (81.21 + (2.228)) \times$$

$$37.699 = 8\,025\,154.7 \text{ W} = \boxed{8\,025.2 \text{ kW}}$$

$$\text{Hydraulic efficiency } \eta_h = \frac{2 (V_{w1} + V_{w2}) u}{V_1^2} = \frac{2 \times (81.21 + (2.228)) \times 37.699}{81.21^2} = \boxed{95.39 \%}$$

$$\text{Water power (input)} = \rho g Q H = 9810 \times 2.551 \times 350 = 8\,759.8 \text{ kW}; \text{ Shaft power} = \eta_m \times \text{runner}$$

$$\text{power} = 0.9 \times 8\,025.2 = 7\,222.6 \text{ kW}$$

$$\text{Overall efficiency } \eta_o = \frac{\text{Shaft power}}{\text{Water power}} = \frac{7\,222.6}{8\,759.8} = \boxed{82.452 \%}$$

$$\text{Specific speed } N_s = \frac{N \sqrt{P}}{H^{5/4}} \quad (P = \text{shaft power in kW}) = \frac{360 \times \sqrt{7\,222.6}}{350^{5/4}} = \frac{30\,595}{1\,513.9} = \boxed{20.21}$$

(Velocities 1 M, power 2 M, hydraulic efficiency 1.5 M, specific speed 1.5 M)

c) Francis turbine vs Kaplan turbine.

6 M

Sr.	Francis turbine	Kaplan turbine
1	Inward radial / mixed flow reaction turbine.	Axial flow reaction turbine.
2	Used for medium head (about 60 m to 250 m) and medium discharge.	Used for low head (below about 60 m) and large discharge.
3	Runner has a large number of vanes (16 to 24) fixed to the crown and band.	Runner has a small number of blades (3 to 8) mounted on a hub (boss).
4	Runner vanes are fixed (not adjustable).	Runner blades are adjustable — angle changes with load.
5	Specific speed is medium (about 51 to 255).	Specific speed is high (about 255 to 860).
6	Efficiency falls considerably at part load.	Efficiency remains high over a wide range of load (flat efficiency curve).
7	Frictional losses are more (larger wetted area of vanes).	Frictional losses are less (fewer blades).

(Any six points, 1 M each)

Q.6

a) Centrifugal pump — inlet angle, outlet velocity, manometric efficiency, power.

6 M

Given: $D_1 = 0.2$ m, $D_2 = 0.4$ m, $N = 1450$ rpm, $\phi = 30^\circ$, $V_{f1} = V_{f2} = 2.8$ m/s, $b_2 = 0.04$ m, $H_m = 60$ m, $\eta_o = 0.72$

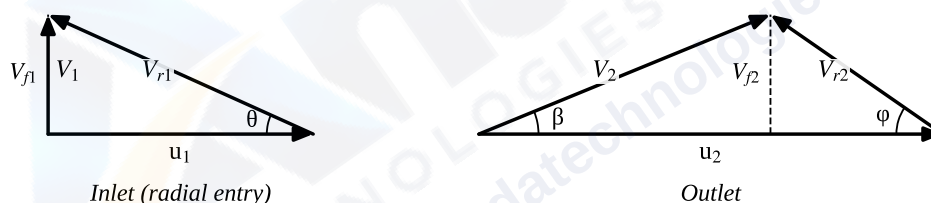


Fig.: Velocity triangles at inlet and outlet (not to scale)

$$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.2 \times 1450}{60} = 15.184 \text{ m/s}; u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.4 \times 1450}{60} = 30.369 \text{ m/s}$$

Radial entry: $V_{w1} = 0$, $V_1 = V_{f1} = 2.8$ m/s. Velocity of flow constant: $V_{f2} = 2.8$ m/s

(i) **Vane angle at inlet:** $\tan \theta = \frac{V_{f1}}{u_1} = \frac{2.8}{15.184} = 0.1844 \Rightarrow \theta = 10.45^\circ$

Outlet: $V_{w2} = u_2 - \frac{V_{f2}}{\tan \phi} = 30.369 - \frac{2.8}{\tan 30^\circ} = 30.369 - 4.85 = 25.519 \text{ m/s}$

(ii) **Work done per kg of water** $= V_{w2} u_2 = 25.519 \times 30.369 = 774.98 \text{ N}\cdot\text{m/kg}$ (= 78.999 m of head)

$$\text{Discharge } Q = \pi D_2 b_2 V_{f2} = \pi \times 0.4 \times 0.04 \times 2.8 = 0.1407 \text{ m}^3/\text{s}$$

$$\text{Work done per second} = \rho Q V_{w2} u_2 = 1000 \times 0.1407 \times 774.98 = 109.07 \text{ kW}$$

$$\text{(iii) Manometric efficiency } \eta_{\text{mano}} = \frac{g H_m}{V_{w2} u_2} = \frac{9.81 \times 60}{774.98} = \boxed{75.95 \%}$$

$$\text{Absolute velocity at outlet } V_2 = \sqrt{(V_{w2}^2 + V_{f2}^2)} = \sqrt{(25.519^2 + 2.8^2)} = 25.672 \text{ m/s at } \beta = \tan^{-1}\left(\frac{2.8}{25.519}\right) = 6.26^\circ \text{ to the wheel tangent}$$

$$\text{(iv) Power required} = \frac{\rho g Q H_m}{\eta_o} = \frac{9810 \times 0.1407 \times 60}{0.72} = 115\,057.7 \text{ W} = \boxed{115.06 \text{ kW}}$$

(Inlet angle 1 M, V_2 and direction 1.5 M, manometric efficiency 1.5 M, power 2 M)

b) Effect of air vessels on the indicator diagram.

6 M

An **air vessel** is a closed chamber containing compressed air in its upper part, fitted to the suction and delivery pipes close to the cylinder.

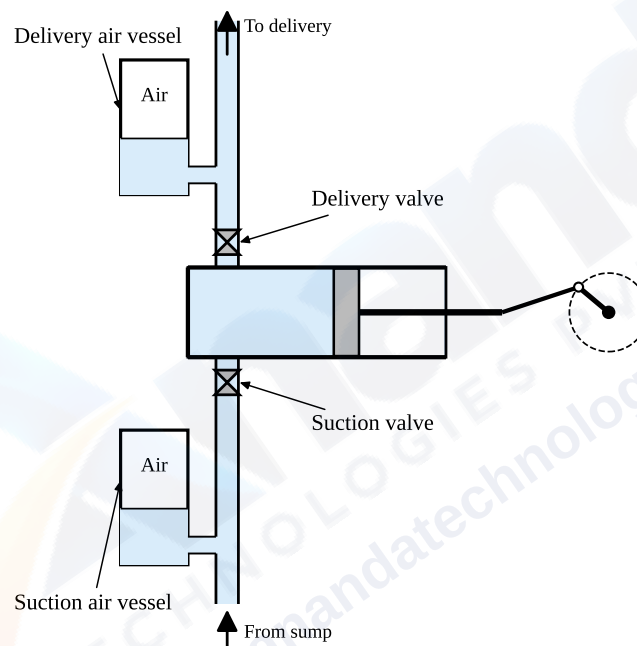


Fig.: Single acting reciprocating pump fitted with air vessels on suction and delivery pipes

c) Operating characteristic curves and Muschel curves of turbines.

Operating characteristic curves (constant speed): A turbine coupled to a generator must run at constant (synchronous) speed while the load changes. The head is also nearly constant. The discharge is changed by the gate opening and the power and efficiency are plotted against the discharge or the percentage of full load.

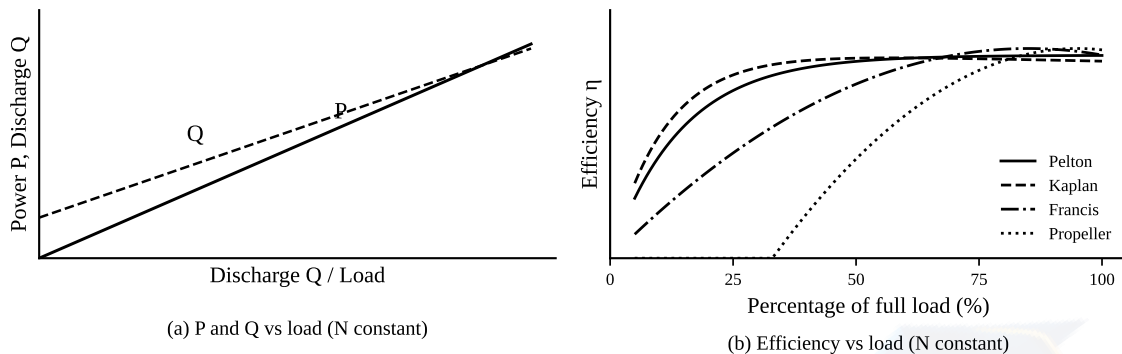


Fig.: Operating characteristic curves of turbines (constant speed)

- Power P increases almost linearly with discharge; some discharge is needed even at zero load to overcome friction (Q -axis intercept).
- Efficiency increases with load and is maximum near full load (design load).
- Kaplan and Pelton turbines maintain high efficiency over a wide range of load (flat curves); Francis and especially fixed-blade propeller turbines have low efficiency at part load.

Constant efficiency (Muschel) curves: obtained from the main characteristic curves (N_u vs Q_u and N_u vs η_o) for different gate openings. Points having the same efficiency are joined on a speed–discharge plot, giving closed loop curves (like shells).

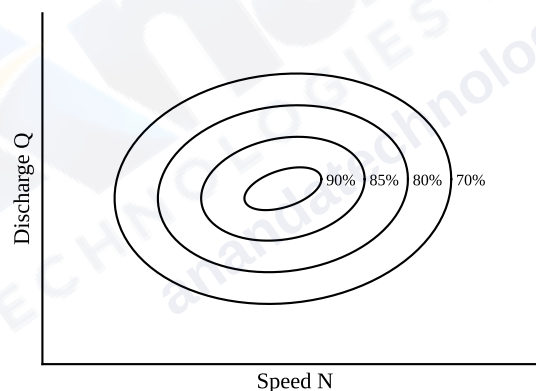


Fig.: Constant efficiency (Muschel) curves

- The innermost loop shows the zone of maximum efficiency (best operating condition).
- They show the range of speed and discharge over which the turbine works at a given efficiency and help in selecting the best operating conditions.

(Operating curves 3 M, Muschel curves 3 M)

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