Fluid Mechanics and Machinery (313309)

Model Question Paper — Set A · Model Answers

Q.1

a) Define dynamic viscosity and specific weight. 2 M

(i) Dynamic viscosity (μ): It is the property of a fluid by which it offers resistance to the movement of one layer of fluid over an adjacent layer. By Newton's law of viscosity $\tau = \mu (du/dy)$, so μ is the shear stress required to produce unit rate of shear strain. Unit: N·s/m² (Pa·s); 1 poise = 0.1 N·s/m².

(ii) Specific weight (w): It is the weight of fluid per unit volume, $w = \rho g$. Unit: N/m³. For water w = 9810 N/m³.

(1 M each)

b) Pascal's law and its applications. 2 M

Pascal's law: The intensity of pressure at any point in a fluid at rest is the same in all directions, i.e. $p_x = p_y = p_z$.

Applications: hydraulic press, hydraulic jack, hydraulic brakes of vehicles, hydraulic lift / crane.

(Statement 1 M, any two applications 1 M)

c) Continuity equation. 2 M

Statement: For steady flow, the mass of fluid flowing per second through every cross-section of a pipe (stream tube) remains constant (law of conservation of mass).

$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$; for incompressible fluid ($\rho_1 = \rho_2$): $\mathbf{Q = A_1V_1 = A_2V_2 = constant}$

(Statement 1 M, expression 1 M)

d) Laws of fluid friction for turbulent flow. 2 M

1. The frictional resistance is proportional to V^n, where n varies from 1.72 to 2.0 (approximately proportional to V^2).
2. It is proportional to the density of the fluid.
3. It is proportional to the area of the surface in contact (wetted area).
4. It depends on the nature (roughness) of the surface in contact.
5. It is slightly affected by temperature and is independent of pressure.

(Any four, ½ M each)

e) Function and types of draft tube. 2 M

Function:

- 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.

Types: (1) Conical draft tube (2) Simple elbow draft tube (3) Moody spreading draft tube (4) Elbow draft tube with circular inlet and rectangular outlet.

(Function 1 M, any two types 1 M)

f) Define NPSH. 2 M

Net Positive Suction Head (NPSH) is the net head available at the suction (inlet) of the pump above the vapour pressure head of the liquid. It is the absolute pressure head plus velocity head at the pump inlet minus the vapour pressure head:

$$NPSH = (p_1/\rho g + V_1^2/2g) - p_v/\rho g = H_{atm} - h_s - h_{fs} - H_v$$

NPSH available must be more than the NPSH required by the pump, otherwise cavitation occurs.

(Definition 1 M, expression 1 M)

g) Turbine for given conditions. 2 M

Condition	Turbine
(i) Head above 250 m, small discharge	Pelton wheel (impulse)
(ii) Head 60–250 m, medium discharge	Francis turbine
(iii) Low head, large discharge, adjustable blades	Kaplan turbine
(iv) Low head, large discharge, fixed blades	Propeller turbine

(½ M each)

Q.2

a) Total pressure and centre of pressure on vertical circular plate. 4 M

Given: d = 2.4 m, depth of centre $\bar{h}$ = 3.5 m, ρ = 1000 kg/m³

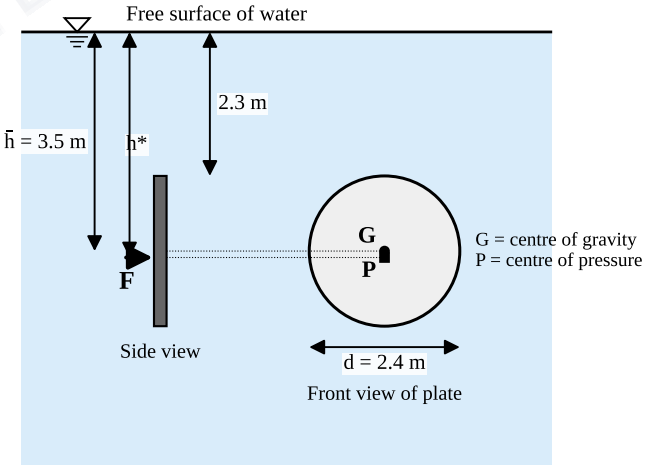


Fig.: Vertical circular plate immersed in water

Area, $A = \frac{\pi}{4} \times 2.4^2 = 4.524\ m^2$

Depth of centroid below free surface, $\bar{h} = 2.3 + \frac{2.4}{2} = 3.5\ m$

Total pressure, $F = \rho\, g\, A\, \bar{h} = 1000 \times 9.81 \times 4.524 \times 3.5 = 155\,327.9\ N =$ **155.33 kN**

M.I. about centroidal axis, $I_G = \frac{\pi\, d^4}{64} = \frac{\pi \times 2.4^4}{64} = 1.629\ m^4$

Centre of pressure, $h^* = \frac{I_G}{A\,\bar{h}} + \bar{h} = \frac{1.629}{4.524 \times 3.5} + 3.5 = 0.1029 + 3.5 =$

3.603 m below free surface

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

b) Bourdon tube pressure gauge. 4 M

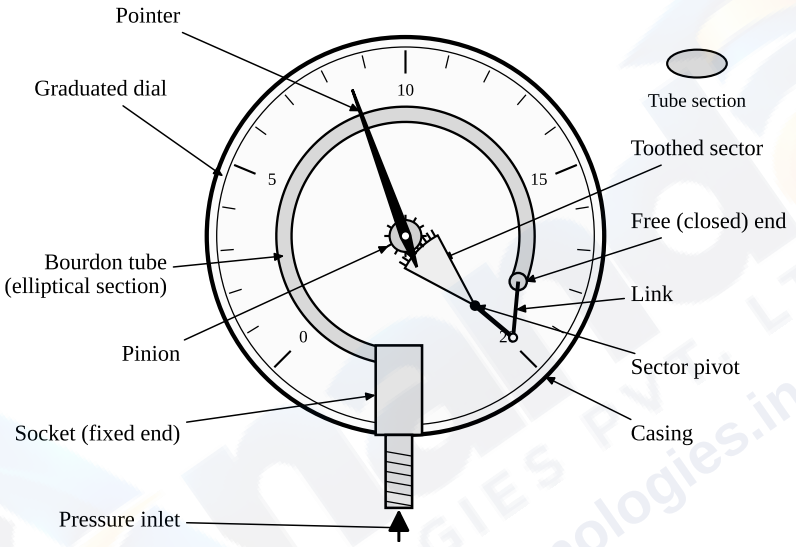


Fig.: Bourdon tube pressure gauge

Principle: When an elliptical, curved tube is subjected to internal pressure, it tends to straighten; the deflection of its free end is proportional to the pressure.

Construction:

- **Bourdon tube:** a hollow tube of elliptical cross-section, bent in the form of letter C (about 270°), made of brass / phosphor bronze / steel.
- **Fixed end (socket):** one end of the tube is fixed to the socket through which the fluid under pressure enters.
- **Free end:** the other end is closed and free to move; it is connected by a **link** to a **toothed sector**.
- **Sector, pinion and pointer:** the toothed sector meshes with a **pinion** on whose spindle the **pointer** is mounted; the pointer moves over a **graduated dial**.

Working: When fluid under pressure enters the tube, the elliptical cross-section tends to become circular. Because of this the tube tries to straighten out and its free end moves outward. This small movement is magnified by the link, sector and pinion and is shown by the pointer on the dial, which is calibrated directly in pressure units (kN/m², bar, kgf/cm²).

For vacuum (negative) pressure, the tube bends more (curls inward) and the pointer moves in the opposite direction — the same gauge type is used as a vacuum gauge.

(Sketch 2 M, construction & working 2 M)

c) Types of flow. 4 M

- **Steady flow:** flow in which the fluid characteristics (velocity, pressure, density) at a point do not change with time, $\partial V/\partial t = 0$. Example: flow through a pipe at constant pump speed.
- **Unsteady flow:** characteristics at a point change with time, $\partial V/\partial t \neq 0$. Example: flow when a valve is being opened or closed; emptying of a tank.
- **Uniform flow:** velocity at a given time does not change with distance along the flow, $\partial V/\partial s = 0$. Example: flow through a long straight pipe of constant diameter.
- **Non-uniform flow:** velocity changes from point to point along the flow, $\partial V/\partial s \neq 0$. Example: flow through a tapering pipe or a bend.
- **Laminar flow:** fluid particles move in smooth layers (laminae) along well-defined paths, one layer sliding over the other; Re < 2000 in a pipe. Example: flow of oil / honey at low velocity.
- **Turbulent flow:** particles move in a zig-zag, irregular manner with mixing between layers; Re > 4000 in a pipe. Example: flow of water in a river in flood.
- **Rotational flow:** fluid particles rotate about their own mass centres while moving along streamlines. Example: forced vortex (liquid in a rotating tank).
- **Irrotational flow:** fluid particles do not rotate about their own mass centres. Example: flow outside the boundary layer, free vortex.

(Each pair 1 M)

d) Pressure in pipe by simple U-tube manometer. 4 M

Given: S_1 (oil) = 0.9 ⇒ ρ_1 = 900 kg/m³; S_2 (mercury) = 13.6 ⇒ ρ_2 = 13600 kg/m³; difference of mercury levels h_2 = 22 cm = 0.22 m. Centre of pipe is 12 cm below the mercury level in the open limb, so height of oil above the lower mercury surface h_1 = 22 − 12 = 10 cm = 0.10 m.

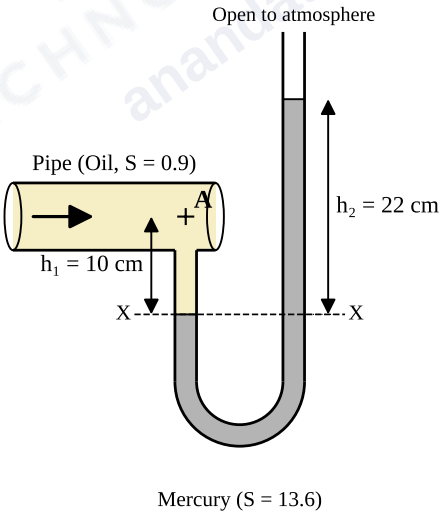


Fig.: Simple U-tube manometer

Take datum X–X at the mercury surface in the left limb. Pressures at the same level in the same continuous liquid are equal.

Pressure above X–X in left limb = $p + \rho_1 g h_1$; Pressure above X–X in right limb = $0 + \rho_2 g h_2$ (gauge)

$p + \rho_1 g h_1 = \rho_2 g h_2 \Rightarrow p = \rho_2 g h_2 - \rho_1 g h_1$

$p = 13\,600 \times 9.81 \times 0.22 - 900 \times 9.81 \times 0.1 = 29\,351.5 - 882.9$

$p =$ **$28\,468.6\ N/m^2 = 28.469\ kN/m^2$ (gauge)**

In metres of water: $\frac{p}{\rho_w g} = \frac{28\,468.6}{9810} =$ **2.902 m of water**

(Equation 2 M, answer 2 M)

Q.3

a) Discharge through venturimeter. 4 M

Given: $d_1 = 200$ mm = 0.2 m, $d_2 = 100$ mm = 0.1 m, x = 18 cm = 0.18 m of mercury, $C_d = 0.97$, $S_m = 13.6$, $S_w = 1$

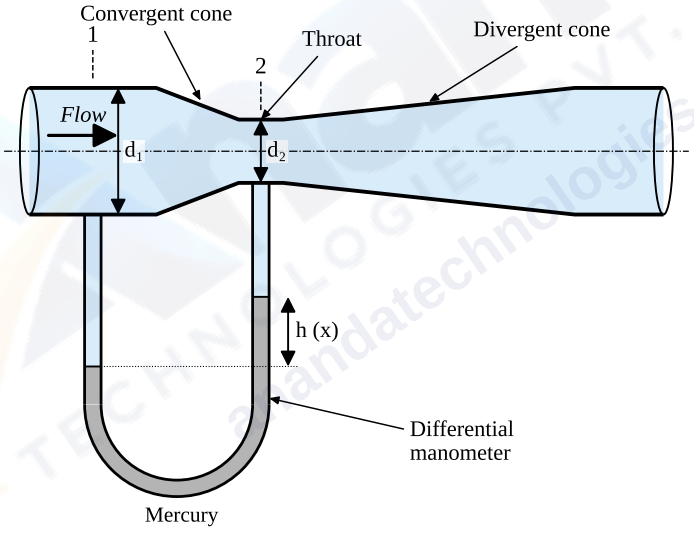


Fig.: Venturimeter with differential U-tube manometer

$a_1 = \frac{\pi}{4} \times 0.2^2 = 0.0314\ m^2$; $a_2 = \frac{\pi}{4} \times 0.1^2 = 0.00785\ m^2$

Differential head, $h = x\left(\frac{S_m}{S_l} - 1\right) = 0.18 \times \left(\frac{13.6}{1.0} - 1\right) = 2.268$ m of water

$$Q = \frac{C_d\, a_1\, a_2\, \sqrt{(2gh)}}{\sqrt{(a_1^2 - a_2^2)}} = \frac{0.97 \times 0.0314 \times 0.00785 \times \sqrt{(2 \times 9.81 \times 2.268)}}{\sqrt{(0.0314^2 - 0.00785^2)}}$$

$Q = \frac{0.0016}{0.0304} =$ **$0.0525\ m^3/s = 52.486$ litre/s**

(Area & head 1 M, formula 1 M, answer 2 M)

b) Hydraulic gradient line and total energy line. 4 M

- **Hydraulic gradient line (HGL):** the line joining the points of piezometric head ($p/\rho g + Z$) at different sections of a pipe. It is the level up to which water would rise in piezometer tubes fitted along the pipe.
- **Total energy line (TEL):** the line joining the points of total head ($p/\rho g + V^2/2g + Z$) at different sections of the pipe.

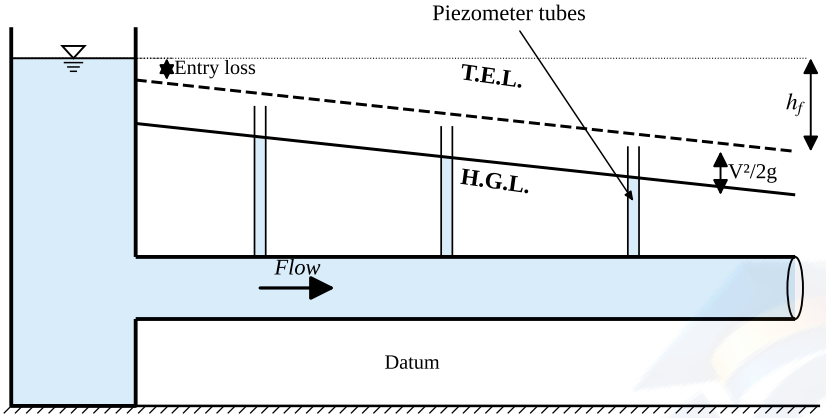


Fig.: Hydraulic gradient line (HGL) and total energy line (TEL)

- TEL always lies above HGL by the velocity head $V^2/2g$.
- For a pipe of uniform diameter the HGL and TEL are parallel.
- TEL always slopes downward in the direction of flow (because of losses) unless energy is added by a pump; HGL may rise (e.g. at a sudden enlargement).
- The vertical drop of TEL between two sections gives the head lost between them.

(Definitions 2 M, sketch 2 M)

c) Water hammer — causes 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**.

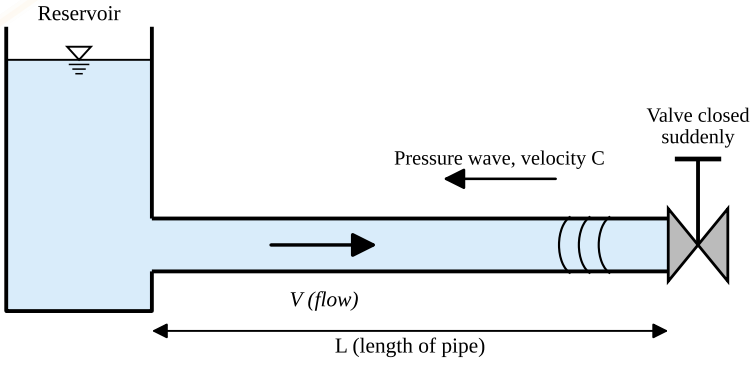


Fig.: Water hammer — sudden closure of valve at the end of a long pipe

Pressure rise for sudden closure: $p = \rho V C$.

Causes:

- Sudden closing of a valve (or sudden opening) at the end of a long pipe.
- Sudden stopping of a pump or failure of power supply to the pump.
- Sudden change in load on a water turbine causing sudden closing of the turbine gates.

Remedial measures:

- 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 1 M, causes 1 M, remedies 2 M)

d) Bernoulli's theorem with sketch and assumptions. 4 M

Statement: For steady, ideal (non-viscous), incompressible flow, the total energy per unit weight of the fluid — i.e. the sum of pressure head, kinetic (velocity) head and potential (datum) head — remains constant at every section along a streamline.

$p/\rho g + V^2/2g + Z$ = constant, i.e. $p_1/\rho g + V_1^2/2g + Z_1 = p_2/\rho g + V_2^2/2g + Z_2$

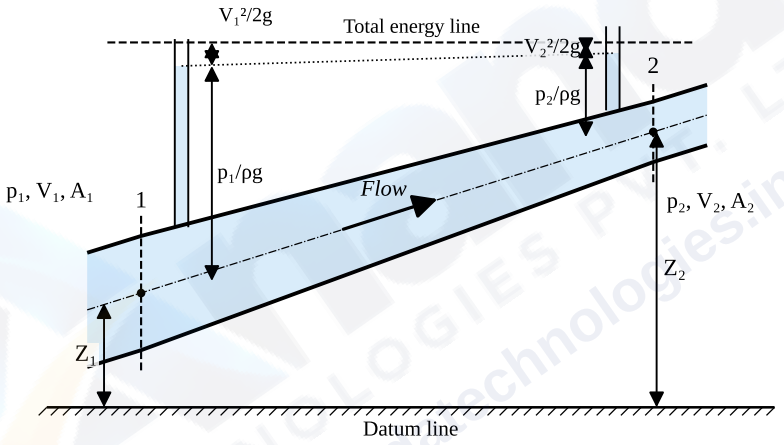


Fig.: Bernoulli's theorem — flow through a non-uniform inclined pipe

In the figure, as the pipe narrows towards section 2, the velocity head increases and the pressure head decreases, but the total head (TEL) stays the same for an ideal fluid.

Assumptions:

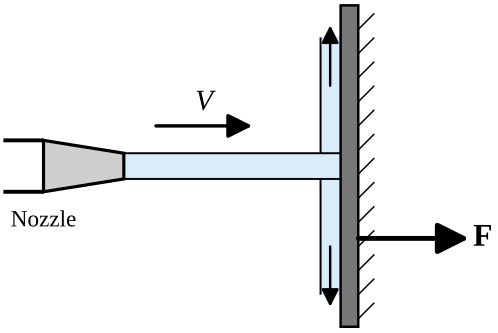
1. The fluid is ideal (viscosity is zero, no friction losses).
2. The flow is steady.
3. The flow is incompressible (ρ = constant).
4. The flow is irrotational and along a streamline.
5. No energy is added or removed (no pump / turbine) and no heat transfer.

(Statement & equation 1 M, sketch 1 M, assumptions 2 M)

Q.4

a) Jet on stationary and moving flat plate. 4 M

Given: d = 50 mm = 0.05 m, V = 25 m/s, u = 10 m/s



Fixed vertical plate

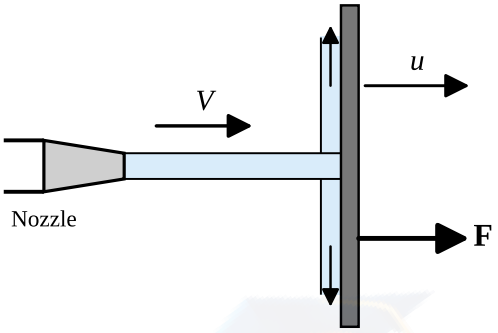


Plate moving with velocity u

Area of jet a = $\frac{\pi}{4} \times 0.05^2 = 0.00196$ m²

(i) Plate fixed: F = ρ a V² = 1000 × 0.00196 × 25² = **1 227.2 N**

(Work done on a fixed plate = 0, since the plate does not move.)

(ii) Plate moving with u = 10 m/s in direction of jet: F = ρ a (V − u)² = 1000 × 0.00196 × (25 − 10)² = **441.79 N**

Work done per second = F × u = 441.79 × 10 = **4 417.9 W = 4.418 kW**

Kinetic energy of jet per second = ½ ρ a V³ = ½ × 1000 × 0.00196 × 25³ = 15 339.8 W

Efficiency η = $\frac{\text{Work done/s}}{\text{K.E. of jet/s}} = \frac{4\,417.9}{15\,339.8}$ = **28.8 %**

(Fixed plate 1 M, moving plate 1 M, work done 1 M, efficiency 1 M)

b) Layout of hydro-electric power plant. 4 M

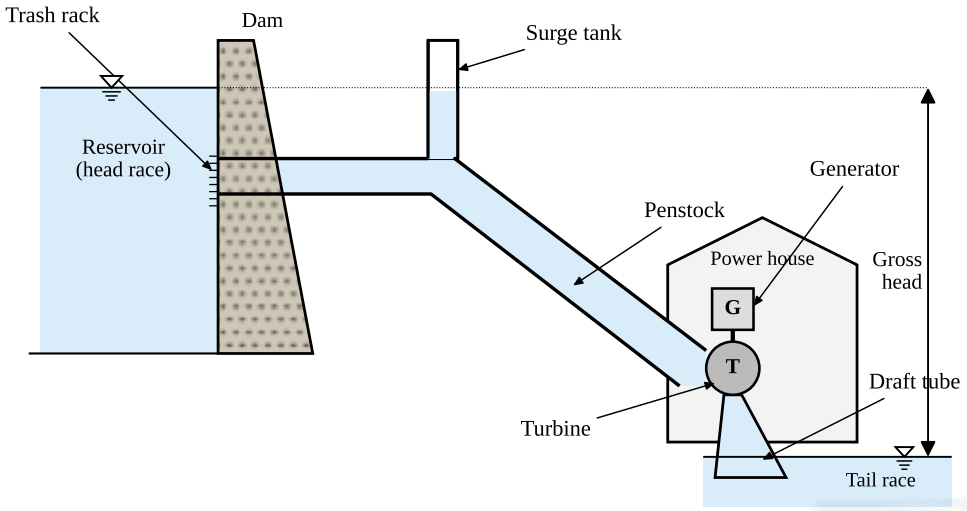


Fig.: General layout of a hydro-electric power plant

- **Reservoir (head race) and dam:** store water at a high level and create the head.
- **Trash rack / intake:** prevents debris and floating matter from entering the conduit.
- **Surge tank:** absorbs sudden pressure rise (water hammer) when the load on the turbine decreases and supplies water when the load increases.
- **Penstock:** large pipe carrying water from the surge tank to the turbine under high pressure.
- **Turbine:** converts hydraulic energy into mechanical energy; it drives the **generator**, which produces electricity.
- **Draft tube and tail race:** carry the water leaving the turbine to the river / canal (tail race) with recovery of kinetic energy.

(Layout 2 M, functions 2 M)

c) Impulse turbine vs reaction turbine. 4 M

Sr.	Impulse turbine	Reaction turbine
1	Whole available energy of water is converted into kinetic energy in the nozzle before entering the runner.	Only a part of the energy is converted into kinetic energy before entering the runner; the rest remains as pressure energy.
2	Pressure of water remains atmospheric throughout (runner works in open air).	Pressure of water changes as it flows through the runner (runner always full of water, under pressure).
3	Casing has no hydraulic function; it only prevents splashing.	Air-tight casing (scroll casing) is essential.
4	Water is admitted only to a few buckets at a time (partial admission).	Water is admitted over the full circumference of the runner (full admission).
5	Draft tube is not used; turbine is placed above tail race.	Draft tube is used; runner may be below or above tail race.
6	Suitable for high head and low discharge, e.g. Pelton wheel.	Suitable for medium/low head and large discharge, e.g. Francis, Kaplan.

(Any four points, 1 M each)

d) Single acting reciprocating pump. 4 M

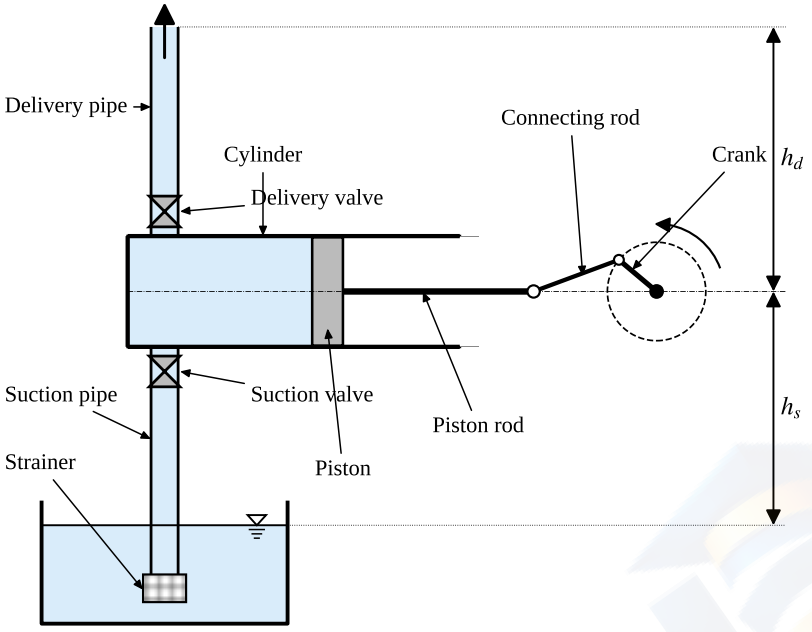


Fig.: Single acting reciprocating pump

Construction: It consists of a cylinder with a piston, piston rod, connecting rod and crank driven by a motor; a suction pipe with strainer dipping in the sump; a delivery pipe; and two non-return valves — suction valve and delivery valve — at the cylinder end.

Working:

- **Suction stroke:** the crank rotates from θ = 0° to 180°, the piston moves away from the valve end (to the right). A partial vacuum is created in the cylinder, the suction valve opens and water from the sump enters the cylinder; the delivery valve remains closed.
- **Delivery stroke:** from θ = 180° to 360° the piston moves towards the valve end. Water in the cylinder is pressurised, the suction valve closes and the delivery valve opens; water is forced into the delivery pipe.
- Water is delivered only during one stroke of every revolution, so discharge is pulsating. Theoretical discharge $Q_{th} = A\,L\,N / 60$.

(Sketch 2 M, working 2 M)

e) Pressure head and units for 2.5 bar. 4 M

Given: Gauge pressure p = 2.5 bar = 250 000 N/m²; $S_{oil} = 0.85$; $S_{Hg} = 13.6$; atmospheric pressure = 101.325 kN/m²

$$\text{Pressure head } h = \frac{p}{\rho g} = \frac{p}{S \times 1000 \times 9.81}$$

(i) Water: $h = \dfrac{250\,000}{1000 \times 9.81} =$ **25.484 m of water**

(ii) Oil: $h = \dfrac{250\,000}{0.85 \times 1000 \times 9.81} =$ **29.981 m of oil**

(iii) Mercury: $h = \frac{250\,000}{13.6 \times 1000 \times 9.81}$ = **1.874 m of mercury**

(iv) In kgf/cm²: 1 kgf/cm² = 9.81×10^4 N/m², so $p = \frac{250\,000}{98100}$ = **2.548 kgf/cm²**

(v) Absolute pressure = gauge + atmospheric = 250 + 101.325 = **351.32 kN/m²**

(Each head 1 M (any four parts))

Q.5

a) Maximum power transmitted through a pipe. 6 M

Given: L = 4 km = 4000 m, D = 300 mm = 0.3 m, H = 450 m, f = 0.0065 ($h_f = 4fLV^2/2gD$)

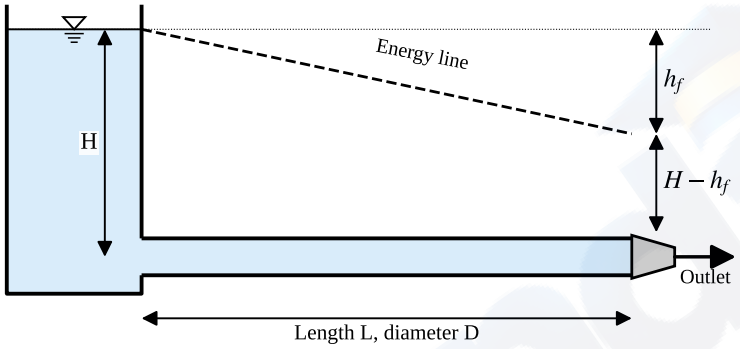


Fig.: Transmission of power through a pipe

Condition for maximum power: Power at outlet $P = \rho g Q(H - h_f)$ with $h_f = 4fLV^2/2gD \propto Q^2$. Putting $dP/dV = 0$ gives $H - 3h_f = 0$, i.e. **$h_f = H/3$**.

Condition for maximum power: $h_f = H/3 = 450/3 = 150$ m

Darcy: $h_f = \frac{4 f L V^2}{2 g D} \Rightarrow 150 = \frac{4 \times 0.0065 \times 4000 \times V^2}{2 \times 9.81 \times 0.3}$

$V^2 = \frac{150 \times 2 \times 9.81 \times 0.3}{4 \times 0.0065 \times 4000} = 8.489 \Rightarrow V = 2.914$ m/s

$Q = A V = \frac{\pi}{4} \times 0.3^2 \times 2.914 = 0.0707 \times 2.914 = 0.206$ m³/s

Maximum power transmitted $P_{max} = \rho g Q (H - h_f) = 9810 \times 0.206 \times (450 - 150)$

$P_{max} = 606\,124.4$ W = **606.12 kW**

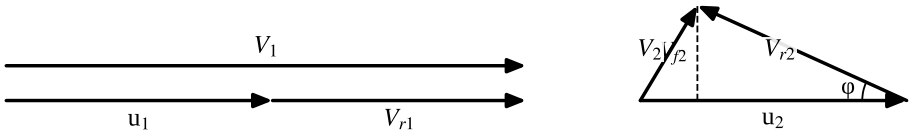
Efficiency of transmission at maximum power $\eta = (H - h_f)/H$ = **66.67 %**

(Condition 1 M, velocity 2 M, discharge 1 M, power 1 M, efficiency 1 M)

b) Pelton wheel — power and efficiencies. 6 M

Given: D = 1.8 m, N = 360 rpm, H = 280 m, d = 0.15 m, deflection = 165°, $V_{r2} = 0.9\ V_{r1}$, $C_v = 0.98$,

$\eta_m = 0.90$



Inlet triangle (straight line) *Outlet triangle*

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

Jet velocity $V_1 = C_v \sqrt{(2gH)} = 0.98 \times \sqrt{(2 \times 9.81 \times 280)} = 72.636$ m/s

Bucket speed $u = u_1 = u_2 = \frac{\pi D N}{60} = \frac{\pi \times 1.8 \times 360}{60} = 33.929$ m/s

Inlet triangle (straight line): $V_{w1} = V_1 = 72.636$ m/s; $V_{r1} = V_1 - u = 38.707$ m/s

Outlet: angle $\varphi = 180° - 165° = 15°$; $V_{r2} = 0.9 \times V_{r1} = 34.837$ m/s

$V_{w2} = V_{r2} \cos \varphi - u = 34.837 \times \cos 15° - 33.929 = -0.2797$ m/s (negative: in direction of u, so it is subtracted)

Discharge $Q = a V_1 = \frac{\pi}{4} \times 0.15^2 \times 72.636 = 0.0177 \times 72.636 = 1.284$ m³/s

Power developed by runner $= \rho Q (V_{w1} + V_{w2}) u = 1000 \times 1.284 \times (72.636 + (-0.2797)) \times 33.929 = 3\,151\,226.6$ W = **3 151.2 kW**

Hydraulic efficiency $\eta_h = \frac{2 (V_{w1} + V_{w2}) u}{V_1^2} = \frac{2 \times (72.636 + (-0.2797)) \times 33.929}{72.636^2} =$ **93.062 %**

Water power (input) $= \rho g Q H = 9810 \times 1.284 \times 280 = 3\,525.8$ kW; Shaft power $= \eta_m \times$ runner power $= 0.9 \times 3\,151.2 = 2\,836.1$ kW

Overall efficiency $\eta_o = \frac{\text{Shaft power}}{\text{Water power}} = \frac{2\,836.1}{3\,525.8} =$ **80.439 %**

(Velocities 1 M, triangles 1 M, power 2 M, efficiencies 2 M)

c) Indicator diagram — effect of acceleration and friction. 6 M

Indicator diagram is the graph of the pressure head in the cylinder against the distance travelled by the piston (stroke length) during one revolution of the crank. Its area is proportional to the work

done by the pump per cycle.

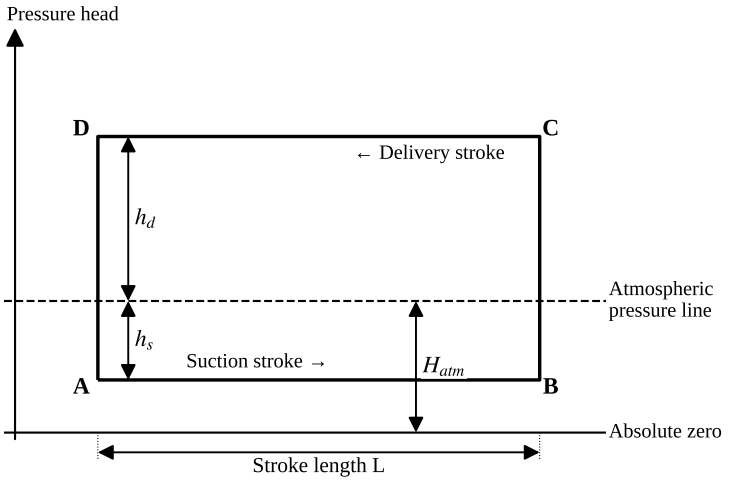


Fig.: Ideal (theoretical) indicator diagram of a reciprocating pump

Ideal diagram ABCD: suction stroke AB at pressure head ($H_{atm} - h_s$), delivery stroke CD at ($H_{atm} + h_d$). Work done $\propto$ area ABCD $\propto (h_s + h_d)$ L.

(i) Effect of acceleration head: the piston (and the water in the pipes) has accelerated motion. Acceleration head $h_a = (l/g)(A/a)\ \omega^2\ r \cos\theta$ — maximum (+) at the beginning and maximum (−) at the end of each stroke and zero at mid-stroke. So the suction line becomes the sloping line starting h_{as} below AB and ending h_{as} above it, and the delivery line starts h_{ad} above CD and ends h_{ad} below it. The area (work done) is unchanged.

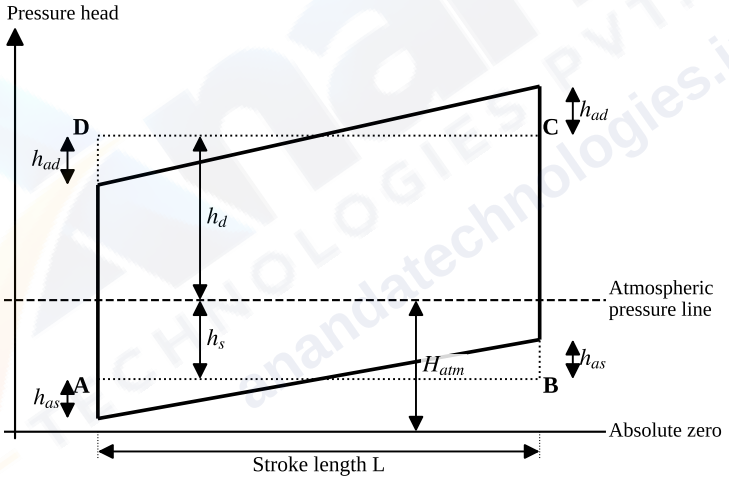


Fig.: Effect of acceleration head on indicator diagram

(ii) Effect of friction head: $h_f = (4fl/2gd)[(A/a)\ \omega\ r \sin\theta]^2$ — zero at the ends and maximum at the middle of each stroke. Suction line becomes a parabola below AB (max h_{fs} at mid-stroke) and

delivery line a parabola above CD (max h_{fd}). The extra area represents work done against friction (work increases by ⅔ h_f L for each stroke).

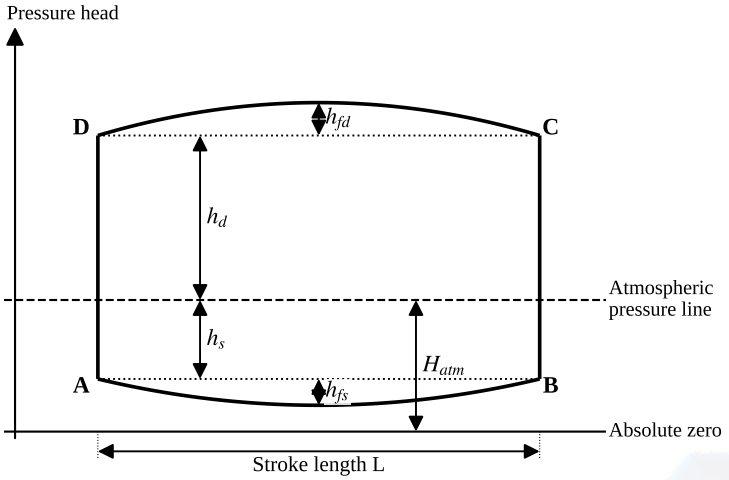


Fig.: Effect of friction head on indicator diagram

Combined effect of acceleration and friction is shown below:

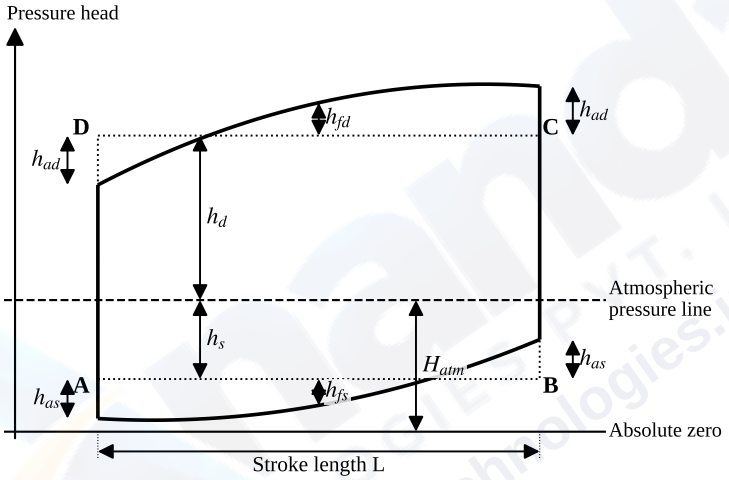


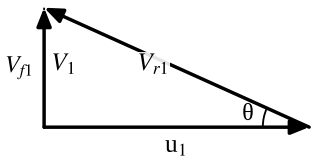
Fig.: Indicator diagram — combined effect of acceleration and friction (dotted = ideal)

(Ideal diagram 1 M, acceleration effect 2 M, friction effect 2 M, combined 1 M)

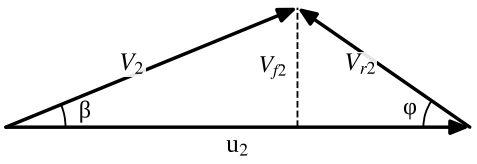
Q.6

a) Centrifugal pump — vane angle, work done, efficiency, power. 6 M

Given: D_1 = 0.3 m, D_2 = 0.6 m, N = 800 rpm, φ = 30°, $V_{f1} = V_{f2}$ = 2.5 m/s, b_2 = 0.05 m, H_m = 40 m, η_o = 0.70



Inlet (radial entry)



Outlet

Fig.: Velocity triangles at inlet and outlet of impeller, not to scale ($V_1 = Vf_1$, $Vw_1 = 0$)

$u_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.3 \times 800}{60} = 12.566$ m/s; $u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.6 \times 800}{60} = 25.133$ m/s

Radial entry: $V_{w1} = 0$, $V_1 = V_{f1} = 2.5$ m/s. Velocity of flow constant: $V_{f2} = 2.5$ m/s

(i) **Vane angle at inlet**: $\tan\theta = \frac{V_{f1}}{u_1} = \frac{2.5}{12.566} = 0.1989 \Rightarrow$ **$\theta = 11.25°$**

Outlet: $V_{w2} = u_2 - \frac{V_{f2}}{\tan\varphi} = 25.133 - \frac{2.5}{\tan 30°} = 25.133 - 4.33 = 20.803$ m/s

(ii) **Work done per kg** of water = $V_{w2}\, u_2 = 20.803 \times 25.133 =$ **522.83 N·m/kg** (= 53.295 m of head)

Discharge $Q = \pi D_2 b_2 V_{f2} = \pi \times 0.6 \times 0.05 \times 2.5 = 0.2356$ m³/s

Work done per second = $\rho Q V_{w2} u_2 = 1000 \times 0.2356 \times 522.83 = 123.19$ kW

(iii) **Manometric efficiency** $\eta_{mano} = \frac{g H_m}{V_{w2} u_2} = \frac{9.81 \times 40}{522.83} =$ **75.054 %**

Absolute velocity at outlet $V_2 = \sqrt{(V_{w2}^2 + V_{f2}^2)} = \sqrt{(20.803^2 + 2.5^2)} = 20.952$ m/s at $\beta = \tan^{-1}\left(\frac{2.5}{20.803}\right) = 6.85°$ to the wheel tangent

(iv) **Power required** = $\frac{\rho g Q H_m}{\eta_o} = \frac{9810 \times 0.2356 \times 40}{0.7} = 132\,081.5$ W = **132.08 kW**

(Vane angle 1 M, work done 1 M, manometric efficiency 2 M, power 2 M)

b) Performance characteristic curves of hydraulic turbines. 6 M

Turbines are tested at different heads, speeds, discharges and gate openings; the results are plotted as characteristic curves using unit quantities $N_u = N/\sqrt{H}$, $Q_u = Q/\sqrt{H}$, $P_u = P/H^{3/2}$. The important curves are:

1. Main characteristic curves (constant head): head is kept constant; speed is varied by changing the load, for different gate openings. Q_u, P_u and η_o are plotted against N_u. For a reaction turbine Q_u falls slightly as speed rises; P_u and η_o are parabolic with a maximum at the design speed.

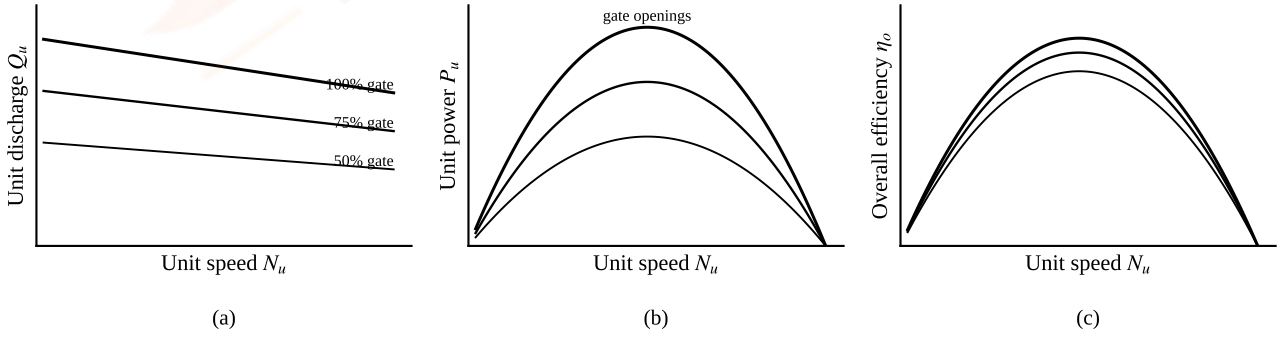
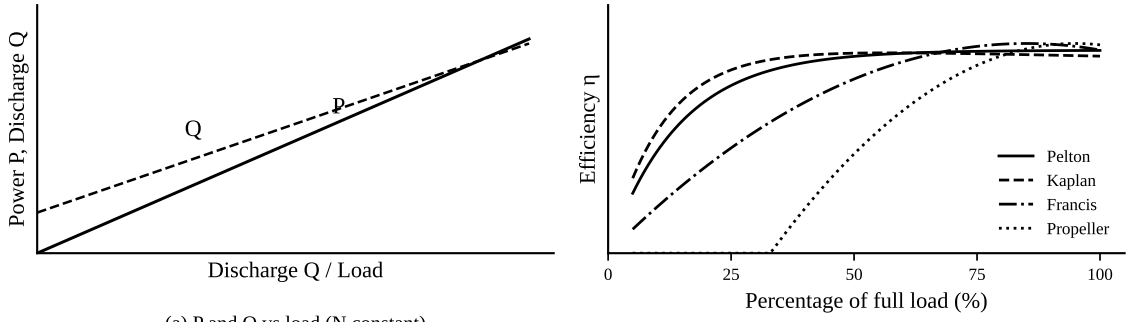


Fig.: Main characteristic curves of a reaction turbine (constant head, varying speed)

2. Operating characteristic curves (constant speed): turbines run at constant (synchronous) speed. P and η are plotted against discharge / percentage load. Kaplan and Pelton turbines keep

high efficiency over a wide range of load; Francis and propeller turbines have lower part-load efficiency.



(a) P and Q vs load (N constant)

(b) Efficiency vs load (N constant)

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

3. Constant efficiency (Muschel) curves: obtained from the main characteristic curves; points of equal efficiency are joined on a Q–N plot. They show the region of best efficiency and are used to select the operating range.

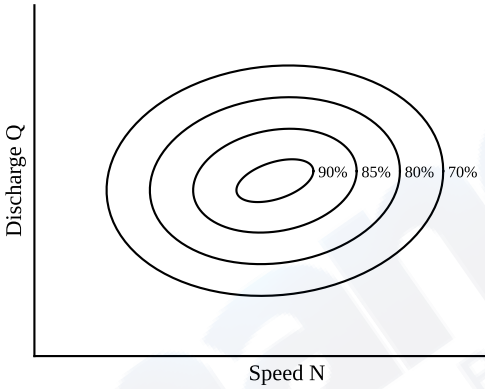


Fig.: Constant efficiency (Muschel) curves

(Each type with curve 2 M)

c) Francis turbine — construction and working. 6 M

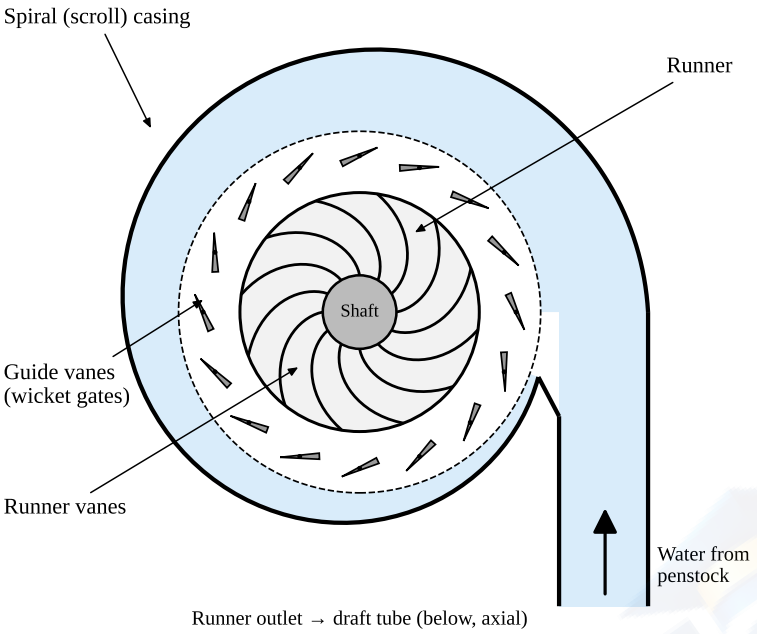


Fig.: Francis turbine — spiral casing, guide vanes and runner (plan view)

Francis turbine is an **inward (mixed) flow reaction turbine** used for medium head (about 60–250 m) and medium discharge.

Main parts:

- **Spiral (scroll) casing:** water from the penstock enters the casing whose cross-section decreases gradually, so that water enters the guide ring with uniform velocity all round.
- **Stay vanes and guide vanes (wicket gates):** guide vanes direct water onto the runner at the correct angle without shock and regulate the discharge; they are pivoted and operated by the governor.
- **Runner:** circular wheel with 16–24 fixed curved vanes between crown and band, keyed to the shaft. Water enters radially at the outer periphery and leaves axially at the centre.
- **Draft tube:** a diverging tube from runner outlet to tail race; recovers kinetic energy and allows the turbine to be placed above tail race.

Working: Water under pressure enters the scroll casing and is distributed all round the guide vanes. The guide vanes convert part of the pressure energy into kinetic energy and direct the water onto the runner vanes. While flowing through the runner, both pressure energy and kinetic energy of water decrease; the reaction of water on the curved vanes produces torque and the runner rotates. Water leaves the runner axially and is discharged through the draft tube to the tail race. The speed is kept constant under varying load by adjusting the guide vane opening with the governor.

(Sketch 2 M, parts 2 M, working 2 M)

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