Chapter 5

Turbo Machinery

CHAPTER HIGHLIGHTS

- *Impact of Free Jets*
- *Force Exerted by a Jet on a Stationary Flat Plate*
- *force Exerted by a Jet on a Stationary Curved Plate*
- *Single Vane*
- *Series of Radial Curved Vanes*
- *Hydraulic Turbines*
- *Pelton Wheel*
- *Blade Friction Coefficient*
- *Gross and Net (effective) Heads*
- *Euler Head or Runner Head (Hr)*
- *Reaction Turbines*
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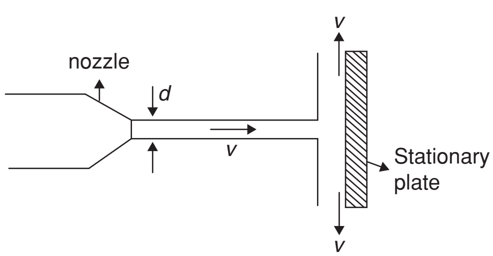
Impact of Free Jets

A fluid jet is a fluid system issuing from a nozzle with a high velocity and thus has a high kinetic energy. When a jet impinges on a plate or vane, due to the change in momentum, the jet exerts a force on the plate or vane.

Force Exerted By a Jet on a Stationary Flat Plate

A Flat Plate Held Normal to a Jet



$$F_X = \rho a v^2$$

Where f_x is the force exerted by the jet on the plate (assumed to be smooth, i.e., no friction between the jet and the plate) in the direction of the jet (x-direction). Here 'a' and 'v' are the cross sectional area and velocity of the jet respectively $\left(a = \frac{\pi d^2}{4}, d \text{ being the diameter of the jet}\right)$

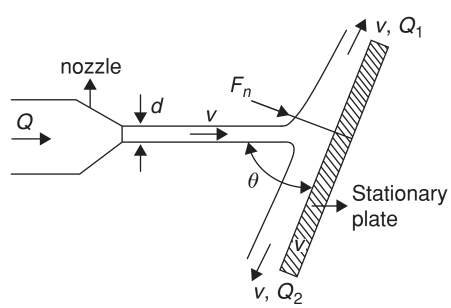
Final velocity of the jet in the x-direction is zero.

The force exerted in the x-direction by the jet is the change in momentum

$$= \rho a v \times v - (\rho a v) \times 0 = \rho a v^2 \frac{\text{kg}}{\text{sec}} \frac{\text{m}}{\text{s}} = \rho a v^2 \text{ Newtons}$$

Since the plate is stationary, the work done on the plate is zero.

A Flat Plate Held Inclined to a Jet



The force exerted by the jet on the plate (assumed to be smooth) in the normal direction,

$$F_n = \rho a v^2 \sin\theta$$

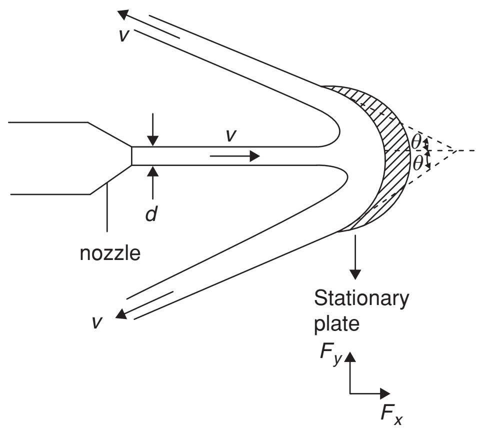
$$Q = Q_1 + Q_2$$

Ratio of discharges $\boxed{\frac{Q_1}{Q_2} = \frac{1+\cos\theta}{1-\cos\theta}}$

Since the plate is stationary, work done on the plate is zero.

Force Exerted by a Jet on a Stationary Curved Plate

Jet Strikes the Curved Plate at the Centre



Force exerted by the jet in the x-direction (direction of the jet).

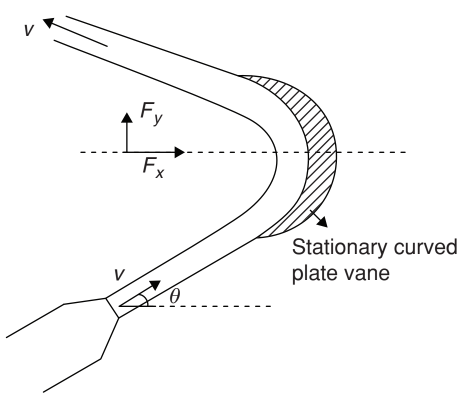
$$\boxed{F_x = \rho a v^2 (1+\cos\theta)}$$

Fore exerted by the jet in the y-direction (normal to the jet direction).

$$\boxed{F_y = -\rho a v^2 \sin\theta}$$

It is to be noted that the curved plate is assumed to be smooth. The angle of deflection of the jet = (180° – θ).

Jet Strikes the Curved Plate at One End tangentially When the Plate is Symmetrical



The stationary curved plate plane is symmetrical about the x-axis and it is assumed to be smooth.

$$\boxed{F_x = 2\rho a v^2 \cos\theta}$$

$$\boxed{F_y = 0}$$

Here θ is the angle made by the jet with the x-axis at in let tip of the curved plate.

Jet Strikes the Curved Plate or Vane at One End Tangentially When the Plate is Unsymmetrical

In this case, the curved plate is unsymmetrical about the x-axis. Let θ and ϕ be the angles made by the tangents drawn at the inlet and outlet tips of the plate/vane respectively with the x-axis.

$$\boxed{F_x = \rho a v^2 (\cos\theta + \cos\phi)}$$

$$\boxed{F_y = \rho a v^2 (\sin\theta + \sin\phi)}$$

Force Exerted by a Jet on a Moving Flat Plate

A Moving Flat Plate Held Normal to a Jet

$$\boxed{Fx = \rho a (v-u)^2}$$

Work done per second on the plate,

$$\boxed{W = \rho a (v-u)^2 \times u}$$

A Moving Flat Plate Held Inclined to a Jet

$$\boxed{F_x = \rho (v-u)^2 \sin\theta}$$

$$\boxed{W = \rho a (v-u)^2 \sin\theta \times u}$$

Force Exerted on a Moving Curved Plate or Vane

Single Vane

$$\boxed{\begin{aligned} F_x &= \rho a (v-u)^2 (1+\cos\theta) \\ W &= \rho a (v-u)^2 (1+\cos\theta) u \end{aligned}}$$

Efficiency,

$$\boxed{\eta = \frac{2(v-u)^2(1+\cos\theta)u}{v^3}}$$

For a given jet velocity, the efficiency is maximum when the vane velocity is one-third of the jet velocity, i.e., $u = \frac{v}{3}$.

Series of Vanes

$$F_x = \rho av(v-u)(1+\cos\theta)$$

$$W = \rho av(v-u)(1+\cos\theta)u$$
$$\eta = \frac{2u(v-u)(1+\cos\theta)}{v^2}$$

For a wheel, consisting of a series of vanes, the efficiency is maximum when the peripheral speed (u) is one-half the jet velocity, i.e., $u = \frac{v}{2}$.

$$\eta_{\max} = \frac{1+\cos\theta}{2}$$

θ_m When the vanes are semicircular i.e., $\theta = 0$, $\eta_{\max} = 100\%$.

Jet Striking a Moving Curved Vane Tangentially at One Tip and Leaving Other

Single Vane

Let, v_1, v_2 $\rightarrow$ Jet's absolute velocities at inlet and outlet u_1, u_2 $\rightarrow$ Vane's peripheral velocities at inlet and outlet (tangential velocity of vane)

V_{r_1}, V_{r_2} $\rightarrow$ Relative velocities of the jet at inlet and outlet with respect to the vane.

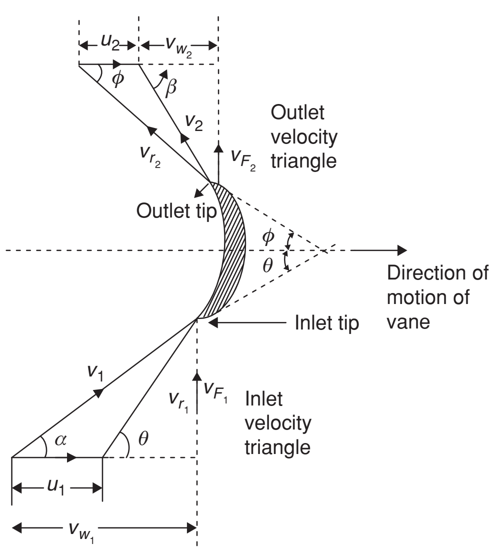
V_{f_1}, V_{f_2} $\rightarrow$ Flow velocities at inlet and outlet.

V_{W_1}, V_{W_2} $\rightarrow$ Velocity of whirl at input and output

θ, ϕ $\rightarrow$ Tip angles of inlet and outlet (φ is also called as vane angle at outlet)

α, β $\rightarrow$ Angles made by the absolute velocities at the inlet and outlet

Here, all angles are measured with the direction of motion of the vane. The velocity of whirl is the component of the absolute velocity in the direction of motion of the vane while the flow velocity is the component normal to the direction of motion of vane.



$$F_x = \rho a v_{r_1}(v_{w_1} \pm v_{w_2})$$

In the above equation the plus sign is used when β is an acute angle and the minus sign when β is obtuse. When $\beta = 90°, v_{w_2} = 0.$ Work done per second by the jet on the vane,

$$W = \rho a v_{r_1}(v_{\omega_1} \pm v_{\omega_2}) \times u$$

Work done per second per unit weight of fluid striking,

$$W_\omega = \frac{1}{g}(v_{\omega_1} \pm v_{\omega_2})u$$

Series of Radial Curved Vanes

Let 'ω' be the constant angular speed of the wheel. Let R_1 and R_2 be the radii of the wheel at the inlet and outlet of the vane respectively.

$$\therefore \begin{array}{l} u_1 = \omega R_1 \\ u_2 = \omega R_2 \end{array}$$

The flow system is inward or outward depending upon whether the jet enters the outer or inner periphery.

Torque exerted by water on the wheels

$$\tau = \rho a v_1 (v_{\omega_1} \times R_1 + v_{w_2} \times R_2)$$

Work done per second on the wheel,

$$\rho a v_1 (v_{\omega_1} u_1 \pm v_{\omega_2} u_2)$$

If the discharge is radial, then β = 90°. Efficiency of the radial curved vane,

$$\eta_{vane} = \frac{2[v_{\omega_1} u_1 \pm v_{\omega_2} u_2]}{v_1^2}$$

Hydraulic Turbines

A hydraulic turbines is a hydraulic (or fluid machine) that converts hydraulic energy (energy possessed by water) into mechanical energy which can be further utilized to generate electric power.

In a hydraulic turbine, a wheel on which blades or buckets are mounted is directed against a flow of water to alter the momentum of the flowing water. As the momentum is changed with the passage of the water through the wheel, the resulting force rotates the shaft of the wheel performing work to generate power. Hydraulic turbines belong to the category of rotodynamic machines.

Classification of Hydraulic Turbines

Several criteria are used to classify hydraulic turbines, some of which are given below:

1. According to the action of water on the turbine blades
   (i) Impulse turbine: In an impulse turbine, e.g., Pelton wheel, at the supply nozzle the total head of the incoming fluid is converted into a large velocity

head in the form of a high velocity jet that strikes the buckets. This leads to the rotation of the wheel. The pressure all over the wheel is constant and equal to atmospheric pressure so that energy transfer occurs due to purely impulse action. At the inlet of this type of turbine, only kinetic energy is available.

(ii) Reaction turbines: Reaction turbines, e.g., Francis, Kaplan and Propeller turbines, always runs full where the water enters the turbine under pressure. The rotation of runner or rotor is partly due to impulse action and partly due to change in pressure over the runner blades. At the inlet of this type of turbine, water possesses both kinetic and pressure energy.

2. According to the head at inlet of turbine
   (i) High-head turbine: In this type of turbine, net head varies from 150 m, to 2000 m or more and they require a small quantity of water. For example, Petron wheel
   (ii) Medium-head turbine: The net head varies from 30 m to 150 m and the requirement is a moderate quantity of water for this type of turbines. For example, Francis turbine.
   (iii) Low-head turbine: For this type of turbines, the net head is less than 30 m and the requirement is a large quantity of water. For example, Kaplan turbine.
3. According to the direction of How through the runner
   (i) Tangential flow turbine: In this type of turbine, water flows tangentially to the runner. For example, Pelton turbine.
   (ii) Radial flow turbine: In this type of turbine, water flows in the radial direction. It is further classified. It is further classified as
   Inward radial flow turbine: Here water flows radially from outwards to inwards. For example, old Francis turbine
   Outward radial flow turbine: Here water flows radially from inwards to outwards. For example, Fourneyron turbine.
   (iii) Axial flow turbine: In this type of turbine, water flows parallely to the axis of rotation of the runner. For example, Kaplan turbine.
   (iv) Mixed flow turbine: In this type of turbine, water flows through the runner and leaves axially, i.e., parallel to the axis of rotation of the runner. For example,modern Francis turbines.
4. According to specific speed
   (i) Low specific speed turbine: In these turbines, the specific speed is less than 50 (varying from 10 to 35 for single jet and upto 50 for double jet). For example, Pelton wheel.
   (ii) Medium specific speed turbines: The specific speed varies from 50 to 250 for these turbines. For example, Francis turbine.
   (iii) High specific speed turbine: In these turbines, the specific speed is more than 250. For example, Kaplan turbine.

Pelton Wheel

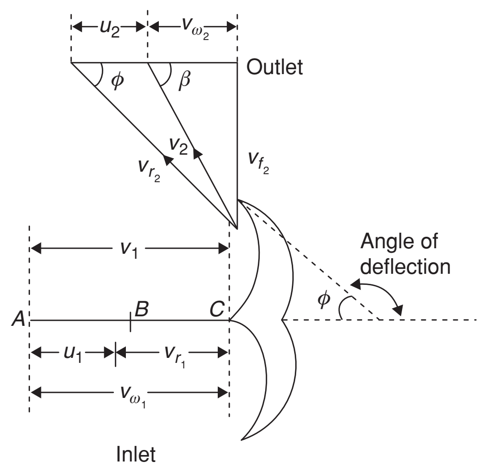
The pelton wheel (or pelton turbine) is a tangential flow impulse turbine. Water from a reservoir flows through penstocks at the outlet of which a nozzle is fitted. The nozzle increases the kinetic energy of the water flowing through the penstocks. At the outlet of the nozzle, water comes out in the form of a jet and strikes the buckets (or vanes) of the runner. This causes the rotation of the runner wherein the kinetic energy gets converted to mechanical energy.

The important parts of a pelton wheel other than the nozzle are:

1. Rotor: At the periphery of the rotor, equally spaced double hemispherical or double ellipsoidal buckets are mounted.
2. Needle spear: It is present in the nozzle and functions to control of the water flow through the nozzle and to provide a smooth flow with negligible losses.
3. Casing: If functions to prevent splashing of water and to discharge water to the tail race.
4. Brake nozzle: If functions to stop the runner in a short time by directing a jet of water, called the braking jet, on the back of the vanes.

Velocity Triangles of a Pelton Wheel

A velocity triangle or velocity diagram is a triangle representing the various components of velocities of the working fluid in a turbo machine. The inlet and outlet velocity triangles for a pelton wheel are shown below:



$$u_1 = u_2 = u = \frac{\pi DN}{60}$$

Where N is the speed of the wheel in rpm and D is the diameter of the wheel.

guide angle (α), the angle between the direction of the jet and direction of motion of the vane/bucket is zero.

Vane angle at inlet (θ), angle made by the relative velocity v_{r_1} with the direction of motion of the inlet, is zero

Runner Types

Depending on the magnitude of the peripheral speed (u), the unit may have a slow, medium or fast runner.

slow runner	$\beta < 90°$	v_{ω_2} is negative
Medium runner	$\beta = 90°$	v_{ω_2} is zero
Fast runner	$\beta > 90°$	v_{ω_2} is positive

Blade Friction Coefficient (k)

$$k = \frac{v_{r_1}}{v_{r_2}}$$

In the absence of friction between fluid and blade surface, $k = 1$

Work Done, Power, Force, and Torque

At inlet

$$U_{r_1} = v_1 - u_1 = v_1 - u$$
$$v_{\omega_1} = v_1$$

At outlet

$$v_{\omega_2} = v_{r_2}\cos\phi - u_2 = v_{r_2}\cos\phi - u$$

Work done per unit weight of water striking,

$$W = \frac{1}{g}(v_{\omega_1} + v_{\omega_2})u$$

Kinetic energy of jet per second

$$KE_{jet} = \frac{1}{2}(\rho a v_1)v_1^2$$

Where $a = \frac{\pi}{4}d^2$, d being the diameter of the Jet and a being the area of the jet. Power delivered to the runner by the water or power developed by the runner,

$$P_r = \rho Q(v_{\omega_1} + v_{\omega_2})u$$
$$= \rho Q(v_1 - u)(1 + k\cos\phi)u$$

Force exerted on the bucket by the water jet, $F = \frac{P}{u}$

$$F = \rho Q(v_{\omega_1} + v_{\omega_2})$$
$$= \rho Q(v_1 - u)(1 + k\cos\phi)$$

Torque acting on the shaft of the Pelton wheel, $\tau = F \times \frac{D}{2}$, when D is the pitch diameter of the Pelton wheel.

$$\tau = \frac{1}{2}\rho DQ(v_1 - u) + (1 + k\cos\phi)$$
$$= \frac{1}{2}\rho QD(v_{\omega_1} + v_{\omega_2})$$

Gross and Net (effective) Heads

Gross head (H_g) is the difference between the head race level (water level of reservoir) and the water level at the tail race. Net or effective head (H) is the head available at the inlet of the turbine. If h_L is the total loss of head between the head race and entrance of the turbine and h is the height of the nozzle above the water level at the tail race, then

$$H = H_g - h_L - h$$

Efficiencies of a Turbine

Let P_i be the power supplied at the turbine inlet by the water jet water power, P_r be the power delivered to the runner by the water or power developed by the runner and P_s be the power available at the turbine shaft (Shaft power)

Hydraulic efficiency (η_h)

$$\eta_\eta = \frac{P_r}{P_i} = \frac{P_r}{\rho g Q H}$$

Where Q is the volume flow rate of the water supplied by the jet to the turbine

$$\eta_\eta = \frac{\text{Work done per second}}{\text{Kinetic energy of jet per second}}$$
$$= \frac{\rho Q g(v_{\omega_1} + v_{\omega_2}) \times u}{g\frac{1}{2}(\rho a v_1)v_1^2}$$

$$\eta_h = \frac{2(v_{\omega_1} + v_{\omega_2}) \times u}{v_1^2}$$
$$= \frac{2(v_1 - u)(1 + k\cos\phi)u}{v_1^2}$$

***Euler head or runner head* (Hr)** It represents the energy transfer per unit weight of water.

$$H_r = \frac{1}{g}(v_{w_1} + v_{w_2})u$$
$$H - H_r = \Delta H$$

Where ΔH is the hydraulic losses within the turbine.

Hydraulic efficiency of a Pelton wheel is maximum when the velocity of the wheel is half the velocity of jet of water at inlet, i.e, $u = \frac{v_1}{2}$.

Maximum hydraulic efficiency of a Pelton wheel,

$$(\eta_h)_{\max} = \frac{1 + k\cos\phi}{2}$$

Mechanical efficiency (η_m)

$$\eta_m = \frac{P_s}{P_r}$$

Overall efficiency (η_0)

$$\eta_0 = \frac{P_s}{P_i} = \frac{P_s}{\rho g Q H}$$

Volumetric efficiency (η_v)

$$\eta_v = \frac{\text{Volume of water actually striking the runner}}{\text{Volume of water supplied by the jet to the turbine}}$$

$$\eta_v = \frac{Q_a}{Q}$$

Where Q_a is the volumetric flow rate of the water actually striking the runner.

NOTE

If $\eta_h = \dfrac{P_r}{\rho g Q_a H}$ (which is the true definition) instead of $\eta_h = \dfrac{P_r}{\rho g Q H}$ (Which is the usual definition assuming volumetric efficiency as 100%), then

$$\eta_0 = \eta_h \times \eta_m \times \eta_v$$

Else

$$\eta_0 = \eta_h \times \eta_m$$

Design Aspects

1. Velocity of jet at inlet,

$$v_1 = C_v\sqrt{2gH}$$

   Where C_v is the coefficient of velocity (= 0.98 or 0.99)
2. Velocity of wheel,

$$u = k_u\sqrt{2gH}$$

   Where k_u is the speed ratio (= 0.43 to 0.48). The speed ratio is defined as the ratio of the velocity of the wheel to the theoretical jet velocity at inlet.
3. Angle of deflection is to be taken as 165° if not given.
4. Mean diameter or pitch diameter D of the pelton wheel is given by

$$D = \frac{60u}{\pi N}$$

5. Jet ratio (m) is given by

$$m = \frac{D}{d}$$

   The range of m lies between 11 and 16 (for maximum hydraulic efficiency) where m = 12 for most cases.
6. Number of buckets on a runner (z) is given by

$$z = 15 + \frac{D}{2d}$$

7. The number of jets is obtained by dividing the total rate of flow through the turbine by the flow rate of water through a single jet.

Solved Examples

Example 1: The head loss (neglecting miner losses) in a penstock of a single jet pelton wheel installation fitted with a frictionless nozzle and generating maximum power is 40 m. If the height of the nozzle above the water level at the tail race is 20 m, then the gross head available to the turbine is

(A) 120 m (B) 180 m
(C) 140 m (D) 160 m

Solution:

Power developed by the runner.

$P_r = \rho Q\,(v_1 - u)(1 + k\cos\phi)\,u$

$\quad = \rho a v_1\,(v_1 - u)(1 + k\cos\phi)\,u$

Where a is the area of the jet

The power will be maximum when $u = \dfrac{v_1}{2}$

That is, $P_{\max} = \dfrac{\rho a(1 + k\cos\phi)}{4} v_1^3$

$P_{\max} = CV_1^3$

Where C is a constant and equal to $\dfrac{\rho a(1 + k\cos\phi)}{4}$

Applying Bernoulli's equation between a point on the water surface at the reservoir and a point at the centre of the nozzle outlet, we get

$$H_g - h = \frac{fL}{D}\frac{v_{pipe}^2}{2g} + \frac{v_1^2}{2g}$$

Where L and D are the length and diameter of the penstock.

$$\therefore\quad v_1^2 = 2g(H_g - h) - \frac{fL}{D}v_{\text{pipe}}^2$$

$$\therefore\quad P_{\max} = C\left[2g(H_g - h) - \frac{fL}{D}v_{\text{pipe}}^2\right]v_1$$

$$= C\left[2g(H_g - h)v_1 - \frac{fL}{D}v_1^3\frac{d^4}{D^4}\right]$$

Here $\dfrac{dp_{\max}}{dv_1} = 0$

$$\Rightarrow\quad 2g(H_g - h) = f\frac{L}{D}3v_1^2\frac{d^4}{D^4}$$

$\therefore \quad H_g - h = 3h_L$

Given $h_L = 40$ m

$h = 20$ m

$\therefore \quad H_g = 3 \times 40 + 20 = 140$ m.

Example 2: A pelton wheel operates with a speed ratio of 0.46. Both the coefficient of velocity and the blade friction coefficient are having the same value of 0.98. If the bucket angle at the outlet is 165° and the mechanical efficiency is 95% then the overall efficiency is

(A) 93.2% (B) 76.4%
(C) 63.5% (D) 88.5%

Solution:

given $k_u = 0.46$

$C_v = 0.98$

$K = 0.98$

$\phi = 180° - 165° = 15°$

$$\text{Overall efficiency} = \eta_0 = \frac{P_s}{P_i}$$

$$= \frac{\eta_m \times p_r}{p_i}$$

$$= \frac{\eta_m \times \rho Q(v_1 - u)(1 + k\cos\phi)u}{\rho Q H g}$$

$$= \frac{\eta_m \times u^2 \left(\frac{v_1}{u} - 1\right)(1 + k\cos\phi)}{gH}$$

Now $v_1 = C_v\sqrt{2gH}$ and

$$u = k_u\sqrt{2gH}$$

$$\therefore \quad \frac{v_1}{u} = \frac{C_v}{k_u}$$

$$\therefore \quad \eta_0 = \frac{\eta_m \times k_u^2 2gH\left(\frac{C_v}{k_u} - 1\right)(1 + k\cos\phi)}{gH}$$

$$= 0.95 \times 2 \times 0.46^2 \times \left(\frac{0.98}{0.46} - 1\right)(1 + 0.98 \times \cos 150°)$$

$= 88.5\%$.

Example 3: A Pelton wheel operates with a bucket peripheral speed of u m/s where the actual jet velocity at the inlet is given by v_1 m/s. The blade friction coefficient is k whereas the angle of deflection is θ. Assuming that volumetric efficiency is 100% and that an additional loss of head given by $\frac{k_1(v_1 - u)^2}{2g}$ (k_1 being a constant) occurs due to the bucket friction and shock, then the maximum efficiency of the pelton wheel occurs when $\frac{u}{v_1}$ is equal to:

(A) $\dfrac{1 + k\cos\theta + k_1}{k_1 + 2(1 + k\cos\theta)}$

(B) $\dfrac{1 - k\cos\theta + k_1}{k_1 + 2(1 - k\cos\theta)}$

(C) $\dfrac{1 - k\cos\theta + k_1}{0.5k_1 + 2(1 - k\cos\theta)}$

(D) $\dfrac{1 - k\cos\theta - k_1}{-k_1 + 2(1 - k\cos\theta)}$

Solution:

The work done per unit weight of water striking,

$$W = \frac{1}{g}(v_1 - u)(1 + k\cos\phi)u$$

$$= \frac{1}{g}(v_1 - u)(1 + k\cos(180° - \theta))u$$

$$= \frac{1}{g}(v_1 - u)(1 - k\cos\theta)u$$

Considering the losses due to bucket friction and shock, we have:

$$W = \frac{1}{g}(v_1 - u)(1 - k\cos\phi)u - \frac{k_1(v_1 - u)^2}{2g}$$

$$\therefore \eta_h = \frac{\text{Work done per second}}{\text{Kinetic energy of jet per second}}$$

$$= \frac{\rho Q g\left[\frac{1}{g}(v_1 - u)(1 - k\cos\theta)u - \frac{k_1(v_1 - a)^2}{2g}\right]}{\frac{1}{2}(\rho Q g)v_1^2}$$

$$= \frac{\frac{2}{g}(v_1 u - u^2)(1 - k\cos\theta) - k_1\frac{(v_1 - u)^2}{g}}{v_1^2}$$

For the efficiency to be maximum:

$$\frac{d\eta_h}{du} = 0$$

$$\Rightarrow \quad \frac{2}{g}(1 - k\cos\theta)(v_1 - 2u) + \frac{k_1^2}{g}(v_1 - u) = 0$$

Or efficiency is maximum when

$$\frac{u}{v_1} = \frac{1 - k\cos\theta + k_1}{k_1 + 2(1 - k\cos\theta)}.$$

Reaction Turbines

In reaction turbines, as water flows through the stationary parts of the turbine, whole of its pressure energy is not transformed into kinetic energy. When the water flows through the moving parts, the pressure and absolute velocity of flow of water reduces. Important reaction turbines are Francis, Kaplan and Propeller.

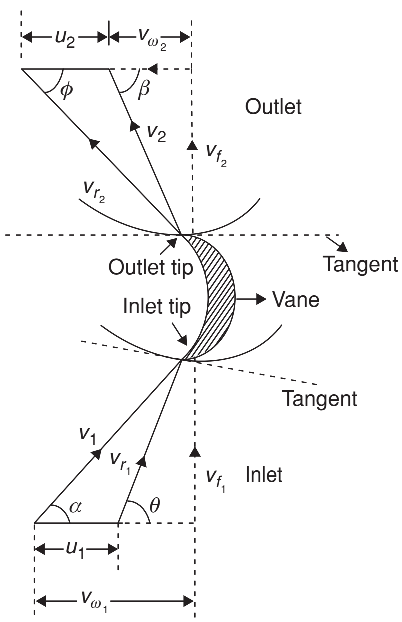
Francis Turbine

The main parts of a Francis turbine are

1. Spiral/scroll casing: It constitutes a closed passage whose cross-sectional area gradually decreases along the direction of flow where the area is maximum at inlet and minimum at exit.
2. Guide vanes/wicket gates: The vane direct the water into the runner at the desired angle
3. Runner and runner blades
4. Draft tube: It is a gradually expanding tube which discharges water from the runner to the tail race.

Velocity Triangles

The velocity triangles for an inward flow reaction turbine are shown below:



Net Head

It is also called available or working or operation head.

$$H = H_g - h_L$$

Here h_L is the loss of head in the penstock.

H = Total head available at exit from the penstock – Total head available at exist from the draft tube.

$$H = H_g - h_L - \frac{v_d^2}{2g}$$

Where v_d is the velocity at the exit of the draft tube.

Work Done

$$W = \rho Q(v_{\omega_1}u_1 \pm v_{\omega_2}u_2)$$

Where Q is the discharge through the runner. The maximum output is obtained when the tangential or whirling component of the velocity of the outlet becomes zero, i.e., $v_{\omega 2} = 0$. This is made to keep the kinetic energy at outlet a minimum.
∴ Maximum work is given by:

$$W_{\max} = \rho Q(v_{\omega_1}u_1)$$

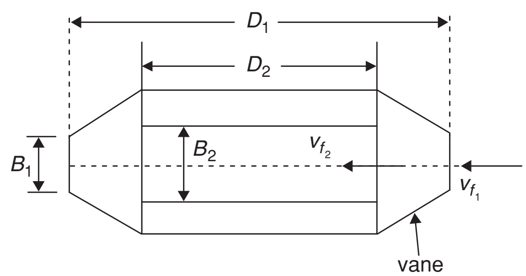
The discharge in this case is radial, i.e., absolute velocity at exit is radial.

Hydraulic Efficiency (η_h)

$$\eta_h = \frac{v_{\omega_1}u_1 \pm v_{\omega_2}u_2}{gH}$$

Working Proportions of a Francis Turbine

The entry of flow to a runner vane is shown below:



1. $\frac{B_1}{D_1} = n$

   The value of n varies from 0.1 to 0.45.
2. Flow ratio (k_f): It is the ratio of the velocity of flow at inlet to the theoretical jet velocity.

$$k_f = \frac{v_{f_1}}{\sqrt{2gH}}$$

   The value of k_f varies from 0.15 to 0.30
3. Speed ratio (k_u)

$$k_u = \frac{u}{\sqrt{2gH}}$$

   The value of k_u varies from 0.6 to 0.9.

Design Aspects of a Francis Turbine Runner

1. Total area of the outer periphery, i.e., at the runner inlet

$$A = k_{t_1}\pi D_1 B_1$$

where k_{t_1} is called the vane thickness factor/coefficient. K_{t_1} always has a value less than unity.

2. Discharge,

$$Q = k_{t_1}\pi D_1 B_1 v_{f_1}$$
$$Q = k_{t_2}\pi D_2 B_2 v_{t_2}$$

3. Guide vane angle (α)

$$\tan\alpha = \frac{v_{f_1}}{v_{\omega_1}}$$

4. Runner vane angle (θ)

$$\tan\theta = \frac{v_{f_1}}{v_{\omega_1} - u_1}$$

The angle θ is 90° when the runner vanes are radial at inlet.

5. Runner vane angle at exit (ϕ)

$$\tan\phi = \frac{v_{f_2}}{u_2}$$

This is obtained by assuming the discharge at the runner exit to be radial ($\beta = 90°$).

6. In order to avoid periodic impulse, the number of vanes should be either one more or one less than the number of guide vanes.
7. Tangential velocity (peripheral velocity) of the runner at inlet (u_1) and at outlet (u_2)

$$u_1 = \frac{\pi D_1 N}{60}$$

$$u_2 = \frac{\pi D_2 N}{60}$$

θ, $\phi \rightarrow$ vane angle at inlet and outlet
$B_1, B_2 \rightarrow$ width of the runner vane at inlet and outlet
$D_1, D_2 \rightarrow$ Diameter of the runner (wheel diameter) at the inlet and outlet.

8. The blade efficiency is given by:

$$\eta_b = \frac{2v_{f_1}^2 \cot\alpha(\cot\alpha - \cot\beta)}{v_{f_2}^2 + 2v_{f_1}^2 \cot\alpha(\cot\alpha - \omega t\beta)}$$

Example 4: Water flows at 6 m/s through a penstock of 5.5 m diameter into a Francis turbine that develops a shaft power of 83.5 MW. The static pressure head in the penstock measured just before entry to the runner is 59 m, with the point of measurement being 46 m above the tail race level. If the velocity of water in the tail race level is 8 m/s, then the overall efficiency of the turbine is:

(A) 84.2% (B) 91.52%
(C) 96.04% (D) 98.32%

Solution:
Given, $D_p = 5.5$ m
Power, $P_\omega = 83.5 \times 10^6$ W

$$\frac{P}{\rho g} = 59 \text{ m}$$
$$Z = 4.6 \text{ m}$$
$$V = 6 \text{ m/s}$$
$$V_t = 8 \text{ m/s}$$

Net head (H) = Head at entry to the runner – Kinetic energy in tail race

$$H = \frac{P}{\rho g} + \frac{v^2}{2g} + z - \frac{v_t^2}{2g}$$
$$= 59 + \frac{6^2}{2\times 9.81} + 4.6 - \frac{8^2}{2\times 9.81}$$
$$= 62.173 \text{ m}$$
$$\eta_0 = \frac{P_\omega}{\rho g Q H}$$
$$= \frac{83.5\times 10^6}{1000\times 9.81\times \frac{\pi}{4}\times 5.5^2\times 6\times 62.173}$$
$$= 96.04\%.$$

Example 5: From the guide vanes of an inward radial flow turbine, which has an available net head of 15 m and water leaves at an angle of 12° to the tangent to the wheel. The vane angle of entry to the wheel is 90°. If the turbine is operating at maximum efficiency and the velocity of flow through the wheel is constant, then the peripheral speed of the wheel at the entry is:

(A) $\left(\frac{2gH}{2+\tan^2\alpha}\right)^{\frac{1}{2}}$ (B) $\left(\frac{2gH}{2+\tan\alpha}\right)^{\frac{1}{2}}$

(C) $\left(\frac{2gH}{2+\tan(\alpha^2)}\right)^{\frac{1}{2}}$ (D) $\left(\frac{2gH}{1+\tan^2\alpha}\right)^{\frac{1}{2}}$

Solution:
Head supplied = Work done + kinetic head of exit

$$H = \frac{v_{\omega 1}u_1 \pm v_{\omega 2}u_2}{g} + \frac{v_2^2}{2g}$$

Given, $\alpha = 12°$
$H = 15$ m
$\theta = 90°$
Since $\theta = 90°$,
$v_{\omega 1} = u_1$
And $v_{f1} = u_1\tan\alpha$
Since velocity of flow is constant,

$$v_{f2} = v_{f1} = u_1\tan\alpha$$

For conditions of maximum efficiency, the flow leaves the runner radially.

That is, $v_{\omega 2} = 0$ and $v_2 = v_{f_2} = u_1 \tan\alpha$

$$\therefore \quad H = \frac{u_1^2}{g} + \frac{u_1^2 \tan^2 \alpha}{2g}$$

Or
$$u_1 = \sqrt{\frac{2gH}{2+\tan^2 \alpha}}.$$

Example 6: An inward flow reaction turbine, with an available net head of 15 m, has a blade thickness coefficient of 0.9025. On changing the blades of the turbine, keeping the wheel's diameter and width of the inlet and outlet and the flow ratio unchanged, only 5% of the area of flow was now blocked by the blade thickness. If the discharge through be runner is still be same as before, then the net head available to the turbine is:

(A) 16.62 m (B) 13.54 m
(C) 15.79 m (D) 14.25 m

Solution:
Before changing the blades, blade thickness coefficient, k_{t1} = 0.9025 net lead, H_1 = 15 m.
After changing the blades, blade thickness coefficient,

$$k_{t_2} = \left(1 - \frac{5}{100}\right)$$
$$= 0.95$$
$$Q = k_t \pi D B v_f$$
$$V_f = kf \frac{h}{\rho V C_p} = \frac{h}{\rho V C p} \times \left(\frac{L}{k} \times \frac{k}{L}\right) \times \frac{\mu}{\mu}$$
$$\therefore \quad Q = k_t \pi D B \, \text{kg} \;\; \rho = 995 \frac{\text{kg}}{\text{m}^3}, \mu = 7.65 \times 10^{-4} \frac{\text{kg}}{\text{m-s}}$$

With Q, D, B and k_f being constant, we have $k_t\sqrt{H}$ = constant

$$\therefore \quad k_{t2}\sqrt{H_2} = k_{t1}\sqrt{H_1}$$

Or
$$H_2 = \left(\frac{kt_1}{kt_2}\right)^2 \times H_1$$
$$= \left(\frac{0.9025}{0.95}\right)^2 \times 15$$
$$= 13.5375.$$

Axial Flow Reaction Turbines – Kaplan and Propeller Turbines

The shaft of such turbines is vertical and the lower end of the shaft which is made larger is known as the hub or boss. The vanes are fixed on the hub and these acts as a runner for this type of turbines. In such a turbine, water enters the runner in an axial direction and leaves axially with the energy transfer being due to the reaction effect, i.e., change in the relative velocity's magnitude across the blades.

The pressure of the inlet of the blades is larger than that of the exit of the blades. In a propeller turbine, the runner blades are fixed and non-adjustable while in a Kaplan turbine they are adjustable. The water leaving the guide vanes of an axial flow reaction turbine undergoes a whirl which is assumed to become a free vertex.

Important Points for a Propeller or Kaplan Turbine

1. Expressions for work done, efficiency and power developed are identical to those of a Francis turbine:

$$\boxed{n = \frac{D_b}{D_o}}$$

Where D_o is the outside diameter of the runner and D_b is the diameter of the hub or boss. The value of n varies from 0.55 to 0.6.

2. Inlet and outlet peripheral velocities are the same since the flow is axial, i.e.,

$$\boxed{u_1 = u_2 = u = \frac{\pi D_o N}{60}}$$

3. Velocity of flow at inlet and outlet are equal, i.e.,

$$\boxed{vf_1 = v_{f2} = v_f}$$

4. Area of flow at inlet = area of flow at outlet

$$= \frac{\pi}{4}(D_o^2 - D_b^2)$$

5. The discharge Q flowing through the runner is given by

$$\boxed{Q = \frac{\pi}{4}(D_o^2 - D_b^2)v_f}$$

6. The flow ratio

$$\boxed{k_f = \frac{v_f}{\sqrt{2gH}}}$$

Where the value of k_f is around 0.7 for a Kaplan turbine.

7. Peripheral velocity of the runner blade is dependent upon the diameter under consideration and varies from section to section along the blade.

Degree of Reaction

The amount of energy transferred, per unit weight of the fluid, between the fluid and the rotor (H) is given by

$$\boxed{H = \frac{1}{2g}\left[\left(v_1^2 - v_2^2\right) + \left(u_1^2 - u_2^2\right) + \left(v_{r_2}^2 - v_{r_4}^2\right)\right]}$$

Where H is also called as the work head. The first term of the above equation represents the change in the dynamic head of the fluid while flowing through the rotor whole the sum of the second and third terms represent the change in the static head of the fluid.

Degree of reaction (R) is defined as the ratio of energy transfer by the change in static head to the total energy transfer in the rotor.

$$R = \frac{\frac{1}{2g}\left[\left(u_1^2 - u_2^2\right) + \left(v_{r_2}^2 - v_{r_4}^2\right)\right]}{H}$$

For an impulse machine, the change in static head in the rotor is zero, hence $R = 0$

For a reaction turbine,

$$R = 1 - \frac{\cot\alpha}{2(\cot\alpha - \cot\theta)}$$

Example 7: The degree of reaction will be zero for a
(A) Francis turbine (B) Kaplan turbine
(C) Pelton wheel (D) Propeller turbine

Solution: (C)

Runaway Speed

It is the maximum speed, with the disengagement of the governor, at which a turbine would run when there is no external load but with the turbine operating under design head and discharge. If N denotes the rated speed of a turbine, then the practical runaway speeds for the different types of a turbine are:

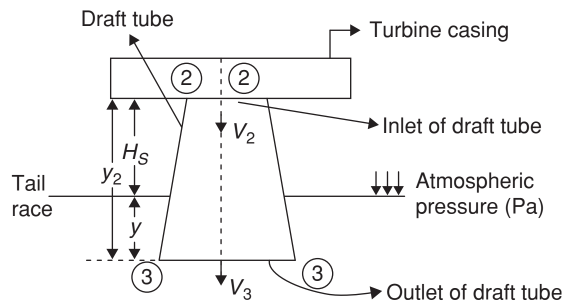
Pelton wheel	1.8 to 1.9 N
Francis turbine	2.0 to 2.2 N
Kaplan turbine	2.5 to 3.0 N

Draft Tube

In the case of mixed and axial flow turbines, a large portion of the available energy still remains with the water as it leaves the runner. As this energy cannot be used in the runner, it is necessary to extract the unused energy. This can be done using a draft tube.

Draft tube is an integral part of mixed and axial flow turbines. The draft tube helps to make it possible to have the pressure at the runner outlet much below the atmospheric pressure. A draft tube serves the following two purposes:

1. It allows the turbine to be set above the tail-water level, without loss of head, and thus can be easily maintained.
2. If regains a major portion of the kinetic energy delivered to it from the runner.



$$\frac{p_2}{\rho g} = \frac{p_a}{\rho g} + (y - y_2) - \left(\frac{v_2^2 - v_3^2}{2g} - h_L\right)$$

The above equation is valid for the straight conical draft tube shown above. The term $(y - y_2)$ is called as the suction head of the drat tube. (H_S). The term $\left(\frac{v_2^2 - v_3^2}{2g}\right)$ is called the dynamic head.

$$\frac{p_2}{\rho g} = \frac{p_a}{\rho g} + H_S - \left[\frac{v_2^2 - v_3^2}{2g} - h_L\right]$$

The term $\frac{p_2}{\rho g}$ is less than atmospheric pressure

Efficiency of a draft tube (η_d)

$$\eta_d = \frac{\text{Net gain in pressure head}}{\text{velocity head at entrance of draft tube}}$$

$$\eta_d = \frac{\left[\frac{v_2^2 - v_3^2}{2g} - h_L\right]}{\frac{v_2^2}{2g}}$$

Note that v_2 is the velocity of water at the inlet of the draft tube while v_3 is the velocity of water at the outlet of the draft tube.

The most commonly used draft tube types are

1. Straight conical or concentric tube and
2. He elbow type

Specific Speed

The specific speed of a turbine is defined as the speed of a geometrically similar turbine that would develop unit power (1 kw) under unit head (1 m).

$$N_S = \frac{N\sqrt{p}}{H^{\frac{5}{4}}}$$

Where Ns is the specific speed (in rpm) of the turbine, N is the speed (in rpm) of the actual turbine, P is the shaft power in kw and H is the head, in metres, under which the turbine is working.

Example 8: In a hydroelectric station, water is supplied to the turbine to the turbine at 0.225 m^3/s and under a net head of 20 m. The turbines run at 1000 rpm and have a specific speed of 8.5 rpm. If there are 3 identical turbines in the station, then the overall efficiency of the turbines is

(A) 96.2% (B) 87.8%
(C) 74.3% (D) 81.9%

Solution:

Given $Q = 0.225\ m^3/s$

$H = 20$ m
$N = 1000$ rpm
$N_S = 85$ rpm

Shaft power developed by a turbine,

$$P_t = \frac{N_S^2}{N^2} \times H^{\frac{5}{2}}$$

$$= \left(\frac{85}{100}\right)^2 \times (20)^{\frac{5}{2}} = 12.92 \text{ kW}$$

Shaft power developed by all the turbines, $P_s = \rho g Q H \eta_0$

$\Rightarrow \quad 3 \times 12.92 \times 10^3 = 1000 \times 9.81 \times 0.225 \times 20 \times \eta_0$

$\therefore\ \eta_0 = 87.8\%$.

Unit Quantities

The unit quantities of a turbine are the unit speed (N_u), unit discharge (Q_u) and unit power (P_u)

$$N_u = \frac{N}{\sqrt{H}}$$

$$Q_u = \frac{Q}{\sqrt{H}}$$

$$P_u = \frac{P}{H^{\frac{3}{2}}}$$

Unit speed (Nu) is defined as the hypothetical speed of the turbine operating under one metre head. Similarly unit power and unit discharge can be defined.

> Geometrically, similar turbines will have the same unit characteristics under similar operating conditions

If a turbine is working under different heads, the behaviour of the turbine can be ascertained from the values of the unit quantities as follows:

$$N_u = \frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}$$

$$Q_u = \frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$$

$$P_u = \frac{P_1}{H_1^{\frac{3}{2}}} = \frac{P_2}{H_2^{\frac{3}{2}}}$$

Example 9: For a head of 120 m, a hydraulic turbine develops 1500 kW. The power developed by the turbine, when the head is reduced to 15 m, is:

(A) 64.32 kW (B) 8.28 kW
(C) 66.29 kW (D) 23.44 kW

Solution:

Given $H_1 = 120$ m
$P_1 = 1500$ kW
$H_2 = 15$ m

$$\frac{P_1}{H_1^{\frac{3}{2}}} = \frac{P_2}{H_2^{\frac{3}{2}}}$$

$$\therefore \quad P_2 = P_1 \times \left(\frac{H_2}{H_1}\right)^{\frac{3}{2}}$$

$$= 1500 \times \left(\frac{15}{120}\right)^{\frac{3}{2}}$$

$$= 66.29 \text{ kW.}$$

Model Relationship

1. Head coefficient (C_H)

$$C_H = \frac{H}{N^2 D^2} = \text{constant}$$

2. Capacity or flow coefficient (C_ϕ)

$$C_Q = \frac{Q}{ND^3} = \text{constant}$$

3. Power co-efficient (C_P)

$$C_P = \frac{P}{N^3 D^5} = \text{constant}$$

Here D refers to a linear dimension. Using the above relations, it is possible to determine the behaviour of a prototype from the test runs carried out on a geometrically similar model. It is to be noted that the model and the prototype are assumed to have the same values of speed ratio (k_u), flow ratio $\left(k_f = \frac{v_f}{\sqrt{2gH}}\right)$ and specific speed. Geometrically similar machines (i.e., a homologous series of machines) have the same values of C_H, C_Q or C_P of their combinations.

Direction for questions 10 and 11: A model that has a runner speed of 178 rpm and with a supplied head of 6 m is used to test a geometrically similar hydraulic turbine. The turbine is expected to develop 30 mW, when supplied with a head of 60 m, with a runner speed of 100 rpm.

Example 10: The power developed by the model of the supplied head is:

(A) 29.942 kw (B) 29.942 MW
(C) 29.942 W (D) 29.942×10^{-3} W

Solution:
The subscript m stands for the model and p stands for the protype.

Given $P_p = 30 \times 10^3$ kw
$H_p = 60$ m
$N_p = 100$ rpm
$N_m = 178$ rpm
$H_m = 6$ m

$$(N_s)_p = \frac{N_p \times \sqrt{P_p}}{(H_p)^{\frac{5}{4}}}$$

$$= \frac{100\sqrt{30 \times 10^3}}{(60)^{\frac{5}{4}}}$$

$$= 103.722 \text{ rpm}$$

For geometrically similar machines such as the model and prototype, it is assumed that they have the same specific speed.

$\therefore \quad (N_s)_m = (N_s)_p = 103.722\ rpm$

$$\Rightarrow \quad 103.722 = \frac{178\sqrt{p_m}}{(6)^{\frac{5}{4}}}$$

$$P_m = 29.942.$$

Example 11: The model to prototype scale ratio is

(A) 0.5 : 1 (B) 0.8236 : 1
(C) 0.432 : 1 (D) 0.177 : 1

Solution:
Note that if the model to prototype scale is, say, 1: r, then:

$$\boxed{\frac{D_m}{D_p} = \frac{1}{r}}$$

For geometrically similar machines power coefficient is constant.

$$\Rightarrow \quad \frac{p_m}{N_m^3 D_m^5} = \frac{P_p}{N_p^3 D_p^5}$$

$$\left(\frac{D_m}{D_p}\right)^5 = \frac{p_m \times N_p^3}{p_p \times N_m^3}$$

$$\therefore \quad \frac{D_m}{D_p} = \left[\left(\frac{29.942}{30 \times 10^3}\right) \times \left(\frac{100}{178}\right)^3\right]^{\frac{1}{5}}$$

$$= 0.17765.$$

Scale Effects

The equations for C_H, C_Q and C_P can be used to develop relationships between certain variable corresponding to a prototype and its model only if the model and prototype operate at identical Reynolds numbers and are exactly geometrically similar.

Unfortunately the geometric similarity between a prototype and model cannot be extended to surface roughness and hence they will have different efficiencies. This aspect is referred to as scale effect. It is generally observed that with an increase in size, a geometrically similar mixed or axial flow turbines has greater efficiency than that of the model operating under hydraulically similar conditions.

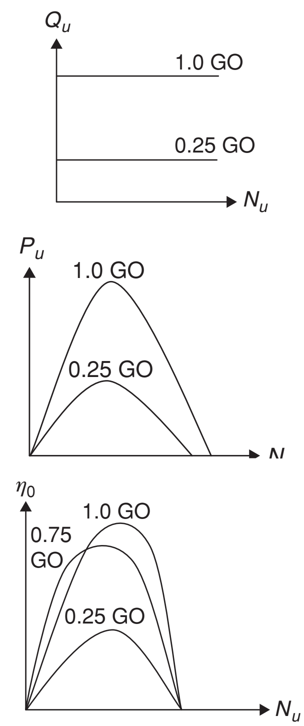
Moody's empirical formula, as given below, is generally used to account for scale effects.

$$\boxed{\frac{1-\eta_p}{1-\eta_m} = \left(\frac{D_m}{D_p}\right)^{0.2}}$$

Where η_p, η_m are the overall efficiencies of the prototype and model respectively and D_p, D_m refer to a linear dimension of the prototype and model respectively.

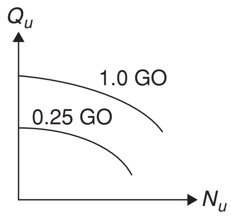
Performance Characteristics of Hydraulic Turbines

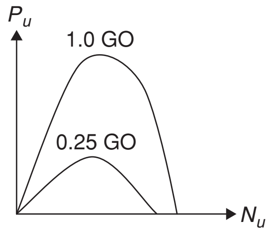
1. Main or constant head characteristic curves here head and gate opening (GO) is maintained constant.
   (i) For Pelton wheel

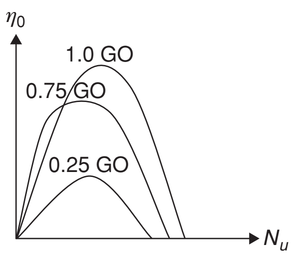


It is to be noted that for a Pelton wheel, the discharge Q_u depends only on the gate opening and not on N_u.

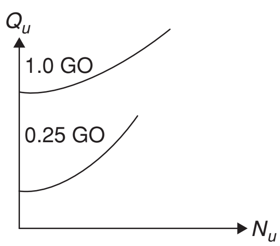
(ii) For Francis turbine







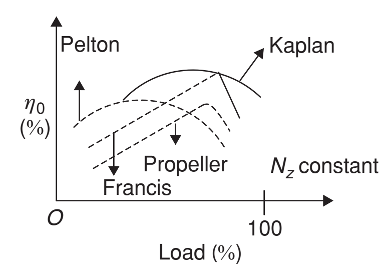
(iii) For Kaplan turbine



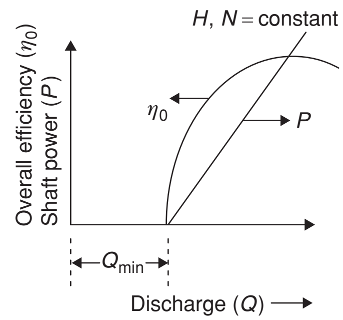
The Pa vs Nu and η_0 vs N_u curves are similar to the corresponding curves for a Francis turbine.

The maximum efficiency for a Pelton wheel occurs at the same speed for all gate openings. In the case of reaction turbines, maximum efficiency occurs at different speeds for different gate openings.

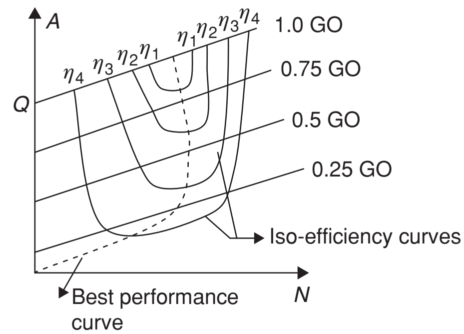
2. Operating or constant speed characteristics curves



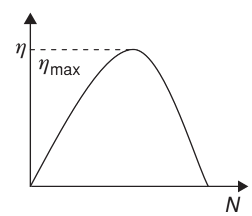
3. Overall efficiency and output power versus discharge curves



4. Constant efficiency or iso-efficiency or Muschel curves



The efficiency (η) vs speed (N) curve for a turbine is parabolic in nature, there exists two speeds for one value of efficiency, except for maximum efficiency which occurs at one speed only.



Governing of Hydraulic Turbines

Governing of a hydraulic turbine, i.e., speed regulation, is necessary as it is required to run the electric generator that is directly coupled to the turbine at a constant speed under all fluctuating load conditions. In an impulse turbine, governing is achieved by spear regulation, deflector regulation or by combined spear and deflector regulation. In reaction turbines, the governing (discharge) is achieved by varying area of flow between adjacent guide vanes.

Cavitation

In a flow field, when the pressure at any point equals the vapour pressure of the liquid at that temperature vapour cavities (bubbles of vapour) begin to appear. The cavities formed, due to liquid motion, are carried to higher pressure regions where the vapour condenses and they suddenly collapse. This formation, growth and collapse of vapour filled cavities or bubbles in a liquid flow due to decrease in liquid pressure is called cavitation.

Cavitation produces erosion of material (called pitting), noise and vibration which lead to a drop in the output and efficiency.

In reaction turbines, cavitation may occur at the runner exit or the draft tube inlet where the pressure is negative

Net positive section head (NPSH)

$$\text{NPSH} = \frac{p_e}{\rho g} + \frac{v_e^2}{2g} - \frac{p_v}{\rho g}$$

Where p_e, v_e are the static pressure and velocity of the liquid at the outlet of the runner (or at the inlet of the draft tube) and p_v is the vapour pressure of the liquid at the working temperature.

If the frictional losses in the draft tube and the velocity of the discharge from the draft tube are considered to be negligibly small, then

$$\text{NPSH} = \frac{p_{atm}}{\rho g} - \frac{p_v}{\rho g} - Hs$$

Thomas's cavitation parameter (of factor) is defined as

$$\sigma = \frac{\text{NPSH}}{H} = \frac{\frac{p_{atm}}{\rho g} - \frac{p_v}{\rho g} - Hs}{H}$$

The critical value of the cavitation parameter (or factor) is defined as

$$\sigma_c = \frac{\frac{p_{atm}}{\rho g} - \frac{p_e}{\rho g} - Hs}{H}$$

For cavitation to not occur, $\sigma > \sigma_c$ (since $p_e > p_v$)

The critical cavitation factor depends on the specific speed of the turbine.

Surge Tanks

A surge tanks is a small reservoir or tank in which the water level changes to reduce the pressure swings so that they are not transmitted in full to a closed circuit. A surge tank serves generally the following two purposes

1. to prevent water hammer effect and to protect the upstream tunnel from high pressure rises.
2. to serve as a supply or storage tank. under respectively increased or reduced load conditions.

Exercises

Practice Problems I

1. Match the following:

List-I		List-II
P. Curtis	**1.**	Pressure compounding
Q. Francis	**2.**	Outward flow reaction
R. Pelton	**3.**	Inward flow reaction
S. Rateau	**4.**	Axial flow impulse
	5.	Velocity compounding
	6.	Tangential flow impulse

(A) P–1, Q–2, R–4, S–5
(B) P–5, Q–3, R–6, S–1
(C) P–5, Q–4, R–3, S–1
(D) P–1, Q–6, R–2, S –5

2. The bucket of a Pelton wheel revolves at 1600 rpm where the difference in the water jet velocity and the bucket peripheral speed is 60 m/s. For a volumetric flow rate of the jet being 0.15 m^3/s, the power developed by the runner is 1.0616 mw. Assuming the flow to be ideal, the pitch diameter of the pelton wheel is
(A) 56.42 mm (B) 41.32 mm
(C) 716.2 mm (D) 938.67 mm

3. Water is supplied at 0.04 m^3/s through a frictionless nozzle as a jet (jet diameter is 100 mm) to a Pelton wheel with a jet ratio of 15. If the water leaves the bucket without any whirl, then the torque exerted by the water on the wheel is
(A) 100.9 Nm (B) 63.54 Nm
(C) 152.8 Nm (D) 172.6 Nm

4. The net available head to a Pelton wheel is 180 m. The velocity coefficient for the nozzle supplying the water to the Pelton wheel is 0.97 while the relative velocity of the water at the exit from the bucket is 0.895 times that at the inlet. If the bucket peripheral speed is 28 m/s, then the sum of the head loss in the nozzle and the bucket is equal to
(A) 8.91 m (B) 19.54 m
(C) 10.638 m (D) 1.728 m

5. For a Pelton wheel revolving at 770 rpm, the bucket peripheral speed is 60 m/s while the actual jet velocity at the inlet is 137.27 m/s. The Pelton wheel has to develop 15 mw under a net head of 1000 m and a jet ratio of 16. If the overall efficiency is 0.85, then the number of jets required is:
(A) 1 (B) 2
(C) 3 (D) 4

6. The straight conical draft tube fitted to a Kaplan turbine is set 5 m above the tail race level where the level of water at the tail race is 3 m above the outlet of the draft tube. The efficiency of the draft tube is determined to be 60%. For the same discharge and inlet diameter, if the water level in the tail race recedes by 1 m and the efficiency of the draft tube is improved to 72%, then the pressure at the inlet of the draft tube:
(A) Increases by 20% (B) Decreases by 20%
(C) Increases by 80% (D) Decreases by 80%

7. Under a net head of 5 m and with an overall efficiency of 88%, a Kaplan turbine develops 2.5 mW. The draft tube has an inlet diameter of 3 m and an efficiency of 75%. To prevent cavitation if the pressure head at the draft tube inlet should not drop more than 4.5 m below atmosphere, then the maximum height at which the runner may be set above the tail race level is:
(A) 1.82 m (B) 1.68 m
(C) 2.53 m (D) 1.93 m

8. Two turbines P and Q are operating with the same overall efficiency and at the discharge of 4 and 1 m^3/s. The net available head for turbine Q is 16 times that available for turbine P. If the speed of turbine P is twice the speed of turbine Q, then the ratio of their specific speeds is:
(A) 16: 1 (B) 8: 1
(C) 32: 1 (D) 1: 16

9. For a hydraulic turbine operating at two different heads, the ratio of their speeds will be 3:1. For the two heads, the ratio of the shaft power developed by the turbine will be:
(A) 9: 1 (B) 243:1
(C) 3: 1 (D) 27: 1

10. The heads available to the geometrically similar hydraulic turbines A and B are in the ratio of 1 : 9. The discharge through a jet in turbine B is nine times the discharge through a jet in turbine A. The jets are assumed to be identical in a turbine. If the number of jets in turbine B is 2, then the number of jets in turbine A is:
(A) 2 (B) 1
(C) 54 (D) 6

11. The unit speed and unit discharge associated with a turbine are given by N_u and Q_u respectively. If the overall efficiency of the turbine is 87%, then the specific speed of the turbine is equal to:
(A) $92.38 N_u \times Q_u$ (B) $92.38 N_u \times \sqrt{Q_u}$
(C) $92.8 \dfrac{N_u}{Q_u}$ (D) $92.38 N_u \times (Q_u)^2$

12. The ratio of the shaft power developed between a prototype and a model that is used to test the prototype is 4 : 1. If the head available to the prototype and the geometrically similar model is 4 :1, then the ratio of the mass flow rates between the prototype and the model is:
(A) 1 : 1 (B) 16 : 1
(C) 4 : 1 (D) 64 : 1

13. A model hydraulic turbine develops shaft power of 3.8 kw when operating under a head of 2 m and with a discharge of 0.2 m^3/s. If it is used to test a full scale hydraulic turbine (prototype model scale ratio = 20 : 1)
Taking the scale effects into account, the efficiency of the prototype would be:
(A) 98.26% (B) 77.47%
(C) 96.84% (D) 98.5%

14. A Francis turbine develops 15 mw under a head of 50 m with its runner being 2.925 m above the tail race level. The turbine operates with a cavitation factor that is 10% more than its critical value (σ_c) given by the relation: $\sigma_c = 6.55 \times 10^{-6} \times (N_s)^2$, where N_s is the specific speed. If the mean atmospheric pressure head is 10 metres of water and the vapour pressure head is 0.2 metres of water, then the turbine runs at:
(A) 100 rpm (B) 150 rpm
(C) 200 rpm (D) 250 rpm

15. In a hydroelectric plant, turbines are to be designed to develop 10 mw while operating under a head of 40 m and angular velocity of 55 rpm. The turbine that would be best suited for this purpose is:
(A) Kaplan (B) Propeller
(C) Francis (D) Pelton

16. The net head available to a Francis turbine, having a flow ratio of 0.18, is 20 m. The guide blades of the turbine make an angle of 15° to the tangent of the wheel. If the runner blades of the turbine at the inlet are radial, then the hydraulic efficiency of the turbine is
(A) 82.9% (B) 98.2%
(C) 64.36% (D) 90.16%

17. A Francis turbine has a discharge of 0.47 m^3/s and a constant velocity of flow of 6 m/s through the runner. The ratio of the wheel width to the wheel diameter at the inlet is 0.1. Assume blade thickness to be negligible. If the diameter of the wheel at the outlet is 0.25 m, then the ratio of the wheel width at the outlet to the wheel width at the inlet is
(A) 0.25 : 1 (B) 1 : 0.75
(C) 0.5 : 1 (D) 2 : 1

18. An inward flow reaction turbine, that has the same radial velocity of flow at the inlet and exit, develops a power of 670 W. The mass flow rate is 1.2 kg/s while the velocity of whirl at the inlet is 25 m/s. The tangential velocity at the outlet is 0.5 times the tangential velocity at the inlet. Water enters the runner making an angle of 15° to the direction of motion of the blades at inlet. If the water leaves radially, then the angle between the relative velocity of water and tangential velocity of the runner at exit is
(A) 31° (B) 15°
(C) 62° (D) 45°

19. A Kaplan turbine develops 5 mW under a net head of 8 m. The tangential velocity of the runner is 18 m/s while the flow velocity is 8.6 m/s. The hub diameter of the runner is 0.38 times the outside diameter. If the speed of the turbine is 100 rpm, then the turbine's overall efficiency is
(A) 86.27% (B) 98.21%
(C) 93.16% (D) 89.63%

20. A Kaplan turbine has a hydraulic efficiency of 87% and works under a head of 22 m. The flow velocity is 12 m/s. At the outlet, the whirl is zero. If the guide blade angle at the extreme edge of the runner is 36°, then the outlet vane angle of the runner is

(A) 32.13° (B) 54.33°
(C) 15.26° (D) 46.56°

Practice Problems 2

Direction for questions 1 to 30: Select the correct alternative from the given choices.

1. Match the following:

List-I		List-II	
P.	Kaplan	**1.**	Steam turbine
Q.	Parsons	**2.**	inward flow reaction
R.	Fournegron	**3.**	outward flow reaction
S.	Francis	**4.**	gas turbine

(A) P–3, Q–4, R–3, S–3
(B) P–3, Q–1, R–3, S–2
(C) P–2, Q–1, R–3, S–2
(D) P–2, Q–4, R–4, S–2

2. Match the following:

List-I		List-II	
P.	Certis	**1.**	prepare compounding
Q.	Kaplan	**2.**	tangential flow
R.	Ratean	**3.**	low head
S.	Pelton	**4.**	mixed flow
		5.	velocity compounding

(A) P–5, Q–2, R–1, S–3
(B) P–5, Q–3, R–1, S–2
(C) P–1, Q–4, R–5, S–3
(D) P–1, Q–3, R–5, S–4

3. The buckets of a Pelton wheel revolve on a mean diameter of 1.5 m and 1500 $\frac{\text{rev}}{\text{min}}$. The water jet velocity is 188.48 m/W with the jet deflection angle being 120°. Assuming the flow to be ideal, the volumetric flow rate of water required to develop 1.5 mW is:
(A) 0.12 m^3/s (B) 0.075 m^3/s
(C) 0.36 m^3/s (D) 0.096 m^3/s

4. For the standard design of a desired Pelton wheel, it is observed that if the jet ratio is increased by five times, then the number of buckets on the runner has to be doubled. If the diameter of the jet is 0.15 m, then the pitch diameter of the desired pelton wheel is:
(A) 0.3 m (B) 3 m
(C) 0.75 m (D) 1.5 m

5. For a Pelton wheel, with a bucket peripheral velocity of 22.24 m/s, the relative velocity of water at the exit from the bucket is 0.9 times the relative velocity at the inlet. The side clearance angle is 20° and the velocity coefficient is 0.98. If the prepare behind the nozzle is 1030 KN/m^2, then the hydraulic efficiency of the turbine is:
(a) 92.28% (B) 79.56%
(C) 76.45% (C) 96.98%

6. A Pelton wheel nozzle, with a velocity coefficient C_v, is h metres below the water surface of a lake. The friction factor of the penstock (pipeline of length L and diameter D) is given as f. If d is the jet diameter and m is the jet ratio, then the actual velocity of the jet is equal to:

(A) $\sqrt{2C_v^2 gh}$ (B) $\sqrt{\dfrac{2gh}{\left(\dfrac{1}{C_v^2}+\dfrac{fLd^4}{D^5}\right)}}$

(C) $\sqrt{\dfrac{2gh}{\left(\dfrac{1}{C_v^2+\dfrac{fL}{mD}}\right)}}$ (D) $\sqrt{\dfrac{2gh}{\left(\dfrac{1}{C_v^2+\dfrac{fD}{mL}}\right)}}$

7. For the two straight conical draft tubes A and B, the head loss due to friction in the tube is equal to 0.4 times the velocity head at outlet. If the inlet to outlet diameter ratio for draft tube A is 0.56 while for draft tube B is 0.71, then which one of the following statements is only correct for the same discharge through the tubes?
(A) Draft tube B is more efficient
(B) Draft tube A is more efficient
(C) Both the draft tubes have the same efficiency
(D) Efficiency of the draft tubes cannot be determined.

8. The overall efficiency of a Kaplan turbine that develops 2 MW under a head of 6 m is 80%. A straight conical draft tube fitted to the turbine has a diameter of 2.5 m at the inlet and is set 2 m above the tail race level. If the draft tube efficiency is 70% and head loss due to friction is negligible in the tube, then a vacuum gauge connected to the inlet of the draft tube would indicate a reading of (in metres of water)
(A) 2 (B) 2.671
(C) 4.671 (D) 2.17

9. Water is available at the same volumetric flow rate and under the same amount of head to the turbines of the hydroelectric stations A and B. The ratio of the actual running speeds and the ratio of the specific speeds of the turbines in the station B to those in station A are 6 : 1 and 2 : 1 respectively. If there are two turbines in station A, then the number of turbines in station B would be:
(A) 18 (B) 6
(C) 9 (D) 288

10. For a hydraulic turbine working of two different head, the ratio of the discharges through the turbine is determined to be $1:\sqrt{7}$. For the two heads, the ratio of the shaft powers developed by the turbine will be

(A) $1:\sqrt{7}$ (B) $1:49$

(C) $1:7$ (D) $1:7\sqrt{7}$

11. An hydraulic turbine is working under a head of 20 m. If the head available to the turbine in increased by 700%, then the speed of the turbine will

(A) Increase by 700%

(B) Increase by 2162.7%

(C) Decrease by 64.6%

(D) Increase by 182.8%

12. A hydraulic turbine is to be total with the help of a geometrically similar 1 : 5 model. If the ratio of the heads available between the prototype and the model is 25 : 1, then the ratio of the mass flow rates between the prototype and modal is

(A) 125 : 1 (B) 625 : 1

(C) 5 : 1 (D) 25 : 1

13. The ratio of the heads available between a prototype and the model, that is used to test it, is 4 : 1. If the model –prototype scale ratio is 1 : 3, then the ratio of the shaft power available between the prototype and the geometrically similar modal is

(A) 72 : 1 (B) 576 : 1

(C) 12 : 1 (D) 144 : 1

14. The bullet peripheral speeds of a hydraulic turbine and a geometrically similar model are u_p and u_m respectively. The model is to be used to test the behaviour of the turbine. If the model and the prototype run at the same speeds, then the ratio of the shaft power available between the prototype and model is equal to:

(A) $\left(\frac{u_p}{u_m}\right)^4$ (B) $\left(\frac{u_p}{u_m}\right)^5$

(C) $\left(\frac{u_p}{u_m}\right)^3$ (D) $\left(\frac{u_p}{u_m}\right)$

15. In order to build a new turbine for a new dam site, it is decided to geometrically scale up an existing hydraulic turbine that operates under a head of 100 m and at a runner speed of 200 rpm. The new turbine is expected to operate at a head of 400 m and runner speed of 200 rpm. If the density of the water at the new dam site is only 0.993 times the density of water feel to the existing turbine, then the ratio of the shaft powers developed between the now and old turbine is:

(A) 32 : 1 (B) 7.94 : 1

(B) 1016.83 : 1 (D) 31.78 : 1

16. An engineer was a model hydraulic turbine of scale 1 : 10 and overall efficiency of 82% to test a geometrically similar full scale hydraulic turbine. He reports that the full scale hydraulic turbine would deliver a shaft power of 15 MW. On further realization that scale effect had to be taken into account, he corrects his reported shaft power value to:

(A) 13.88 MW (B) 18.29 MW

(C) 16.21 MW (D) 16.92 MW

17. The head available to both the Francis turbines *A* and *B* is 80 m. The mean atmospheric pressure is 101.043 kPa and the vapour pressure for water is 2.943 kPa. The height of the runner of turbine *A* above the tail water level is 0.5 m and for turbine *B* it is 0.6 m. If the critical cavitation factor is 0.1144, then which one of the following statements is only correct?

(A) Cavitation does not occur in the turbines

(B) Cavitation occurs both the turbines

(C) Cavitation occurs only in turbine *A*

(D) Cavitation occurs only in turbine *B*

18. Two dam sites I and II require turbines that develop 10 MW under a head of 40 m. But dam site I required turbines that run at 52 rpm while dam site II requires turbines that run at 9 rpm Based on your recommendations, the best suitable turbines for dam sites I and II would respectively be:

(A) Francis turbines

(B) Pelton turbines

(C) A Pelton wheel and Francis turbine

(D) A Francis turbine and Pelton wheel

19. You are observing, inside a hydroelectric dam installation, turbine that have an overall efficiency of 85% and that non at 2 revolution per second (rps). The available head and discharge for each turbine are 130 m and 0.9225 m^3/s respectively. If the best suitable turbines were installed, then you would be looking at:

(A) Kaplan turbines (B) Pelton wheels

(C) Francis turbines (D) Propeller turbines.

20. In an inward flow reaction turbine, the runner vanes at the inlet and the discharge at the outlet are radial. The velocity of flow through the runner is constant. The hydraulic efficiency of the turbine, which has as available net head of 15 m, is 98%. If the absolute velocity of water at outlet is 5 m/s, then the absolute velocity of water at inlet is:

(A) 12 m/s (B) 10.9 m/s

(C) 13.93 m/s (D) 13 m/s

21. For a Francis turbine, the ratio of the tangential velocities at the inlet to the outlet is 2.5 : 1 with the flow velocity through the runner being constant. Assume blade thickness to be negligible If, at the inlet the wheel width is 0.2 times the wheel diameter, then at the outlet, the ratio of the wheel width to wheel diameter is:

(A) 0.8 : 1 (B) 2.5 : 1

(C) 0.2 : 1 (D) 1.25 : 1

22. The angle between the relative velocity of water and the tangential velocity of the runner at the exit in a Francis turbine is 30°. The radial velocity of flow at exit is 6 m/s while the whirl velocity at the exit and inlet are 0 m/s and 28 m/s respectively. If the ratio of the wheel diameter at the inlet to that at the outlet are 2 and the man flow rate is 1.5 kg/s, then the power developed by the turbine is:
(A) 582 W (B) 873 W
(C) 291 W (D) 194 W

23. A Francis turbine is supplied with water at a volumetric flow rate of 120 m^3/s through a penstock of 5 m diameter. The turbine develops a shaft power of 70 MW. The static pressure head in the penstock measured just before entry to the runner is 58 m, with the point of measurement being 5 m above the tail race level. If the overall efficiency of the turbine is 95.81%, then the velocity of water in the tail race is:
(A) 6.11 m/s (B) 9.6 m/s
(C) 7.5 m/s (D) 4.3 m/s

24. A Kaplan turbine develops 10 MW when working under a head of 7 m. The tangential velocity of the runner is 20 m/s while the flow velocity is 8 m/s. The hub diameter of the runner is 0.36 times the outside diameter. If the overall efficiency of the turbine is 87%, then the speed of the turbine (in rpm) is
(A) 69 (B) 34
(C) 140 (D) 93

25. A Kaplan turbine with an overall efficiency of 86% works under a head of 20 m. The flow velocity is 11 m/s. If the guide blade angle at the entremic edge of the runner is 35°, then the inlet vane angle of the runner is:
(A) 77.69° (B) 45.69°
(C) 54.69° (D) 65.69°

26. If W denotes the velocity of fluid relative to the rotor and the subscripts 1 and 2 refer to the inlet and outlet respectively, then for an axial flow impulse turbo machine the ratio $\frac{W_2}{W_1}$ will be equal to:
(A) 0.5 (B) 2
(C) 1 (D) 0

27. For a reaction turbine, the guide blade angle at the inlet is 12°. For this turbine, the degree of reaction is zero when the vane angle at the inlet is:
(A) 7° (B) 23°
(C) 12° (D) 0°

28. The structural modifications of a Pelton wheel have increased the water jet velocity at the inlet by 100% and the speed ratio by 10%. The modifications would have then led to an increase of the bucket peripheral velocity by:
(A) 10% (B) 110%
(C) 100% (D) 120%

29. For a pelton wheel operating at maximum power, the angle of deflection is 160° and the blade friction coefficient is 0.48. The absolute value of the ratio of the whirl velocity at the inlet to the whirl velocity at the outlet is:
(A) 1.38 (B) 0.275
(C) 0.726 (D) 3.64

30. The velocity of flow of a Francis turbine is constant and equal to one-fourth the tangential velocity of the blade at the inlet. The blade tip angle at the inlet is 90°. If at the outlet, the flow velocity is equal to the absolute velocity, then the blade efficiency is:
(A) 96.97% (B) 76.32%
(C) 82.48% (D) 91.85%

Previous Years' Questions

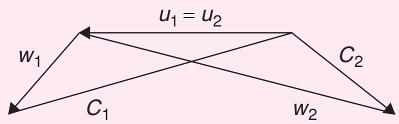
1. At a hydro electric power plant site, available head and flow rate are 24.5 m and 10.1 m^3/s respectively. If the turbine to be installed is required to run at 4.0 revolution per second (rps) with an overall efficiency of 90%, the suitable type of turbine for this site is: **[2004]**
(A) Francis (B) Kaplan
(C) Pelton (D) Propeller

2. Match the following: **[2004]**

List-I		List-II	
P.	Reciprocating pump	1.	Plant with Power output below 100 kW
Q.	Axial flow pump	2.	Plant with Power output Between 100 kW to 1 MW
R.	Microhydel plant	3.	Positive Displacement
S.	Backward curved vanes	4.	Draft tube
		5.	High flow rate, low pressure ratio
		6.	Centrifugal pump impeller

(A) P–3, Q–5, R–6, S–2
(B) P-3, Q–5, R–2, S–6
(C) P–3, Q–5, R–1, S–6
(D) P–4, Q–5, R–1, S–6

3. In the following velocity diagram, u = blade velocity, C = absolute fluid velocity and W = relative velocity of fluid and the subscripts 1and 2 refer to inlet and outlet. This diagram is for: **[2005]**



(A) An impulse turbine
(B) A reaction turbine
(C) A centrifugal compressor
(D) An axial flow compressor

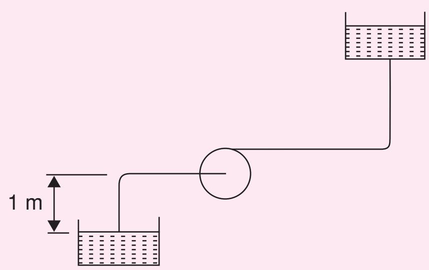
4. In a Pelton wheel, the bucket peripheral speed is 10 m/s, the water jet velocity is 25 m/s and volumetric flow rate of the jet is 0.1 m^3/s. If the jet deflection angle is 120° and the flow is ideal, the power developed is: **[2006]**

(A) 7.5 kW (B) 15.0 kW
(C) 22.5 kW (D) 37.5 kW

5. A large hydraulic turbine is to generate 300 kW at 1000 rpm under a head of 40 m. For initial testing, a 1: 4 scale model of the turbine operates under a head of 10 m.The power generated by the model (in kW) will be: **[2006]**

(A) 2.34 (B) 4.68
(C) 9.38 (D) 18.75

6. A horizontal shaft centrifugal pump lifts water at 65°C. The suction nozzle is one meter below pump centerline. The pressure at this point equals 200 kPa gauge and velocity is 3 m/s. Steam tables show saturation pressure at 65°C is 25 kPa, and specific volume of the saturated liquid is 0.001020 m^3/kg. The pump net positive suction head (NPSH) in meters is: **[2006]**



(A) 24 (B) 26
(C) 28 (D) 30

7. The inlet angle of runner blades of a Francis turbine is 90°. The blades are so shaped that the tangential component of velocity at blade outlet is zero. The flow velocity remains constant throughout the blade passage and is equal to half of the blade velocity at runner inlet. The blade efficiency of the runner is: **[2007]**

(A) 25% (B) 50%
(C) 80% (D) 89%

8. A model of a hydraulic turbine is tested at a head of 1/4th of that under which the full scale turbine works. The diameter of the model is half of that of the full scale turbine. If N is the rpm of the full scale turbine, then the rpm of the model will be: **[2007]**

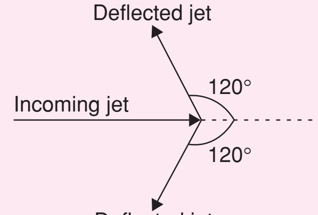
(A) N/4 (B) N/2
(C) N (D) 2 N

9. Match the items in columns I and II and choose the correct answer. **[2007]**

	Column I		Column II
(P)	Centrifugal compressor	(1)	Axial flow
(Q)	Centrifugal pump	(2)	Surging
(R)	Pelton wheel	(3)	Priming
(S)	Kaplan turbine	(4)	Pure impulse

(A) P–2, Q–3, R–4, S–1
(B) P–2, Q–3, R–1, S–4
(C) P–3, Q–4, R–1, S–2
(D) P–1, Q–2, R–3, S–4

10. Water, having a density of 1000 kg/m^3, issues from a nozzle with a velocity of 10 m/s and the jet strikes a bucket mounted on a Pelton wheel. The wheel rotates at 10 rad/s. The mean diameter of the wheel is 1 m. The jet is split into two equal streams by the bucket, such that each stream is deflected by 120°, as shown in the figure. Friction in the bucket may be neglected. Magnitude of the torque exerted by the water on the wheel, per unit mass flow rate of the incoming jet, is:

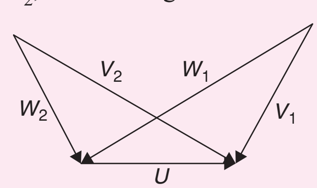


(A) 0 (N.m)/(kg/s) (B) 1.25 (N.m)/(kg/s)
(C) 2.5 (N.m)/(kg/s) (D) 3.75 (N.m)/(kg/s)

11. A hydraulic turbine develops 1000 kW power for a head of 40 m. If the head is reduced to 20 m, the power developed (in kW) is: **[2010]**

(A) 177 (B) 354
(C) 500 (D) 707

12. The velocity triangles at the inlet and exit of the rotor of a turbo machine are shown. V denotes the absolute velocity of the fluid, W denotes the relative velocity of the fluid and U denotes the blade velocity. Subscripts 1 and 2 refer to inlet and outlet respectively. If $V_2 = W_1$ and $V_1 = W_2$, then the degree of reaction is: **[2012]**



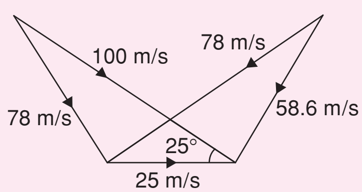
(A) 0 (B) 1
(C) 0.5 (D) 0.25

13. In order to have maximum power from a Pelton turbine, the bucket speed must be: **[2013]**
(A) Equal to the jet speed
(B) Equal to half of the jet speed
(C) Equal to twice the jet speed
(D) Independent of the jet speed

14. An ideal water jet with volume flow rate of 0.05 m^3/s strikes a flat plate placed normal to its path and exerts a force of 1000 N. Considering the density of water as 1000 kg/m^3, the diameter (in mm) of the water jet is ____ **[2014]**

15. Steam at a velocity of 10 m/s enters the impulse turbine stage with symmetrical blading having blade angle 30°. The enthalpy drop in the stage is 100 kJ. The nozzle angle is 20°. The maximum blade efficiency (in percent) is _____ **[2014]**

16. At the inlet of an axial impulse turbine rotor, the blade linear speed is 25 m/s, the magnitude of absolute velocity is 100 m/s and the angle between them is 25°. The relative velocity and the axial component of velocity remain the same between the inlet and outlet of the blades. The blade inlet and outlet velocity triangles are shown in the figure. Assuming no losses, the specific work (in J/kg) is ______. **[2014]**

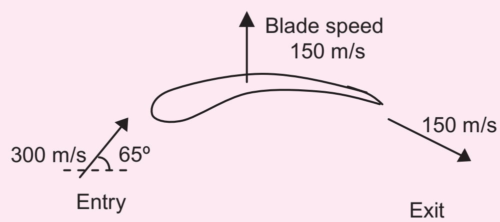


17. Kaplan water turbine is commonly used when the flow through its runner is: **[2014]**
(A) Axial and the head available is more than 100 m
(B) Axial and the head available is less than 10 m
(C) Radial and the head available is more than 100 m
(D) Mixed and the head available is about 50 m

18. Which of the following statements are TRUE, when the cavitation parameter $\sigma = 0$? **[2015]**
(i) The local pressure is reduced to vapor pressure
(ii) Cavitation starts
(iii) Boiling of liquid starts
(iv) Cavitation stops
(A) (i), (ii) and (iv) (B) only (ii) and (iii)
(C) only (i) and (iii) (D) (i), (ii) and (iii)

19. Consider two hydraulic turbines having identical specific speed and effective head at the inlet. If the speed ratio (N_1/N_2) of the two turbines is 2, then the respectively power ratio (P_1/P_2) is ________. **[2016]**

20. The blade and fluid velocities for an axial turbine are as shown in the figure.



The magnitude of absolute velocity at entry is 300 m/s at an angle of 65° to the axial direction, while the magnitude of the absolute velocity at exit is 150 m/s. The exit velocity vector has a component in the downward direction. Given that the axial (horizontal) velocity is the same at entry and exit, the specific work (in kJ/kg) is ______. **[2016]**

Answer Keys

Exercises

Practice Problems 1

1. B	**2.** C	**3.** C	**4.** B	**5.** B	**6.** A	**7.** D	**8.** D	**9.** D	**10.** D
11. B	**12.** A	**13.** A	**14.** B	**15.** C	**16.** D	**17.** D	**18.** A	**19.** C	**20.** D

Practice Problems 2

1. C	**2.** B	**3.** A	**4.** D	**5.** A	**6.** B	**7.** B	**8.** C	**9.** A	**10.** D
11. D	**12.** A	**13.** A	**14.** B	**15.** D	**16.** C	**17.** A	**18.** D	**19.** B	**20.** D
21. D	**22.** B	**23.** C	**24.** A	**25.** D	**26.** C	**27.** B	**28.** D	**29.** D	**30.** A

Previous Years' Questions

1. A	**2.** B	**3.** B	**4.** C	**5.** A	**6.** A	**7.** C	**8.** C	**9.** A	**10.** D
11. B	**12.** C	**13.** B	**14.** 56 to 57		**15.** 85.1 to 89.9		**16.** 3250 to 3300		**17.** B
18. D	**19.** 0.25	**20.** 50–54							

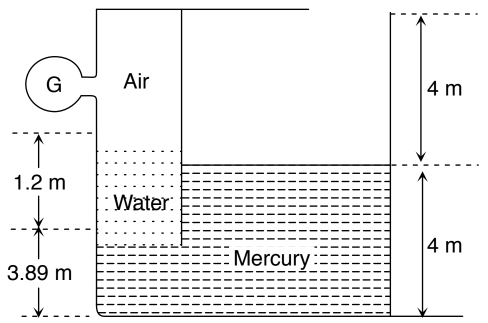
Test

Fluid Mechanics

Time: 60 Minutes

Direction for questions 1 to 30: Select the correct alternative from the given choices.

1. A partitioned tank contains water (specific gravity = 1) and mercury (specific gravity = 13.6) in equilibrium as shown below. The reading (in kPa) of the pressure guage (*G*) attached on the left chamber will approximately be:
   (A) 11.7 (B) 2.9
   (C) 104.2 (D) 90.6



2. Two identical rectangular surfaces *ABCD* and *PQRS* are submerged in water such that the horizontal surface *ABCD* lies at the centroid level of the surface *PQRS* which is inclined at an angle of 45° to the free surf ace of water. The hydrostatic forces acting on the surfaces *ABCD* and *PQRS* are in the ratio of:
   (A) $\sqrt{2}:1$ (B) $1:\sqrt{2}$
   (C) $1:2$ (D) $1:1$

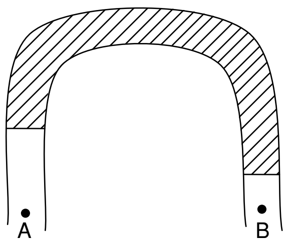
3. If the viscosity of air 200°K is 1.3289×10^{-5} Kg/ms, then the viscosity of air 350°K will probably (in Kg/ms) be:
   (A) 0.6925×10^{-5} (B) 1.3289×10^{-5}
   (C) 0.8274×10^{-5} (D) 2.075×10^{-5}

4. Two ideal gases, having the ratio of their densities as *m*: 1 and the ratio of their temperatures as 1:*r*, will have the ratio of their bulk modulus of compressibility as:
   (A) *m*:*r* (B) *r*:*m*
   (C) 1:*r* (D) 1:*rm*

5. The velocity potential satisfies Laplace's equation for:
   (A) A compressible rotational flow
   (B) A compressible irrotational flow
   (C) An incompressible irrotational flow
   (D) An incompressible rotational flow

6. Two identical metallic cubes *A* and *B* are lying immersed in a lake, such that cube *B* is immersed a little deeper. If F_A and F_B are the buoyant forces acting on the cubes *A* and *B* respectively, then:
   (A) $F_A > F_B$ (B) $F_A = F_B$
   (C) $F_B > F_A$ (D) $F_A = F_B = 0$

7. The pressure differential between the points *A* and *B*, lying on the same horizontal plane, at the walls of a duct flow is 588.6 Pa. If the manometric reading of the inverted U–tube manometer fitted at these points, as shown in the following figure, is 0.3 m, then the difference between the density of the liquid flowing in the duct and the density of the manometric liquid is:
   (A) 1962 Kg/m^3 (B) 86 Kg/m^3
   (C) 200 Kg/m^3 (D) 60 Kg/m^3



8. A rectangular tank of height *h* meters is filled with a liquid upto a height of h_1 metres ($h_1 < h$). The perpendicular distance between the free surface of the liquid and the point at which the total pressure force acts on a vertical wall of the tank is:
   (A) $\frac{h_1}{3}$ (B) $\frac{2h_1}{3}$
   (C) $\frac{h_1}{2}$ (D) $\frac{h}{2}$

9. Points *M*, *G* and *B* represent the metacentre, centre of gravity and centre of buoyancy of a floating body respectively. The body can be said to be stable or unstable if:
   (A) *M* lies above *G*. (B) *M* lies below *G*.
   (C) *M* lies below *B*. (D) *M* lies above *B*.

10. The velocity field of a steady, three dimensional flow is given by:
    $\vec{V} = 6(y+z)\hat{i} + 7y^2\hat{j} + (a+6-14y)z\hat{k}$. If the flow field represents an incompressible flow field, then the value of *a* is:
    (A) –6 (B) 14*y* – 6
    (C) 8 (D) 6

11. A cylindrical buoy of weight 187.7 KN and height 10 m is floating vertically in sea water (density = 1015 Kg/m^3)

such that only 4 m of its height is exposed to the atmosphere. If a load of 62563 N is placed on the top of the buoy and the cylinder still floats vertically, then the height of the cylinder exposed to the atmosphere will be:

(A) 2 m (B) 3 m
(C) 0.5 m (D) 1 m

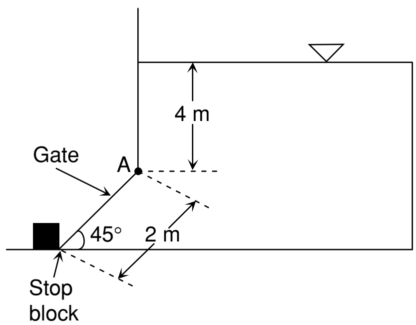
12. A two dimensional steady incompressible flow in the *xy* plane has the velocity field given by:

$\vec{V} = \frac{A}{x}\hat{i} + \frac{y}{x^2}\hat{j}$. If a streamline in this flow field has the equation $y = 5x$, then the time required by a fluid particle to move from $x = 1$ m to $x = 3$ m in this flow field is:

(A) 0.2 seconds (B) 2 seconds
(C) 4 seconds (D) 20 seconds

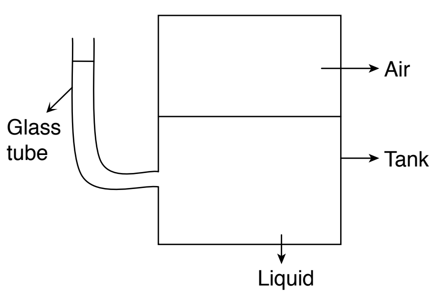
13. A rectangular gate of height 2 m and width 4 m is held in place by a stop block that exerts only a horizontal force F_B Newtons. The gate, on whose right a water body is present, is hinged at *A* as shown in the following figure. If the weight of the gate is considered to be negligible, then the magnitude of force F_B (in Newtons) is:

(A) 24242.8 (B) 34284.5
(C) 32651.9 (D) 48485.6



14. A glass tube, as shown in the following figure, is used to determine the pressure exerted by the air trapped in a tank containing a liquid of density 1200 Kg/m^3 and surface tension 0.082 N/M. Unaccounting for the surface tension corrections, the air guage pressure is estimated to be equivalent to a liquid height of 22 cm in the glass tube of diameter 1mm. If the contact angle of the liquid in the glass tube is 0°, then the true guage pressure of the air is:

(A) 2589.84 Pa (B) 1830.2 Pa
(C) 2917.84 Pa (D) 2261.84 Pa



15. Loads are present on the tops of the large and small pistons of a hydraulic lift such that the pistons are at the same levels. The pistons still remain at the same levels if the mass on the smaller piston is reduced by half and an additional force of 1000 Newtons is applied on the piston. If the ratio of the diameters of the pistons having negligible weights is 3; 1, then the mass on the larger piston is:

(A) 611.62 Kg (B) 1834.86 Kg
(C) 22.65 Kg (D) 917.43 Kg

16. The pressures inside a spherical water bubble and a spherical water droplet, both exposed to the same atmospheric pressure, are the same. If the radius of the water droplet is *r*, then the radius of the water bubble will be:

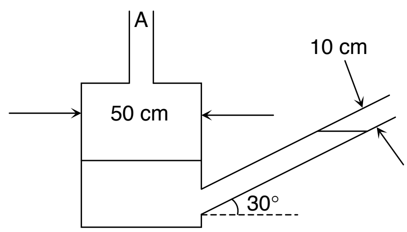
(A) *r* (B) 0.5*r*
(C) 2*r* (D) 4*r*

17. For a converging nozzle, the fluid flow through which is approximated by a one dimensional velocity distribution, the fluid velocity varies from 1 m/s at the entrance to 5 m/s at the exit. If the acceleration at the midpoint of the nozzle is 0.8 m/s^2, then the length of the nozzle is:

(A) 15 m (B) 5 m
(C) 12 m (D) 4 m

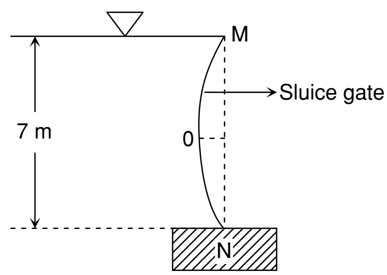
18. The pressure changes at a point *A* in a gas flow pipe are measured with a mercury filled inclined leg manometer as shown in the following figure. If the mercury has moved a distance of 15 cm in the inclined leg, then the guage pressure at point *A* is:

(A) 10006.2 Pa (B) 10806.7 Pa
(C) 9205.7 Pa (D) 20012.4 Pa



19. A sluice gate MN has a water body to its left as shown in the figure below. If the resultant hydrostatic force acting on the gate makes an angle of 8.13° with the horizontal, then the magnitude of the resultant hydrostatic force, per unit length of the gate into the plane of the paper is:

(A) 34335 N (B) 240345 N
(C) 242785 N (D) 274680 N



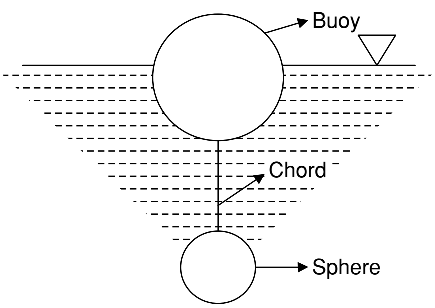
20. The velocity potential of a two dimensional flow is $\varphi = 2xy + y^2 - x^2$. If C is used to denote any integration constant, then a stream function for this flow would be:
(A) $-y^2 - 2xy + x^2 + C$
(B) $-y^2 - 2xy - x^2 + C$
(C) $y^2 - 2xy - x^2 + C$
(D) $y^2 + 2xy + x^2 + C$

21. A double edged blade is placed on the surface of a water body (surface tension = 7.3×10^{-2} N/m). The total length of the sides of the blade is 0.35 m while its mass is 3×10^{-3} Kg. If the surface tension forces acting on the blade make an angle θ with the water surface, then which one of the following statements is ONLY correct?
(A) Blade will float if $\theta = 30°$
(B) Blade will float if $\theta = 45°$
(C) Blade will float if $\theta = 60°$
(D) Blade will never float

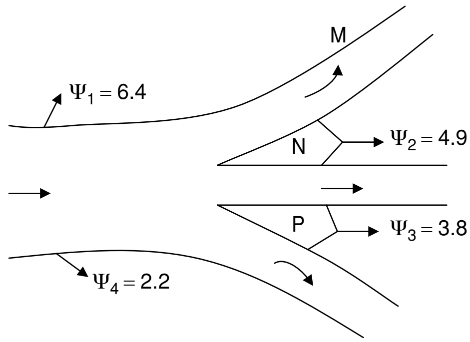
22. For a startified stationary body of liquid, the liquid density ρ varies with the depth h as $\rho = \rho_0 (1 + h)$. If P denotes the pressure at a depth h and $P_0 = \rho_0 hg$, then the differnce in pressures, i.e., $P - P_0$, for any depth h is equal to:
(A) $\rho_0 h^2 g$ (B) $\rho_0 h^2 g/2$
(C) $\rho_0 h^2 g/4$ (D) $\rho_0 h^2 g/8$

23. A large upper plate moving at 3 m/s is seperated paralleley from a stationary large lower plate by a Newtoniam fluid layer of 3 cm thickness. If a linear velocity profile is assumed for the fluid layer, the shear stress on the upper plate is 90 N/m^2. If a parabolic velocity distribution is assumed for the fluid layer, then the shear stress acting on the upper plate would be:
(A) 45 N/m^2 (B) 180 N/m^2
(C) 67.5 N/m^2 (D) 90 N/m^2

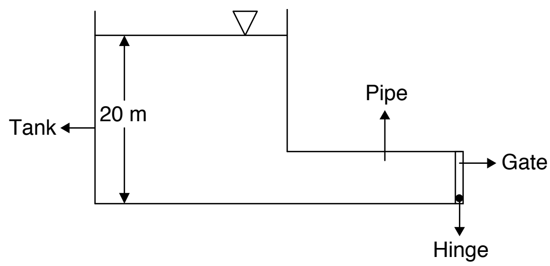
24. A buoy (density = 500 Kg/m^3) has a volume of 2 m^3, out of which 1.05 m^3 volume is submerged in a body of liquid. The buoy is attached by a flexible chord to a metalllic sphere (density = 3000 Kg/m^3, volume = 0.15 m^3). That is fully immersed in the liquid as shown in the following figure. If the buoy and the sphere are in equilibrium, then the density of the liquid is:
(A) 674.42 Kg/m^3 (B) 1611.11 Kg/m^3
(C) 783.78 Kg/m^3 (D) 1208.33 Kg/m^3



25. A main duct branches into three smaller ducts as shown in the following figure. The values of the following function (Q) on the ducts are displayed in the figure. The ratio of the discharge in duct M to the discharge in duct P is.
(A) 32:19 (B) 113:60
(C) 15:16 (D) 49:22



26. A vertical circular gate of 7 m diameter and hinged at the bottom is present at the end of a circular pipe attached to a large open tank containing water as shown in the figure. If the depth of the water in the tank is 20 m, then the minimum horizontal force that is to be applied at the centre of the gate to hold the gate closed is.
(A) 161865 N
(B) 763078 N
(C) 153078 N
(D) 771865 N



Direction for questions 27 and 28: A shaft of diameter 600 mm is rotating at 42 rad/s inside a bearing of length 200 mm. The gap between the bearing and the shaft is filled by 2 mm thick layer of a Newtonian oil of viscosity 0.7 $P_a.s$.

27. The torque necessary to overcome the viscous resistance is:
(A) 498.75 Nm (B) 11083.54 Nm
(C) 1764 Nm (D) 2770.88 Nm

28. The power necessarily employed to overcome the viscous resistance is:
(A) 116.38 Kw (B) 465.51 Kw
(C) 74.09 Kw (D) 20.95 Kw

Direction for questions 29 and 30: A solid cone, made up of a material with a specific gravity of 0.85, has a diameter of 40 cm and a height of 35 cm. The cone floats with its vertex vertically downwards in water such that 10 cm of its height is only above the free surface of water. The cone has a semi vertex angle of 30°.

29. The distance between the centre of buoyancy and the centre of gravity is:
(A) 75 mm (B) 100 mm
(C) 50 mm (D) 25 mm

30. If the equilibrium condition of the cone is attempted to be determined, then it:
(A) Could be stable, unstable or neutral.
(B) Is table
(C) Is unstable
(D) Is neutral.

Answer Keys

1. B	**2.** D	**3.** D	**4.** A	**5.** C	**6.** B	**7.** C	**8.** B	**9.** D	**10.** A
11. A	**12.** C	**13.** B	**14.** D	**15.** B	**16.** C	**17.** A	**18.** B	**19.** C	**20.** C
21. D	**22.** B	**23.** A	**24.** D	**25.** C	**26.** B	**27.** A	**28.** D	**29.** A	**30.** C