Chapter 8

Turbo Machinery

CHAPTER HIGHLIGHTS

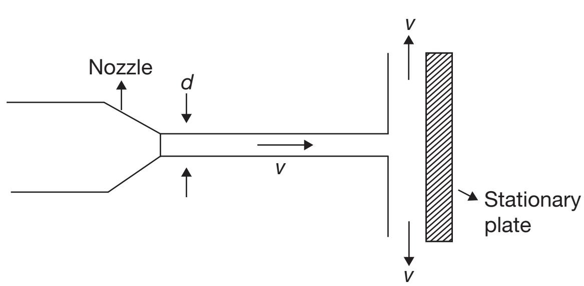
- *Impact of free jets*
- *Hydraulic turbines*
- *Pelton wheel*
- *Reaction turbines*
- *Unit quantities*
- *Performance characteristics of hydraulic turbines*
- *Governing of hydraulic turbines*
- *Pumps*

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:

1. A flat plate held normal to a jet:



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

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.

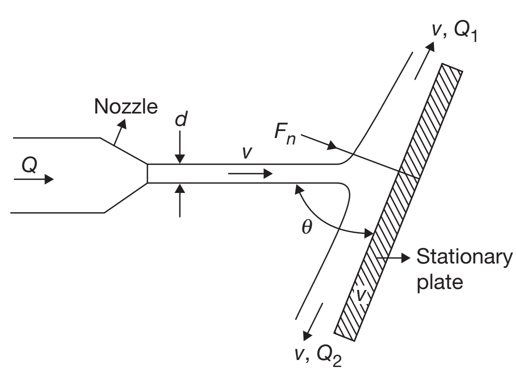
The force exerted in the x-direction by the jet is the change in momentum,

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

$$= \rho a v^2 \text{ N.}$$

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

2. 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,

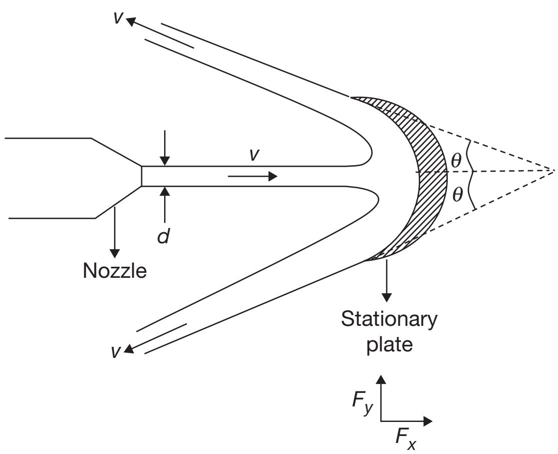
$$\boxed{\begin{aligned}F_n &= \rho a v^2 \sin\theta \\ Q &= Q_1 + Q_2\end{aligned}}$$

Ratio of discharges, $\dfrac{Q_1}{Q_2} = \dfrac{1+\cos\theta}{1-\cos\theta}$

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

Force exerted by a jet on a stationary curved plate:

1. **Jet strikes the curved plate at the centre:**



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

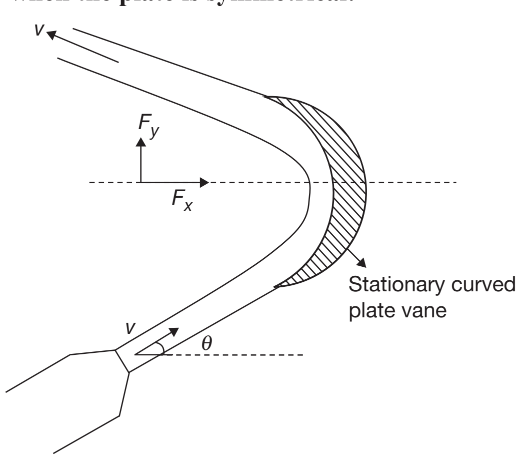
$$\boxed{F_x = \rho a v^2 (1 + \cos\theta).}$$

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).

2. **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.

$$\boxed{\begin{aligned}F_x &= 2\rho a v^2 \cos\theta \\ F_y &= 0\end{aligned}}$$

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

3. **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.

$$\boxed{\begin{aligned}F_x &= \rho a v^2 (\cos\theta + \cos\phi) \\ F_y &= \rho a v^2 (\sin\theta - \sin\phi)\end{aligned}}$$

Force exerted by a jet on a moving flat plate:

1. **A moving flat plate held normal to a jet:**

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

Work done per second on the plate,

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

2. **A moving flat plate held inclined to a jet:**

$$\boxed{\begin{aligned}F_x &= \rho (v-u)^2 \sin\theta \\ W &= \rho a\,(v-u)^2 \sin\theta \times u\end{aligned}}$$

Force exerted on a moving curved plate or vane:

1. **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,

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

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

2. **Series of vanes:**

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

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., $u = \dfrac{v}{2}$.

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

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

Jet striking a moving curved vane tangentially at one tip and leaving other:

1. **Single vane:** Let, v_1, v_2 = Jet's absolute velocities at inlet and outlet.

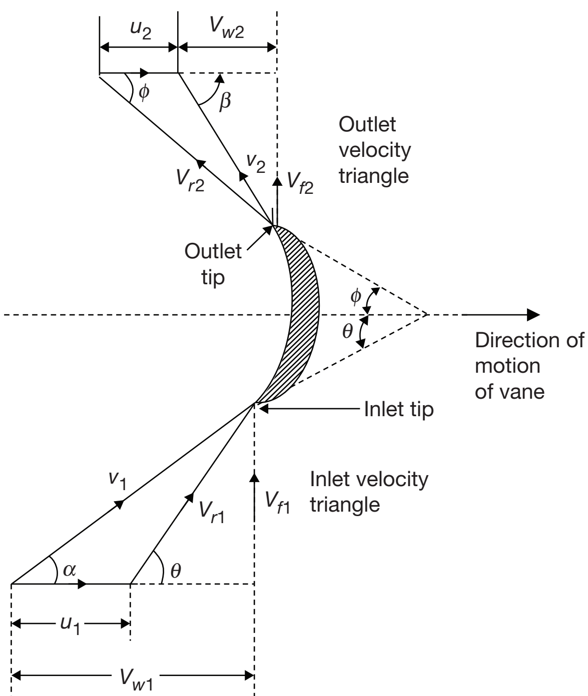
u_1, u_2 = Vane's peripheral velocities at inlet and outlet (tangential velocity of vane). V_{r_1}, V_{r_2} = Relative velocities of the jet at inlet and outlet with respect to the vane. V_{f_1}, V_{f_2} = Flow velocities at inlet and outlet.

V_{W_1}, V_{W_2} = Velocity of whirl at input and output.

θ, ϕ = Tip angles of inlet and outlet (ϕ is also called as vane angle at outlet).

α, β = 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.



$$\boxed{F_x = \rho a v_{r_1}\left(v_{w_1} \pm v_{w_2}\right).}$$

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

$$\boxed{W = \rho a \;\; v_{r_1}\left(v_{\omega_1 \pm v_{\omega_2}}\right) \times u.}$$

Work done per second per unit weight of fluid striking,

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

2. **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.

$$\therefore \qquad u_1 = \omega R_1$$
$$u_2 = \omega R_2$$

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,

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

Work done per second on the wheel,

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

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

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

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 as follows:

1. **According to the action of water on the turbine blades:**

   (a) 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.

   (b) 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:**
   (a) **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.
   Example: Petron wheel
   (b) **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.
   Example: Francis turbine.
   (c) **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.
   Example: Kaplan turbine.
3. **According to the direction of how through the runner:**
   (a) **Tangential flow turbine:** In this type of turbine, water flows tangentially to the runner.
   Example: Pelton turbine
   (b) **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 radically from outwards to inwards.
   Example: Old Francis turbine
   Outward radial flow turbine: Here water flows radically from inwards to outwards.
   Example: Fourneyron turbine
   (c) **Axial flow turbine:** In this type of turbine, water flows parallely to the axis of rotation of the runner.
   Example: Kaplan turbine.
   (d) **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.
   Example: Modern Francis turbines.
4. **According to specific speed:**
   (a) **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).
   Example: Pelton wheel.
   (b) **Medium specific speed turbines:** The specific speed varies from 50 to 250 for these turbines.
   Example: Francis turbine.
   (c) **High specific speed turbine:** In these turbines, the specific speed is more than 250.
   Example: Kaplan turbine.

Pelton Wheel

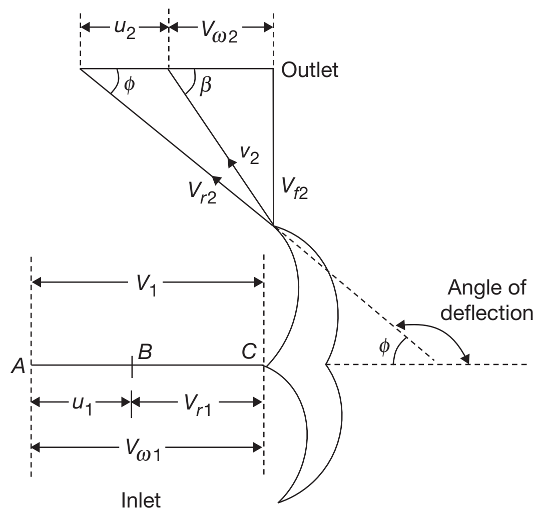
The pelton wheel (or pelton turbine) is a tangential flow impulse turbine. Water from a reservoir flows through penstocks at the outlet of which a nozzle is fitted. The nozzle increases the kinetic energy of the water flowing through the penstocks. At the outlet of the nozzle, water comes out in the 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:



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

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_{r1} 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	$\beta < 90°$	$v_{\omega 2}$ is negative
Medium runner	$\beta = 90°$	$v_{\omega 2}$ is zero
Fast runner	$\beta > 90°$	$v_{\omega 2}$ is positive

Blade Friction Coefficient (*k*)

$$\boxed{k = \frac{v_{r1}}{v_{r2}}}$$

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

Work done, Power, Force, Torque

At inlet.

$$\boxed{\begin{aligned} U_{r1} &= v_1 - u_1 = v_1 - u \\ v_{\omega 1} &= v_1 \end{aligned}}$$

At outlet.

$$\boxed{v_{\omega 2} = v_{r2}\cos\phi - u_2 = v_{r2}\cos\phi - u}$$

Work done per unit weight of water striking,

$$\boxed{W = \frac{1}{g}\left(v_{\omega 1} + v_{\omega_2}\right)u}$$

Kinetic energy of jet per second.

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

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,

$$\boxed{\begin{aligned} P_r &= \rho Q(v_{\omega 1} + v_{\omega 2})\text{u} \\ &= \rho Q(v_1 - u)\,(1 + k\cos\phi)u \end{aligned}}$$

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

$$\boxed{\begin{aligned} F &= \rho Q(v_{\omega 1} + v_{\omega 2}) \\ &= \rho Q(v_1 - u)\,(1 + k\cos\phi). \end{aligned}}$$

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.

$$\boxed{\begin{aligned} \tau &= \frac{1}{2}\rho DQ\left(v_1 - u\right) + \left(1 + k\cos\varphi\right) \\ &= \frac{1}{2}\rho QD\left(v_{\omega 1} + v_{\omega 2}\right). \end{aligned}}$$

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

$$\boxed{H = H_g - h_L - h}$$

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*).

1. **Hydraulic efficiency (η_h):**

$$\boxed{\eta_h = \frac{P_r}{P_i} = \frac{P_r}{\rho g Q H}}$$

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

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

$$= \frac{\rho Q g\left(v_{\omega 1} + v_{\omega 2}\right)\times u}{g\frac{1}{2}\left(\rho a v_1\right)v_1^2}$$

$$\boxed{\begin{aligned} \eta_h &= \frac{2\left(v_{\omega 1} + v_{\omega 2}\right)\times u}{v_1^2} \\ &= \frac{2\left(v_1 - u\right)\left(1 + k\cos\varphi\right)u}{v_1^2}. \end{aligned}}$$

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

$$\boxed{\begin{aligned} H_r &= \frac{1}{g}(v\omega_1 + v\omega_2)u \\ H - H_r &= \Delta H \end{aligned}}$$

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,

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

2. **Mechanical efficiency (η_m):**

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

3. **Overall efficiency (η_0):**

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

4. **Volumetric efficiency (η_v):**

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

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

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

$$\boxed{\eta_0 = \eta_h \times \eta_m \times \eta_v}$$

else,

$$\boxed{\eta_0 = \eta_h \times \eta_m}$$

Design Aspects

1. **Velocity of jet at inlet:**

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

Where, C_v is the *coefficient of velocity* = 0.98 or 0.99.

2. **Velocity of wheel:**

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

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**: It is to be taken as 165° if not given.

4. **Mean diameter or pitch diameter:** D of the pelton wheel is given by,

$$\boxed{D = \frac{60u}{\neq N}}$$

5. **Jet ratio** (m) is given by,

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

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,

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

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.

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

Where a is the area of the jet.

The power will be maximum when $u = \frac{v_1}{2}$ i.e., $P_{max} = \dfrac{\rho a(1 + k\cos\varphi)}{4} v_1^3$

$$P_{max} = CV_1^3$$

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

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.

Hence, the correct answer is option (C).

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 $k_u = 0.46$

$$C_v = 0.98$$
$$K = 0.98$$
$$\phi = 180° - 165° = 15°$$

Overall efficiency $= \eta_0 = \dfrac{P_s}{P_i}$

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

$$= \frac{\eta_m \times \rho Q(v_1 - u)(1 + k\cos\varphi)u}{\rho QHg}$$

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

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

$$u = k_u\sqrt{2gH}\ ; \quad \therefore \frac{v_1}{u} = \frac{C_v}{k_u}$$

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

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

$$(1 + 0.98 \times (\cos 15°))$$

$= 88.5\%$.

Hence, the correct answer is option (D).

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 $\dfrac{k_1(v_1 - u)^2}{2g}$ (k_1 being a constant) occurs due to the bucket friction and shock, then the maximum efficiency of the pelton wheel occurs when $\dfrac{u}{v_1}$ 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,

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

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

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

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

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

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

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

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

For the efficiency to be maximum,

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

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

or efficiency is maximum when,

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

Hence, the correct answer is option (B).

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

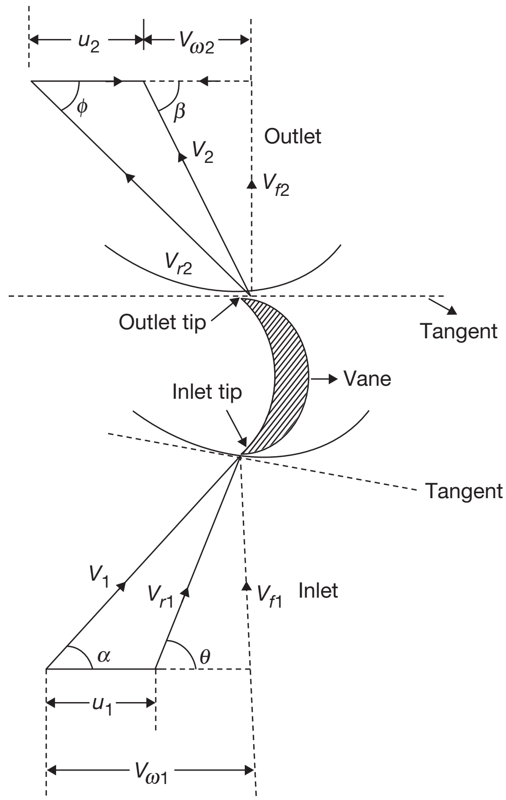
The main parts of a Francis turbine are:

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 exit 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,

$$W_{\max} = \rho Q\,(v_{\omega 1}\, u_1)$$

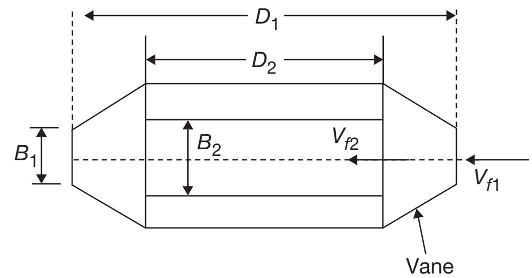
The discharge in this case is radial, i.e., absolute velocity at exit is radial.

Hydraulic Efficiency (η_h)

$$\eta_h = \frac{v_{\omega 1}u_1 \pm v_{\omega 2}u_2}{gH}$$

Working Proportions

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



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

   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

$$k_f = \frac{v_{f1}}{\sqrt{2gH}}$$

   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 Spects of a Francis Turbine Runner

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

$$A = k_{t1}\,\pi D_1 B_1$$

   Where, k_{t1} is called the *vane* thickness factor/coefficient. K_{t1} always has a value less than unity.

2. Discharge,

$$Q = k_{t1}\,\pi D_1 B_1 v_{f1}$$
$$Q = k_{t2}\,p D_2 B_2 v_{f2}$$

3. Guide vane angle (α),

$$\tan\alpha = \frac{v_{f1}}{v_{\omega 1}}$$

4. Runner vane angle (θ),

$$\tan\theta = \frac{v_{f1}}{v_{\omega 1} - u_1}$$

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

5. Runner vane angle at exit (ϕ),

$$\tan\phi = \frac{v_{f2}}{u_2}$$

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),

$$u_1 = \frac{\neq D_1 N}{60}, \quad u_2 = \frac{\neq D_2 N}{60}$$

Where

θ, ϕ = Vane angle at inlet and outlet

B_1, B_2 = Width of the runner vane at inlet and outlet

D_1, D_2 = Diameter of the runner (wheel diameter) at the inlet and outlet.

8. The blade efficiency is given by,

$$\eta_b = \frac{2v_{f_1}^2 \cot\alpha\left(\cot\alpha - \cot\beta\right)}{v_{f2}^2 + 2v_{f1}^2 \cot\alpha\left(\cot\alpha - \omega t\beta\right)}$$

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 4.6 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, $P_w = 83.5 \times 10^6$ 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

$$H = \frac{P}{\rho g} + \frac{v^2}{2g} + z - \frac{v_t^2}{2g}$$

$$= 59 + \frac{6^2}{2\times 9.81} + 4.6 - \frac{8^2}{2\times 9.81}$$

$$= 62.173 \text{ m}$$

$$\eta_0 = \frac{P_\omega}{\rho g Q H}$$

$$= \frac{83.5\times 10^6}{1000\times 9.81\times \frac{\pi}{4}\times 5.5^2\times 6\times 62.173}$$

$$= 96.04\%.$$

Hence, the correct answer is option (C).

Example 5

From the guide vanes of an inward radial flow turbine, which has an available net head of 15 m, 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, where α is inlet nozzle angle.

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

(C) $\left(\frac{2gH}{2+\tan\left(\alpha^2\right)}\right)^{\frac{1}{2}}$ (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, $\alpha = 12°$

$$H = 15 \text{ m}$$
$$\theta = 90°$$

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, $v_{w2} = 0$ and $v_2 = v_{f2} = u_1\tan\alpha$

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

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

Hence, the correct answer is option (A).

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,

$$k_{t2} = \left(1 - \frac{5}{100}\right) = 0.95$$

$$Q = k_t \pi D B v_f$$

$$v_f = kf\sqrt{2gH}$$

$$\therefore \quad Q = k_t \, \pi DBkg\sqrt{2gH}$$

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

$$\therefore \quad k_{t2}\sqrt{H}_2 = k_{t1}\sqrt{H_1}$$

$$\text{Or} \quad H_2 = \left(\frac{k_{t1}}{kt_2}\right)^2 \times H_1$$

$$= \left(\frac{0.9025}{0.95}\right)^2 \times 15$$

$$= 13.5375.$$

Hence, the correct answer is option (C).

Axial Flow Reaction Turbines—Kaplan and Propeller Turbines

The shaft of such turbines are 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 this 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,

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

Where, D_0 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.,

$$u_1 = u_2 = u = \frac{\pi D_0 N}{60}$$

3. Velocity of flow at inlet and outlet are equal, i.e., $vf_1 = v_{f2} = v_f$
4. Area of flow at inlet = Area of flow at outlet = $\frac{\pi}{4}\left(D_o^2 - D_b^2\right)$
5. The discharge Q flowing through the runner is given by,

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

7. The flow ratio,

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

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

8. 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,

$$H = \frac{1}{2g}\left[\left(v_1^2 - v_2^2\right) + \left(u_1^2 - u_2^2\right) + \left(v_{r_2}^2 - v_{r_1}^2\right)\right]$$

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 while 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.

$$R = \frac{\frac{1}{2g}\left[\left(u_1^2 - u_2^2\right) + \left(v_{r2}^2 - v_{r1}^2\right)\right]}{H}$$

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

For a reaction turbine,

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

Example 7

The degree of reaction will be zero for a

(A) Francis turbine (B) Kaplan turbine
(C) Pelton wheel (D) Propeller turbine

Solution

The correct answer is option (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:

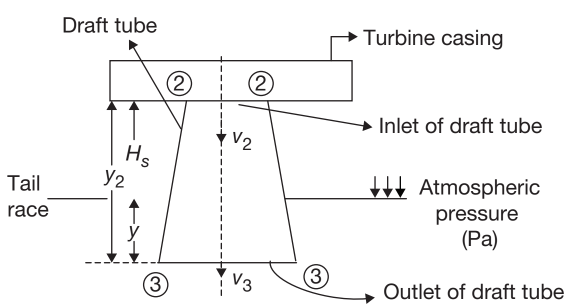
Pelton wheel	1.8–1.9 N
Francis turbine	2.0–2.2 N
Kaplan turbine	2.5–3.0 N

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. the 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).

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

Where, N_s 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 85 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,

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

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

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

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

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

Hence, the correct answer is option (B).

Unit Quantities

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

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

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

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

Unit speed (N_u) 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:

$$N_u = \frac{N_1}{\sqrt{H_1}} = \frac{N_2}{\sqrt{H_2}}, \; Q_u = \frac{Q_1}{\sqrt{H_1}} = \frac{Q_2}{\sqrt{H_2}}$$

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

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

$$H_1 = 120 \text{ m}$$
$$P_1 = 1500 \text{ kW}$$
$$H_2 = 15 \text{ m}$$

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

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

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

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

Hence, the correct answer is option (C).

Model Relationship

1. **Head coefficient (C_H):**

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

2. **Capacity or flow coefficient (C_f):**

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

3. **Power coefficient (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 solved examples 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

$$P_p = 30 \times 10^3 \text{ kW}$$
$$H_p = 60 \text{ m}$$
$$N_p = 100 \text{ rpm}$$
$$N_m = 178 \text{ rpm}$$
$$H_m = 6 \text{ m}$$

$$(N_s)_p = \frac{N_p \times \sqrt{P_\rho}}{(H\;)^{\frac{5}{4}}}$$

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

$$= 103.722 \text{ rpm}$$

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

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

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

$$P_m = 29.942.$$

Hence, the correct answer is option (A).

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.

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

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

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

$$= 0.17765.$$

Hence, the correct answer is option (D).

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.

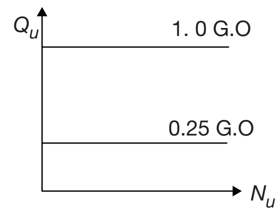
$$\frac{1-\eta_p}{1-\eta_m} = \left(\frac{D_m}{D_p}\right)^{0.2}$$

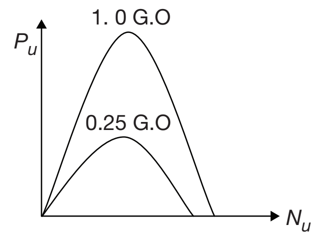
Where, η_p, η_m are the overall efficiencies of the prototype and model respectively and D_p, D_m refer to a linear dimension of the prototype and model respectively.

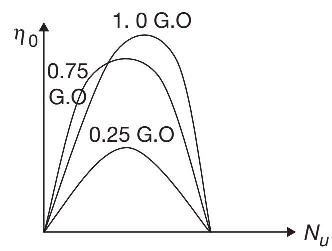
Performance Characteristics of Hydraulic Turbines

1. Main or constant head characteristic curves. Here head and gate opening (GO) are maintained constant.

(a) For pelton wheel,

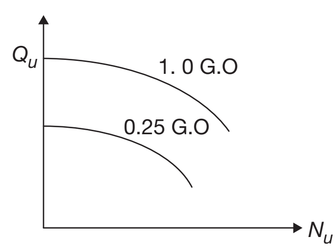


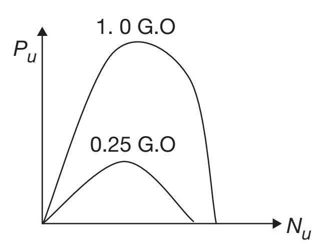


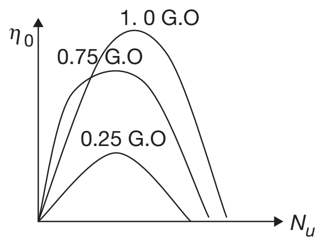


It is to be noted that for a pelton wheel, the discharge Q_u depends only on the gate opening and not on N_u.

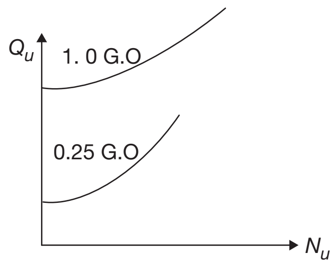
(b) For Francis turbine,







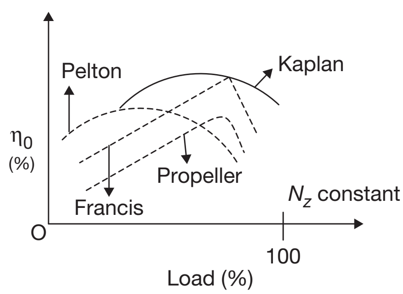
(c) For Kaplan turbine,



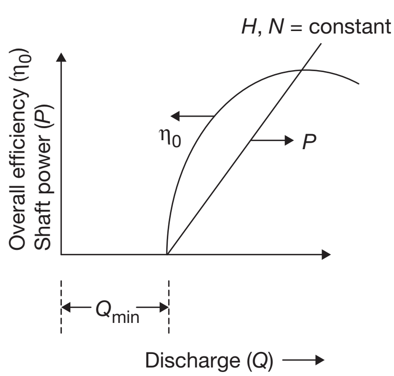
The P_u vs N_u and η_0 vs N_u curves are similar to the corresponding curves for a Francis turbine.

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.

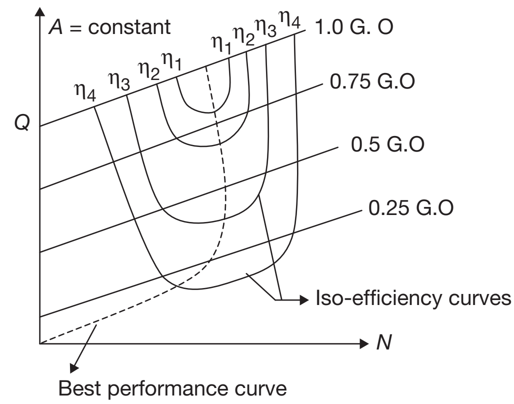
1. **Operating or constant speed characteristics curves:**



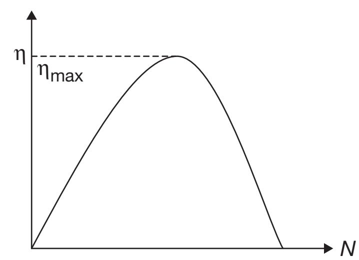
2. **Overall efficiency and output power versus discharge curves:**



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

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

Thoma's caviation parameter (of factor) is defined as,

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

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.

Pumps

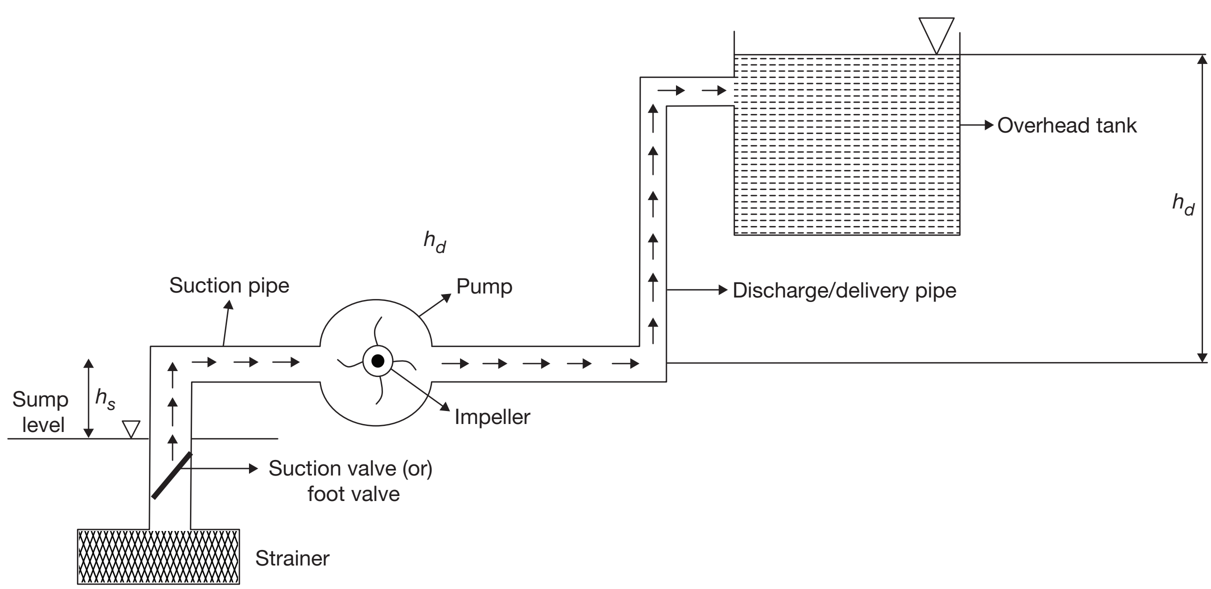
Pump is a device which is used to transfer mechanical energy of motor into pressure and kinetic energy of water.

Centrifugal Pump

Centrifugal pumps are classified as rotodynamic type of pumps in which a dynamic pressure is developed which enables the lifting of liquids from a lower level to a higher level.

The basic principle on which a centrifugal pump works is that, when a certain mass of liquid is made to rotate by an external force, it is thrown away from the central axis of rotation and a centrifugal head is impressed which enables it to rise to a higher level.

Now if more liquid is constantly made available at the centre of rotation, a continuous supply of liquid at a higher level may be ensured.



General layout of the pump

Exercises

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

(A) francis. (B) kaplan.
(C) pelton. (D) propeller.

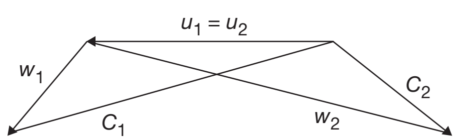
2. Match the following:

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

Codes:

(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 velocity diagram shown below, 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



(A) an impulse turbine.
(B) a reaction turbine.
(C) a centrifugal compressor.
(D) an axial flow compressor.

4. The gross head available to a hydraulic power plant is 100 m. The utilized head in the runner of the hydraulic turbine is 72 m. If the hydraulic efficiency of the turbine is 90%, the pipe friction head is estimated to be

(A) 20 m (B) 18 m
(C) 16.2 m (D) 1.8 m

5. Consider the following statements regarding the specific speeds of a centrifugal pump:

I. Specific speed is defined as the speed of a geometrically similar pump developing unit power under unit head.
II. At the same specific speed, the efficiency is greater with larger capacity.
III. The specific speed increases with increase in outer blade angle.
IV. The specific speed varies directly as the square root of the pump discharge.

Which of these statements are correct?

(A) I and II (B) II and IV
(C) III and IV (D) II and III

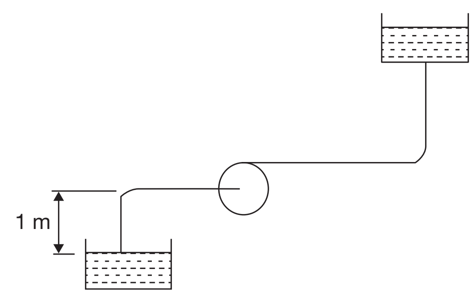
6. In a pelton wheel, the bucket peripheral speed is 10 m/s, the water jet velocity is 25 m/s and volumetric flow rate of the jet is 0.1 m^3/s. If the jet deflection angle is 120° and the flow is ideal, the power developed is:

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

7. 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

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

8. A horizontal shaft centrifugal pump lifts water at 65°C. The suction nozzle is one metre 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 metres is:



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

9. Which of the following purposes are served by the volute casing of a centrifugal pump?

I. Increase in the efficiency of the pump.
II. Conversion of part of the pressure head to velocity head.
III. Giving uniform flow of the fluid coming out of the impeller.

Select the correct answer using the codes given below:

(A) I and II (B) I and III
(C) II and III (D) I, II and III

10. Consider the following pumps:

I. Centrifugal pump, single-stage
II. Centrifugal pump, multi-stage

III. Reciprocating pump
IV. Jet pump

The pump(s) which can be used to lift water through a suction head of 12 m from a well would include

(A) II Only (B) I, III and IV
(C) IV Only (D) I and III

11. A pump is installed at a height of 5 m above the water level in the sump. Frictional loss on the suction side is 0.6 m. If the atmospheric pressure is 10.3 m of water and vapor pressure head is 0.4 m (abs), the NPSH (net positive suction head) will be

(A) 3.7 m (B) 4 m
(C) 4.3 m (D) 4.6 m

12. Water is required to be lifted by a 10 kW pump from a depth of 100 m. If the pump is unable to lift the water, then which one of the following is correct?

(A) A greater capacity pump has to be used.
(B) A larger diameter delivery pipe has to be used.
(C) A larger diameter suction pipe has to be used.
(D) A multistage pump has to be used.

13. Each of the next consists of two statements, one labelled as the **'Assertion (A)'** and other as **'Reason (R)'**. You are to examine these two statements carefully and select the answers to these items using the codes given below:

Assertion (A): The volute casing of a centrifugal pump helps in creating the high velocity head necessary for enabling water flow upwards to a higher level.

Reason (R): The water flows through a diverging passage in the volute chamber.

(A) Both A and R are individually true and R is the correct explanation of A.
(B) Both A and R are individually true but R is not the correct explanation of A.
(C) A is true but R is false.
(D) A is false but R is true.

14. A centrifugal pump with radial vane tips at the outlet has an impeller of 100 mm outer diameter. If the retentive speed is 3000 rpm and manometric efficiency 0.8, then what is the net head developed?

(A) 10 m (B) 20 m
(C) 30 m (D) 40 m

15. A mixed flow pump is driven by a 8 kW motor running at 1000 rpm. It delivers water at the rate of 1000 litres/min against a total head of 25 m. What is the specific speed of the pump in metre-minutes?

(A) 90 (B) 50
(C) 45 (D) 75

16. 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%.

17. Air vessels are used in reciprocating pumps in order to

(A) increase the delivery head.
(B) reduce suction head.
(C) minimize delivery head fluctuation.
(D) reduce accelerating head.

18. A centrifugal pump is fully primed, but on starting it fails to deliver fluid. The probable reasons are listed below:

I. Leaky foot valve or suction line.
II. Suction head is very low.
III. Insufficient motor speed.
IV. Damaged or closed delivery valve.

Which of these reasons are correct?

(A) I, II, III and IV
(B) I, II and III only
(C) II, III and IV only
(D) I, III and IV only

19. An impulse turbine

(A) always operates submerged.
(B) makes use of draft tube.
(C) operates by initial complete conversion to kinematic energy.
(D) converts pressure head into velocity head throughout the vanes.

20. A hydraulic turbine develops a power of 10^4 metric horse power while running at a speed of 100 revolutions per minute, under a head of 40 m. Its specific speed is nearest to

(A) 100 (B) 628
(C) 523 (D) 314

21. A hydraulic turbine has a discharge of 5 m^3/s, when operating under a head of 20 m with a speed of 500 rpm. If it is to operate under a head of 15 m, for the same discharge, the rotational speed in rpm will approximately be

(A) 433 (B) 403
(C) 627 (D) 388

22. Water is to be lifted by a net head of 150 m. Identical pumps each with specific speed of 30 and rotational speed of 1450 rpm with design discharge of 0.2 m^3/s are available. The minimum number of pumps required is ______.

23. The expression for the specific speed of a pump

(A) does not include the diameter of the impeller.
(B) yield larger values for radial pumps than for axial flow pump.

(C) is necessarily non-dimensional.
(D) includes power as one of the variables.

24. At a rated capacity of 44 cumecs, a centrifugal pump develops 36 m of head when operating at 1450 rpm. Its specific speed is
(A) 654 (B) 509
(C) 700 (D) 90

25. If the pump head is 75 m, discharge is 0.464 m^3/s and the motor speed is 1440 rpm at rated condition, the specific speed of the pump is about
(A) 4 (B) 26
(C) 38 (D) 1440

26. A pump can lift water at a discharge of 0.15 m^3/s to a head of 25 m. The critical cavitation number (σ_c) for the pump is found to be 0.144. The pump is to be installed at a location where the barometric pressure is 9.8 m of water and the vapor pressure of water is 0.30 m of water. The intake pipe friction loss is 0.40 m. Using the minimum value of NPSH (net positive suction head), the maximum allowable elevation above the sump water surface at which the pump can be located is
(A) 9.80 m (B) 6.20 m
(C) 5.50 m (D) None of these

27. The allowable net positive suction head (NPSH) for a pump provided by the manufacturer for a flow of 0.05 m^3/s is 3.3 m. The temperature of water is 30°C (vapour pressure head absolute = 0.44 m), atmospheric pressure is 100 kPa absolute and the head loss from the reservoir to pump is 0.3 N-m/N. The maximum height of the pump above the suction reservoir is
(A) 10.19 m (B) 6.89 m
(C) 6.15 m (D) 2.86 m

28. Identify the FALSE statement from the following. The specific speed of the pump increases with
(A) increase in shaft speed.
(B) increase in discharge.
(C) decrease in gravitational acceleration.
(D) increase in head.

Direction for questions 29 and 30:

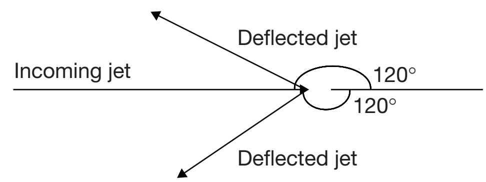
A propeller turbine is to develop 6250 kW under a head of 5 m, having given that speed ratio 'k_u' based on outer diameter = 2.10, flow ratio ψ = 0.65, diameter of boss = 0.35 times external diameter of the runner and overall efficiency is 85%.

29. The diameter of the runner in 'm' is
(A) 5.81 (B) 4.91
(C) 5.21 (D) 6.35

30. The speed of the turbine in rpm is
(A) 78.27 (B) 68.37
(C) 58.35 (D) 48.22

31. Draft tube is a pipe used in
(A) reaction turbine for discharge and it has gradually decreasing cross-sectional area.
(B) reaction turbine for discharge and it has gradually increasing cross-sectional area.
(C) Impulse turbine for discharge and it has gradually decreasing cross-sectional area.
(D) Impulse turbine for discharge and it has gradually increasing cross-sectional area.

32. 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 (Nm)/(Kg/s) (B) 1.25 (Nm)/(Kg/s)
(C) 2.5 (Nm)/(Kg/s) (D) 3.75 (Nm)/(Kg/s)

Direction for questions 33 and 34:

A pelton wheel has to be designed for the following data. Power to be developed = 6000 kW, net head available = 300 m, speed = 550 rpm, ratio of jet diameter to wheel diameter = $\frac{1}{10}$ and overall efficiency = 85%.

33. Find the number of jets required
(A) 1 (B) 2
(C) 3 (D) 4

34. The diameter of jet is in mm
(A) 135.2 (B) 116.4
(C) 141.2 (D) 186.4

35. The maximum hydraulic efficiency for a pelton wheel is given by the expression (θ is the vane angle of outlet).
(A) $\frac{1-\cos\theta}{2}$ (B) $\frac{1+\cos\theta}{2}$
(C) $\frac{\cos\theta}{2}$ (D) None of these

36. Incase of pelton turbine installed in a hydraulic power plant the gross head available is the vertical distance between
(A) forebay and tailrace.
(B) reservoir level and turbine inlet.
(C) forebay and turbine inlet.
(D) reservior level and tail race.

37. A turbine operates under a head of 30 m at 200 rpm with a discharge of 10 m^3/s and efficiency 90%. Speed of the turbine (in rpm) under a head of 20 m will be
(A) 178.9 (B) 163.3
(C) 152.4 (D) 146.5

38. In a hydroelectric power station water is available at a rate of 170 m^3/s under a head of 20 m. Turbines run at 150 rpm with 80% overall efficiency. If maximum specific speed is 460, power available at a turbine shaft is
(A) 12624 kW
(B) 16823 kW
(C) 20162 kW
(D) 22346 kW

Previous Years' Questions

1. The inlet angle of runner blades of a Francis turbine is 90º. 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 **[GATE, 2007]**
(A) 25% (B) 50%
(C) 80% (D) 89%

2. A model of a hydraulic turbine is tested at a head of 1/4 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 **[GATE, 2007]**
(A) $n/4$ (B) $n/2$
(C) n (D) $2n$

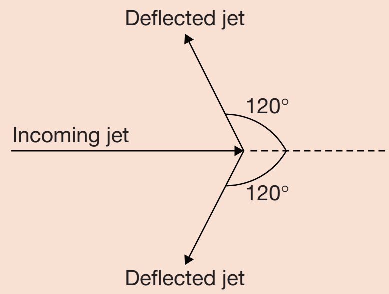
3. Match the items in List I and List II and choose the correct answer: **[GATE, 2007]**

	List I		List II
P.	Centrifugal compressor	**1.**	Axial flow
Q.	Centrifugal pump	**2.**	Surging
R.	Pelton wheel	**3.**	Priming
S.	Kaplan turbine	**4.**	Pure impulse

Codes:
(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

4. A horizontal water jet with a velocity of 10 m/s and cross-sectional area of 10 mm^2 strikes a flat plate held normal to the flow direction. The density of water is 1000 kg/m^3. The total force on the plate due to the jet is **[GATE, 2007]**
(A) 100 N (B) 10 N
(C) 1 N (D) 0.1 N

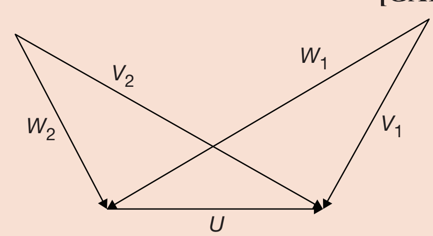
5. Water, having a density of 1000 kg/m^3, issues from a nozzle with a velocity of 10 m/s and the jet strikes a bucket mounted on a pelton wheel. The wheel rotates at 10 rad/s. The mean diameter of the wheel is 1 m. The jet is split into two equal streams by the bucket, such that each stream is deflected 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 **[GATE, 2008]**



(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)

6. 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 **[GATE, 2010]**
(A) 177 (B) 354
(C) 500 (D) 707

7. 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 **[GATE 2012]**



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

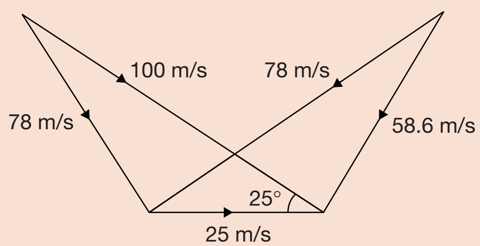
8. In order to have maximum power from a pelton turbine, the bucket speed must be **[GATE, 2013]**
(A) equal to the jet speed.
(B) equal to half of the jet speed.

(C) equal to twice the jet speed.
(D) independent of the jet speed.

9. 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 ______. **[GATE, 2014]**

10. 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 20°. The maximum blade efficiency (in per cent) is ______. **[GATE, 2014]**

11. 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 ______. **[GATE, 2014]**



12. Kaplan water turbine is commonly used when the flow through its runner is **[GATE, 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.

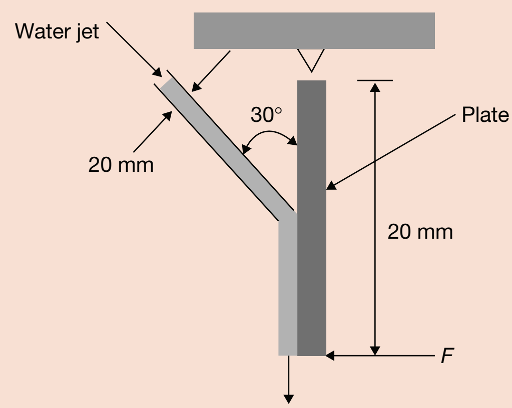
13. A horizontal jet of water with its cross-sectional area of 0.0028 m^2 hits a fixed vertical plate with a velocity of 5 m/s. After impact the jet splits symmetrically in a plane parallel to the plane of the plate. The force of impact (in N) of the jet on the plate is **[GATE 2014]**
(A) 90 (B) 80
(C) 70 (D) 60

14. A horizontal nozzle of 30 mm diameter discharges a steady jet of water into the atmosphere at a rate of 15 litres per second. The diameter of inlet to the nozzle is 100 mm. The jet impinges normal to a flat stationary plate held close to the nozzle end. Neglecting air friction and considering the density of water as 1000 kg/m^3, the force exerted by the jet (in N) on the plate is ______. **[GATE, 2014]**

15. Which of the following statements are TRUE, when the cavitation parameter $\sigma = 0$?
I. The local pressure is reduced to vapor pressure
II. Cavitation starts
III. Boiling of liquid starts
IV. Cavitation stops **[GATE, 2015]**
(A) I, II and IV (B) Only II and III
(C) Only I and III (D) I, II and III

16. The relationship between the length scale ratio (L_r) and the velocity scale ratio (V_r) in hydraulic models, in which Froude dynamic similarly is maintained, is **[GATE, 2015]**
(A) $V_r = L_r$ (B) $L_r = \sqrt{V_r}$
(C) $V_r = L_r^{1.5}$ (D) $V_r = \sqrt{L_r}$

17. A square plate is suspended vertically from one of its edges using a hinge support as shown in the figure. A water jet of 20 mm diameter having a velocity of 10 m/s strikes the plate at its mid-point, at an angle of 30° with the vertical. Consider g as 9.81 m/s^2 and neglect the self-weight of the plate. The force F (expressed in N) required to keep the plate in its vertical position is ________. **[GATE, 2016]**



18. A penstock of 1 m diameter and 5 km length is used to supply water from a reservoir to an impulse turbine. A nozzle of 15 cm diameter is fixed at the end of the penstock. The elevation difference between the turbine and water level in the reservoir is 500 m. Consider the head loss due to friction as 5% of the velocity head available at the jet. Assume unit weight of water = 10 kN/m^3 and acceleration due to gravity (g) = 10 m/s^2. If the overall efficiency is 80%, power generated (expressed in kW and rounded to nearest integer) is ______________. **[GATE, 2016]**

Answer Keys

Exercises

1. A	**2.** B	**3.** B	**4.** A	**5.** C	**6.** C	**7.** A	**8.** A	**9.** B	**10.** C
11. C	**12.** D	**13.** D	**14.** B	**15.** A	**16.** A	**17.** C	**18.** D	**19.** C	**20.** A
21. A	**22.** 3	**23.** C	**24.** A	**25.** D	**26.** C	**27.** C	**28.** D	**29.** A	**30.** B
31. B	**32.** D	**33.** C	**34.** B	**35.** B	**36.** B	**37.** B	**38.** B		

Previous Years' Questions

1. C	**2.** C	**3.** A	**4.** C	**5.** D	**6.** B	**7.** C	**8.** B	**9.** 56 to 57
10. 85.1 to 89.9		**11.** 3250 to 3300		**12.** B	**13.** C	**14.** 318.3	**15.** D	**16.** D
17. 7.85	**18.** 6570							