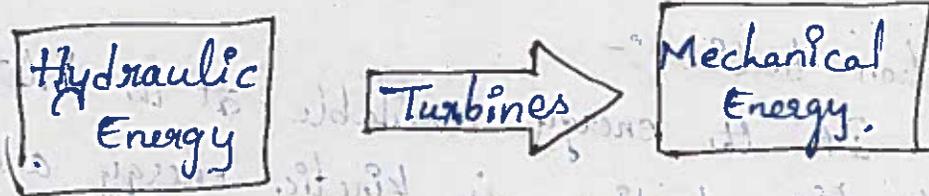


Hydraulic Turbines - I

Hydraulic turbines and its classification

Hydraulic turbines are basically defined as the hydraulic machines which convert hydraulic energy into mechanical energy and this mechanical energy will be given to a generator to produce electric energy.



Now there will be one question that how this mechanical energy will be given to electric generator. Electric generator will be directly coupled with the hydraulic turbine and therefore mechanical energy developed by hydraulic turbines, will be transmitted to electric generator and hence use mechanical energy will be converted into electrical energy.

Electric power developed from hydraulic energy will be considered as hydroelectric power. We have used here term i.e hydraulic energy that indicates the energy of water.

Classification of hydraulic turbines :-

Hydraulic turbines will be classified on the basis of the type of energy available at the inlet of the hydraulic turbine, direction of flow through the vanes, head at the inlet of the hydraulic turbine and specific speed of the hydraulic turbine.

Let us find out here a brief classification of hydraulic turbines as mentioned here.

According to the type of energy at the inlet of the turbine.

(i) Impulse turbine

(ii) Reaction turbine.

Impulse :-

If the energy available at the inlet of the hydraulic turbine is only kinetic energy, the hydraulic turbine will be considered as Impulse turbine.

Reaction turbine :-

If the energy available at the inlet of the hydraulic turbine is kinetic energy and pressure energy, the hydraulic turbine will be considered as Reaction turbine.

According to the direction of flow through runner

1) Tangential flow turbine.

2) Radial flow turbine.

3) Axial flow turbine.

4) Mixed flow turbine.

Tangential flow turbine :-

If the water flow along the tangent of the runner, the hydraulic turbine will be considered as Tangential flow turbine.

Radial flow turbine :-

If the water flows in radial direction through the runner, the hydraulic turbine will be considered as radial flow turbine.

Axial flow turbine :-

If the water flows through the runner along the directions parallel to the axis of rotation of the runner, the hydraulic turbine will be considered as axial flow turbine.

Mixed flow turbine:-

If the water flow through the runner is in the radial direction but leaves in the direction parallel to the axis of rotation of the runner, the hydraulic turbine will be considered as mixed flow turbine.

According to the head at the inlet of the turbine:-

- (i) High head turbine
- (ii) Medium head turbine
- (iii) Low head turbine.

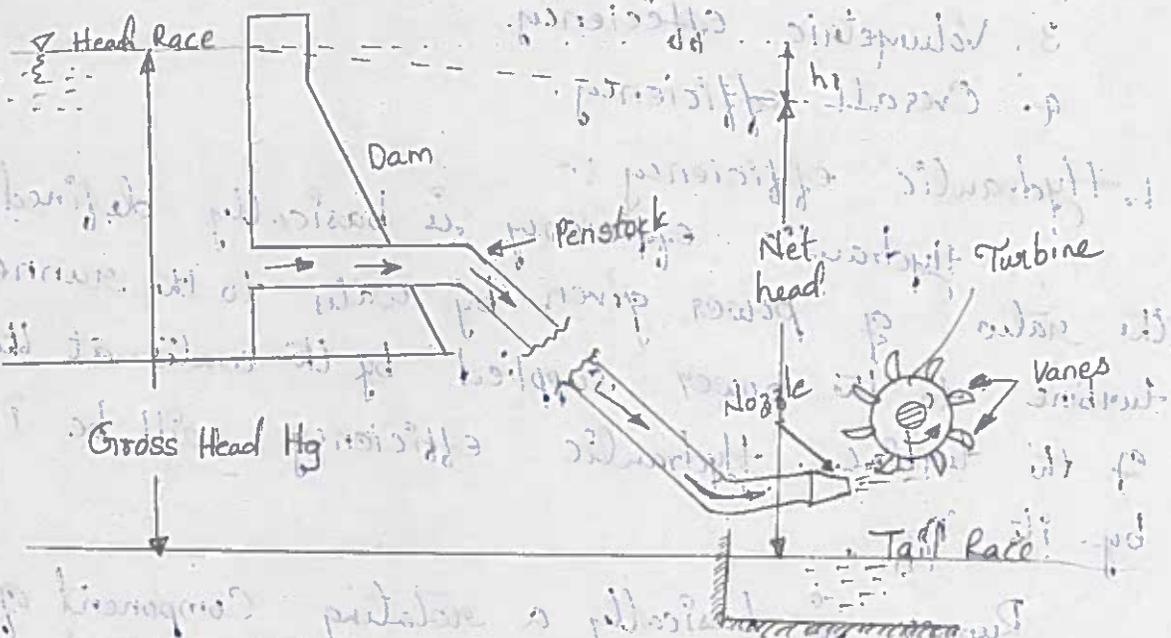
According to the specific speed of the turbine:-

- (i) Low specific speed turbine
- (ii) Medium specific speed turbine
- (iii) High specific speed turbine.

* Definitions of Heads and Efficiencies of Turbines:-

Gross Head :-

Gross head is basically defined as the difference b/w the head race level and tail race level when water is not flowing. Gross head will be indicated by H_g as displayed here in following figure.



Net head :-

Net head is basically defined as the head available at the inlet of the turbine. Net head is also simply called as effective head.

When water will flow from head race to the turbine there will be some losses of head due to friction of water and penstock. There will also be other losses of head such as loss of head due to bend, fitting, at entrance of penstock etc. We must note it here that these losses will be very less and could be neglected when we compare with head loss due to friction.

Net head available at the inlet of turbine could be written as mentioned here.

$$\text{Net head, } H = \text{Gross head (H}_g\text{)} - \text{head losses due to friction (h}_f\text{)}$$

Loss of head due to friction will be given by

Darcy - Weisbach equation and we can find it here.

* Efficiencies of a turbine :-

There are following important efficiencies that we will discuss here in this post.

1. Hydraulic efficiency.
2. Mechanical efficiency.
3. Volumetric efficiency.
4. Overall efficiency.

1. Hydraulic efficiency :-

Hydraulic efficiency is basically defined as the ratio of power given by water to the runner of turbine to the power supplied by the water at the inlet of the turbine. Hydraulic efficiency will be indicated by its η_h .

Runner is basically a rotating component of a turbine and buckets or vanes will be fixed at the circumference of the runner.

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Vanes or buckets fixed on the runner are not smooth and hence will be hydraulic losses when water will flow through these vanes of the turbine. Therefore, power given by water to the runner of the turbine will be less than the power supplied by the water at the inlet of the turbine.

Hydraulic efficiency of a turbine could be written as mentioned here.

Hydraulic efficiency (η_h) = Power delivered to the runner of turbine / power supplied at the inlet of turbine hydraulic efficiency. (η_h) = R.P / W.P.

R.P = Power delivered to the runner of turbine

W.P = Power supplied at the inlet of turbine of turbine or water power.

Mechanical efficiency:-

Mechanical efficiency is basically defined as the ratio of power available at the shaft of the turbine to the power delivered to the runner of the turbine.

Mechanical efficiency will be indicated by η_m .

Mechanical efficiency of a turbine could be written as mentioned here

Mechanical efficiency (η_m) = Power available at the shaft of the turbine / Power delivered to the runner of the turbine.

Mechanical efficiency (η_m) = S.P / R.P.

S.P = Power available at the shaft of the turbine.

R.P = Power delivered to the runner of turbine.

Volumetric Efficiency:-

The volume of the water striking the runner of a turbine will be slightly less than the volume of the water supplied to the turbine as some amount of water will be discharged to the tail race without striking the runner.

to the tail race without striking the runner of the turbine.

Volumetric efficiency of a turbine could be written as mentioned here.

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

Overall Efficiency (η_o).

It is defined as the ratio of power available at the shaft of the turbine to the power supplied by the water at the inlet of the turbine. It is written as.

$$\eta_o = \frac{\text{Volume available at the shaft of the turbine}}{\text{power supplied at the inlet of the turbine.}}$$

$$= \frac{\text{Shaft power}}{\text{Water power.}}$$

$$= \frac{S.P}{W.P} \Rightarrow \frac{S.P}{W.P} \times \frac{R.P}{R.P}$$

$$= \frac{S.P}{R.P} \times \frac{R.P}{W.P}$$

$$= \eta_m \times \eta_h.$$

If shaft power (Sp) is taken in kW then water power should be taken in kW. Shaft power is commonly represented by P. But from equation,

$$\text{Water power in kW} = \frac{\rho \times g \times Q \times H}{1000}$$

$$\eta_o = \frac{\text{Shaft power in kW}}{\text{water power in kW}} = \frac{P}{\left(\frac{\rho \times g \times Q \times H}{1000}\right)}$$

where

P = shaft power.

37)

* Pelton Wheel (or Turbine):-

The pelton wheel or Pelton turbine is a tangential flow impulse turbine. The water strikes the bucket along the tangent of runner. The energy available at the inlet of the turbine is only kinetic energy. The pressure at the inlet and outlet of the turbine is atmosphere. This turbine is used for high heads and is named after L. A. Pelton, an American Engineer.

Fig. shows the layout of a hydroelectric power plant in which the turbine is pelton wheel. The water from the reservoir flows through the penstock at the outlet of which a nozzle is fitted. The nozzle increase the kinematic energy of the water flowing through the penstock. At the outlet of the nozzle, the water comes out in the form of a jet and strikes the buckets of the runner. The main parts of the pelton turbine are.

1. Nozzle and flow regulating arrangement.
2. Runner and buckets.
3. Casing
4. Breaking jet.

1. Nozzle and Flow regulating Arrangement:-

The amount of water striking the buckets of the runner is controlled by providing a spear in the nozzle as shown in fig. The spear is a conical needle which is operated either by a hand wheel or automatically in an axial direction depending upon the size of the unit. When the spear is pushed forward into the

nozzle the amount of water striking the runner is reduced. On the other hand, if the spear is pushed back, the amount of water striking the runner increases.

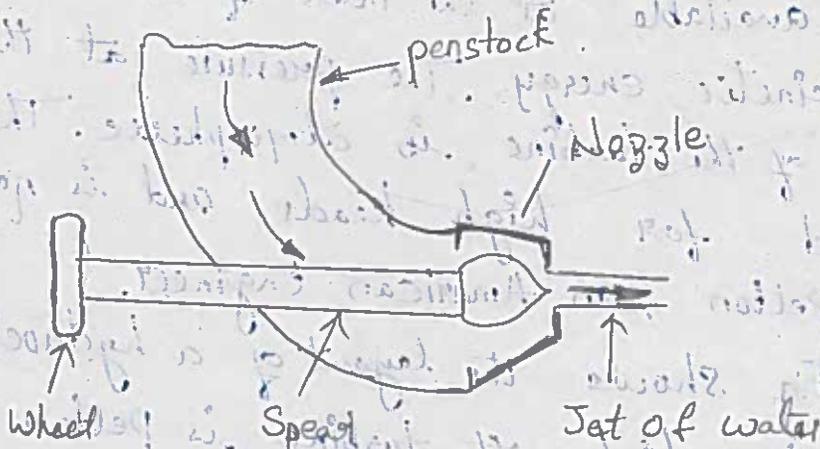


Fig Nozzle with a spear to regulate flow.

2. Runner with Buckets

Figure shows the runner of a pelton wheel. It consists of a circular disc, on the periphery of which a number of buckets evenly spaced are fixed. The shape of the buckets is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as splitter.

The jet of water strikes on the splitter. The splitter divides the jet into two equal parts and the jet comes out at the outer edge of the bucket. The buckets are shaped in such a way that the jet gets deflected through 160° or 170° . The buckets are made of cast iron, cast steel, bronze or stainless steel depending upon the head at the inlet of the turbine.

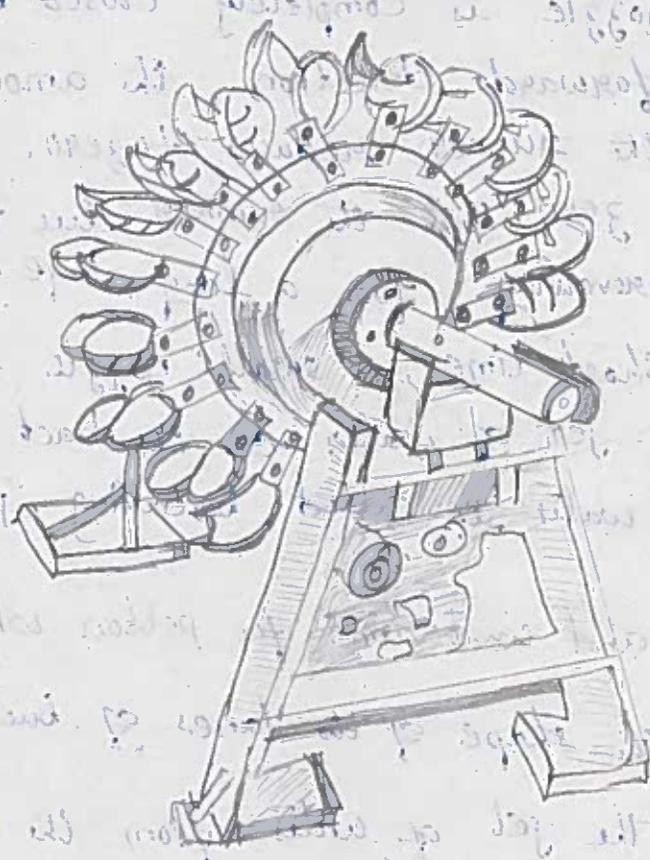


Fig Runner of a pelton wheel.

3. Casing:- Figure shows a pelton turbine with a casing. The function of the casing is to prevent the splashing of the water and to discharge water to all race. It also acts as safeguard against accidents. It is made of cast iron or fabricated steel plates. The casing of the pelton wheel does not perform any hydraulic function.

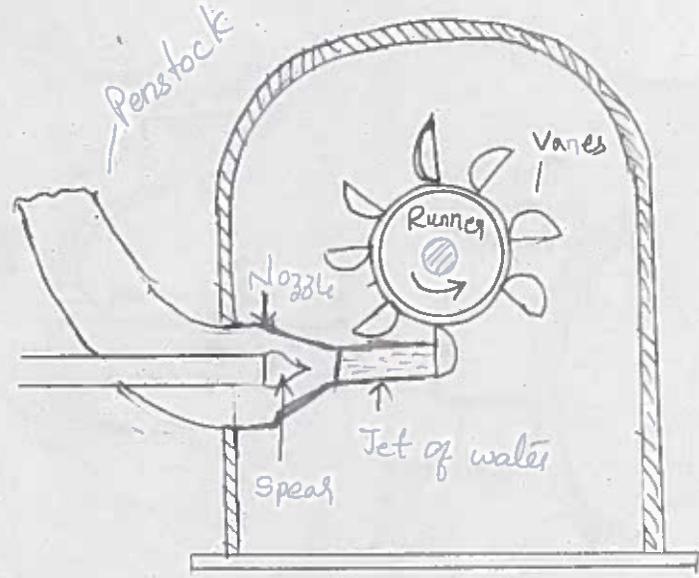


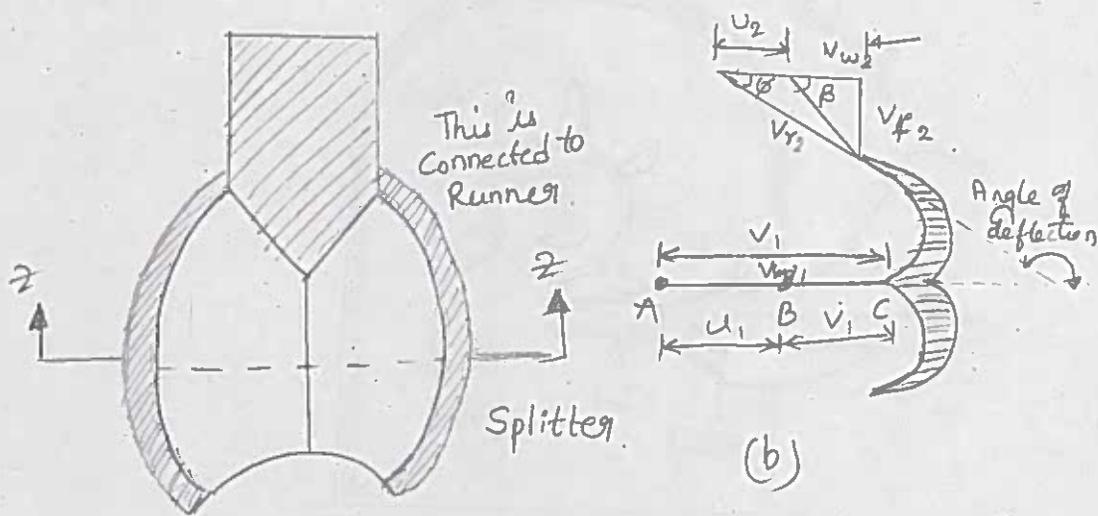
Fig Pelton-turbine.

4. Breaking jet :-

When the nozzle is completely closed by moving the spear in the forward direction, the amount of water striking the runner reduces to zero. But the runner reduces to zero. But the runner due to inertia goes on revolving for a long time. To stop the runner in a short time, a small nozzle is provided which directs the jet of water on the back of the vanes. This jet of water is called breaking jet.

Velocity triangles and work done for pelton wheel :-

Fig shows the shape of the vanes of buckets of the pelton wheel. The jet of water from the nozzle strikes the bucket at the splitter, which splits up the jet into two parts of the jet, glides over the inner surface and comes out at the outer edge. Fig shows the section of the buckets at $z-z$. The splitter is the inlet tip and outer edge of the bucket is the outlet tip of the bucket. The inlet velocity triangle is drawn at the splitter and outlet velocity triangle is drawn at the outer edge of the bucket, by the same method as explained.



(a)

(b)

Fig shape of bucket.

39)

Let $H =$ Net head acting on the pelton wheel.

$$= H_g - h_f$$

where $H_g =$ Gross head and $h_f = \frac{4 f l v^2}{D^* \times 2g}$.

$D^* =$ Dia of penstock

$D =$ Diameter of the wheel.

$n =$ speed of the wheel in rpm.

$d =$ diameter of the jet.

then

$$V_1 = \text{velocity of jet at inlet} = \sqrt{2gH}$$

$$u = u_1 = u_2 = \frac{\pi D n}{60}$$

The velocity triangle at inlet will be a straight line

$$V_{r1} = V_1 - u_1 = V_1 - u$$

$$V_{w1} = V_1$$

$$\alpha = 0^\circ \text{ and } \theta = 0$$

From the velocity triangle at outlet we have

$$V_{r2} = V_{r1} \text{ and } V_{w2} = V_{r2} \cos \beta + u_2$$

The force exerted by the jet of water in the direction of motion is given by equation.

$$F_x = \rho a v_1 [V_{w1} + V_{w2}]$$

As the angle β is an acute angle +ve sign should be taken. Also this is the case of series of vane the mass of water striking is $\rho a v_1$ and not $\rho a v_2$. In equation 'a' is the area of the jet which is given as

$$a = \text{Area of jet} = \frac{\pi}{4} d^2$$

Now work done by the jet on the runner per second.

$$= F_x \times u = \rho a v_1 [V_{w1} + V_{w2}] \times u \text{ Nm/s.}$$

power given to the runner by the jet

$$= \frac{\rho a v_1 [V_{w1} + V_{w2}] \times u}{1000} \text{ kw.}$$

Work done per unit weight of water striking

$$= \frac{\rho a v_1 [v_{w1} + v_{w2}] \times u}{\text{weight of water striking}}$$

$$= \frac{\rho a v_1 [v_{w1} + v_{w2}] \times u}{\rho a v_1 \times g}$$

$$= \frac{1}{g} [v_{w1} + v_{w2}] \times u$$

The energy supplied to the jet at inlet is in the form of kinetic energy and is equal to $\frac{1}{2} m v^2$

$$\therefore \text{K.E of jet per second} = \frac{1}{2} (\rho a v_1) \times v_1^2$$

Hydraulic efficiency

$$\eta_h = \frac{\text{work done per second}}{\text{K.E of jet per second}}$$

$$= \frac{\rho a v_1 [v_{w1} + v_{w2}] \times u}{\frac{1}{2} (\rho a v_1) \times v_1^2} = \frac{2 [v_{w1} + v_{w2}]}{v_1^2} \times u$$

$$v_{w1} = v_1, \quad v_{w2} = v_1 - u = (v_1 - u)$$

$$v_{w2} = (v_1 - u)$$

$$v_{w2} = v_{v2} \cos \phi - u = (v_1 - u) \cos \phi - u$$

Substituting the value of v_{w1} and v_{w2} in eqn.

$$\eta_h = \frac{2 [v_1 + (v_1 - u) \cos \phi - u] \times u}{v_1^2}$$

$$= \frac{2 [v_1 - u + (v_1 - u) \cos \phi] \times u}{v_1^2}$$

$$= \frac{2 (v_1 - u) (1 + \cos \phi) u}{v_1^2}$$

The efficiency will be maximum for a given value of v_1 when

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

(or)

$$u = \frac{v_1}{2}$$

40)

Equation states that hydraulic efficiency of a pelton wheel will be maximum when the velocity of the wheel is half the velocity of the jet of water at inlet. The expression for maximum efficiency will be obtained by substituting the value of $u = \frac{V_1}{2}$ in eqn.

$$\text{Max. } \eta_h = \frac{2 \left(V_1 - \frac{V_1}{2} \right) (1 + \cos \phi) \times \frac{V_1}{2}}{V_1^2}$$

$$= \frac{2 \times \frac{V_1}{2} (1 + \cos \phi) \frac{V_1}{2}}{V_1^2}$$

$$\text{max. } \eta_h = \frac{(1 + \cos \phi)}{2}$$

$= 0 =$

* Working proportions for pelton wheel.

(i) The velocity of the jet at inlet is given by $V_1 = C_v \sqrt{2gH}$

$C_v =$ Co-efficient of velocity $= 0.98$

$H =$ Net head on turbine.

(ii) The velocity of wheel (u) is given by $u = \phi \sqrt{2gH}$.

$\phi =$ speed ratio. The value of speed ratio

varies from 0.43 to 0.48.

(iii) The angle of deflection of the jet through buckets is taken at 165° if no angle of deflection is given.

(iv) The mean diameter or the pitch diameter D of the pelton wheel is given by.

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

(v) Jet ratio Z is defined as the ratio of the pitch diameter (D) of the pelton wheel to the diameter of the jet (d). Z is denoted by m and is given as

$$m = \frac{D}{d} (= 12 \text{ for most cases})$$

(vi) Number of buckets on a runner is given by

$$Z = 15 + \frac{D}{2d} = 15 + 0.5m$$

where $m = \text{jet ratio}$

(vii) Number of jet :- It is obtained by dividing the total rate of flow through the turbine by the rate of flow of water through a single jet.

Problem

A Pelton wheel has a mean bucket speed of 10 m/s with a jet of water flowing at the rate of 700 lit/s under a head of 30 m . The buckets deflect the jet through an angle of 160° . Calculate the power given by water to the runner and the hydraulic efficiency of the turbine. Assume co-efficient of velocity as 0.98 .

Sol Given,

speed of buckets $u = u_1 = u_2 = 10 \text{ m/s}$

Discharge $Q = 700 \text{ lit/s} = 0.7 \text{ m}^3/\text{s}$

head of water $H = 30 \text{ m}$.

Angle of deflection $= 160^\circ$

\therefore Angle $\phi = 180^\circ - 160^\circ = 20^\circ$

Co-efficient of velocity $C_v = 0.98$.

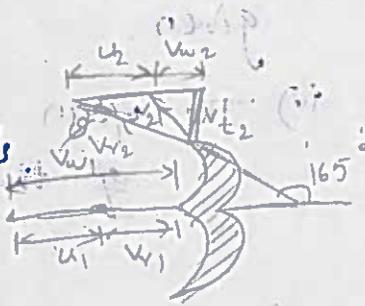
The velocity of jet $= V_1 = C_v \sqrt{2gH}$

$= 0.98 \sqrt{2 \times 9.81 \times 30} = 23.77 \text{ m/s}$.

$V_{r1} = V_1 - u_1 = 23.77 - 10$

$V_{r1} = 13.77 \text{ m/s}$

$V_{w1} = V_1 = 23.77 \text{ m/s}$



from outlet velocity triangle

$V_{r2} = V_{r1} = 13.77 \text{ m/s}$

$V_{w2} = V_{r2} \cos \phi - u_2$

$= 13.77 \cos 20^\circ - 10 = 2.94 \text{ m/s}$

work done by the jet per second on the runner is given by equation

$= \rho a V_1 [V_{w1} + V_{w2}] \times u$

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$$= 1000 \times 0.9 \times [23.77 + 2.94] \times 10$$

$$= 186970 \text{ Nm/s}$$

power given to turbine = $\frac{186970}{1000} = 186.97 \text{ kW}$.

∴ The hydraulic efficiency of the turbine is given by equation as

$$\eta_h = \frac{2[V_{w1} + V_{w2}] \times u}{V_1^2}$$

$$= \frac{2[23.77 + 2.94] \times 10}{23.77 \times 23.77}$$

$$\eta_h = 0.945$$

∴ 94.54%

= 0 =

* Francis turbine :-

The inward flow reaction turbine having radial discharge at outlet is known as Francis Turbine. After the name of J.B Francis, an American engineer who in the beginning designed mixed radial flow reaction type of turbine. In the modern Francis turbine the water enters the runner of the turbine in the radial direction at outlet and leaves in the axial direction at the inlet of the runner. Thus the modern Francis turbine is a mixed flow type turbine.

Radial flow reaction Turbines :-

Radial flow turbine are those turbines in which the water flows in the radial direction. The water may flow radially from outwards to inwards, or from inwards to outwards radial flow turbine.

Reaction turbine means that the water at the inlet of the turbine possesses kinetic energy as well as pressure energy. As the water flows through the runner, a part of pressure energy goes on changing into kinetic energy. Thus the water through the runner is under pressure. The runner is completely enclosed in an air-tight casing and casing and the runner is always full of water.

Main Parts of a radial flow reaction Turbine :-

The main parts of a radial flow reaction turbine are .

1. Casing
2. Guide mechanism
3. Runner
4. Draft tube.

1. Casing :- As mentioned above that in case of reaction turbine, casing and runner are always full of water. The water from the penstock enters the casing which is of spiral shape in which area of cross section of the casing goes on decreasing gradually. The casing completely surrounds the runner of the turbine. The casing as shown in fig. is made of spiral shape, so that the water may enter the runner at constant velocity throughout the circumference of the runner. The casing is made of concrete cast steel or plate steel.

2. Guide Mechanism :-

It consists of a stationary circular wheel all around the runner of the turbine. The stationary guide vanes are fixed on the guide mechanism. The guide vanes allow the water to strike the vanes fixed on the runner without shock at inlet. Also by a suitable arrangement, the width

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vanes of guide mechanism can be altered so that the amount of water striking the runner can be varied.

3. Runner:- It is a circular wheel on which a series of radial curved vanes are fixed. The surface of the vanes are made very smooth. The radial curved vanes are so shaped that the water enters and leaves the runner without shock. The runner are made of cast steel, cast iron. They are keyed to the shaft. ^{water from Penstock}

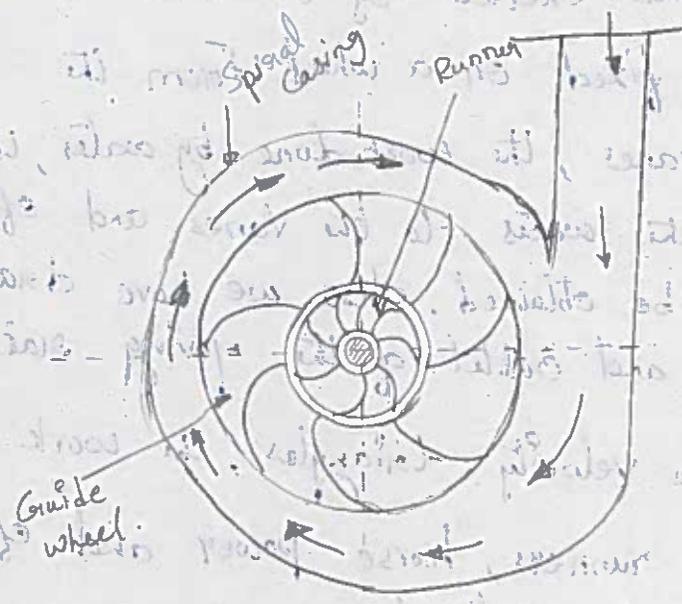


Fig:- Main parts of a radial reaction turbines.

4. Draft tube :-

The pressure at the exit of the runner of a reaction turbine is generally less than atmospheric pressure. The water at exit cannot be directly discharged to the tail race. A tube or pipe of gradually increasing area for discharging water from the exit of the tail race. This tube of increasing area is called draft tube.

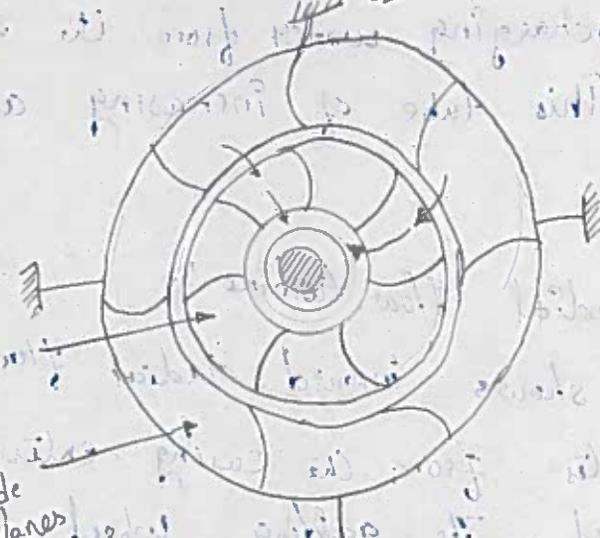
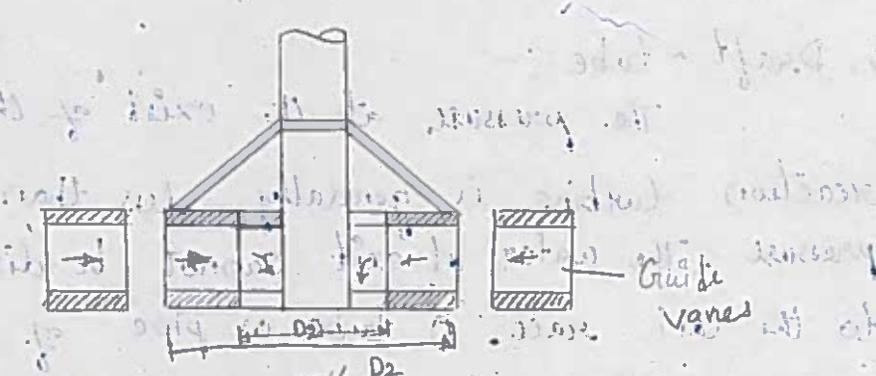
* Inward radial flow turbine :-

Fig shows inward radial flow turbine, in which case the water from the casing enters the stationary guiding wheel. The guiding wheel consists of guide

Vanes which direct the water to enter the runner which consists of moving vanes. The water flows over the vanes in the inward radial direction and is discharged at the inner diameter of the runner. The outer diameter of the runner is the inlet and the inner diameter is the outlet.

Velocity Triangles and work done by water on Runner.

In chapter we have discussed in detail the force exerted by the water on the radial curved vanes fixed on a wheel. From the force exerted on the vanes, the work done by water, the horse power given by the water to the vanes and efficiency of the vanes can be obtained. Also we have drawn velocity triangles at inlet and outlet of the moving radial vanes in Fig. From the velocity triangles, the work done by the water on the runners, horse power and efficiency of the turbine can be obtained.



Inward radial flow turbine. SPIT-4; PG-12/55

The work done per second of the runner by water is given by equation as

$$= \rho a v_1 (v_{w1} u_1 \pm v_{w2} u_2)$$

$$= \rho Q (v_{w1} u_1 \pm v_{w2} u_2)$$

The equation also represents the energy transfer per second to the runner.

where, v_{w1} = Velocity whirl at inlet

v_{w2} = Velocity of whirl at outlet.

u_2 = Tangential velocity of wheel at outlet.

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

u_1 = Tangential velocity of wheel at inlet.

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

The work done per second per unit weight of water per second.

$$= \frac{\text{Work done per second}}{\text{weight of water striking per second}}$$

$$= \frac{\rho Q (v_{w1} u_1 \pm v_{w2} u_2)}{\rho Q \times g} = \frac{1}{g} (v_{w1} u_1 \pm v_{w2} u_2)$$

The equation represents the energy transfer per unit weight to the runner. This equation is known by Euler's equation of hydrodynamic machines. This is also known as fundamental equation of hydrodynamic machine. This is also known as fundamental equation of hydrodynamic machine. The equation was given by Swiss Scientist L. Euler.

In equation, +ve sign is taken if angle β is an acute angle. If β is an obtuse angle then -ve sign is taken. If $\beta = 90^\circ$ then $v_{w2} = 0$ and work done per second per unit weight of water striking becomes as

$$= \frac{1}{g} v_{w1} u_1$$

Hydraulic efficiency is obtained from equation as

$$\eta_h = \frac{R.P}{W.P} = \frac{W}{1000g} \frac{[V_{w1} u_1 \pm V_{w2} u_2]}{W \times H} = \frac{(V_{w1} u_1 \pm V_{w2} u_2)}{gH}$$

where R.P = Runner power i.e. power delivered by water to the runner.

W.P = water power.

If the discharge is radial at outlet, then

$$V_{w2} = 0$$

$$\eta_h = \frac{V_{w1} u_1}{gH}$$

Working proportions for Francis Turbines:-

The following are the important relations

for Francis Turbines.

1. The ratio of width of the wheel to its diameter

is given as $n = \frac{B_1}{D_1}$. The value of n varies from 0.1 to 0.40

2. The flow ratio is given as

Flow ratio = $\frac{V_{f1}}{\sqrt{2gH}}$ and varies from 0.15 to 0.30

3. The speed ratio

$\frac{u_1}{\sqrt{2gH}}$ varies from 0.6 to 0.9.

Design of Francis turbine:-

It is designed to develop known power P , running at speed N , Under head H & probable

values of η_h, η_o, η & q are assumed.

The design of runner involves determination of size & vane angle.

(i) Determine width required discharge Q from.

$$\eta_o = \frac{S.P}{W.P} = \frac{P}{\frac{\rho g Q H}{1000}} \Rightarrow P = \frac{\eta_o \rho g Q H}{1000}$$

2) Determine width and diameter of runner from

$$Q = \text{Circumferential area} \times \text{velocity of flow}$$

$$= \pi B D_1 \times V_{F1} = \pi B_2 D_2 \times \frac{V_{F1}}{2}$$

$$\eta = \frac{B_1}{D_1} \quad B = \eta D_1$$

$$Q = \pi A D^2 \times V_F$$

3) Calculate tangential velocity $u_1 = \frac{\pi D_1 N}{60}$

4) Calculate wheel velocity from $\eta = \frac{V_{w1}}{gH}$

5) From inlet vel. triangles find angle

$$\tan \alpha = \frac{V_{F1}}{V_{w1}} \quad \tan \theta = \frac{V_{F1}}{V_{w1} - u_1}$$

6) The runner diameter at outlet $D_2 = \frac{1}{2} D_1$

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

7) Calculate width & diameter of runner at outlet

$$\pi D_1 B_1 V_{F1} = \pi D_2 B_2 V_{F2}$$

8) Generally it is designed to have $V_{w2} = 0$

then $\beta = 0$, $V_{F2} = V_2$ then

$$\tan \theta = \frac{V_{F1}}{u_1}$$

9) The no. of runner vanes should be one less than

momentum then no. of guide vanes.

no. of guide vanes = no. of runner vanes + 1

no. of guide vanes = no. of runner vanes + 1

no. of guide vanes = no. of runner vanes + 1

no. of guide vanes = no. of runner vanes + 1

no. of guide vanes = no. of runner vanes + 1

* Axial Flow Reaction Turbine:-

If the water flows parallel to the axis of the turbine of its rotation of the shaft, the turbine is known as axial flow turbine. And if the head at the inlet of the turbine is the sum of pressure energy and kinetic energy and during the flow of water through runner a part of pressure energy is converted into kinetic energy, the turbine is known as reaction turbine.

For the axial flow reaction turbine:

the shaft of the turbine is vertical. The lower end of the shaft is made larger which is known as hub or boss. The vanes are fixed on the hub and hence hub or boss. The vanes are fixed on the hub and hence hub acts as a runner for axial flow reaction turbine. The following are the important type of axial flow reaction turbine.

- 1) Propeller turbine.
- 2) Kaplan Turbine.

When the vanes are fixed to the hub and they are not adjustable, the turbine is known as propeller turbine. But if the vanes on the hub are adjustable, the turbine is known as a Kaplan turbine, after the name of V Kaplan, an Austrian engineer. This turbine is suitable where a large quantity of water at low head is available.

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On the hub, the adjustable vanes are fixed as shown in fig.

- 1) Scroll Casing
- 2) Guide vane mechanism.
- 3) Hub with vanes or runner of the turbine
- 4) Draft tube.

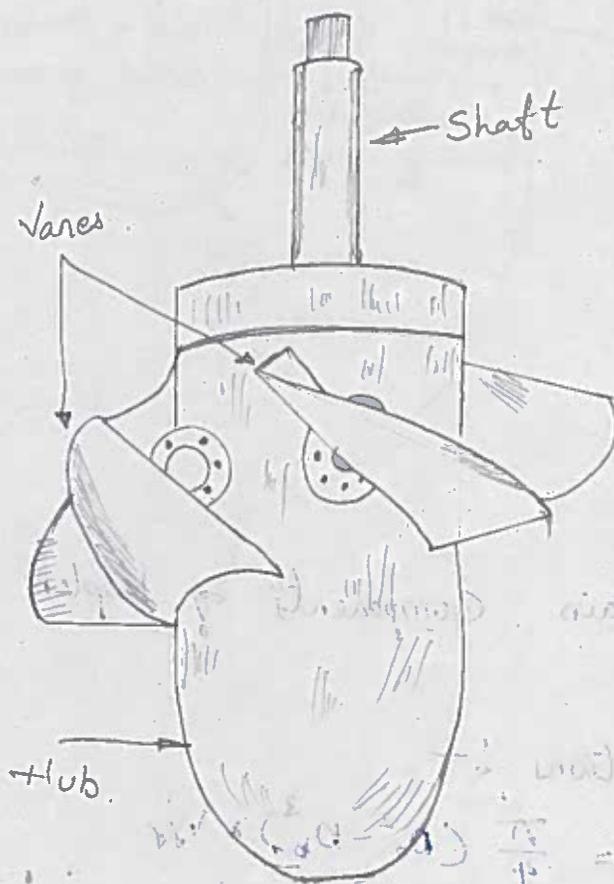


fig Kaplan turbine runner.

Fig shows all parts of a Kaplan turbine. The water from penstock enters the scroll casing and then moves to the guide vanes. From the guide vanes, the water turns through 90° and flows axially through the runner as shown in fig. The discharge through the runner is obtained as.

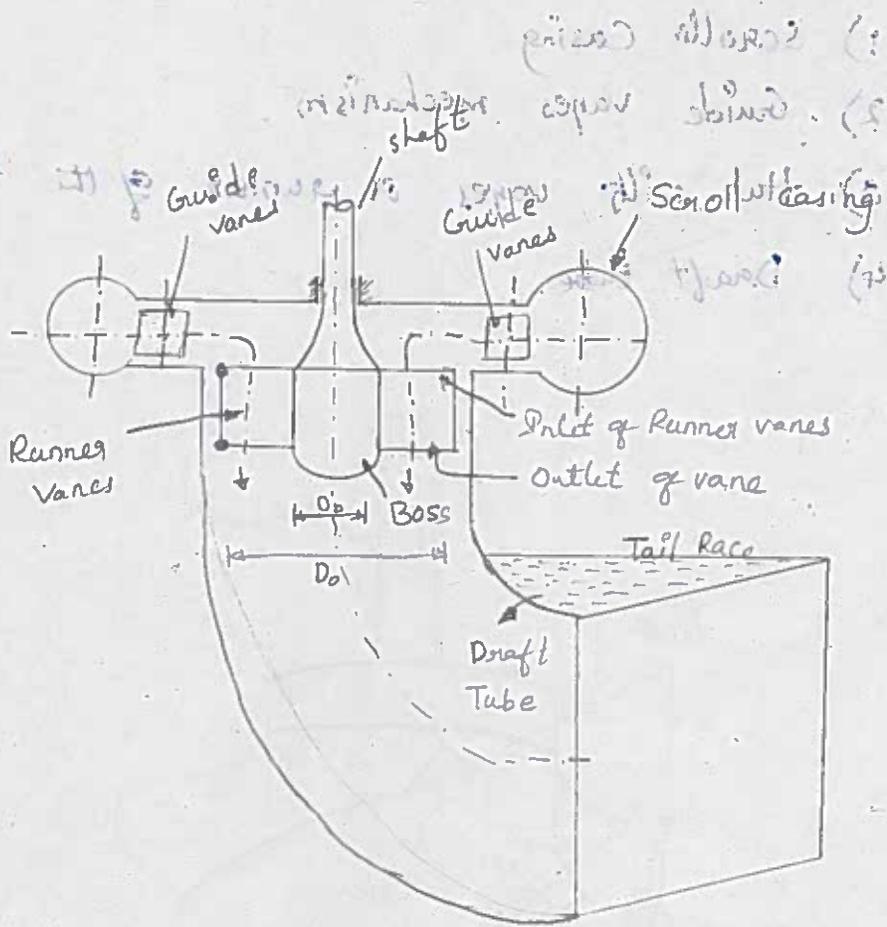


Fig. Main Components of Kaplan turbine.

* Working proportions :-

$$Q = \frac{\pi}{4} (D_o^2 - D_b^2) \times v_{eff}$$

$$\eta = \frac{D_b}{D_o} \approx 0.35 \text{ to } 0.60$$

D_o = Outer diameter of its runner.

D_b = Diameter of hub

v_{eff} = velocity of flow at inlet

The inlet and outlet velocity triangles are drawn at its extreme edge of its runner vane corresponding to its points 1 and 2.

46.
* Some Important point for propeller (Kaplan Turbine).

The following are the important points for propeller & Kaplan turbine:

1) The peripheral velocity at inlet and outlet are equal.

Circumferential Velocity

$$u_1 = u_2 = \frac{\pi D_o N}{60}, \text{ where } D_o = \text{Outer dia of runner}$$

2) Velocity of flow at inlet and outlet are equal

$$V_{f1} = V_{f2} = 4\sqrt{2gH} \quad [4 \approx 0.70]$$

3) Area of flow at inlet = Area of flow at outlet

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

Problem

A Kaplan turbine working under a head of 20 m develops 11772 kW shaft power. The outer dia of the runner is 3.5 m and hub diameter is 1.75 m with the guide blade angle at the extreme edge of the runner 35°. The hydraulic and overall efficiencies of the turbine are 88% and 84% respectively. If the velocity of whirl is zero at outlet determine

- (i) Runner vane angles at inlet and outlet at the extreme edge of the runner and.
- (ii) Speed of the turbine.

Sol Given

Head

$$H = 20 \text{ m}$$

Shaft power S.P = 11772 kW

$$D_o = 3.5 \text{ m}$$

$$D_b = 1.75 \text{ m}$$

$$\alpha = 35^\circ$$

$$\eta_h = 88\%$$

$$\eta_o = 84\%$$

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$$\eta_o = \frac{\text{S.P}}{\text{W.P}}$$

$$\text{where, W.P} = \frac{\text{W.P}}{1000} = \frac{\rho \times g \times Q \times H}{1000}$$

$$0.84 = \frac{11772}{\frac{\rho \times g \times Q \times H}{1000}}$$

$$= \frac{11772 \times 1000}{1000 \times 9.81 \times Q \times 20}$$

$$Q = \frac{11772 \times 1000}{0.84 \times 1000 \times 9.81 \times 20} = 71.42 \text{ m}^3/\text{s}$$

Using equation

$$Q = \frac{\pi}{4} (D_o^2 - D_b^2) \times V_{f1}$$

$$V_{f1} = \frac{71.428}{7.216} = 9.9 \text{ m/s}$$

Inlet velocity triangle $\tan \alpha = \frac{V_{f2}}{V_{w1}}$

$$V_{w1} = \frac{V_{f1}}{\tan \alpha} = \frac{9.9}{\tan 35^\circ} = 14.14 \text{ m/s}$$

Using the relation for hydraulic efficiency,

$$\eta_h = \frac{V_{w1} u_1}{gH}$$

$$0.88 = \frac{14.14 \times u_1}{9.81 \times 20}$$

$$u_1 = \frac{0.88 \times 9.81 \times 20}{14.14} = 12.21 \text{ m/s}$$

(ii) Runner vane angles at inlet and outlet at its extreme edge of its runner are given as

$$\tan \theta = \frac{V_f}{V_{w1} - u_1} = \frac{9.9}{(14.14 - 12.21)} = 5.13$$

$$\theta = \tan^{-1} 5.13 = 78.97^\circ$$

For Kaplan turbine

$$u_1 = u_2 = 12.21 \text{ m/s and } v_{f1} = v_{f2} = 9.9 \text{ m/s}$$

∴ From outlet velocity triangle, $\tan \phi = \frac{v_{f1}}{u_1}$

$$= \frac{9.9}{12.21} = 0.811$$

$$\phi = \tan^{-1} 0.811 = 39.035^\circ$$

(ii) Speed of turbine is given by $u_1 = u_2 = \frac{\pi D_o \omega}{60}$

$$12.21 = \frac{\pi \times 3.5 \times \omega}{60}$$

$$\omega = \frac{60 \times 12.21}{\pi \times 3.5} = 66.63 \text{ r.p.m}$$

* Draft tube :-

The draft tube is a pipe of gradually increasing area which connects the outlet of the runner to the tail race. It is used for discharging water from the exit of the turbine to the tail race. This pipe of gradually increasing area is called draft tube. The draft tube, in addition to serve a passage for water discharge, has the following two purposes also.

1) It permits a negative head to be established at the outlet of the runner and thereby increases the net head on the turbine. The turbine may be placed above the tail race without any loss of net head and hence turbine may be inspected properly.

2) It converts a large proportion of the kinetic energy ($\frac{v_2^2}{2g}$) rejected at outlet of the turbine into useful pressure energy. Without the draft-tube the kinetic energy rejected at the outlet of the turbine will go waste to the tail race.

Hence by using draft-tube, the net head on the turbine increases. The turbine develops more power and also the efficiency of the tail race.

If a reaction turbine is not fixed not fitted with a draft-tube the pressure at the outlet of runner will be equal to atmospheric pressure. The water from the outlet of the runner will discharge freely into the tail race. The net head on the turbine will be less than that of a reaction turbine fitted with a draft tube.

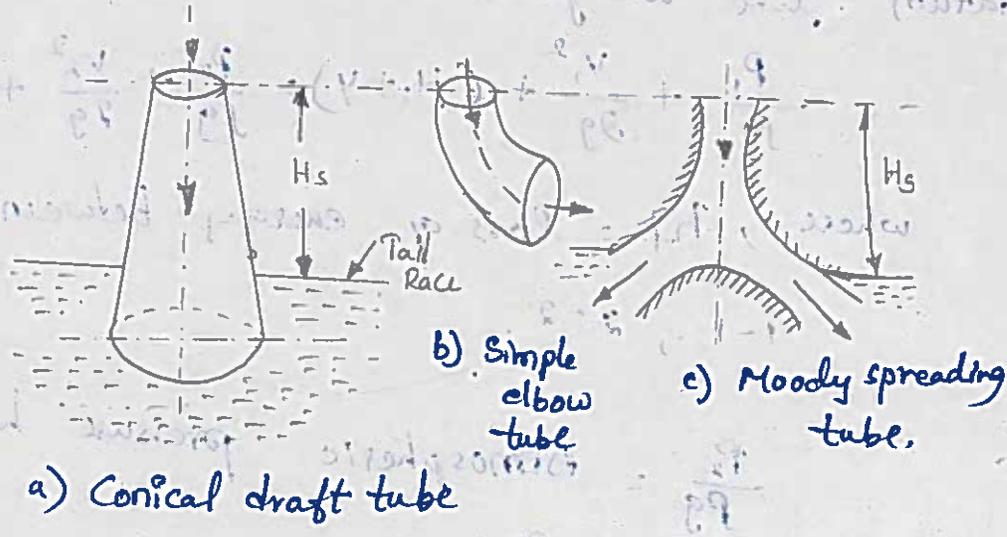
Also without a draft-tube, the kinetic energy ($\frac{v_2^2}{2g}$) rejected at the outlet of the runner will go waste to the tail race.

* Types of Draft tubes:-

The following are the important types of draft-tubes which are commonly used:

1. Conical draft tubes,
2. Simple elbow tubes
3. Moody Spreading tube.
4. Elbow draft-tubes with circular inlet and rectangular outlet.

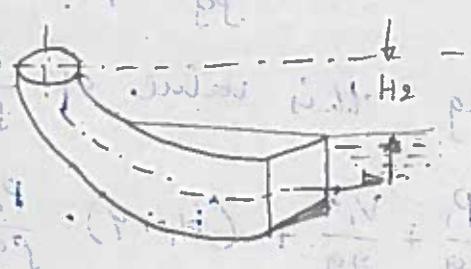
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a) Conical draft tube

b) Simple elbow tube

c) Moody spreading tube.



d) Draft-tube with circular inlet and rectangular outlet.

Fig Types of draft tubes.

These different types of draft tubes are shown in fig. The conical draft tubes and moody spreading draft tubes are most efficient while simple elbow tubes and elbow draft tubes with circular inlet and rectangular outlet require less space as compared to other draft-tubes.

Draft tube theory:-

Consider a conical draft-tube as shown in fig.
 H_s = Vertical height of draft tube above the tail race.
 Y = Distance of bottom of draft tube from tail race.

Applying Bernoulli's equation to inlet and outlet of the draft tube and taking section 2-2 as the datum line we get.

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + (H_1 + y) = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + 0 + h_f$$

where, h_f = loss of energy between sections

1-1, 2-2

$$\frac{P_2}{\rho g} = \text{Atmospheric pressure head} + y$$

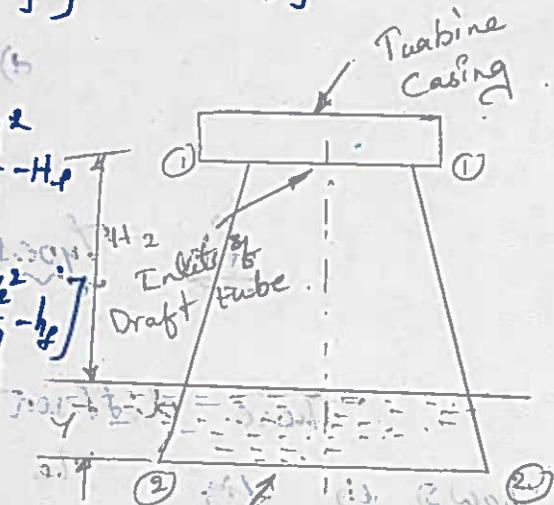
$$= \frac{P_a}{\rho g} + y$$

Substituting this value of $\frac{P_2}{\rho g}$ in equation (1) we get

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + (H_1 + y) = \frac{P_a}{\rho g} + y + \frac{V_2^2}{2g} + h_f$$

$$\frac{P_1}{\rho g} = \frac{P_a}{\rho g} + \frac{V_2^2}{2g} + h_f - \frac{V_1^2}{2g} - H_1$$

$$= \frac{P_a}{\rho g} - H_3 - \left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f \right]$$



∴ In equation

$$\frac{P_1}{\rho g}$$

is less than atmospheric pressure.

By Draft tube theory

* Efficiency of Draft Tube :-

The efficiency of a draft tube is defined as the ratio of actual conversion of kinetic head into pressure head in the draft tube to its kinetic head at the inlet as of the draft tube. Mathematically, it is written as.

$\eta_d = \frac{\text{Actual Conversion of kinetic head into pressure head}}{\text{kinetic head at the inlet of draft tube}}$

$V_1 = \text{Velocity of water at inlet of draft tube}$

$V_2 = \text{Velocity of water at outlet of draft tube}$

$h_f = \text{loss of head in the draft tube}$

$\frac{\text{Theoretical Conversion of kinetic head into pressure head in draft tube}}{\text{kinetic head at the inlet of draft tube}}$

$= \left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right]$

Actual Conversion of kinetic head into pressure head

head = $\left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right] - h_f$

$\eta_d = \frac{\left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g} \right] - h_f}{\left[\frac{V_1^2}{2g} \right]}$

$\left[\frac{V_1^2}{2g} \right]$

$\neq 0$

Hydraulic Turbines - II* Governing of Turbines :-

The governing of a turbine is defined as the operation by which the speed of the turbine is kept constant under all conditions of working. It is done automatically by means of a governor, which regulates the rate of flow through the turbine according to the changing load conditions on the turbine.

When the load on the generator decreases, the speed of the generator increases beyond the normal speed. Then the speed of the turbine also increases beyond the normal speed. If the turbine or the generator is to run at constant speed the rate of flow of water to the turbine should be decreased till the speed becomes normal. This process by which the speed of the turbine is kept constant under varying conditions of load is called governing.

* Governing of Pelton Turbine (Impulse Turbine).

Governing of Pelton turbine is done by means of oil pressure governor, which consists of the following parts:

- 1) Oil sump
- 2) Gear pump also called oil pump, which is driven by power obtained from turbine shaft.

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- 3) The servomotor also called the relay cylinder
 - 4) The control valve or the distribution valve or relay valve.
 - 5) The centrifugal governor or pendulum which is driven by belt or gear from the turbine shaft
 - 6) Pipes connecting the oil sump with the control valve and control valve with servomotor and
 - 7) The spear rod or needle.

Figure shows the position of the piston in the relay cylinder, position of control or relay valve and fly balls of the centrifugal governor, when the turbine is running at the normal speed.

sp When the load on the generator decreases, the speed of the generator increases. This increase the speed of the turbine beyond the normal speed. The centrifugal governor, which is connected to the turbine main shaft, will be rotating at an increased speed. Due to increase in the speed of the centrifugal governor, the fly balls move upwards due to the increased centrifugal force on them. Due to the upward moment of the fly balls, the sleeve and piston rod of the control valve. As the sleeve moves up the lever turn about the fulcrum and the piston rod of the control valve moves downwards. This closes the valve V_1 and opens the valve V_2 as shown in figure.

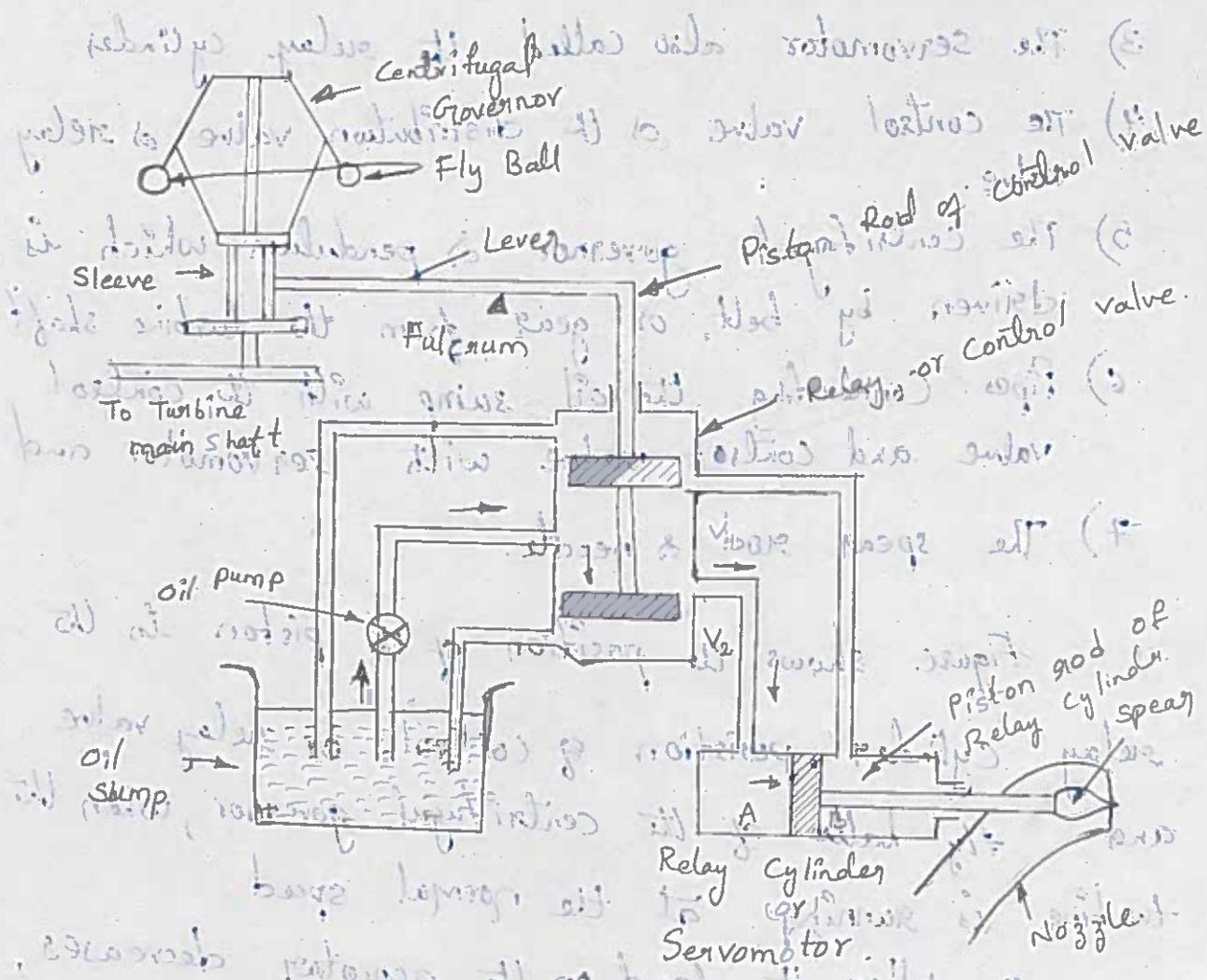


Fig:- Governing of pelton turbine.

When the load on the generator increases, the speed of the generator and hence of the turbine decrease. The speed of the centrifugal governor also decreases and hence centrifugal force acting on the fly-balls also reduces. This brings the fly balls in the downward direction. Due to this, the sleeve moves downward and the lever or turns about the fulcrum moves the piston rod of the control valve in the upward direction. This closes the valve V_2 to opens the valve V_1 . The oil under pressure from the control valve, will move through valve V_1 to the servomotor and will exert a force on the face B of the piston. This will move the piston along with the piston rod and spear toward left, increasing the area of flow of water at the outlet of the nozzle.

* Runaway speed

For a turbine working under maximum head and full gate opening, if its external load suddenly drops to almost zero value and at the same time the governing mechanism of the turbine also fails, then the turbine runner will tend to race up and it will attain the maximum possible speed. This maximum or limiting speed of the turbine runner is known as runaway speed. Obviously for safe design the various rotating components are designed for the runaway speed. For a pelton wheel the runaway speed normally ranges from 1.8 to 1.9 times its normal speed, for a francis turbine it normally ranges from 2 to 2.2 times its normal speed and for a kaplan turbine its normally ranges from 2.5 to 3 times its normal speed.

* Surge tanks:-

As indicated earlier when the load on the generation decreases the governor reduces the rate of flow of water striking the runner in order to maintain the constant speed for the runner. But the sudden reduction of the rate of flow in the penstock may lead to setting up of water hammer in the pipe, which may cause excessive inertia pressure in the pipeline due to which the pipe may burst. Two devices viz, deflectors and relief valve as described earlier are thus provided to avoid the sudden reduction of the rate of flow in penstock. But neither of these devices certain other devices such as surge tank and forebay are usually employed

Surge tanks are employed in the case of high and medium head hydro-power plants where the length of the penstock is short.

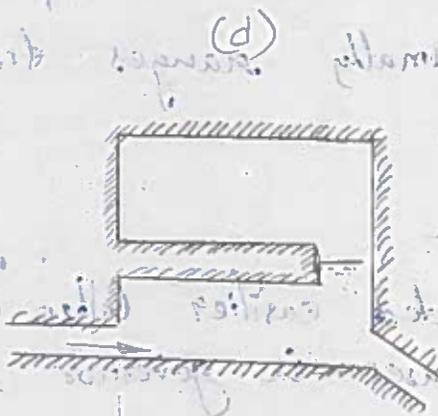
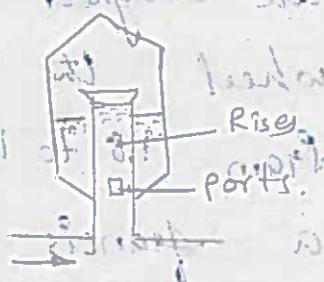
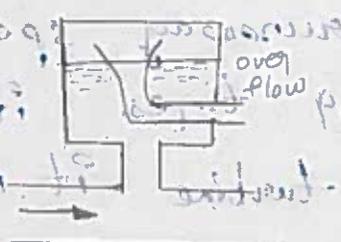
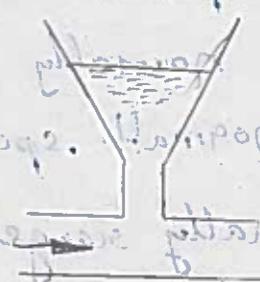
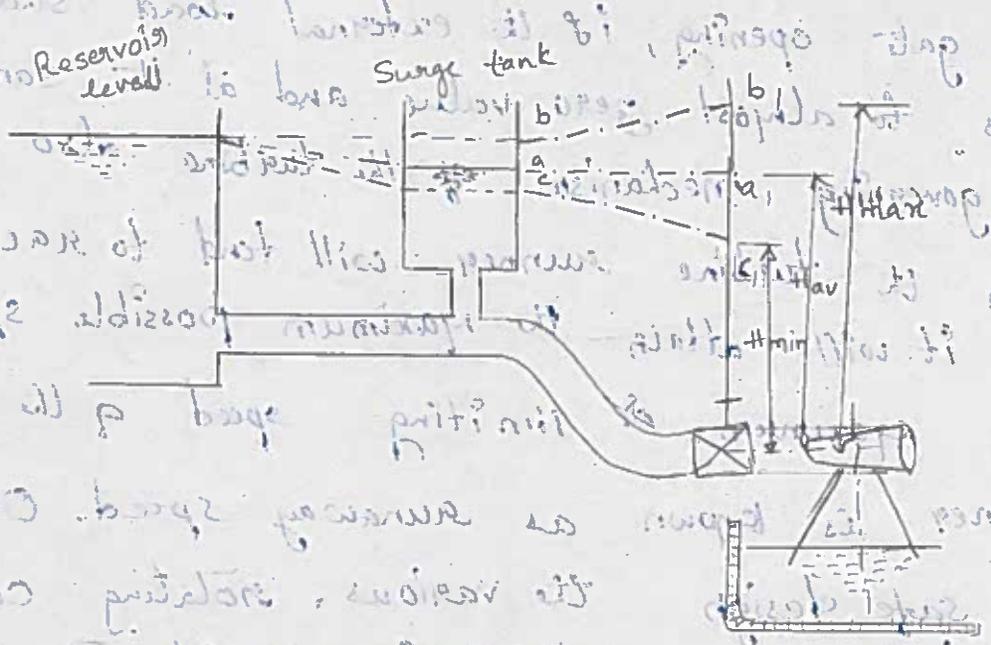


Fig Different types of surge tanks.

An ordinary surge tank is a cylindrical open topped storage reservoir as shown fig. which is connected to the penstock at a point as close as possible to the turbine. when the load on the turbine is steady and normal and there are no velocity variations in the pipe line there will be a normal pressure gradient. The water surface in the surge tank will be lower than the reservoir surface by an amount equal to friction head loss in the pipe.

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connecting the reservoir and the surge tank.

When the load on the generator increases the governor opens the turbine gates to increase the rate of flow entering the runner. As such as the water level in the surge tank falls and a falling pressure gradient occurs developed. In other words the surge tank develops an accelerating head which increases the velocity of flow in the pipeline to a value corresponding to the increased discharge required by the turbine.

The various other types of surge tanks are also shown fig. Type (a) is a conical type surge tank, (b) has an internal bell mouth spillway which permits the overflow to be easily disposed at type (c) is known as the differential surge tank.

which provided with a central riser pipe having small ports or holes at its lower end. This is so because in a differential tank retarding and accelerating heads are developed more promptly than in a simple surge tank in which the heads only built up gradually as the tank fills. Moreover no water is spilled to waste from the differential tank. Type (d) is also similar in performance to the differential tank, but it is suitable when appropriate earth or rock excavation can be carried out.

* Specific speed :-

It is defined as the speed of a turbine which is identical in shape, geometrical dimensions, blade angles, gate opening etc., with the actual turbine but of such a size that will develop unit power when working under unit head. It is denoted by symbol N_s , the specific speed is used in comparing the different types of turbines, all every type of turbine has different specific speed.

In M.K.S units, unit power is taken as one horse power and unit head as one meter. But in S.I units, unit power is taken as one kilowatt and unit head as one meter.

* Derivation of the specific speed :-

The overall efficiency (η_o) of any turbine is given by

$$\eta_o = \frac{\text{Shaft power}}{\text{Water power}} = \frac{\text{Power developed}}{\frac{\rho \times g \times Q \times H}{1000}}$$

$$\eta_o = \frac{P}{\frac{\rho \times g \times Q \times H}{1000}} \quad \text{--- (i)}$$

H = head under which the turbine is working

Q = Discharge through turbine.

P = power developed

form equation (i)
$$P = \eta_o \times \frac{\rho \times g \times Q \times H}{1000}$$

D = Diameter of actual turbine

N = speed of actual turbine

u = Tangential velocity of the turbine

N_s = specific speed of the turbine.

V = Absolute velocity of water.

The absolute velocity, tangential velocity and head on the turbine are related as,

$$u \propto v, \text{ where } v \propto \sqrt{H}$$

But the tangential velocity u is given by

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

\therefore From equation (i) and (ii) we have

$$\sqrt{H} \propto DN \text{ and } D \propto \frac{\sqrt{H}}{N}$$

The discharge through turbine is given by

$$Q = \text{Area} \times \text{Velocity}$$

$$\text{Area} \propto \beta \times \theta$$

$$\propto D^2$$

But

$$\text{velocity} \propto \sqrt{H}$$

And

$$Q \propto D^2 \times \sqrt{H}$$

$$\propto \left(\frac{\sqrt{H}}{N}\right)^2 \times \sqrt{H}$$

$$\propto \frac{H}{N^2} \times \sqrt{H} \propto \frac{H^{3/2}}{N^2}$$

Substituting the value of Q in equation (ii) we get

$$P \propto \frac{H^{3/2}}{N^2} \times H \propto \frac{H^{5/2}}{N^2}$$

$$P = k \frac{H^{5/2}}{N^2}$$

k is constant of proportionality.

If $P=1$, $H=1$ the speed $N = \text{speed } N_s$

Substituting these values in the above eqn.

$$1 = \frac{k \times 1^{5/2}}{N^2}$$

$$P = N_s^2 \frac{H^{5/2}}{N^2}$$

$$N_s = \sqrt{\frac{N^2 P}{H^{5/2}}} = \frac{\sqrt{P}}{H^{5/4}}$$

In equation, if P is taken in metric horse power the specific speed is obtained in k.k.s units. But if P taken in kilowatt the specific speed is obtained in S.I units.

* Significance of specific speed :-

Specific speed plays an important role for selecting the type of the turbine. Also the performance of a turbine can be predicted by knowing the specific speed of the turbine.

The type of turbine for different specific speed is given in Table 18.1 as.

Table 18.1

s.no	specific speed		Types of turbine
	(M.k.s)	(S.I)	
1	10 to 35	8.5 to 30	Pelton wheel with single jet
2	35 to 60	30 to 51	pelton wheel with two or more jets
3	60 to 300	51 to 340	Francis turbine
4	300 to 1000	340 to 826	Kaplan or propeller turbine

Problem
A turbine develops 7225 kW power under a head of 25 m at 135 rpm. Calculate the specific speed of the turbine and state the type of the turbine.

Sol
Given,
Power developed $P = 7225 \text{ kW}$.
Head $H = 25 \text{ m}$
speed $N = 135 \text{ rpm}$

54)

specific speed of the turbine (N_s).
Using equations

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$= \frac{135 \times \sqrt{7225}}{25^{5/4}}$$

$$N_s = 205.28$$

From table 18.1 for specific speeds (SI) b/w 51 and ²²⁵340 the type of turbine is Francis. As the specific speed 205.28 lies in this range and hence type of turbine is Francis.

* Unit Quantities:-

In order to predict the behaviour of turbine working under varying conditions of head, speed, output and gate opening, the results are expressed in terms of quantities which may be obtained, when the head on the turbine is reduced to unity. The conditions of the turbine under unit head are such that the efficiency of the turbine remains unaffected. The following are the three important unit quantities which must be studied under unit head.

1. Unit speed
2. Unit discharge
3. Unit power.

* Unit speed:-

It is defined as the speed of a turbine working under a unit head.

It is denoted by N_u . The expression for

$N_u =$ speed of a turbine under a head H .

$H =$ Head under which a turbine is working.

$u =$ Tangential velocity.

The tangential velocity, absolute velocity of water and head on the turbine are related as,

$$u \propto V \propto \sqrt{H}$$

Also tangential velocity (u) is given by. - (1)

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

For a given turbine, the diameter (D) is constant.

$$u \propto N$$

$$N = k_1 \sqrt{H} \quad \text{--- (2)}$$

where k_1 is a constant of proportionality.

If head on the turbine becomes unity, the

speed becomes unit speed.

$$H = 1, \quad N = N_u$$

Substituting the value in equation (ii) we get

$$N_u = k_1 \sqrt{1.0} = k_1$$

Substituting the value of k_1 in equation (ii)

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

* Unit Discharge :-

It is defined as the discharge passing through a turbine, which is working under a unit head. It is denoted by the symbol Q_u .

The expression for unit discharge is given as.

let $H =$ Head of water on the turbine.

$Q =$ Discharge passing through turbine when head is H on the turbine.

$a =$ Area of flow of water.

55)
The discharge passing through a given turbine under a head H is given by,

$$Q = \text{Area of flow} \times \text{velocity}$$

But for a turbine, area of flow is constant and velocity is proportional to \sqrt{H} .

$$Q \propto \text{velocity} \propto \sqrt{H}$$

$$Q = k_2 \sqrt{H}$$

where k_2 is constant of proportionality

$$\text{If } H = 1, Q = Q_u$$

Substituting these values in eqn (iii) we get

$$Q_u = k_2 \sqrt{1} = k_2$$

Substituting the value in eqn (iii) we get

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

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

* Unit power :-

It is defined as the power developed by a turbine, working under a unit head. It is denoted by the symbol P_u . The expression for unit power is obtained as

let H = Head of water on the turbine.

P = Power developed by the turbine under a head of H ,

Q = Discharge through turbine under a head of H .

The overall efficiency (η_o) is given as

$$\eta_o = \frac{\text{Power developed}}{\text{water power}} = \frac{P}{\rho \times g \times Q \times H}$$

$$P = \frac{\eta_o \times \rho \times g \times Q \times H}{1000}$$

$$P \propto \eta_o$$

$$Q \propto Q \times H$$

$$Q \propto \sqrt{H} \times H$$

$$Q \propto H^{3/2}$$

$$P = k_3 H^{3/2} \quad \text{--- (iv)}$$

where k_3 is a constant of proportionality

$$H = 1 \text{ m} \quad P = P_u$$

$$P_u = k_3 (1)^{3/2} = k_3$$

Substituting the value of k_3 in equation (iv).

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

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

Problem

A turbine develops 9000 kW when runner at 100 rpm the head on the turbine is 30m. If the head on the turbine is reduced to 18m determine the speed and power developed by the turbine.

Sol

$$P = 9000 \text{ kW}$$

$$N_1 = 100 \text{ r.p.m.}$$

$$H_1 = 30 \text{ m}$$

$$H_2 = 18 \text{ m}$$

$$= N_2$$

$$= P_2$$

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

$$N_2 = \frac{N_1 \sqrt{H_2}}{\sqrt{H_1}} = \frac{100 \sqrt{18}}{\sqrt{30}} = \frac{100 \times 4.2426}{5.4772}$$

$$N_2 = 77.46 \text{ rpm}$$

Also we have

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

$$P_2 = \frac{P_1 H_2^{3/2}}{H_1^{3/2}} = \frac{9000 \times 18^{3/2}}{30^{3/2}}$$

$$= \frac{100 \times 687307.78}{164.31}$$

$$P_2 = 4182.84 \text{ kW}$$

* Characteristics Curves of hydraulic Turbines.
 Characteristic curves of a hydraulic turbine are the curves with the help of which the exact behaviour and performance of the turbine under different working conditions, can be known. These curves are plotted from the results of the tests performed on the turbine under different working conditions.

The important parameters which are varied during test on a turbine are:

- 1) speed (N)
- 2) Head (H)
- 3) Discharge (Q)
- 4) Power (P)
- 5) Overall efficiency (η_o)

6) Gate opening.

Out of the above six parameters, three parameters namely speed (N) Head (H) and discharge (Q) are independent parameters.

Out of the three independent parameters (N, H, Q) one of the parameters kept constant and constant variation of the other four parameters with respect to any one of the remaining two independent variables are plotted and various curves are obtained. These curves are called characteristic curves. The following are important characteristic curves of turbine.

1) Main characteristic curves of constant head curves.

2) Operating characteristic curves of constant head speed curves.

3) Muschel curves of constant efficiency curves.

* Main characteristics curves of constant head curves:-

Main characteristic curves are obtained by maintaining a constant head and a constant gate opening on the turbine. The speed of the turbine is varied by changing load on the turbine. For each value of the speed, the corresponding values of the power (P) and discharge (Q) are obtained.

These curves are called characteristic curves.

The following are the important characteristic curves of a turbine.

By changing the gate opening the values of Q_u , P_u and η_o and N_u are determined and taking N_u as abscissa the values of Q_u , P_u and η_o are plotted. Shows the main characteristic curves for pelton wheel and shows the main characteristics curves for reaction turbines.

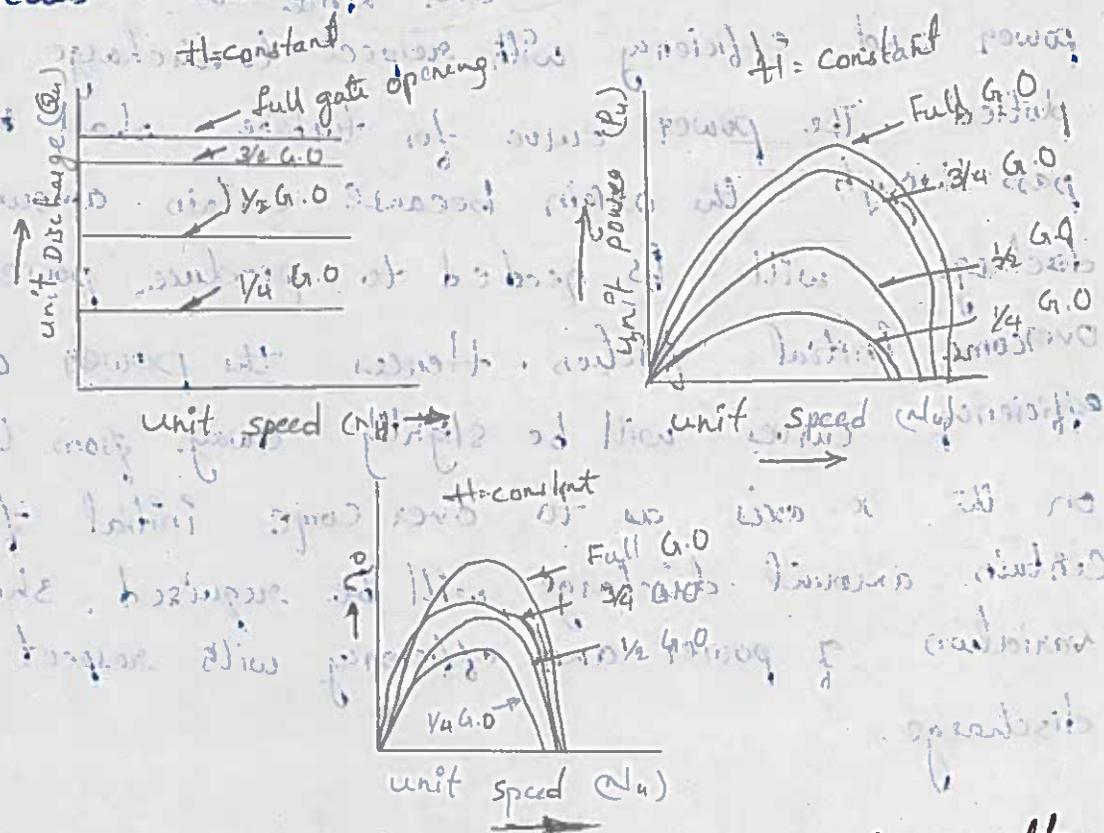
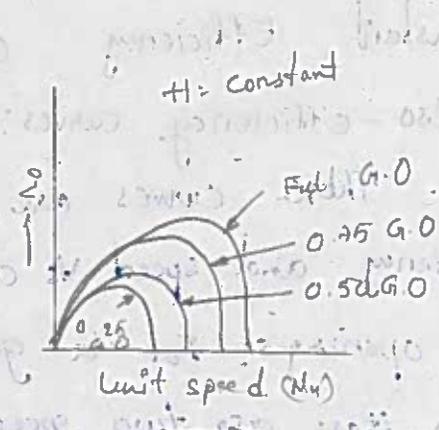
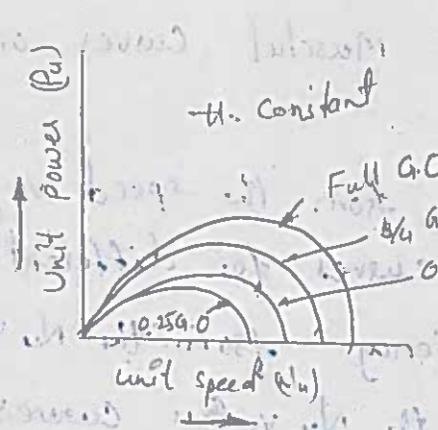
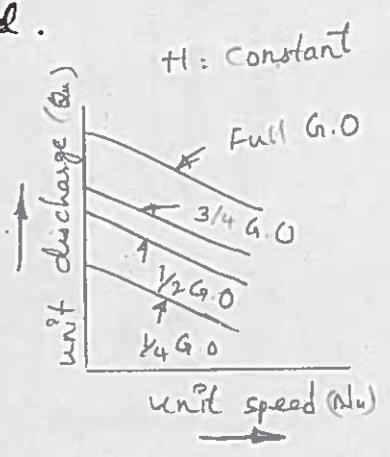
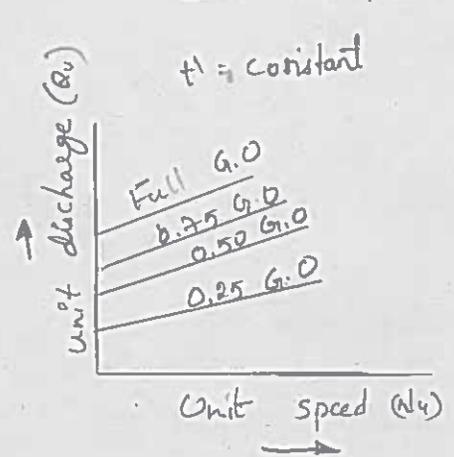
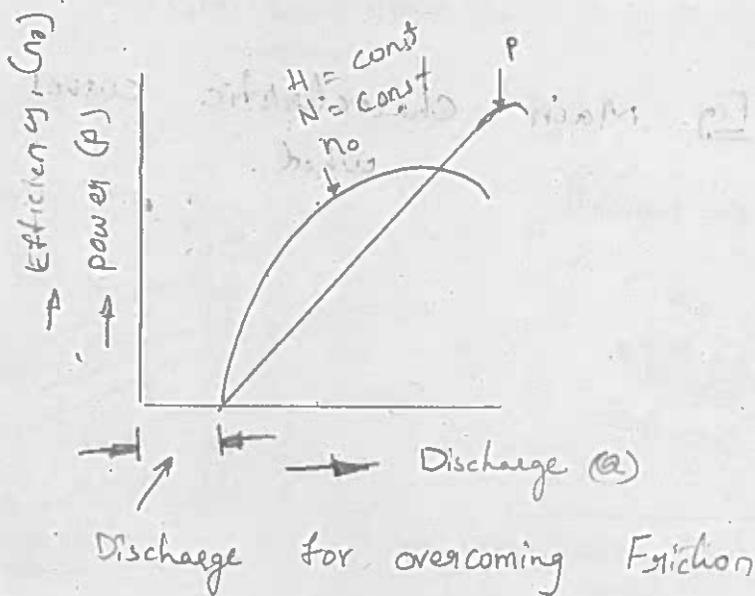


Fig Main characteristic curves for a pelton wheel.



* Operating characteristic curves or Constant speed curves:-

Operating characteristic curves are plotted when the speed on the turbine is constant. In case of turbines, the head and is generally constant. As mentioned in art 18.13, there are three independent parameters namely N , H and Q . For operating characteristics N and H are constant and hence the variation of power and efficiency with respect to discharge Q are plotted. The power curve for turbine shall not pass through the origin because certain amount of discharge will be needed to produce power to overcome initial friction. Hence the power and efficiency curves will be slightly away from the origin on the x-axis as to overcome initial friction certain amount discharge will be required. Shows the variation of power and efficiency with respect to discharge.



* Constant Efficiency curves or Muschel curves or

Iso-efficiency curves:-

These curves are obtained from the speed vs, efficiency and speed vs discharge curves for different gate openings. For a given efficiency from the N_u vs η curves, there are two speeds. From the N_u vs Q_u curves corresponding to two values of speeds there are two

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Hence, for a given efficiency there are two values of speeds. These are two values of discharge for a particular gate opening. This means for given efficiency there are two values of speeds and two values of the discharge for a given gate opening.

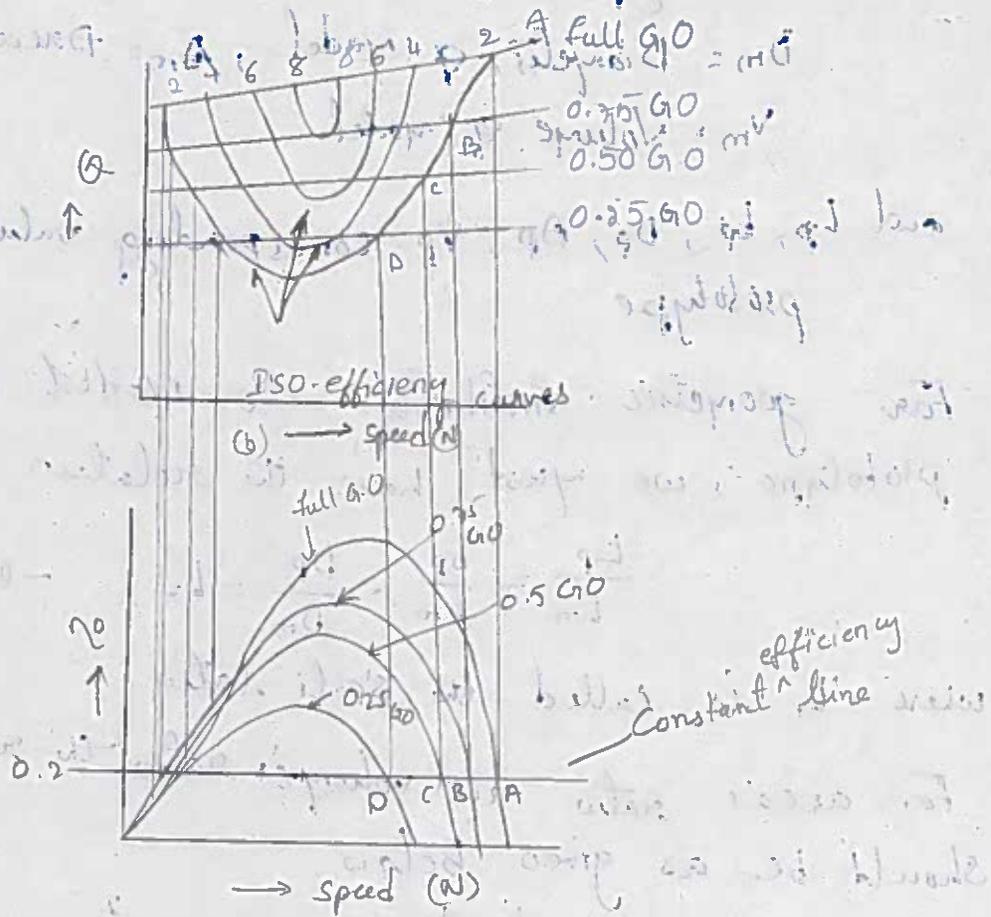


Fig Constant efficiency curve.

If the efficiency is maximum there is only one value. These two values of speed and two values of discharge corresponding to a particular gate opening are plotted as shown fig. The procedure is repeated for different gate openings and curves Q vs N are plotted. The points having the same efficiency are called iso-efficiency curves.

For plotting the iso-efficiency curves, horizontal lines representing the same efficiency are drawn on the η -speed curves.

(i) Geometric similarity :-

The geometric similarity is said to exist b/w the model and the prototype. The ratio of all corresponding linear dimensions in the model and prototype are equal.

l_m = length of model, b_m = breadth of model.

D_m = Diameter of model, A_m = Area of model.

V_m = Volume of model

and l_p, b_p, D_p, A_p, V_p = corresponding value of the prototype

For geometric similarity b/w Model and prototype, we must have the relation

$$\frac{l_p}{l_m} = \frac{b_p}{b_m} = \frac{D_p}{D_m} = L_r \quad \text{--- (1.6)}$$

where L_r is called the scale ratio

For area's ratio and volume's ratio the relations should be as given below.

$$\frac{A_p}{A_m} = \frac{l_p \times b_p}{l_m \times b_m} = L_r \times L_r = L_r^2 \quad \rightarrow 12.7$$

$$\frac{V_p}{V_m} = \left(\frac{l_p}{l_m}\right)^3 = \left(\frac{b_p}{b_m}\right)^3 = \left(\frac{D_p}{D_m}\right)^3 \quad \rightarrow 12.8$$

* Cavitation :-

Cavitation is defined as the phenomenon of formation of vapour bubbles of a flowing liquid in a region where the pressure of the liquid falls below its vapour pressure and the sudden collapsing of these vapour bubbles in a region of the liquid fall below its vapour pressure collapse a very high pressure is created.

The metallic surface, above which these vapour bubbles collapse, at very high these high pressures, which cause pitting action on the surface. These cavities are formed on the metallic surface and also considerable noise and vibration are produced.

Cavitation includes formation of vapour bubbles of flowing liquid and collapsing of the vapour bubbles. Formation of vapour bubbles of the flowing liquid take place only whenever the pressure in any region falls below vapour pressure. When the pressure of the flowing liquid is less than its vapour pressure, the liquid starts boiling and vapour bubbles are formed. These vapour bubbles are carried along with flowing liquid to high pressure zones where the vapour bubbles are carried along with the flowing collapse. Due to sudden collapsing of the bubbles on the metallic surface high pressure is produced and metallic surface are subjected to high local stresses. Thus the surface are damaged.

* Precaution Against Cavitation :-

The following precautions should be taken against Cavitation:

- (i) The pressure of the flowing liquid in any part of the hydraulic system should not be allowed to fall below its vapour pressure. D + the flowing

liquid is water, when then its absolute pressure head should not be below 2.5 m of water.

(ii) The special materials and coatings such as aluminum bronze and stainless steel which are cavitation resistant material, should be used.

Effects of Cavitation.

The following are the effects of cavitation.

(i) The metallic surface are damaged and cavities are formed on the surface.

(ii) Due to sudden collapse of vapour bubble considerable noise and vibrations are produced.

Hydraulic machines subjected to Cavitation:-

The hydraulic machines subjected to cavitation are reaction turbines and centrifugal pump.

* Cavitation in Turbines:-

In turbine, only reaction turbines are subjected to cavitation. In reaction turbine its cavitation may occur at the outlet of its runner or at the inlet of the draft.

Due to cavitation, its metal of below of the runner vanes and draft-tube is gradually eaten away, which result in lowering the efficiency of the turbine. Hence, the cavitation in a reaction turbine can be noted by a sudden drop in efficiency. In order to determine whether cavitation will occur in any portion of a reaction turbine, the critical value of Thomas' cavitation factor is calculated.

Thoma's Cavitation Factor for Reaction Turbine:-

Prof. D. Thoma suggested a dimensionless number, called after his name Thoma's Cavitation factor σ which can be used for determining the region where cavitation takes place in reaction turbines. The mathematical expression for the Thoma's Cavitation factor is given by

$$\sigma = \frac{H_b - H_s}{H} = \left(\frac{H_{atm} - H_v}{H} \right) - V_s \quad \text{--- (19.23)}$$

where,

H_b = Barometric pressure head in m of water

H_{atm} = Atmospheric pressure head in m of water

H_v = Vapour pressure head in m of water,

H_s = Suction pressure at the inlet of reaction turbine in m of water or height of turbine runner above the tail water surface

H = Net head on the turbine in m.

Moreover it has been found that σ depends on N_s of the turbine and for a turbine of particular N_s the factor σ can be reduced upto a certain value upto which its efficiency η_o remains constant. A further increase in the value of σ results in a sharp fall in η_o ... The value of σ at this turning point is called Critical Cavitation factor σ_c . The values of σ_c for different turbines may be determined with the help of the following empirical relationships.

For Francis turbines.

$$\sigma_c = 0.625 \left[\frac{N_s}{380.78} \right]^2 \quad \text{--- (22.22)}$$

For propeller turbines.

$$\sigma_c = 0.28 + \left[\frac{1}{7.5} \left(\frac{N_s}{380.78} \right)^3 \right] \quad \text{--- (22.23)}$$

For Kaplan turbines, values of σ_c obtained by eqn 22.23 should be increased by 10 percent. Unit-4; Pg-53/55

* Selection of turbines :-

The selection of a suitable type of turbine is usually governed by the following factors.

(i) Head and specific speed :- It has been found that there is a range of head and specific speed for which each type of turbine is most suitable which is given in Table 22.1.

Table 22.1

s.No	Head in meters	Type of turbine	Specific speed.
1	300 or more	Pelton wheel Single or multiple jet	8.5 to 47 (In SI units) 10 to 155 (in metric unit)
2	150 to 300	Pelton or Francis	20 to 85 (In SI unit) 35 to 100 (in metric unit)
3	60 to 150	Francis or Deriaz	85 to 188 (In SI unit) 100 to 200 (in metric unit)
4	Less than 60	Kaplan or propeller or Deriaz or Tubular	188 to 860 (In SI units) 220 to 1000 (metric units)

However, as a general rule it may be stated that as far as possible a turbine with highest permissible specific speed should be chosen, which will not only be the cheapest in itself but its relatively small size and high rotational speed will reduce the size of the generator as well as the power house. But the specific speed cannot be increased indefinitely, because higher specific speed turbine is generally more liable to cavitation. The cavitation may, however, be avoided by installing the turbine at a lower level with respect to the tail race.

* (ii) Part load operation:-

The turbine may be stated that as required to work with considerable load variations as the load deviates from normal working load the efficiency would also vary. If fig a plot turbines b/w η_0 and % of full load has been shown. At part load the performance of Kaplan and Pelton turbine is better in comparison to that of Francis and Propeller turbines. The variability of load will influence the choice of type of turbine if the head lies b/w 150 m to 300 m. For head below 30 m, Kaplan turbine is preferable for part load operation in comparison to Propeller turbine.

In addition to above mentioned factor there are certain other factors to be considered for the selection of the suitable type of a turbine.

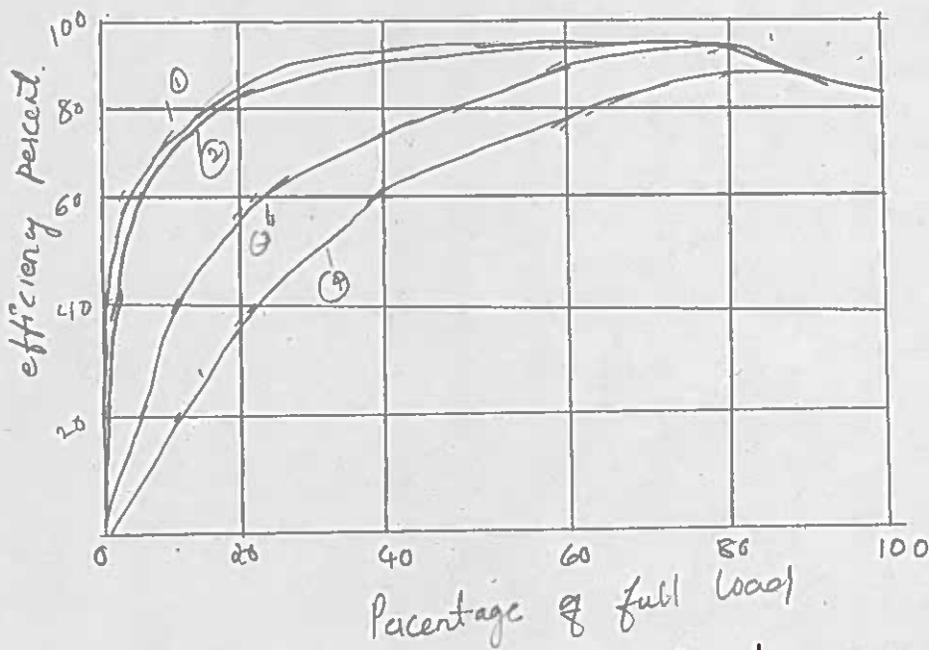


Fig Percentage of full load vs η_0 curves for different type of turbines.

The overall cost which includes the initial cost and the running cost should be considered. The cavitation characteristics of the turbine should also be considered since it affects the installation of a reaction turbine.

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