Unit – I:

Hydraulic Turbines

(Part 1)

Fluid Machinery

EMEC-3320

Classification Of Turbines

1. According to the type of energy at inlet :
   (*a*) Impulse turbine, and (*b*) Reaction turbine.
2. According to the direction of flow through runner :
   (*a*) Tangential flow turbine,
   (*b*) Radial flow turbine,
   (*c*) Axial flow turbine, and
   (*d*) Mixed flow turbine.
3. According to the head at the inlet of turbine :
   (*a*) High head turbine,
   (*b*) Medium head turbine, and
   (*c*) Low head turbine.
4. According to the specific speed of the turbine :
   (*a*) Low specific speed turbine,
   (*b*) Medium specific speed turbine, and
   (*c*) High specific speed turbine.

Classification Of Turbines

The main classification depends upon the type of action of the water on the turbine.

These are *: (i) Impulse Turbine (ii) Reaction Turbine*.

(i) Impulse turbine

In the case of impulse turbine all the potential energy is converted to kinetic energy in the nozzles. The impulse provided by the jets is used to turn the turbine wheel. The pressure inside the turbine is atmospheric. This type is found suitable when the available potential energy is high and the flow available is comparatively low.

Classification Of Turbines

(ii) **Reaction Turbine**

In reaction turbines the available potential energy is progressively converted in the turbines rotors and the reaction of the accelerating water causes the turning of the wheel.

These are again divided into (a) radial flow, (b) mixed flow and (c) axial flow machines.

Radial flow machines are found suitable for *moderate levels of potential energy and medium quantities of flow*. The axial machines are suitable for low levels of potential energy and large flow rates.

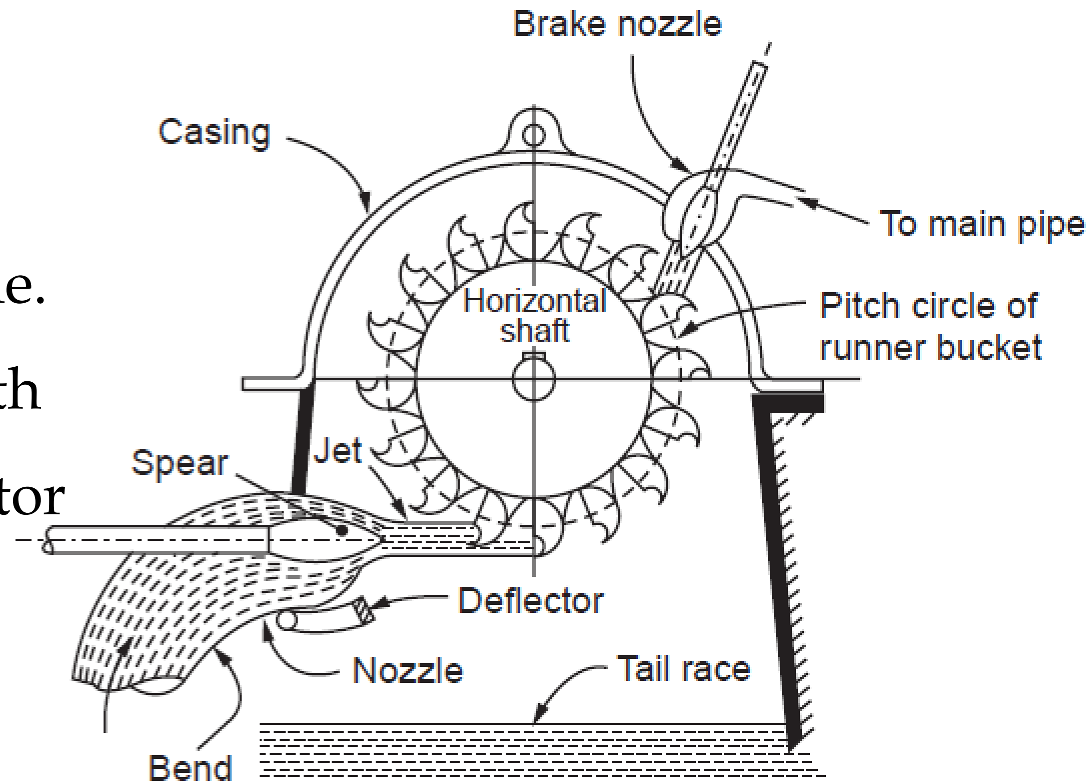
The potential energy available is generally denoted as “head available”. With this terminology plants are designated as “high head”, “medium head” and “low head” plants.

PELTON TURBINE

This is the only type used in high head power plants. This type of turbine was developed and patented by L.A. Pelton in 1889 and all the type of turbines are called by his name to honour him.

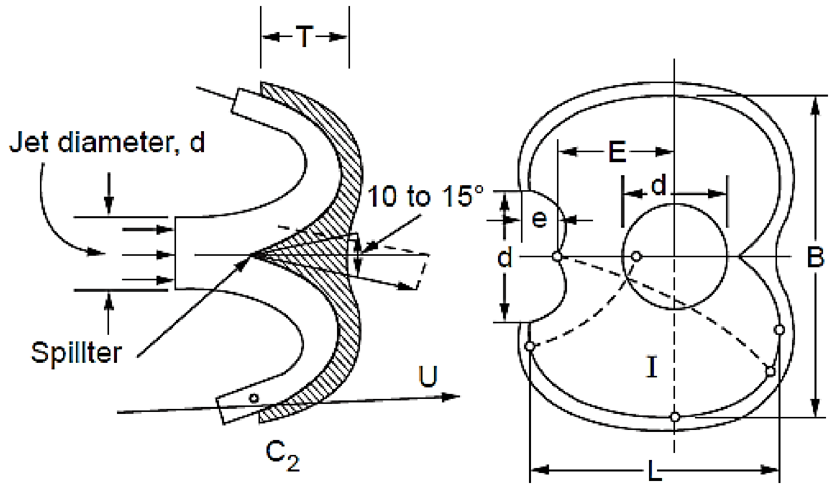
A sectional view of a horizontal axis Pelton turbine is shown in figure. The main components are:

1. The runner with the (vanes) buckets fixed on the periphery of the same.
2. The nozzle assembly with control spear and deflector
3. Brake nozzle and
4. The casing.



PELTON TURBINE (CONSTRUCTION)

The rotor or runner consists of a circular disc, fixed on suitable shaft, made of cast or forged steel. Buckets are fixed on the periphery of the disc. The spacing of the buckets is decided by the runner diameter and jet diameter and is generally more than 15 in number. These buckets in small sizes may be cast integral with the runner. In larger sizes it is bolted to the runner disc.



The buckets are formed in the shape of two half ellipsoids with a splilter connecting the two

PELTON TURBINE (CONSTRUCTION)

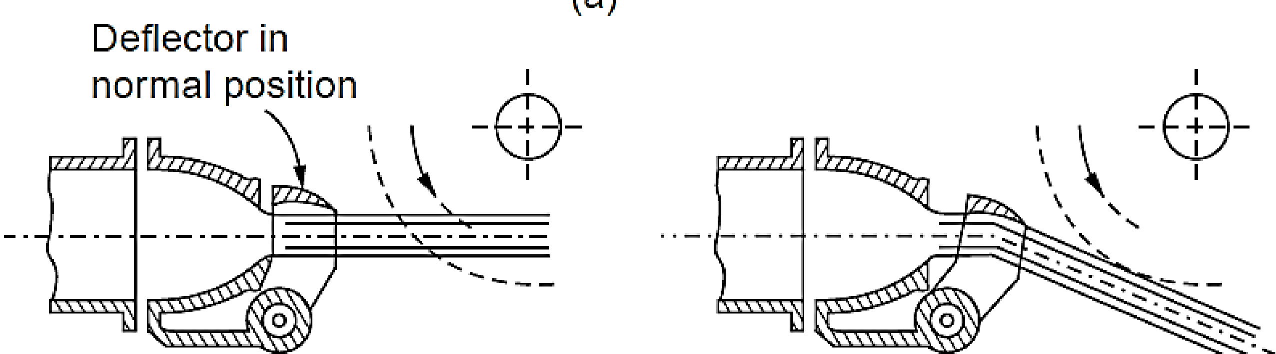
Equations are available to calculate the number of buckets on a wheel. The number of buckets, *Z*,

$$Z = (D/2d) + 15$$

where *D* is the runner diameter and *d* is the jet diameter.

D/d	B/d	L/d	T/d	Notch width
14 – 16	2.8 – 4	2.5 – 2.8	0.95	1.1 d + 5 mm

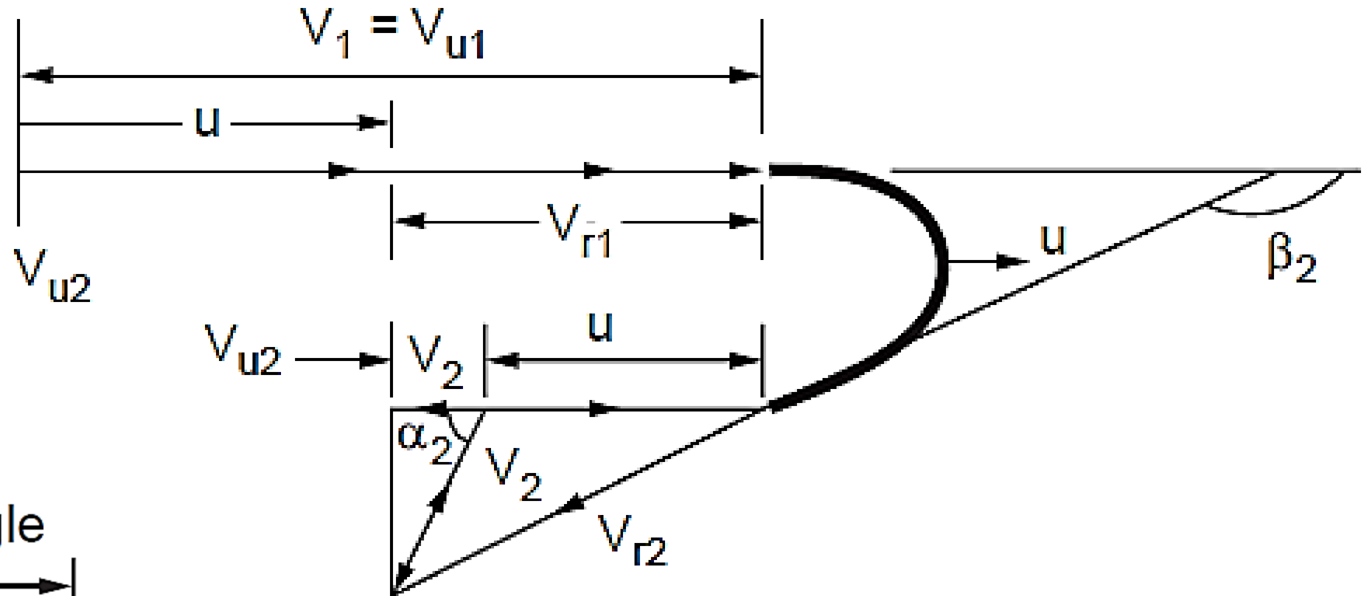
(a)

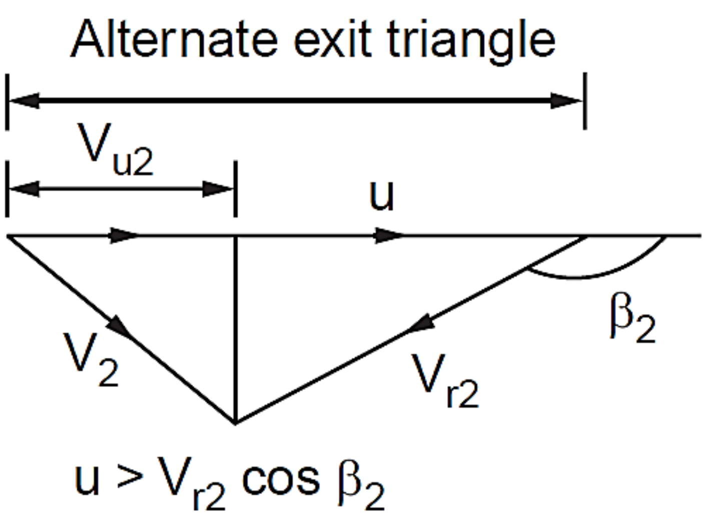


(b)

POWER DEVELOPMENT IN A PELTON TURBINE

The bucket splits the jet into equal parts and changes the direction of the jet by about 165°. Velocity diagram for Pelton turbine is shown in fig.





In this case the jet direction is parallel to the blade velocity or the tangential velocity of the runner.

Hence

$$V_{u1} = V_1$$

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

Power Development in a Pelton Turbine

In the ideal case: $V_{r2} = V_{r1}$

But due to friction, $V_{r2} = kV_{r1}$

and $u_2 = u_1$

Therefore,

$$F = \dot{m}\,(V_{u1} \pm V_{u2})$$

$$\tau = \dot{m}\,(V_{u1} \pm V_{u2})\, r$$

$$P = \dot{m}\,(V_{u1} \pm V_{u2})\, u$$

where $\dot{m}$ is given by ρAV at entry.

Hydraulic efficiency is given as:

$$\eta_h = \frac{\dot{m}\,(V_{u1} \pm V_{u2})\, u}{\dot{m}\, V_1^{\,2} / 2} = \frac{2u\,(V_{u1} \pm V_{u2})\, u}{V_1^{\,2}}$$

Once the effective head of turbine entry is known

$$V_1 = C_v\sqrt{2gH}$$

For various values of u, power developed & efficiency will be different.

Optimum Peripheral Speed for a Pelton Turbine

It is desirable to arrive at the optimum value of u for a given value of V_1. Hydraulic Efficiency Equation can be modified by using the following relations.

$$V_{u1} = V_1,\ V_{u2} = V_{r2} \cos \beta_2 - u = kV_{r1} \cos \beta_2 - u = k(V_1 - u) \cos \beta_2 - u$$

$$V_{u1} + V_{u2} = V_1 + k\,V_1 \cos \beta_2 - u \cos \beta_2 - u$$

$$= V_1 (1 + k \cos \beta_2) - u(1 + k \cos \beta_2)$$

$$= (1 + k \cos \beta_2)(V_1 - u)$$

Substituting in the hydraulic efficiency equation:

$$\eta_H = \frac{2u}{V_1^2} \times (1 + k \cos \beta_2)(V_1 - u)$$

$$= 2(1 + k \cos \beta_2)\left[\frac{u}{V_1} - \frac{u^2}{V_1^2}\right]$$

$\frac{u}{V_1}$ is called speed ratio and denoted as ϕ.

Optimum Peripheral Speed for a Pelton Turbine

Therefore,
$$\eta_H = 2(1 + k \cos \beta_2)\,[\phi - \phi^2]$$

To arrive at the optimum value of φ, this expression is differentiated with respect to φ and equated to zero.

$$\frac{d\eta_H}{d\phi} = 2(1 + k \cos \beta_2)\,(1 - 2\,\phi)$$

$$\boldsymbol{\phi = \frac{u}{V_1} = \frac{1}{2} \quad \text{or} \quad u = 0.5\, V_1}$$

Substituting this optimum condition in the expression of hydraulic efficiency we get,

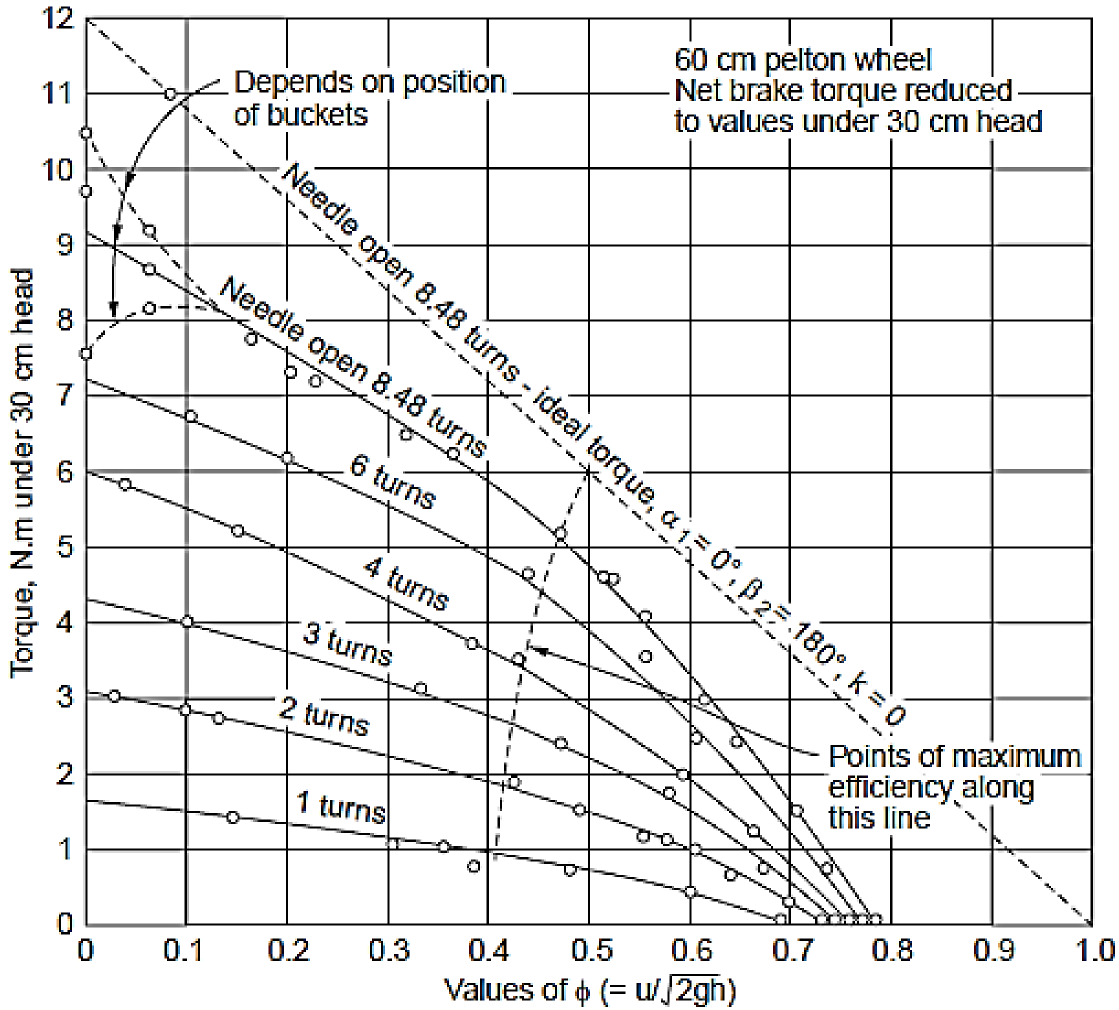
$$\eta_H = 2(1 + k \cos \beta_2)\,[0.5 - 0.5^2] \quad = \frac{1 + k \cos \beta_2}{2}$$

It may be seen that in the case k = 1 and β = 180°,

$$\eta_H = 1 \quad \text{or} \quad 100 \text{ percent.}$$

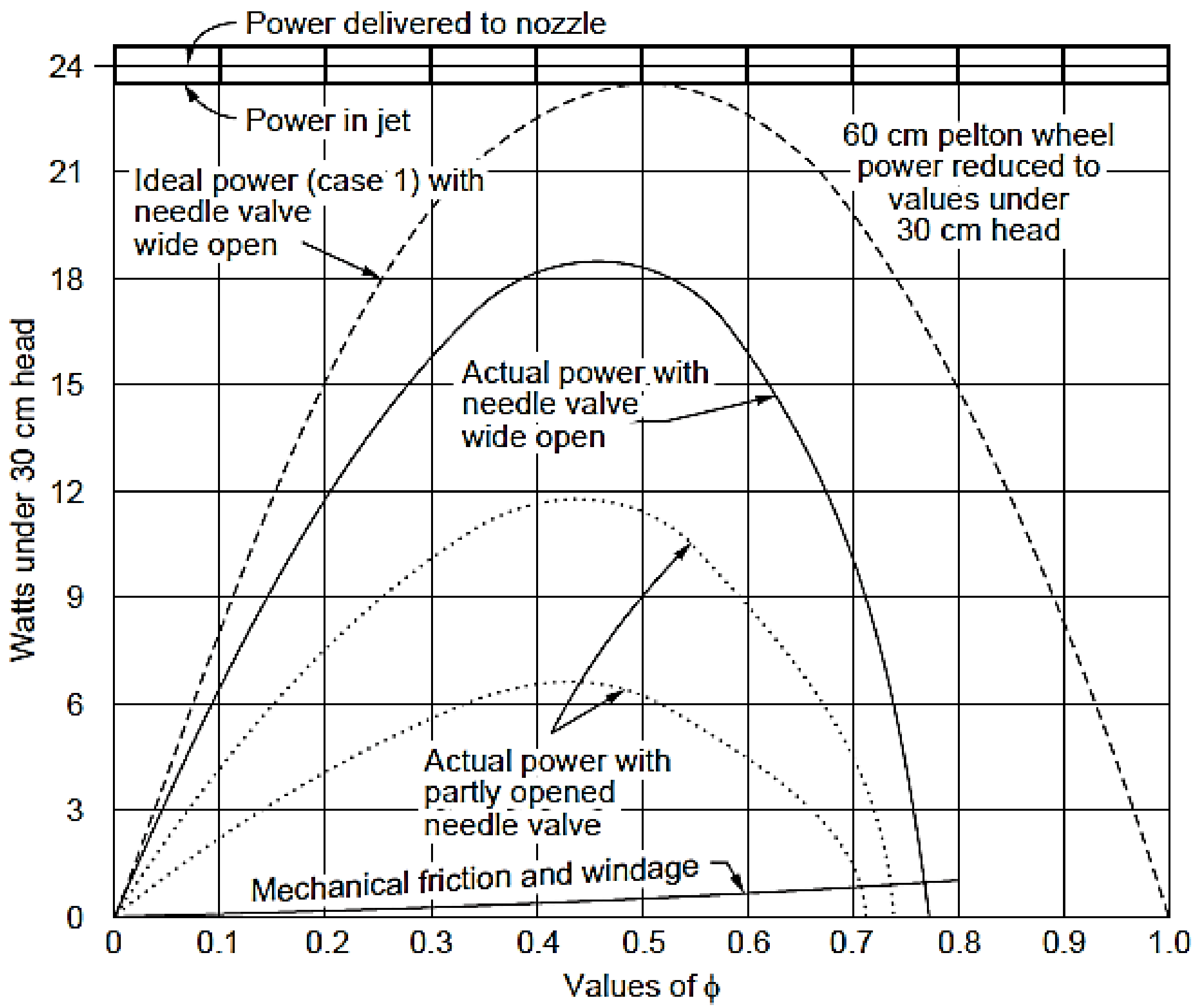
But the actual efficiency in well designed units lies between 85 and 90%.

Torque and Power and Efficiency Variation with Speed



The actual variation of torque with speed ratio is shown in figure. It is noted that the maximum efficiency lies in all cases between φ = 0.4 and 0.5. Also torque is found to be zero at values less than φ = 1. This is due to friction and exit loss ($V_2^2/2$) variation with various values of u.

THE POWER VARIATION FOR CONSTANT VALUE OF V_1 WITH Φ



The power can be calculated from the torque curves. In the ideal case power is zero both at φ = 0 and φ = 1. In the actual case power is zero even at φ is between 0.7 and 0.8. As the torque versus φ is not a straight line, the actual power curve is not a parabola.

DEFINITIONS OF HEADS AND EFFICIENCIES

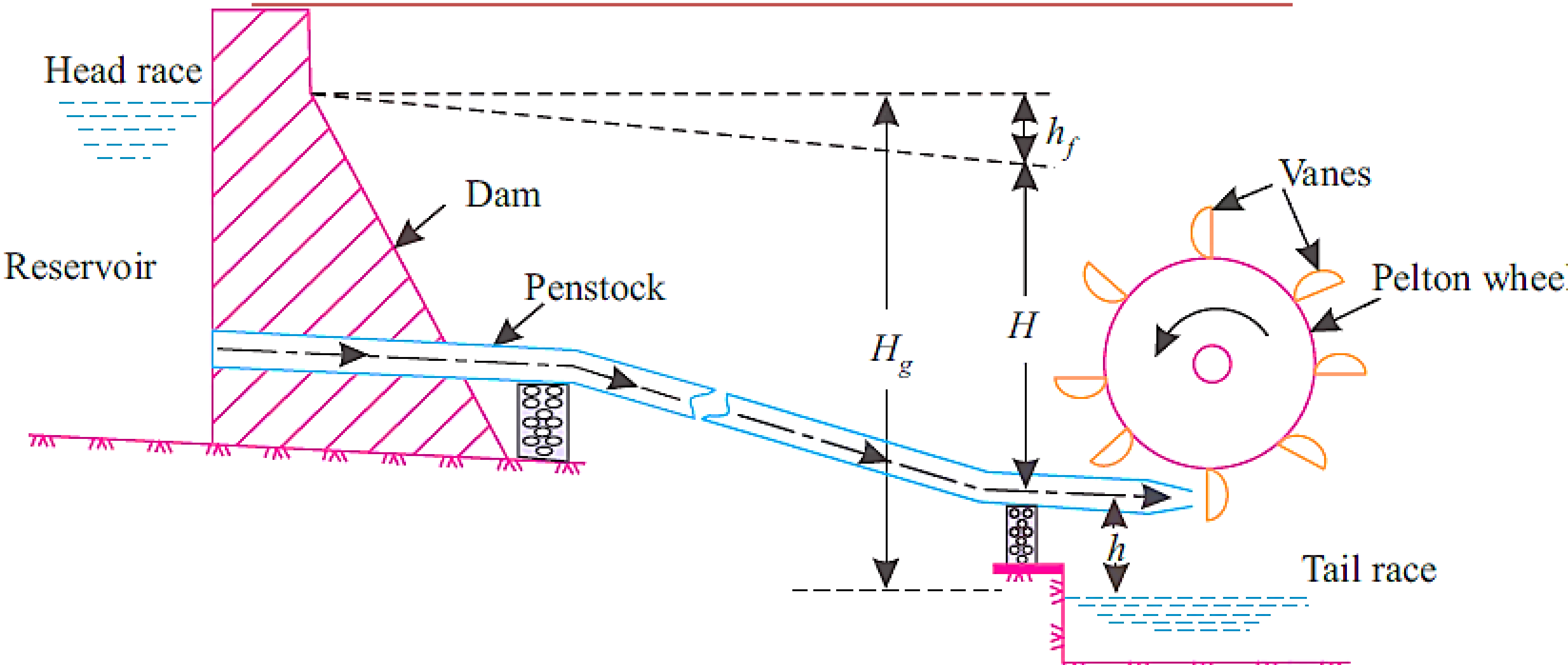


Fig. shows a general layout of a hydroelectric power plant using an impulse turbine (Pelton wheel).

1. Gross head. The gross (total) head is the difference between the water level at the reservoir (also known as the *head race*) and the water level at the tail race. It is denoted by H_g.

2. Net or effective head. The head available at the inlet of the turbine is known as net or effective head. It is denoted by *H* and is given by:

$$H = H_g - h_f - h$$

where, h_f is Total loss of head between the head race & entrance of turbine

DEFINITIONS OF HEADS

$$h_f = \frac{4fLV^2}{D \times 2g}$$

Where,
(L = length of penstock, D = diameter of penstock,
V = velocity of flow in penstock), and h = Height of nozzle above the tail race.

EFFICIENCIES

The following are the important efficiencies of turbine:

(i) Hydraulic efficiency (η_h).

It is defined as the ratio of power developed by the runner to the power supplied by the jet at entrance to the turbine.

Mathematically,

$$\eta_h = \frac{\text{Power developed by the runner}}{\text{Power supplied at the inlet of turbine}}$$

$$= \frac{\rho Q_a (V_{w1} \pm V_{w2})\, u}{wQ_a H} = \frac{\left(\frac{w}{g}\right) Q_a (V_{w1} \pm V_{w2})\, u}{wQ_a H}$$

$$= \frac{(V_{w1} \pm V_{w2})\, u}{gH} = \frac{H_r}{H}$$

EFFICIENCIES

The parameter,

$$H_r = \frac{1}{g} (V_{w1} + V_{w2})\, u$$

represents the energy transfer per unit weight of water and is referred to as the '*runner head*' or '*Euler head*'.

Hydraulic losses within the turbine $= \Delta H$

$$\Delta H = H - H_r$$

(ii) Mechanical efficiency (η_m)

It is defined as the ratio of the power obtained from the shaft of the turbine to the power developed by the runner. These two powers differ by the amount of mechanical losses, viz.,bearing friction, etc.

EFFICIENCIES

Mathematically, Mechanical efficiency (η_m)

$$\eta_m = \frac{\text{Power available at the turbine shaft}}{\text{Power developed by turbine runner}} = \frac{\text{Shaft power}}{\text{Bucket power}}$$

$$= \frac{P}{wQ_a\left(\dfrac{V_{w1}+V_{w2}}{g}\right)u} = \frac{P}{wQ_aH_r}$$

Values of mechanical efficiency for a Pelton wheel usually lie between 97 to 99 percent depending on size and capacity of the unit.

(iii) <u>Volumetric efficiency</u> (η_v)

The volumetric efficiency is the ratio of the volume of water actually striking the runner to the volume of water supplied by the jet to the turbine. That is, $\eta_v = \dfrac{\text{Volume of water actually striking the runner } (Q_a)}{\text{Total water supplied by the jet to the turbine } (Q)}$

$$\eta_v \simeq 0.97 \text{ to } 0.99.$$

(iv) Overall efficiency (η_o)

It is defined as the ratio of power available at the turbine shaft to the power supplied by the water jet. That is:

$$\eta_0 = \frac{\text{Power available at the turbine shaft}}{\text{Power available from the water jet}} = \frac{\text{Shaft power}}{\text{Water power}} = \frac{P}{wQH}$$

The values of overall efficiency for a Pelton wheel lie between 0.85 to 0.90. The individual efficiencies may be combined to give,

$$\eta_0 = \eta_h \times \eta_m \times \eta_v$$
$$= \frac{H_r}{H} \times \frac{P}{wQ_aH_r} \times \frac{Q_a}{Q} = \frac{P}{wQH}$$

If η_g is the efficiency of a generator, then power output of hydro-unit (turbine + hydro-generators) $= (wQH) \times \eta_0 \times \eta_g$

The product $\eta_0 \times \eta_g$ is known as hydroelectric plant efficiency.

Design Aspects of Pelton wheel

The following points should be considered while designing a Pelton wheel.

1. Velocity of jet.

The velocity of jet at inlet is given by,

$$V_1 = C_v \sqrt{2gH}$$

$$C_v = \text{Co-efficient of velocity } (= 0{\cdot}98 \text{ or } 0{\cdot}99), \text{ and}$$

$$H = \text{Net head on turbine.}$$

2. Velocity of wheel.

The velocity of wheel (u) is given by,

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

where, Ku = Speed ratio. It varies from *0.43 to 0.48*.

3. Angle of deflection of the jet.

The angle of deflection of the jet through the buckets is taken as 165° if no angle of deflection is given.

4. Mean diameter of the wheel (D).

The mean diameter or pitch diameter D of the Pelton wheel is given by,

$$u = \frac{\pi DN}{60} \text{ or } D = \frac{60\,u}{\pi N}$$

5. Jet Ratio.

It is defined as the ratio of pitch diameter (D) of the Pelton wheel to the diameter of the jet (d). It is denoted by *m* and is given as

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

6. Number of buckets.

The number of buckets on a runner is given by

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

where m = Jet ratio

7. Number of Jets.

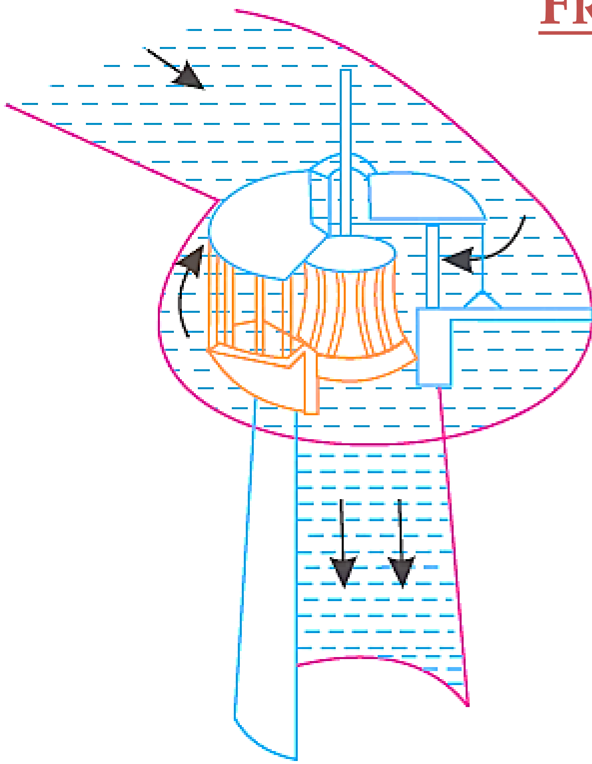
It is obtained by dividing the total rate of flow through the turbine by the rate of flow of water through a single jet.

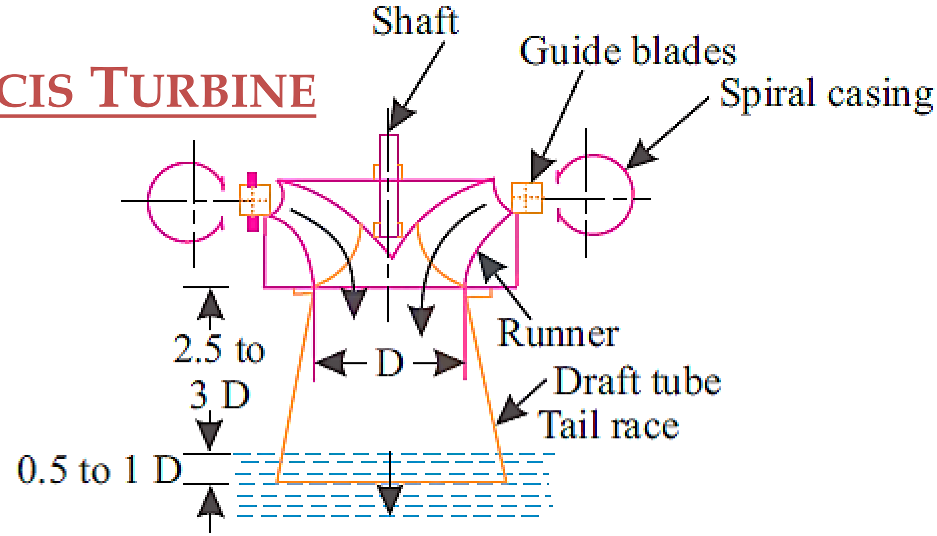
REACTION TURBINES

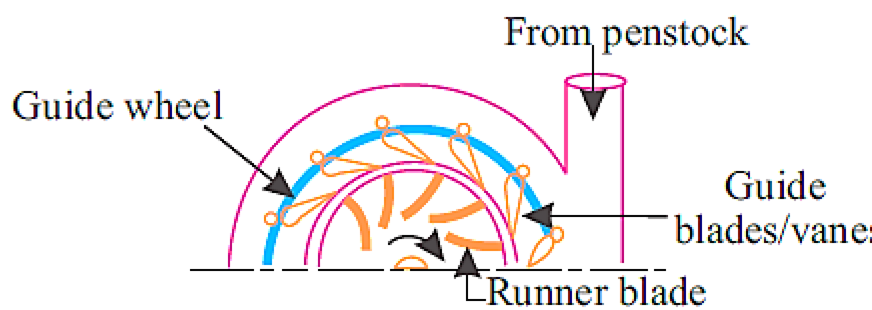
In reaction turbines, runner utilizes *both potential & kinetic energies*.

- As the water flows through the stationary parts of the turbine, *whole of its pressure energy is not transformed to kinetic energy* and when the water flows through the moving parts, there is a change both in pressure and in the direction and velocity of flow of water.
- As the water gives up its energy to the runner, both its pressure and absolute velocity get reduced. The water which acts on the runner blades is under a pressure above atmospheric and the runner passages are always completely filled with water.
- Important reaction turbines are *Francis, Kaplan and Propeller*.

FRANCIS TURBINE







The Francis turbine *operates under medium heads and also requires medium quantity of water*. It is employed in the medium head power plants. This type of turbine covers a wide range of heads.

The head acting on the turbine is *partly transformed into kinetic energy and the rest remains as pressure head*. There is a difference of pressure between the guide vanes and the runner which is called the reaction pressure and is responsible for the motion of the runner.

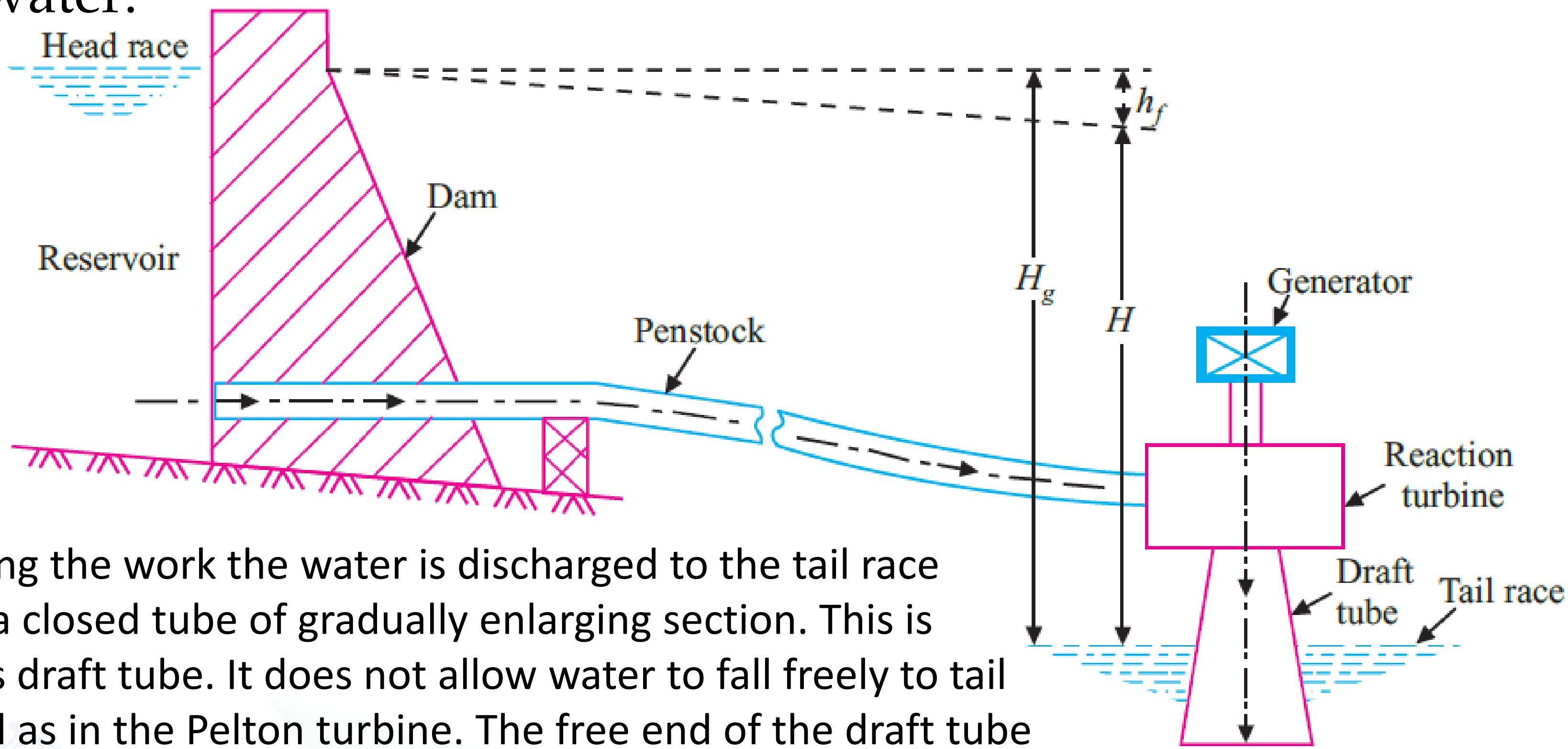
Francis Turbine : Main Parts

The main parts of a Francis turbine are:

1. **Penstock:** It is a large size conduit which conveys water from the upstream of the dam/reservoir to the turbine runner.
2. **Spiral/scroll casing:** It constitutes a closed passage whose cross-sectional area gradually decreases along the flow direction, area is maximum at inlet and nearly zero at exit.
3. **Guide vanes/wicket gates:** These vanes direct the water onto the runner at an angle appropriate to the design. The motion to them is given by means of a hand wheel or automatically by a governor.
4. **Governing mechanism:** It changes the position of the guide blades/vanes to affect a variation in water flow rate, when the load conditions on the turbine change.
5. **Runner and runner blades:** The driving force on the runner is both due to impulse and reaction effects; The number of runner blades usually varies between 16 to 24.
6. **Draft tube:** It is a gradually expanding tube which discharges water, passing through the runner, to the tail race.

LAYOUT OF HYDRO-ELECTRIC POWER PLANT WITH FRANCIS TURBINE

In Francis turbine the pressure at inlet is more than that at the outlet. This means that the water in the turbine must flow in a closed conduit. Unlike the Pelton type, where the water strikes only a few of the runner buckets at a time, in the Francis turbine the runner is always full of water.



After doing the work the water is discharged to the tail race through a closed tube of gradually enlarging section. This is known as draft tube. It does not allow water to fall freely to tail race level as in the Pelton turbine. The free end of the draft tube is submerged deep in tail water making, thus, the entire water passage, right from the head race up to the tail race, totally enclosed.

Important differences between Inward and Outward Flow Reaction Turbines

S.No.	Aspects	Inward flow reaction turbine	Outward flow reaction turbine
1	*Entry of water*	Water enters at the outer periphery, flows inward and towards the centre of the turbine and discharges at the outer periphery.	Water enters at the inner periphery flows outward and discharges at the outer periphery.
2	*Centrifugal head imparted*	Negative (negative centrifugal head reduces the relative velocity of water at the outlet).	Positive (Positive centrifugal head increases the relative velocity of water at the outlet).
3	*Discharge*	Does not increase.	The discharge increases.
4	*Speed control*	Easy and effective.	Very difficult.
5	*Tendency of the wheel to race*	Nil. The turbine adjusts the speed by itself.	If the turbine speed increases the wheel tends to race; the turbine cannot adjust the speed by itself.
6	*Suitability*	Quite suitable for medium high heads; best suitable for large outputs and units.	Quite suitable for low or medium heads.
7	*Application*	For power projects.	Practically obsolete.

WORK DONE AND EFFICIENCY OF FRANCIS TURBINE

Net head at the turbine runner:

$$H = H_g - h_f$$

$$H = \begin{bmatrix} \text{Total energy available at exit} \\ \text{from the penstock} \end{bmatrix} - \begin{bmatrix} \text{total energy available at exit} \\ \text{from the draft tube} \end{bmatrix}$$

$$= \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{penstock}} - \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{draft tube}}$$

If the draft tube exit is at tail race level, and the datum is also taken at that level, then,

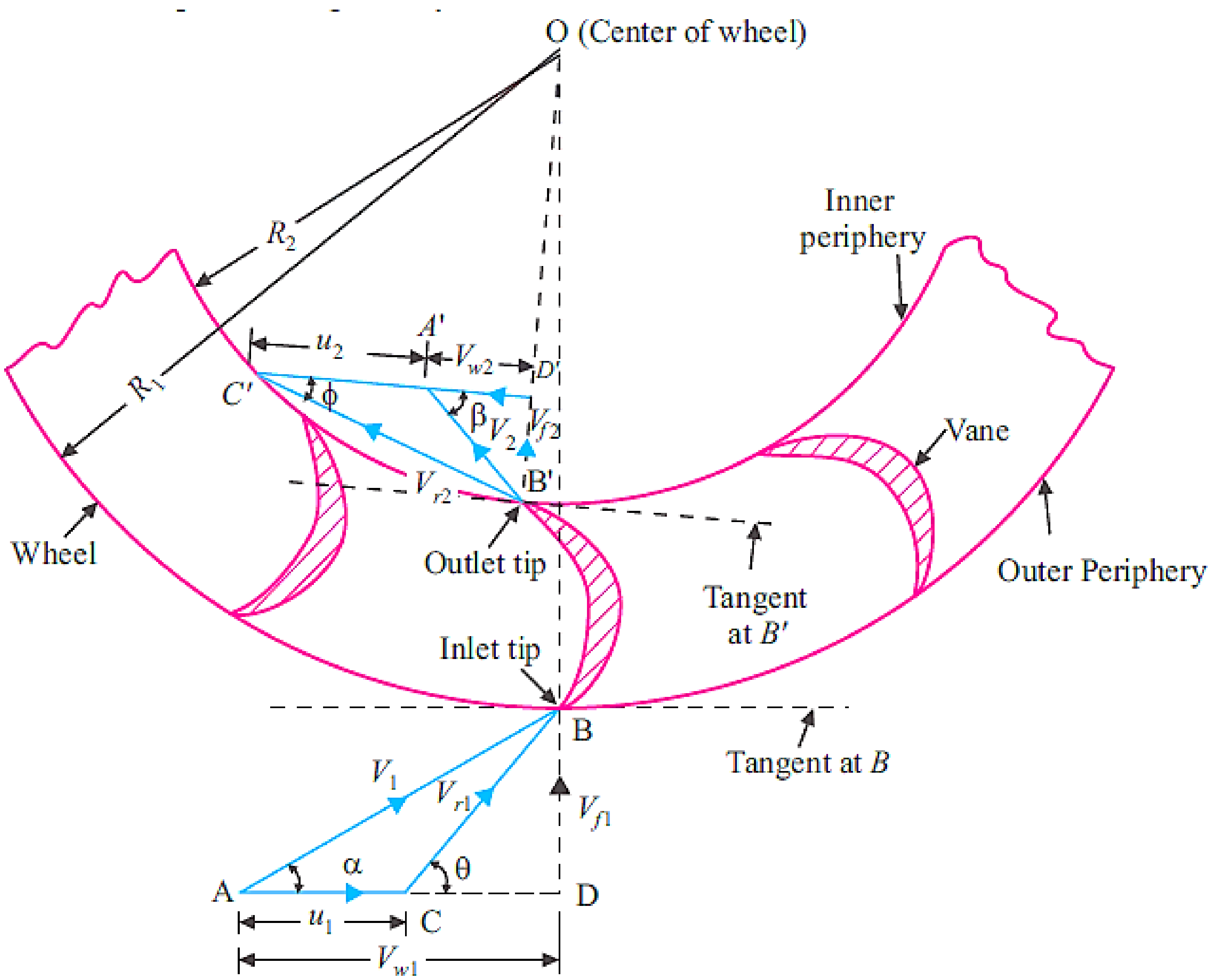
$$H = \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)_{\text{penstock}} - \frac{V_d^2}{2g}$$

(where, V_d = velocity at the exit of the draft tube)

Neglecting the velocity at the draft tube exit (V_d), we have:

$$H = \left(\frac{p}{w} + \frac{V^2}{2g} + z\right)$$

WORK DONE BY THE RUNNER



Velocity diagrams for an inward flow reaction turbine.

Work done by the runner

$$\text{Work done} = \rho Q\,(V_{w1}\,u_1 \pm V_{w2}\,u_2)$$

$$= \frac{wQ}{g}\,(V_{w1}\,u_1 \pm V_{w2}\,u_2)$$

where, Q = Discharge through the runner, m^3/s.

The maximum output under given conditions is obtained when

$$V_{w2} = 0$$

Thus, the maximum work done is given by,

$$\text{Work done} = \frac{wQ}{g}\,(V_{w1}\,u_1)$$

This discharge in this case is radial. For radial discharge, the absolute velocity at exit is radial.

HYDRAULIC EFFICIENCY

If H is the net head, then input to the turbine = $w\ Q\ H$

$$\eta_h = \frac{\text{Power developed by the runner}}{\text{Power supplied to the turbine (water power)}} = \frac{\frac{wQ}{g}(V_{w1}\ u_1)}{wQH}$$

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

However, if the velocity of whirl at the exit is zero, then

$$\eta_h = \frac{V_{w1}\ u_1 \pm V_{w2}\ u_2}{gH}$$

The hydraulic efficiency of the Francis turbine varies from 85 to 90%.

Mechanical And Overall Efficiency

Mechanical efficiency, η_m :

The mechanical efficiency is given by:

$$\eta_m = \frac{\text{Shaft power (P)}}{\text{Power developed by the runner}}$$

Overall efficiency, η_0 :

The overall efficiency is given as:

$$\eta_0 = \frac{\text{Shaft water}}{\text{Water power}} = \frac{P}{wQH}$$

and,

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

The overall efficiency varies from 80 to 90 percent.

WORKING PROPORTIONS OF A FRANCIS TURBINE

The following working proportions pertain to a Francis turbine :

1. Ratio of width to diameter B/D:

The value of (B/D) varies from *0.10 to 0.45*.

2. Flow ratio (K_f):

Flow ratio is the ratio of the velocity of flow at inlet to the theoretical jet velocity. Thus, Flow ratio, $K_f = \frac{V_{f1}}{\sqrt{2gH}}$

The value of Kf varies from *0.15 to 0.30*.

3. Speed ratio (K_u):

Speed ratio is the ratio of the peripheral speed at inlet to the theoretical jet velocity.

$$\text{Speed ratio, } K_u = \frac{u}{\sqrt{2gH}}$$

The value of K_u ranges from 0.6 to 0.9.

ADVANTAGES /DISADVANTAGES OF FRANCIS TURBINE W.R.T. PELTON

Advantages :

The Francis turbine claims following advantages over Pelton wheel :

1. In Francis turbine the variation in the operating head can be more easily controlled.

2. In Francis turbine the ratio of maximum and minimum operating heads can be even two.

3. The operating head can be utilized even when the variation in the tail water level is relatively large when compared to the total head.

4. The mechanical efficiency of Pelton wheel decreases faster with wear than Francis turbine.

5. The size of the runner, generator and power house required is small and economical if the Francis turbine is used instead of Pelton wheel.

Advantages and disadvantages of a Francis turbine over a Pelton wheel

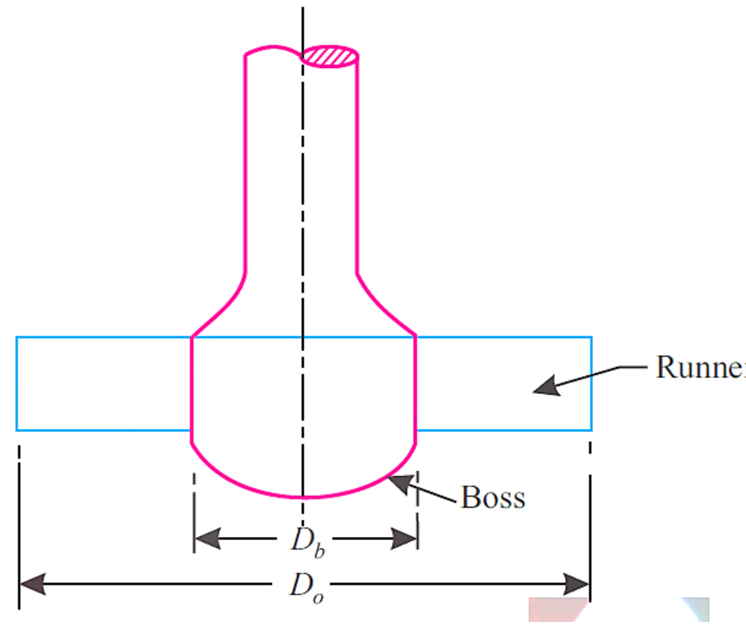
Disadvantages :

As compared with Pelton wheel, the Francis turbine has the following drawbacks/ shortcomings:

1. Water which is not clean can cause very rapid wear in high head Francis turbine.
2. The overhaul and inspection is much more difficult comparatively,
3. Cavitation is an ever-present danger.
4. The water hammer effect is more troublesome with Francis turbine.
5. If Francis turbine is run below 50 percent head for a long period it will not only lose its efficiency but also the cavitation danger will become more serious.

AXIAL FLOW REACTION TURBINES

- It has been observed that with increasing specific speed the flow tends to be axial.
- If water flows parallel to the axis of the rotation of the shaft, the turbine is known as *axial flow turbine*;
- The shaft of an axial flow reaction 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 it acts as runner for axial flow reaction turbine.



PROPELLER AND KAPLAN TURBINES-AXIAL FLOW REACTION TURBINES

Two important axial flow reaction turbines are:

(i) Propeller turbine, and

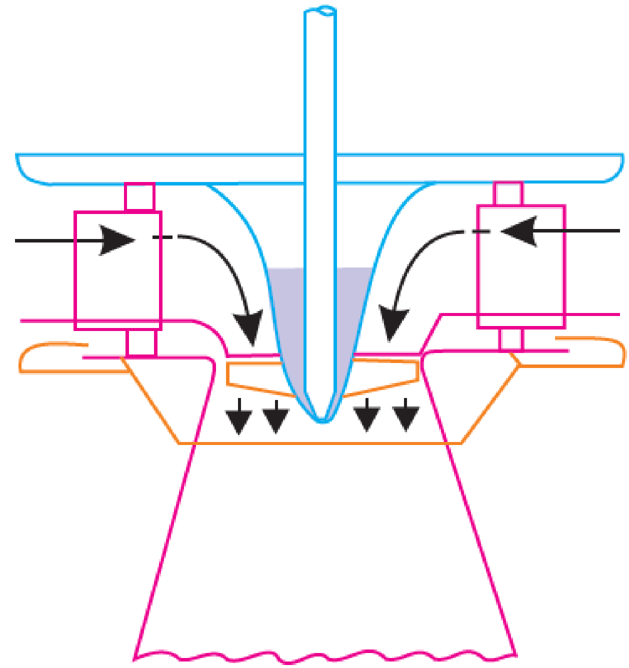
(ii) Kaplan turbine.

- In these turbines all parts such as spiral casing, stay vanes, guide vanes, control vanes, and draft tube are similar to mixed-flow turbines in design.
- But the water enters the runner in an axial direction and during the process of energy transfer, it travels across the blade passage in axial direction and leaves axially.
- The pressure at the inlet of the blades is larger than pressure at the exit of the blades. Energy transfer is due to the reaction effect, i.e. the change in the magnitude of relative velocity across the blades.

Some Aspects of Design of Axial Flow Reaction Turbines

- In an axial flow turbine the number of blades are fewer and hence the loading on the blade is larger. Smaller contact area causes less frictional loss compared to mixed flow turbines, but the peripheral speed of the turbine is larger.
- Axial flow rotors do not have a rim at the outer end like the Francis rotors; but the blades are enclosed in a cylindrical casing.
- Tip clearance between the blades and the cylindrical casing is small; hence the flow past blades can be considered two-dimensional. The water coming out from the guide vanes undergoes a whirl which is assumed to satisfy the law of free vortex ($V_w = C/r$). Accordingly the whirl is largest near the hub and smallest at the outer end of blade. *Hence the blade is twisted along its axis.*

PROPELLER TURBINE



The propeller turbine is a reaction turbine used for heads between 4 m and 80 m. It is purely axial-flow device providing the largest possible flow area that will utilize a large volume of water and still obtain flow velocities which are not too large.

The propeller turbine consists of an axial-flow runner with four to six or at the most ten blades of air-foil shape. The runner is generally kept horizontal, *i.e* the shaft is vertical. The blades resemble the propeller of a ship. In the propeller turbine, as in Francis turbine, the runner blades *are fixed* and *non-adjustable*. The *spiral casing* and *guide blades* are similar to those in Francis turbine.

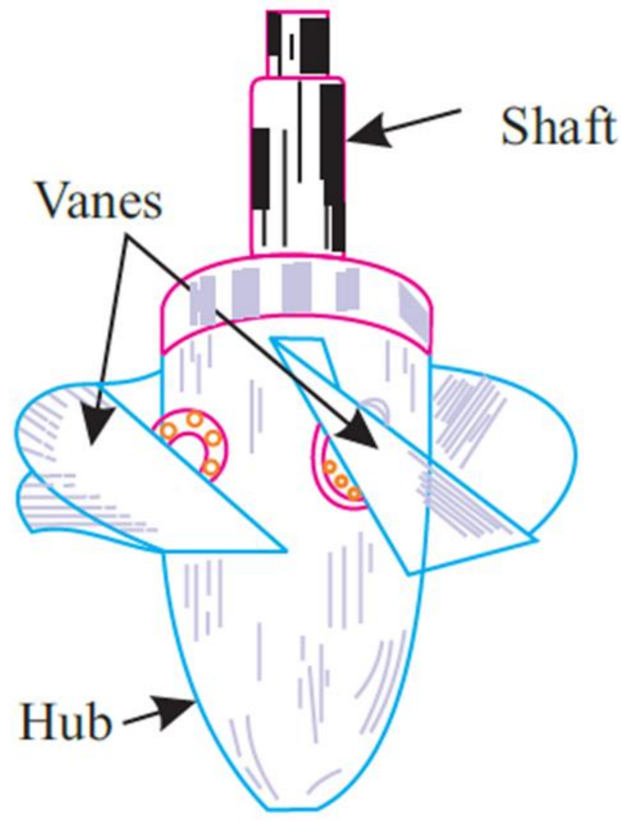
KAPLAN TURBINE

A propeller turbine is quite suitable when the load on the turbine remains constant. At part load its efficiency is very low; since the blades are fixed, the water enters with shock (at part load) and eddies are formed which reduce the efficiency.

This defect of the propeller turbine is removed in Kaplan turbine.

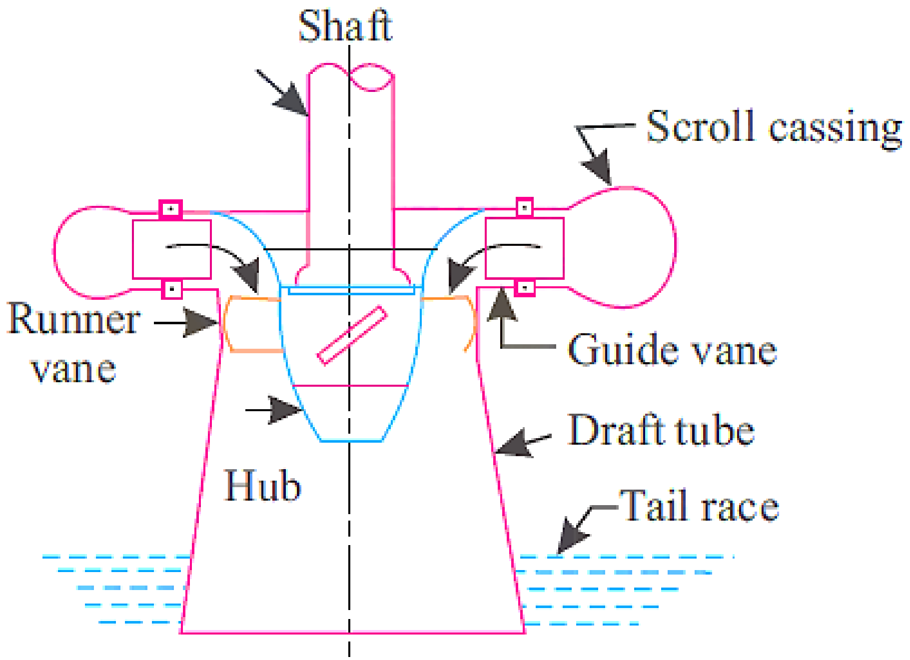
In a Kaplan turbine the runner blades are adjustable and can be rotated about pivots fixed to the boss of the runner.

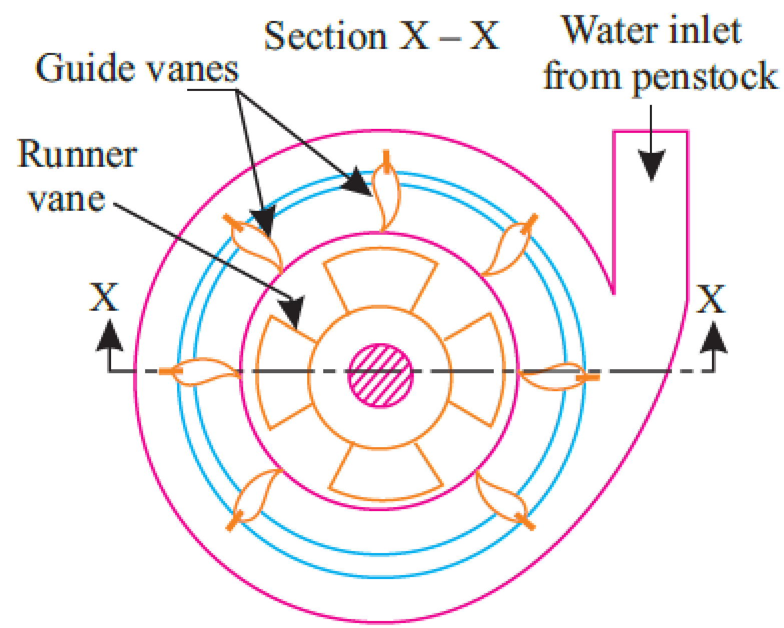
The Kaplan turbine has purely axial flow. Usually it has 4-6 blades having no outside rim.



KAPLAN TURBINE

The blades are adjusted automatically by servomechanism so that at all loads the flow enters them without shock. Thus, a high efficiency is maintained even at part load. The servomotor cylinder is usually accommodated in the hub.





Kaplan is also known as a variable-pitch propeller turbine since the pitch of the turbine can be changed because of adjustable vanes. The Kaplan turbine behaves like a propeller turbine at full-load conditions.

WORKING PROPORTIONS: KAPLAN TURBINE

The expressions for work done, efficiency & power developed by axial flow propeller & Kaplan turbines are identical to those of a Francis turbine, & the *working proportions* are obtained in an identical fashion. However, the following deviations need to be noted carefully:

1. In case of a propeller/Kaplan turbine, the ratio n is taken as

$$\frac{D_b}{D_0}\left(\text{and not } \frac{B}{D}\right)$$

where, D_0 = Outside diameter of the runner, and

D_b = Diameter of boss (or hub).

Discharge, Q = Area of flow × velocity of flow

$$= \frac{\pi}{4}(D_0^2 - D_b^2) \times V_f$$

$$= \frac{\pi}{4}(D_0^2 - D_b^2) \times K_f\sqrt{2gH} \qquad \text{(where, } K_f = \text{flow ratio)}$$

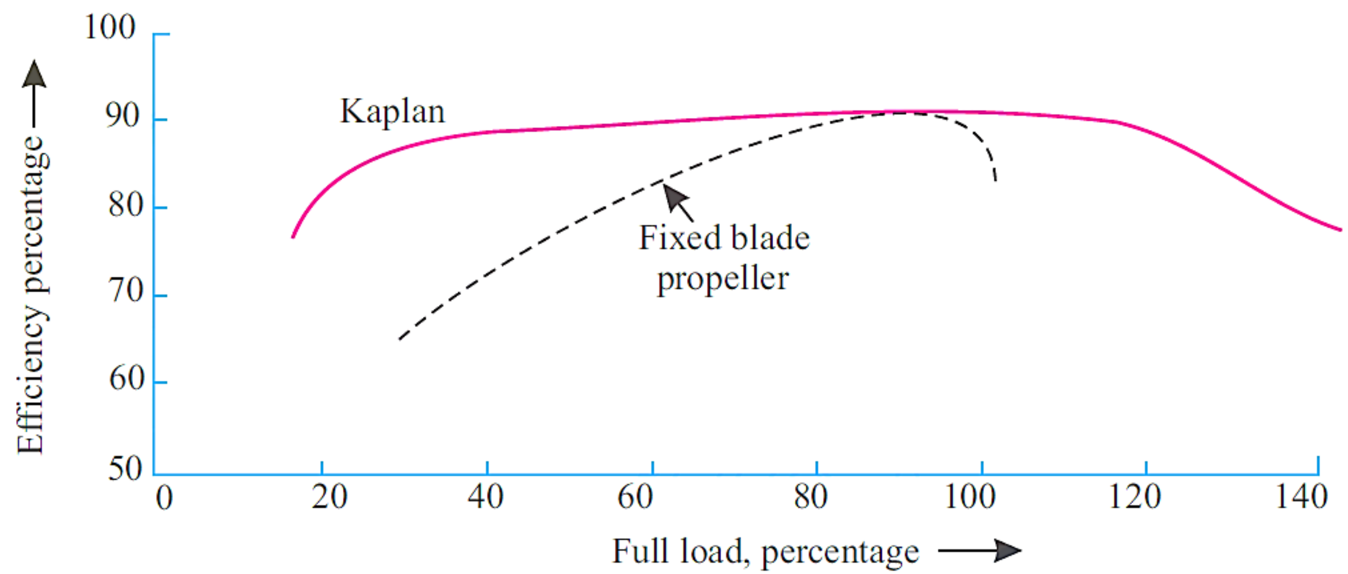
or, $$Q = \frac{\pi}{4} D_0^2\,(1 - n^2) \times K_f\sqrt{2gH}$$

WORKING PROPORTIONS: KAPLAN TURBINE

2. The peripheral velocity u of the runner vanes depends upon the radius of the point under consideration and thus the blade angles vary from the rim to the boss and the vanes are warped; this is necessary to ensure shock free entry and exit.

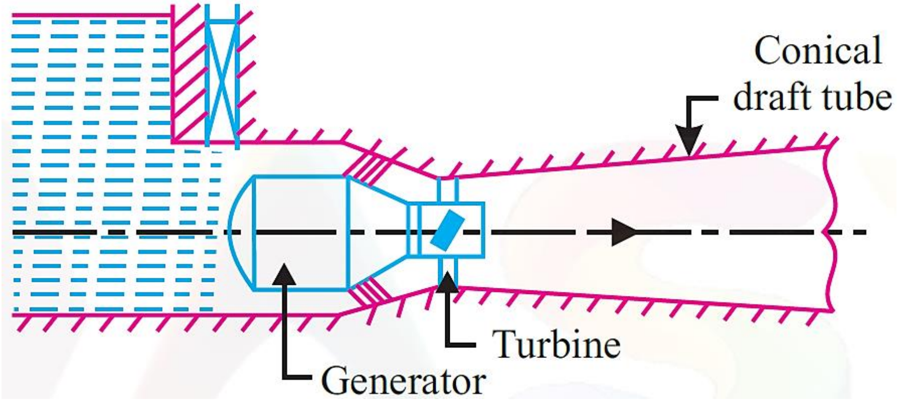
3. The velocity of flow remains constant throughout.

Fig. shows the comparison of efficiencies of propeller (fixed blades) and Kaplan turbine.



Tubular Or Bulb Turbines

The electric generator coupled to the Kaplan turbine is enclosed and works inside a straight passage having the shape of a bulb. The water tight bulb is submerged directly into a stream of water, and the bends at inlet to casing, draft tube, etc. which are responsible for the loss of head are dispensed with. The unit then needs less installation space with a consequent reduction in excavation and other civil engineering works. These turbines are referred to as tubular or bulb turbines. The tubular turbine, a modified axial flow turbine, was developed in Germany by Arno Fischer in 1937. The economical harnessing of fairly low heads on major rivers is now possible with high-output bulb turbines.



Features of Tubular Or Bulb Turbines

The following features are worth noting :

- A tubular bulb turbine is an axial flow turbine with either adjustable or non-adjustable runner vanes (and hence similar to Kaplan or propeller turbines).
- In such a turbine the scroll casing is not provided but the runner is placed in a tube extending from head water to the tail water (and hence it is called tubular turbine).
- It is a low head turbine and is employed for heads varying from 3 to 15 m.
- The disposition of shaft in a tubular turbine may be vertical, or inclined or horizontal.
- The turbo-generator set using tubular turbine has an outer casing having the shape of a bulb. Such a set is now termed as bulb set and the turbine used for the set is called a bulb turbine