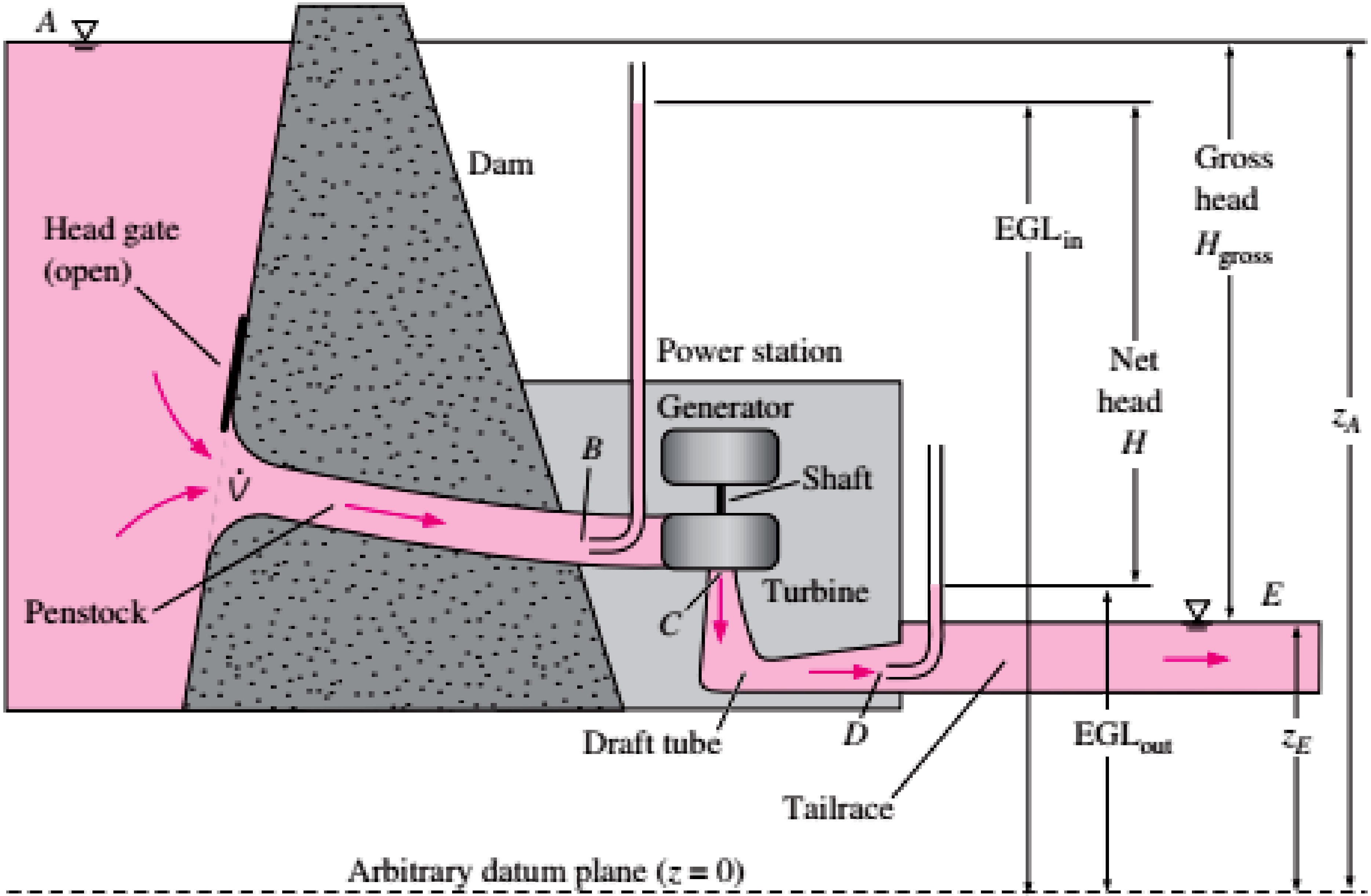
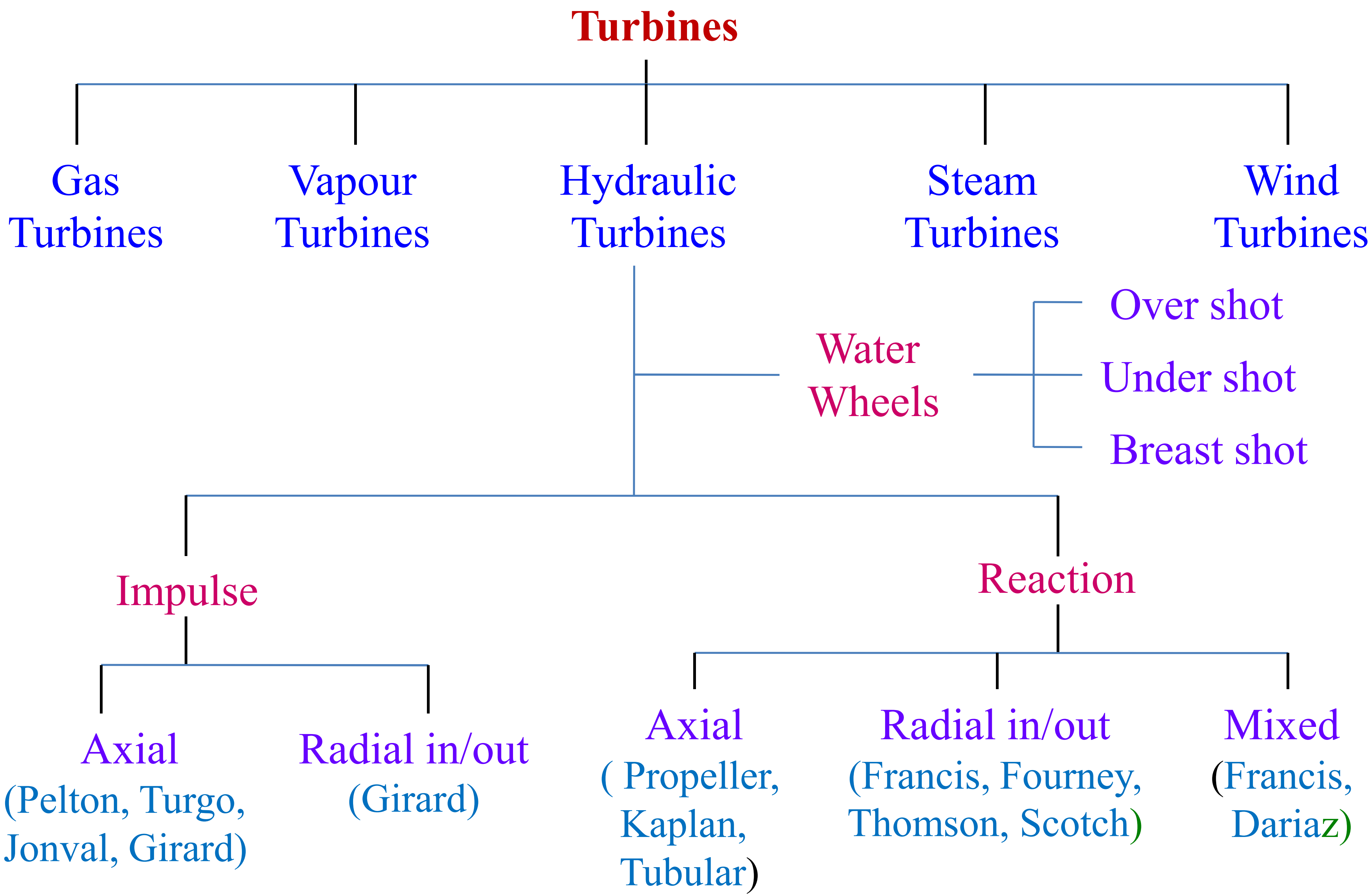
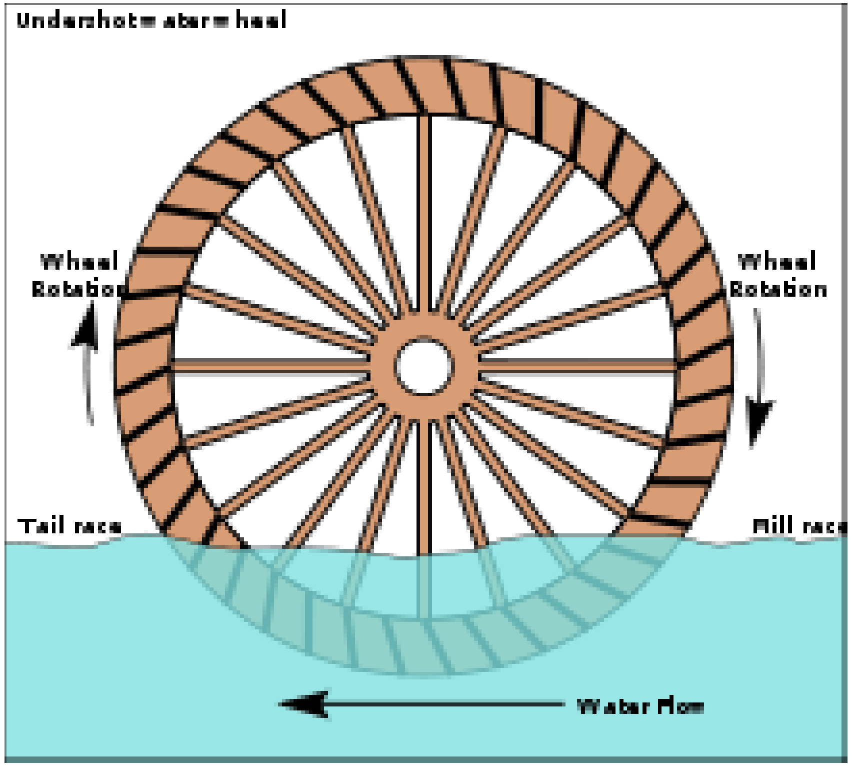
Fluid Machines



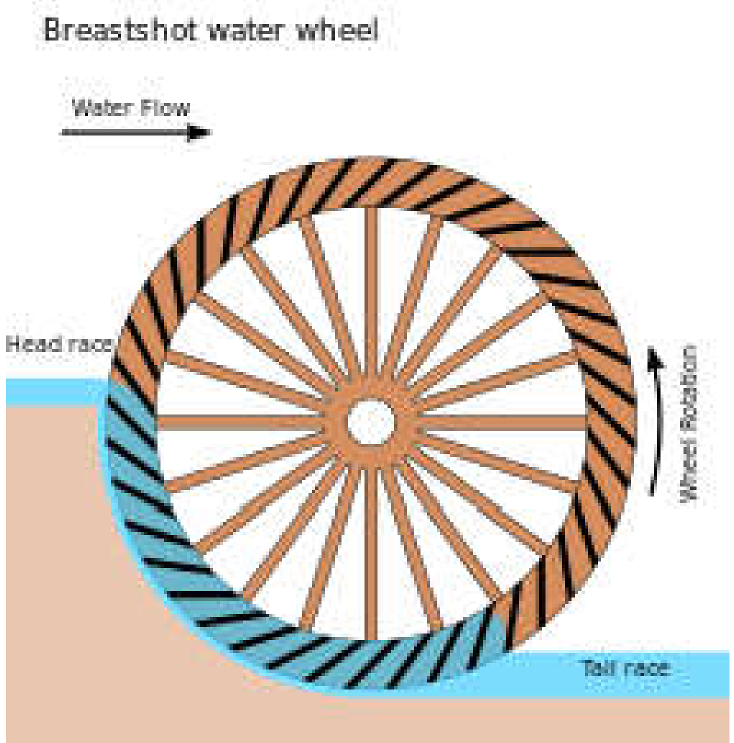
Turbine

Turbines are devices that extract energy from a flowing fluid.

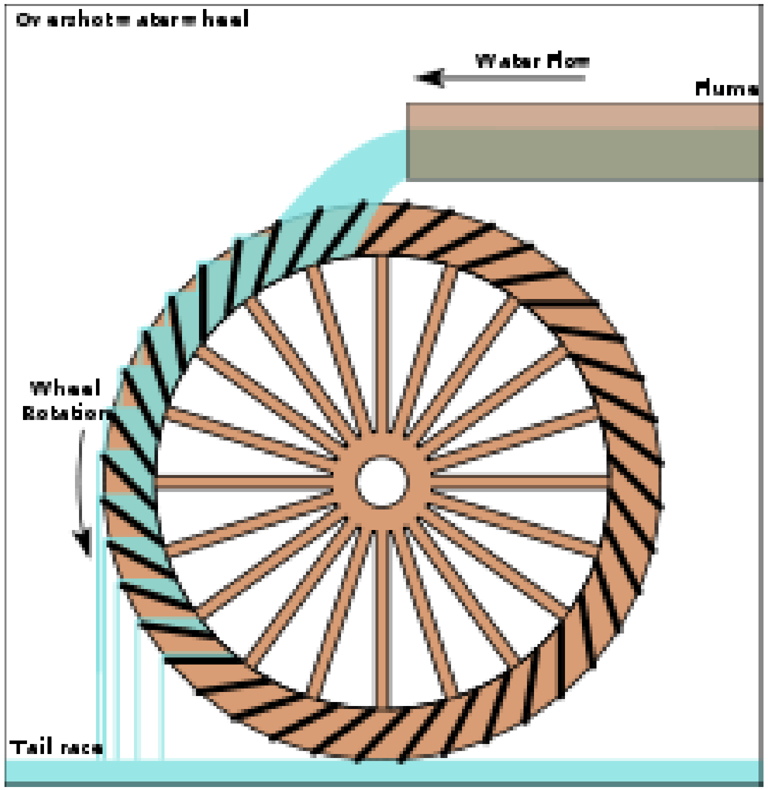




Undershot water wheel



Breastshot water wheel



Overshot water wheel

Hydraulic Turbines

Hydraulic turbines ***are the machines which convert the hydraulic energy into mechanical energy***. In other words, hydraulic turbines are prime movers which run with hydraulic energy. ***The mechanical energy produced by a hydraulic turbine can be converted into electricity by coupling the turbine to an electric generator***. The entire system consisting of a hydraulic turbine, a generator and other allied units is known as hydro electric project/plant.

Classification of Turbines

Hydraulic turbine may be classified according to the following criteria:

(a) Hydraulic action,

(b) Direction of flow of water,

(c) Disposition of the shaft,

(d) Head,

(e) Specific speed.

(a) Hydraulic action

(i) Impulse Turbine

In an impulse turbine, the pressure of water does not change while flowing through the rotor of the machine. In these turbines, the pressure change occur only in the nozzles of the machine and not in the rotor. The ***available head is totally converted into kinetic energy*** by passing the water through a nozzle. The water comes out the nozzle as ***a free jet***. It ***works under the atmospheric pressure***. **Exam:** Pelton wheel, Girard turbine, Banki turbine, Jonval turbine.

(ii) Reaction Turbine

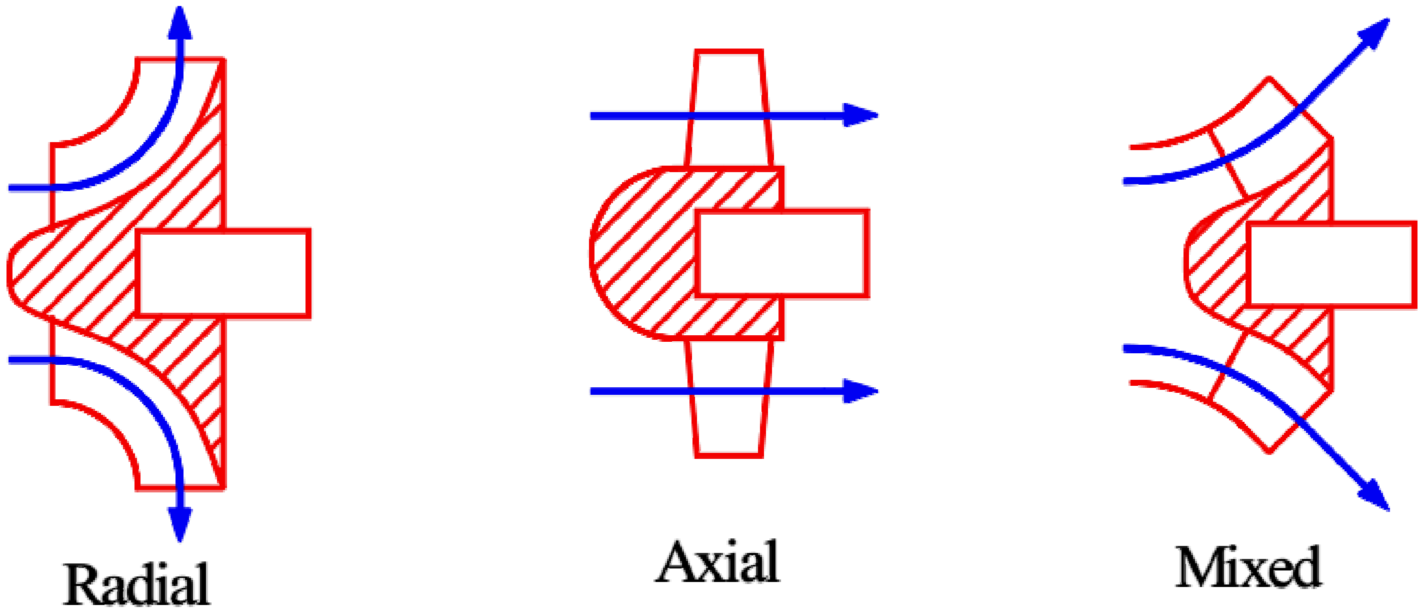
In a reaction turbine, the pressure of water changes while it flows through the rotor of the machine. The change in water velocity and reduction in its pressure causes a reaction on the turbine blades. ***A part of the total head is converted into kinetic energy and the rest remains in the form of pressure energy.*** ***The pressure at the inlet to the turbine is more than the atmospheric pressure and the pressure changes*** from inlet to tail race of the turbine. **Exam:** Francis turbine, Kaplan turbine, Propeller turbine, Thomson turbine.

(b) Direction of flow of water

(i) Axial Flow

The ***water flows in the direction parallel to the axis of the shaft***.

Exam: The Kaplan and Propeller turbines.



(ii) Radial flow turbine

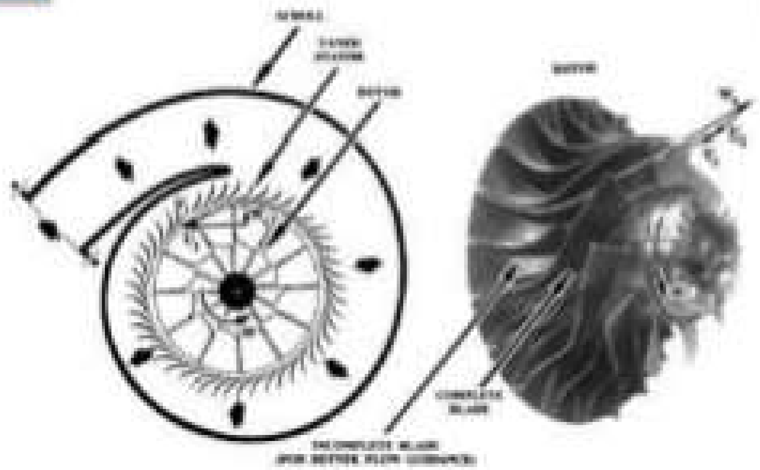
Here ***the water strikes in the radial direction***. A radial flow turbine may be ***inward flow or outward flow***, depending upon whether the flow is inward from the periphery to the centre or outward from the centre to the periphery.

Exam: old Francis turbine was an inward flow turbine.

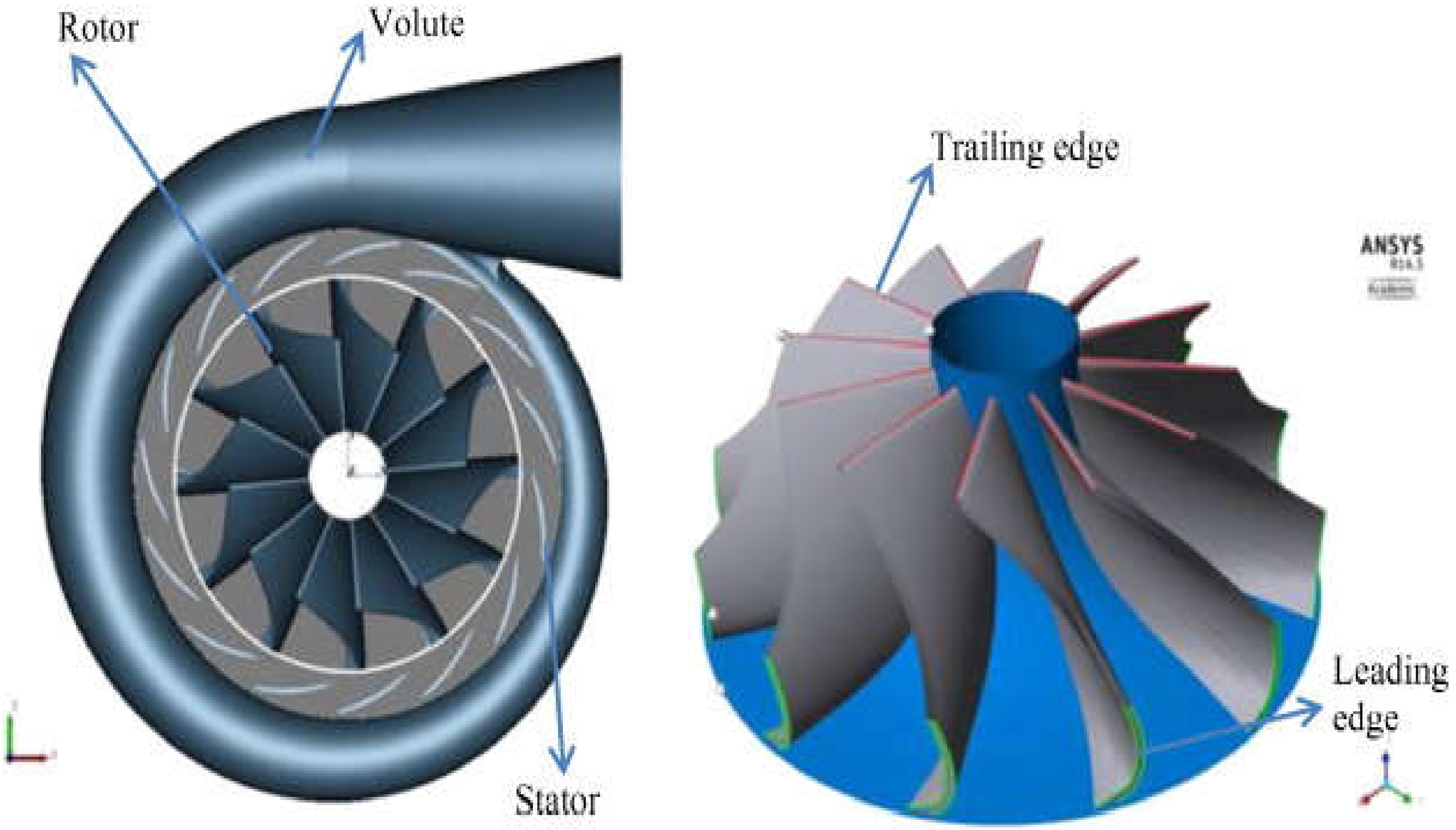
Tangential flow turbines

Mixed flow turbine

Axial flow turbine



Radial flow turbines


Rotor
Volute
Stator
Trailing edge
ANSYS
Leading
edge

(iii) Tangenial flow turbine

The water strikes the runner in the direction of tangent to the wheel.

Exam: Pelton wheel.

(iv) Mixed flow turbine

Here the water ***enters the runner in radial direction and leaves in axial direction***. **Exam:** Modern Francis turbine.

(c) Disposition of tbe shaft

(i) Vertical shaft turbine - the shaft is vertical and the runner is horizontal.

(ii) Horizontal shaft turbine - the shaft is horizontal and the runner is vertical.

(d) Head (under which it works)

(i) High head turbine - the ***net head varies from 150m to 2000m or more***. ***High head turbine requires a small quantity of water***.

Exam: The Pelton wheel

(ii) Medium head turbine – It works under a ***head of 30m to 150m***. The water requirements of a medium head turbine are moderate.

Exam: Francis turbine

(iii) Low head turbine - The turbine working under a head ***less than 30m is termed*** the low head turbine. A low head turbine requires a large quantity of water. **Exam:** Kaplan turbine, propeller turbine.

(e) Specific speed

The specific speed of a turbine is defined as the speed of a geometrically similar turbine working under a unit head and developing unit power.

(i) Low sp. speed - turbine that have the specific speed ***less than 60***.

Exam: Pelton wheel.

(i) Medium sp. speed - have the specific speed ***60 to 300***.

Exam: Francis turbine

(i) High sp. speed - have the ***specific speed greater than 300***.

Exam: Kaplan turbine

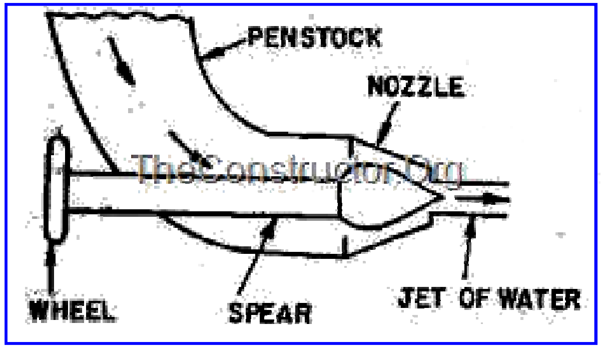
Pelton wheel

It is also called ***Free Jet Turbine***. The Pelton wheel is the most commonly used impulse turbine. It was designed by Lester A, Pelton in 1880. It works under a high head and requires small quantity of water. Water is taken to the turbine from the reservoir through penstocks. The penstock is a large pipe fitted with a nozzle at the end. Water comes out of the nozzle in the form of a jet. The whole of hydraulic energy is converted into kinetic energy at the nozzle. The jet of water issuing from the nozzle strikes the buckets the nozzle in the direction tangential to the wheel. The impact of water imparts a dynamic force to the wheel and the wheel starts moving. The water coming out the wheel is discharged into the tail race.

The main components of a Pelton wheel installation are described below:

(a) Nozzle with control mechanism:

A nozzle is fitted at the end of the penstock to convert hydraulic energy into kinetic energy. ***The nozzle is provided with a spear mechanism*** to control the quantity of water. The ***spear changes the opening in the nozzle by moving in or out and is usually operated by hand***.



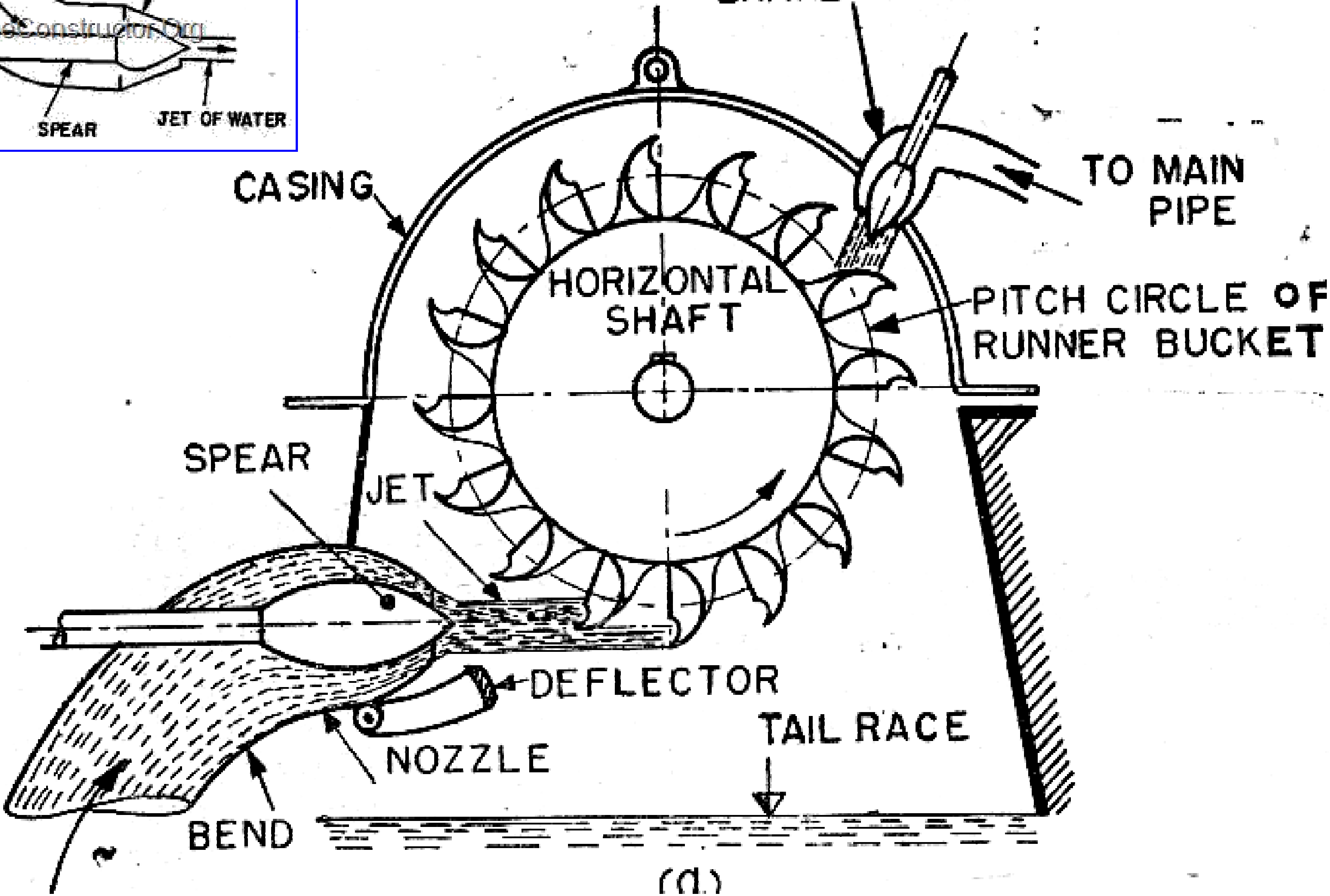


Fig: Single Jet, Horizontal Shaft Pelton Turbine

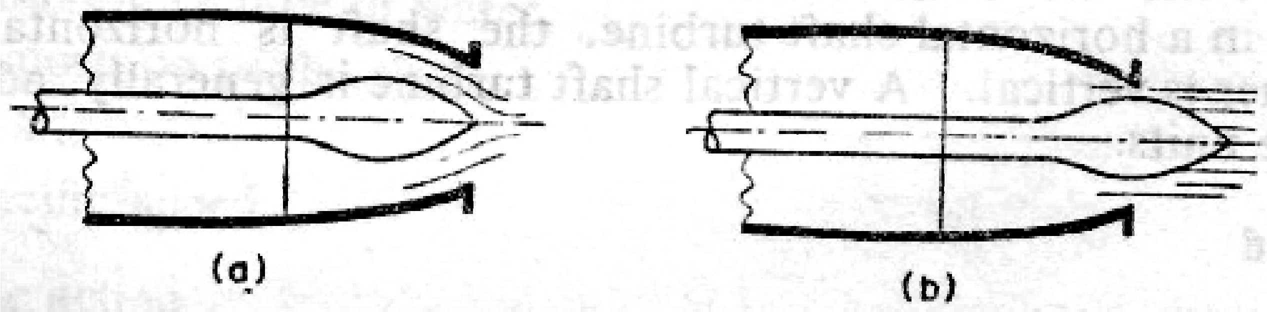
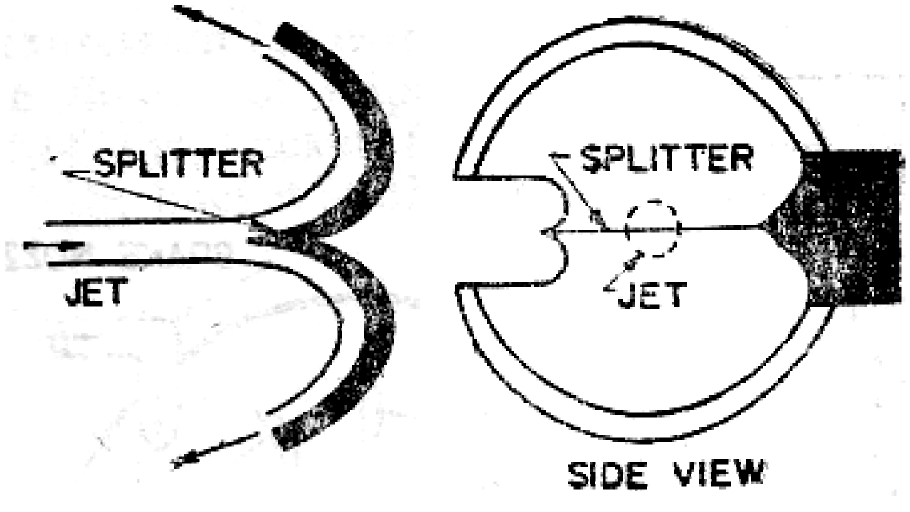


Fig: Pelton Wheel Nozzle

Note that the **jet velocity remains the same and only the discharge is changed by the mechanism.**

(b) Buckets and runner:

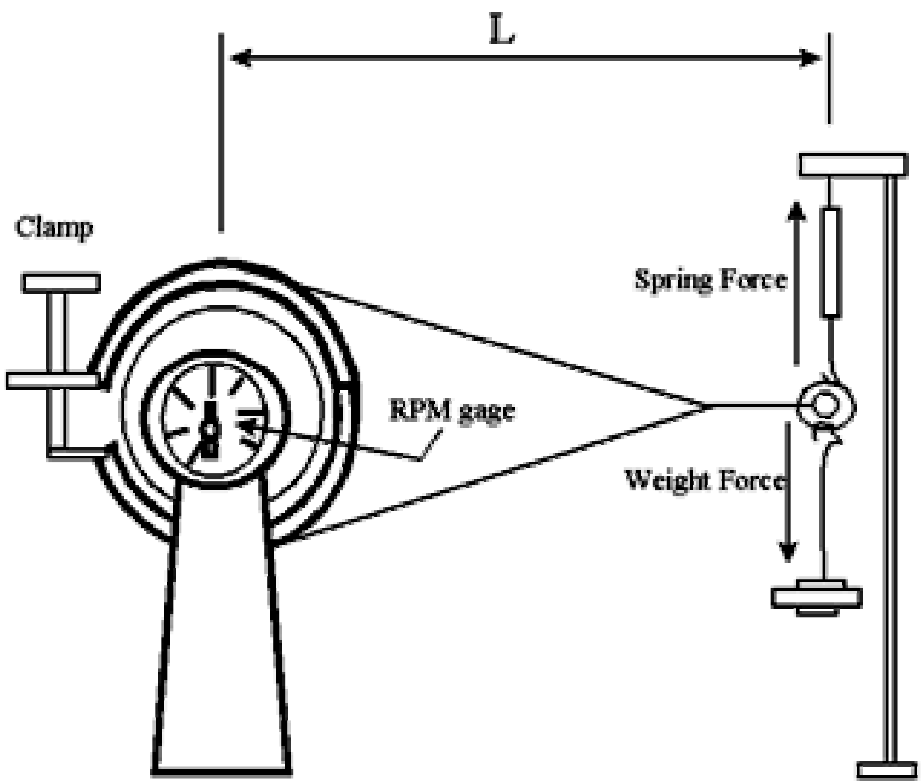
A Pelton wheel is fitted with buckets having the shape of **double-hemispherical cup**. The advantage of such shape is that **because of symmetry the axial thrust on the shaft is zero. The force in axial direction at the two ends are equal and opposite and balance each other**. The buckets should be properly designed to withstand the impact of jet. **For low heads**, the buckets are made of **cast iron**. But **for high heads**, the buckets are made of **bronze, cast steel or stainless steel.**



Each bucket is divided into two parts by a sharp vertical edge at the centre known as *splitter*. ***The angle at the outlet tip varies from 10° to 20°, thus deflecting the jet by 170° to 160°.*** The inner surface of the buckets should be properly polished to reduce the frictional resistance.

(c) Casing:

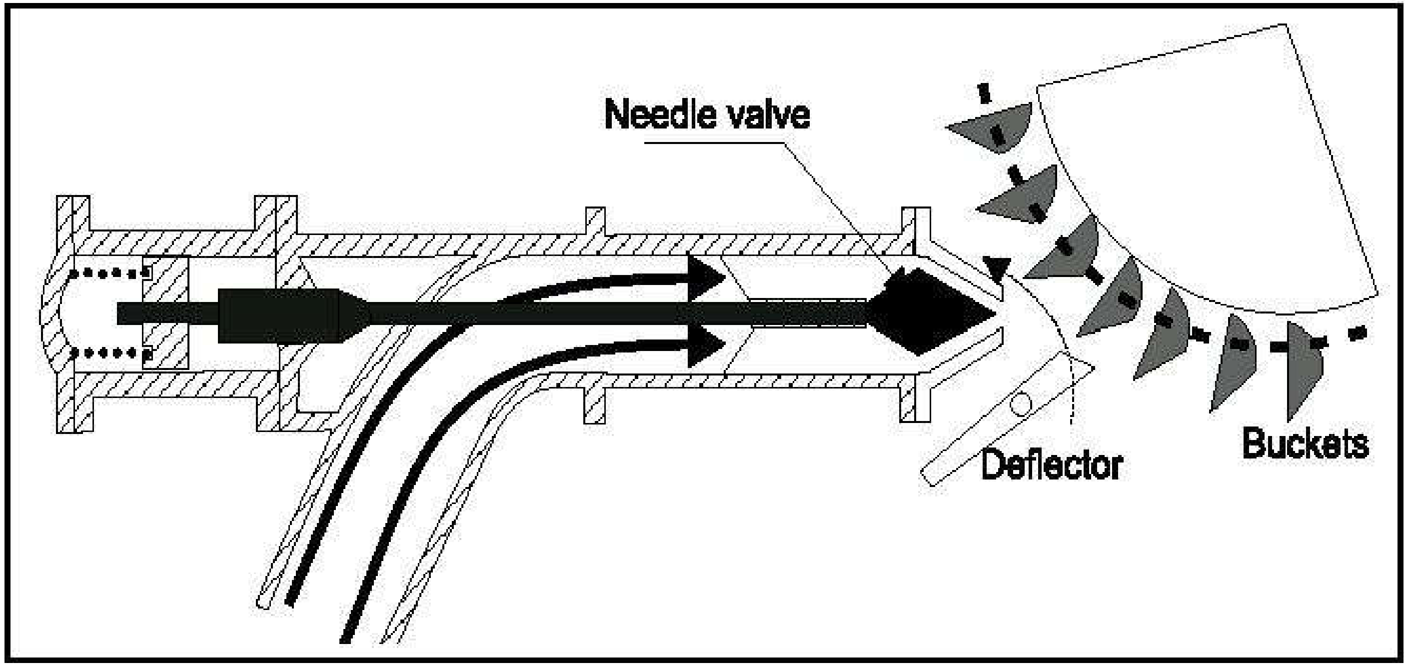
The casing has no hydraulic action to perform. However, the casing is usually provided **to prevent the splashing of water** and **to safe-gaurd against the accidents.**



(d) Hydraulic brake:

The hydraulic brake is provided to bring the runner to standstill position in a short time after the nozzle has been closed.

(e) Deflector:



When the load on the turbine decreases suddenly, the water supply has to be cut off by closing the nozzle. The sudden closure of the nozzle may cause excessive water hammer. To avoid water, the nozzle is closed slowly and some of the water coming out the nozzle is deflected away from the bucket by a movable steel plate known as deflector. ***Thus the deflector works only during the period when the nozzle is being adjusted.***

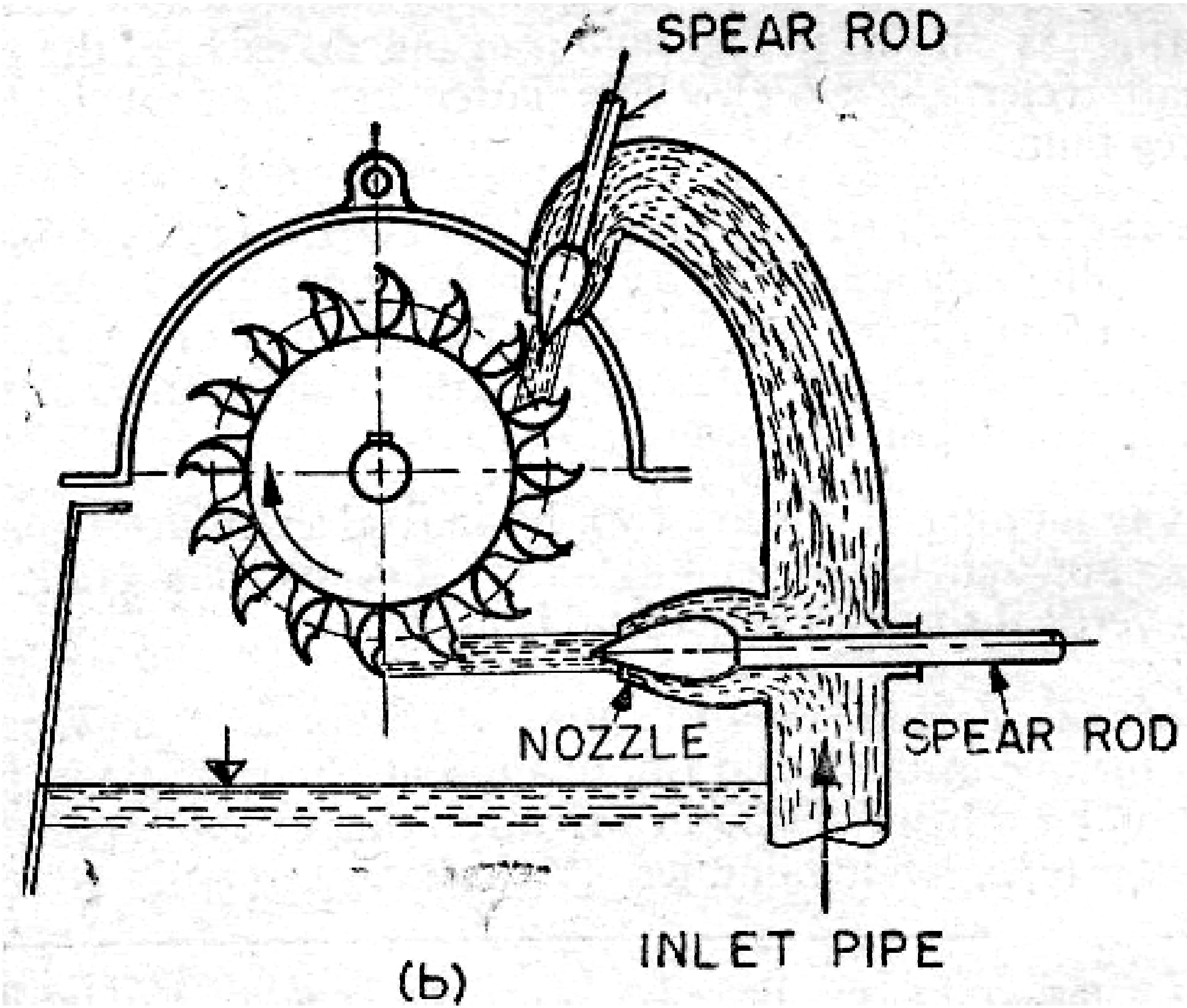
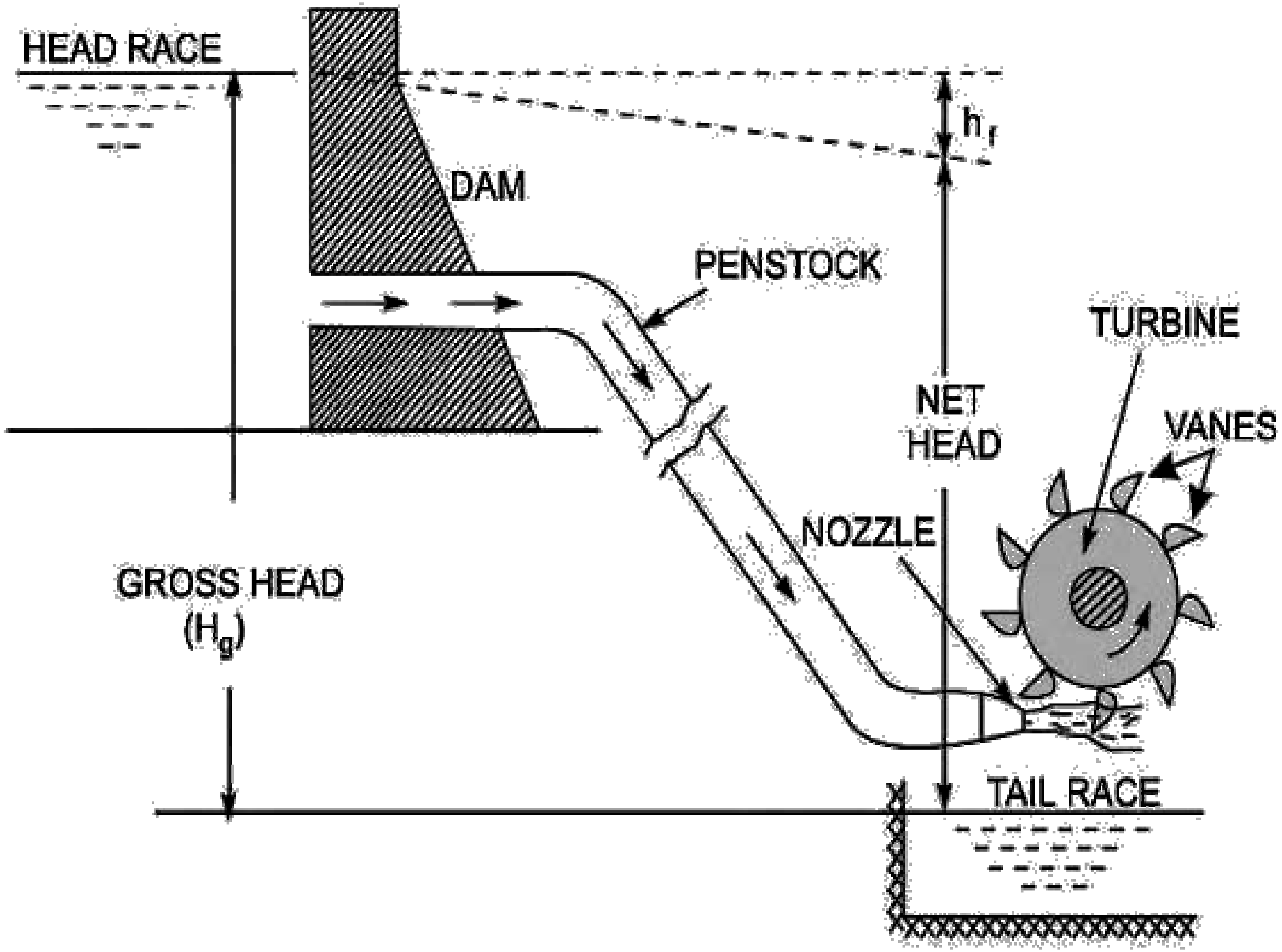
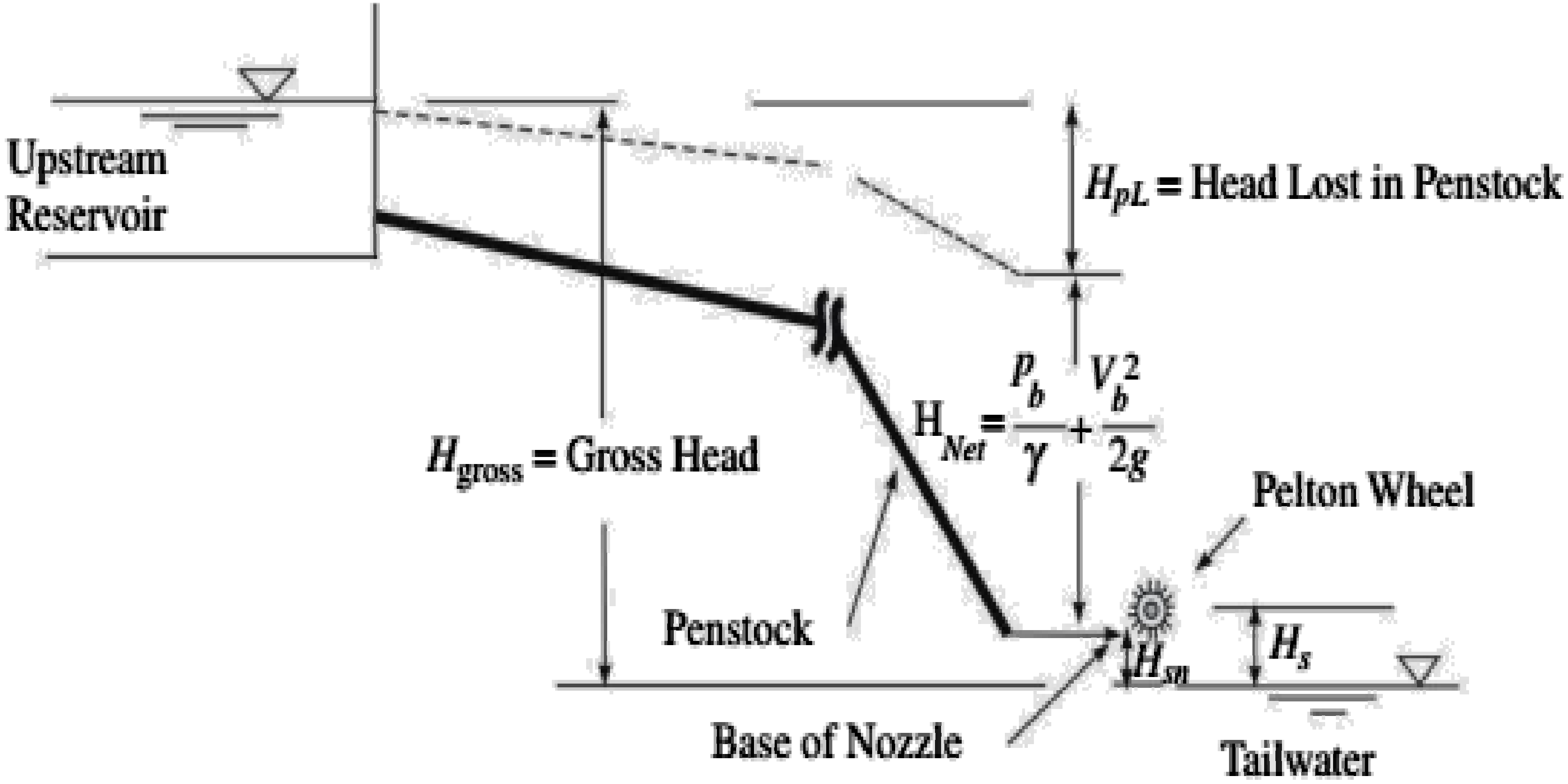


Fig: Double Jet, Horizontal Shaft Pelton Turbine

Expression for Work Done on pelton Wheels





Net head = H = $H_{gross} - H_{pL} - H_{sn}$

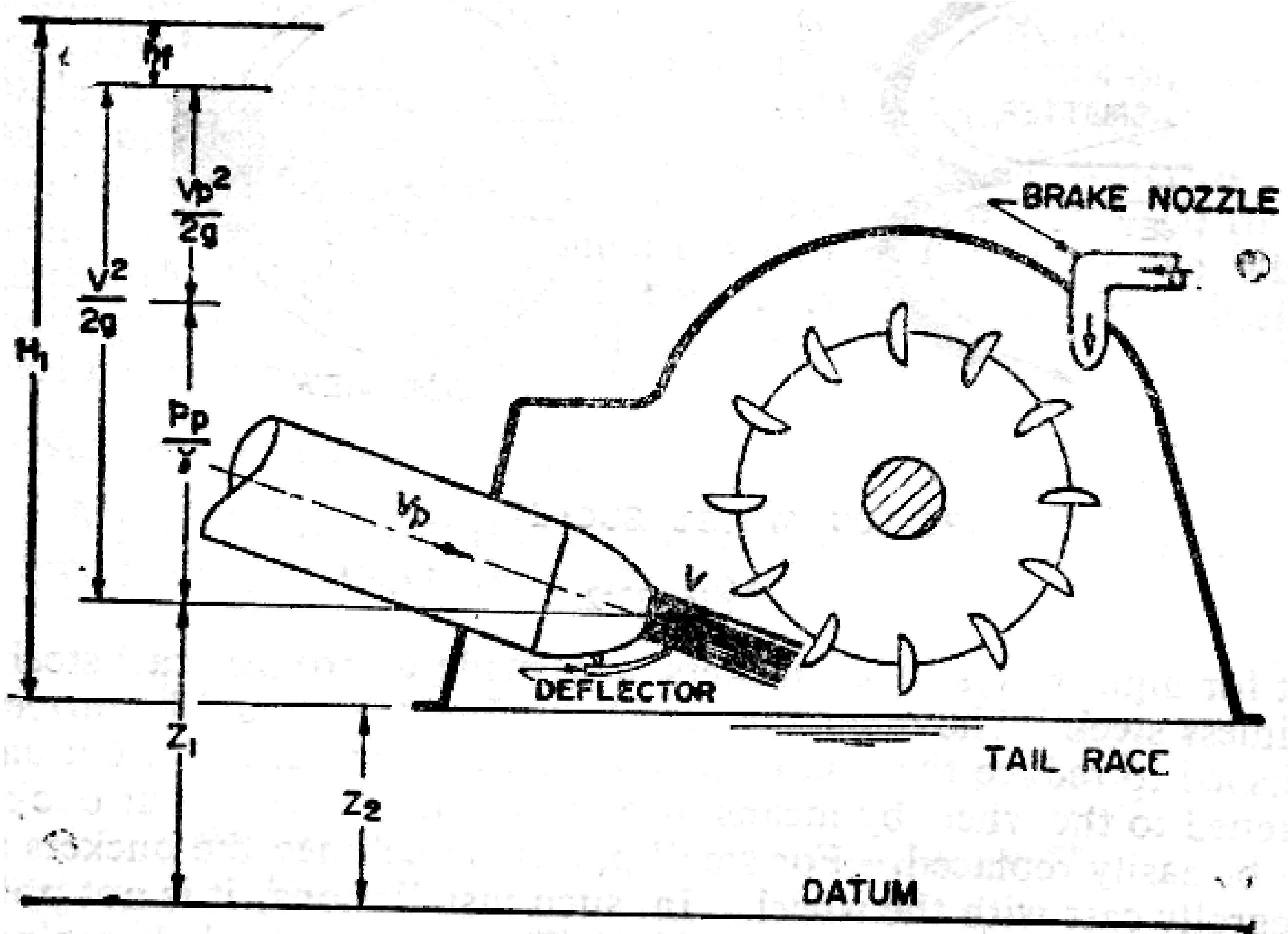
Also, $H = \frac{p_b}{\gamma} + \frac{V_b^2}{2g}$

where H_{gross} = Gross head = Difference in water surface elevation of upstream reservoir and tailwater level.

H_{pL} = Head loss in the penstock

H_{sn} = Height of the lowest nozzle above the tailwater level

$\frac{p_b}{\gamma}$ = Pressure head at the base of the nozzle and $\frac{V_b^2}{2g}$ = Velocity head at the base of the nozzle.



PELTON WHEEL

The gross head (H_1) is the ***difference of levels between the head race and the tail race***. However, in impulse turbine as available head is only upto the level of the nozzle, the **gross head is** taken as (**H_1 - Z**), where Z is the height of the nozzle above the tailrace level. Obviously, **$Z = Z_1 - Z_2$**, **The net head, also called the effective head**, is the ***head available at the entrance to the turbine*** and is given by **$H = \text{Gross head} - h_f$**

Where *h_f is the loss of head in the penstock*.

Or, $$H = H_1 - (Z_1 - Z_2) - h_f$$

The net head is also given by

$$H = \frac{p_p}{\gamma} + \frac{V_p^2}{2g}$$

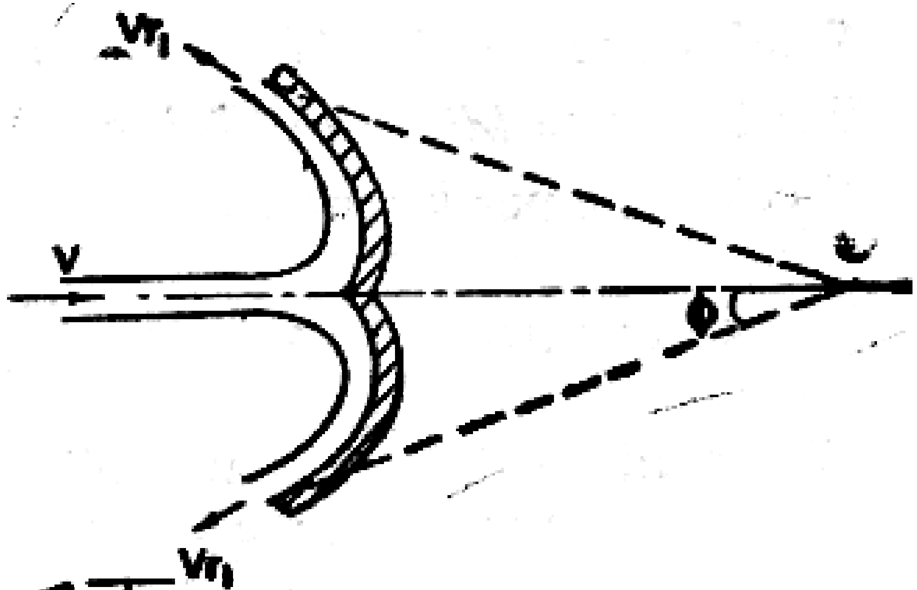
$$H = \frac{V^2}{2g} + \text{Losses in the nozzle}$$

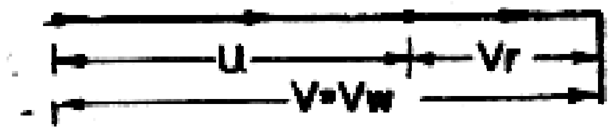
where P_p = pressure at the entrance to the nozzle

V_p = velocity in the Penstock

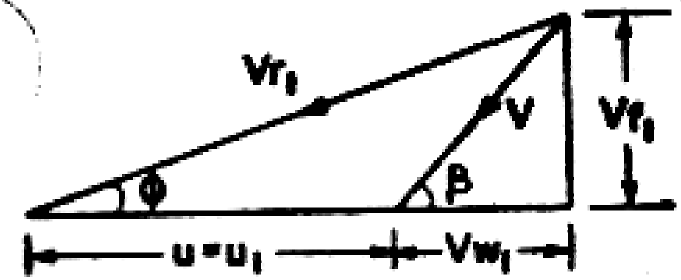
V = Velocity of the jet.

$$v = \sqrt{2gH}$$





INLET VELOCITY DIAGRAM



OUTLET VELOCITY TRIANGLE

The jet strikes at the centre of the bucket and after doing work, it leaves at the ends. The previous fig. shows the inlet velocity diagram and the exit velocity diagram. The inlet velocity diagram is a straight line. **The relative velocity and the velocity of whirl** is given by

$$\mathbf{V_r = V - u} \qquad \text{and} \qquad \mathbf{V_w = V}$$

The velocity triangle at outlet has been drawn for the case when the angle β is acute. ***As the inlet and outlet tips are at the same radial distance from the centre***,

$$\mathbf{u = u_1}$$

The relative velocity at the outlet tip **V_{r1} is *k* times V_r**, where **"*k*" is a constant which takes friction into account**. However, **for smooth buckets, *k* is unity**. From the outlet velocity triangle

$$V_{w1} = V_{r1} \cos \phi - u_1$$

$$= kV_r \cos \phi - u$$

$$= k(V - u) \cos \phi - u$$

The **work done per second** on the wheel is given by, using the Euler equation

$$\text{Work done} \qquad = M(V_w u \pm V_{w1} u_1)$$

(+) sign for $\beta < 90^0$ and (-) sign for $\beta > 90^0$.

Taking plus sign,

$$\text{Work done} = M[V + k(V-u)\cos\phi - u]u$$

or $$\text{Work done} = M(V-u)(1+k\cos\phi)u$$

$$\text{Work done per kg mass} = (V-u)(1+k\cos\phi)u$$

$$\text{Work done per unit weight} = \frac{(V-u)(1+k\cos\phi)u}{g}$$

Neglecting the losses in the nozzle,

$$\text{Input} = \frac{M}{2}(V^2)$$

The hydraulic efficiency (η_h) is defined as the ratio of the work done by the runner to the input.

$$\eta_h = \frac{\text{Work done}}{\text{Input}}$$

or $$\eta_h = \frac{M(V-u)(1+k\cos\phi)u}{\frac{1}{2}(M)V^2}$$

$$= \frac{2(V-u)(1+k\cos\phi)u}{V^2} \quad \text{...(23·4)}$$

For a given jet velocity V, the hydraulic efficiency is a maximum, if $\frac{d\eta_h}{du}=0$

or $$\frac{d}{du}[2(V-u)(1+k\cos\phi)]u=0$$

or $$V-2u=0$$

or $$u=\frac{V}{2}$$

Thus $$(\eta_h)_{max}=\frac{2(V-V/2)(1+k\cos\phi)(V/2)}{V^2}$$

or $$(\eta_h)_{max}=\tfrac{1}{2}(1+k\cos\phi) \quad \ldots(23\cdot5)$$

If $k=1$, *i.e.*, the vanes are smooth,

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

If **the buckets deflect the jet exactly through 180°**, $\phi = 0$ and **the hydraulic efficiency becomes 100 per cent.**

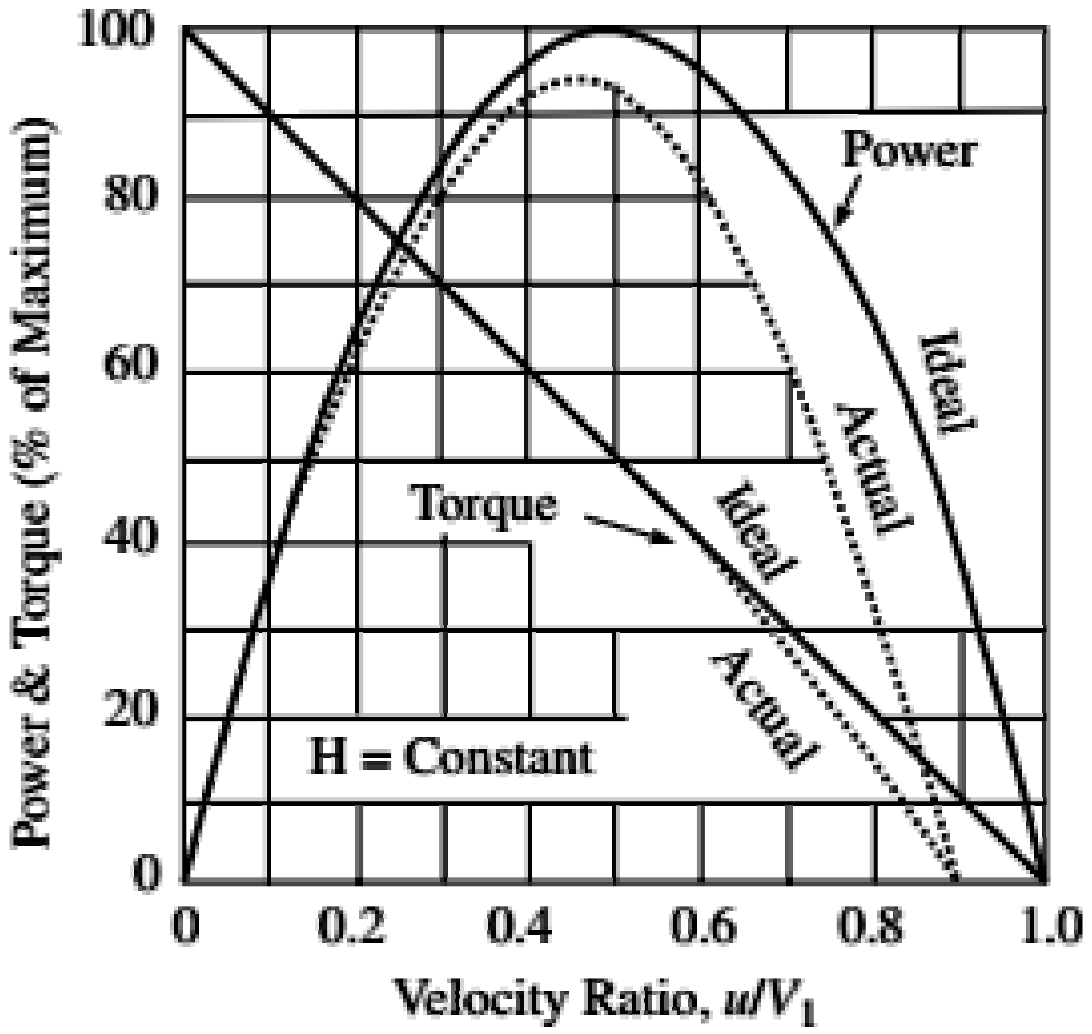


Fig. 4.5 *Variation of power and torque with speed ratio*

However, in practice, the maximum efficiency seldom exceeds 95%. The efficiency is less than 100% because :

(1) "k" is always less than unity,

(2) the angle $\phi \neq 0$.

If the angle $\phi = 0$, the jet leaving a **bucket strikes on the back of the following bucket and exerts a retarding force**. In order to keep the jet clear of the following bucket, the angle is usually kept between 10° to 20°. This angle ϕ is known as the **side clearance angle**.

The **work done by a Pelton wheel may also be expressed** as ***the difference of the kinetic energy at the inlet and outlet***. Thus

$$\text{Work done} = M(V^2 - V_1^2)/2$$

Therefore,

$$\eta_h = \frac{M(V^2 - V_1^2)/2}{MV^2/2}$$

$$\eta_h = \frac{V^2 - V_1^2}{V^2}$$

The **hydraulic efficiency** is also defined as ***the ratio of the power developed (P) to the power supplied to the turbine (P_t)***. Thus

$$\eta_h = P/P_t$$

The **mechanical efficiency (η_m)** is defined as t*he ratio of the power obtained at the shaft (S.P.) to the power developed by the runner (P)*. Thus

$$\eta_m = S.P / P$$

The difference between the shaft power (S.P.) and the power developed is *due to mechanical losses*.

The **overall efficiency (η_o)** of the turbine is the *ratio of the shaft power (S.P.) to the power supplied to the turbine (P_t)*. Thus

$$\eta_0 = \frac{S.P.}{P_t}$$

$$\eta_0 = \frac{S.P.}{P} \times \frac{P}{P_t}$$

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

The *overall efficiency is equal to* *the product of the mechanical efficiency and the hydraulic efficiency*. The overall efficiency of the Pelton wheel **ranges from 0.85 to 0.90**.

The volumetric efficiency (η_v) is the ***ratio of the volume of water actually striking the runner to the volume of water being supplied to the turbine***. Thus

$$\eta_v = \frac{Q}{Q + \Delta Q}$$

The **difference ΔQ is due to slippage of water directly to the tailrace**, **without striking the buckets**. ***The volumetric efficiency of the Pelton wheel varies from 97 to 99% and is usually taken as 100%.***

Working Proportions of Pelton Wheel

A Pelton wheel has the following working proportions. These proportions should be kept in view while dealing with a Pelton wheel.

(1) Velocity of Jet: The **theoretical velocity** of the jet is given by

$$V_{theo} = \sqrt{2gH}$$

where **H is the net head**. The **actual velocity (*V*)** is given by

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

where C_v is the **coefficient of velocity (nozzle)**. The value of C_v varies from ***0.98 to 0.99***.

(2) Theoretical power (P_t): The power available from water can be estimated from the expression

$$P_t = \gamma Q H$$

The ***theoretical power is also called the water power (W.P.)***. In MKS system it is called ***water horse power (W.H.P)***. Power supplied to the runner (P_1) is given by

$$P_1 = P_t - \text{Power lost in nozzle}$$
$$P_1 = P_t = \text{W.P.}$$

If the losses in the nozzle are neglected. The actual powers developed by the Pelton, wheel is

$$\text{S.P.} = \eta_0 \,(\text{W.P.}),$$
where η_0 = overall efficiency.

Unless otherwise mentioned, the losses in nozzle is neglected.

(3) Angle (ϕ): The angle ϕ **varies from 10° to 20°**. The **average** value of **ϕ is 15°**.

(4) Diameter of the Jet (d): The diameter of the jet may obtained if the discharge is known.

$$(\pi/4)\ d^2 = \frac{Q}{C_v\sqrt{2gH}}$$

$$d = \left(\frac{4Q}{\pi C_v\sqrt{2gH}}\right)^{1/2}$$

(5) Speed ratio (ϕ or Ku): The speed ratio is the ***ratio of the velocity (u) of the wheel at pitch circle to the theoretical velocity of the jet***. Thus

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

In practice, the value of the speed ratio varies from 0.43 to 0.47. Its average value is 0.45.

(6) Mean diameter of the wheel (D): The diameter of the wheel ***measured up to the centers of the buckets*** is called the mean diameter. The mean may be obtained from the peripheral speed (u). Thus $u = \pi DN/60$

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

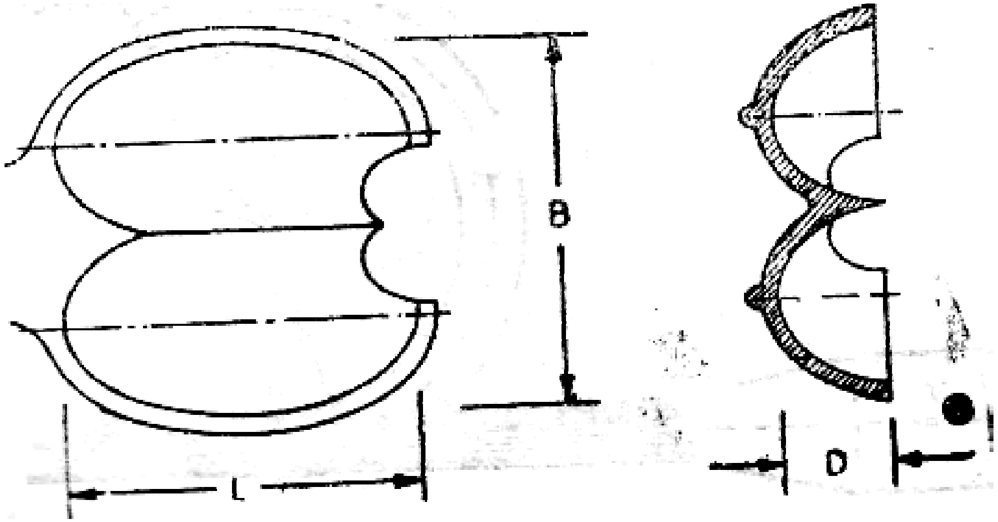
'D' is also known as **the pitch diameter**.

(7) Jet ratio (m): The ***ratio of the pitch diameter to the jet diameter*** is known as the jet ratio.

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

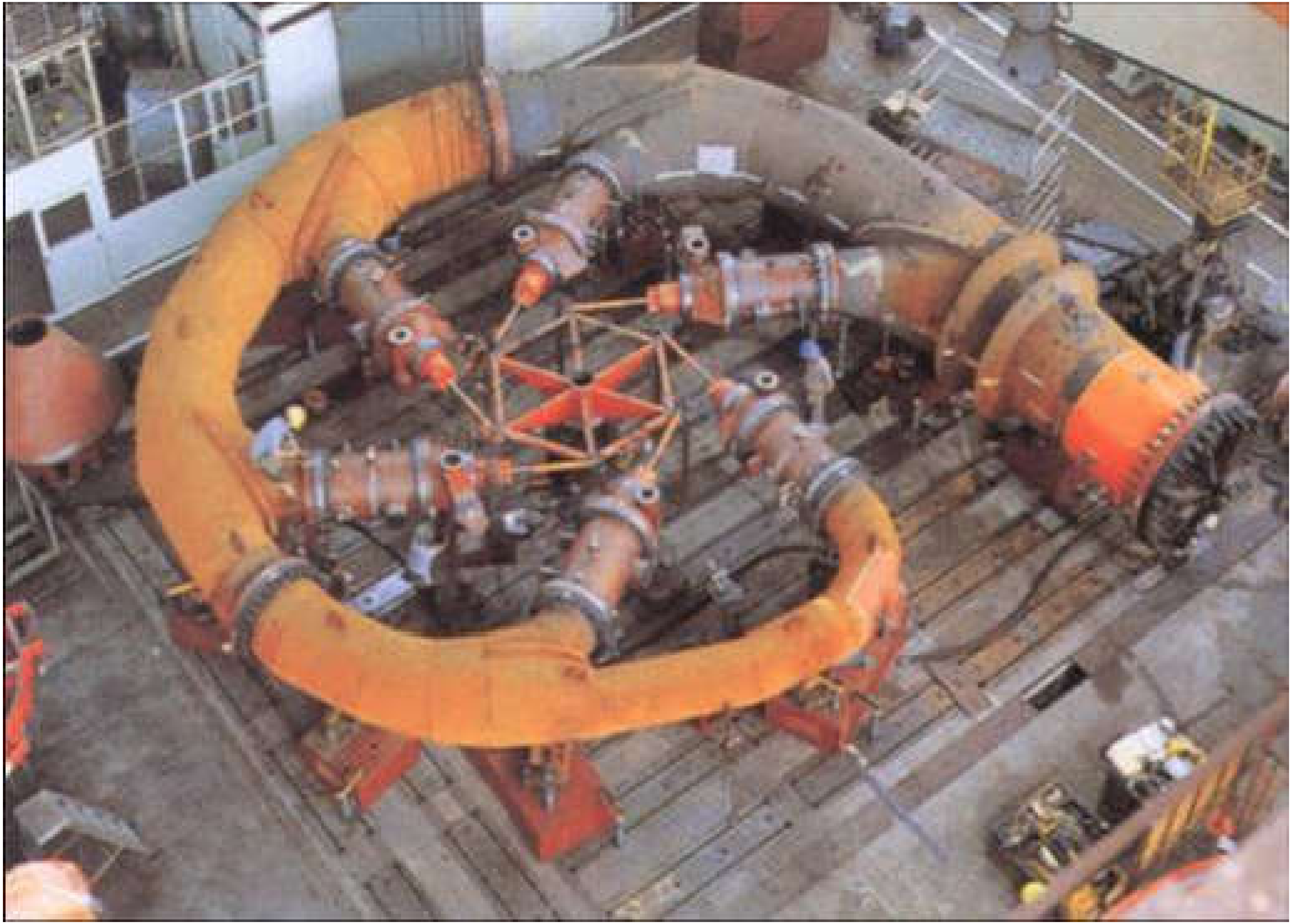
The jet ratio ***varies from 11 to 14***. The average value ***is 12***.

(8) Size of buckets: The following proportions of the bucket are usually adopted.



Radial length of bucket, L= 2 to 3d
Axial width of bucket, B = 3 to 5d
Depth of bucket, D = 0.8 to l.2 d
where d is the diameter of the jet.

(9) Number of jets (n): Ordinary, Pelton wheels have a single jet. But when the Pelton wheel has to develop more power, it is fitted with a number of jets. Then it is called **multiple jet Pelton wheel**. If P is the power developed by the Pelton wheel working under one jet, then **power developed** by the same wheel when working under n jets is ***nP***,



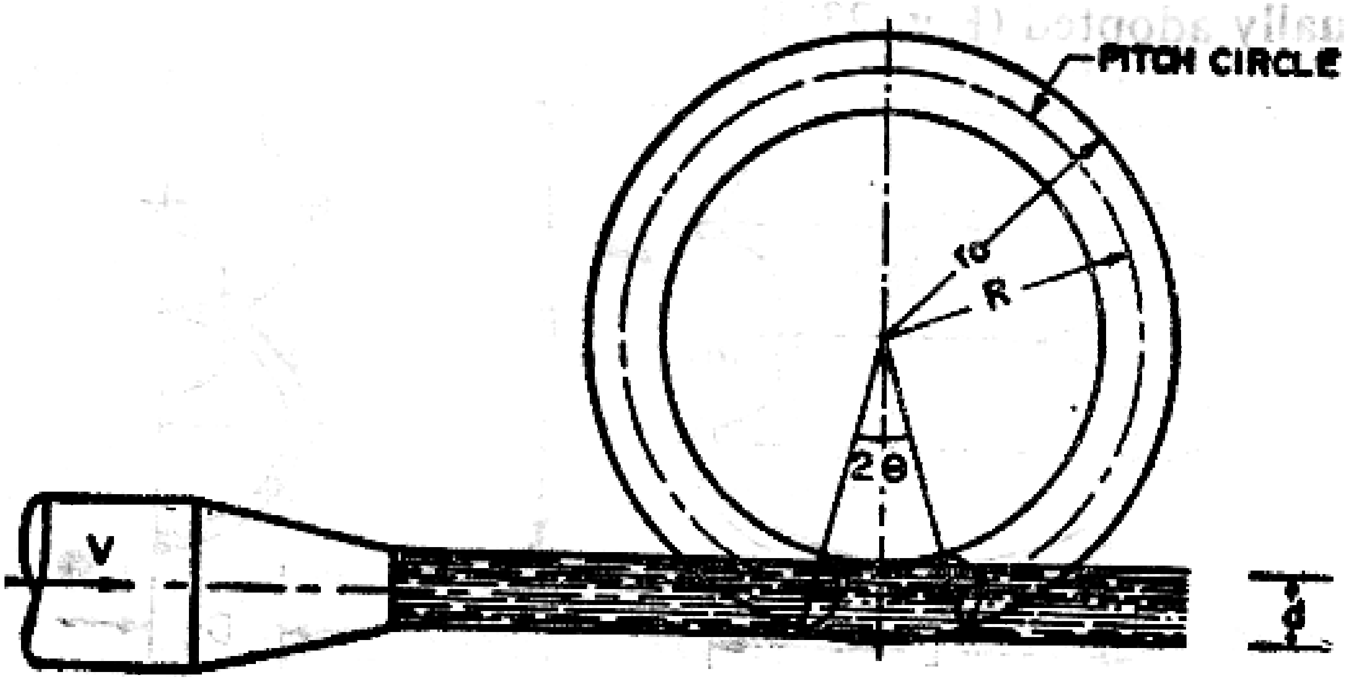
other conditions remaining the same.

The **jet should have sufficient spacing** ***so that one jet after striking the wheel does not interfere with the other***. Ordinarily, **more than six jets are not provided.**

{10) Number of buckets (z): The number of buckets ***should be as small as possible so as to keep the frictional losses to a minimum***. But the number of buckets should also be sufficiently large so that the jet is always intercepted by one bucket or the other. The jet will always be intercepted by one buckets if the angle between two successive buckets is equal to or less than 2, such that

$$\cos\theta = \frac{R + d/2}{r_0}$$

Where "*R*" is the radius of pitch circle and "r_o" is the radius of the outer circle and "*d*" is the diameter of the jet.



Therefore, $\sin\theta = \sqrt{1-\cos^2\theta}$

$$= \left[1 - \frac{(R+d/2)^2}{r_0^2}\right]^{1/2}$$

For small values of θ, $\sin\theta \simeq \theta$.

Therefore, $$\theta = \left[1 - \frac{(R+d/2)^2}{r_0^2}\right]^{1/2}$$

Therefore, the number of buckets is given by

$$z = \frac{2\pi}{2\theta} = \frac{\pi}{\left[1 - \frac{(R+d/2)^2}{r_0^2}\right]^{1/2}}$$

The number of buckets is usually obtained from the empirical formula given by Taygun.

$$z = \frac{D}{2d} + 15 = 0{\cdot}5\,m + 15$$ where z is the jet ratio.

Problem:

The jet of water coming out a nozzle strikes the buckets of a Pelton wheel which when stationary would deflect the jet through 165°. If the relative velocity of water at the exit is 0.9 times that at the inlet und the bucket speed is 0.45 times the jet speed, **determine the hydraulic efficiency and diameter of the Peltan wheel**. The speed of the Pelton wheel is 300 r.p.m. and the effective head is 150 m. Take $C_v = 0.98$.

Solution

Actual Velocity of Jet:

$$V = C_v\sqrt{2gH} = 0{\cdot}98\sqrt{2\times 9{\cdot}81\times 150} = 53{\cdot}2 \text{ m/sec}$$

Theoretical Velocity of Jet: $V_{theo} = \sqrt{2gH}$

The bucket speed

$$u = \phi\sqrt{2gH} = 0{\cdot}45\sqrt{2\times 9{\cdot}81\times 150} = 24{\cdot}4 \text{ m/sec}$$

Hydraulic Efficiency:

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

$$= \frac{2(53{\cdot}2-24{\cdot}4)(1+0{\cdot}9\times0{\cdot}966)\,24{\cdot}4}{(53{\cdot}2)^2}$$

$$= 0{\cdot}925 \text{ or } 92{\cdot}5\%.$$

diameter of the Pelton wheel

$$u = \pi DN/60$$

$$24{\cdot}4 = \pi \times D \times 300/60$$

$$D = 1{\cdot}55 \text{ m}$$

Problem:

A Pelton wheel is required to develop 9196.875 kW (12,500 h.p.) at the shaft when working under a head of 300 m. Assuming the values of C_v, ϕ and '*m*' as 0.98, 0.45 and 12, respectively, **determine (a) the number of jets, (b) the diameter of the wheel, (c) the quantity of water required and (d) the diameter of the jet**. Take the overall efficiency as 85% and the speed of the wheel as 550 r.p.m.

Solution

The bucket speed

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

$$u = 0{\cdot}45\sqrt{2\times9{\cdot}81\times300} = 34{\cdot}5 \text{ m/sec}$$

Actual Velocity of Jet: $V = C_v\sqrt{2gH}$

$= 0.98\sqrt{2 \times 9.81 \times 300} = 75.2$ m/sec

Overall efficiency: $\eta_0 = \dfrac{S.P.}{P_t}$ and $P_t = \gamma Q H$

$S.P. = \eta_0 \times P_t$

Therefore, $9196.875 = 0.85 \times 9.81 \times Q \times 300$

The quantity of water required $Q = 3.68$ cumecs

The diameter of the wheel $D = \dfrac{60\,u}{\pi N}$

$$D = \frac{60u}{\pi N} = \frac{60 \times 34.5}{\pi \times 550} = 1.20 \text{ m}$$

The diameter of the jet $d = \dfrac{D}{m} = \dfrac{1.20}{12} = 0.10$ m

Discharge from one jet $= \pi/4 \times (0.10)^2 \times 75.2 = 0.591$ cumecs

No. of jets $= \dfrac{3.68}{0.591} = 6.23 = 7$ (say)

Problem:

In a laboratory test on an impulse turbine, the following data were obtained: (a) Diameter of the jet = 3.90 cm, (b) Discharge = 0.022 m^3/s, (c) Head on the nozzle = 15 m, (d) Power available at the shaft = 2.354 kW (3.2 h.p.). Assuming a mechanical efficiency of 90%, **find the power (a) lost in the nozzle, (b) rejected to the tail race, (c) lost in mechanical friction.**

Solution. Overall efficiency, $\eta_0 = \dfrac{\text{S.P.}}{\text{Water Power}}$

or $$\eta_0 = \frac{2{\cdot}354}{9{\cdot}81 \times 0{\cdot}02 \times 15} = 0{\cdot}80$$

Water power $= 9{\cdot}81 \times 0{\cdot}02 \times 15 = 2{\cdot}943$ kW

Overall efficiency, $\eta_0 = \eta_m \times \eta_h$

$$0{\cdot}80 = 0{\cdot}90 \times \eta_h \quad \text{or} \quad \eta_h = 0{\cdot}89$$

Velocity of the jet $= \dfrac{0{\cdot}02}{\pi/4 \times (0{\cdot}039)^2} = 16{\cdot}74$ m/sec

Power available at the nozzle,

$$P_1 = \left(\frac{M}{2} \times V^2 \right) = \frac{\gamma Q V^2}{2g}$$

$$= \frac{9{\cdot}81 \times 0{\cdot}02 \times (16{\cdot}74)^2}{9{\cdot}81 \times 2}$$

$$= 2{\cdot}802 \text{ kW } (3{\cdot}82 \text{ h.p.})$$

Power lost in the nozzle = Water horse power − power at the nozzle

$= 2{\cdot}943 - 2{\cdot}802 = \mathbf{0{\cdot}141\ kW}$

Power developed $P = \eta_h \times$ Water power

$= 0{\cdot}89 \times 2{\cdot}943 = 2{\cdot}619$ kW

Power rejected to the tail race $= 2{\cdot}802 - 2{\cdot}619$

$= \mathbf{0{\cdot}183\ kW}$

Power lost in mechanical friction $= 2{\cdot}619 - 2{\cdot}354$

$= 0{\cdot}265$ kW (**0·36 h.p.**).