

Centrifugal pumps

FMDHM

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Introduction :-

The hydraulic machines which convert the mechanical energy are called pumps. The hydraulic energy is in the form of pressure energy. If the mechanical energy is converted into pressure energy by means of centrifugal force acting on the fluid, the hydraulic machine is called Centrifugal pump.

The centrifugal pump acts as a reverse of an inward radial flow reaction turbine. This means that the flow in centrifugal pumps is in the radial outward directions. The centrifugal pump works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise in pressure head of the rotating liquid takes place. The rise in pressure head at any point of the rotating liquid is proportional to the square of tangential velocity of the liquid at that point.

$$\therefore \text{rise in pressure head} = \frac{V^2}{2g}$$

Thus at the outlet of the impeller, where radius is more, the rise in pressure head will be more and more, the rise in pressure head will be discharged at its outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level.

Main parts of a centrifugal pump :-

The following are the main parts of a centrifugal pump :

- 1) Impeller
- 2) Casing
- 3) Suction pipe with a foot valve and a strainer.
- 4) Delivery pipe.

All the main parts of the centrifugal pump are shown.

1. Impeller :- The rotating part of a centrifugal pump is called impeller. It consists of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to the shaft of an electric motor.

2. Casing ! -

The casing of a centrifugal pump is similar to the casing of a reaction turbine. It is an air tight passage surrounding the impeller and is designed in such a way that the kinetic energy of the water discharged at the outlet of the impeller is converted into pressure energy before the water leaves the casing and enters the delivery pipe.

The following three types of the casings are commonly adopted

- a) Volute Casing as shown in fig
- b) Vortex Casing as shown in fig. (a)
- c) Casing with guide blades as shown in fig (b)

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a) Volute Casing :-

Fig. shows the volute casing, which surrounds the impeller. It is of spiral type in which area of flow increases gradually. The increase in area of flow decreases the velocity of flow. The decrease in velocity increases the pressure of the water flowing through the casing. It has been observed that in case of volute casing, the efficiency of the pump increases slightly as a large amount of energy is lost due to the formation of eddies in this type of Casing.

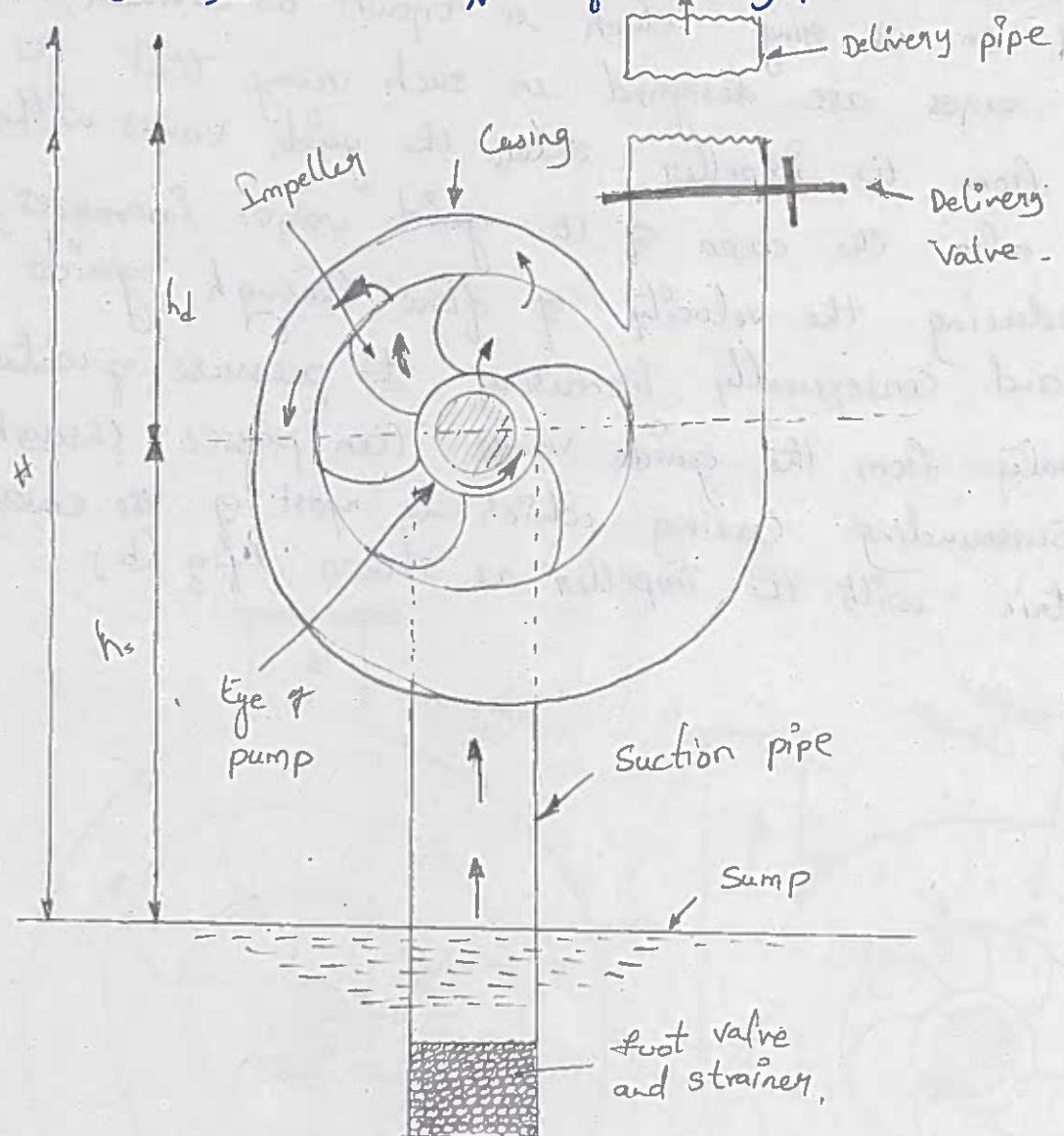


Fig Main parts of a centrifugal pump.

b) Vortex Casing:-

If a circular chamber is introduced b/w the Casing and the impeller as shown in fig (a)

the casing is known as vortex casing. By introducing the circular chamber, the loss of energy due to the formation of eddies is reduced to a considerable extent.

Thus the efficiency of the pump is more than the efficiency when only volute casing is provided.

c) Casing with Guide Blades:-

This casing is shown in fig (b) in which the impeller is surrounded by a series of guide blades mounted on a ring which is known as diffuser. The guide vanes are designed in such away that the water from the impeller enters the guide vanes without shock also the area of the guide vanes increases, thus reducing the velocity of flow through guide vanes and consequently increasing the pressure of water. The water from the guide vanes then passes through the surrounding casing which is most of the cases concentric with the impeller as shown fig (b).

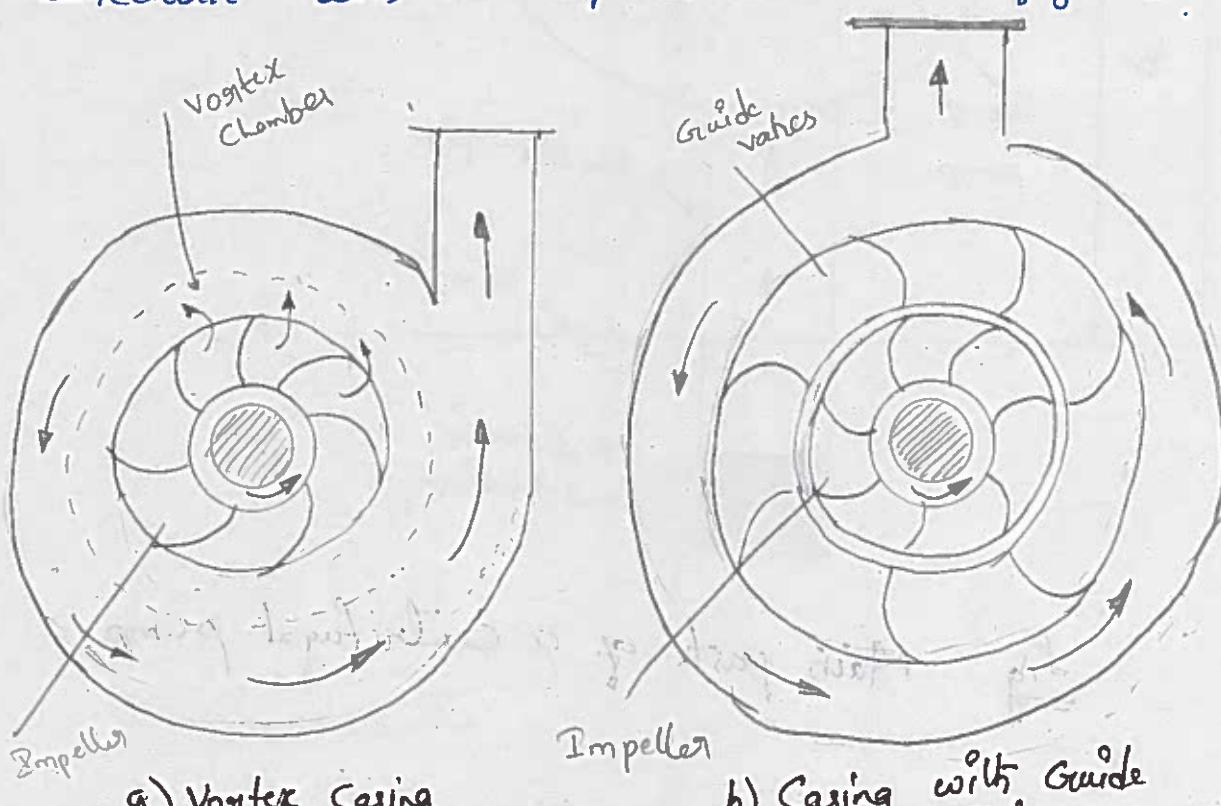


fig Different types of Casing. UNIT-5; Pg - 47/39

3) Suction pipe with a foot valve and a strainer:-

A pipe whose one end is connected to the inlet of the pump and other end dips into water in a sump is known as suction pipe. A foot valve which is a non-return valve or one-way type of valve is fitted at the lower end of the suction pipe. The foot valve opens only in the upward direction. A strainer is also fitted at the lower end of the suction pipe.

4) Delivery pipe :-

A pipe whose one end is connected to the outlet of the pump and other end delivers the water at a required height is known as delivery pipe.

* Work done by the centrifugal pump on water.

In case of the centrifugal pump, work is done by the impeller on the water. The

expression for the work done by the impeller on the water is obtained by drawing velocity triangles at inlet for best efficiency of the pump,

which means the absolute velocity of its motion

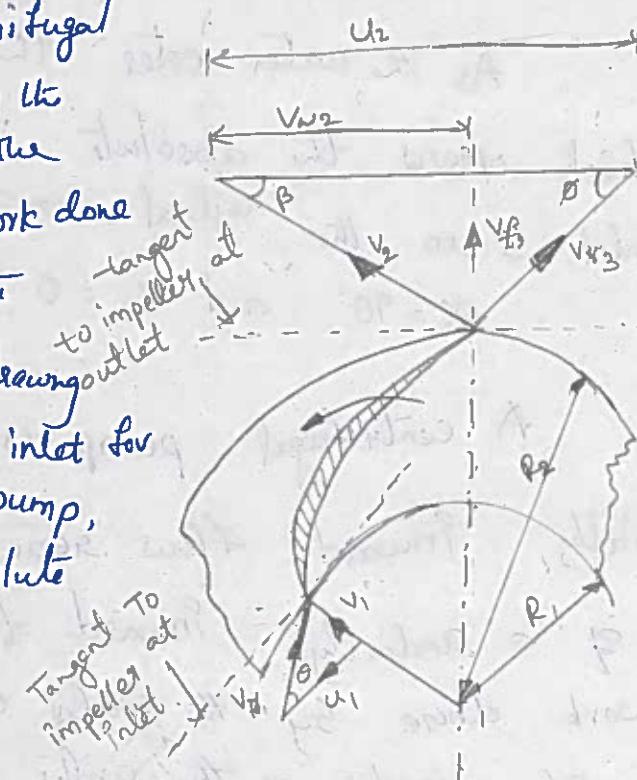
of impeller at inlet.

Hence angle $\alpha = 90^\circ$

and $V_{A1} = 0$. For drawing

the velocity triangle the ~~for~~ velocity triangles at inlet and outlet.

same notations are used as that for turbines. Fig shows the velocity triangles at the inlet and outlet tips of the vanes fixed to an impeller.



Let N = speed of its impeller in rpm.

D_1 = Diameter of impeller at inlet.

u_1 = Tangential velocity of impeller at inlet.

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

D_2 = Diameter of impeller at outlet

u_2 = Tangential velocity of impeller at outlet

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

v_1 = Absolute velocity of water at inlet.

v_{r1} = Relative velocity of water at inlet.

α = Angle made by absolute velocity (v_r) at inlet with the direction of motion of vane.

θ = Angle made by relative velocity (v_{r1})

at inlet with the direction of motion

of vane and $v_r, v_{r2}, \beta, \theta$ and ϕ are the corresponding values at outlet.

As the water enters the impeller radially

which means the absolute velocity of water at inlet is in the radial direction and hence angle $\alpha = 90^\circ$ and $v_{r1} = 0$.

A centrifugal pump is the reverse of a

radially inward flow reaction turbine. But in

case of a radially inward flow reaction turbine, the work done by the water on the runner per second per unit weight of the water striking per second is given by equation as

$$\text{Work} = \frac{1}{g_w} [v_w u_1 - v_w u_2]$$

\therefore Work done by the impeller on the water per second
per unit weight of water striking per second.
 \therefore [Work done in case of turbine].

$$\begin{aligned} &= - \left[\frac{1}{g} [v_{w_1} u_1 - v_{w_2} u_2] \right] \\ &= \frac{1}{g} [v_{w_2} u_2 - v_{w_1} u_1] \\ &= \frac{1}{g} v_{w_2} u_2 \quad : v_{w_1} = 0. \quad -(19.1) \end{aligned}$$

Work done by impeller on water per second.

$$= \frac{W}{g} v_{w_2} u_2$$

W = weight of water = $\rho \times g \times Q$.

Q = volume of water

$$Q = \pi D_1 B_1 \times V_{f_1}$$

$$= \pi D_2 B_2 \times V_{f_2}$$

where B_1 and B_2 are width of impeller at inlet
and outlet and V_{f_1} and V_{f_2} are velocities of flow at
inlet and outlet.

Equation (19.1) gives the head imparted
to the water by the impeller or energy given by
impeller to water per unit weight per second.

* Definitions of Heads and efficiencies of a Centrifugal pump.

1) Suction Head (h_s):- It is the vertical height of
the centre line of the centrifugal pump above the
water surface in its tank or pump from which
water is to be lifted as shown fig 19.1.
This height is also called suction lift and
denoted by h_s .

2) Delivery Head (h_d):

The vertical distance between the centre line of the pump and the water surface in the tank to which water is delivered is known as delivery head.

This is denoted by h_d .

3) Static head (H_s) :-

The sum of suction head and of delivery head is known as static head. This is represented by ' H_s ' and is written as

$$H_s = h_s + h_d. \quad \text{---(3)}$$

4) Manometric head (H_m)

The manometric head is defined as the head against which a centrifugal pump has to work. It is denoted by ' H_m '. It is given by the following expressions:-

a) H_m = Head imparted by impeller to the water - loss of head in the pump.

$$= \frac{V_o \omega_2 u_2}{g} - \text{Loss of head in Impeller and casing} \quad \text{---(4)}$$

b) H_m : Total head at outlet of the pump - Total at the inlet of the pump.

$$= \left[\frac{P_o}{\rho g} + \frac{V_o^2}{2g} + z_o \right] - \left[\frac{P_i}{\rho g} + \frac{V_i^2}{2g} + z_i \right]$$

$\frac{P_o}{\rho g}$ = pressure head at outlet of the pump

$\frac{V_o^2}{2g}$ = velocity head at outlet of the pump.

= Velocity head in delivery pipe = $\frac{V_d^2}{2g}$

z_o = Vertical height of the outlet of the pump from datum line.

$\frac{P_i}{\rho g}$, $\frac{V_i^2}{2g}$, z_i = Corresponding values of pressure head, velocity head and datum head at the inlet of the pump.

h_s , $\frac{V_s^2}{2g}$ and z_s respectively.

$$h_m = h_s + h_d + h_{fs} + \frac{V_d^2}{2g}$$

where, h_s = suction head, h_d = delivery head.

h_{fs} = frictional head loss in suction pipe.

h_{fd} = frictional head loss in delivery pipe.

V_d = Velocity of water in delivery pipe.

5) Efficiencies of a centrifugal pump:-

In case of a centrifugal pump, the power is transmitted from the shaft of the electric motor to the shaft of the pump and then to the impeller. From the impeller, the power is given to the water. Thus power is decreasing from the shaft of the pump to the impeller and then to the water. The following are the important efficiencies of a centrifugal pump.

a) Manometric efficiency, η_{man}

b) Mechanical efficiency η_m

c) overall efficiency η_o .

a) Manometric efficiency (η_{man}). The ratio of the manometric head to the head imparted by the impeller to the water is known as manometric efficiency.

Mathematically, it is written as

$$\text{UNIT-5; Pg-9/39} \quad \eta_{man} = \frac{\text{Manometric head}}{\text{Head imparted by impeller to water.}}$$

$$= \left[\frac{\frac{1}{2} \rho u_2^2}{\frac{g H_m}{\rho}} \right] = \frac{g H_m}{\rho u_2^2}$$

The power at the impeller of the pump is more than the power given to the water at outlet of the pump. The ratio of the power to the power available at the impeller, is known as manometric efficiency.

The power given to water at outlet of the pump

$$= \frac{\omega H_m}{1000} \text{ k.w.}$$

The power at the impeller = $\frac{\text{Work done by Impeller per second}}{1000 \text{ k.w.}}$

$$= \frac{\omega}{g} \times \frac{V_{w2} \times u_2}{1000} \text{ k.w.}$$

$$\eta_{man} = \frac{\frac{\omega \times H_m}{1000}}{\frac{\omega}{g} \times \frac{V_{w2} \times u_2}{1000}} = \frac{g \times H_m}{V_{w2} \times u_2}$$

b) Mechanical efficiency (η_m) :-

The power at the shaft of the centrifugal pump is more than the power available at the impeller to the water. It is known as manometric efficiency.

power at the shaft of the centrifugal pump is known as η_{man} = mechanical efficiency.

$$\eta_m = \frac{\text{Power at the impeller}}{\text{Power at the shaft}}$$

$$\text{The power at the impeller in k.w.} = \frac{\text{Work done by impeller per second}}{1000}$$

$$= \frac{\omega}{g} \times \frac{V_{w2} u_2}{1000}$$

$$\eta_m = \frac{\frac{\omega}{g} \left[\frac{V_{w2} u_2}{1000} \right]}{S.P}$$

where, S.P = shaft power.

⇒ Overall efficiency (η_o) :-

It is defined as ratio of power output of the pump to the power input to the pump. The power output of the pump in kw.

$$= \frac{\text{weight of water lifted} \times H_m}{1000} = \frac{wH_m}{1000}$$

power input to the pump = power supplied by its electric motor.
= s.p of the pump.

$$\eta_o = \frac{\left(\frac{wH_m}{1000}\right)}{\text{s.p.}}$$

$$\eta_o = \eta_{\text{man}} \times \eta_m$$

A) Losses :- The various losses occurring during the operation of a centrifugal pump may be classified as follows

- 1) hydraulic losses.
- 2) Mechanical losses.
- 3) Leakage losses.

1) hydraulic losses :- The hydraulic losses that may occur in a centrifugal pump installation may be grouped as

- a) Hydraulic losses in the pump.
- b) Others hydraulic losses.

The hydraulic losses that may occur within the pump consist of the following

- (i) shock or eddy losses at the entrance to and the exit from the impeller.
- (ii) Friction losses in the impeller.
- (iii) ~~Friction~~ Friction and eddy losses in the guide vanes and casting.

It can be seen from fig. that for the given values of the blade angle θ and ϕ , and the speed of rotation, there can be only one rate of discharge that will ensure tangential entry to the impeller and tangential exit from the impeller. But often the pump is required to operate under varying conditions which result in the variation in the rate of discharge. As such at the entrance and the exit of the impeller the shock losses generally occurs.

The other hydraulic losses consist of the following.

- (i) Friction and other minor losses in the suction pipe.
- (ii) Friction and other minor losses in the delivery pipe.

2) Mechanical losses :-

The mechanical losses occurs in the centrifugal pump on account of the following

- (i) Disc friction b/w the impeller and the liquid which fills the clearance spaces b/w the impeller and the casing.
- (ii) Mechanical friction of the main bearings and glands.

3) Leakage losses :-

In centrifugal pumps as ordinary built, it is not possible to provide a completely water tight seal b/w the delivery and suction spaces. As such there is always a certain amount of liquid which slips or leaks from the high pressure to the low pressure points in the pump and it never passes through the delivery pipe. The liquid which escapes or leaks from from a high pressure zone to a low pressure zone carries with it energy which is subsequently

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wasted in eddies. This loss of energy due to leakage of liquid represent the leakage loss.

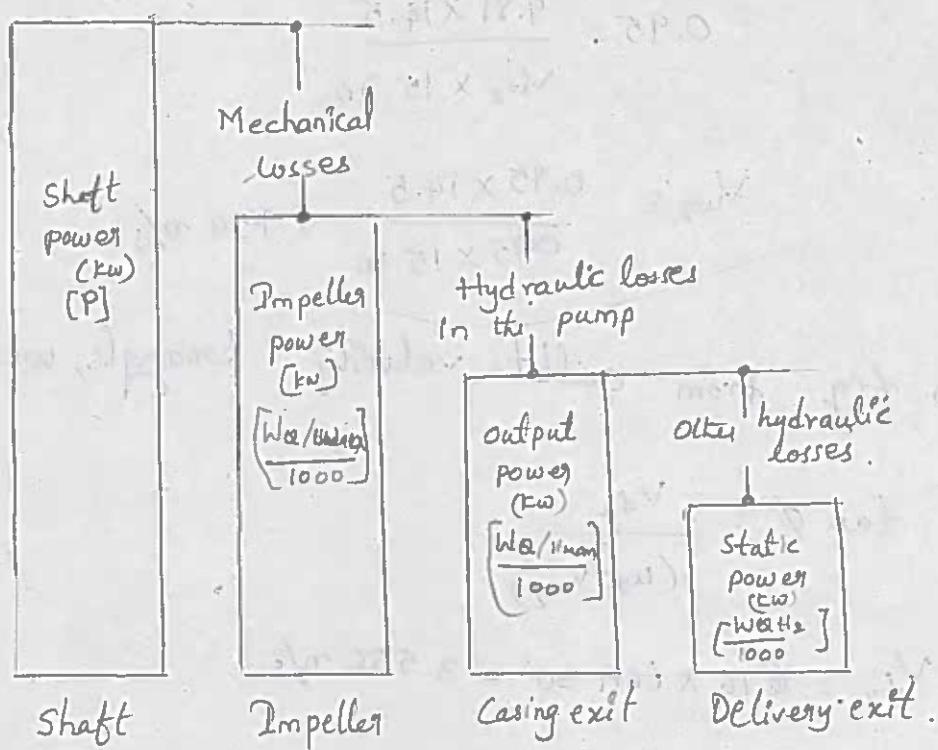


Fig Head losses in Centrifugal pump.

Problem

A centrifugal pump delivers water against a net head of 14.5 m and a design speed of 1000 r.p.m. The vanes are curved back to an angle of 30° with the periphery. The impeller diameter is 300 mm and outlet width is 50 mm. Determine the discharge of the pump if manometric efficiency is 95%.

Sol :-

Given,

$$H_m = 14.5 \text{ m}$$

$$N = 1000 \text{ r.p.m}$$

$$\phi = 30^\circ$$

$$D_2 = 300 \text{ mm} = 0.30 \text{ m}$$

$$B_2 = 50 \text{ mm} = 0.05 \text{ m}$$

$$\gamma_{\text{man}} = 95\% = 0.95$$

$$u_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.30 \times 1000}{60} = 15.70 \text{ m/s}$$

Now using equation, $\eta_{man} = \frac{gH_m}{V_{w2} \times u_2}$

$$0.95 = \frac{9.81 \times 14.5}{V_{w2} \times 15.30}$$

$$V_{w2} = \frac{0.95 \times 14.5}{0.95 \times 15.30} = 9.54 \text{ m/s}$$

Refer to fig from outlet velocity triangle, we have

$$\tan \phi = \frac{V_{f2}}{(u_2 - V_{w2})}$$

$$V_{f2} = 6.16 \times \tan 30^\circ = 3.556 \text{ m/s}$$

$$Q = \pi D_2 B_2 \times V_{f2} \\ = \pi \times 0.30 \times 0.05 \times 3.556 \text{ m}^3/\text{s}$$

$$Q = 0.1675 \text{ m}^3/\text{s}$$

* Minimum speed for starting A Centrifugal pump.

If the pressure rise in the impeller is more than or equal to manometric head, the centrifugal pump will start delivering water. Otherwise, the pump will not discharge any water, through the impeller is rotating. When impeller is rotating, the water in contact with the impeller is also rotating. This is the case of forced vortex, the centrifugal head is head due to pressure rise in the impeller.

$$= \frac{\omega^2 r_2^2}{2g} - \frac{\omega^2 r_1^2}{2g}$$

where, ωr_2 = Tangential velocity of impeller at outlet u_2

ωr_1 = Tangential velocity of impeller at inlet = u_1 .

\therefore Head due to pressure rise in impeller = $\frac{u_2^2}{2g} - \frac{u_1^2}{2g}$

The flow of water will commence only if

Head due to pressure rise in impeller $\geq H_m$.

For minimum speed, we must have $\frac{U_2^2}{2g} - \frac{U_1^2}{2g} = H_m$.

But from equation we have.

$$\eta_{\text{man}} = \frac{g H_m}{V_{w_2} U_2}$$

$$H_m = \eta_{\text{man}} \times \frac{V_{w_2} U_2}{g}$$

Substituting this value of H_m in equation

$$\frac{U_2^2}{2g} - \frac{U_1^2}{2g} = \eta_{\text{man}} \times \frac{V_{w_2} U_2}{g}$$

Now, $U_2 = \frac{\pi D_2 N}{60}$ and $U_1 = \frac{\pi D_1 N}{60}$

Substituting the values of U_2 and U_1 in eqn.

$$\frac{1}{2g} \left(\frac{\pi D_2 N}{60} \right)^2 - \frac{1}{2g} \left(\frac{\pi D_1 N}{60} \right)^2 = \eta_{\text{man}} \times \frac{V_{w_2} \times \pi D_2 N}{g \times 60}$$

Dividing by $\frac{\pi N}{g \times 60}$ we get $\frac{\pi N D_2^2}{120} - \frac{\pi N D_1^2}{120} = \eta_{\text{man}} \times V_{w_2} \times D_2$

(or) $\frac{\pi N}{120} [D_2^2 - D_1^2] = \eta_{\text{man}} \times V_{w_2} \times D_2$

$$N = \frac{120 \times \eta_{\text{man}} \times V_{w_2} \times D_2}{\pi (D_2^2 - D_1^2)}$$

equation gives the minimum starting speed of the centrifugal pump.

* Multistage Centrifugal pumps:

If a centrifugal pump consists of two or more impellers, the pump is called a multistage centrifugal pump. The impellers may be mounted on the same shaft or on different shaft. A multistage pump is having the following two important functions.

- 1) To produce a high head
- 2) To discharge a large quantity of liquid.

If a high head is to be developed. The impellers are connected in series or in parallel.

* Multistage Centrifugal pumps for high heads:-

For developing a high head, a number of impellers are mounted in series or not on the same shaft as shown in fig.

The water from suction pipe enters the 1st impeller at inlet and is discharged at outlet with increased pressure. The water with increased pressure from the outlet of the 1st impeller is taken to the inlet of the 2^d impeller with the help of a connecting pipe as shown in fig. At the outlet of the 2^d impeller the pressure of water will be more than the pressure of water at the outlet of the 1st impeller. Thus if more impellers are mounted on the same shaft, the pressure at outlet will be increased further.

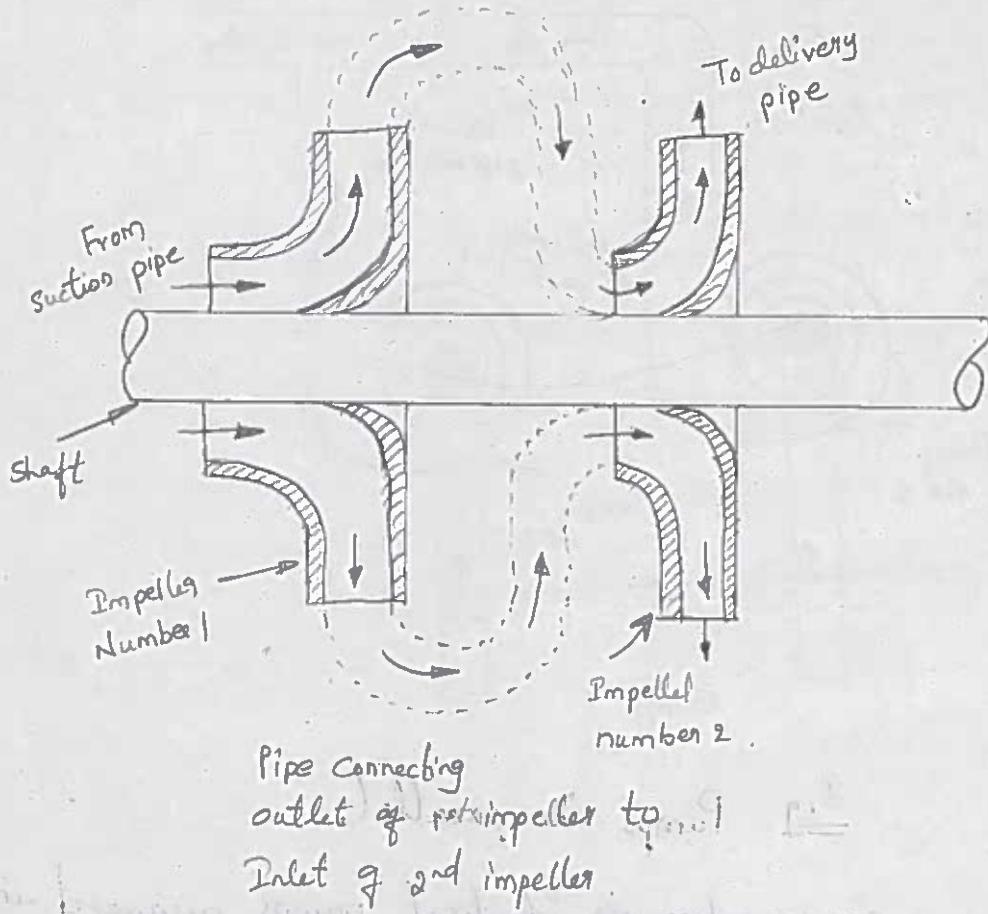


fig Two-stage pumps with impellers in series.

Let η = Number of identical impellers mounted on the same shaft.

H_m = Head developed by each impeller.

Then total head developed.

$$= n \times H_m$$

The discharge passing through each impeller is same.

* Multistage Centrifugal pumps for high Discharge:-

For obtaining high discharge, the pumps should be connected in parallel as shown in fig. Each of the pump lifts the water from a common pump and discharges water to a common pipe to which the delivery pipes of each pump is connected. Each of the pump is working against the same head.

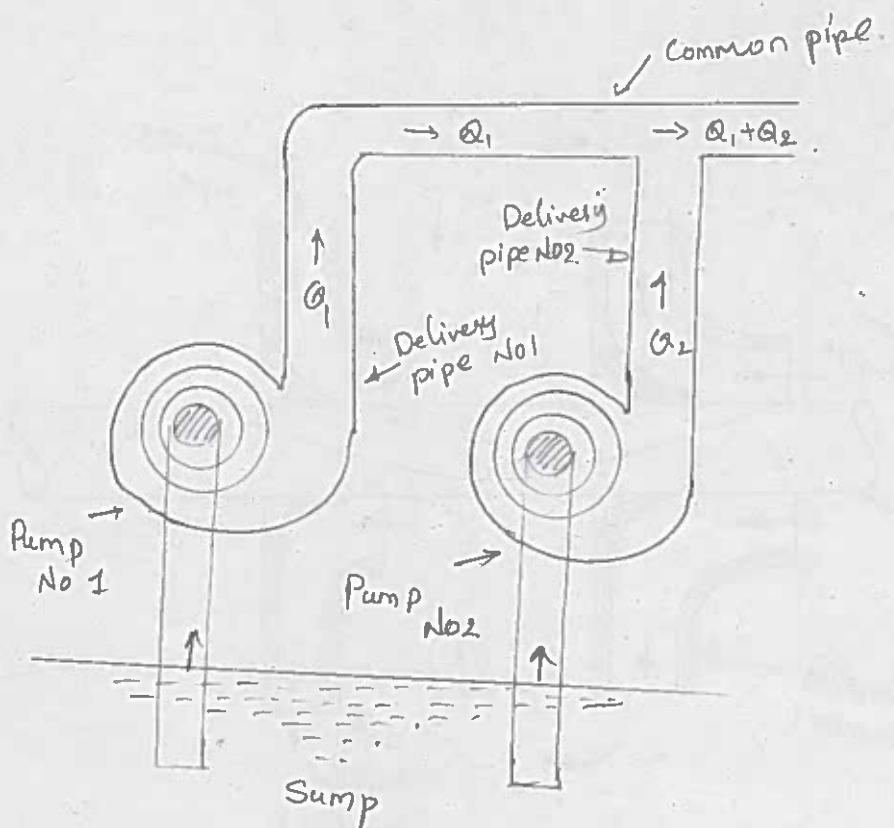


Fig Pumps in parallel.

γ = number of identical pumps arranged in parallel.

Q = Discharge from one pump.

$$\text{Total discharge} = \gamma \times Q$$

* Specific speed of a centrifugal pump (N_s).

The specific speed of a centrifugal pump is defined as the speed of a geometrically similar pump which would deliver one cubic metre of liquid per second against a head of one metre. It is denoted by ' N_s '.

* Expression for specific speed of a pump! -

The discharge Q for a centrifugal pump is given by the relation

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

$$= \pi D \times B \times V_f$$

D = Diameter of impeller of the pump.

B = width of the impeller.

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we know that, $B \propto D$.

from eqn (i) we have $Q \propto B^2 \times V_f$.
we also know that tangential velocity is given by

$$u = \frac{\pi D N}{60} \propto DN.$$

Now the tangential velocity (u) and velocity of flow (V_f) are related to the manometric head as,

$$u \propto V_f \propto \sqrt{H_m}.$$

Substituting the value of D in eqn (i)

$$Q \propto \frac{H_m}{N^2} \times V_f.$$

$$\propto \frac{H_m}{N^2} \times \sqrt{H_m}.$$

$$\propto \frac{H_m^{3/2}}{N^2}$$

$$Q = k \frac{H_m^{3/2}}{N^2}$$

where k is a constant of proportionality.

If $H_m = 1m$ and $Q = 1 m^3/s$ it becomes -

Substituting these values in equation

$$1 = k \frac{1^{3/2}}{N_s^2} = \frac{k}{N_s^2}$$

$$k = N_s^2.$$

Substituting the value of k in equation (V) we get

$$Q = N_s^2 H_m^{-1/2}$$

(or)

$$N_s^2 = \frac{N^2 Q}{H_m^{3/2}}$$

$$N_s = \frac{N \sqrt{Q}}{H_m^{3/4}},$$

$\approx 02.$

* Model Testing of Centrifugal pumps :-

Before manufacturing the large size pumps, their models which are in complete similarity with the actual pumps are made. Test are conducted on the models and performance of the prototypes are predicated. The complete similarity b/w the model and actual pump will exist if the following conditions are satisfied.

1. Specific speed of model = specific speed of prototype

$$(N_s)_m = (N_s)_p$$

2) Tangential Velocity (v_t) is given by

$$v_t = \frac{\pi DN}{60} \text{ also } v_t \propto \sqrt{H_m}$$

$$\sqrt{H_m} \propto DN$$

$$\frac{\sqrt{H_m}}{DN} = \text{constant}$$

$$(O^*) \quad \left(\frac{\sqrt{H_m}}{DN} \right)_m = \left[\frac{\sqrt{H_m}}{DN} \right]_p$$

3. From equation (i) of Art. 19.7.1 we have

$$Q \propto D^2 \times V_t$$

$$\propto D^2 \times DN$$

$$\propto D^3 \times N$$

$$\frac{Q}{D^3 N} = \text{constant}$$

4. Power of the pump , $P = \frac{\rho \times g \times Q \times H_m}{75}$

$$P \propto Q \times H_m$$

$$\propto D^3 N \times D^2 N^2$$

$$\propto D^5 N^3$$

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$$\frac{P}{D^5 N^3} = \text{constant} \rightarrow (19.22)$$

$\therefore =$

Problem

Find the number of pumps required to take water from a deep well under a total head of 89m. All the pumps are identical and are running at 800 rpm. The specific speed of each pump is given as 25 while the rated capacity of each pump is $0.16 \text{ m}^3/\text{s}$.

Sol

Given,

$$\text{Total head} = 89 \text{ m}$$

$$\text{Speed } N = 800 \text{ rpm.}$$

$$\text{Specific speed } N_s = 25$$

$$\text{Rate capacity } Q = 0.16 \text{ m}^3/\text{s.}$$

H_m = Head developed by each pump.

using eqn.

$$N_s = \frac{N \sqrt{Q}}{H_m^{3/4}}$$

$$25 = \frac{800 \times \sqrt{0.16}}{H_m^{3/4}}$$

$$H_m = (12.8)^{4/3} = 29.94 \text{ m}$$

\therefore Number of pumps required = $\frac{\text{Total head}}{\text{Head developed by one pump}}$

$$= \frac{89}{29.94} = 3.$$

As the total head is more than the head developed by one pump, the pump should be connected in series.

* Characteristic Curves of Centrifugal pumps:-

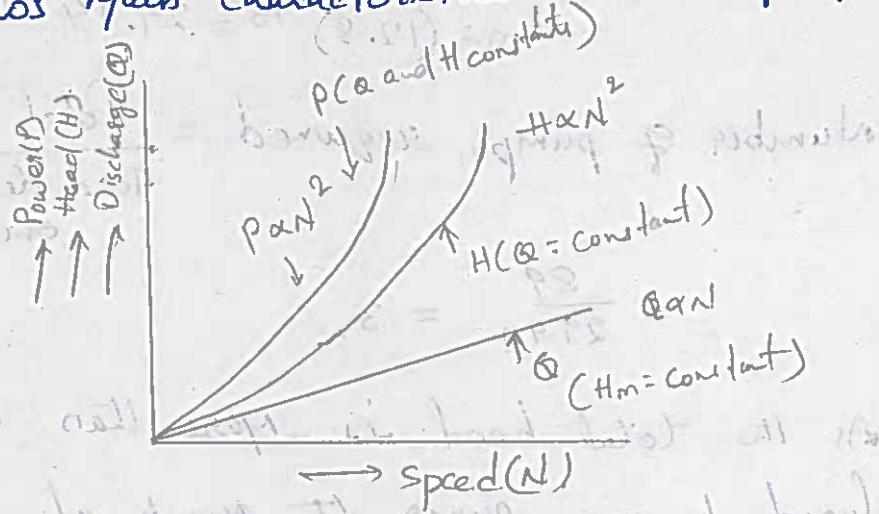
Characteristics curve of centrifugal pump are defined as those curve which are plotted from the results of the number of tests on the centrifugal pump. These curves are necessary to predict the behaviour and performance of the pump when the pump is working under different flow rate, head and speed. The following are the important characteristic curves for pumps.

1. Main characteristic curves.
2. Operating characteristic curves.
3. Constant efficiency or Nussel curves.

(i) Main characteristic curves:-

The main characteristic curves of a centrifugal pump consists of variation of head, power and discharge with respect to speed. For plotting curves of η manometric head versus speed, discharge is kept constant. And for plotting curves of discharge versus speed, manometric head is kept constant. And for plotting curves of power versus speed, both manometric head and discharge are kept constant.

Fig shows main characteristics curves of a pump.



~~Fig~~ Main characteristic Curves of a pump.

For plotting the graph of H_m versus speed (N), the discharge is kept constant. From equation it is clear that $\sqrt{H_m}/DN$ is constant. This means the head developed by a pump is proportional to N^2 . Hence the curve of H_m vs N is a parabolic curve as shown in fig 19.14.

From equation (19.22) it is clear that P/DN^3 is a constant. Hence $P \propto N^3$. This means that the curve P vs N is a cubic curve as shown fig. 19.14

Equation (19.21) shows that $\frac{Q}{DN} = \text{constant}$. This means $Q \propto N$ for a given pump. Hence the curve Q vs N is a straight line as shown in fig 19.14.

2) Operating Characteristic Curves :-

If the speed is kept constant, the variation of manometric head power and efficiency with respect to discharge gives the operating characteristic of the pump. Fig 19.15 shows the operating characteristic curve of a pump.

The input power curve for pumps shall not pass through the origin. It will be slightly away from the origin on the y-axis as even at zero discharge some power is needed to overcome mechanical losses.

The head curve will have maximum value of head when discharge is zero.

The output power curve will start from origin as at $Q=0$. Output power will be zero.

The efficiency curve will start from origin as at $Q=0, \eta=0$.

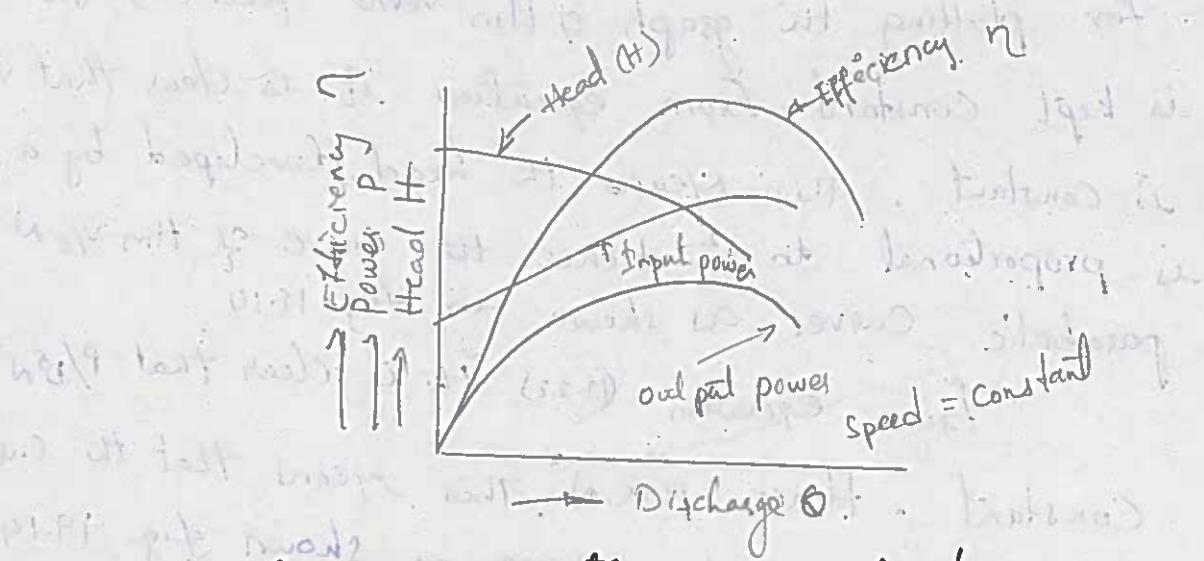


Fig. 19.15 Operating characteristic curves of pump.

3) Constant Efficiency Curves :-

For obtaining Constant efficiency Curves for a pump, the head versus discharge Curves and Efficiency versus discharge Curves for different speed are used, Fig 19.16 (a) shows the head versus discharge curves for different speed. The efficiency versus discharge curves for the different speeds are shown (19.16) (b) By Combing these curve ($H \sim Q$ curves and $\eta - Q$ curves) constant efficiency curves are obtained as shown in Fig 19.16 (c).

For plotting the constant efficiency curves horizontal lines representing constant efficiencies are drawn on the $\eta \sim Q$ curves. The points at which these lines cut the efficiency curves at various speed, are transferred to the corresponding $H \sim Q$ curves. The points having the same efficiency are then joined by smooth curves. These smooth curves represent the iso-efficiency curves.

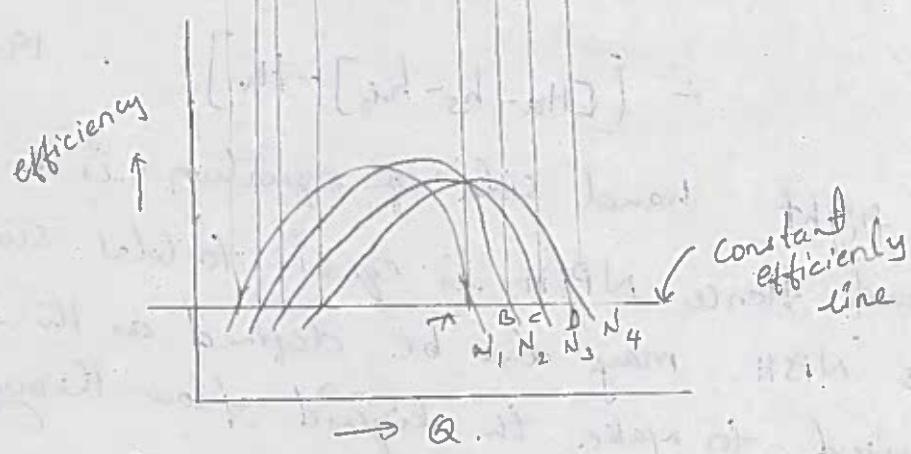
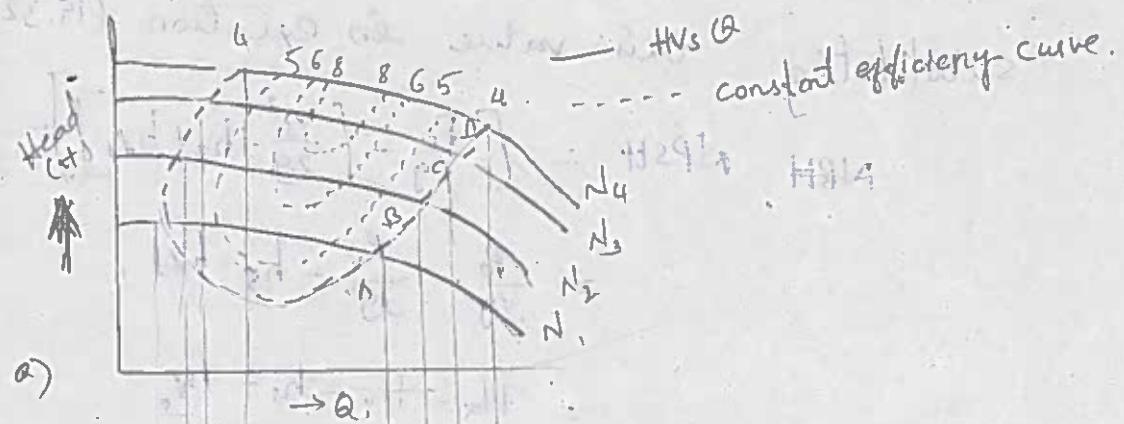


Fig 19.16. Constant efficiency curves of a pump.

* Net positive suction head (NPSH).

The term NPSH is very commonly used in the pump industry. Actually the minimum suction conditions are more frequently specified in terms of NPSH.

The net positive suction head (NPSH) is defined as the absolute pressure head at the inlet to the pump minus the vapour pressure head plus the velocity head.

$$\therefore \text{NPSH} = \text{Absolute pressure head at inlet of the pump} - \text{vapour pressure head} + \text{velocity head.}$$

$$= \frac{P_1}{\rho g} - \frac{P_v}{\rho g} + \frac{V^2}{2g} \quad (19.32),$$

But from equation (17) of Art. 19.12 the absolute pressure head at inlet of the pump is given by as.

$$\frac{P_1}{\rho g} = \frac{P_0}{\rho g} - \left[\frac{V_s^2}{2g} + h_s + h_{fs} \right].$$

Substituting this value in equation (19.32) we get.

$$\begin{aligned} \text{NPSH}_{\text{A}} &= \left[\frac{P_0}{\rho g} - \left[\frac{V_s^2}{2g} + h_s + h_{fs} \right] \right] - \frac{P_v}{\rho g} + \frac{V_s^2}{2g} \\ &= \frac{P_0}{\rho g} - \frac{P_v}{\rho g} - h_g - h_{fs} \\ &= H_a - H_v - h_s - h_{fs} \\ &= [H_a - h_s - h_{fs}] - H_v. \end{aligned}$$

— 19.33 —

The right hand side of equation is the total suction head. Hence NPSH is equal to total suction head. Thus NPSH may also be defined as the total head required to make the liquid flow through the suction pipe to the pump impeller.

For any pump installation, a distinction is made b/w the required NPSH and available NPSH. The value of required NPSH is given by manufacturer. This value can also be determined at which the pump gives maximum efficiency without any operational noise.

When the pump is installed, the available NPSH is calculated from equation. In order to have cavitation free operation of centrifugal pump its available NPSH should be greater than the NPSH.

* Cavitation in Centrifugal pumps :-

In centrifugal pumps the cavitation may occur at the inlet of the impeller of the pump or at the suction side of the pump, where the pressure of the liquid less than the of the pump, where the pressure of the liquid can be cavitation may occur. The cavitation in a pump can be noted by a sudden drop in efficiency and head. In order to determine whether cavitation will occur in any position of the suction side of the pump, the critical value of Thomas' cavitation factor or is calculated.

Thomas' Cavitation Factor for Centrifugal pumps !

The mathematical expression for Thomas' cavitation factor for centrifugal pump is given by.

$$\sigma = \frac{(H_b) - H_s - h_{fs}}{H}$$

$$= \frac{(H_{atm} - H_v) - H_b - h_{fs}}{H} \rightarrow (19.24)$$

where H_{atm} = Atmospheric pressure head in m of water
or absolute pressure head at the liquid surface in pump.

H_v = Vapour pressure head in m of water

H_s = Suction pressure head in m of water

h_{fs} = Head lost due to friction in suction pipe

H_b = Head developed by the pump.

Thomas' Cavitation factor is used to indicate whether cavitation will occur in pumps. Equation (19.24) gives the values of Thomas' Cavitation factor for pumps are

$$G = \frac{(H_{atm} - H_r) - H_s - h_{s_s}}{H}$$

$$\frac{H_a + H_v - H_s - H_w}{H_m}$$

But from equation (19.33), we have

$$H_a - H_r - h_S - h_{\bar{S}} = \Delta P_{SH}$$

$$G = \frac{NPA}{Hm}$$

If the value of σ is less than the critical value σ_c ,

When cavitation will occur in the pumps. The value

q & σ_c depends upon the specific speed of the pump.

$$N_S = \frac{\sqrt{A}}{H_m^{3/4}}$$

The following empirical relation is used to determine the value σ_c

$$\sigma_c = 0.103 \left(\frac{N_s}{1000} \right)^{4/3}$$

$$= 0.103 \frac{N_3^{4/3}}{(10^3)^{4/3}} = \frac{0.103 N^{4/3}}{10^4}$$

$$= 1.03 \times 10^3 N^{4/3} - \underline{\underline{19.35}}$$

→ o =

Part-II.

Hydro power Engineering :-

* Components of Hydropower plants:-

The various components of hydropower plants include

- (i) Storage or diversion dam ;
- (ii) Waterways such as tunnel , power channel or penstocks , with forebay and necessary appurtenances such as intake structures , air vent valve , surge tanks, etc
- (iii) Power house with turbine , generator and other appurtenances
- (iv) Switch yard for transmission of power .

Dams:-

A dam is a structure constructed across a river at a suitable site to develop an artificial reservoir for the storage of water and to create head. Dams may be of different types such as earth dam , masonry dam , concrete dam etc. Out of these concrete dams are quite common . The different types of gates used are plain sliding gates , wheeled or roller gates etc.

Similarly the common type of valves used are butterfly and needle valves .

Waterways:- A waterway is a passage through which water is carried from its storage reservoir to the power house . It may consist of a tunnel , channel or penstock . If a hill intervenes the reservoir and the power house then a tunnel may be driven to provide a water way . It may flow full or pressure conduit or partly full as the open channel .

Penstocks :- Penstocks are the pipes of large diameters used as waterways for conveying water from the reservoir to the power house. The thickness of penstock is determined on the basis on the magnitude of stresses developed due to static and water hammer loads.

Air valves are provided at the sections of the penstock where there are steep changes in the gradient. In the case of long steel penstocks, expansion joints should be provided to take care of expansion and contraction due to changes in temperature.

Alternatively an open channel to be followed by penstocks along the slopes may also be the choice as waterway. The power channel may be rectangular, trapezoidal or circular and may be lined or unlined.

Forebay :-

A forebay is enlarged body of water provided just in front of the penstocks. The main function of the forebay is to provide a small balancing storage upstream of the power house to store temporarily the water rejected by the plant when the load is reduced and to meet the instantaneous increased demand when the load is instantaneously increased.

When water is carried through a channel, the forebay may be developed by enlarging the channel just upstream of the intake for the penstocks leading water to the turbines in the power house.

Intake structure :-

The water from the reservoir of forebay is let into the penstocks through intake structure. The main components of an intake structure are trash racks and gates. A trash rack is a grating made of a series of steel bars of rectangular cross-section set parallel to each other and placed across the entire intake opening in an inclined position. It is provided to prevent the debris entering into the water passage of the hydropower plant which may otherwise damage the plant equipment. A debris cleaning device is usually fitted on the trash rack. This is so because if the air inlets are not provided the sudden drainage of the penstock may lead to its collapse.

Surge tank :-

As explained in chapter 21 a surge tank is cylindrical open-topped storage reservoir which is provided to protect the penstocks against water hammer pressure. It however provides protection for only that portion of the penstock which lies on upstream of it. As such a surge tank is given in chapter 21.

Power House :-

A power house of a hydroelectric scheme houses the various hydraulic and electric equipment. The various hydraulic equipment are turbines, gates or gate valve, governors etc. The various electrical equipment include generator, transformer, switching equipment

transmission lines and transmission structures, auxiliary electrical equipment etc. A switch yard for the transmission of power is usually located outdoors near the power house.

Types of hydropower plants :-

- Hydropower plants may be classified according to several considerations. According to the storage being provided or not hydropower plants may be classified as

- (i) Run-off-river plants
- (ii) Reservoir or storage plant
- (iii) Pumped storage plants
- (iv) Tidal plants.

(i) Run off river plant :-

Run-off-river plants are those which utilize the flow as it comes, without any storage being provided. As such these plants would be feasible only on such rivers which have a minimum dry weather flow of such magnitude which makes the development worthwhile. A weir or barrage may be constructed across the river close to the power plant to maintain a given water level.

These are generally low head plants and often at times of flood, tail water rises to such an extent that the plants are inoperative.

Many run off river plants are provided with pondage, which enables them to take care of hour to hour fluctuations in load on the plant throughout the period of a week.

(i) Reservoir plants:-

These are the hydropower plants which take their flow from large storage reservoir developed by constructing dams across rivers. Depending on the storage volume, these plants can hold over surplus water from the period when stream flow.

(ii) Pumped storage plants:-

Pumped storage hydro plants are those which pump all or portion of the water used by these plants, back to the head-water pond, to be made available again for the power generation. Essentially they consist of a tail-water pond and a head water pond. During the times of peak load, water is drawn from the head water pond through the penstock to operate hydroelectric generating units. Power for operating the pumps is provided by some of peak thermal or hydropower plant. Alternatively pumped storage plants may have separate pumping and power generating units, especially for high heads for which multistage centrifugal pumps are used for pumping water and high head Francis turbines are installed for power generation.

(iv) Tidal plants:-

Tidal plants do not involve any storage of water. These plants work on the principle that there is a rise in sea water during the high tide period and a fall during the low tide or ebb tide period.

Sea water rises and falls twice a day, each full cycle occupying about 12 hours 25 mins. The tidal range or the difference b/w the high tide and the low tide level is utilized to generate power.

Hydropower plants may also be classified on a functional basis as

(i) Base Load plants.

(ii) Peak load plants.

(i) Base load plant :-

As the name indicates base load plants are those which are capable of substantially continuous operating in the base of the load curve throughout the year. Both run of river plants as well as reservoir plants can be used as base load hydro plants. However, when run of river plants without pondage are used as base load hydro plants, their full plant discharge is seldom more than the minimum flow of river.

(ii) Peak load plants :-

A peak load plant is one designed and constructed primarily for taking care of the peak load of a power system. Pumped storage plants are peak - load plants. Run of river plants with pondage can operate both as peak load and base - load plants at river flow permits.

Hydroplants may also be classified, on the basis of head under which they operating as

(i) Low head plants

(ii) Medium head plants

(iii) High head plants.

A low head plant is the one which is operating under a head of less than about 30m. Run of river plants are usual low head plants. A medium head plant is the one which is operating under a head b/w 30 to 250 m. A high head plant is the one which is operating under a head of more than about 250m. On other hand the higher ranges of medium head as well as high heads may be obtained by constructing dams of sufficient height and locating the power house either at the toe of the dam close to it, or in a deep depression if available at a certain distance away from the dam.

The above noted head ranges for the different types of hydropower plant are however arbitrary and it is customary to associate these head ranges with the type of turbine used. With the advances in the turbine design it has become possible to use axial and mixed flow turbines for higher heads. Consequently the ranges of head indicated above also move up.

* Load Factor, Utilisation Factor And Capacity Factor.

Load Factor :-

Load Factor is defined as the ratio of the average load during a certain period to the maximum or peak load during that period. The load factor is thus related to a certain period and therefore, there will be daily load factors, weekly load factors, load factor for

a gives monthly and yearly load factors. For instance, during a certain week a power plant generates 8400000 kwh and peak load during the week is 100000 kw. Therefore, the load factor during the week is

$$\frac{8400000}{24 \times 7 \times 100000} = 0.50 \text{ or } 50\%.$$

The load factor of a power plant would vary greatly with the character of the load. In highly industrialised area the load factors will be high, but in residential areas the load factors may be as low as 25 to 30%. The installed capacity of a power plant has to be equal to peak load but the total number of units generated will be governed by the average load. Conversely, If the load factor of a power plant is high the generating capacity is better utilised and the cost of generation is relatively less.

Utilization Factor :-

Utilization factor is defined as the ratio of the peak load developed during a certain period to the installed capacity of the plant. It thus represents the maximum proportion of the installed capacity utilized during that period. With constant head, utilization factor would also be the ratio of water actually utilized for generating maximum power corresponding to peak load to that available in the river, and usually there will be little difference in this factor, whether expressed as a ratio of power or water.

Capacity factor :-

Capacity factor may be defined as the ratio of the energy that the plant actually produces during any period to the energy that it might have produced if operated at full capacity throughout this period.

For instance, if during a particular week the peak load on a power plant with a capacity of 100,000 kw was 65,000 kw and if the energy produced by the plant was 6720000 kwh, the capacity factor for the week (168 hr).

$$= \frac{6720000}{100000 \times 168} = 0.4 \text{ or } 40\%$$

During the same period the load factor.

$$= \frac{6720000}{65000 \times 168} = 0.616 \text{ or } 61.6\%$$

Capacity factor will be identical with load factor when the maximum of peak load just equal the plant capacity. For a hydroelectric plant, capacity factor commonly varies from about 0.25 to 0.70 more, depending on load factor plant, capacity, available pondage and storage etc..

The thermal power plants may be operated at any desired capacity factor, whereas the annual capacity factor at which hydro power plants may operate is usually limited by the variation in the flow of water in the river. Moreover, there is always a decline in the annual capacity factor of thermal power plant with its age. Pg-37

* Load Factor:-

= Avg load for a given interval (L_{av})

Max (or) peak load for same time interval (L_{max}).

The value of load factor is always less than one.

This particular period of time may be one day.

(1 week, 1 month).

Capacity / plant factor :-

= Avg load (L_{av})

rated Capacity of plant (C).

The value will be zero, if there is no load on hydro electric Power plant.

Utilisation factor :-

= Max (or) Peak load (L_{max})

rated capacity of the plant (C)

Inter relation along load, capacity, utilisation factor.

$$\text{Load Factor} = \frac{L_{av}}{L_{max}} \times \frac{C}{C}$$

$$= \frac{L_{av}/C}{L_{max}/C}$$

UNIT-5 \Rightarrow Capacity factor is nothing but utilization factor.

Pg - 33 \Rightarrow Capacity factor = Load factor \times Utilization factor.

* Estimation of hydropower potential.

Hydropower plant convert water power to electric power (energy)

$$P = \frac{\rho g Q (H - h_f) \times \eta}{1000} \text{ (kw)}$$

η = eff. of station

Q = Rate of water flow = $A \times h$.

A = Catchment Area.

h = Rainfall intensity.

$H - h_f$ = net eff. head = h_e .

Problem

a.) The gross head & discharge of a hydroelectric plant are 40m & $200 \text{ m}^3/\text{sec}$ resp. The losses in penstock are 12%. If the turbine works with an efficiency of 92.1. what will be the power developed

Sol $h_e = 40 - (40 \times \frac{12}{100}) = 35.2 \text{ m}$

$$P = \frac{\rho g Q (h_e) \times \eta}{1000}$$

$$= \frac{1000 \times 9.81 \times 200 \times 35.2 \times 0.92}{1000}$$

$$\boxed{P = 63.537 \text{ kw.}}$$

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