Hydraulic Machines BME-51

Unit-3

(Lecture 1)

Lecture contains

- **Centrifugal Pumps**
- **Classifications of centrifugal pumps**
- **Construction and working**

Introduction

A pump is a hydraulic machine which converts mechanical energy into hydraulic energy or pressure energy.

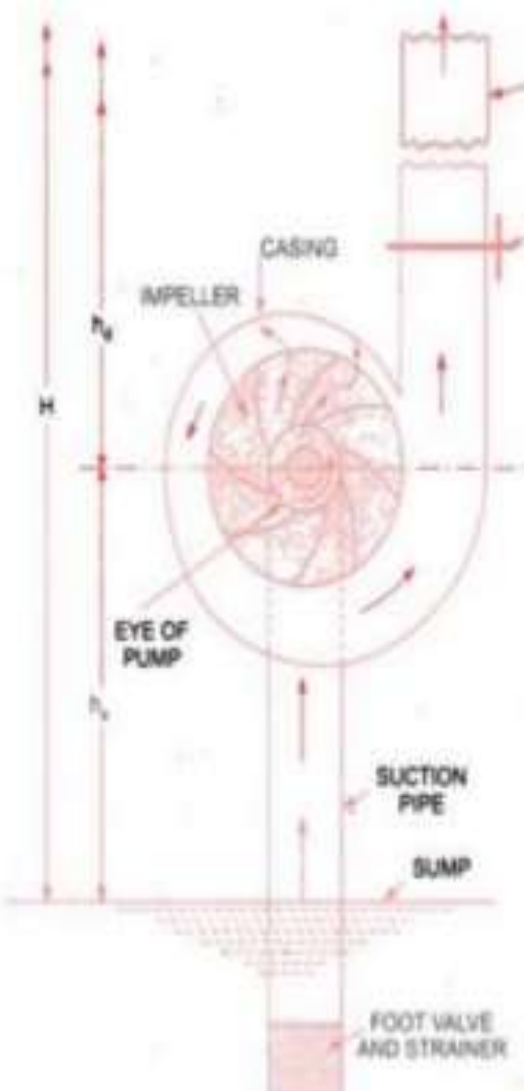
A centrifugal pump works on the principle of centrifugal force.

In this type of pump the liquid is subjected to whirling motion by the rotating impeller which is made of a number of backward curved vanes. The liquid enters this impeller at its center or the eye and gets discharged into the casing enclosing the outer edge of the impeller.

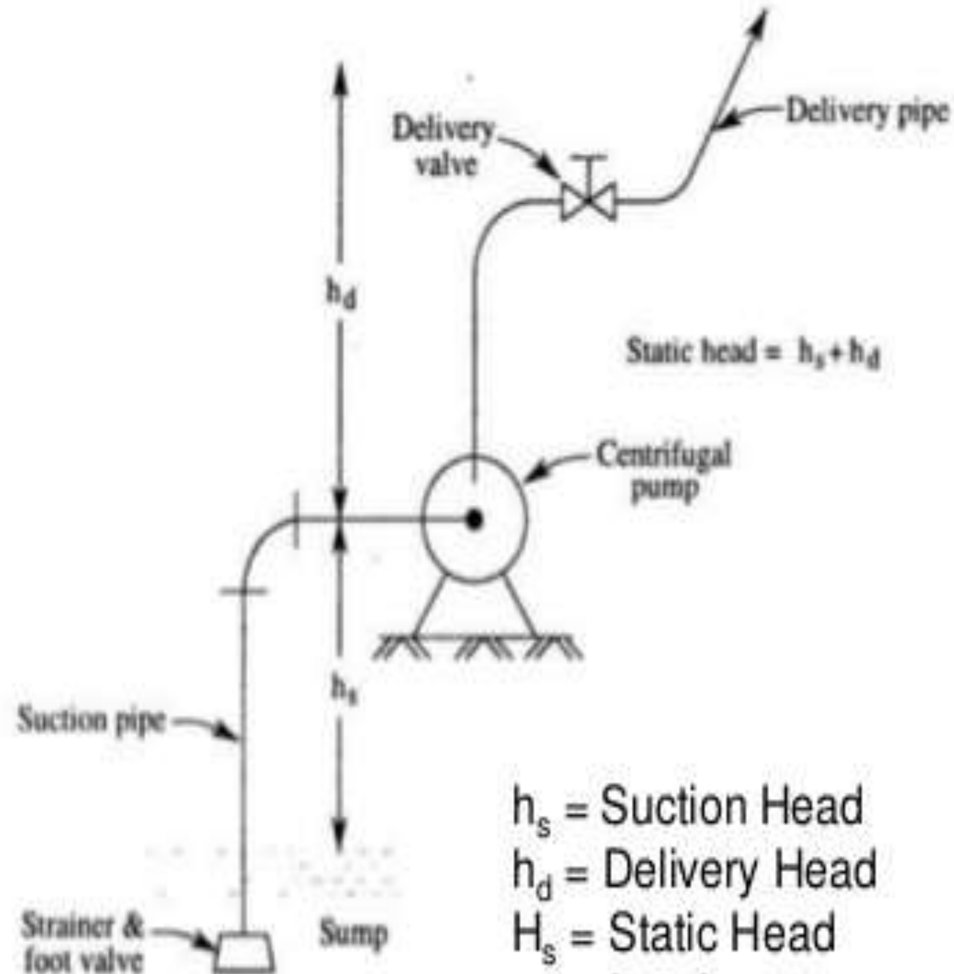
Generally centrifugal pumps are made of the **radial flow** type only ($\alpha = 90^0$)

Classification of Pumps

1. According to No. of Impellers
   a) Single Stage Pump
   b) Multistage Pump
2. According to Disposition of Shaft
   a) Vertical Shaft Pump
   b) Horizontal Pump
3. According to Head
   a) Low Head Pump - $H < 15m$
   b) Medium Head Pump - $15m < H < 40m$
   c) High Specific Speed Turbine - $H > 40m$



Main parts of a centrifugal pump.

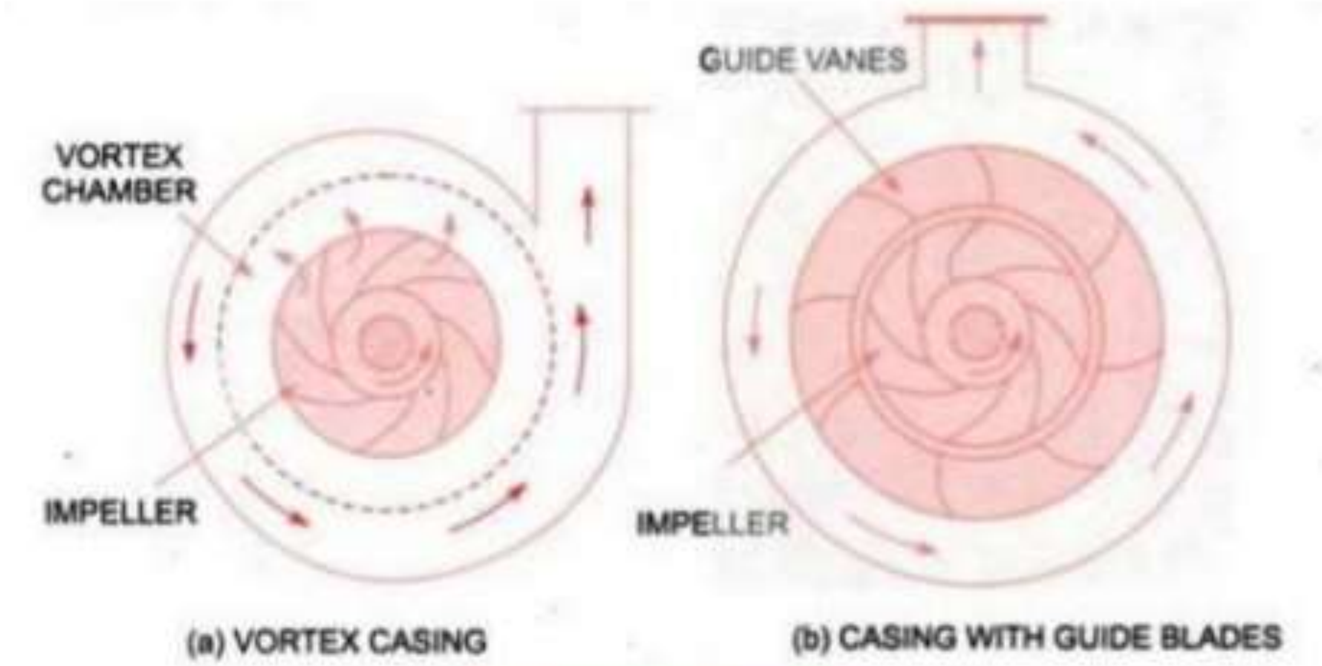


h_s = Suction Head
h_d = Delivery Head
H_s = Static Head
= $h_s + h_s$

Components of Centrifugal Pump

Components of Pump

1. Strainer and Foot Valve
2. Suction Pipe and its fittings
3. Pump
4. Delivery Valve
5. Delivery Pipe and its fittings



Different types of casing.

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(Lecture 2)

Lecture contains

- Work done by impellor
- Manometric Head

WORK DONE BY THE CENTRIFUGAL PUMP (OR BY IMPFLLER) 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 and outlet of the impeller in the same way as for a turbine. The water enters the impeller radially at inlet for best efficiency of the pump, which means the absolute velocity of water at inlet makes an angle of 90° with the direction of motion of the impeller at inlet. Hence angle $\alpha = 90°$ and $V_{w_1} = 0$. For drawing the velocity triangles, the 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 the impeller in r.p.m.,

D_1 = Diameter of impeller at inlet,

u_1 = Tangential velocity of impeller at inlet,

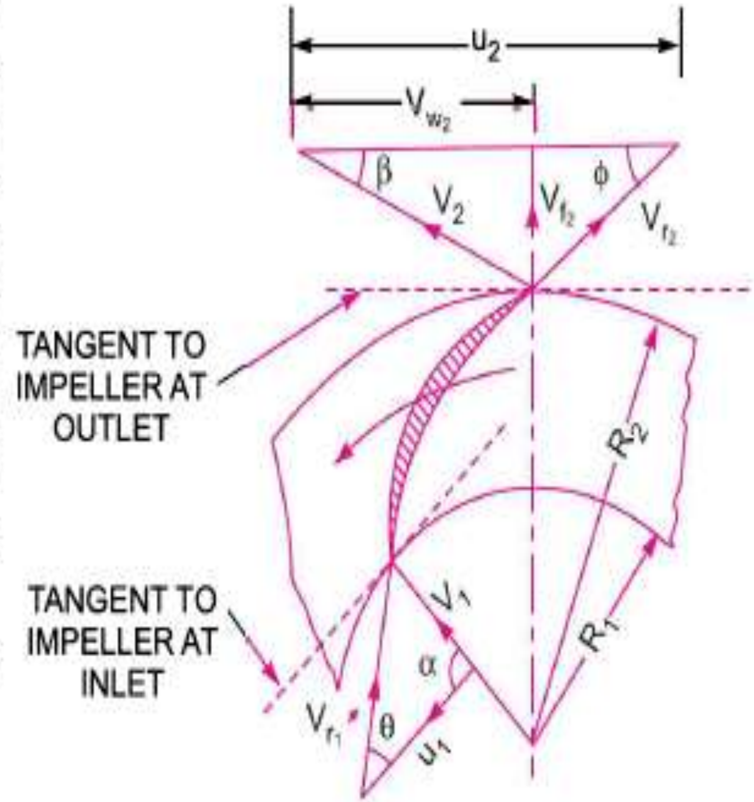

Fig. *Velocity triangles at inlet and outlet.*

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

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_{r_1} = Relative velocity of water at inlet,

α = Angle made by absolute velocity (V_1) at inlet with the direction of motion of vane,

θ = Angle made by relative velocity (V_{r_1}) at inlet with the direction of motion of vane, and V_2, V_{r_2}, β 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 $\alpha = 90°$ and $V_{w_1} = 0$.

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

$$= \frac{1}{g}[V_{w_1}u_1 - V_{w_2}u_2]$$

$\therefore$ Work done by the impeller on the water per second per unit weight of water striking per second

$$= -\text{[Work done in case of turbine]}$$

$$= -\left[\frac{1}{g}\left(V_{w_1}u_1 - V_{w_2}u_2\right)\right] = \frac{1}{g}\left[V_{w_2}u_2 - V_{w_1}u_1\right]$$

$$= \frac{1}{g}V_{w_2}u_2 \qquad (\because \quad V_{w_1} = 0 \text{ here})$$

Work done by impeller on water per second

$$= \frac{W}{g} . V_{w_2} u_2$$

where W = Weight of water = $\rho \times g \times Q$

where Q = Volume of water

and Q = Area × Velocity of flow = $\pi D_1 B_1 \times V_{f_1}$

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

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

Manometric head

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 the impeller to the water – Loss of head in the pump

$$= \frac{V_{w_2} u_2}{g} - \text{Loss of head in impeller and casing}$$

$$= \frac{V_{w_2} u_2}{g} \quad \text{...if loss of pump is zero}$$

(*b*) H_m = Total head at outlet of the pump – Total head at the inlet of the pump

$$= \left(\frac{P_o}{\rho g} + \frac{V_o^2}{2g} + Z_o\right) - \left(\frac{p_i}{\rho g} + \frac{V_i^2}{2g} + Z_i\right)$$

where $\frac{p_o}{\rho g}$ = Pressure head at outlet of the pump = h_d

$\frac{V_o^2}{2g}$ = 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, and

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

i.e., h_s, $\frac{V_s^2}{2g}$ and Z_s respectively.

(*c*) $$H_m = h_s + h_d + h_{f_s} + h_{f_d} + \frac{V_d^2}{2g}$$

where h_s = Suction head, h_d = Delivery head,
h_{f_s} = Frictional head loss in suction pipe, h_{f_d} = Frictional head loss in delivery pipe, and
V_d = Velocity of water in delivery pipe.

(Lecture 3)

Lecture contains

- Efficiencies of centrifugal pumps

- Problems

Efficiencies of centrifugal pumps:-

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 and
(*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

$$\eta_{man} = \frac{\text{Manometric head}}{\text{Head imparted by impeller to water}}$$

$$= \frac{H_m}{\left(\dfrac{V_{w_2} u_2}{g}\right)} = \frac{gH_m}{V_{w_2} u_2}$$

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

The power given to water at outlet of the pump $= \dfrac{WH_m}{1000}$ kW

The power at the impeller $= \dfrac{\text{Work done by impeller per second}}{1000}$ kW

$$= \frac{W}{g} \times \frac{V_{w_2} \times u_2}{1000} \text{kW}$$

$$\eta_{man} = \frac{\dfrac{W \times H_m}{1000}}{\dfrac{W}{g} \times \dfrac{V_{w_2} \times u_2}{1000}} = \frac{g \times H_m}{V_{w_2} \times u_2}.$$

(*b*) **Mechanical Efficiency (η_m).** The power at the shaft of the centrifugal pump is more than the power available at the impeller of the pump. The ratio of the power available at the impeller to the power at the shaft of the centrifugal pump is known as mechanical efficiency. It is written as

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

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

$$= \frac{W}{g} \times \frac{V_{w_2} u_2}{1000}$$

$$\eta_m = \frac{\frac{W}{g}\left(\frac{V_{w_2} u_2}{1000}\right)}{\text{S.P.}}$$

where S.P. = Shaft power.

(*c*) **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 the electric motor

= S.P. of the pump.

$$\therefore \qquad \eta_o = \frac{\left(\frac{WH_m}{1000}\right)}{\text{S.P.}}$$

Also $\qquad \eta_o = \eta_{man} \times \eta_m.$

Problem *The internal and external diameters of the impeller of a centrifugal pump are 200 mm and 400 mm respectively. The pump is running at 1200 r.p.m. The vane angles of the impeller at inlet and outlet are 20° and 30° respectively. The water enters the impeller radially and velocity of flow is constant. Determine the work done by the impeller per unit weight of water.*

Problem *A centrifugal pump delivers water against a net head of 14.5 metres 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%.*

Problem *A centrifugal pump having outer diameter equal to two times the inner diameter and running at 1000 r.p.m. works against a total head of 40 m. The velocity of flow through the impeller is constant and equal to 2.5 m/s. The vanes are set back at an angle of 40° at outlet. If the outer diameter of the impeller is 500 mm and width at outlet is 50 mm, determine :*

(i) Vane angle at inlet, *(ii) Work done by impeller on water per second, and*

(iii) Manometric efficiency.

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(Lecture 4)

Lecture contains

- Minimum starting speed of Centrifugal pump
- Specific speed
- Model testing

Minimum Starting Speed of Pump

A centrifugal pump will start delivering liquid only if the head developed by the impeller is more than the manometric head (H_m). If the head developed is less than H_m no discharge takes place although the impeller is rotating. When the impeller is rotating, the liquid in contact with the impeller is also rotating. This is a forced vertex, in which the increase in head in the impeller is given by

$$\text{Head rise in impeller} = \frac{u_2^2}{2g} - \frac{u_1^2}{2g}$$

Discharge takes place only when

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g} \geq H_m$$

substituting for u_1, u_2 and H_m in Equation (10.13), we obtain

$$N = \frac{120\eta_m V_{w_2} D_2}{\pi(D_2^2 - D_1^2)}$$

which is the minimum speed for the pump to discharge liquid.

Specific Speed of Pump

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'.

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

MODEL TESTING OF CENTRIFUGAL PUMPS

Before manufacturing the large sized pumps, their models which are in complete similarity with the actual pumps (also called prototypes) are made. Tests are conducted on the models and performance of the prototypes are predicted. The complete similarity between the model and actual pump (prototype) will exist if the following conditions are satisfied :

1. Specific speed of model = Specific speed of prototype

$$(N_s)_m = (N_s)_p \quad \text{or} \quad \left(\frac{N\sqrt{Q}}{H_m^{3/4}}\right)_m = \left(\frac{N\sqrt{Q}}{H_m^{3/4}}\right)_p$$

2. Tangential velocity (u) is given by $u = \dfrac{\pi DN}{60}$ also $u \propto \sqrt{H_m}$

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

$$\therefore \quad \frac{\sqrt{H_m}}{DN} = \text{Constant}$$

or $$\left(\frac{\sqrt{H_m}}{DN}\right)_m = \left(\frac{\sqrt{H_m}}{DN}\right)_p$$

3. From equation (*ii*) of Art. 19.7.1, we have

$$Q \propto D^2 \times V_f \qquad \text{where } V_f \propto u \propto DN$$
$$\propto D^2 \times D \times N$$
$$\propto D^3 \times N$$

$$\therefore \quad \frac{Q}{D^3N} = \text{Constant} \quad \text{or} \quad \left(\frac{Q}{D^3N}\right)_m = \left(\frac{Q}{D^3N}\right)_p$$

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

$$\therefore \quad P \propto Q \times H_m$$
$$\propto D^3 \times N \times H_m \qquad (\because \ Q \propto D^3N)$$
$$\propto D^3N \times D^2N^2 \qquad (\because \ \sqrt{H_m} \propto DN)$$
$$\propto D^5N^3$$

$$\therefore \quad \frac{P}{D^5N^3} = \text{Constant} \quad \text{or} \quad \left(\frac{P}{D^5N^3}\right)_m = \left(\frac{P}{D^5N^3}\right)_p$$

(Lecture 5)

Lecture contains

- **Multistage centrifugal pump**
- **Pump in series and parallel**

Multistage centrifugal pump:-

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 shafts. A multistage pump is having the following two important functions :

1. To produce a high head, and
2. To discharge a large quantity of liquid.

If a high head is to be developed, the impellers are connected in series (or on the same shaft) while for discharging large quantity of liquid, the impellers (or pumps) are connected in parallel.

1 Multistage Centrifugal Pumps for High Heads. For developing a high head, a number of impellers are mounted in series or on the same shaft as shown in Fig. 1

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 2nd impeller with the help of a connecting pipe as shown in Fig. 1 At the outlet of the 2nd 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 the outlet will be increased further.

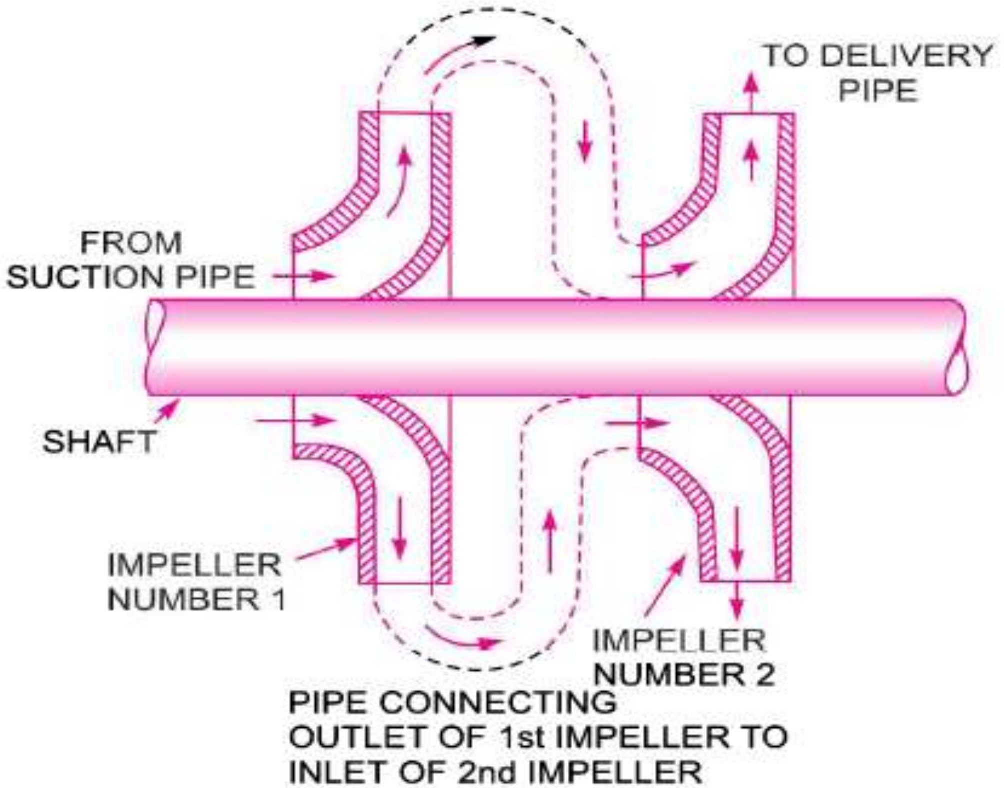

Fig. 1 *Two-stage pumps with impellers in series.*

Let n = 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

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2 Multistage Centrifugal Pumps for High Discharge. For obtaining high discharge, the pumps should be connected in parallel as shown in Fig. 2. Each of the pumps 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.

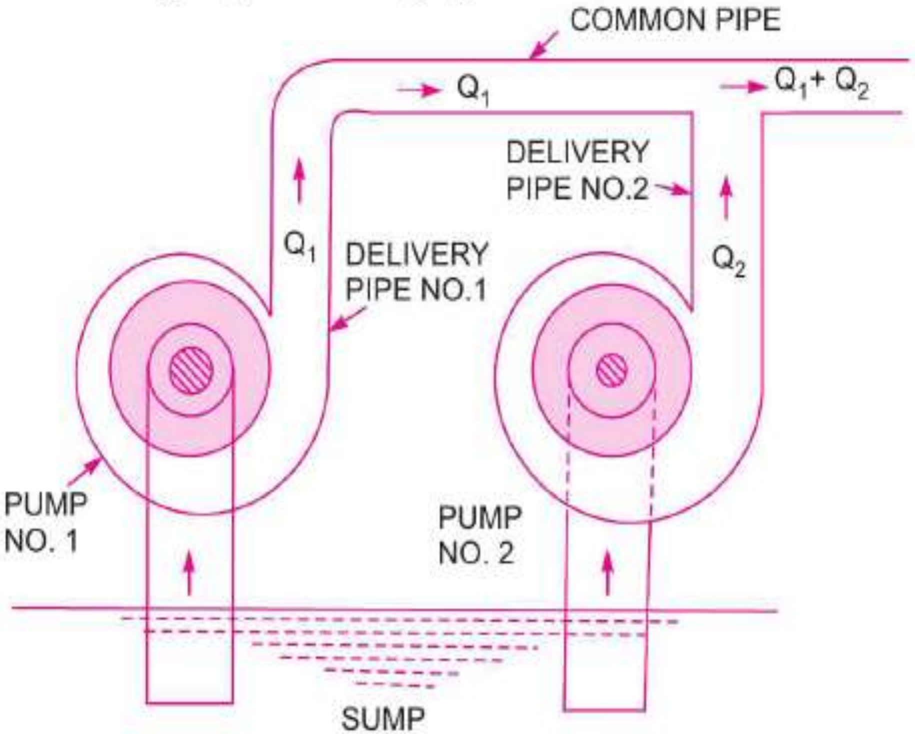

Fig. 2 *Pumps in parallel.*

Let n = Number of identical pumps arranged in parallel.

Q = Discharge from one pump.

$\therefore$ Total discharge $= n \times Q$

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Unit-3 (Lecture 5)

Problem *A three stage centrifugal pump has impellers 40 cm in diameter and 2 cm wide at outlet. The vanes are curved back at the outlet at 45° and reduce the circumferential area by 10%. The manometric efficiency is 90% and the overall efficiency is 80%. Determine the head generated by the pump when running at 1000 r.p.m. delivering 50 litres per second. What should be the shaft horse power ?*

Problem *A four-stage centrifugal pump has four identical impellers, keyed to the same shaft. The shaft is running at 400 r.p.m. and the total manometric head developed by the multistage pump is 40 m. The discharge through the pump is 0.2 m^3/s. The vanes of each impeller are having outlet angle as 45°. If the width and diameter of each impeller at outlet is 5 cm and 60 cm respectively, find the manometric efficiency.*

Problem *A single-stage centrifugal pump with impeller diameter of 30 cm rotates at 2000 r.p.m. and lifts 3 m^3 of water per second to a height of 30 m with an efficiency of 75%. Find the number of stages and diameter of each impeller of a similar multistage pump to lift 5 m^3 of water per second to a height of 200 metres when rotating at 1500 r.p.m.*

Problem *A one-fifth scale model of a pump was tested in a laboratory at 1000 r.p.m. The head developed and the power input at the best efficiency point were found to be 8 m and 30 kW respectively. If the prototype pump has to work against a head of 25 m, determine its working speed, the power required to drive it and the ratio of the flow rates handled by the two pumps.*

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Unit-3

(Lecture 6)

Lecture contains

- **Priming of centrifugal pump**
- **Performance characteristics**

Priming of centrifugal pump:-

Priming of a centrifugal pump is defined as the operation in which the suction pipe, casing of the pump and a portion of the delivery pipe upto the delivery valve is completely filled up from outside source with the liquid to be raised by the pump before starting the pump. Thus the air from these parts of the pump is removed and these parts are filled with the liquid to be pumped.

The work done by the impeller per unit weight of liquid per sec is known as the head generated by the pump. Equation (19.1) gives the head generated by the pump as $= \frac{1}{g} V_{w_2} u_2$ metre. This equation is independent of the density of the liquid. This means that when pump is running in air, the head generated is in terms of metre of air. If the pump is primed with water, the head generated is same metre of water. But as the density of air is very low, the generated head of air in terms of equivalent metre of water head is negligible and hence the water may not be sucked from the pump. To avoid this difficulty, priming is necessary.

Performance characteristics:-

CHARACTERISTIC CURVES OF CENTRIFUGAL PUMPS

Characteristic curves of centrifugal pumps are defined as those curves which are plotted from the results of a 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, and
3. Constant efficiency or Muschel curves.

I Main Characteristic Curves. The main characteristic curves of a centrifugal pump consists of variation of head (manometric head, H_m), power and discharge with respect to speed. For plotting curves of manometric head *versus* speed, discharge is kept constant. For plotting curves of discharge *versus* speed, manometric head (H_m) is kept constant. And for plotting curves of power *versus* speed, the manometric head and discharge are kept constant. Fig. 1 shows main characteristic curves of a pump.

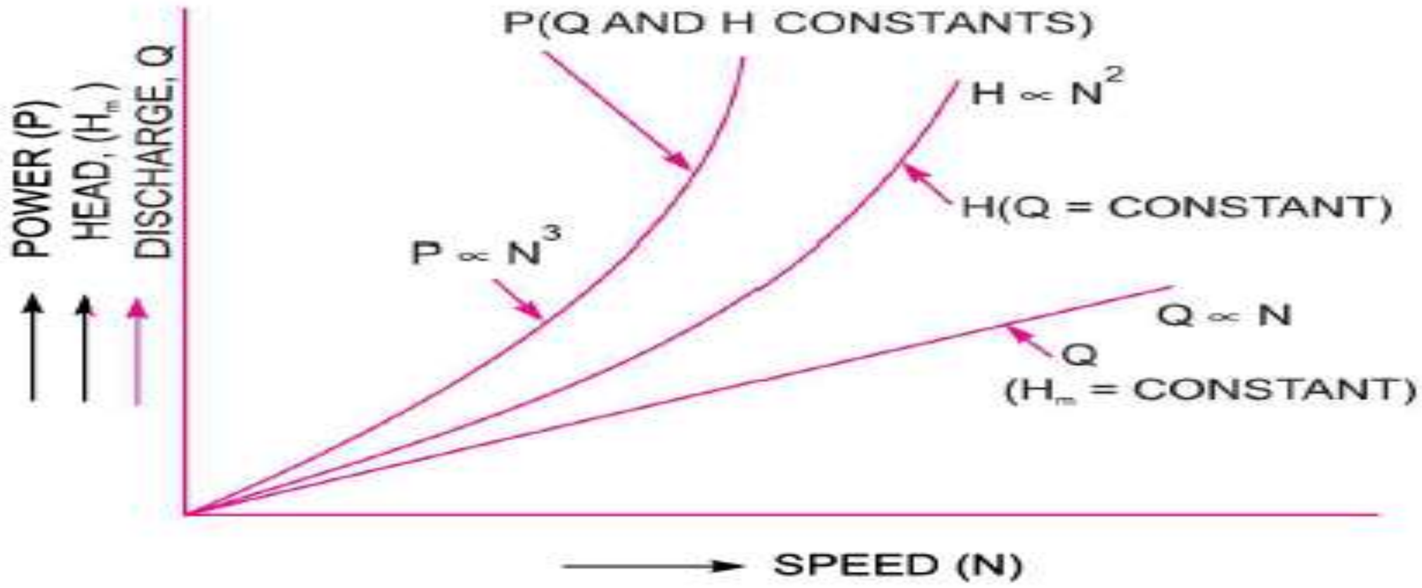

Fig. 1 *Main characteristic curves of a pump.*

For plotting the graph of H_m versus speed (N), the discharge is kept constant.

, it is clear that $\sqrt{H_m}/DN$ is a constant or $H_m \propto N^2$. This means that head developed by a pump is proportional to N^2. Hence the curve of H_m v/s N is a parabolic curves as shown in Fig. 1

it is clear that P/D^5N^3 is a constant. Hence $P \propto N^3$. This means that the curve P v/s N is a cubic curve as shown in Fig. 1

shows that $\dfrac{Q}{D^3N}$ = constant. This means $Q \propto N$ for a given pump. Hence the curve Q v/s N is a straight line as shown in Fig. 1

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Unit-3

(Lecture 7)

Lecture contains

- **Performance characteristics**

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Unit-3 (Lecture 7)

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 characteristics of the pump. Fig. 1 shows the operating characteristic curves 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 (ρQgH) will be zero.

The efficiency curve will start from origin as at $Q = 0$, $\eta = 0$ $\left(\because \quad \eta = \dfrac{\text{Output}}{\text{Input}}\right)$

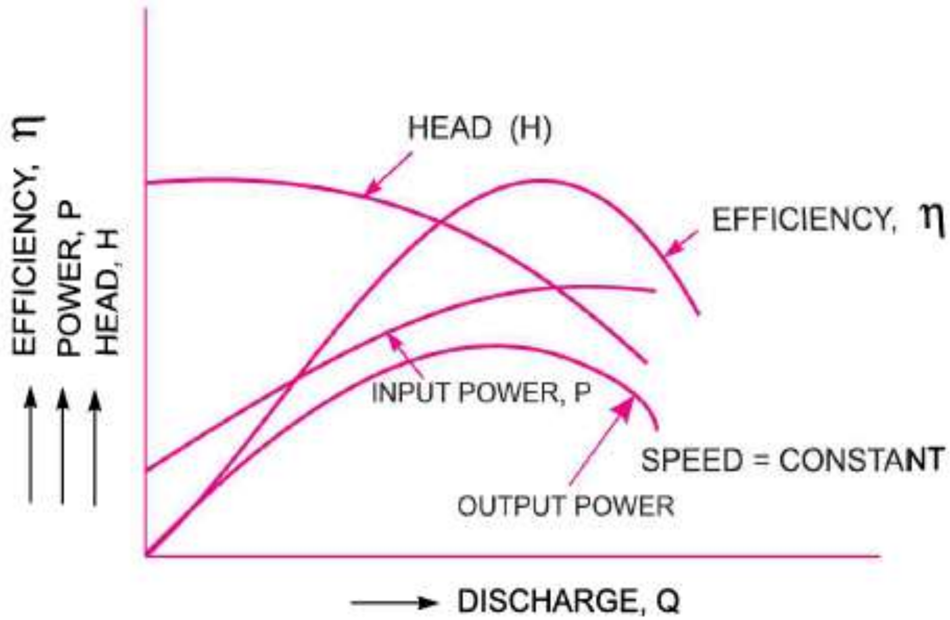

Fig. 1 *Operating characteristic curves of a pump.*

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. 1 (*a*) shows the head *versus* discharge curves for different speeds. The efficiency *versus* discharge curves for the different speeds are as shown in Fig. 2 (*b*). By combining these curves ($H \sim Q$ curves and $\eta \sim Q$ curves), constant efficiency curves are obtained as shown in Fig. 3 (*a*).

For plotting the constant efficiency curves (also known as iso-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 speeds, are transferred to the corresponding $H \sim Q$ curves. The points having the same efficiency are then joined by smooth curves. These smooth curves represents the iso-efficiency curves.

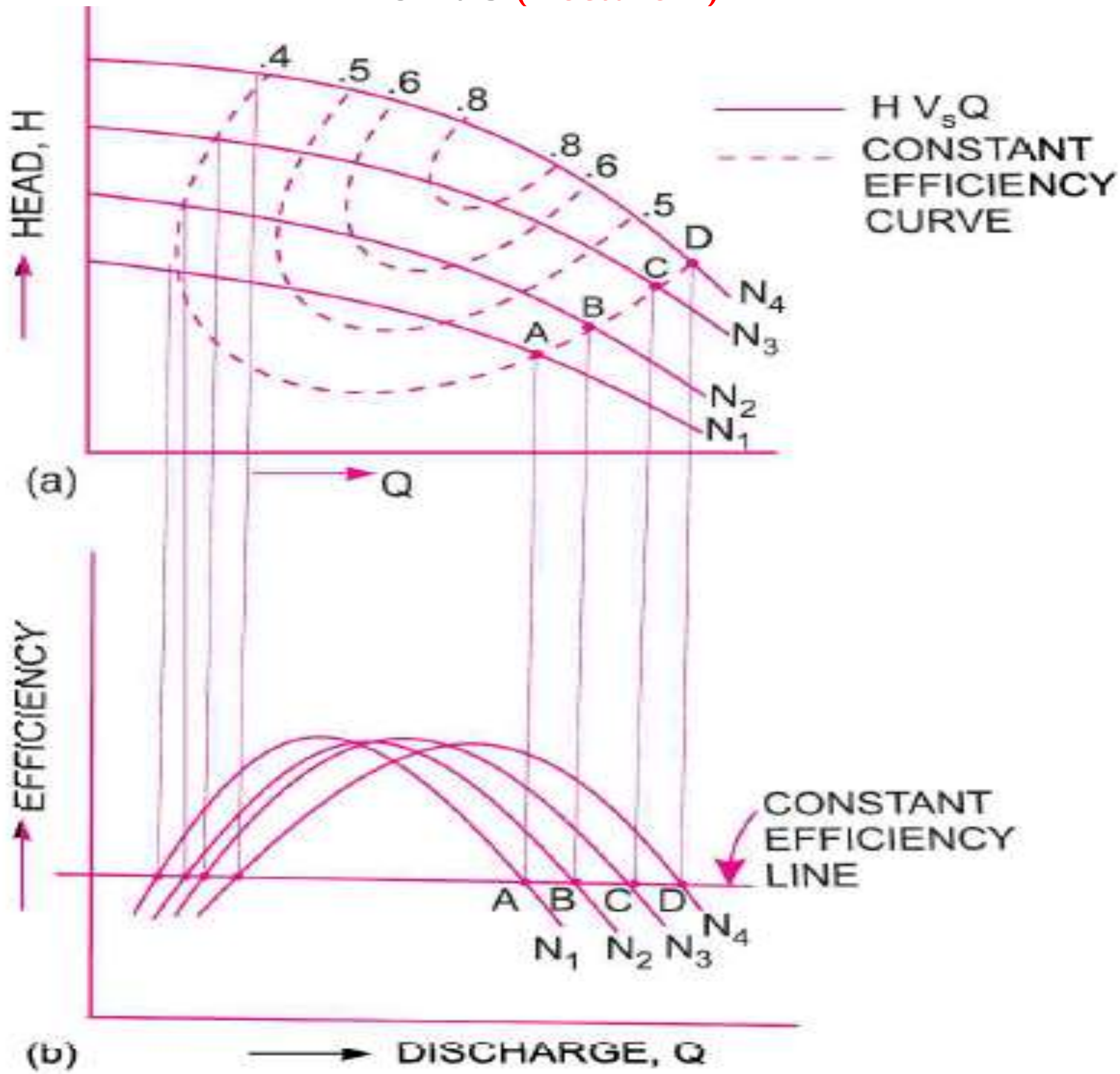

Fig.3 *Constant efficiency curves of a pump.*

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Unit-3

(Lecture 8)

Lecture contains

- **Cavitation**

Cavitation

Cavitation is defined as the phenomenon of formation of vapour bubbles of a flowing liquid in a region where the pressure of the liquid falls below its vapour pressure and the sudden collapsing of these vapour bubbles in a region of higher pressure. When the vapour bubbles collapse, a very high pressure is created. The metallic surfaces, above which these vapour bubbles collapse, is subjected to these high pressures, which cause pitting action on the surface. Thus cavities are formed on the metallic surface and also considerable noise and vibrations are produced.

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

I **Precaution Against Cavitation.** The following precautions should be taken against cavitation :

(*i*) The pressure of the flowing liquid in any part of the hydraulic system should not be allowed to fall below its vapour pressure. If the flowing liquid is water, then the absolute pressure head should not be below 2.5 m of water.

(*ii*) The special materials or coatings such as aluminium-bronze and stainless steel, which are cavitation resistant materials, should be used.

2 Effects of Cavitation. The following are the effects of cavitation :

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

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

(*iii*) The efficiency of a turbine decreases due to cavitation. Due to pitting action, the surface of the turbine blades becomes rough and the force exerted by water on the turbine blades decreases. Hence, the work done by water or output horse power becomes less and thus efficiency decreases.

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 pumps, where the pressure is considerably reduced. Hence if the pressure at the suction side of the pump drops below the vapour pressure of the liquid then the 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 portion of the suction side of the pump, the critical value of Thoma's cavitation factor (σ) is calculated.

Thoma's Cavitation Factor for Centrifugal Pumps. The mathematical expression for Thoma's cavitation factor for centrifugal pump is given by

$$\sigma = \frac{(H_b) - H_S - h_{LS}}{H} = \frac{(H_{atm} - H_V) - H_S - h_{LS}}{H}$$

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Unit-3

(Lecture 9)

Lecture contains

- Net positive Section Head
- Separation

NET POSITIVE SUCTION HEAD (NPSH)

The term NPSH (Net Positive Suction Head) 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 (in absolute units) plus the velocity head.

$\therefore$ NPSH = Absolute pressure head at inlet of the pump – vapour pressure head (absolute units) + velocity head

$$= \frac{p_1}{\rho g} - \frac{p_v}{\rho g} + \frac{v_s^2}{2g} \quad (\because \text{ Absolute pressure at inlet of pump} = p_1)$$

But from equation the absolute pressure head at inlet of the pump is given by as

$$\frac{p_1}{\rho g} = \frac{p_a}{\rho g} - \left(\frac{v_s^2}{2g} + h_s + h_{f_s}\right)$$

Substituting this value in equation we get

$$\text{NPSH} = \left[\frac{p_a}{\rho g} - \left(\frac{v_s^2}{2g} + h_s + h_{f_s}\right)\right] - \frac{p_v}{\rho g} + \frac{v_s^2}{2g}$$

$$= \frac{p_a}{\rho g} - \frac{p_v}{\rho g} - h_s - h_{f_s}$$

$$= H_a - H_v - h_s - h_{f_s}$$

$$\left(\because \frac{p_a}{\rho g} = H_a = \text{Atmospheric pressure head}, \frac{p_v}{\rho g} = H_v = \text{Vapour pressure head}\right)$$

$$= \left[\left(H_a - h_s - h_{f_s}\right) - H_v\right] \qquad \ldots(1 \qquad)$$

The right hand side of equation (1) 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 between the required NPSH and the available NPSH. The value of required NPSH is given by the pump manufacturer. This value can also be determined experimentally. For determining its value, the pump is tested and minimum value of h_s is obtained at which the pump gives maximum efficiency without any objectional noise (*i.e.*, cavitation free). The required NPSH varies with the pump design, speed of the pump and capacity of the pump.

When the pump is installed, the available NPSH is calculated from equation (1). In order to have cavitation free operation of centrifugal pump, the available NPSH should be greater than the required NPSH.