9.2 Pump

Pumps (and compressors) are mechanical systems used in our day to day life. For example, a pump is used to raise pressure of liquids as in municipal water works and drainage system, agricultural and irrigation system, tube well, oil pumps in lubrication system, crude oil in petroleum industry, etc. In fact, a pump is used to increase pressure of liquid and can handle any kind of liquid, even contaminated with solid particulates to some degree. In power plants pump is used to handle ash slurry made with water, which is highly erosive in nature. In chemical industries, it is used to handle corrosive fluids, and so on. But its performance is affected by the increase in the viscosity of the liquid and the presence of contamination in it.

The function of the pump is to raise pressure of fluid and impart desirable velocity to it so that fluid may be transported from one place to another; more is the velocity the more is the flow rate and hence less time required for transportation. However, we have learnt that the head loss occurring in transportation increases as square of the flow velocity. It therefore requires pressure of the fluid at the outlet of the pump to be more to overcome these losses. Thus, a compromise is required between the pressure and the velocity at the pump exit. It must, however, be emphasised that the purpose of a pump is mainly to raise the pressure and not the velocity of the fluid. The Following chart gives a broad spectrum of the pumps available in the market.
In dynamic pressure pump, work is transferred to fluid flowing over rotating blades of the impeller by changing direction of flow. This increases both pressure and velocity of fluid; mechanism of increase in pressure will be discussed later. Therefore fluid flows in the direction of increasing pressure and chances of separation and back flow are quite high, which results in decrease in efficiency and stable operation of the pump. For this reason their use is limited to those applications where moderate pressure rise is required. These are preferred as compared to reciprocating pump when required volume flow rate is high and/or pressure rise is not large. Dynamic pressure type systems include both turbine and rotary pump, which can handle compressible as well as incompressible fluids. Further, fluid may flow from outer to inner side or vice versa, as in radial flow machines or along the shaft axis of such machines. Methods like multi-staging have been developed to use these for high-pressure applications also. It is a technique in which discharge of the first stage is fed to the next stage and so on. This way a high-pressure ratio can be obtained. Here pressure ratio is defined as ratio of discharge pressure to suction pressure.

Amongst positive displacement type, reciprocating pump is commonly used. This system can deliver very high pressure since the cylinder piston surface does not allow back flow, whence the name. However, capacity, which is defined as mass flow rate of the device per minute, is low because of limited size of the cylinder. Plunger—barrel arrangement is also a reciprocating system like piston cylinder. It is used for diesel fuel pumps as quantity of fuel required is very low but discharge pressure is quite high. Plunger has length comparatively large with respect to its diameter; cylinder is called barrel. Other techniques include gear, lobe, vane, and screw pumps, which are rotary devices, in which back flow does not occur. The rotary type gear pump has wide application in industry mainly for raising pressure of lubricating oil. This pump can be used for extremely large pressure ratio. For this reason, these are used extensively in hydraulic controls.

If the pressure ratio is high then the pumps are placed in series with each other. Or if large quantity is required then many pumps operate with same pressure ratio and the discharge is fed to a common pipe. This arrangement is called parallel arrangement. Both the arrangements are often used in industrial application.
Some of the parameters that decide the selection of a pump are:
- Pressure and capacity of liquid being handled
- Speed of rotation and power requirement
- Properties such as viscosity, corrosiveness, grittiness, etc. of fluid
- Availability of space for positioning of pump
- Initial and maintenance cost

9.2.1 Centrifugal Pump

This is a radial machine in which fluid enters at its centre and comes out from the periphery of a rotating element called impeller. Side view of a centrifugal compressor having volute casing is shown in Fig. 9.2.1 (a) along with the nomenclature of its different parts. Impeller is made of a hub on which number of vanes are casted at equiangular positions either on its one or both faces making passage for the fluid to glide through it for which the curvature of vanes is so designed that fluid flows without separation. Pump is known as single or double suction pump accordingly. A double suction pump is used where large mass flow rate is required. A shaft connects hub to motor and passes through the pump casing, which provides bearing for the shaft, see Fig. 9.2.1(b). Seal is provided between casing and pump shaft to stop leakage of fluid.

Pressure across the passage between two consecutive vanes is assumed to be same but due to rotation of the vane pressure on forward face becomes more and less on trailing face. That is, on one side of a vane there will be positive pressure and on its other side (in another passage) it will be negative, as shown in figure. Therefore, fluid will have tendency to move around the edge in the space between casing and impeller causing losses. Fixing a ring around the vanes can prevent this. This ring is called shroud. Further, pressure at impeller exit is more than that at its entrance and that the exit and entrance are connected through space between casing and blade edge. Therefore, there are chances of back flow from exit to suction side of impeller through this casing space, see Fig. 9.1.3(a). Reducing clearance space can decrease it.

Fluid coming out of impeller has sufficiently high velocity and is converted to pressure using any or a combinations of (i) diffuser (ii) spiral (also known as volute) casing. A diffuser is a diverging passage having a suitable angle of divergence so that separation of flow does not occur. Pressure loss at its entry is reduced by designing the entrance angle of the diffuser so that liquid coming from impeller enters into this passage smoothly. For this reason,
velocity of the entering fluid should be in the direction of tangent to diffuser blade at its entrance. It should be noted that the number of diffuser vanes is kept less than that of moving vanes to make the flow uniform along peripheral direction. Therefore, a diffuser passage is fed by more than one moving vane and the streamlines at the entrance are curved. We know from fluid mechanics that such flows are associated with secondary flow, which cause loss of head. Therefore, initial portion of the passage is designed to straighten the streamlines. This process is completed within a short distance during which the diffuser passage is curved and fluid angles go on changing. The section where streamlines are almost straight is called throat and after this section the diffuser passage is a straight diverging section so that the area of flow goes on increasing. Normally a rectangular section is used with constant depth and varying width. Angle of divergence is around $8^\circ$ to $10^\circ$ to avoid separation of fluid from the wall of the passage.

A volute casing is a passage, which surrounds impeller or diffuser, if later is also used. Fluid enters into it tangentially from different impeller (or diffuser) passages and moves into volute casing. Volute casing is designed for constant velocity of flow. It changes direction of flow from successive passages so that fluid leaves the casing finally at the exit section of the pump. (It is also possible to take fluid from each diffuser channel directly to the place of need). These requirements are met by a spiral casing, whence the name spiral casing. Normally a volute casing is enough, since magnitude of velocity in pump is not high. The volute casing is also called the scroll casing.
For pumping liquid or water from its storage, called sump, piping arrangement is shown in Fig. 9.2.2. Water is sucked from a sump by the impeller action that causes decrease in pressure at the pump entrance and water rises in suction pipe due to higher atmospheric pressure at the free surface in sump. Unlike reciprocating pump, flow in centrifugal pump is continuous with constant velocity because of continuous pumping of fluid. Suction pipe diameter is normally kept large to decrease velocity in pipe to reduce frictional losses. Lower edge of the vane, which protrudes from the surface, is so shaped that fluid enters tangentially at all its points. By proper design of the edge, fluid can be taken on the vane without loss. Water then moves along the vane surface towards outer edge because of centrifugal force. Curvature of the passage is so chosen that fluid glides over it without separation of flow and formation of eddy. Software packages to study flow in such passage help in arriving at suitable curvature.

In Mechanics, we learn that centrifugal force is a reaction force; action for which comes from pressure. Pressure at exit therefore is more than that at inlet to vane. Centrifugal force besides increasing pressure increases velocity also at the impeller exit. In pumps there is decrease in relative velocity of fluid from inlet to exit. Thus all the three terms in Eq. 9.1.12 contribute in energy transfer from impeller to fluid, which is expressed in meter of the fluid. This head is termed Euler head, \( H_e \). A part of velocity head is further converted to pressure head by diffuser and/or volute and fluid velocity is reduced to a value required for transportation of fluid through delivery pipe.

### 9.2.1 Terms Used in Centrifugal Pump

Figure 9.2.2 shows a centrifugal pump placed at a height \( h_s \) from the free surface of liquid. Here the suction pipe is shown as vertical pipe but in practice it need not be so. However, the length of the suction pipe will be minimum if it is vertical and in that case
the frictional losses will also be minimum. Liquid is stored in a tank placed at a height
$h_d$ from the centre line of the pump. We use the following terms in centrifugal pump
analysis, see Fig. 9.2.2:

- **Suction lift head, $h_s$**: It is the vertical distance between free surface of liquid level
in the sump and pump centre line. It should not be confused with the length of suction pipe, which will be more if not vertical.

- **Discharge lift head, $h_d$**: It is the vertical distance between the top surface of liquid level in discharge tank and pump centre line. The sum of suction lift and discharge lift is called **total static** or **vertical lift**, $H$.

- **Suction head, $H_s$**: In addition to suction lift head, pump also has to generate head for overcoming friction loss $h_{fs}$ in suction pipe, entry loss $h_i$ at entrance to suction pipe and running fluid in suction pipe with velocity $V_s$. Thus

$$H_s = h_s + h_i + \frac{V_s^2}{2g} + h_{fs} = \frac{p_s}{w} + \frac{V_s^2}{2g}$$  \(9.2.1\)

where $p_s = w (h_s + h_i + h_{fs})$ is the vacuum below atmospheric pressure that would be
recorded by a pressure gauge placed at the entrance of the pump. This can be verified
by applying modified Bernoulli’s equation between two points one at the entrance section
of the suction pipe and another at its discharge end where the pressure gauge is fitted
and taking free surface in sump as datum for measuring height. If $h$ is depth of suction
end of the pipe from free surface then pressure at this end $= p_a + wh$. This gives:

$$\frac{p_a}{w} + h + \frac{V_s^2}{2g} = \frac{p_s}{w} + \frac{V_s^2}{2g} + (h_s + h) + (h_i + h_{fs})$$

or,

$$p_s = w (h_s + h_i + h_{fs})$$

- **Discharge head, $H_d$**: In addition to discharge lift head, pump has to generate head for overcoming friction loss $h_{fd}$ in discharge pipe and imparting discharge velocity $V_d$. Thus

$$H_d = h_d + \frac{V_d^2}{2g} + h_{fd} = \frac{p_d}{w} + \frac{V_d^2}{2g}$$  \(9.2.2\)

where $p_d$ is the pressure measured by a pressure gauge placed just at exit of the pump.
We may show it in the way it was shown for suction side.

- **Manometric head, $H_m$ and manometric efficiency**: Manometric head is defined as change in total energy head produced by the pump when fluid moves through it. If we neglect the change in potential head from inlet to exit of pump then

$$H_m = H_d + (-H_s) = \left( \frac{p_d}{w} - \frac{p_s}{w} \right) + \frac{V_d^2}{2g} - \frac{V_s^2}{2g}$$  \(9.2.3\)

Since the impeller transfers energy to the fluid, manometric head is nothing but Euler head except when there are energy losses associated in energy transfer process. These losses are discussed later in this chapter. The loss is expressed in terms of manometric efficiency defined as manometric head divided by Euler head.

Manometric efficiency, $\eta_{\text{man}} = \frac{H_m}{H_e}$  \(9.2.4\)
Manometric head may be seen as the head required to provide suction and lift head plus the losses in pipe from inlet of suction pipe to the discharge point of the delivery pipe plus velocity head at the delivery pipe end. That is,

\[ H_m = h_s + h_d + h_{fs} + h_{fd} + \frac{V_d^2}{2g} = H_s + H_d - \frac{V_s^2}{2g}, \]

using Eqs. 9.2.1 and 9.2.2.

**Net positive suction head, NPSH.** Concept of net positive suction head is that vapours should not be formed in the flow as these form bubbles, which are trapped inside liquid. These bubbles collapse when reach with flow in regions of high pressure where actual temperature is lower than the condensation temperature. This results in formation of cavity and flow of liquid from all the sides resulting in tremendous rise in pressure. This phenomenon results in mechanical loss as well as loss in efficiency of the machine. It is called cavitation and has to be avoided for better performance and long life of the machine. Formation of vapour depends on the surrounding temperature. Therefore the minimum pressure in suction line should not fall below vapour pressure of the liquid, \( p_v \). It is seen from Fig. 9.2.2 that minimum pressure in suction line will be at the impeller eye. Pressure at this point can be obtained by applying modified Bernoulli equation between the points on free surface in sump and near eye, neglecting velocity of liquid on the free surface and taking this surface as datum. We get

\[
\frac{p_a}{w} = \frac{p_1}{w} + h_s + h_{fs} + \frac{V_s^2}{2g} \\
\Rightarrow \frac{p_1}{w} = \frac{p_a}{w} - \left( h_s + h_{fs} + \frac{V_s^2}{2g} \right) > \frac{p_v}{w} \\
\text{or,} \quad \frac{p_a}{w} - \frac{p_v}{w} - \left( h_s + h_{fs} \right) > \frac{V_s^2}{2g}
\]

In literature, term \( \frac{p_a}{w} - \frac{p_v}{w} - \left( h_s + h_{fs} \right) \) is named as net positive suction head, NPSH. Thus from Eq. 9.2.6, \( \text{NPSH} > \frac{V_s^2}{2g} \), which is assumed 3m of water. Eq. 9.2.6 helps in deciding positioning of the centre line of pump above sump level. Normally a large diameter pipe with minimum length between sump and pump is used to reduce the losses so that vertical height of the pump from sump may be more. However, in practice, suction lift head is not kept more than 7m of water as vibration and cavitation has been reported for higher values.

### 9.2.2 Effect of Vane Shape and Operating Variables

Figure 9.2.3 shows three types of vanes used in practice. They differ in the way the vanes are curved with reference to direction of rotation, whence the name. For example,

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3 All liquids start changing their phase to vapour when they reach saturated liquid state. This depends on the pressure and temperature of the liquid. If pressure is decreased at a given temperature then at a particular pressure the liquid will start boiling. This pressure is called vapour pressure.
a backward curved vane is curved opposite to direction of rotation whereas forward curved in the direction of rotation. We can draw velocity triangles at exit in three cases for the same peripheral velocity $u_2$, which are shown in Fig. 9.2.4. We observe from Fig. 9.2.4 that (i) $\beta_2$, $C_{w2}$, and $C_2$ increase as we go through backward, radial to forward curved vanes and (ii) $\beta_2$ is $< 90^\circ$ for backward, $\beta_2 = 90^\circ$ for radial and $\beta_2 > 90^\circ$ for forward curved vane. Therefore energy transfer to fluid and absolute velocity at pump exit is least in backward curved and maximum in forward curved case. However, chances of back flow in backward curved pump are less. Moreover, when the impeller rotates at high rpm, as in rotating compressor, centrifugal force acting on vane tries to straighten it out. For these reasons, radial type is mostly preferred; it is easy to manufacture too. But

![Fig. 9.2.3 Different vane shapes](image)

backward curved pump is finding more and more application with improvement in manufacturing techniques because of their higher efficiency. In all the three type, entry is along axial direction having zero angular momentum in the plane of rotation. Euler head developed using Eq. 9.1.1 and Fig. 9.2.4 is

$$H_e = C_{w2} \frac{u_2}{g} = (u_2 - C_{r2} \cot \beta_2) \frac{u_2}{g} = (u_2 - Q \cot \beta_2/A_2) \frac{u_2}{g}$$  \hspace{1cm} (9.2.7)

![Fig. 9.2.4 Velocity diagram for different vanes](image)

Where $Q$ is the volume flow rate and $A_2$ is the area of flow at impeller exit, which gives, $C_{r2} = Q/A_2$. The rest of the symbols have the usual meaning. This equation shows
the effect of vane angle at its exit on pressure head development. For a given rpm of
impeller, variables in Eq. 9.2.7 are \( H_e, Q \) and \( \beta_2 \). Therefore, we get a parametric family for
different \( \beta_2 \) between \( H_e \) and \( Q \); this is shown in Fig. 9.2.5(a). Curve for radial blade is
horizontal which means there is no effect of discharge on the head developed.

![Graph showing ideal operating curves and actual operating curve for backward curved vane pump](image)

**Fig. 9.2.5** \( H \) vs \( Q \) curve and losses

We assumed that flow leaves vanes along tangent at its exit. But in practice, this
does not happen because (i) the fluid has inertia and opposes turning of fluid in the
direction of rotation and (ii) that the fluid rotates in direction opposite to the direction
of rotation in the impeller passage to have zero vorticity as it was zero at entrance to
the impeller, by vorticity theorem. Though no change in vorticity is true only for
frictionless flow but for the small distance travelled by fluid, an assumption of no
change in vorticity is justified. Also velocity distribution along impeller periphery is
non-uniform even at the design operating conditions. That is, \( C_2 \) varies along the
impeller periphery, both in magnitude and direction resulting in decrease in whirl
velocity at exit. This is shown in Fig. 9.2.6. It results in less energy transfer to fluid and
hence actual head produced is less than Euler head. The reduction in whirl velocity is
expressed by the term \( \text{slip} \), defined as ratio of actual to theoretical whirl velocity. Thus
ideal head produced is \( H_i = \sigma H_e \), where \( \sigma \) is slip factor. Stodola and others, as given
below, obtained following relations for determination of slip factor based on vorticity
theorem.

**Stodola relation:** \[
\sigma = 1 - \frac{\pi \sin \beta_2}{Z \left[ 1 - \left( \frac{C_f}{u_2} \right) \cot \beta_2 \right]}
\]

**Busemann relation:** \[
\sigma = A - B \left( \frac{C_f}{u_2} \right) \cot \beta_2,
\]
where \( A \) and \( B \) are function of \( r_2 / r_1, \beta_2 \) and \( Z \)

\(^6\) A pump is designed for a given pressure head, mass flow rate and rpm. Under these conditions, it has
maximum efficiency. These conditions are known as design operating conditions.
Stanitz relation:

$$\sigma = 1 - \frac{0.63 \sin \beta_2}{Z \left[ 1 - \left( C_f/u_2 \right) \cot \beta_2 \right]}$$

Fig. 9.2.6 Effect of inertia in backward curved pump

Slip factor improves if the number of vanes is increased. However, this increases the blockage ratio, that is, area of flow reduces and hence the mass flow rate too. Therefore a compromise is to be made.

The result of slip and non-uniform flow is to reduce the pressure head produced at impeller exit and the effect is shown by vertical distance $A$ in Fig. 9.2.5(b). This loss decreases with increase in $Q$. Losses occurring in diffuser further decreases actual head $H$; vertical distance $B$ is measure of this loss and we observe from Fig. 9.2.5(b) that this loss first decrease upto some $Q$ and then starts increasing with increasing $Q$. The loss consists of friction loss in diffuser passage and the aerodynamic loss at the entrance to it because of deviation between actual flow angle and diffuser blade angle. The later loss changes with $Q$ and hence explains the trend of the diffuser loss. So the losses in diffuser will be minimum only at the design point and therefore the actual head $H$ varies with $Q$ as shown in Fig. 9.2.5(b). Ratio of $H$ to $H_i$ is known as hydraulic efficiency,

$$\eta_h = H / H_i \quad \text{or} \quad H = \eta_h H_i = \eta_h \sigma H_e \quad (9.2.8)$$

Besides the losses mentioned earlier, other losses also occur. Fig. 9.2.7 gives a picture of flow of energy and the loss occurring in different sections. Power from electric motor is known as shaft power. But due to mechanical losses in bearing, power that reaches pump rotor is less. Thus

$$\text{rotor power} = w \left( Q + q \right) H_i$$

where $(Q + q)$ is the discharge from the sump. This loss is expressed by mechanical efficiency, defined as

$$\eta_{\text{mech}} = \text{rotor or impeller power/ shaft power} \quad (9.2.9)$$

Mechanical efficiency usually is 0.95 to 0.98.
Leakage of fluid occurs from the seals and through the clearance space between the casing and the shroud that surrounds the impeller. Therefore volume of fluid handled by impeller is more than it is discharged. If \( Q \) is the volume discharged and \( q \) the volume leakage then volumetric efficiency is defined as ratio of \( Q \) and \( (Q + q) \).

\[
\eta_{vol} = \frac{Q}{(Q + q)}
\]  \hspace{1cm} (9.2.10)

A pump may be considered as a black box, which is fed with shaft power and water power \( wQH \) leaves it. We may express performance by introducing the term overall efficiency,

\[
\eta_o = \frac{wQH}{\text{shaft power}} = \frac{w(Q + q)H_i}{w(Q + q)H_i} \times \frac{Q}{Q + q}
\]

We multiply and divide this equation by rotor power while writing expression for the rotor power in the denominator. This gives overall efficiency

\[
\eta_o = \frac{wQH}{\text{shaft power}} \times \frac{\text{Rotor power}}{w(Q + q)H_i} = \frac{H}{H_i} \times \frac{Q}{Q + q} \times \frac{\text{Rotor power}}{\text{shaft power}} = \eta_h \eta_{vol} \eta_{mech} \hspace{1cm} (9.2.11)
\]

We may also express manometric head in terms of \( Q \) and \( N \) to see their effect on it, since these are operating variables which are changed in actual set up. Let suffix 1 and 2 represent inlet and exit of impeller and say a fraction \( k \) of the kinetic energy head at exit of impeller \( V_2^2/2g \) is recovered as pressure head in diffuser and/or volute casing. Then manometric head is,

\[
H_m = \frac{(p_2 - p_1)}{w} + k V_2^2/2g
\]

Assuming no loss in impeller, Bernoulli’s equation gives:

\[
p_1/w + V_1^2/2g + C_{w2}u_2/g = p_2/w + V_2^2/2g
\]

or,

\[
(p_2 - p_1)/w = (V_1^2 - V_2^2)/2g + C_{w2}u_2/g
\]

But from outlet velocity triangle,

\[
C_{w2} = u_2 - C_{f2} \cot \beta_2 \quad \text{and} \quad C_2^2 = C_{w2}^2 + C_{f2}^2
\]

\[
C_2^2 = C_{w2}^2 + C_{f2}^2
\]

\[
(p_2 - p_1)/w = \left(C_{f1}^2 + u_2^2 - C_{f2}^2 \csc^2 \beta_2\right)/2g
\]

Here it is assumed that the entry to pump is radial, which means \( C_1 = C_{f1} \), and has been used in above relation. We further assume that there is no change in velocity of flow, that is \( C_{f1} = C_{f2} \). Substituting for \( (p_2 - p_1)/w, \ C_2^2 \) in the equation for manometric head and simplifying, we get
\[ H_m = (AN^2 + BNQ + CQ^2)/2g, \]  
\hspace{2cm} (9.2.12) 

where  
\[ A = (1 + k) \left( \pi D_2/60 \right)^2, \quad B = -2k \cot \beta_2/\left( \pi D_2 \beta \right), \]

and  
\[ C = (1 + k \cot^2 \beta_2 - \cosec^2 \beta_2) / \left( \pi D_2 b \right)^2 \]

Here \( b \) is depth of impeller passage at its exit.

### 9.2.3 Cavitation, Minimum Speed of Operation and Priming in Pump

Cavitation and priming are two important phenomena associated with pump. We have explained earlier as to why cavitation occurs. It is specific characteristic of hydraulic machines caused by vaporisation of liquid. We mentioned earlier that the pressure is minimum at impeller eye. Let the pressure at the section is \( p_v \), which is saturation pressure of the liquid at the temperature at this section. Any further decrease in pressure would result in formation of vapour. Under this critical condition, the flow velocity \( V_s \) at eye can be obtained by applying modified Bernoulli equation. This gives

\[ \frac{V_s^2}{2g} = \frac{p_a}{w} - \frac{p_v}{w} - h_s - h_{fp} = NPSH \]

However, \( V_s^2/2g \) should be less than this values so that cavitation does not occur. Thoma gave cavitation number \( \sigma \), defined as

\[ \sigma = \frac{V_s^2}{2gH} \leq NPSH / H, \]  
\hspace{2cm} (9.2.13) 

where \( H \) is head produced by pump; \( \sigma \) should therefore be less than \( NPSH/H \) for cavitation free flow. Critical value of Thoma’s cavitation factor \( \sigma_c \), depends on specific speed, efficiency of the pump and number of vanes and is given by empirical relation

\[ \sigma_c = 1.042 \times 10^{-3} \left( N_s \right)^{4/5}, \text{ where } N_s \text{ is pump specific speed} \]  
\hspace{2cm} (9.2.14) 

Thus cavitation factor of given installation should always be greater than \( \sigma_c \). That is,

\[ \sigma_c < \sigma < NPSH / H \]  
\hspace{2cm} (9.2.15) 

Manufacturer of pump gives the required \( NPSH \), which depends on the design of the pump. When in use, available \( NPSH \) is obtained for those conditions for which the pump operates at maximum efficiency without any objectionable noise. Therefore available \( NPSH \) must be more than the \( NPSH \) prescribed by manufacturer for cavitation free operation. Thus Thoma’s factor decides the positioning of the pump.

On the other hand, pump fails to lift liquid at the time of its starting. This happens because suction pipe is filled with air when liquid flows back to sump due to failure of one-way valve fitted at the entrance of suction pipe. Since air is lighter than liquid, rotation of impeller does not create enough suction head to raise liquid from sump. It is overcome by filling the suction pipe and the space around the impeller with liquid by pouring it through a funnel fixed in delivery pipe or by sucking air from the pump using an extraction pump that also pulls liquid from sump and fills the whole space around impeller. This process is called priming.

A pump can operate only when its rpm is sufficient to move liquid over the vane. This will happen when head developed by pump at the time of start is more than the manometric head. But at this condition, since flow is to start, the only force to develop head is centrifugal head \( = (u_2^2 - u_1^2)/2g \). Therefore,
\[
\frac{(u_2^2 - u_1^2)}{2g} \geq H_m \quad \text{or} \quad \frac{(u_2^2 - u_1^2)}{2g} \geq \eta_m \frac{H_e}{C_{w2}} u_2 / g
\]

\[\Rightarrow \quad N^2 \geq 2g \left( \frac{60}{\pi} \right)^2 \frac{H_e \eta_m}{D_2^2 - D_1^2} \tag{9.2.16}\]

Where \(D_1\) and \(D_2\) are diameter of mid point of eye and impeller outlet respectively. Equation 9.2.16 gives the minimum \text{rpm} for a given pump. Normally an electric motor is used to run the pump and therefore value of \(N\) is determined by motor \text{rpm}. In that case \(D_2\) is checked from Eq. 9.2.16; actual pump diameter should be more than this diameter so that pump may be used at that motor \text{rpm}.

### 9.2.4 Performance Characteristics of Centrifugal Pump

Three types of performance characteristic curves, namely, main, operating and iso-efficiency characteristic curves are used for a centrifugal pump. Experiments are performed on the given pump running at a given (fixed) \text{rpm} for different values of the discharge through it while measuring power consumption and manometric head.

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![Diagram](image_url)

**Fig. 9.2.8** Pump's performance curves
produced for each setting. From these observations, efficiency is obtained for a given value of Q and the results are plotted on a graph with Q along abscissa and H, P and efficiency along ordinate. This experiment is repeated at a few other rpm and the results plotted on the same graph. This gives the parametric family in the parameter rpm. Curves so obtained are main characteristic, shown in Fig. 9.2.8(a). This chart helps in selection of the pump for given operating conditions.

Like any other system, pump will also operate at its maximum efficiency when run at its design rpm. Therefore, manufacturers give operation graph, which is nothing but the graph from main characteristic at the design rpm. The normal operation condition of pump corresponds to the point at which the efficiency is maximum. Since the efficiency curve is quite sharp near the operating point, it is desirable to run the pump at the design flow rate as far as possible.

Yet another graph of importance is the iso-efficiency graph, also known as Muschel graph because discharge requirement during operation may vary and we would like to run the pump almost at the same efficiency. For this, Muschel chart is very useful since it gives the desired operating speed. It is a graph between Head and flow rate with the efficiency as the parameter of the family of the curves. These curves are obtained from main characteristic at a given efficiency by drawing a horizontal line CD, see Fig. 9.2.8(a). This line meets the efficiency curve for a particular rpm \(N_1\) (say) at C and D. Now draw vertical lines through points, C and D to meet the \(H = \text{constant}\) line corresponding to \(N = N_1\) at \(C'\) and \(D'\) and read values of \(H\) and \(Q\) corresponding to points \(C'\) and \(D'\). This gives two points \(C'\) and \(D'\) on Muschel graph. The horizontal line will also meet the constant efficiency curves at different rpm from which we get other points in the same manner and plot on the Muschel graph. This gives an iso-efficiency curve. The procedure is then repeated for different values of efficiency. We get these curves from the main characteristics only. We can also join the points for which rpm is the same. The results are shown in Fig. 9.2.8(c).

It is observed that as efficiency increases, width of the curve decreases and ultimately at some efficiency it will shrink to a point that corresponds to the point of maximum efficiency for that pump. Also for each RPM there is a point of maximum efficiency and hence there will be only one point for it, which is shown as points A, B, etc. in the graph. Thus, the line joining these points is the line at which the pump is to be operated for that efficiency.

All the three types of characteristics is shown in Fig. 9.2.8.