Module 5: CENTRIFUGAL PUMPS

5.1 Introduction:

Centrifugal pump is a power absorbing turbomachine used to raise liquids from a lower level to a higher level by creating the required pressure with help of centrifugal action. Thus it can be defined as a machine which converts mechanical energy into pressure energy (hydraulic energy) by means of centrifugal action on the liquids.

When a certain amount of liquid is rotated by an external energy (mechanical energy) inside the pump casing, a forced vortex is set up, which raises the pressure head of the rotating liquid purely by centrifugal action.

5.2 Working Principal:

Figure 5.1(a) shows the working principal of a centrifugal pump. The liquid to be pumped enters the centre of the impeller which is known as eye of the pump and discharge into the space around the casing and hence filling the space. Due to the rotation of the impeller inside the pump casing a forced vortex is set up which imparts pressure head to the liquid purely by centrifugal action.

The pressure head developed by centrifugal action is entirely by the velocity imparted to the liquid by rotating impeller and not due to any displacement or impact. Thus the mechanical action of the pump is to impart velocity to liquid so that its speed is enough to produce necessary centrifugal head for discharging.

5.3 Classification of Centrifugal Pumps:

Centrifugal pumps are classified based on following aspects:

5.3.1 Based on Type of Casings:

Question No 5.1: With neat sketch explain the various types of casings used in centrifugal pumps. ***(VTU, Jun/Jul-08, Jun/Jul-09)***

Answer: Based on the type of casing centrifugal pumps may be classified as volute pump and diffusion pump.

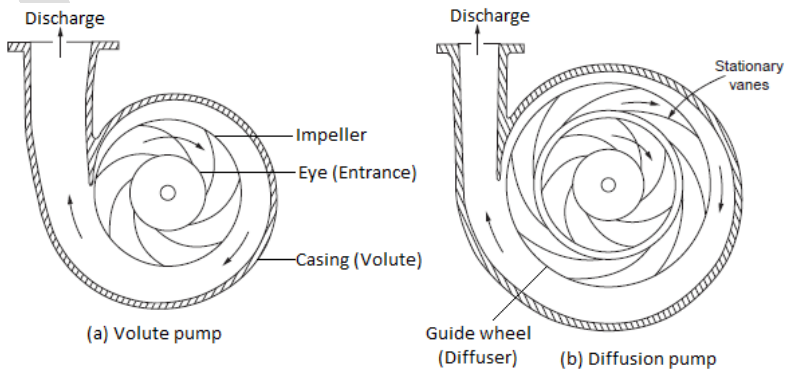

Fig. 5.1 Classification based on pump casings

1. Volute Pump: A volute pump is used to discharge water at a high velocity. The pump consists of a volute casing of spiral form with gradually increasing cross sectional area towards the discharge end. Mean velocity of flowing fluid remains constant as cross sectional area at any point is proportional to amount of water flowing through that section. Loss of kinetic head due to eddy formation is avoided due to the spiral shape. A volute pump is shown in figure 5.1(a).

The volute casing provided serves the following purpose:

a) To collect water from the periphery of the impeller and to transmit it to the delivery pipe at a constant velocity.

b) To eliminate the head loss due to change in velocity, as the velocity of water leaving the impeller equals the velocity of flow in the volute.

c) To increase the efficiency of the pump by eliminating loss of head.

2. Diffusion Pump or Turbine Pump: The diffusion pump consists of an impeller surrounded by guide wheel fitted with stationary vanes or diffusers as shown in figure 5.1(b). The cross sectional area of these vanes increases gradually so that pressure of the fluid increases by decreasing velocity as fluid passes over the vanes. The guide vanes are so designed that the angle made at the entrance should perfectly match with the direction of absolute velocity at the outlet of the impeller.

Diffusion pumps may be either horizontal or vertical shaft type. The vertical shaft type pump is suitable for deep wells where space consideration is more important. These pumps are used in narrow wells and mines.

5.3.2 Based on Fluid Entrance: Based on fluid entrance centrifugal pumps are classified as single entry and double entry pumps as shown in figure 5.2.

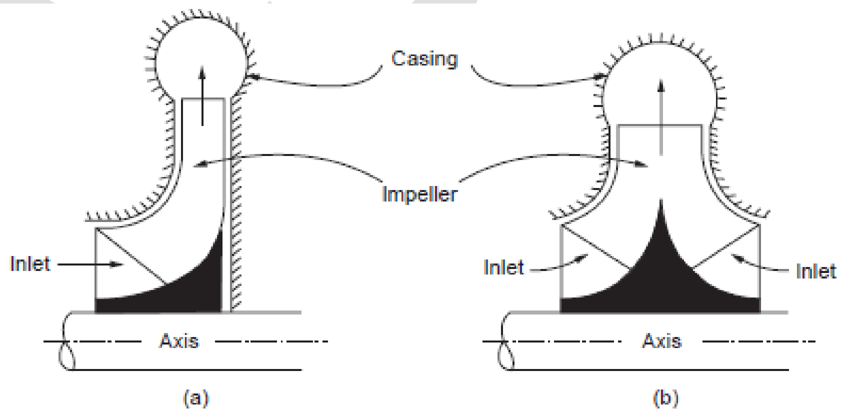

Fig. 5.2 Classification based on fluid entrance

1. Single Entry Pump: In single entry pumps water is admitted from one side of the impeller only.

2. Double Entry Pump: In double entry pumps water enters from both sides of the impeller. This arrangement neutralizes the axial thrust, producing high heads. It is used for pumping large quantities of liquid.

5.3.3 Based on Type of Impeller:

Question No 5.2: With a neat sketch explain different types of centrifugal pump impellers and list their merits and demerits.

Answer: Based on the type of impeller centrifugal pumps are classified as closed impeller pump, semi-closed (semi-open) impeller pump and open impeller pump as shown in figure 5.3.

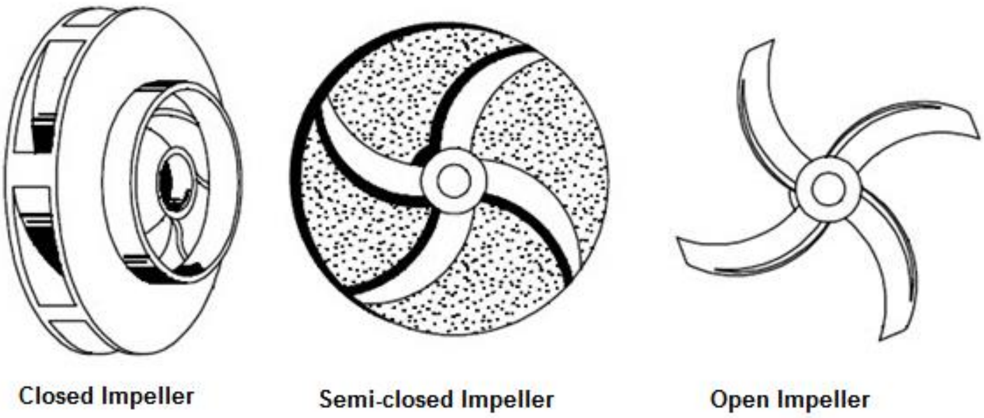

Fig. 5.3 Classification based on type of impeller

1. Closed Impeller Pump: If the impeller vanes are covered with shrouds on both sides of the impeller, then it is a closed impeller pump. This pump can handle non-viscous liquids like water (hot and cold), hot oil, acids etc.

2. Semi-closed (Semi-open) Impeller Pump: If the vanes are covered with shroud on one side of the impeller only, then it is a semi-closed impeller pump. The height of the vanes are increased and number of vanes are reduced so as avoid clogging of impeller. This pump can be used to discharge sewage water, pulp etc.

3. Open Impeller Pump: If the vanes are not covered with shroud, then it is open impeller pump. The impeller is made of forged steel and is designed to work under rough operating conditions. This pump can be used discharge mixtures of water, sand, clay etc.

The closed and semi-closed type impellers cover most industrial applications due to their superior performance and efficiency. The open type with no shroud is employed due to its simplicity, low cost and negligible maintenance. But its efficiency is very poor since it has poor flow confinement and the vanes are not formed to provide the best efficiency.

5.4 Heads of Centrifugal Pump:

Question No 5.3: Define the following with respect to centrifugal pumps: (i) Static head (ii) Manometric head (iii)Total head with the help of a schematic diagram. (VTU, Dec-09/Jan-10, May/Jun-10, Dec-12)

Answer: The different types of heads used in centrifugal pump are as shown in figure 5.4, and are defined as follows.

1. Static Head (*h*): The main purpose of the pump is to lift water from the sump and deliver it at the overhead tank, the vertical height between the liquid level in the sump and the liquid level in the delivery tank is called the static head. It consists of two parts, the suction head (h_s) and the delivery head (h_d). i.e., $h = h_s + h_d$

Where suction head is the vertical height between the centre line of the pump to the liquid level in the sump and delivery head is the vertical height the centre line of the pump to the liquid level in the delivery tank.

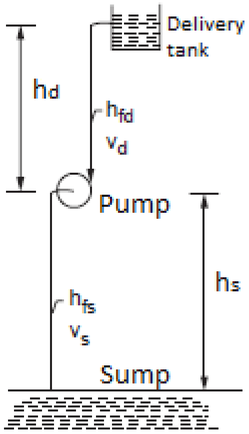

Fig. 5.4 Schematic diagram with suction and delivery pipes

2. Manometric Head (H_m): It is the effective head that must be produced by the pump to satisfy the external requirements. It includes all the losses like frictional losses, leakage losses etc.

$$H_m = h_s + h_d + h_{fs} + h_{fd} + \frac{V_s^2}{2g}$$

$$H_m = h + h_{fs} + h_{fd} + \frac{V_s^2}{2g}$$

Where h_{fs} and h_{fd} are the head loss due to friction in suction and delivery pipes respectively and V_s is the velocity of the fluid in the suction pipe.

The head imparted to the liquid by the impeller is equal to sum of manometric head and loss of head (H_L) in the impeller and casing.

$$\frac{U_2 V_{u2}}{g} = H_m + H_L$$

$$H_m = \frac{U_2 V_{u2}}{g} - H_L$$

The manometric head is measured by installing pressure gauges in the delivery and suction lines of the pump as close as possible to the pump inlet and the exit.

$$H_m = \frac{P_d - P_s}{\rho g}$$

Where P_s and P_d are the pressure denoted by the gauges on the suction and delivery sides respectively.

The pressure at the suction of the pump is,

$$P_s = P_a - \rho g h_s - \rho g h_{fs} - \frac{\rho V_s^2}{2}$$

Similarly, pressure at the delivery of the pump is,

$$P_d = P_a + \rho g h_d + \rho g h_{fd}$$

Where P_a is the atmospheric pressure.

3. Total Head (H_e): The total head is the net head produced by the pump to overcome the static head, the total loss in the system due to friction, turbulence, foot-valves and bends, etc., and to provide the kinetic energy of water at the delivery tank.

$$H_e = h + h_{fs} + h_{fd} + \frac{V_d^2}{2g}$$

Where h_{fs} and h_{fd} are the head loss due to friction in suction and delivery pipes respectively and V_d is the velocity of the fluid in the delivery tank.

The manometric head and the total head are the same if the suction and delivery pipes have the same diameter (i.e. V_s=V_d)

4. Net Positive Suction Head (*NPSH*):

Question No 5.4: What do you mean by NPSH? Is it desirable to have a lower or higher value of NPSH? Justify your answer with the help of relevant equations.

Answer: It is the head required at the pump inlet to keep the local pressure everywhere inside the pump above the vapour pressure.

Net positive suction head (*NPSH*) is defined as the difference between the pump's suction stagnation pressure head and vapour pressure head.

$$NPSH = \left[\frac{P_s}{\rho g} + \frac{V_s^2}{2g}\right] - \frac{P_v}{\rho g}$$

Where V_s is the velocity of the water in suction side, P_s and P_v are the static pressure at the suction and the vapour pressure respectively.

In the suction the fluid is at atmospheric temperature so the vapour pressure remains constant, to increase the static head one has to increase net positive suction head. Therefore NPSH should be higher value. In order to have cavitation free operation of centrifugal pump available NPSH should be greater than the minimum NPSH.

$$h = \frac{P_s}{\rho g} = NPSH + \frac{P_v}{\rho g} - \frac{V_s^2}{2g}$$

5.5 Efficiencies of Centrifugal Pump:

Question No 5.5: Explain the following with mathematical expression: (i) Manometric efficiency (ii) Mechanical efficiency (iii) Volumetric efficiency (iv) Overall efficiency. ***(VTU, Jun-12)***

Answer: The different types of efficiencies expressed in centrifugal pump are as follows:

1. Manometric Efficiency: It is the ratio of the manometric head to the ideal head imparted by the impeller to the fluid.

$$\eta_{ma} = \frac{H_m}{\left(\frac{U_2 V_{u2}}{g}\right)} = \frac{\left(\frac{U_2 V_{u2}}{g}\right) - Hydraulic\ losses}{\left(\frac{U_2 V_{u2}}{g}\right)} = \frac{H_m}{H_m + Hydraulic\ losses}$$

2. Mechanical Efficiency: It is the ratio of the energy transferred by the impeller to the fluid to the mechanical energy delivered to the pump at the shaft.

$$\eta_m = \frac{w_{impeller}}{w_{shaft}} = \frac{U_2 V_{u2}}{U_2 V_{u2} + Mechanical\ losses}$$

3. Volumetric Efficiency: It is the ratio of the amount of fluid delivered by the delivery pipe to the amount of fluid entering the impeller though suction pipe.

$$\eta_v = \frac{Actual\ discharge}{Theoretical\ discharge} = \frac{Q_d}{Q_s} = \frac{Q_d}{Q_d + Q_L}$$

Where Q_L is the amount of fluid leakage loss.

4. Overall Efficiency: It is the ratio of actual hydraulic energy output by the pump to the mechanical energy input to the pump at the shaft. The overall pump efficiency is the product of hydraulic efficiency, volumetric efficiency and mechanical efficiency.

$$\eta_o = \eta_H \eta_v \eta_m$$

Where hydraulic efficiency is the ratio of the useful pump output head (static head) to the ideal head imparted by the impeller to the fluid

$$\eta_H = \frac{h}{\left(\frac{U_2 V_{u2}}{g}\right)}$$

5.6 Cavitation and Priming:

Question No 5.6: What is cavitation in centrifugal pump? What are the causes of cavitation? Explain the steps to be taken to avoid cavitation. ***(VTU, Jun/Jul-11, Dec-11)***

Answer: If the pressure at any point in a suction side of centrifugal pump falls below the vapour pressure, then the water starts boiling forming saturated vapour bubbles. Thus, formed bubbles moves at very high velocity to the more pressure side of the impeller blade and strikes the surface of the blade and collapse there. In this way, as the pressure further decreases, more bubbles will be formed and collapses on the surface of the blades, physically enables to erosion and pitting, forming a cavities on blades. This process takes place many thousand times in a second and damages the blade of a centrifugal pump. This phenomenon is known as cavitation.

Causes of cavitation: The causes of cavitation are as follows,

1. The metallic surfaces damaged and cavities are formed on the impeller surface.
2. Considerable noise and vibration are produced due to the sudden collapse of vapour bubble.
3. The efficiency of the machine reduces.

Steps to avoid cavitation: The following steps should be taken to avoid cavitation,

1. The suction losses should be minimised through the use of large diameter suction tubes with fewer bends than in the delivery pipe.
2. The pressure of the fluid flow in any part of the system should not fall below the vapour pressure.
3. The impeller should be made of better cavitation resistant materials such as aluminium, bronze and stainless steel.

Question No 5.7: Explain the term priming related to centrifugal pump. Why priming is necessary for a centrifugal pump. (VTU, Dec-07/Jan-08) Or,

What is priming? How priming will be done in centrifugal pumps? (VTU, Dec-12)

Answer: Priming is the process of removing the air present in the suction pipe and impeller casing. To remove the air the suction pipe, casing of the pump and portion of the delivery pipe are completely filled with water before starting the pump. If the pump is running with air it develops the head in meters of air. If the pump is running with water, the head produced is in terms of meters of water. But as the density of air is very low, the head generated in terms of equivalent of water is negligible. It is therefore the pressure rise by the pump, when the air is present, may not be sufficient to suck water from the sump. To eliminate this difficulty, the pump is to be primed with water properly before start.

5.7 Work Done by the Centrifugal Pump:

Question No 5.8: Derive a theoretical head capacity (H-Q) relationship for centrifugal pumps and compressors and explain the influence of outlet blade angle. (VTU, Jul/Aug-05, Dec-11)

Answer: The velocity diagram for centrifugal pumps and compressor with V_{u1}= 0 is as shown in figure 5.5.

Energy transfer of a centrifugal compressor and pump is given as:

$$e = gH = U_2V_{u2} - U_1V_{u1}$$

(Because, in centrifugal pump and compressor the working fluid is usually water or oil)

Or, $$gH = U_2V_{u2}$$ (Because, V_{u1}= 0)

From outlet velocity triangle, $V_{u2} = U_2 - x_2$

But, $cot\beta_2 = \frac{x_2}{V_{m2}} \Longrightarrow x_2 = V_{m2}cot\beta_2$

$$V_{u2} = U_2 - V_{m2}cot\beta_2$$

Therefore, $$gH = U_2(U_2 - V_{m2}cot\beta_2)$$

Or, $$H = \frac{U_2^2}{g} - \frac{U_2 V_{m2}}{g} cot\beta_2$$

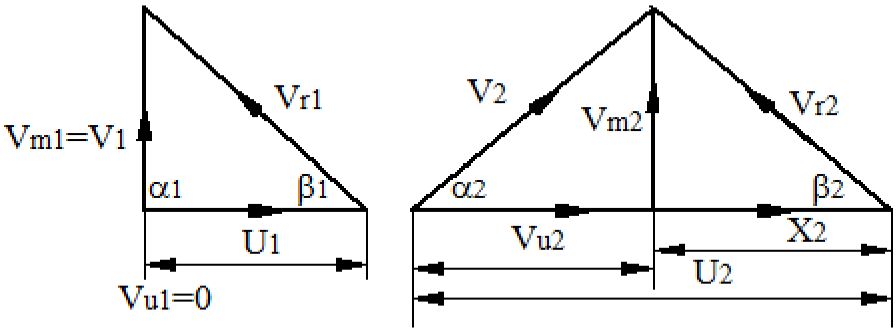

Fig. 5.5 Velocity triangles for centrifugal pump

Discharge at outer radius of centrifugal machine = Area of flow × Flow velocity

$$Q = \pi D_2 B_2 \times V_{m2}$$

$$V_{m2} = \frac{Q}{\pi D_2 B_2}$$

Then,

$$\boldsymbol{H = \frac{U_2^2}{g} - \left(\frac{U_2}{g}\right)\left(\frac{Q}{\pi D_2 B_2}\right) cot\beta_2}$$

By using above equation, H-Q characteristic curve of a given impeller exit blade angle β_2 for different values of discharge is drawn in figure 5.6.

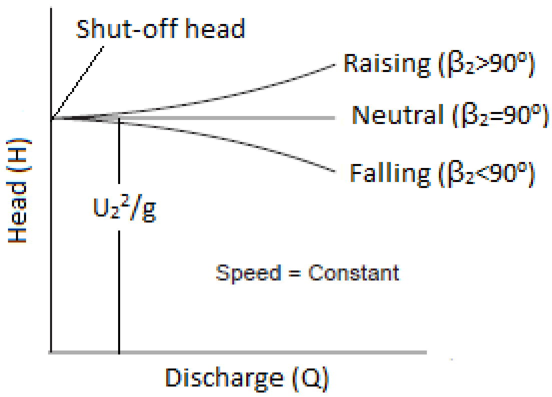

Fig. 5.6 H-Q characteristic curve

Question No 5.9: Draw the inlet and outlet velocity triangles for a radial flow power absorbing turbomachines with (i) Backward curved vane (ii) Radial vane (iii) Forward curved vane. Assume inlet whirl velocity to be zero. Draw and explain the head-capacity relations for the above 3 types of vanes. ***(VTU, Dec-08/Jan-09)***

Answer: There are three types of vane shapes in centrifugal machines namely, (i) Backward curved vane (ii) Radial vane (iii) Forward curved vane.

The vane is said to be backward curved if the angle between the rotor blade-tip and the tangent to the rotor at the exit is acute (β_2<90°). If it is a right angle (β_2=90°) the blade said to be radial and if it is greater than 90^o, the blade is said to be forward curved. Here the blade angles measured with respect to direction of rotor (clockwise direction). The velocity triangles at the outlet of centrifugal machines are shown figure 5.7.

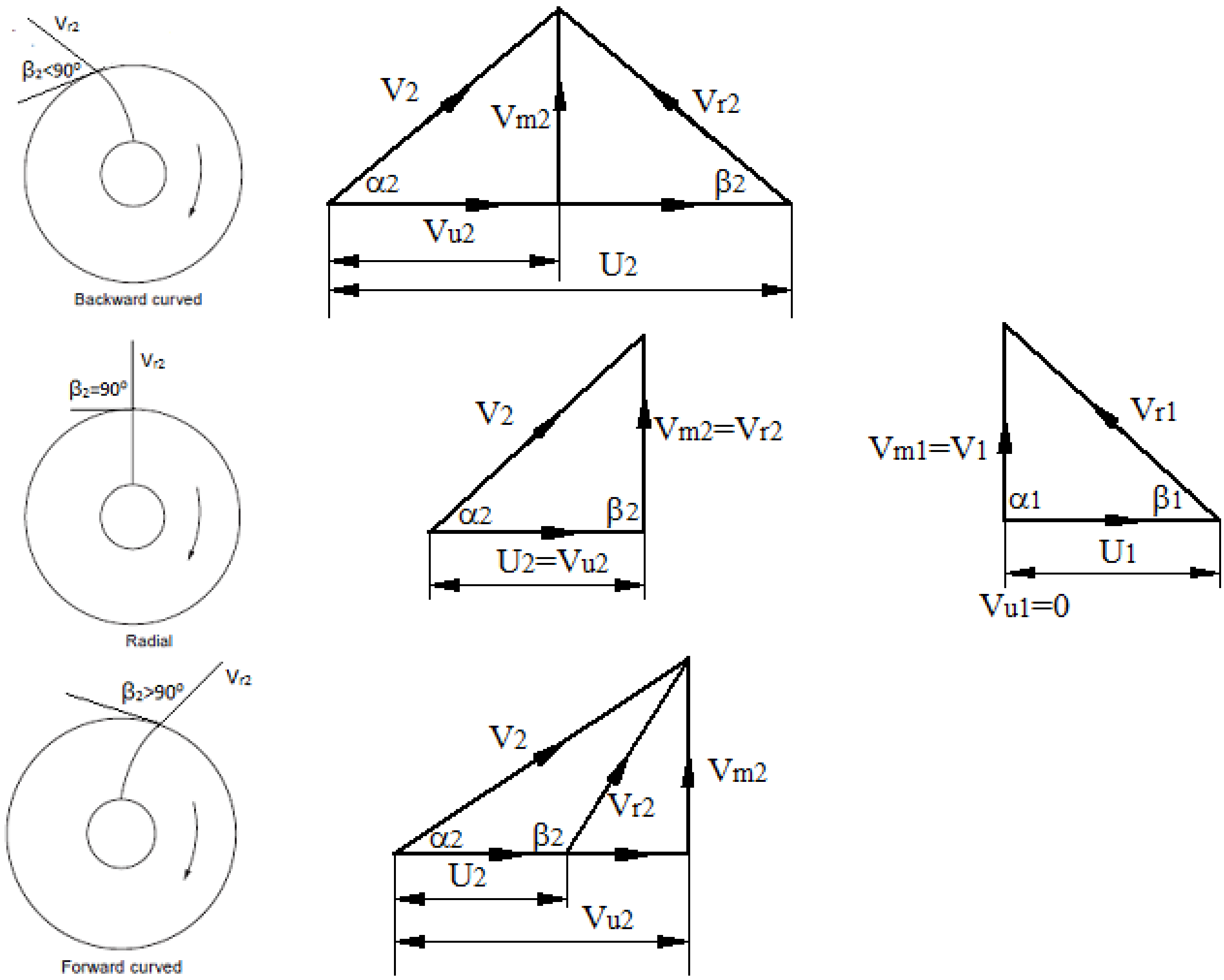

Fig. 5.7 Types of centrifugal vanes

The head-capacity characteristic curve for the above 3 types of vanes is given in figure 5.6, if β_2 lies between 0 to 90^o (backward curved vanes), cotβ_2 in H-Q relation is always positive. So for backward curved vanes the head developed by the machine falls with increasing discharge. For values of β_2 between 90^o to 180^o, cotβ_2 in H-Q relation is negative. So for forward curved vanes the head developed by the machine continuously rise with increasing discharge. For β_2=90^o (radial vanes), the head is independent of flow rates and is remains constant. For centrifugal machines usually the absolute velocity at the entry has no tangential component (i.e., V_{u1}= 0), thus the inlet velocity triangle for all the 3 types of vanes is same.

5.8 Static Pressure Rise in Centrifugal Pump:

Question No 5.10: Derive an expression for the static pressure rise in the impeller of a centrifugal pump with velocity triangles. ***(VTU, Dec-06/Jan-07, Jun/Jul-14)*** ***Or, For a centrifugal pump, show that***

the static head rise in the impeller neglecting the friction and other losses is given by $\frac{1}{2g}\left[V_{m1}^2 + U_2^2 - V_{m2}^2 cosec^2\beta_2\right]$ where V_{m1} and V_{m2} are velocities of flow at inlet and outlet, U_2 is tangential velocity of impeller at outlet and β_2 is vane angle at outlet.

(VTU, Jan/Feb-06, Dec-08/Jan-09, Dec-09/Jan-10)

Answer: The velocity diagram of a centrifugal pump is given by,

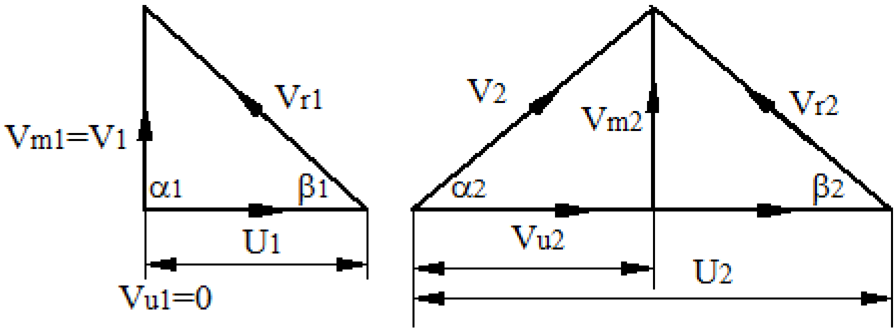

Fig. 5.8 Velocity triangles for centrifugal pump

Energy transfer due to static pressure change is given by,

$$e_{static} = \frac{(U_2^2 - U_1^2)}{2} - \frac{(V_{r2}^2 - V_{r1}^2)}{2}$$

From inlet velocity triangle, $\boldsymbol{V_{r1}^2 = U_1^2 + V_{m1}^2}$

From outlet velocity triangle, $V_{r2}^2 = V_{m2}^2 + x_2^2$

But, $cot\beta_2 = \frac{x_2}{V_{m2}} \Longrightarrow x_2 = V_{m2}cot\beta_2$

Then, $\boldsymbol{V_{r2}^2 = V_{m2}^2 + V_{m2}^2 cot^2\beta_2}$

Then,

$$e_{static} = \frac{(U_2^2 - U_1^2)}{2} - \frac{(V_{m2}^2 + V_{m2}^2 cot^2\beta_2 - U_1^2 - V_{m1}^2)}{2}$$

$$e_{static} = \frac{1}{2}[V_{m1}^2 + U_2^2 - V_{m2}^2(1 + cot^2\beta_2)]$$

Therefore static energy rise is given by,

$$\boldsymbol{e_{static} = \frac{1}{2}\left[V_{m1}^2 + U_2^2 - V_{m2}^2 cosec^2\beta_2\right]}$$

The static pressure rise is given by, $(P_2 - P_1) = \rho e_{static}$

$$\boldsymbol{(P_2 - P_1) = \frac{\rho}{2}\left[V_{m1}^2 + U_2^2 - V_{m2}^2 cosec^2\beta_2\right]}$$

The static head rise is given by, $h = \frac{(P_2 - P_1)}{\rho g} = \frac{e_{static}}{g}$

$$\boldsymbol{\frac{(P_2 - P_1)}{\rho g} = \frac{1}{2g}\left[V_{m1}^2 + U_2^2 - V_{m2}^2 cosec^2\beta_2\right]}$$

5.9 Minimum Starting Speed:

Question No 5.11: What is minimum starting speed of a centrifugal pump? Obtain an expression for the minimum starting speed of a centrifugal pump.

(VTU, Jun/Jul-09, May/Jun-10, Jun/Jul-14, Dec-14/Jan-15)

Answer: When the pump is started there will be no flow until the pressure rise in the impeller is more than or equal to the manometric head. In other words the centrifugal head should be greater than the manometric head. Therefore, minimum starting speed is the speed of centrifugal pump at which centrifugal head is equal to manometric head.

For minimum starting speed condition, *centrifugal head=manometric head*

$$\frac{(U_2^2 - U_1^2)}{2g} = H_m = \frac{\eta_{ma} U_2 V_{u2}}{g}$$

$$\frac{\left[\left(\frac{\pi D_2 N}{60}\right)^2 - \left(\frac{\pi D_1 N}{60}\right)^2\right]}{2} = \eta_{ma}\left(\frac{\pi D_2 N}{60}\right) V_{u2}$$

$$\frac{\left(\frac{\pi N}{60}\right)^2 [D_2^2 - D_1^2]}{2} = \eta_{ma}\left(\frac{\pi N}{60}\right) D_2 V_{u2}$$

$$\frac{\pi N D_2^2 \left[1 - \frac{D_1^2}{D_2^2}\right]}{120} = \eta_{ma} D_2 V_{u2}$$

$$\frac{\pi N D_2 \left[1 - \frac{D_1^2}{D_2^2}\right]}{120} = \eta_{ma} V_{u2}$$

Then minimum starting speed in rpm is,

$$\boldsymbol{N_{min} = \frac{120\eta_{ma} V_{u2}}{\pi D_2 \left[1 - \frac{D_1^2}{D_2^2}\right]}}$$

5.10 Multistage Pump:

Question No 5.12: Write a note on multistage centrifugal pumps. (VTU, Dec-06/Jan-07)

Question No 5.13: What are the applications of multi-stage centrifugal pumps? With a neat sketch, explain centrifugal pumps in series and parallel. (VTU, Dec-12)

Answer: If the centrifugal pump consists of two or more impellers connected in series on the same shaft, then the pump is called multistage pump. The water enters the impeller 1 of a multistage pump with two impellers as shown in figure 5.9 and pressure increases in it. The high pressure water from impeller 1 is then entering the impeller 2 where the pressure increases further. The flow generally is same in both the impeller. The head produced by the combined impellers will be higher than either of one, but need not be sum of them. If 'n' be the number of identical impellers and each produce a head of H_m, then the total head produced is given by $H_T = nH_m$.

Multistage pump is used for the following reasons:

1. To decrease the size of the impeller.
2. To develop high head.
3. To keep the outlet blade angle less than 45°.

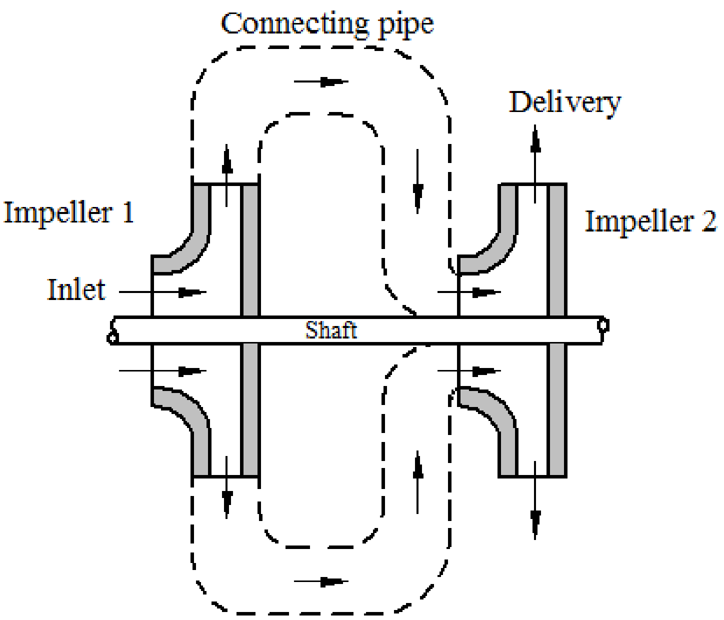

Fig. 5.9 Two stage centrifugal pump

5.11 Double-suction Pump:

In a double-suction pump two identically designed separate single stage pumps placed back-to-back (connected in parallel) as shown in figure 5.10. They draw water from a common source at a low level and discharge water to a common tank through a single delivery pipe. Each of the pump works against the same head. If 'n' be the number of identical impellers, each delivers the same flow rates which works under same head, then the total discharge is given by $Q_T = nQ$.

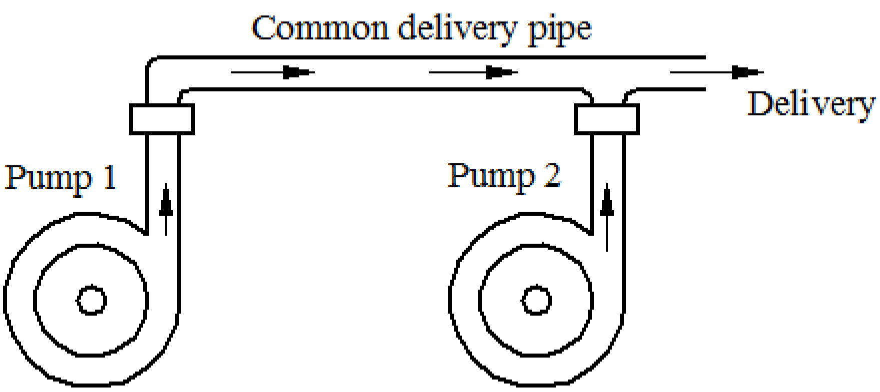

Fig. 5.10 Pumps connected in parallel

5.12 Slip and Slip Co-efficient:

In deriving the Euler's equation, it was assumed that the velocity and pressure distributions are uniform over the impeller cross-sectional area. But in actual practice this assumption is not correct, because the velocity and pressure are not uniform over an impeller cross-sectional area as shown in figure 5.11. Due to uneven pressure distribution and hence the velocity distribution the tangential component of the velocity (whirl velocity) reduces, thus head developed by the machine is always less than that developed at the ideal condition.

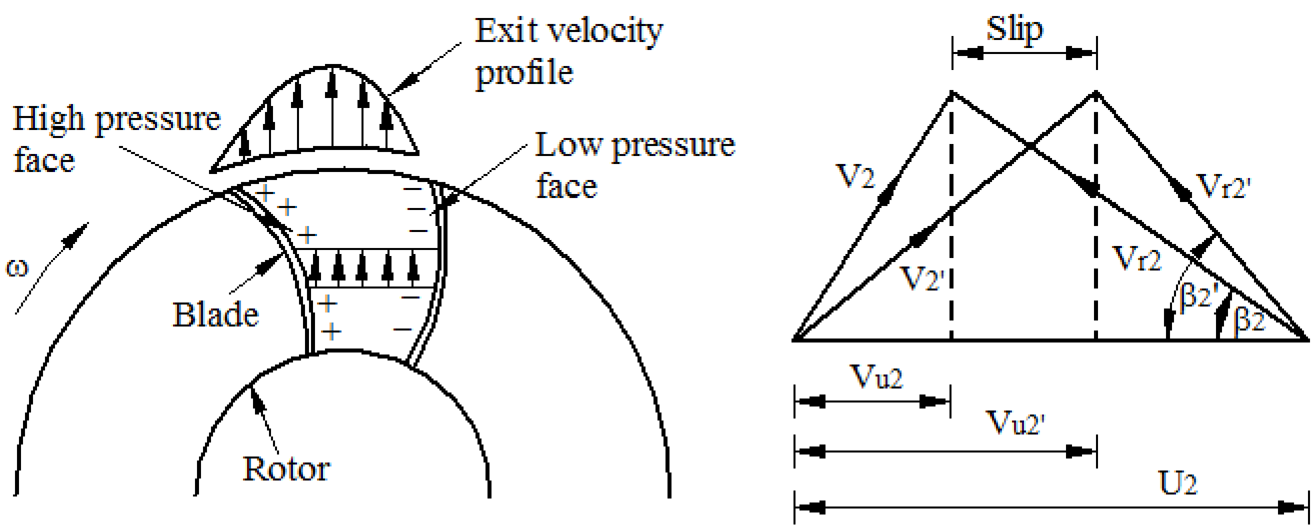

Fig. 5.11 Velocity and pressure distribution over impeller

Thus slip may be defined as the phenomenon observed in centrifugal machines due to uneven pressure distribution and the velocity distribution, which results in the reduction of tangential component of the velocity (whirl velocity).

If $V_{u2'}$ is the tangential component of the velocity without slip and (V_{u2}) is the tangential component of the velocity with slip, then slip (S) is $S = V_{u2'} - V_{u2}$

The ratio of ideal head (H_i) with slip to the Euler's head (H_e) without slip is called the slip co-efficient (μ).

$$\mu = \frac{H_i}{H_e} = \frac{V_{u2}}{V_{u2'}}$$

Then work done on fluid by a centrifugal pump with slip is given as:

$$w = gH = \mu U_2 V_{u2'}$$

Although above equation modified by the slip coefficient, it is still the theoretical work done on the fluid, since slip will be present even if the fluid is frictionless (ideal fluid).

Module 5: CENTRIFUGAL COMPRESSORS

5.1 Introduction:

Question No 5.1: Distinguish between fans, blowers and compressors and mention one application area for each. Or, Classify centrifugal compressors based on pressure developed. (VTU, Jul-07)

Answer: Centrifugal compressors work very much like centrifugal pumps except that they handle gases instead of liquids. Compressors as well as blowers and fans are the devices used to increase the pressure of a compressible fluid (gas).

A fan usually consists of a single rotor with or without a stator element and causes only a small rise in stagnation pressure of the flowing fluid, perhaps as low as 20 to 30 mm of water and very rarely in excess of 0.07 bar. Fans are used to provide strong circulating air currents or for air circulation and ventilation of buildings.

A blower may consist of one or more stages of compression with the rotors mounted on a common shaft. The air is compressed in a series of successive stages and is often led through a diffuser located near the exit. Blowers may run at very high shaft speeds and cause overall pressure rise in the range 1.5 to 2.5 bar. Blowers are used in ventilators, power stations, workshops, etc.

A compressor is a device used to produce large pressure changes ranging from 2.5 to 10 bar or more. Centrifugal compressors are mainly used in turbo-chargers.

The advantages of centrifugal compressor over the axial flow compressor are smaller length and wide range of mass flow rate of gas. The disadvantages are larger frontal area and lower maximum efficiency.

5.2 Components of Centrifugal Compressor:

Question No 5.2: Explain the various components of typical centrifugal compressors with the help of a schematic diagram. Discuss the actual pressure and velocity variations of flow across the impeller and diffuser. (VTU, Jul-07)

Answer: The principal components are the impeller and the diffuser. When the impeller is rotating at high speed, air is drawn in through the eye of the impeller. The absolute velocity of the inflow air is axial. The air then flows radially through the impeller passages due to centrifugal force. The total mechanical energy driving the compressor is transmitted to the fluid stream in the impeller where it is converted into kinetic energy, pressure and heat due to friction. The function of the diffuser is to convert the kinetic energy of air that leaving the impeller into pressure. The air leaving the diffuser is collected in a spiral casing from which it is discharged from compressor. The pressure and velocity variation across the centrifugal compressor is shown in figure 5.1.

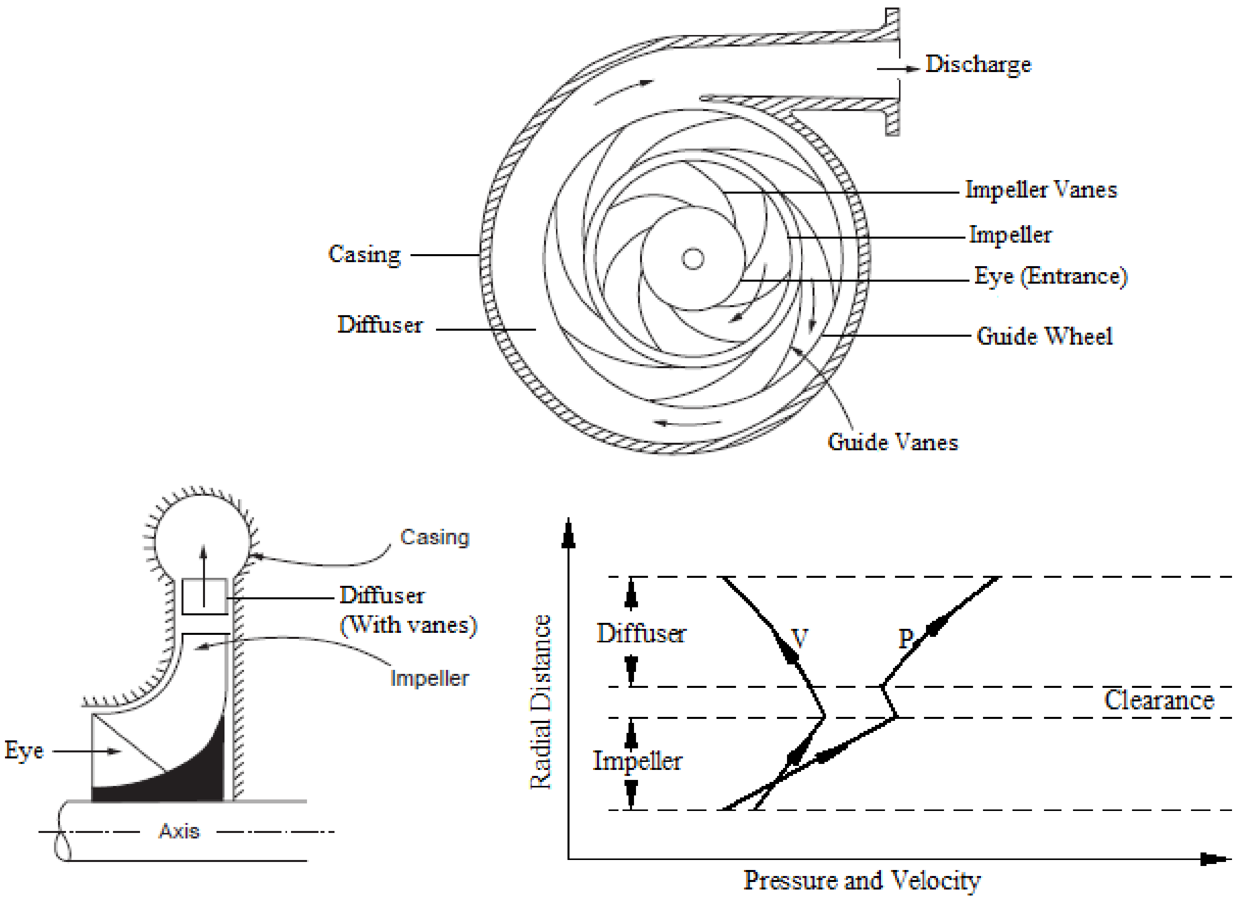

Fig. 5.1 Pressure and velocity diagram for centrifugal compressor

5.3 Types of Vane Shapes:

Question No 5.3: With a neat sketch and velocity triangles, explain different vane shapes of the centrifugal compressor. Draw the inlet velocity triangle assuming $V_{u1} = 0$. (VTU, May/Jun-10)

Answer: There are three types of vane shapes in centrifugal machines namely, (i) Backward curved vane (ii) Radial vane (iii) Forward curved vane.

The vane is said to be backward curved if the angle between the rotor blade-tip and the tangent to the rotor at the exit is acute ($\beta_2<90^o$). If it is a right angle ($\beta_2=90^o$) the blade said to be radial and if it is greater than 90^o, the blade is said to be forward curved. Here the blade angles measured with respect to direction of rotor (clockwise direction). The velocity triangles at the outlet of centrifugal machines are shown below.

For centrifugal machines usually the absolute velocity at the entry has no tangential component (i.e., $V_{u1}= 0$), thus the inlet velocity triangle for all the 3 types of vanes is same.

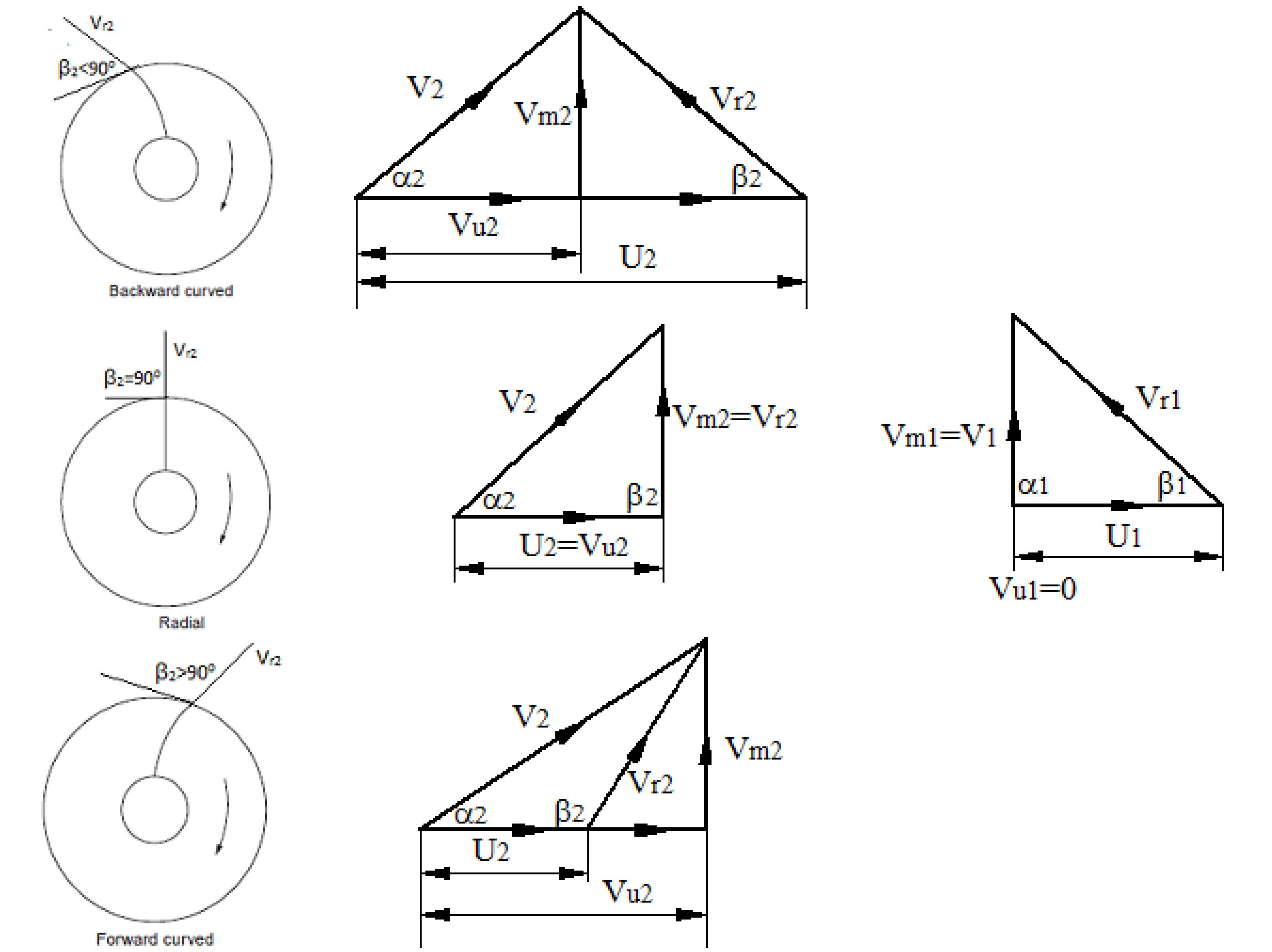

Fig. 5.2 Different vane shapes in centrifugal compressors.

5.4 Influence of Impeller Vane Shape:

Question No 5.4: Discuss with velocity diagram why backward curved vanes are preferred for radial flow (centrifugal) compressors.

Answer: Figure 5.2 gives the velocity diagram for different vane shapes. For compressor which absorb a specific amount of energy and runs at a given speed, the diameter (D_2) and whirl velocity (V_{u2}) can be varied to maintain U_2V_{u2} at the required value. Hence if β_2 is large (as in forward curved vane), V_{u2} is also large and U_2 has to be small that is diameter should be decreased. Similarly, if β_2 is small (as in backward curved vane), V_{u2} is small. Hence U_2 has to be large and the diameter should be increased appropriately to provide the required performance. This implies that compressor with backward curved vanes are larger in size than those with forward curved (or radial) vanes of the same capacity.

Now consider fluid flow through compressors running at the same tip speed (U_2) and with the same radial velocity (V_{m2}). Then, an increase in exit vane angle (β_2) increases exit fluid velocity (V_2) consequently, a very efficient diffuser is needed to obtain a pressure rise using all to of the kinetic energy at the exit. Because of irreversibilities due to adverse pressure gradients and thick boundary layers, complete diffusion of the exit kinetic energy with a pressure rise corresponding to the theoretical, is impossible. Therefore the compressors with large exit angles will be less efficient overall than compressors with small exit angles. So when high compressor efficiency is desired, machines with backward curved vanes must be used. This is one of the reasons that compressors with backward curved vanes are preferred. In some cases, where a large pressure rise is needed with a compressor of small size, radial blades are used though the efficiency may not be as high as that of a compressor with backward curved vanes of similar capacity. Compressors with forward curved vanes are even less common than those of radial type.

5.5 Slip and Slip Factor:

Question No 5.5: Briefly explain the slip and slip coefficient in centrifugal compressors. (VTU, Jun-12)

Answer:

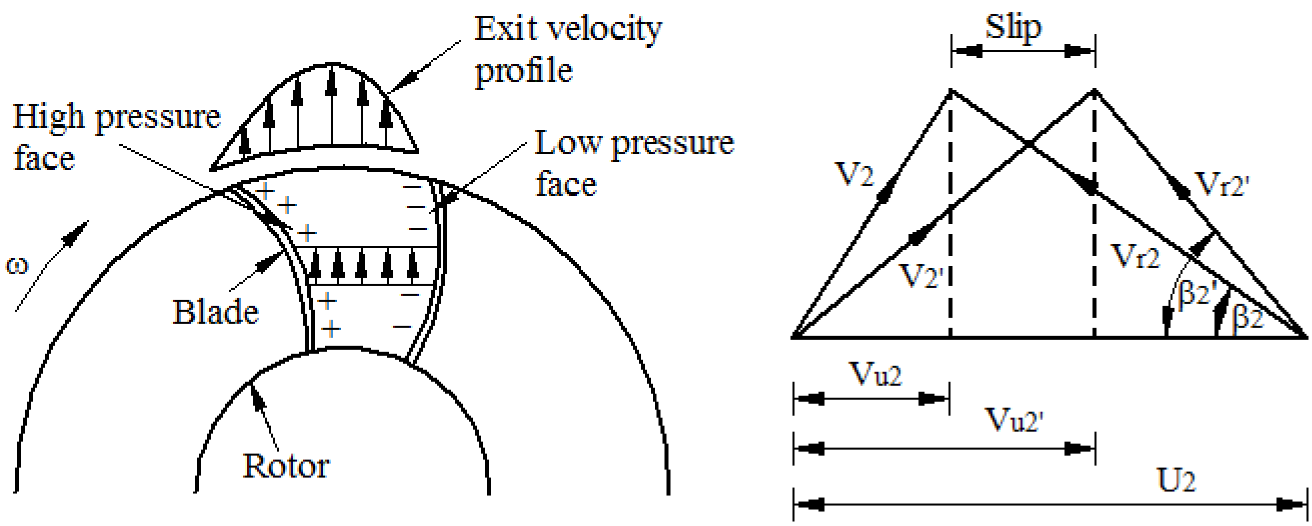

Fig. 5.3 Velocity and pressure distribution over impeller

In deriving the Euler's equation, it was assumed that the velocity and pressure distributions are uniform over the impeller cross-sectional area. But in actual practice this assumption is not correct, because the velocity and pressure are not uniform over an impeller cross-sectional area as shown in figure 5.3. Due to uneven pressure distribution and hence the velocity distribution the tangential component of the velocity (whirl velocity) reduces, thus head developed by the machine is always less than that developed at the ideal condition.

Thus slip may be defined as the phenomenon observed in centrifugal machines due to uneven pressure distribution and the velocity distribution, which results in the reduction of tangential component of the velocity (whirl velocity).

If $V_{u2'}$ is the tangential component of the velocity without slip and (V_{u2}) is the tangential component of the velocity with slip, then slip (S) is $S = V_{u2'} - V_{u2}$

The ratio of ideal head (H_i) with slip to the Euler's head (H_e) without slip is called the slip co-efficient (μ).

$$\mu = \frac{H_i}{H_e} = \frac{V_{u2}}{V_{u2'}}$$

Then theoretical work done on gas by a centrifugal compressor with slip is given as:

$$w_{the} = \Delta h_o = \mu U_2 V_{u2'} = \mu e$$

Although above equation modified by the slip coefficient, it is still the theoretical work done on the gas, since slip will be present even if the fluid is frictionless (ideal fluid).

5.6 Power Input Factor and Pressure Coefficient:

The losses that occur in a compressor are due to:

(i) Friction between air and the sides of the passages of flow or between disks.

(ii) The effects of shock (due to improper incidence), separation in regions of high adverse pressure gradients and turbulence.

(iii) Leakage between the tip of the rotor and casing.

(iv) Mechanical losses in bearings etc.

The frictional losses in the rotor and leakage losses between the tip of the rotor and casing make the actual work absorbed by the rotor, lesser than the theoretical. This fact is expressed by a quantity called the power input factor or the work factor, which is defined as the ratio of actual work supplied to the compressor to the theoretical work supplied to the same machine.

$$\Omega = \frac{w}{w_{the}} = \frac{w}{\mu U_2 V_{u2'}}$$

Then actual work supplied to the compressor is,

$$w = \Omega \mu U_2 V_{u2'} = \Omega \mu e$$

The frictional losses in the diffuser and the loss due to exit kinetic energy make the total static pressure rise, less than the theoretical maximum specified by the impeller speed. This fact is expressed by a quantity called the pressure or loading coefficient, which is defined as the ratio of the isentropic work needed to cause the observed rise to the isentropic work specified by the impeller tip-speed (Euler's work).

$$\psi_p = \frac{w_{ise}}{e}$$

But,

$$isentropic\ work\ (w_{ise}) = actual\ work \times isentropic\ efficiency = w \times \eta_c$$

$$w_{ise} = \eta_c \Omega \mu e$$

Then pressure coefficient is,

$$\psi_p = \frac{w \times \eta_c}{e} = \frac{\eta_c \Omega \mu U_2 V_{u2'}}{U_2 V_{u2'}}$$

$$\psi_p = \eta_c \Omega \mu$$

5.7 Work Done and Pressure Ratio:

Question No 5.6: Derive an expression for total-to-total pressure ratio in terms of impeller tip speed for a radial vanes centrifugal compressor. (VTU (Ph.D), Jan-12)

Answer: Figure 5.4 shows the enthalpy-entropy diagram for a centrifugal compressor stage. State points with superscript are correspond to isentropic compression processes. Air enters the impeller vanes with lower absolute velocity V_1 and leaves with large absolute velocity V_2. This absolute velocity reduces to V_3 when it passes through the diffuser vanes, which is slightly higher than V_1. It can be observed that stagnation pressure P_{o1} will be higher than static pressure P_1 by an amount $\frac{V_1^2}{2}$. Similarly P_{o2} is much higher than static pressure P_2 by an amount $\frac{V_2^2}{2}$.

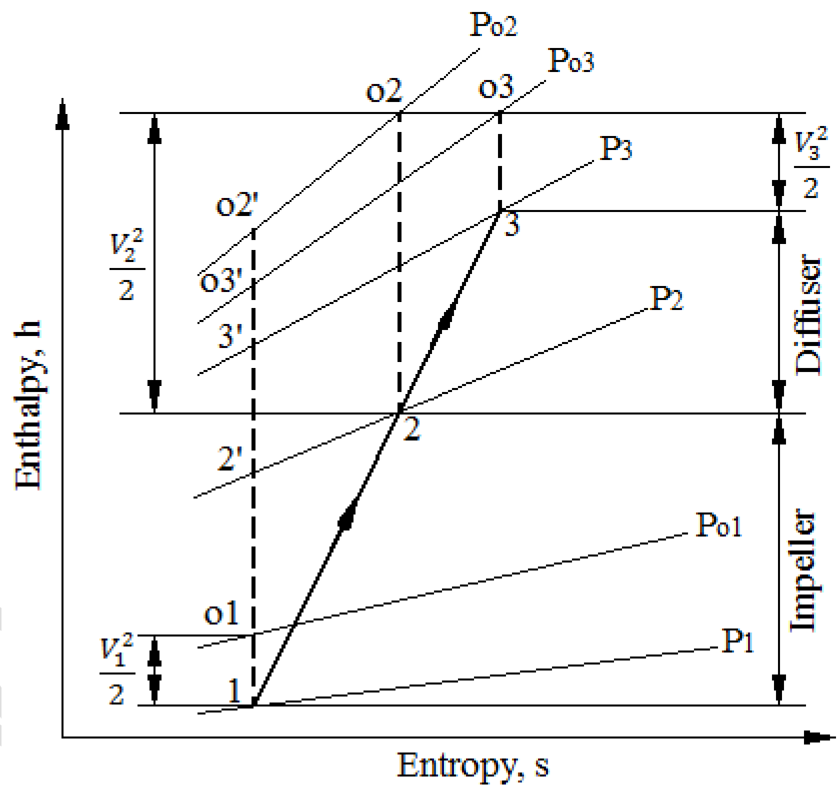

Fig. 5.4 Enthalpy-entropy diagram of a centrifugal compressor

Euler's work done on gas by a centrifugal compressor is given as:

$$e = U_2 V_{u2'}$$

(Because, $V_{u1} = 0$)

For backward curved vanes,

$$V_{u2'} = U_2 - V_{m2} \cot \beta_2$$

Then, $$\boldsymbol{e = U_2(U_2 - V_{m2}\cot\beta_2)}$$

For forward curved vanes,

$$V_{u2\prime} = U_2 + V_{m2}\cot(150^o - \beta_2) = U_2 - V_{m2}\cot\beta_2$$

Then, $$\boldsymbol{e = U_2(U_2 - V_{m2}\cot\beta_2)}$$

For radial vanes,

$$V_{u2\prime} = U_2$$

Then, $$\boldsymbol{e = U_2^2}$$

The stage efficiency of the compressor based on stagnation conditions at entry and exit is given by,

$$\eta_c = \frac{Total\ isentropic\ enthalpy\ rise\ between\ inlet\ and\ outlet}{Actual\ enthalpy\ rise\ between\ same\ total\ pressure\ limits}$$

$$\eta_c = \frac{h_{o3\prime} - h_{o1}}{h_{o3} - h_{o1}} = \frac{c_p(T_{o3\prime} - T_{o1})}{h_{o3} - h_{o1}}$$

Since, no work is done in the diffuser, $h_{o2} = h_{o3}$.

$$h_{o3} - h_{o1} = h_{o2} - h_{o1} = w = \Omega\mu U_2 V_{u2\prime}$$

$$\eta_c = \frac{c_p T_{o1}\left(\frac{T_{o3}\prime}{T_{o1}} - 1\right)}{\Omega\mu U_2 V_{u2\prime}} = \frac{c_p T_{o1}\left[\left(\frac{P_{o3}}{P_{o1}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\Omega\mu U_2 V_{u2\prime}}$$

Then pressure ratio of centrifugal compressor is,

$$P_{ro} = \frac{P_{o3}}{P_{o1}} = \left[1 + \frac{\eta_c \Omega\mu U_2 V_{u2\prime}}{c_p T_{o1}}\right]^{\frac{\gamma}{\gamma-1}}$$

For backward curved vanes,

$$\boldsymbol{P_{ro} = \left[1 + \frac{\eta_c \Omega\mu U_2(U_2 - V_{m2}\cot\beta_2)}{c_p T_{o1}}\right]^{\frac{\gamma}{\gamma-1}}}$$

For forward curved vanes,

$$\boldsymbol{P_{ro} = \left[1 + \frac{\eta_c \Omega\mu U_2(U_2 - V_{m2}\cot\beta_2)}{c_p T_{o1}}\right]^{\frac{\gamma}{\gamma-1}}}$$

For radial vanes,

$$\boldsymbol{P_{ro} = \left[1 + \frac{\eta_c \Omega\mu U_2^2}{c_p T_{o1}}\right]^{\frac{\gamma}{\gamma-1}}}$$

The interesting part of the above equation is that they permit direct evaluations of pressure ratio and work output, once the initial conditions and the rotor tip-speed are given and slip-coefficient, power input factor and efficiency are estimated.

5.5 Compressibility and Pre-whirl:

Question No 5.7: Discuss the compressibility for a centrifugal compressor. (VTU, Dec-12)

Answer: The Mach number is responsible for the compressibility of a flow. The higher the Mach number the greater will be the compressibility effect and hence it reduces the compressor efficiency. The Mach number at the inlet of the impeller eye is mainly depending on the relative velocity of the impeller at the inlet. It is therefore necessary to keep the relative velocity value as low as possible.

For a given flow rate, impeller eye may be either large or small. For large eye, velocity V_1 is low and eye tip speed U_1 is high. For small eye it is opposite. Both these conditions result in higher value of $V_{r1,}$ but it is minimum in between these two. Figure 5.5 shows the variation of relative Mach number with the eye tip diameter.

Question No 5.5: What is the necessity of providing the pre-whirl at the inlet of the centrifugal compressor? (VTU, Dec-11)

Answer: When the relative velocity is too high for efficient operation of a compressor and if flow rate and speed cannot be altered, still the relative velocity can be reduced by giving the fluid some initial positive pre-rotation. This is known as *Prewhirl*. This is usually done by providing inlet guide vanes installed directly in front of the eye as shown in figure 5.6. Sufficient prewhirl at the eye tip can avoid reaching the condition of critical Mach number leading to a prewhirl component. Positive prewhirl is disadvantageous because a positive inlet whirl velocity reduces energy transfer by an amount equal to U_1V_{u1}.

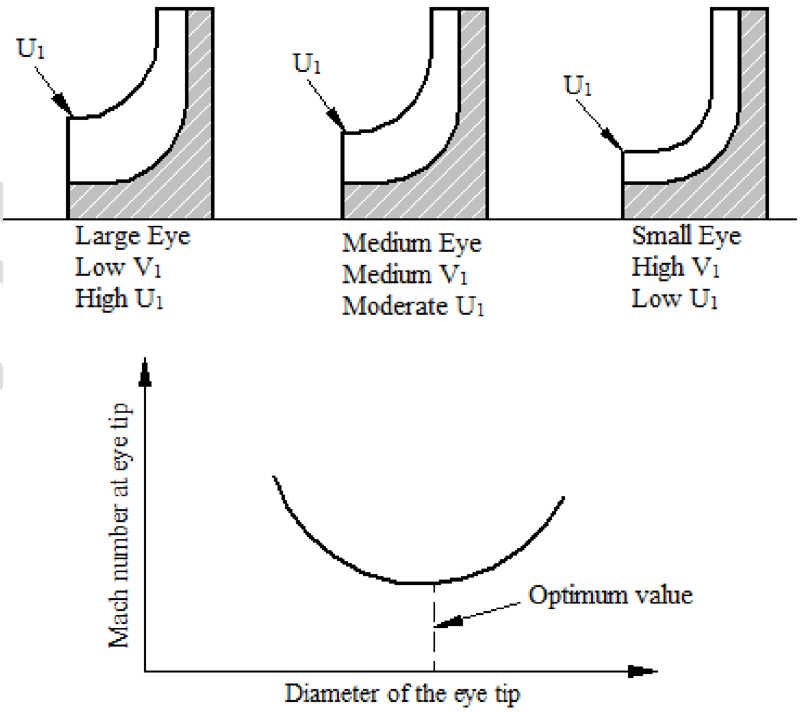

Fig. 5.5 Variation of Mach number with the eye tip diameter

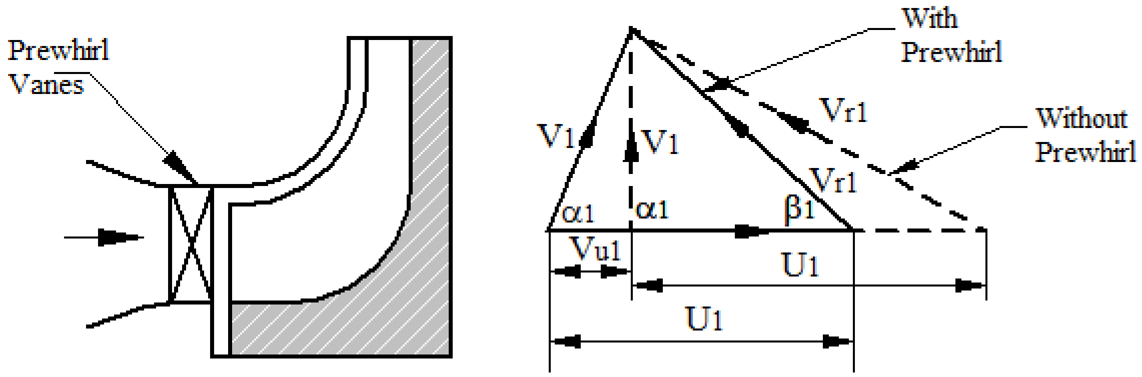

Fig. 5.6 Prewhirl at impeller eye

Note: C***ompressibility*** *is a measure of the relative volume change of a fluid or solid as a response to a pressure change.*

5.9 Diffuser:

Question No 5.9: What is the function of a diffuser? Name different types of diffusers used in centrifugal compressor and explain them with simple sketches. (VTU, Jun/Jul-09)

Answer: Diffuser plays an important role in the overall compression process of a centrifugal compressor. The impeller imparts energy to the air by increasing its velocity. The diffuser converts this imported kinetic energy into pressure rise. For a radial vanned impeller, the diffuser does compress and increase the pressure equal to 50 percent of the overall static pressure rise.

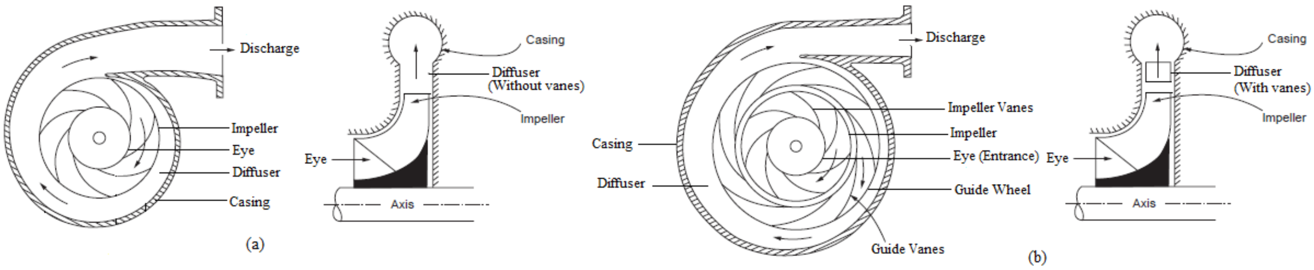

Fig. 5.7 Types of diffusers

5.9.1 Vaneless Diffuser: In this type, the diffusion process will take place in the vaneless space around the impeller before the air leaves the compressor stage through volute casing. A vaneless diffuser is shown in figure 5.7 (a). A vaneless diffuser has wide range of mass flow rate. But for a large reduction in the outlet kinetic energy, diffuser with a large radius is required. Because of long flow path with this type of diffuser, friction effects are important and the efficiency is low.

5.9.2 Vanned Diffuser: In the vanned diffuser as shown in figure 5.7 (b), the vanes are used to diffuse the outlet kinetic energy at a much higher rate, in a shorter length and with a higher efficiency than the vaneless diffuser. A ring of diffuser vanes surrounds the impeller at the outlet, and after leaving the impeller, the air enters the diffuser vanes

The diffuser efficiency defined as the ratio of ideal enthalpy rise to the actual enthalpy rise in the diffuser.

$$\eta_d = \frac{Ideal\ enthalpy\ rise}{Actual\ enthalpy\ rise}$$

5.10 Centrifugal Compressor Characteristics:

An idealized centrifugal compressor characteristic curve is shown in figure 5.5. Consider a centrifugal compressor delivering through a flow control valve situated after the diffuser. There is a certain pressure head, even if the valve is fully closed and is indicated by state 1. This pressure head is merely due to the churning action of the impeller vanes. The pressure head so developed is called “shut off” head. As flow control valve is opened, the air starts flowing and diffuser increases the pressure head. Thus, at state 2, the maximum pressure head is reached but the efficiency is just below the maximum efficiency.

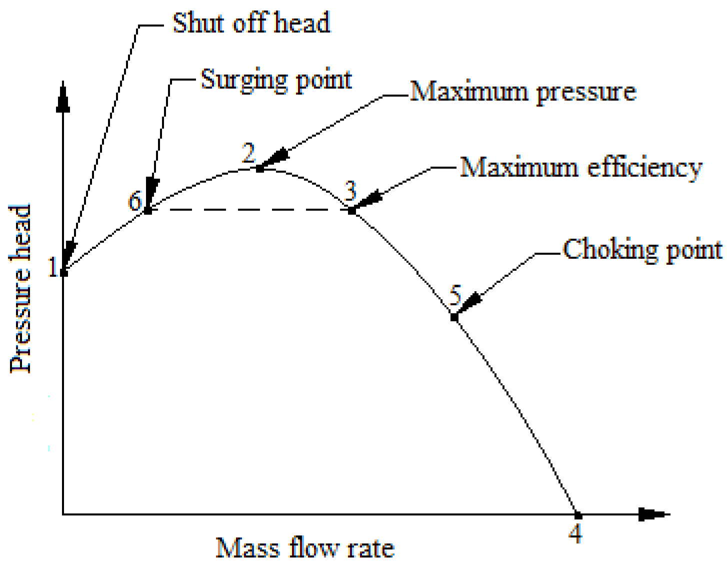

Fig. 5.5 Characteristic curve of centrifugal compressor

Further increase in mass flow reduces the pressure head to state 3. But at this state, the efficiency is maximum compared with state 2. Thus the value corresponding to state 3 is said to be design mass flow rate and pressure head.

Further increase in mass flow decreases the pressure head and reaches zero at state 4. Corresponding to this state, all the power absorbed by the compressor is used to overcome the internal friction and thus the compressor efficiency is zero.

5.10.1 Surging:

Question No 5.10: Explain the surging phenomena in centrifugal compressors with the help of head-discharge curves. (VTU, Dec-05/Jan-09, Dec-11, Jun-12, Jun/Jul-13)

Answer: The phenomenon of a momentary increase in the delivery pressure resulting in unsteady, periodic and reversal of flow through the compressor is called surging.

Consider compressor is operating at the state 3 as shown in figure 5.5, if the mass flow is reduced by gradual closing of the flow valve, the operating point move on to the left. Further reduction in mass flow increases the pressure head until it reaches the maximum value. Any further decrease in flow will not increase the pressure head and hence reduces the pressure head to state 6. At this condition there is a large pressure in the exit pipe than at compressor delivery and the flow stops momentarily, and may even flow in the reverse direction. This reduces the exit pipe pressure, then compressor again starts to deliver the air and the operating point quickly shifts to 3 again. Once again the pressure starts increasing and operating point moves from right to left. If the exit pipe conditions are remain unchanged then once again the flow will breakdown after state 2 and cycle will be repeated with high frequency. This phenomenon is called surging.

If the surging is severe enough then the compressor may be subjected to impact loads and high frequency vibration leads to failure of the compressor parts.

5.10.2 Choking:

Question No 5.11: Explain the choking phenomena in centrifugal compressors. (VTU, Dec-11,)

Answer: When the mass flow is increased to the right of point 3 on the characteristic curve (as in figure 5.5) a state 5 is reached, where no further increase in mass flow is possible no matter how wide open the flow control valve is. This indicates that the flow velocity in the passage reaches the speed of sound at some point within the compressor and the flow chokes. Choking means fixed mass flow rate regardless of pressure ratio. Choking may take place at the inlet, within the impeller, or in the diffuser section. It will occur in the inlet if stationary guide vanes are fitted.