Lesson
21
Centrifugal Compressors

The specific objectives of this lesson are to:

1. Explain the working principle of a centrifugal compressor (*Section 21.1*)
2. Present the analysis of centrifugal compressors (*Section 21.2*)
3. Discuss the selection of impeller diameter and speed of a centrifugal compressor using velocity diagrams (*Section 21.3*)
4. Discuss the effect of blade width on the capacity of centrifugal compressor (*Section 21.4*)
5. Discuss the methods of capacity control of a centrifugal compressor (*Section 21.5*)
6. Discuss the performance aspects and the phenomenon of surging in centrifugal compressors (*Section 21.6*)
7. Compare the performance of a centrifugal compressor with a reciprocating compressor vis-á-vis condensing and evaporator temperatures and compressor speed (*Section 21.6*)
8. Describe commercial refrigeration systems using centrifugal compressors (*Section 21.7*)

At the end of the lecture, the student should be able to:

1. Explain the working principle of a centrifugal compressor with suitable diagrams
2. Analyse the performance of a centrifugal compressor using steady flow energy equation and velocity diagrams
3. Calculate the required impeller diameter and/or speed of a centrifugal compressor
4. Explain the limitations on minimum refrigeration capacity of centrifugal compressors using velocity diagrams
5. Explain the methods of capacity control of centrifugal compressor
6. Explain the phenomenon of surging
7. Compare the performance aspects of centrifugal and reciprocating compressors

21.1. Introduction:

Centrifugal compressors; also known as turbo-compressors belong to the roto-dynamic type of compressors. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour by a high-speed impeller into static pressure. Unlike reciprocating compressors, centrifugal compressors are steady-flow devices hence they are subjected to less vibration and noise.

Figure 21.1 shows the working principle of a centrifugal compressor. As shown in the figure, low-pressure refrigerant enters the compressor through the eye of the impeller (1). The impeller (2) consists of a number of blades, which form flow passages (3) for refrigerant. From the eye, the refrigerant enters the flow passages formed by the impeller blades, which rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser (4). In the diffuser, the refrigerant is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The vapour from the diffuser enters the volute casing (5) where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally, the pressurized refrigerant leaves the compressor from the volute casing (6).

The gain in momentum is due to the transfer of momentum from the high-speed impeller blades to the refrigerant confined between the blade passages. The increase in static pressure is due to the self-compression caused by the centrifugal action. This is analogous to the gravitational effect, which causes the fluid at a higher level to press the fluid below it due to gravity (or its weight). The static pressure produced in the impeller is equal to the static head, which would be produced by an equivalent gravitational column. If we assume the impeller blades to be radial and the inlet diameter of the impeller to be small, then the static head, h developed in the impeller passage for a single stage is given by:

$$\mathbf{h} = \frac{\mathbf{V^2}}{\mathbf{g}} \qquad (21.1)$$

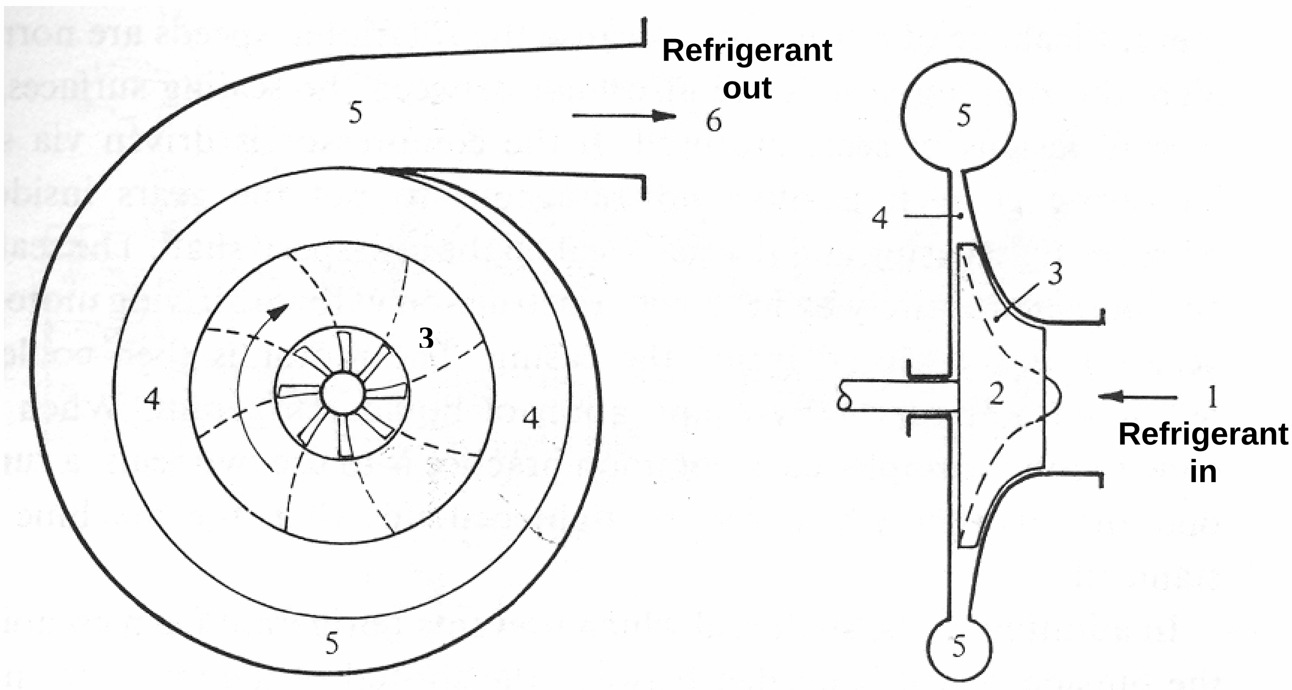
where h = static head developed, m

V = peripheral velocity of the impeller wheel or tip speed, m/s

g = acceleration due to gravity, m/s^2

Hence increase in total pressure, ΔP as the refrigerant flows through the passage is given by:

$$\Delta \mathbf{P} = \rho \mathbf{g h} = \rho \mathbf{V^2} \qquad (21.2)$$



***21.1**. Centrifugal Compressor*
1: Refrigerant inlet (eye); 2: Impeller; 3: Refrigerant passages
4: Vaneless diffuser; 5: Volute casing; 6: Refrigerant discharge

Thus it can be seen that for a given refrigerant with a fixed density, the pressure rise depends only on the peripheral velocity or tip speed of the blade. The tip speed of the blade is proportional to the rotational speed (RPM) of the impeller and the impeller diameter. The maximum permissible tip speed is limited by the strength of the structural materials of the blade (usually made of high speed chrome-nickel steel) and the sonic velocity of the refrigerant. Under these limitations, the maximum achievable pressure rise (hence maximum achievable temperature lift) of single stage centrifugal compressor is limited for a given refrigerant. Hence, multistage centrifugal compressors are used for large temperature lift applications. In multistage centrifugal compressors, the discharge of the lower stage compressor is fed to the inlet of the next stage compressor and so on. In multistage centrifugal compressors, the impeller diameter of all stages remains same, but the width of the impeller becomes progressively narrower in the direction of flow as refrigerant density increases progressively.

The blades of the compressor or either forward curved or backward curved or radial. Backward curved blades were used in the older compressors, whereas the modern centrifugal compressors use mostly radial blades.

The stationary diffuser can be vaned or vaneless. As the name implies, in vaned diffuser vanes are used in the diffuser to form flow passages. The vanes can be fixed or adjustable. Vaned diffusers are compact compared to the vaneless diffusers and are commonly used for high discharge pressure applications. However, the presence of vanes in the diffusers can give rise to shocks, as the refrigerant velocities at the tip of the impeller blade could reach sonic velocities in large, high-speed centrifugal compressors. In vaneless diffusers the velocity of refrigerant in the diffuser decreases and static pressure increases as the radius increases. As a result, for a required pressure rise, the required size of the vaneless diffuser could be large compared to vaned diffuser. However, the problem of shock due to supersonic velocities at the tip does not arise with vaneless diffusers as the velocity can be diffused smoothly.

Generally adjustable guide vanes or pre-rotation vanes are added at the inlet (eye) of the impeller for capacity control.

21.2. Analysis of centrifugal compressors:

Applying energy balance to the compressor (Fig.24.2), we obtain from steady flow energy equation:

$$-Q+m(h_i+\frac{V_i^2}{2}+gZ_i)=-W_c+m(h_e+\frac{V_e^2}{2}+gZ_e) \qquad (21.3)$$

where Q = heat transfer rate from the compressor
W = work transfer rate to the compressor
m = mass flow rate of the refrigerant
V_i,V_e = Inlet and outlet velocities of the refrigerant
Z_i,Z_e = Height above a datum in gravitational force field at inlet and outlet

Neglecting changes in kinetic and potential energy, the above equation becomes:

$$-Q+mh_i=-W_c+mh_e \qquad (21.4)$$

In a centrifugal compressor, the heat transfer rate Q is normally negligible (as the area available for heat transfer is small) compared to the other energy terms, hence the rate of compressor work input for adiabatic compression is given by:

$$W_c=m(h_e-h_i) \qquad (21.5)$$

The above equation is valid for both reversible as well as irreversible adiabatic compression, provided the actual enthalpy is used at the exit in case of irreversible compression. In case of reversible, adiabatic compression, the power input to the compressor is given by:

$$W_{c,isen}=m(h_e-h_i)_{isen} \qquad (21.6)$$

then using the thermodynamic relation, *Tds=dh–vdp*; the isentropic work of compression is given by:

$$\mathbf{w_{c,isen} = (h_e - h_i)_{isen} = \int_{Pi}^{Pe} vdp\Big|_{isen}} \qquad (21.7)$$

Thus the expression for reversible, isentropic work of compression is same for both reciprocating as well as centrifugal compressors. However, the basic difference between actual reciprocating compressors and actual centrifugal compressors lies in the source of irreversibility.

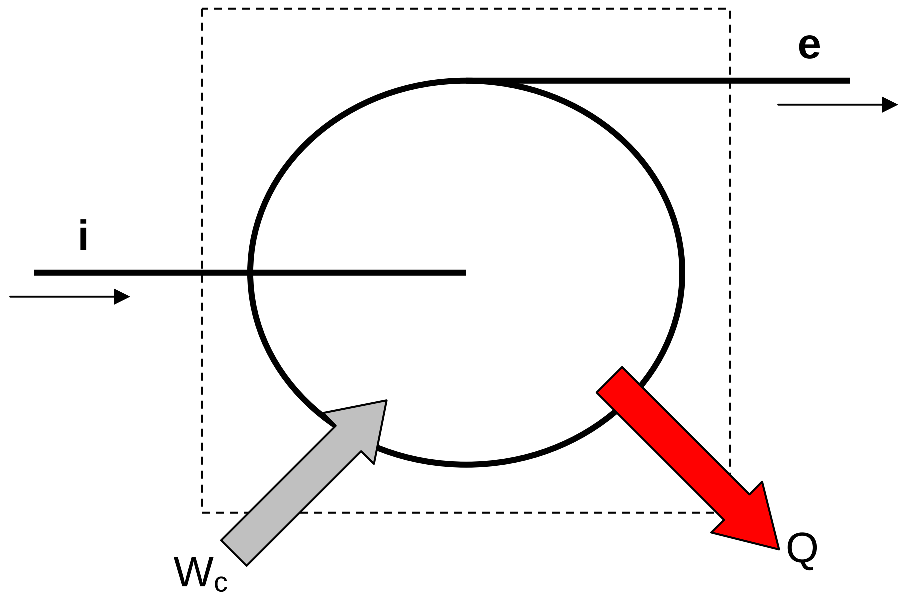


Fig.21.2*. Energy balance across a compressor*

In case of reciprocating compressors, the irreversibility is mainly due to heat transfer and pressure drops across valves and connecting pipelines. However, in case of centrifugal compressors, since the refrigerant has to flow at very high velocities through the impeller blade passages for a finite pressure rise, the major source of irreversibility is due to the viscous shear stresses at the interface between the refrigerant and the impeller blade surface.

In reciprocating compressors, the work is required to overcome the normal forces acting against the piston, while in centrifugal compressors, work is required to overcome both normal pressure forces as well as viscous shear forces. The specific work is higher than the area of P-v diagram in case of centrifugal compressors due to irreversibilities and also due to the continuous increase of specific volume of refrigerant due to fluid friction.

To account for the irreversibilities in centrifugal compressors, a polytropic efficiency η_{pol} is defined. It is given by:

$$\eta_{pol} = \frac{w_{pol}}{w_{act}} = \frac{\int_{Pi}^{Pe} vdp}{(h_e - h_i)} \qquad (21.8)$$

where w_{pol} and w_{act} are the polytropic and actual works of compression, respectively.

The polytropic work of compression is usually obtained by the expression:

$$w_{pol} = \int_{Pi}^{Pe} vdP = f\left(\frac{n}{n-1}\right)Pivi\left[\left(\frac{Pe}{Pi}\right)^{\frac{n-1}{n}} - 1\right] \qquad (21.9)$$

where n is the index of compression, f is a correction factor which takes into account the variation of n during compression. Normally the value of f is close to 1 (from 1.00 to 1.02), hence it may be neglected in calculations, without significant errors.

If the refrigerant vapour is assumed to behave as an ideal gas, then it can be shown that the polytropic efficiency is equal to:

$$\eta_{pol} = \left(\frac{n}{n-1}\right)\left(\frac{\gamma - 1}{\gamma}\right) \qquad (21.10)$$

where γ = specific heat ratio, cp/cv (assumed to be constant).

Though refrigerant vapours do not strictly behave as ideal gases, the above simple equation is often used to obtain the polytropic efficiency of the centrifugal compressors by replacing γ by isentropic index of compression, k, i.e., for actual refrigerants the polytropic efficiency is estimated from the equation:

$$\eta_{pol} = \left(\frac{n}{n-1}\right)\left(\frac{k - 1}{k}\right) \qquad (21.11)$$

For actual centrifugal compressors, the polytropic efficiency is found to lie in the range of 0.7 to 0.85. The index of compression n is obtained from actual measurements of pressures and specific volumes at the inlet and exit of the compressor and then using the equation Pv^n = constant. This procedure usually gives fairly accurate results for refrigerants made of simple molecules such as water, ammonia. The deviation between actual efficiency and polytropic efficiency evaluated using the above equations can be significant in case of heavier molecules such as R 22, R 134a.

When the refrigerant velocities are high, then the change in kinetic energy across the compressor can be considerable. In such cases, these terms have to be included in the steady flow energy equation. If the heat transfer rate is negligible and change in kinetic energy is considerable, then the rate of work input to the compressor is given by:

$$\mathbf{W_c = m(h_{t,e} - h_{t,i})} \qquad (21.12)$$

where $h_{t,e}$ and $h_{t,i}$ are the total or stagnation enthalpies at the exit and inlet to the compressor, respectively. The stagnation enthalpy of the refrigerant h_t is given by:

$$\mathbf{h_t = h + \frac{V^2}{2}} \qquad (21.13)$$

where h is the specific enthalpy of the refrigerant and V is its velocity. Similar to stagnation enthalpy, one can also define stagnation temperature and stagnation pressure. The stagnation pressure P_t is defined as the pressure developed as the refrigerant is decelerated reversibly and adiabatically from velocity V to rest. Then from energy balance,

$$\mathbf{\int_{P}^{Pt} vdp\Big|_{isen} = h_t - h = \frac{V^2}{2}} \qquad (21.14)$$

Stagnation pressure and temperature of moving fluids can be measured by pressure and temperature sensors moving with the fluid at the same velocity.

<u>For an ideal gas:</u>

$$\mathbf{(h_t - h) = \frac{V^2}{2} = Cp(T_t - T)} \qquad (21.15)$$

where T_t is the total or stagnation temperature given by:

$$\mathbf{T_t = T + \frac{V^2}{2Cp}} \qquad (21.16)$$

where T is the static temperature and Cp is the specific heat at constant pressure.

For an incompressible fluid (density ≈ constant):

$$\int_{P}^{Pt} vdp\Big|_{isen} = \frac{V^2}{2} \approx v(P_t - P) \qquad (21.17)$$

hence the stagnation pressure of an incompressible fluid is given by:

$$P_t = P + \frac{1}{2}\frac{V^2}{v} \qquad (21.18)$$

21.3. Selection of impeller speed and impeller diameter:

As the refrigerant vapour flows from the suction flange to the inlet to the impeller, its stagnation enthalpy remains constant as no work is done during this section. However, the velocity of the refrigerant may increase due to reduction in flow area. Depending upon the presence or absence of inlet guide vanes in the eye of the impeller, the refrigerant enters the impeller with a pre-rotation or axially. Then the direction of the refrigerant changes by 90^o as it enters the flow passages between the impeller blades from the inlet. As the refrigerant flows through the blade passages its stagnation enthalpy rises as work of compression is supplied to the refrigerant through the impeller blades. Simultaneously its velocity and static pressure rise due to the momentum transfer and self-compression. However, the relative velocity between refrigerant and impeller blades usually reduces as the refrigerant flows towards the tip. From the tip of the impeller the refrigerant enters the diffuser, where its static pressure increases further due to deceleration, however, its total enthalpy remains constant as no energy transfer takes place to the refrigerant. From the diffuser the refrigerant enters the volute casing where further pressure rise takes place due to conversion of velocity into static pressure, while the total enthalpy remains constant as no energy is added to the refrigerant in the volute casing. Thus the total enthalpy of the refrigerant remains constant everywhere except across the impeller. To establish a relation between the power input and the impeller speed and diameter, it is essential to find the torque required to rotate the impeller. This calls for application of conservation of angular momentum equation to the refrigerant across the impeller.

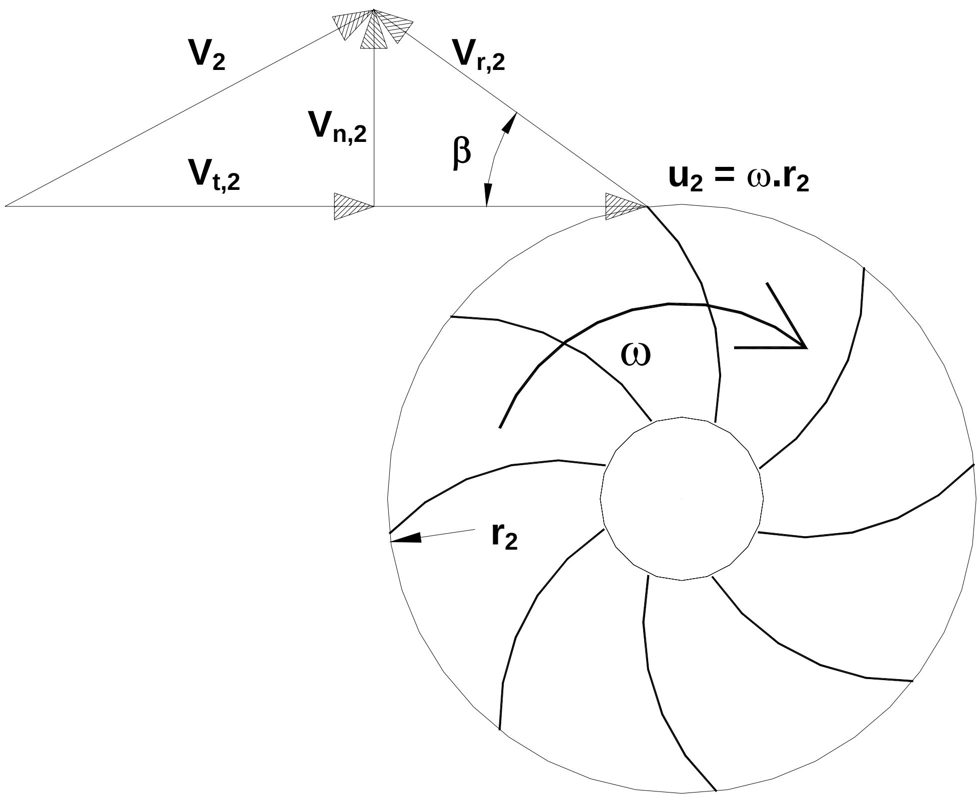
Figure 21.3 shows the velocity diagram at the outlet of the impeller. The torque required to rotate the impeller is equal to the rate of change of the angular momentum of the refrigerant. Assuming the refrigerant to enter the impeller blade passage radially with no tangential component at inlet, the torque τ is given by:

$$\tau = mr_2 V_{t,2} \qquad (21.19)$$

where m is the mass flow rate of the refrigerant, r_2 is the outer radius of the impeller blade and $V_{t,2}$ is the tangential component of the absolute refrigerant velocity V_2 at impeller exit. The power input to the impeller W is given by:

$$\mathbf{P = \tau.\omega = mr_2\omega V_{t,2} = mu_2 V_{t,2}} \qquad (21.20)$$

where u_2 is the tip speed of the impeller blade = $\omega.r_2$. ω is the rotational speed in radians/s and r_2 is the impeller blade radius.



u_2	= $\omega.r_2$ = Tip speed of the impeller
ω	= Rotational speed of impeller
V_2	= Absolute velocity of fluid
$V_{r,2}$	= Relative velocity of fluid w.r.t to the impeller
$V_{t,2}$	= Tangential component of V_2
$V_{n,2}$	= Normal component of V_2

***21.3**: Velocity diagram at the outlet of the impeller of a centrifugal compressor*

The velocity diagram also shows the normal component of refrigerant velocity, $V_{n,2}$ at the impeller outlet. The volume flow rate from the impeller is proportional to the normal component of velocity. From the velocity diagram the tangential component $V_{t,2}$ can be written in terms of the tip speed u_2, normal component $V_{n,2}$ and the outlet blade angle β as:

$$V_{t,2} = u_2 - V_{n,2}\cot\beta = u_2\left(1 - \frac{V_{n,2}\cot\beta}{u_2}\right) \tag{21.21}$$

Hence the power input to the impeller, W is given by:

$$W = mu_2 V_{t,2} = mu_2^{\,2}\left(1 - \frac{V_{n,2}\cot\beta}{u_2}\right) \tag{21.22}$$

Thus the power input to the compressor depends on the blade angle β. The blade angle will be less than 90^o for backward curved blade, equal to 90^o for radial blades and greater than 90^o for forward curved blade. Thus for a given impeller tip speed, the power input increases with the blade angle β.

<u>If the blades are radial</u>, then the power input is given by:

$$W = mu_2^{\,2}\left(1 - \frac{V_{n,2}\cot\beta}{u_2}\right) = mu_2^{\,2};\ \text{for } \beta = 90^o \tag{21.23}$$

If the compression process is reversible and adiabatic, then power input can also be written as:

$$W_{c,isen} = m(h_e - h_i)_{isen} = m\int_{Pi}^{Pe} vdp\Big|_{isen} \tag{21.24}$$

Comparing the above two equations:

$$(h_e - h_i)_{isen} = \int_{Pi}^{Pe} vdP\Big|_{isen} = u_2^{\,2} = (\omega r_2)^2 \tag{21.25}$$

The above equation can also be written as:

$$\int_{Pi}^{Pe} vdP\Big|_{isen} = \left(\frac{k}{k-1}\right) Pivi\left[\left(\frac{Pe}{Pi}\right)^{\frac{k-1}{k}} - 1\right] = (\omega r_2)^2 \tag{21.26}$$

Thus from the above equation, the pressure ratio, r_p = (Pe/Pi) can be written as:

$$r_p = \left(\frac{Pe}{Pi}\right) = \left[1 + \left(\frac{k-1}{k}\right)\left(\frac{1}{Pivi}\right)(\omega r_2)^2\right]^{\frac{k}{k-1}} \tag{21.27}$$

Thus it can be seen from the above expression that for a given refrigerant at a given suction conditions (i.e., fixed k, Pi and vi), pressure ratio is proportional to the rotational speed of the compressor and the impeller blade diameter. Hence, larger the required temperature lift (i.e., larger pressure ratio) larger should be the rotational speed and/or impeller diameter.

Generally from material strength considerations the tip speed, u_2 ($=\omega r_2$) is limited to about 300 m/s. This puts an upper limit on the temperature lift with a single stage centrifugal compressor. Hence, for larger temperature lifts require multi-stage compression. For a given impeller rotational speed and impeller diameter, the pressure rise also depends on the type of the refrigerant used.

<u>For example</u>, for a single stage saturated cycle operating between an evaporator temperature of 0^oC and a condensing temperature of 32^oC, the required tip speed [$V_{t,2} = (he\text{-}hi)_{isen}^{1/2}$) will be 145.6 m/s in case of R134a and 386 m/s in case of ammonia. If the impeller rotates at 50 rps, then the required impeller radius would be 0.4635m in case of R 134a and 1.229m in case of ammonia. In general smaller tip speeds and impeller size could be obtained with higher normal boiling point refrigerants. This is the reason behind the wide spread use of R 11 (NBP = 23.7^oC) in centrifugal compressors prior to its ban.

Similar type of analyses can be carried out for other types of blades (i.e., forward or backward) and also with a pre-rotation at impeller inlet (i.e., $V_{t,1} \neq 0$). However, the actual analyses can be quite complicated if one includes the pre-rotation guide vanes, slip between the refrigerant and impeller blades etc.

In actual compressors, the angle at which fluid leaves the impeller β' will be different from the blade angle β. This is attributed to the internal circulation of refrigerant in the flow passages between the impeller blades. As the refrigerant flows outwards along a rotating radius, a pressure gradient is developed across the flow passage due to the Coriolis component of acceleration. Due to this pressure difference, eddies form in the flow channels as shown in Fig.21.4. As shown, these eddies rotate in a direction opposite to that of the impeller, as a result the actual angle β' at which the refrigerant leaves the impeller will be less than the blade angle β. Due to this, the tangential component of velocity $V_{t,2}$ reduces, which in turn reduces the pressure rise and also the volumetric flow rate of refrigerant. The ratio of actual tangential velocity component ($V_{t,act}$) to the tangential component without eddy formation ($V_{t,2}$) is known as slip factor. The slip factor can be increased by increasing the number of blades (i.e., by decreasing the area of individual flow passages), however, after a certain number of blades, the efficiency drops due increased frictional losses. Hence, the number of blades are normally optimized considering the slip factor and frictional losses.

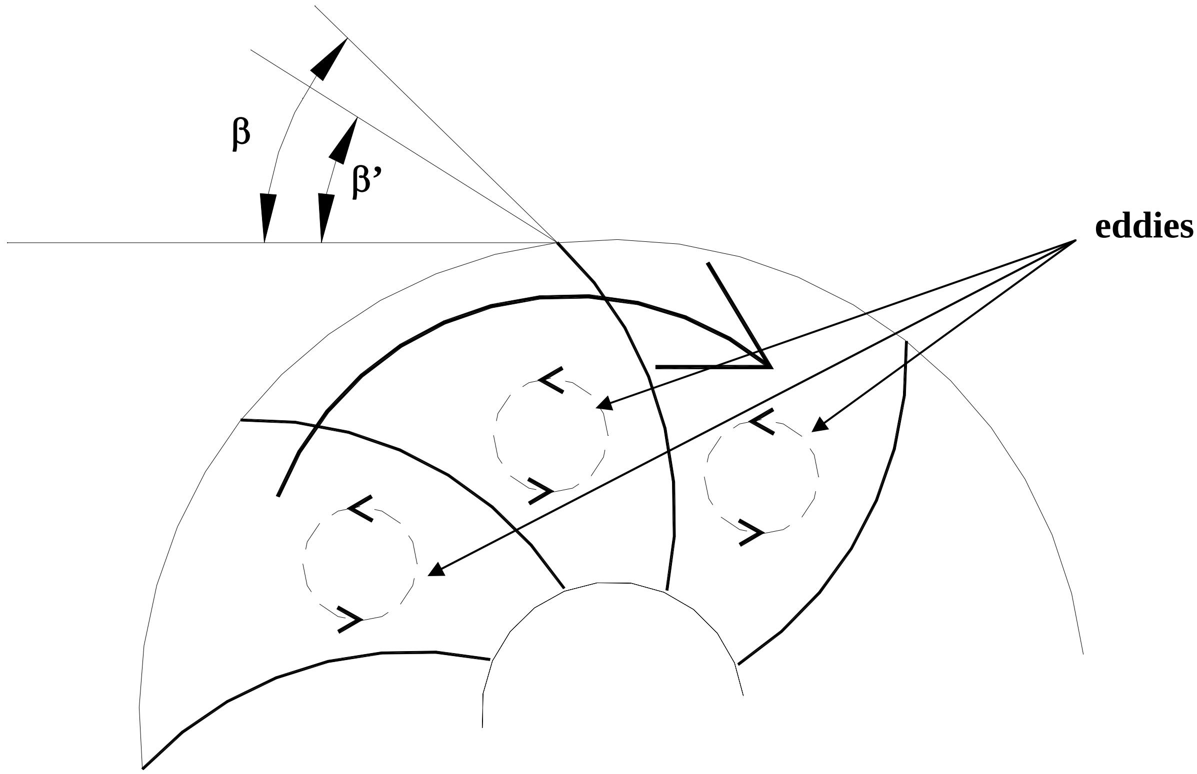


Fig.21.4: *Formation of eddies in a backward curved centrifugal compressor*

21.4. Refrigerant capacity of centrifugal compressors:

The refrigerant capacity of a centrifugal compressor depends primarily on the tip speed and width of the impeller. For a given set of condenser and evaporator temperatures the required pressure rise across the compressor remains same for all capacities, large and small. Since the pressure rise depends on the impeller diameter, number of impellers and rotational speed of the impeller, these parameters must remain same for all compressors of all capacities operating between the same condenser and evaporator temperatures.

The mass flow rate through a centrifugal compressor can be written as:

$$\mathbf{m} = \frac{\mathbf{V_{n,2} A_{f,p}}}{\mathbf{v_2}} \qquad (21.28)$$

where $V_{n,2}$ = Normal component of velocity at the exit
$A_{f,p}$ = Flow area at the periphery
v_2 = Specific volume of the refrigerant at the periphery

For a given blade diameter, the flow area at the periphery depends on the number of blades and the width of the blade. If the number of blades is fixed, then the flow area depends only on the width of the impeller.

Hence, one way to design the compressors for different refrigerant capacities is by controlling the width of the impeller (Fig.21.5). To design the compressor for smaller refrigerant capacity, one has to reduce the width of the impeller. However, as the width of the impeller is reduced frictional losses between the refrigerant and impeller blades increase leading to lower efficiency. Of course another alternative is to reduce both diameter and width of the impeller simultaneously, thereby the frictional losses can be reduced. However, since this reduces the pressure rise across a single impeller, one has to increase the number of stages, which leads to higher manufacturing costs. This puts a lower limit on the refrigerant capacity of centrifugal compressors. In practice, the lower volumetric flow rate is limited to about 0.7 m^3/s and the minimum refrigeration capacities are around 300 kW for air conditioning applications. Since the compressor works more efficiently at higher volumetric flow rates, refrigerants having lower densities (i.e., higher normal boiling points) such as R 11, water are ideal refrigerants for centrifugal compressors. However, centrifugal compressors in larger capacities are available for a wide range of refrigerants, both synthetic and natural.

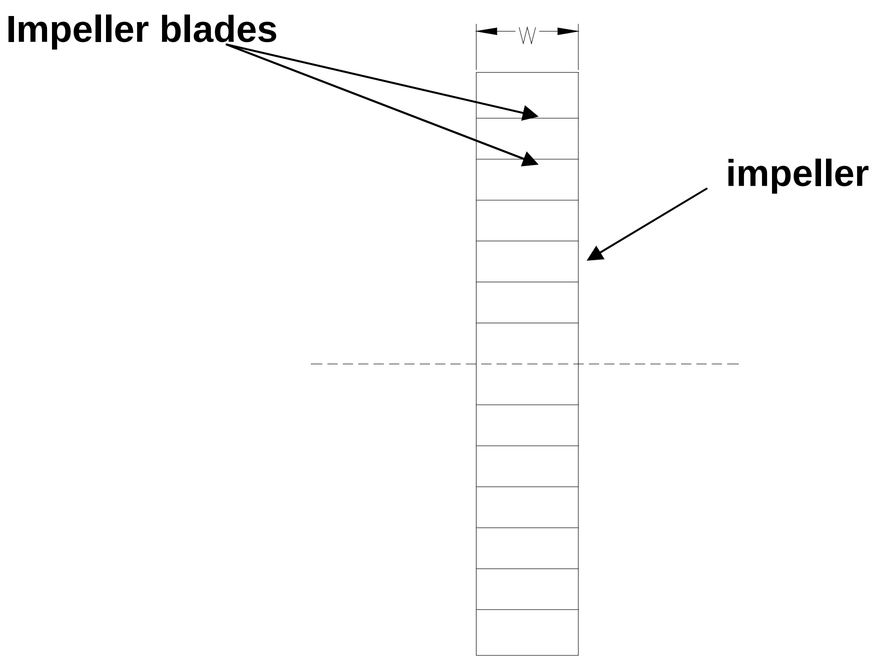


***Fig.21.5**: Impeller of a centrifugal compressor with width w*

21.5. Capacity control:

The capacity of a centrifugal compressor is normally controlled by adjusting inlet guide vanes (pre-rotation vanes). Adjusting the inlet guide vanes provide a swirl at the impeller inlet and thereby introduces a tangential velocity at the inlet to the impeller, which gives rise to different refrigerant flow rates. Figure 21.6 shows the performance of the compressor at different settings of the inlet guide vanes. Use of inlet guide vanes for capacity control is an efficient method as long as the angle of rotation is high, i.e., the vanes are near the fully open condition. When the angle is reduced very much, then this method becomes inefficient as the inlet guide vanes then act as throttling devices.

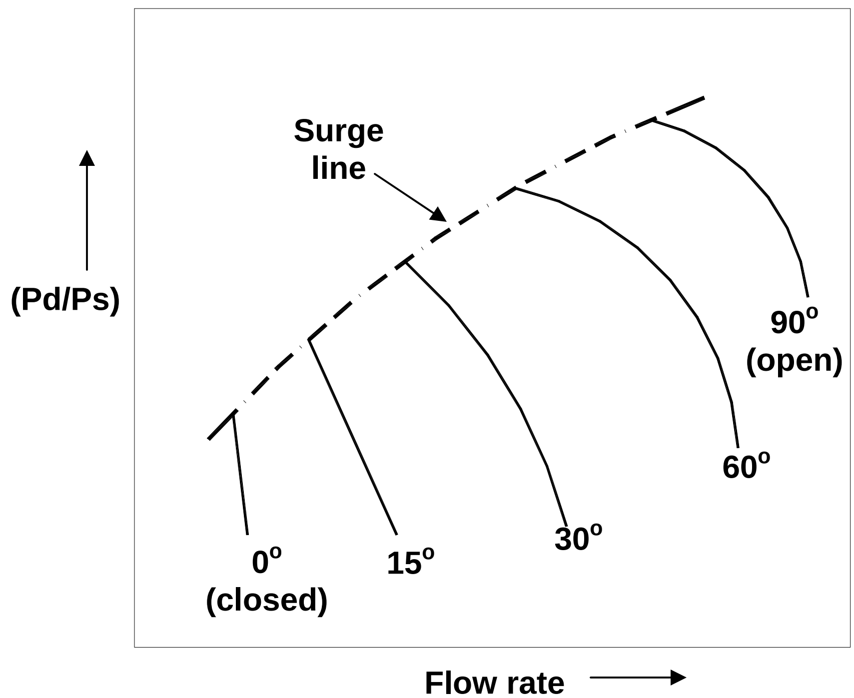


Fig.21.6: *Effect of angle of pre-rotation vanes on capacity of a centrifugal compressor*

In addition to the inlet guide vanes, the capacity control is also possible by adjusting the width of a vaneless diffuser or by adjusting the guide vanes of vaned diffusers. Using a combination of the inlet guide vanes and diffuser, the capacities can be varied from 10 percent to 100 percent of full load capacity.

Capacity can also be controlled by varying the compressor speed using gear drives. For the same pressure rise, operating at lower speeds reduces the flow rate, thereby reducing the refrigeration capacity.

21.6. Performance aspects of centrifugal compressor:

Figure 21.7 shows the pressure-volume characteristics of a centrifugal compressor running at certain speed. As shown in the figure, the relation between pressure and volume is a straight line in the absence of any losses. However, in actual compressors losses occur due to eddy formation in the flow passages, frictional losses and shock losses at the inlet to the impeller. As a result the net head developed reduces as shown in the figure. The entry losses are due to change of direction of refrigerant at the inlet and also due to pre-rotation. These losses can be controlled to some extent using the inlet guide vanes. Due to these losses the net performance curve falls below the ideal characteristic curve without losses, and it also shows an optimum point. The optimum point at which the losses are minimum is selected as the design point for the compressor.

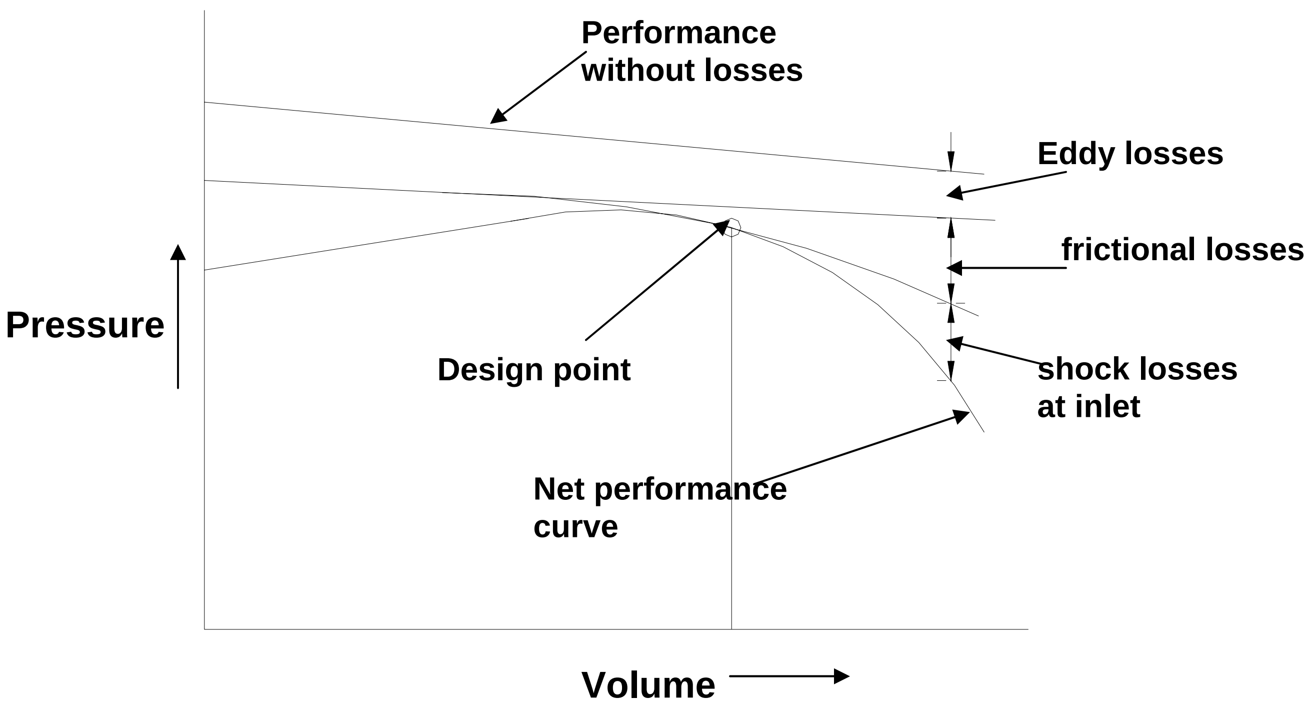


Fig.21.7*: Pressure-volume characteristics of a centrifugal compressor running at certain speed*

Surging:

A centrifugal compressor is designed to operate between a given evaporator and condenser pressures. Due to variations either in the heat sink or refrigerated space, the actual evaporator and condenser pressures can be different from their design values. For example, the condenser pressure may increase if the heat sink temperature increases or the cooling water flow rate reduces. If the resulting pressure difference exceeds the design pressure difference of the compressor, then refrigerant flow reduces and finally stops. Further increase in condenser pressure causes a reverse flow of refrigerant from condenser to evaporator through the compressor. As a result the evaporator pressure increases, the pressure difference reduces and the compressor once again starts pumping the refrigerant in the normal direction. Once the refrigerant starts flowing in the normal direction, the pressure difference increases and again the reversal of flow takes place, as the pressure at the exit of compressor is less than the condenser pressure. This oscillation of refrigerant flow and the resulting rapid variation in pressure difference gives rise to the phenomenon called "*surging*". Surging produces noise and imposes severe stresses on the bearings of the compressor and motor, ultimately leading to their damage. Hence, continuous surging is highly undesirable, even though it may be tolerated if it occurs occasionally. Surging is most likely to occur when the refrigeration load is low (i.e. evaporator pressure is low) and/or the condensing temperature is high. In some centrifugal compressors, surging is taken care of by bypassing a part of the refrigerant from the discharge side to the evaporator, thereby increasing the load artificially. Thus a centrifugal compressor cannot pump the refrigerant when the condensing pressure exceeds a certain value and/or when the evaporator pressure falls below a certain point. This is unlike reciprocating compressors, which continue to pump refrigerant, albeit at lower flow rates when the condenser temperature increases and/or the evaporator pressure falls.

Figures 21.8(a) and (b) show the effect of condensing and evaporating temperatures on the performance of centrifugal compressors and reciprocating compressors. It can be seen from these figures that beyond a certain condenser pressure and below a certain evaporator pressure, the refrigerant capacity of centrifugal compressor decreases rapidly unlike reciprocating compressors where the capacity drop under these conditions is more gradual. However, one advantage with centrifugal compressor is that when operated away from the surge point, the reduction in evaporator temperature with refrigeration load is smaller compared to the reciprocating compressor. This implies that the evaporator temperature of the refrigeration system using a centrifugal compressor remains almost constant over wide variation of refrigeration loads.

Figure 21.9 shows the effect of condensing temperature on power input for both reciprocating as well as centrifugal compressors at a particular evaporator temperature and compressor speed. It can be seen that while the power input increases with condensing temperature for a reciprocating compressor, it decreases with condensing temperature for a centrifugal compressor. This is due to the rapid drop in refrigerant mass flow rate of centrifugal compressor with condensing temperature. This characteristic implies that the problem of compressor overloading at high condensing temperatures does not exist in case of centrifugal compressors.

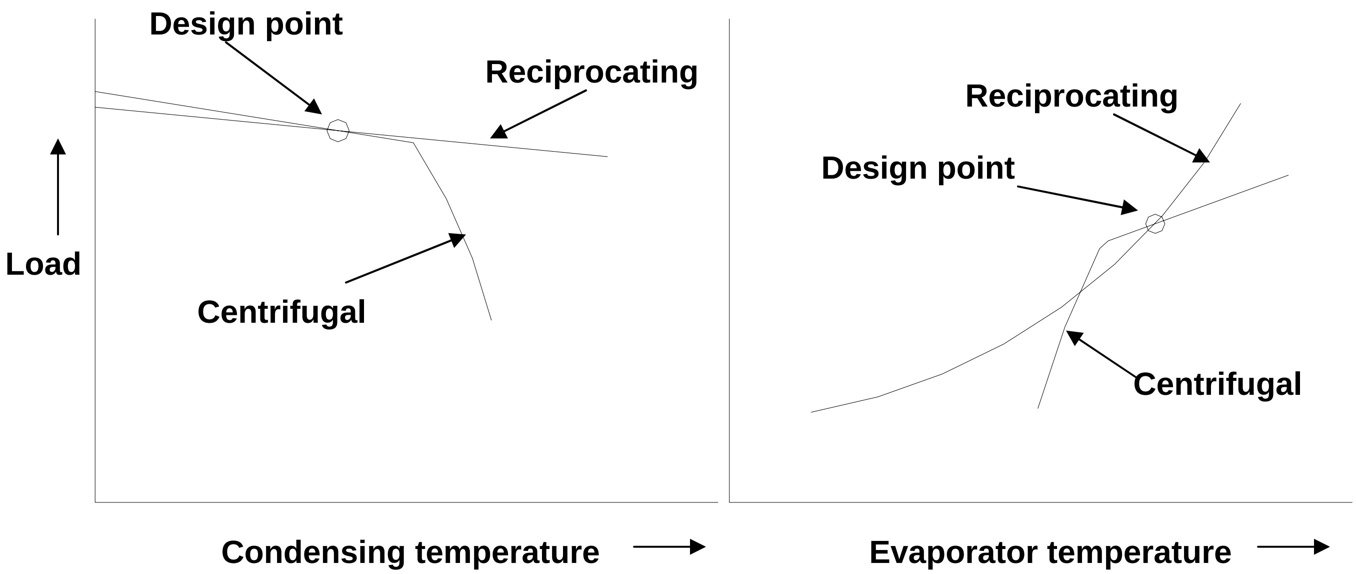


***Fig.21.8(a) and (b)**: Effects of condensing and evaporator temperatures on the performance of reciprocating and centrifugal compressors*

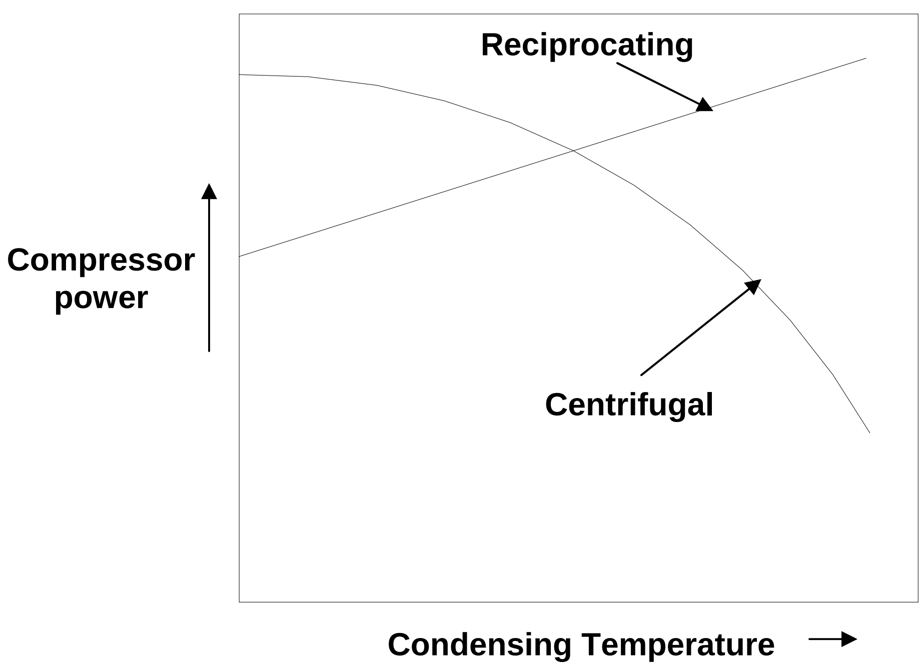


***Fig.21.9**: Effect of condensing temperature on power input for both reciprocating as well as centrifugal compressors at a particular evaporator temperature and compressor speed*

Figure 21.10 shows the effect of compressor speed on the performance of reciprocating and centrifugal compressors. It can be seen from the figure that the performance of centrifugal compressor is more sensitive to compressor speed compared to reciprocating compressors.

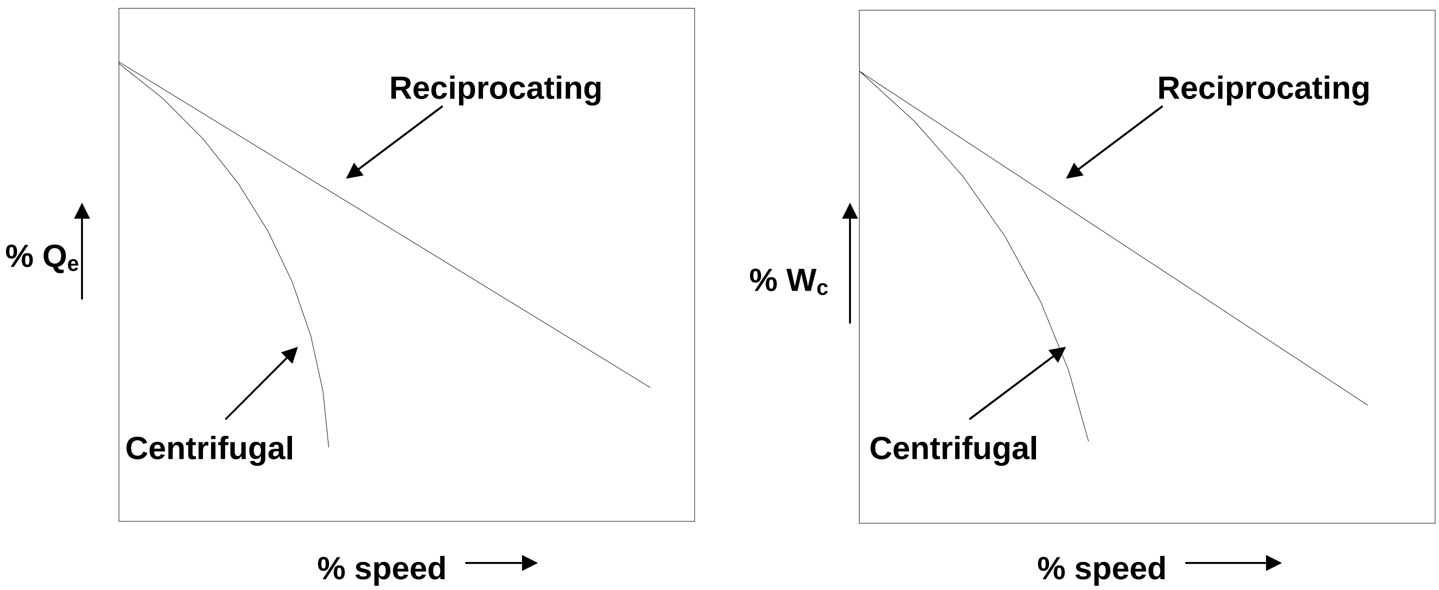


Fig. 21.10*: Effect of compressor speed on the performance of reciprocating and centrifugal compressors at a given condensing and evaporator temperatures*

Figure 21.11 shows the performance characteristics of a centrifugal compressor with backward curved blades. The figure shows the performance at various iso-efficiency values and at different speeds. Such figures are very useful as by using these one can find out, for example the efficiency, flow rate at a given pressure ratio and compressor speed or vice versa. Figure 21.12 shows the sectional view of an actual centrifugal compressor.

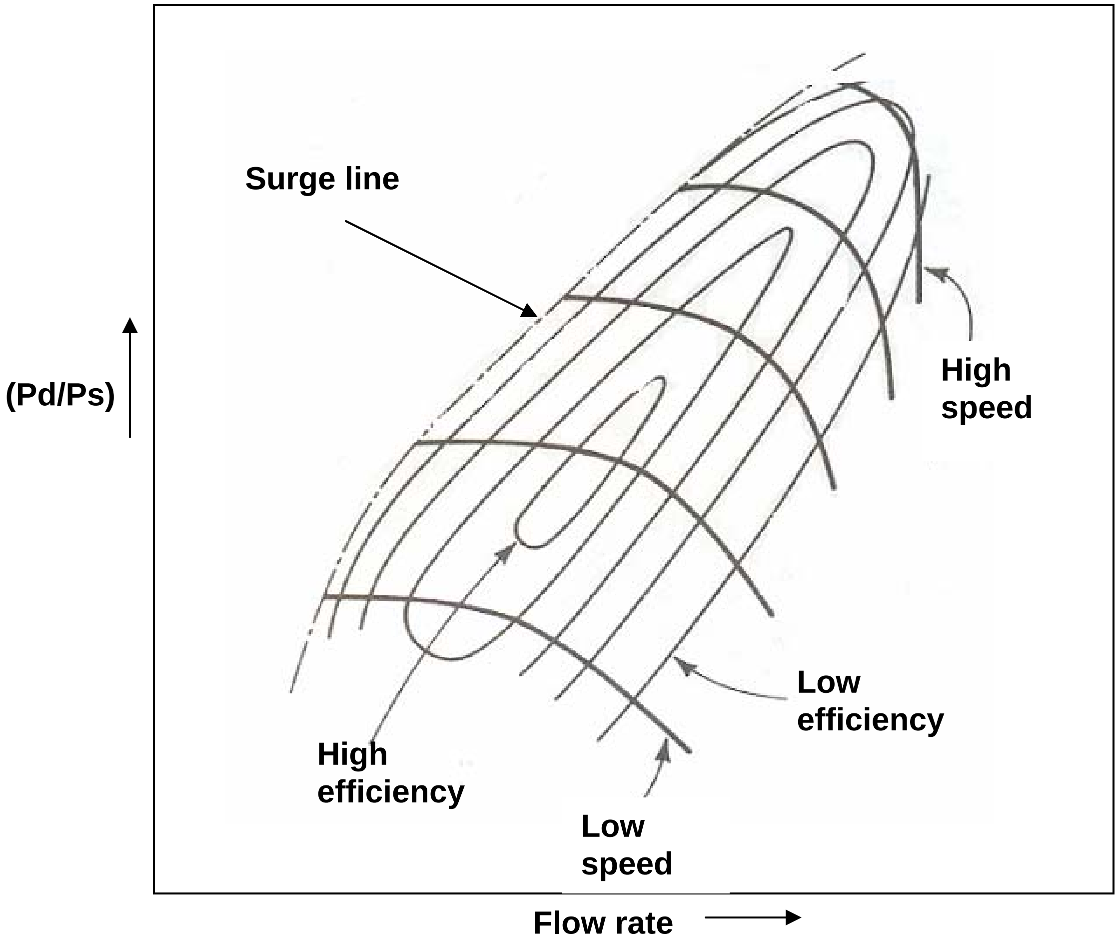


Fig. 21.11: *Performance characteristics of a centrifugal compressor with backward curved blades*

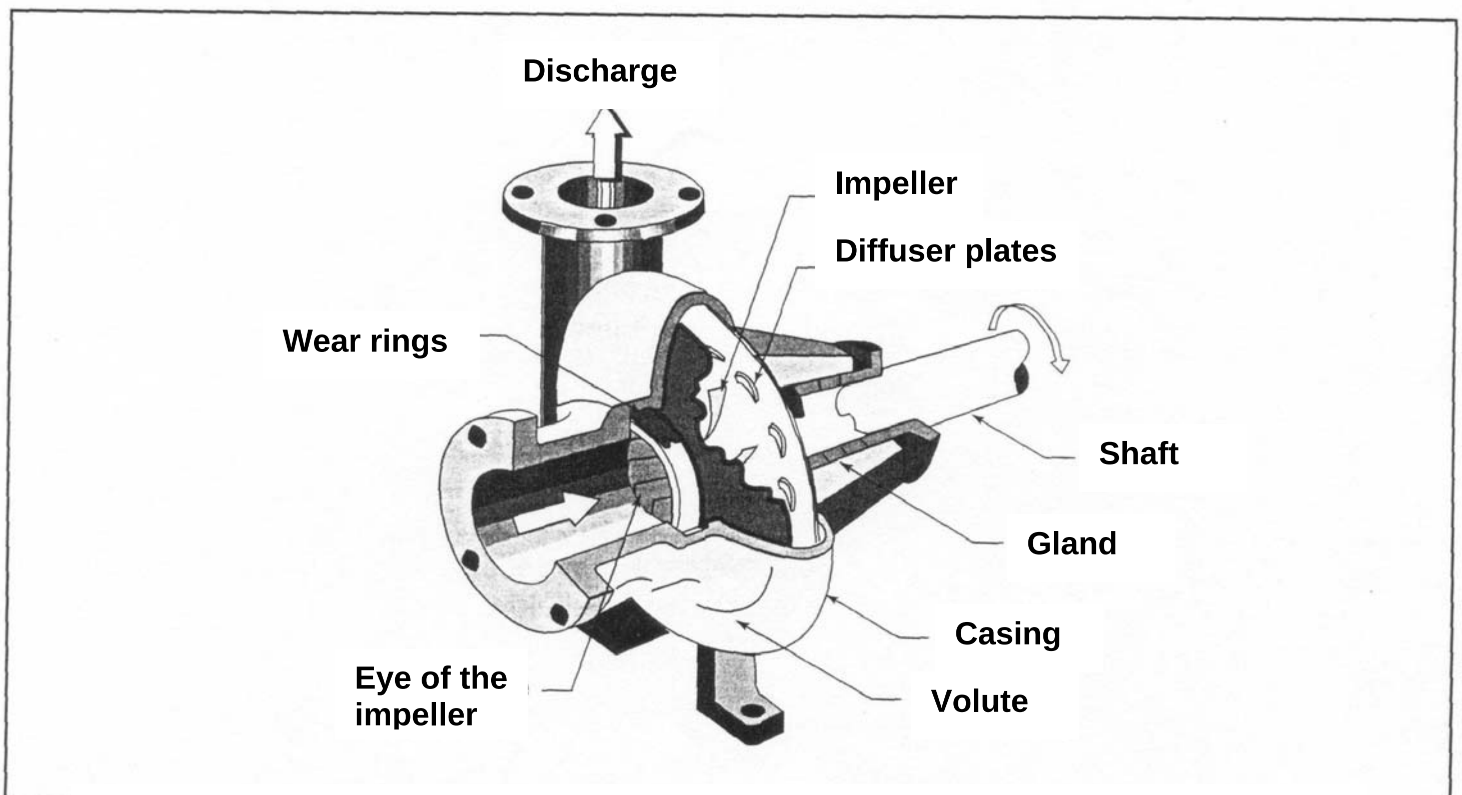


Fig.21.12: *Sectional view of a commercial, single-stage centrifugal compressor*

21.7: Commercial refrigeration systems with centrifugal compressors:

Commercially centrifugal compressors are available for a wide variety of refrigeration and air conditioning applications with a wide variety of refrigerants. These machines are available for the following ranges:

Evaporator temperatures	:	-100^oC to $+10^oC$
Evaporator pressures	:	14 kPa to 700 kPa
Discharge pressure	:	upto 2000 kPa
Rotational speeds	:	1800 to 90,000 RPM
Refrigeration capacity	:	300 kW to 30000 kW

As mentioned before, on the lower side the capacity is limited by the impeller width and tip speeds and on the higher side the capacity is limited by the physical size (currently the maximum impeller diameter is around 2 m).

Since the performance of centrifugal compressor is more sensitive to evaporator and condensing temperatures compared to a reciprocating compressor, it is essential to reduce the pressure drops when a centrifugal compressor is used in commercial systems. Commercial refrigeration systems using centrifugal compressors normally incorporate flash intercoolers to improve the system performance. Since the compressor is normally multi-staged, use of flash intercooler is relatively easy in case of centrifugal compressors.

Centrifugal compressors are normally lubricated using an oil pump (force feed) which can be driven either directly by the compressor rotor or by an external motor. The lubrication system consists of the oil pump, oil reservoir and an oil cooler. The components requiring lubrication are the main bearings, a thrust bearing (for the balancing disc) and the shaft seals. Compared to reciprocating compressors, the lubrication for centrifugal compressors is simplified as very little lubricating oil comes in direct contact with the refrigerant. Normally labyrinth type oil seals are used on the rotor shaft to minimize the leakage of lubricating oil to the refrigerant side. Sometimes oil heaters may be required to avoid excessive dilution of lubricating oil during the plant shutdown.

Commercially both hermetic as well as open type centrifugal compressors are available. Open type compressors are driven by electric motors, internal combustion engines (using a wide variety of fuels) or even steam turbines.

Questions & answers:

1. Which of the following statements concerning centrifugal compressors are true?

a) Centrifugal compressors are subjected to less vibration and noise as they rotate at very high speeds
b) Pressure rise in centrifugal compressor is due to the continuous conversion of angular momentum into static pressure
c) The stagnation enthalpy of refrigerant vapour remains constant everywhere, except across the impeller blades
d) Conversion of dynamic pressure into static pressure takes place in the volute casing due to its convergent shape

Ans.: b) and c)

2. Which of the following statements concerning centrifugal compressors are true?

a) Centrifugal compressors with vaneless diffusers are compact compared to vaned diffusers
b) In multi-stage centrifugal compressors, the width of the blades reduces progressively in the direction of flow
c) In multi-stage centrifugal compressors, the width of the blades increases progressively in the direction of flow
d) Multi-staging in centrifugal compressors is commonly used for high refrigerant capacity applications

Ans.: b)

3. The polytropic efficiency of a centrifugal compressor is found to be 0.85. The isentropic index of compression of the refrigerant, which behaves as an ideal gas, is 1.17. The polytropic index of compression, n is then equal to:

a) 1.206
b) 0.829
c) 0.854
d) 1.141

Ans.: a)

4. Which of the following statements are true:

a) In reciprocating compressors, the irreversibility is mainly due to heat transfer and viscous shear stresses
b) In reciprocating compressors, the irreversibility is mainly due to heat transfer and pressure drops across valves and connecting pipelines
c) In centrifugal compressors, the irreversibility is mainly due to heat transfer and viscous shear stresses
d) In centrifugal compressors, the irreversibility is mainly due to viscous shear stresses

Ans.: b) and d)

5. Which of the following statements are true:

a) Due to slip, the actual pressure rise and volumetric flow rate of a centrifugal compressor is less than that of an ideal compressor
b) For a given impeller diameter, the slip factor decreases as the number of blades increases
c) For a given impeller diameter, the slip factor decreases as the number of blades decreases
d) For a given flow rate, the frictional losses decrease as the number of blades increase

Ans.: a) and c)

6. Which of the following statements are true:

a) The capacity of a centrifugal compressor can be controlled by using inlet guide vanes and by changing the width of the diffuser
b) Surging in centrifugal compressors takes place as evaporator and condenser pressures increase
c) Surging in centrifugal compressors takes place as evaporator pressure increases and condenser pressure decreases
d) Surging in centrifugal compressors takes place as evaporator pressure decreases and condenser pressure increases

Ans.: a) and d)

7. Which of the following statements are true:

a) When operated away from the surge point, the reduction in evaporator temperature with refrigeration load is smaller for centrifugal compressors compared to the reciprocating compressors
b) When operated away from the surge point, the reduction in evaporator temperature with refrigeration load is much larger compared to the reciprocating compressor
c) The problem of compressor motor overloading due to high condenser temperature does not take place in a centrifugal compressor
d) Compared to reciprocating compressor, the performance of centrifugal compressor is less sensitive to speed

Ans.: a) and c)

8. Saturated R134a vapour is compressed isentropically from -18^oC (P_{sat}=144.6 kPa) to a pressure of 433.8 kPa in a single stage centrifugal compressor. Calculate the speed of the compressor at the tip of the impeller assuming that the vapour enters the impeller radially.

Ans.:

From the refrigerant property data, the enthalpy and entropy of ammonia vapour at the inlet to the impeller are 387.8 kJ/kg and 1.740 kJ/kg.K, respectively.

At an exit pressure of 433.8 kPa and an entropy of 1.740 kJ/kg.K (isentropic compression), the exit enthalpy of the vapour is found to be 410.4 kJ/kg.

For radial entry, the velocity of ammonia vapour at the tip of the impeller (u_2) is given by:

$$\mathbf{u_2^2 = (h_{exit}-h_{inlet}) = 410.4-387.8 = 22.6\ kJ/kg = 22600\ J/kg}$$

$$\Rightarrow \mathbf{u_2 = 150.3\ m/s\ (Ans.)}$$

9. A 2-stage centrifugal compressor operating at 3000 RPM is to compress refrigerant R 134a from an evaporator temperature of 0^oC to a condensing temperature of 32^oC. If the impeller diameters of both stages have to be same, what is the diameter of the impeller? Assume the suction condition to be dry saturated, compression process to be isentropic, the impeller blades to be radial and refrigerant enters the impeller axially.

Given:

Refrigerant	= R 134a
Evaporator temperature	= 0^oC
Condensing temperature	= 32^oC
Inlet condition	= Dry saturated
Compression process	= Isentropic (reversible, adiabatic)
Number of stages	= 2
Rotational speed	= 3000 RPM
Impeller blades	= Radial
Tangential velocity at inlet	= 0 m/s
Diameter of impeller	= Same for both stages

Ans.:

From refrigerant property data:

Enthalpy of refrigerant at compressor inlet, h_i = 398.6 kJ/kg
Enthalpy of refrigerant at compressor exit, h_e = 419.8 kJ/kg

Since the blades are radial with no tangential velocity component at inlet, the enthalpy rise across each stage,

$$\mathbf{\Delta h_1 = \Delta h_2 = u_2^2 = \Delta h_{stage}}$$

$\Rightarrow$ enthalpy rise across the compressor, $\mathbf{(h_e\text{-}h_i) = \Delta h_1+\Delta h_2 = 2\Delta h_{stage}}$

$$\mathbf{\Rightarrow \Delta h_{stage} = (h_e\text{-}h_i)/2 = (419.8\text{-}398.6)/2 = 10.6\ kJ/kg}$$

$$\mathbf{\therefore u_2 = (\Delta h_{stage})^{1/2} = (10.6\ X\ 1000)^{1/2} = 103\ m/s}$$
$$\mathbf{u_2 = \omega.r2}$$
$$\mathbf{\omega = 2\pi\ X\ 3000/60 = 100\pi\ rad/s}$$

$$\mathbf{\therefore r_2 = \therefore u_2/\omega = 0.3279\ m \Rightarrow impeller\ diameter = 2r_2 = 0.6558\ m\ \ (Ans.)}$$

10. A backward curved centrifugal compressor is to compress refrigerant R134a. The diameter of the impeller is 0.6 m and the blade angle is 60^o. The peripheral area is 0.002 m^2 and the flow coefficient (ratio of normal component of velocity to tip speed) is 0.5. If the pressure and temperature of refrigerant at the exit of the impeller are found to be 7.702 bar and 40^oC, find the specific work and power input to the compressor. The impeller rotates at 9000 RPM. The tangential component of velocity at the inlet to the impeller may be assumed to be negligible.

Ans.: Given:

Refrigerant	:	R134a
Diameter of impeller	=	0.6 m
Blade angle, β	=	60^o
Peripheral flow area, $A_{f,p}$	=	0.002 m^2
Flow coefficient ($V_{n,2}/u_2$)	=	0.5
Impeller speed	=	9000 RPM
Exit pressure	=	7.702 bar
Exit temperature	=	40^oC

To find: Specific work input (w) and power input (W)

When the tangential component of velocity at the impeller inlet is negligible and the slip factor is unity, then the power input to the compressor is given by:

$$W = mu_2 V_{t,2} = mu_2^2 \left(1 - \frac{V_{n,2} \cot\beta}{u_2}\right)$$

The tip speed, u_2 is obtained from the RPM (N) and the impeller diameter (d) as:

$$u_2 = 2\pi(N/60)(d/2) = 2\pi(9000/60)(0.6/2) = 282.74\,m/s$$

Since the flow coefficient is given as 0.5, the normal component of velocity at the exit of the impeller, $V_{n,2}$ is given by:

$$V_{n,2} = 0.5u_2 = 141.37\,m/s$$

The mass flow rate of refrigerant is obtained from the normal component at the tip ($V_{n,2}$), peripheral area ($A_{f,p}$) and the specific volume of refrigerant at the exit (v_2; obtained from exit pressure and temperature) as:

$$m = \frac{V_{n,2}A_{f,p}}{v_2} = \frac{141.37\,X\,0.002}{0.1846} = 1.532\,kg/s$$

Substituting the values of mass flow rate, tip velocity, normal component of velocity at the impeller exit and the blade angle in the expression for power input, we obtain:

Power input to the compressor, W = 87117 W = 87.117 kW

(Ans.)

Specific work = W/m = 56.865 kJ/kg