Lesson
18
Refrigeration System Components: Compressors

The objectives of this lesson are to:

1. Discuss basic components of a vapour compression refrigeration system (*Section 18.1*)
2. Present classification of refrigerant compressors based on working principle and based on the arrangement of compressor motor or external drive *(Section 18.2.1)*
3. Describe the working principle of reciprocating compressors *(Section 18.3)*
4. Discuss the performance aspects of ideal reciprocating compressors with and without clearance ***(****Section 18.3.1****)***

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

1. List important components of a vapour compression refrigeration system
2. Classify refrigerant compressors based on their working principle and based on the arrangement of compressor motor/external drive
3. Enumerate salient features of positive displacement type compressors, dynamic compressors, open and hermetic compressors
4. Draw the schematic of a reciprocating compressor and explain its working principle
5. Define an ideal reciprocating compressor without clearance using pressure-volume and pressure-crank angle diagrams
6. Calculate the required displacement rate and power input of an ideal compressor without clearance
7. Define an ideal reciprocating compressor with clearance using pressure-volume and pressure-crank angle diagrams
8. Calculate the volumetric efficiency and power input of an ideal compressor with clearance, and
9. Discuss the effects of compression ratio and index of compression on the volumetric efficiency of a reciprocating compressor with clearance

18.1. Introduction

A typical refrigeration system consists of several basic components such as compressors, condensers, expansion devices, evaporators, in addition to several accessories such as controls, filters, driers, oil separators etc. For efficient operation of the refrigeration system, it is essential that there be a proper matching between various components. Before analyzing the balanced performance of the complete system, it is essential to study the design and performance characteristics of individual components. Except in special applications, the refrigeration system components are standard components manufactured by industries specializing in individual components. Generally for large systems, depending upon the design specifications, components are selected from the manufacturers' catalogs and are assembled at site. Even though most of the components are standard off-the-shelf items, sometimes components such as evaporator may be made to order. Small capacity refrigeration systems such as refrigerators, room and package air conditioners, water coolers are available as complete systems. In this case the manufacturer himself designs or selects the system components, assembles them at the factory, tests them for performance and then sells the complete system as a unit.

18.2. Compressors

A compressor is the most important and often the costliest component (typically 30 to 40 percent of total cost) of any vapour compression refrigeration system (VCRS). The function of a compressor in a VCRS is to continuously draw the refrigerant vapour from the evaporator, so that a low pressure and low temperature can be maintained in the evaporator at which the refrigerant can boil extracting heat from the refrigerated space. The compressor then has to raise the pressure of the refrigerant to a level at which it can condense by rejecting heat to the cooling medium in the condenser.

18.2.1. Classification of compressors

Compressors used in refrigeration systems can be classified in several ways:

a) Based on the working principle:

i. Positive displacement type
ii. Roto-dynamic type

In positive displacement type compressors, compression is achieved by trapping a refrigerant vapour into an enclosed space and then reducing its volume. Since a fixed amount of refrigerant is trapped each time, its pressure rises as its volume is reduced. When the pressure rises to a level that is slightly higher than the condensing pressure, then it is expelled from the enclosed space and a fresh charge of low-pressure refrigerant is drawn in and the cycle continues. Since the flow of refrigerant to the compressor is not steady, the positive displacement type compressor is a *pulsating flow device*. However, since the operating speeds are normally very high the flow appears to be almost steady on macroscopic time scale. Since the flow is pulsating on a microscopic time scale, positive displacement type compressors are prone to high wear, vibration and noise level. Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

i. Reciprocating type
ii. Rotary type with sliding vanes (rolling piston type or multiple vane type)
iii. Rotary screw type (single screw or twin-screw type)
iv. Orbital compressors, and
v. Acoustic compressors

In roto-dynamic compressors, the pressure rise of refrigerant is achieved by imparting kinetic energy to a steadily flowing stream of refrigerant by a rotating mechanical element and then converting into pressure as the refrigerant flows through a diverging passage. Unlike positive displacement type, the roto-dynamic type compressors are steady flow devices, hence are subjected to less wear and vibration. Depending upon the construction, roto-dynamic type compressors can be classified into:

i. Radial flow type, or
ii. Axial flow type

Centrifugal compressors (also known as turbo-compressors) are radial flow type, roto-dynamic compressors. These compressors are widely used in large capacity refrigeration and air conditioning systems. Axial flow compressors are normally used in gas liquefaction applications.

b) Based on arrangement of compressor motor or external drive:

i. Open type
ii. Hermetic (or sealed) type
iii. Semi-hermetic (or semi-sealed) type

In open type compressors the rotating shaft of the compressor extends through a seal in the crankcase for an external drive. The external drive may be an electrical motor or an engine (e.g. diesel engine). The compressor may be belt driven or gear driven. Open type compressors are normally used in medium to large capacity refrigeration system for all refrigerants and for ammonia (due to its incompatibility with hermetic motor materials). Open type compressors are characterized by high efficiency, flexibility, better compressor cooling and serviceability. However, since the shaft has to extend through the seal, refrigerant leakage from the system cannot be eliminated completely. Hence refrigeration systems using open type compressors require a refrigerant reservoir to take care of the refrigerant leakage for some time, and then regular maintenance for charging the system with refrigerant, changing of seals, gaskets etc.

In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor. All motors reject a part of the power supplied to it due to eddy currents and friction, that is, inefficiencies. Similarly the compressor also gets heated-up due to friction and also due to temperature rise of the vapor during compression. In Open type, both the compressor and the motor normally reject heat to the surrounding air for efficient operation. In hermetic compressors heat cannot be rejected to the surrounding air since both are enclosed in a shell. Hence, the cold suction gas is made to flow over the motor and the compressor before entering the compressor. This keeps the motor cool. The motor winding is in direct contact with the refrigerant hence only those refrigerants, which have high dielectric strength, can be used in hermetic compressors. The cooling rate depends upon the flow rate of the refrigerant, its temperature and the thermal properties of the refrigerant. If flow rate is not sufficient and/or if the temperature is not low enough the insulation on the winding of the motor can burn out and short-circuiting may occur. Hence, hermetically sealed compressors give satisfactory and safe performance over a very narrow range of design temperature and should not be used for off-design conditions.

The COP of the hermetic compressor based systems is lower than that of the open compressor based systems since a part of the refrigeration effect is lost in cooling the motor and the compressor. However, hermetic compressors are almost universally used in small systems such as domestic refrigerators, water coolers, air conditioners etc, where efficiency is not as important as customer convenience (due to absence of continuous maintenance). In addition to this, the use of hermetic compressors is ideal in systems, which use capillary tubes as expansion devices and are critically charged systems. Hermetic compressors are normally not serviceable. They are not very flexible as it is difficult to vary their speed to control the cooling capacity.

In some (usually larger) hermetic units, the cylinder head is usually removable so that the valves and the piston can be serviced. This type of unit is called a semi-hermetic (or semi-sealed) compressor.

18.3. Reciprocating compressors

Reciprocating compressor is the workhorse of the refrigeration and air conditioning industry. It is the most widely used compressor with cooling capacities ranging from a few Watts to hundreds of kilowatts. Modern day reciprocating compressors are high speed ($\approx$ 3000 to 3600 rpm), single acting, single or multi-cylinder (upto 16 cylinders) type.

Figure 18.1 shows the schematic of a reciprocating compressor. Reciprocating compressors consist of a piston moving back and forth in a cylinder, with suction and discharge valves to achieve suction and compression of the refrigerant vapor. Its construction and working are somewhat similar to a two-stroke engine, as suction and compression of the refrigerant vapor are completed in one revolution of the crank. The suction side of the compressor is connected to the exit of the evaporator, while the discharge side of the compressor is connected to

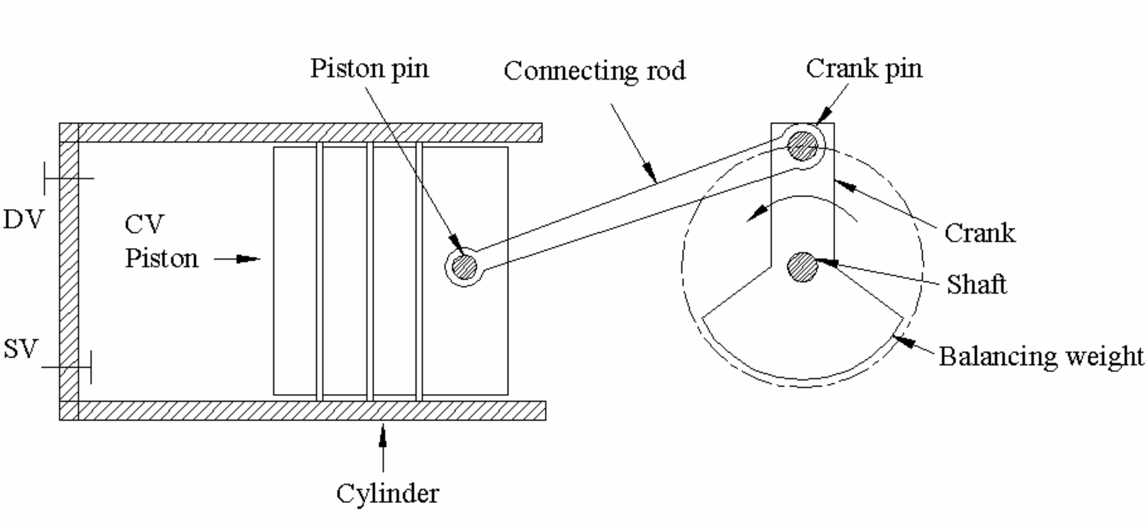

***Fig 18.1**: Schematic of a reciprocating compressor*

the condenser inlet. The suction (inlet) and the discharge (outlet) valves open and close due to pressure differences between the cylinder and inlet or outlet manifolds respectively. The pressure in the inlet manifold is equal to or slightly less than the evaporator pressure. Similarly the pressure in the outlet manifold is equal to or slightly greater than the condenser pressure. The purpose of the manifolds is to provide stable inlet and outlet pressures for the smooth operation of the valves and also provide a space for mounting the valves.

The valves used are of reed or plate type, which are either floating or clamped. Usually, backstops are provided to limit the valve displacement and springs may be provided for smooth return after opening or closing. The piston speed is decided by valve type. Too high a speed will give excessive vapor velocities that will decrease the volumetric efficiency and the throttling loss will decrease the compression efficiency.

18.3.1. Performance of reciprocating compressors

For a given evaporator and condenser pressures, the important performance parameters of a refrigerant compressor are:

a) The mass flow rate (m) of the compressor for a given displacement rate
b) Power consumption of the compressor (W_c)
c) Temperature of the refrigerant at compressor exit, T_d, and
d) Performance under part load conditions

The mass flow rate decides the refrigeration capacity of the system and for a given compressor inlet condition, it depends on the volumetric efficiency of the compressor. The volumetric efficiency, η_V is defined as the ratio of volumetric flow rate of refrigerant to the maximum possible volumetric flow rate, which is equal to the compressor displacement rate, i.e.,

$$\eta_V = \frac{\text{Volumetric flow rate}}{\text{Compressor Displacement rate}} = \frac{\dot{m}.v_e}{\dot{V}_{SW}} \qquad (18.1)$$

where $\dot{m}$ and $\dot{V}_{SW}$ are the mass flow rate of refrigerant (kg/s) and compressor displacement rate (m^3/s) respectively, and v_i is the specific volume (m^3/kg) of the refrigerant at compressor inlet.

For a given evaporator and condenser temperatures, one can also use the volumetric refrigeration capacity (kW/m^3) to indicate the volumetric efficiency of the compressor. The actual volumetric efficiency (or volumetric capacity) of the compressor depends on the operating conditions and the design of the compressor.

The power consumption (kW) or alternately the power input per unit refrigeration capacity (kW/kW) depends on the compressor efficiency (η_C), efficiency of the mechanical drive (η_{mech}) and the motor efficiency (η_{motor}). For a refrigerant compressor, the power input (W_c) is given by:

$$W_C = \frac{W_{ideal}}{\eta_C \eta_{mech} \eta_{motor}} \qquad (18.2)$$

where W_{ideal} is the power input to an ideal compressor.

The temperature at the exit of the compressor (discharge compressor) depends on the type of refrigerant used and the type of compressor cooling. This parameter has a bearing on the life of the compressor.

The performance of the compressor under part load conditions depends on the type and design of the compressor.

<u>a) Ideal reciprocating compressor:</u>

An ideal reciprocating compressor is one in which:

i. The clearance volume is zero, i.e., at the end of discharge process, the volume of refrigerant inside the cylinder is zero.
ii. No pressure drops during suction and compression
iii. Suction, compression and discharge are reversible and adiabatic

Figure 180.2 shows the schematic of an ideal compression process on pressure-volume and pressure-crank angle (θ) diagrams. As shown in the figures, the cycle of operations consists of:

Process D-A: This is an isobaric suction process, during which the piston moves from the Inner Dead Centre (IDC) to the Outer Dead Centre (ODC). The suction valve remains open during this process and refrigerant at a constant pressure P_e flows into the cylinder.
Process A-B: This is an isentropic compression process. During this process, the piston moves from ODC towards IDC. Both the suction and discharge valves remain closed during the process and the pressure of refrigerant increases from P_e to P_c.
Process B-C: This is an isobaric discharge process. During this process, the suction valve remains closed and the discharge valve opens. Refrigerant at a constant P_c is expelled from the compressor as the piston moves to IDC.

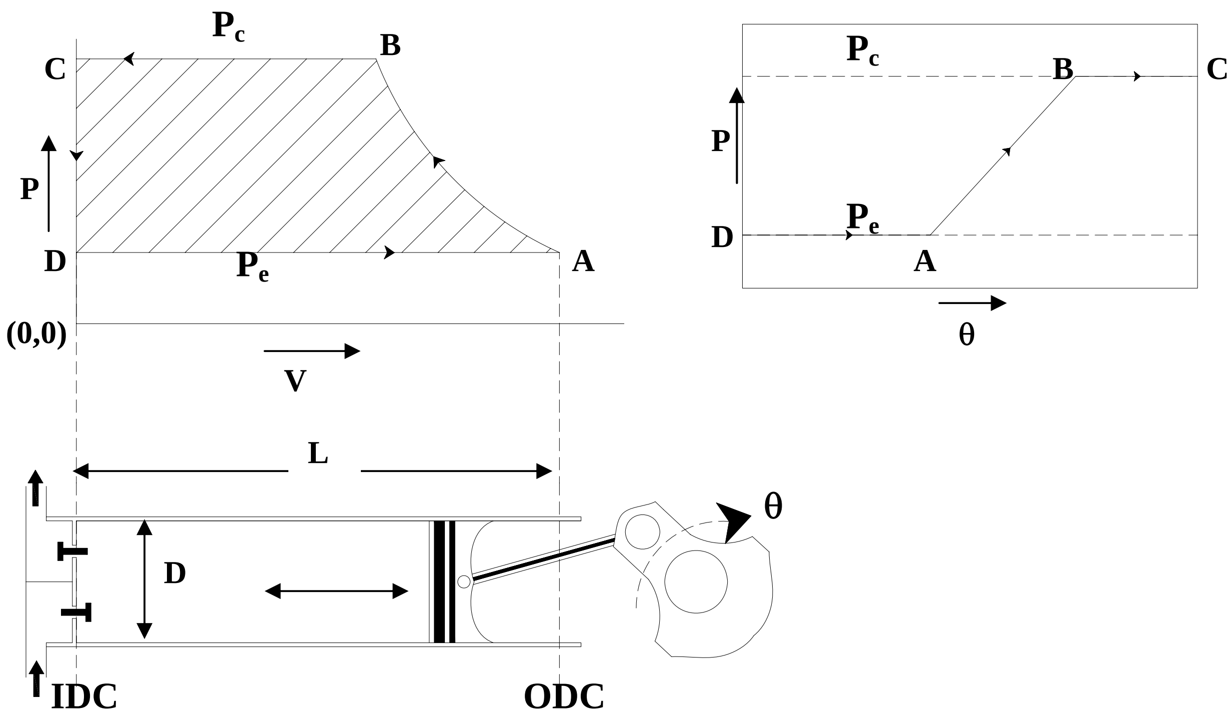

***Fig.18.2**. Ideal reciprocating compressor on P-V and P-θ diagrams*

Since the clearance volume is zero for an ideal compressor, no gas is left in the compressor at the end of the discharge stroke, as a result the suction process D-A starts as soon as the piston starts moving again towards ODC. The volumetric flow rate of refrigerant at suction conditions is equal to the compressor displacement rate hence, the volumetric efficiency of the ideal compressor is 100 percent. The mass flow rate of refrigerant of an ideal compressor is given by:

$$\dot{m} = \frac{\dot{V}_{SW}}{v_e} \qquad (18.3)$$

Thus for a given refrigeration capacity, the required size of the compressor will be minimum if the compressor behaves as an ideal compressor.

The swept volume $\dot{V}_{SW}$ of the compressor is given by:

$$\dot{V}_{SW} = nN\frac{\pi D^2}{4}L \qquad (18.4)$$

where n = Number of cylinders
N = Rotational speed of compressor, revolutions per second
D = Bore of the cylinder, m
L = Stroke length, m

<u>Work input to the ideal compressor:</u>

The total work input to the compressor in one cycle is given by:

$$W_{id} = W_{D-A} + W_{A-B} + W_{B-C} \qquad (18.5)$$

Where,
W_{D-A} = Work done by the refrigerant on the piston during process D-A
= Area under line D-A on P-V diagram = $-P_e.V_A$
W_{A-B} = Work done by the piston on refrigerant during compression A-B
= Area under the curve A-B on P-V diagram = $\int_{V_A}^{V_B} P.dV$
W_{B-C} = Work done by the piston on the refrigerant during discharge B-C
= Area under line B-C = $P_c.V_B$

$\therefore W_{id} = -P_e.V_A + \int_{V_A}^{V_B} P.dV + P_cV_B$ = Area A-B-C-D on P-V diagram = $\int_{Pe}^{Pc} V.dP$

Thus the work input to the ideal compressor per cycle is equal to the area of the cycle on P-V diagram.

The specific work input, w_{id} (kJ/kg) to the ideal compressor is given by:

$$w_{id} = \frac{W_{id}}{M_r} = \int_{Pe}^{Pc} v.dP \qquad (18.6)$$

where M_r is the mass of refrigerant compressed in one cycle and v is the specific volume of the refrigerant.

The power input to the compressor W_c is given by:

$$W_c = \dot{m}\,w_{id} = \frac{\dot{V}_{SW}}{v_e}\int_{Pe}^{Pc} v.dP \qquad (18.7)$$

The mean effective pressure (mep) for the ideal compressor is given by:

$$mep = \frac{W_{id}}{\dot{V}_{SW}} = \frac{1}{v_e}\int_{Pe}^{Pc} v.dP \tag{18.8}$$

The concept of *mean effective pressure* is useful for real compressors as the power input to the compressor is a product of mep and the swept volume rate.

Thus the power input to the compressor and its mean effective pressure can be obtained from the above equation if the relation between v and P during the compression process A-B is known. The above equation is valid for both isentropic and non-isentropic compression processes, however, the compression process must be reversible, as the path of the process should be known for the integration to be performed.

For the isentropic process, Pv^k = constant, hence the specific work of compression w_{id} can be obtained by integration, and it can be shown to be equal to:

$$w_{id} = \int_{Pe}^{Pc} v.dP = P_e v_e \left(\frac{k}{k-1}\right)\left[\left(\frac{Pc}{Pe}\right)^{\frac{k-1}{k}} - 1\right] \tag{18.9}$$

In the above equation, k is the index of isentropic compression. If the refrigerant behaves as an ideal gas, then k = γ. In general, the value of k for refrigerants varies from point to point, and if its value is not known, then an approximate value of it can be obtained from the values of pressure and specific volume at the suction and discharge states as $k \approx \frac{\ln(P_c/P_e)}{\ln(v_e/v_c)}$.

The work of compression for the ideal compressor can also be obtained by applying energy balance across the compressor, Fig.18.3. Since the process is assumed to be reversible and adiabatic and if we assume changes in potential and kinetic energy to be negligible, then from energy balance across the compressor:

$$w_{id} = \frac{W_c}{\dot{m}} = (h_c - h_e) \tag{18.10}$$

The above expression can also be obtained from the thermodynamic relation:

$$Tds = dh - vdP \Rightarrow dh = vdP \ (\because ds = 0 \text{ for isentropic process})$$
$$\therefore w_{id} = \int_{Pe}^{Pc} vdP = \int_{Pe}^{Pc} dh = (h_d - h_e) \tag{18.11}$$

The above expression is valid only for reversible, adiabatic compression.

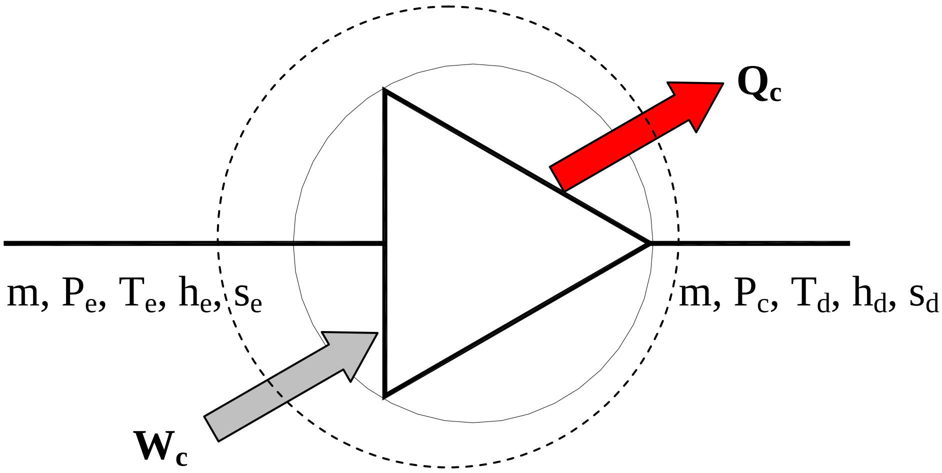

***Fig.18.3**. Energy balance across a steady flow compressor*

b) Ideal compressor with clearance:

In actual compressors, a small clearance is left between the cylinder head and piston to accommodate the valves and to take care of thermal expansion and machining tolerances. As a thumb rule, the clearance C in millimetres is given by:

$$C = (0.005L + 0.5) \text{ mm, where L is stroke length in mm} \tag{18.12}$$

This space along with all other spaces between the closed valves and the piston at the inner dead center (IDC) is called as Clearance volume, Vc. The ratio of the clearance volume to the swept volume is called as Clearance ratio, ε, i.e.,

$$\varepsilon = \frac{V_c}{V_{sw}} \tag{18.13}$$

The clearance ratio ε depends on the arrangement of the valves in the cylinder and the mean piston velocity. Normally ε is less than 5 percent for well designed compressors with moderate piston velocities ($\approx$ 3 m/s), however, it can be higher for higher piston speeds.

Due to the presence of the clearance volume, at the end of the discharge stroke, some amount of refrigerant at the discharge pressure Pc will be left in the clearance volume. As a result, suction does not begin as soon as the piston starts moving away from the IDC, since the pressure inside the cylinder is higher than the suction pressure ($P_c > P_e$). As shown in Fig. 18.4, suction starts only when the pressure inside the cylinder falls to the suction pressure in an ideal compressor with clearance. This implies that even though the compressor swept volume, $V_{SW} = V_A - V_C$, the actual volume of the refrigerant that entered the cylinder during suction stroke is $V_A - V_D$. As a result, the volumetric efficiency of the compressor with clearance, $\eta_{V,cl}$ is less than 100 percent, i.e.,

$$\eta_{V,cl} = \frac{\text{Actual volume of refrigerant compressed}}{\text{Swept volume of the compressor}} = \left(\frac{V_A - V_D}{V_A - V_C}\right) \qquad (18.14)$$

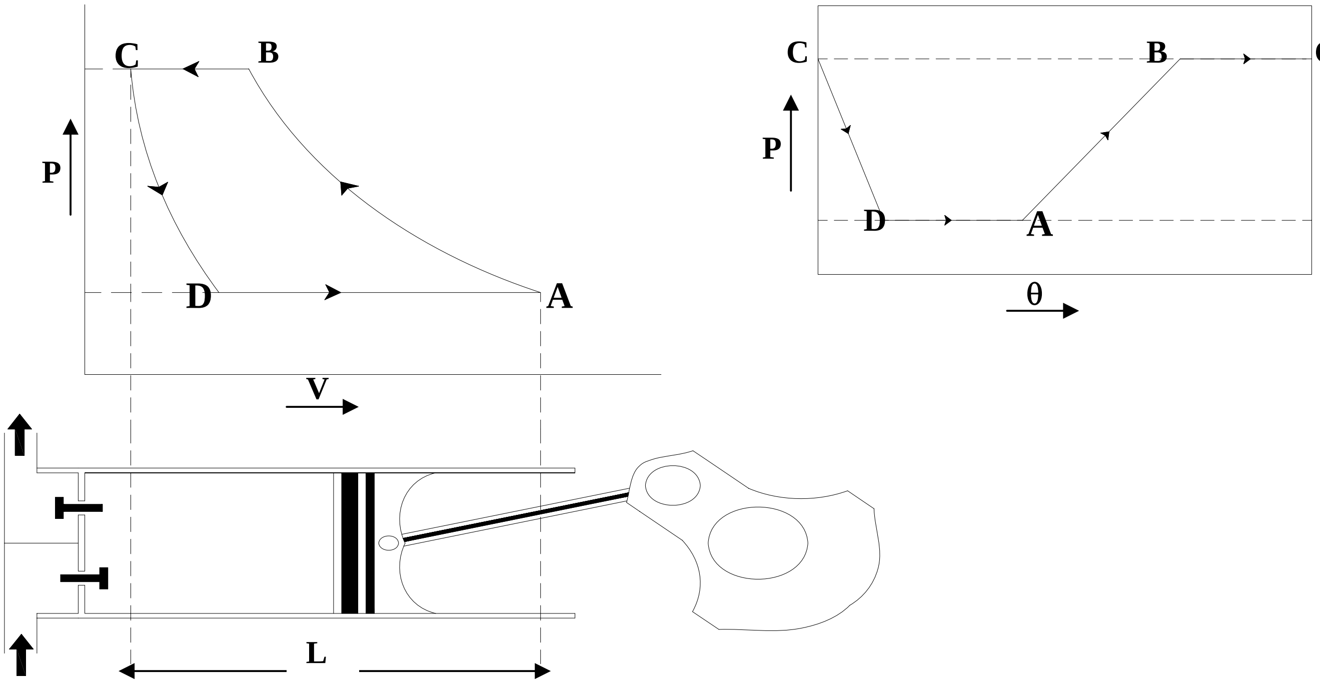

***Fig.18.4**. Ideal reciprocating compressor with clearance*

From Fig.18.4, the clearance volumetric efficiency can be written as:

$$\eta_{V,cl} = \left(\frac{V_A - V_D}{V_A - V_C}\right) = \frac{(V_A - V_C) + (V_C - V_D)}{(V_A - V_C)} = 1 + \left(\frac{(V_C - V_D)}{(V_A - V_C)}\right) \qquad (18.15)$$

Since the clearance ratio, $\varepsilon = \frac{Vc}{V_{SW}} = \frac{V_C}{V_A - V_C} \Rightarrow (V_A - V_C) = \frac{V_C}{\varepsilon}$ (18.16)

Substituting the above equation in the expression for clearance volumetric efficiency; we can show that:

$$\eta_{V,cl} = 1 + \left(\frac{(V_C - V_D)}{(V_A - V_C)}\right) = 1 + \frac{\varepsilon(V_C - V_D)}{V_C} = 1 + \varepsilon - \varepsilon\left(\frac{V_D}{V_C}\right) \qquad (18.17)$$

Since the mass of refrigerant in the cylinder at points C and D are same, we can express the ratio of cylinder volumes at points D and C in terms of ratio of specific volumes of refrigerant at D and C, i.e.,

$$\left(\frac{V_D}{V_C}\right)=\left(\frac{v_D}{v_C}\right) \quad (18.18)$$

Hence, the clearance volumetric efficiency is given by:

$$\eta_{V,cl} = 1+\varepsilon-\varepsilon\left(\frac{V_D}{V_C}\right)=1+\varepsilon-\varepsilon\left(\frac{v_D}{v_C}\right) \quad (18.19)$$

If we assume the re-expansion process also to follow the equation Pv^k=constant, then:

$$\left(\frac{v_D}{v_C}\right)=\left(\frac{P_C}{P_D}\right)^{1/k}=\left(\frac{P_c}{P_e}\right)^{1/k} \quad (18.20)$$

Hence the clearance volumetric efficiency is given by:

$$\eta_{V,cl} = 1+\varepsilon-\varepsilon\left(\frac{P_c}{P_e}\right)^{1/k}=1-\varepsilon\left[r_p^{\ 1/k}-1\right] \quad (18.21)$$

where r_p is the pressure ratio, P_c/P_e.

The above expression holds good for any reversible compression process with clearance. If the process is not reversible, adiabatic (i.e., non-isentropic) but a reversible polytropic process with an index of compression and expansion equal to n, then k in the above equation has to be replaced by n, i.e., in general for any reversible compression process;

$$\eta_{V,cl} = 1+\varepsilon-\varepsilon\left(\frac{P_c}{P_e}\right)^{1/n}=1-\varepsilon\left[r_p^{\ 1/n}-1\right] \quad (18.22)$$

The above expression shows that $\eta_{V,cl}\downarrow$ as $r_p\uparrow$ and $\varepsilon\uparrow$ as shown in Fig.18.5. It can also be seen that for a given compressor with fixed clearance ratio ε, there is a limiting pressure ratio at which the clearance volumetric efficiency becomes zero. This limiting pressure ratio is obtained from the equation:

$$\eta_{V,cl} = 1-\varepsilon\left[r_p^{\ 1/n}-1\right]=0$$
$$\Rightarrow r_{p,max}=\left[\frac{1+\varepsilon}{\varepsilon}\right]^n \quad (18.23)$$

The mass flow rate of refrigerant compressed with clearance $\dot{m}_{cl}$ is given by:

$$\dot{m}_{cl} = \eta_{V,cl} \frac{\dot{V}_{SW}}{v_e} \qquad (18.24)$$

Thus the mass flow rate and hence the refrigeration capacity of the system decreases as the volumetric efficiency reduces, in other words, the required size of the compressor increases as the volumetric efficiency decreases.

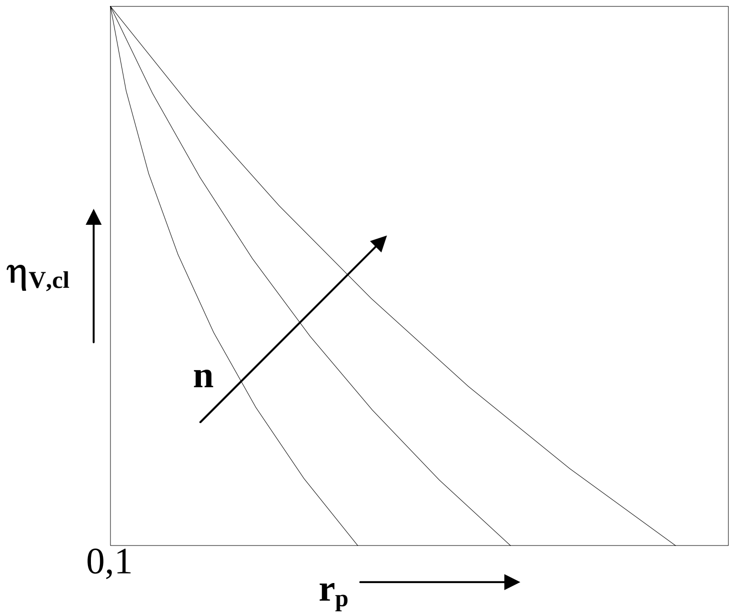

***Fig.18.5**. Effect of pressure ratio (r_p) and index of compression (n) on clearance volumetric efficiency ($\eta_{V,cl}$)*

<u>Work input to the compressor with clearance:</u>

If we assume that both compression and expansion follow the same equation Pv^n = constant (i.e., the index of compression is equal to the index of expansion), then the extra work required to compress the vapour that is left in the clearance volume will be exactly equal to the work output obtained during the re-expansion process. Hence, the clearance for this special case does not impose any penalty on work input to the compressor. The total work input to the compressor during one cycle will then be equal to the area A-B-C-D-A on P-V diagram.

The specific work with and without clearance will be given by the same expression:

$$w_{id} = \int_{Pe}^{Pc} v.dP = P_e v_e \left(\frac{n}{n-1}\right)\left[\left(\frac{Pc}{Pe}\right)^{\frac{n-1}{n}} - 1\right] \qquad (18.25)$$

However, since the mass of refrigerant compressed during one cycle is different with and without clearance, the power input to the compressor will be different with and without clearance. The power input to the compressor and mean effective pressure (mep) with clearance are given by:

$$W_c = \dot{m}\, w_{id} = \left(\eta_{V,cl}\frac{\dot{V}_{SW}}{v_e}\right) w_{id} \qquad (18.26)$$

$$\text{mep} = \eta_{V,cl} \frac{w_{id}}{v_e} \tag{18.27}$$

Thus the power input to the compressor and mep decrease with clearance due to decrease in mass flow rate with clearance.

If the process is reversible and adiabatic (i.e., n = k), then the power input to the compressor with clearance is given by:

$$W_c = = \left(\eta_{V,cl} \frac{\dot{V}_{SW}}{v_e} \right) (h_B - h_A) = \left(\eta_{V,cl} \frac{\dot{V}_{SW}}{v_e} \right) \Delta h_{c,s} \tag{18.28}$$

where $\Delta h_{c,s}$ is the isentropic work of compression (kJ/kg)

Questions and answers:

1. Which of the following is not positive displacement type compressor?

a. Rotary vane compressor
b. Rotary screw type compressor
c. Centrifugal compressor
d. Acoustic compressor

Ans.: c)

2. Compared to a hermetic compressor, an open type compressor:

a. Offers higher efficiency
b. Offers lower noise
c. Offers better compressor cooling
d. Offers serviceability and flexibility

Ans.: a), c) and d)

3. Hermetic compressors are used mainly in smaller systems as they:

a. Yield higher COP
b. Do not require frequent servicing
c. Offer the flexibility of using any refrigerant
d. Can be used under different load conditions efficiently

Ans.: b)

4. In reciprocating compressors, clearance is provided:

a. To improve the volumetric efficiency of the compressor
b. To accommodate valves
c. To account for thermal expansion due to temperature variation
d. To reduce power consumption of the compressor

Ans.: b) and c)

5. The clearance volumetric efficiency of a reciprocating compressor depends on:

a. Properties of the refrigerant
b. Operating temperatures
c. Clearance volume
d. All of the above

Ans.: d)

6. A spacer is used in reciprocating compressors to introduce clearance volume. A refrigerant manufacturer wishes to standardize the components of a reciprocating compressor for refrigeration systems of capacities of 2 kW and 2.5 kW by varying only the spacer. Both the systems use the same refrigerant, which has an isentropic index of compression of 1.116 and operate over a pressure ratio of 5. The operating temperatures are also same for both the systems. If the required clearance factor for the 2.5 kW system is 0.03, what should be the clearance factor for the 2.0 kW system?

Ans.: Given:

Pressure ratio, r_p = 5 and index of compression γ = 1.116 for both the compressors. The clearance factor for the 2.5 kW compressor $\varepsilon_{2.5}$ = 0.03

When all other parameters are same except the capacity, then:

$$(Q_{e,2.5}/Q_{e,2.0}) = 2.5/2.0 = 1.25 = (m_{r,2.5}/m_{r,2.0}) = (\eta_{v,2.5}/\eta_{v,2.0})$$

where Q_e is the refrigeration capacity, m_r is the refrigerant mass flow rate and η_v is the clearance volumetric efficiency of the compressor.

Substituting the expression for volumetric efficiency;

$$\frac{\eta_{V,2.5}}{\eta_{V,2.0}} = \; = \frac{1-\varepsilon_{2.5}\left(r_p^{1/\gamma}-1\right)}{1-\varepsilon_{2.0}\left(r_p^{1/\gamma}-1\right)} = 1.25$$

substituting the values of pressure ratio, index of compression and the clearance factor of 2.5 kW compressor in the above expression, we obtain:

$$\varepsilon_{2.0} = 0.086 \quad \textbf{(Ans.)}$$

7. Water is used in a Standard Single Stage (SSS) vapour compression refrigeration system. The system operates at an evaporator temperature of 4.5^{o}C (pressure = 0.8424 kPa) and a condenser temperature of 38^{o}C (pressure = 6.624 kPa). Assume that the water vapour behaves as an ideal gas with c_p/c_v = 1.322 and calculate the discharge temperature if compression is isentropic. Also calculate COP and volumic refrigeration effect if the refrigeration effect is 2355 kJ/kg. Molecular weight of water = 18 kg/kmol, Universal gas constant = 8.314 kJ/kmol.K

Ans.: Given:

Evaporator temperature, T_e = 4.5^{o}C = 277.5 K
Evaporator pressure, P_e = 0.8424 kPa
Condenser temperature, T_e = 38^{o}C = 311 K
Condenser pressure, P_c = 6.624 kPa
Isentropic index of compression, $\gamma = c_p/c_v$ = 1.322
Refrigeration effect, q_e = 2355 kJ/kg

Gas constant, R = 8.314/18 = 0.462 kJ/kg.K

Specific volume of refrigerant at compressor inlet, $\mathbf{v_e} = \left(\frac{\mathbf{RT_e}}{\mathbf{P_e}}\right) = \mathbf{152.19\,m^3/kg}$

a) Discharge temperature, T_d:

$$\mathbf{T_d = T_e\left(\frac{P_c}{P_e}\right)^{\frac{\gamma-1}{\gamma}} = 458.6\,K}$$

b) Work of compression, w_c:

$$\mathbf{w_c = RT_e\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_c}{P_e}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 343.45\ kJ/kg}$$

c) COP:

$$\mathbf{COP = \frac{q_e}{w_c} = 6.86}$$

d) Volumic refrigeration effect, q_v:

$$\mathbf{q_v = \left(\frac{q_e}{v}\right) = 15.4\ kJ/m^3}$$

8. An ammonia based refrigeration system with a refrigeration capacity of 100TR (1TR=3.5167 kW) operates at an evaporating temperature of -36^oC (saturation pressure = 0.8845 bar) and a condensing temperature of 30^oC (saturation pressure = 11.67 bar). Assume the system to operate on a single stage saturated (SSS) cycle. The compression process may be assumed to be isentropic. Under these conditions, the following property data are available:

Enthalpy of saturated vapour at the exit of evaporator, h_1 = 1414 kJ/kg
Enthalpy of saturated liquid at the exit of condenser,h_4 = 341.8 kJ/kg
Isentropic index of compression, γ = 1.304
The compressor is an 8-cylinder, reciprocating type with a clearance ratio of 0.05 and speed of 1750 RPM. The stroke-to-bore ratio is 0.8. In the absence of superheat data, the refrigerant vapour may be assumed to behave as a perfect gas. The molecular weight of ammonia is 17.03 kg/kmol. Fnd:

a) Power input to the compressor
b) COP and cycle (second law) efficiency
c) Compressor discharge temperature, and
d) Compressor dimensions (diameter and stroke length)

Ans.: Given:

Refrigeration capacity, Q_e = 100 TR = 351.67 kW
Evaporator temperature, $T_e = -36^oC$ = 237 K
Evaporator pressure, P_e = 0.8845 bar = 88.45 kPa
Condenser temperature, $T_e = -36^oC$ = 237 K
Condenser pressure, P_c = 11.67 bar = 1167 kPa
Molecular weight , M = 17.04 kg/kmol
Gas constant, R = 8.314/17.04 = 0.4882 kJ/kg.K
Speed of compressor, N = 1750 RPM
Clearance factor, ε = 0.05
No. of cylinders, n = 8
Stroke-to-bore (L/D) ratio,θ = 0.8

a) Power input to compressor, W_c:

$$\mathbf{W_c = m_r . w_c}$$

where the mass flow rate m_r is given by:

$$\mathbf{m_r = \left(\frac{Q_e}{h_1 - h_4}\right) = 0.328\,kg/s}$$

work of compression, w_c is given by:

$$w_c = RT_e\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_c}{P_e}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 409.6\ kJ/kg$$

Substituting these values, we find that the power input to the compressor is given by:

$$W_c = 134.35\ kW$$

b) COP and second law efficiency

$$COP = \frac{Q_e}{W_c} = 2.618$$

Second law efficiency, η_{II}:

$$\eta_{II} = \frac{COP}{COP_{Carnot}} = COP\left(\frac{T_c - T_e}{T_e}\right) = 0.729$$

c) Discharge temperature, T_d:

$$T_d = T_e\left(\frac{P_c}{P_e}\right)^{\frac{\gamma-1}{\gamma}} = 432.7\,K$$

d) Compressor dimensions, L and D

Swept volume, V_{sw} is given by:

$$V_{sw} = \frac{\pi}{4}D^2 L.N.n = \frac{\pi}{4}D^3.\theta.N.n = \frac{V_e}{\eta_v}$$

The volumetric efficiency η_v is given by:

$$\eta_v = 1 - \varepsilon\left[\left(\frac{P_c}{P_e}\right)^{\frac{1}{\gamma}} - 1\right] = 0.6885$$

The actual volumetric flow rate of refrigerant at compressor inlet, V_e is given by:

$$V_e = m_r.v_e = m_r.\frac{RT_e}{P_e} = 0.4293\,m^3/s$$

Substituting these values in the expression for swept volume V_{sw}, we obtain:

V_{sw} = 0.6235 m^3/s, and

D = 0.162 m and L = 0.8D = 0.1296 m (ans.)