

# Refrigeration System Components: Compressors

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**The objectives of this lesson are to:**

1. Basic components of a vapour compression refrigeration system.
2. Present classification of refrigerant compressors based on working principle and based on the arrangement of compressor motor or external drive.
3. Working principle of reciprocating compressors.
4. Performance aspects of ideal reciprocating compressors with and without clearance.

## **1.1. 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,

## **1.2. 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.

### 1.2.1. 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 encased space and then reducing its volume. Since an altered amount of refrigerant is trapped each time, its pressure rises as its volume is lessened. At the point when the pressure rises to a level that is marginally higher than the condensing pressure, then it is removed from the encased space and a new charge of low-pressure refrigerant is drawn in and the cycle continues. Since the stream of refrigerant to the compressor is not steady, the positive displacement type compressor is a pulsating flow device.

However, since the operating rates are normally high the stream appears to be almost steady on macroscopic time scale. Since the stream is pulsating on a tiny time scale, positive displacement type compressors are inclined to high wear, vibration and commotion level. Depending upon the development, positive displacement type compressors utilized as a part of 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
- 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 utilized as a part of medium to large capacity refrigeration system for all refrigerants and for ammonia (because of its incompatibility with hermetic motor materials). Open type compressors are characterized by high efficiency, flexibility, better compressor cooling and serviceability. Then again, since the shaft has to extend through the seal, refrigerant leakage from the system cannot be eliminated totally.

Hence refrigeration systems using open type compressors oblige a refrigerant repository to take care of the refrigerant leakage for eventually, and then regular maintenance for charging the system with refrigerant, changing of seals, gaskets and so forth. In hermetic compressors, the motor and the compressor are encased in the same housing to anticipate refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for force input attachment. As a consequence of this, there is virtually no plausibility of refrigerant leakage from the compressor. All motors dismiss a part of the force supplied to it because of whirlpool streams and grating, that is, inefficiencies.

Similarly the compressor also gets heated-up because of grinding and also because of temperature ascent of the vapour during compression. In Open type, both the compressor and the motor normally reject heat to the surrounding air for efficient operation.

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.

The cooling rate relies on the stream rate of the refrigerant, its temperature and the thermal properties of the refrigerant. In the event that stream rate is not sufficient and/or if the temperature is not sufficiently low the insulation on the winding of the motor can wear out and short circuiting may happen. Thus, hermetically sealed compressors give satisfactory and safe performance over an exceptionally narrow range of design temperature and ought not to be utilized 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. On the other hand, hermetic compressors are almost universally utilized as a part of small systems such as household refrigerators, water coolers, air conditioners and so forth, where efficiency is not as important as customer accommodation (because of absence of continuous maintenance). In addition to this, the utilization of hermetic compressors is ideal in systems, which utilize capillary tubes as expansion devices and are critically charged systems. Hermetic compressors are normally not serviceable.

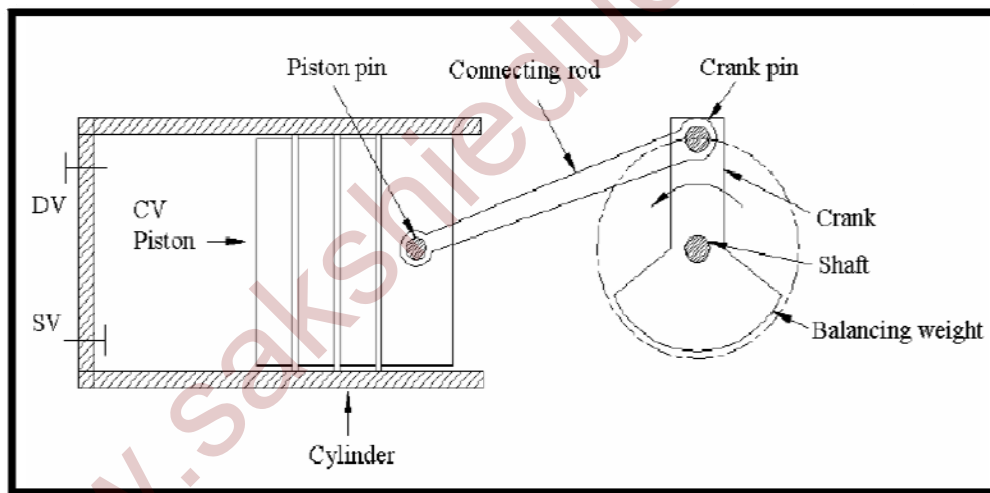
They are not exceptionally adaptable as it is hard to vary their rate to control the cooling capacity. In some (usually larger) hermetic units, the cylinder head is usually removable so that the valves and the piston can be overhauled. This type of unit is called a semi-hermetic (or semi-sealed) compressor.

### **1.3. Reciprocating compressors**

Reciprocating compressor is the workhorse of the refrigeration and air conditioning industry. It is the most generally utilized compressor with cooling capacities ranging from a couple of Watts to several kilowatts. Advanced reciprocating compressors are fast ( $\approx 3000$  to  $3600$  rpm), single

acting, single or multi-cylinder (up to 16 cylinders) type. Figure 1.1 demonstrates the schematic of a reciprocating compressor. Reciprocating compressors comprise of a piston moving back and forth in a cylinder, with suction and discharge valves to achieve suction and compression of the refrigerant vapour. Its development and working are somewhat similar to a two-stroke engine, as suction and compression of the refrigerant vapour are finished in one upset of the crank.

The suction side of the compressor is joined with the way out of the evaporator, while the discharge side of the compressor is associated with the condenser inlet. The suction (inlet) and the discharge (outlet) valves open and close because of pressure contrasts between the cylinder and inlet or outlet manifolds individually. The pressure in the inlet manifold is equal to or somewhat not exactly the evaporator pressure. Similarly the pressure in the outlet manifold is equal to or somewhat greater than the condenser pressure. The reason for the manifolds is to give stable inlet and outlet pressures for the smooth operation of the valves and also give a space to mounting the valve



**Fig 1.1:** Schematic of a reciprocating compressor

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 vapour velocities that will decrease the volumetric efficiency and the throttling loss will decrease the compression efficiency.

### 1.3.1. 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,  $\eta_V$  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.,

$$\eta_V = \frac{\text{Volumetric flow rate}}{\text{Compressor Displacement rate}} = \frac{m \cdot v_e}{V_{sw}}$$

where  $m$  is the mass flow rate of refrigerant (kg/s) and  $V_{sw}$  is the compressor displacement rate (m<sup>3</sup>/s) respectively, and  $v_e$  is the specific volume (m<sup>3</sup>/kg) of the refrigerant at compressor inlet.

For a given evaporator and condenser temperatures, one can also use the volumetric refrigeration capacity (kW/m<sup>3</sup>) 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 ( $\eta_C$ ), efficiency of the mechanical drive ( $\eta_{mech}$ ) and the motor efficiency ( $\eta_{motor}$ ). For a refrigerant compressor, the power input ( $W_c$ ) is given by:

$$W_c = \frac{W_{ideal}}{\eta_C \eta_{mech} \eta_{motor}}$$

Where,  $W_{ideal}$  is the power input to an ideal compressor. The temperature at the exit of the compressor (discharge temperature) 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.

**a) Ideal reciprocating compressor:**

An ideal reciprocating compressor is one in which:

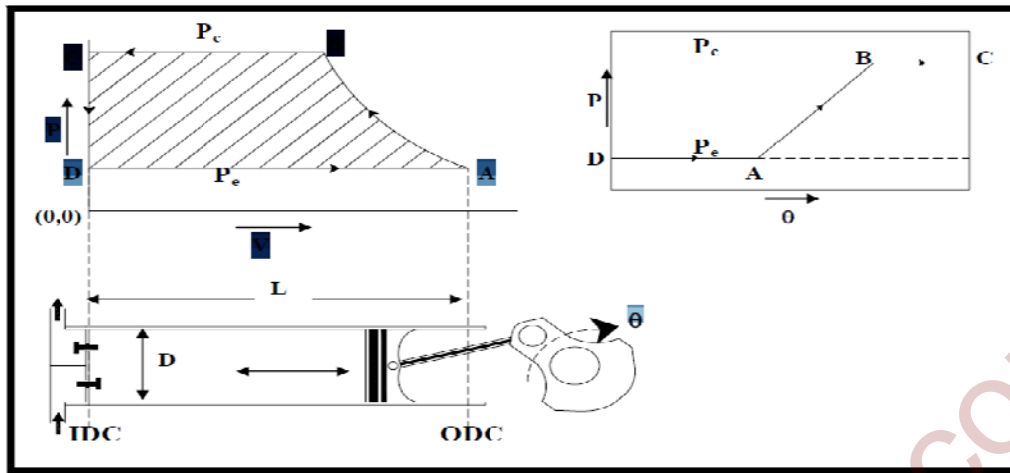
- i. The clearance volume is zero, i.e., at the end of discharge process, the volume of refrigerant inside the cylinder is zero.
- ii. No pressure drops during suction and compression
- iii. Suction, compression and discharge are reversible and adiabatic

Figure 1.2 shows the schematic of an ideal compression process on pressure-volume and pressure-crank angle ( $\theta$ ) 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.1.2.** 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:

$$\dot{m} = \frac{\dot{V}_{sw}}{v_e}$$

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  $V_{sw}$  of the compressor is given by:

$$\dot{V}_{sw} = nN \frac{\pi D^2}{4} L$$

Where,

n = Number of cylinders

N = Rotational speed of compressor, revolutions per second



D = Bore of the cylinder, m

L = Stroke length, m

### Work input to the ideal compressor:

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

$$W_{id} = W_{D-A} + W_{A-B} + W_{B-C}$$

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

$W_{B-C}$  = Work done by the piston on the refrigerant during discharge B-C  
 = Area under line B-C =  $P_c V_B$

$$\therefore W_{id} = -P_e V_A + \int_{V_A}^{V_B} P \cdot dV + P_c V_B = \text{Area A-B-C-D on P-V diagram} = \int_{P_e}^{P_c} V \cdot dP$$

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:

$$w_{id} = \frac{W_{id}}{M_r} = \int_{P_e}^{P_c} v \cdot dP$$

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:

$$W_c = m w_{id} = \frac{V_{sw}}{V_e} \int_{P_e}^{P_c} v \cdot dP$$

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

$$mep = \frac{W_{id}}{V_{sw}} = \frac{1}{v_e} \int_{P_e}^{P_c} v \cdot dP$$

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,  $P v^k = \text{constant}$ , hence the specific work of compression  $w_{id}$  can be obtained by integration, and it can be shown to be equal to:

$$w_{id} = \int_{P_e}^{P_c} v \cdot dP = P_e v_e \left( \frac{k}{k-1} \right) \left[ \left( \frac{P_c}{P_e} \right)^{\frac{k-1}{k}} - 1 \right]$$

In the above equation,  $k$  is the index of isentropic compression. If the refrigerant behaves as an ideal gas, then  $k = \gamma$ . 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.1.3. 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:

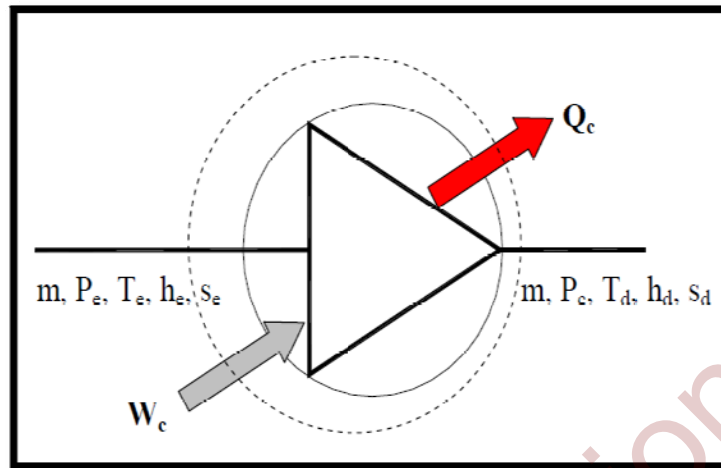
$$w_{id} = \frac{W_c}{\dot{m}} = (h_c - h_e)$$

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

$T ds = dh - v dP \Rightarrow dh = v dP$  ( $\because ds = 0$  for isentropic process)

$$\therefore w_{id} = \int_{P_e}^{P_c} v dP = \int_{P_e}^{P_c} dh = (h_d - h_e)$$

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



**Fig.1.3.** 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:

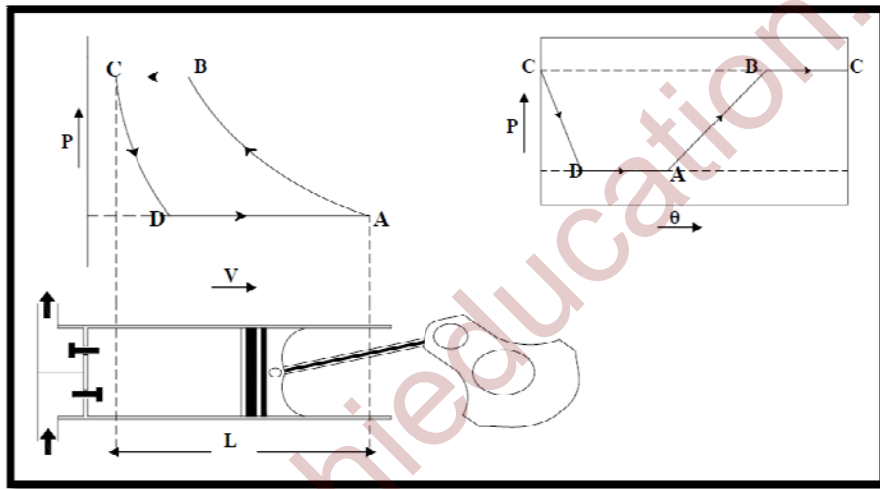
This space along with all other spaces between the closed valves and the piston at the inner dead centre (IDC) is called as Clearance volume,  $V_c$ . The ratio of the clearance volume to the swept volume is called as Clearance ratio,  $\epsilon$ , i.e.,

$$\epsilon = \frac{V_c}{V_{sw}}$$

The clearance ratio  $\epsilon$  relies on upon the arrangement of the valves in the cylinder and the mean piston velocity. Normally  $\epsilon$  is under 5 percent for very much designed compressors with moderate piston speeds ( $\approx 3$  m/s), however, it can be higher for higher piston speeds. Because of the vicinity of the clearance volume, at the end of the discharge stroke, some amount of refrigerant at the discharge pressure  $P_c$  will be left in the clearance volume. Thus, 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 indicated in Fig. 1.4, suction starts just when the pressure inside the cylinder falls to the suction pressure in an ideal compressor with clearance. This suggests that despite the fact that the compressor cleared volume,  $V_{SW} = V_A - V_C$ , the actual volume of the refrigerant that entered the cylinder during suction stroke is  $V_A - V_D$ . Thus, the volumetric efficiency of the compressor with clearance,  $\eta_{V,cl}$  is under 100 percent,

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



**Fig.1.4.** Ideal reciprocating compressor with clearance

From Fig.1.4, the clearance volumetric efficiency can be written as:

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

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

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

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

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.,

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

Hence, the clearance volumetric efficiency is given by:

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

If we assume the re-expansion process also to follow the equation  $Pvk = \text{constant}$ , then:

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

Hence the clearance volumetric efficiency is given by:

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

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;

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

The above expression shows that  $\eta_{V,cl}$   $\propto$  as  $r_p$   $\propto$  and  $\varepsilon$   $\propto$  as shown in Fig.1.5. It can also be seen that for a given compressor with fixed clearance ratio  $\varepsilon$ , there is a limiting pressure ratio at which the clearance volumetric efficiency becomes zero. This limiting pressure ratio is obtained from the equation:

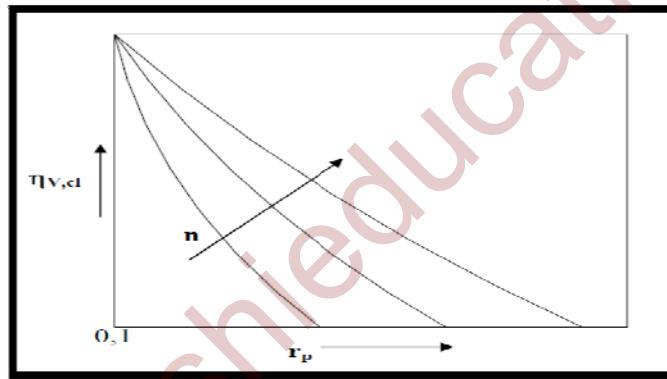
$$\eta_{v,cl} = 1 - \varepsilon \left[ r_p^{1/n} - 1 \right] = 0$$

$$\Rightarrow r_{p,max} = \left[ \frac{1 + \varepsilon}{\varepsilon} \right]^n$$

The mass flow rate of refrigerant compressed with clearance  $m_{cl}$  is given by:

$$\dot{m}_{cl} = \eta_{v,cl} \frac{\dot{V}_{sw}}{v_e}$$

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

### Work input to the compressor with clearance:

If we assume that both compression and expansion follow the same equation  $Pv^n = \text{constant}$  (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:

$$w_{id} = \int_{P_e}^{P_c} v \cdot dP = P_e v_e \left( \frac{n}{n-1} \right) \left[ \left( \frac{P_c}{P_e} \right)^{\frac{n-1}{n}} - 1 \right]$$

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:

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

$$mep = \eta_{v,cl} \frac{W_{id}}{V_e}$$

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:

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

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