Lesson
19
Performance Of Reciprocating Compressors

The specific objectives of this lecture are to:

1. Discuss the performance aspects of ideal reciprocating compressors with clearance, specifically:

   a) Effect of evaporator temperature on system performance at a fixed condenser temperature (*Section 19.1.1*)
   b) Effect of condenser temperature on system performance at a fixed evaporator temperature (*Section 19.1.1*)
   c) Effects of pressure ratio and type of refrigerant on compressor discharge temperature (*Section 19.1.3*)

2. Discuss the performance aspects of actual compressor processes by considering:

   a) Effect of heat transfer in the suction line and compressor (*Section 19.2.1*)
   b) Effects of pressure drops in the suction and discharge lines and across suction and discharge valves of compressor (*Section 19.2.2*)
   c) Effect of refrigerant leakage (*Section 19.2.3*)

3. Describe various methods of capacity control (*Section 19.3*)

4. Discuss methods of compressor lubrication (*Section 19.4*)

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

1. Describe qualitatively the effects of evaporator and condenser temperatures on performance of reciprocating compressors

2. Discuss the effects of heat transfer, pressure drops and refrigerant leakage on performance of actual compressors

3. Explain various methods of regulating the capacity of reciprocating compressors, and

4. Discuss aspects of compressor lubrication

19.1. Ideal compressor with clearance:

19.1.1. Effect of evaporator temperature:

The effect of evaporator temperature on performance of the system is obtained by keeping the condenser temperature (pressure) and compressor displacement rate and clearance ratio fixed. To simplify the discussions, it is further assumed that the refrigeration cycle is an SSS cycle.

a) On Volumetric efficiency and refrigerant mass flow rate:

The volumetric of the compressor with clearance is given by:

$$\eta_{V,cl} = 1+\varepsilon-\varepsilon\left(\frac{P_c}{P_e}\right)^{1/n} = 1-\varepsilon\left[r_p^{\ 1/n}-1\right] \qquad (19.1)$$

For a given condensing temperature (or pressure), the pressure ratio r_p increases as the evaporator temperature (or evaporator pressure) decreases. Hence, from the expression for clearance volumetric efficiency, it is obvious that the volumetric efficiency decreases as evaporator temperature decreases. This is also explained with the help of Fig.19.1, which shows the P-V diagram for different evaporator pressures. As shown, as the evaporator pressure decreases, the volume of refrigerant compressed decreases significantly, since the compressor displacement remains same the clearance volumetric efficiency decreases as evaporator temperature decreases. In fact, as explained in the earlier lecture, at a limiting pressure ratio, the volumetric efficiency becomes zero.

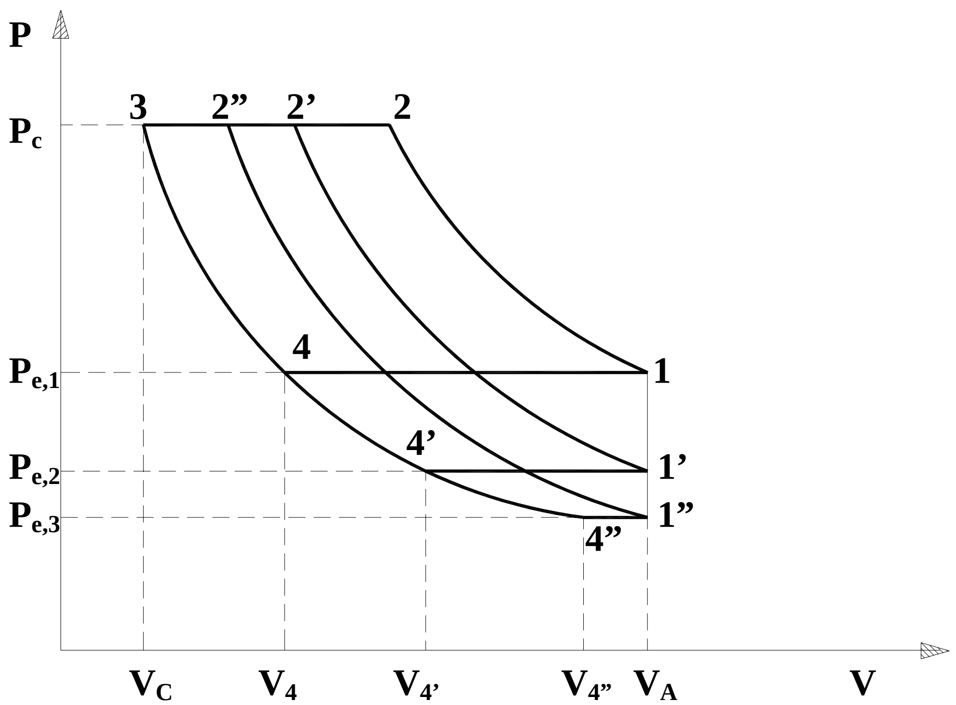

Fig.19.1. *P-V diagram for different evaporator pressures and a fixed condenser pressure*

The mass flow rate of refrigerant $\dot{m}$ is given by:

$$\dot{m} = \eta_{V,cl} \frac{\dot{V}_{SW}}{v_e} \qquad (19.2)$$

As the evaporator temperature decreases the clearance volumetric efficiency decreases and the specific volume of refrigerant at compressor inlet v_e increases. As a result of these two effects, the mass flow rate of refrigerant through the compressor decreases rapidly as the evaporator temperature decreases as shown in Fig.19.2.

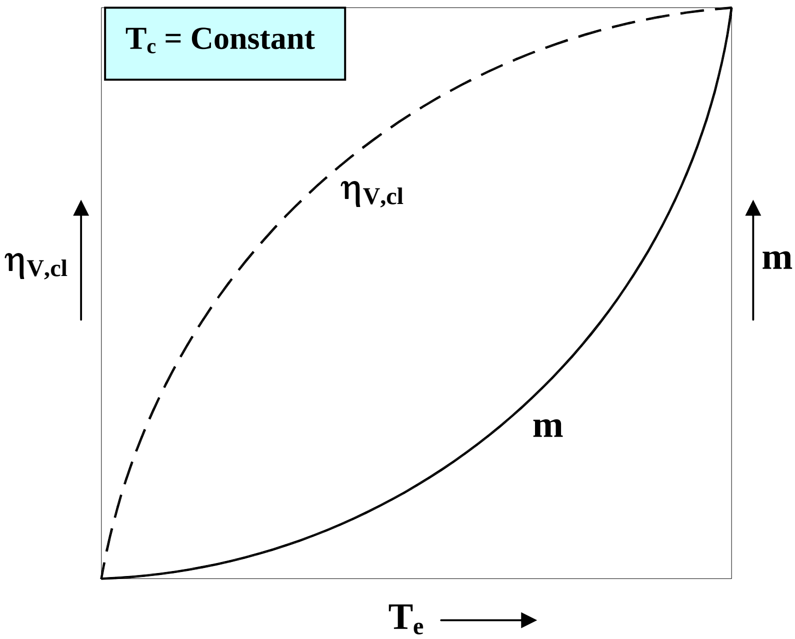

Fig.19.2. *Effect of evaporator temperature on clearance volumetric efficiency and refrigerant mass flow rate*

b) On refrigeration effect and refrigeration capacity:

A compressor alone cannot provide refrigeration capacity. By refrigeration capacity of compressor what we mean is the capacity of a refrigeration system that uses the compressor under discussion. Figure 19.3 (a) shows the SSS cycle on P-h diagram at different evaporator temperatures. It can be seen from the figure that the refrigeration effect, q_e ($q_e = h_1 - h_4$) increases marginally as the evaporator temperature is increased. This is due to the shape of the saturation vapour curve on P-h diagram. The effect of T_e on refrigerant effect is also shown in Fig.19.3(b).

The refrigeration capacity of the compressor Q_e is given by:

$$Q_e = \dot{m}.q_e \qquad (19.3)$$

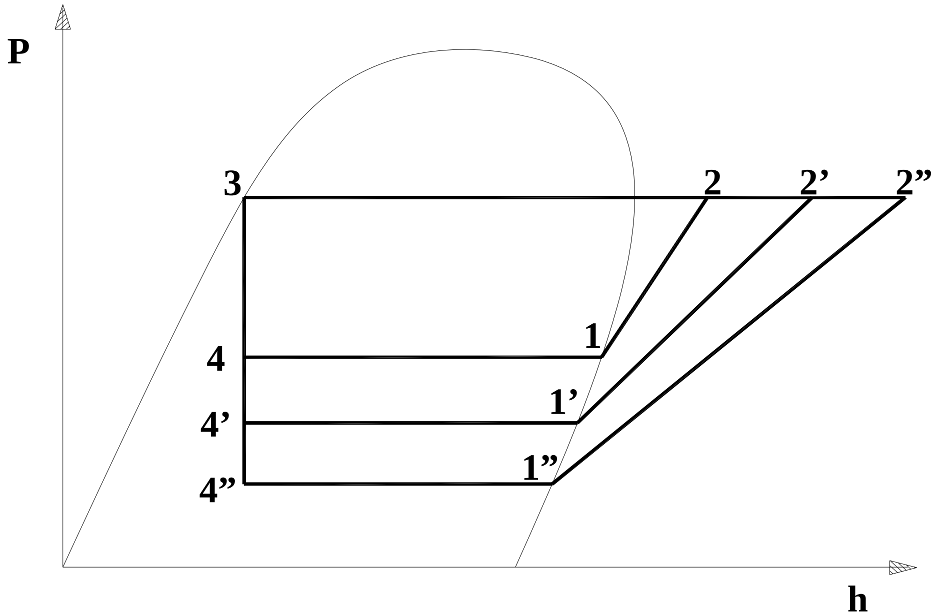

***Fig.19.3(a)**: Effect of evaporator temperature on refrigeration effect on P-h diagram*

Since mass flow rate of refrigerant increases rapidly and refrigerant effect also increases, though marginally with increase in evaporator temperature, the refrigeration capacity increases sharply with increase in evaporator temperature as shown in Fig.19.3(b).

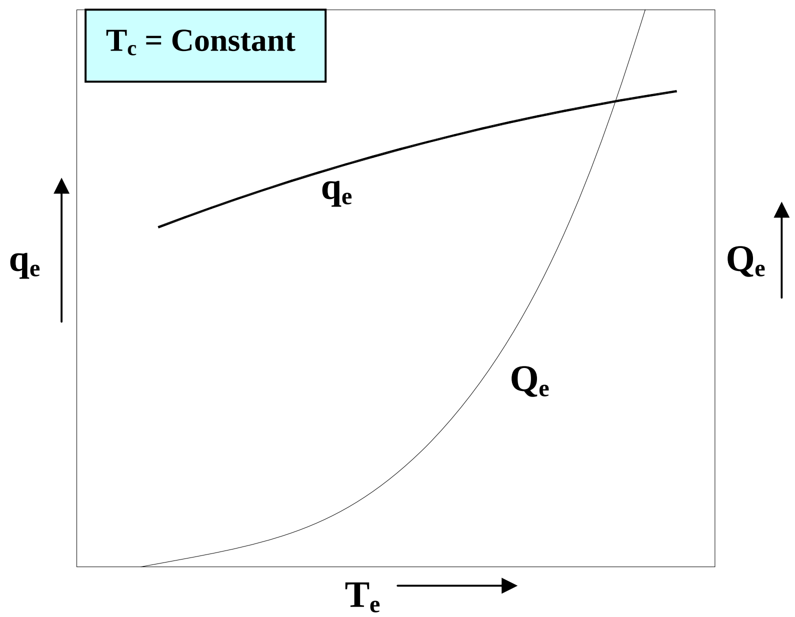

***Fig.19.3(b)**: Effect of evaporator temperature on refrigeration effect and refrigeration capacity*

c) On work of compression and power requirement:

At a constant condenser temperature as evaporator temperature increases the work of compression, Δh_c (= h_2-h_1) decreases as shown in Fig.19.3(a). This is due to the divergent nature of isentropes in the superheated region. The work of compression becomes zero when the evaporator temperature becomes equal to the condenser temperature (T_e=T_c) as shown in Fig. 19.4.

The power input to the compressor is given by:

$$\mathbf{W_c = \dot{m}.\Delta h_c} \qquad (19.4)$$

As discussed before, for a given clearance ratio and condenser temperature, the volumetric efficiency and hence the mass flow rate becomes zero at a lower limiting value of evaporator temperature (Te = $T_{e,lim}$). Since the work of compression becomes zero when the evaporator temperature equals the condenser temperature, the power input to the compressor, which is a product of mass flow rate and work of compression is zero at a low evaporator temperature (at which the mass flow rate is zero). And the power input also becomes zero when evaporator temperature equals condenser temperature (at which the work of compression becomes zero). This implies that as evaporator temperature is increased from the limiting value, the power curve increases from zero, reaches a peak and then becomes zero as shown in Fig.19.4.

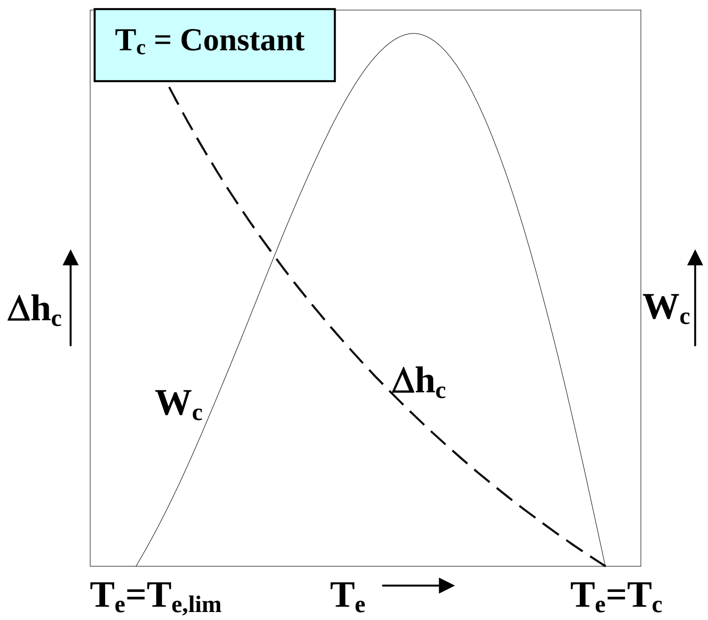

***Fig.19.4**: Effect of evaporator temperature on work of compression (Δh_c) and power input to compressor (W_c)*

The variation of compressor power input with evaporator temperature has a major practical significance. As a mentioned before, there is an evaporator temperature at which the power reaches a maximum value. If the design evaporator temperature of the refrigeration system is less than the evaporator temperature at which the power is maximum, then the design power requirement is lower than the peak power input. However, during the initial pull-down period, the initial evaporator temperature may lie to the left of the power peak. Then as the system runs steadily the evaporator temperature reduces and the power requirement passes through the peak point. If the motor is designed to suit the design power input then the motor gets overloaded during every pull-down period as the peak power is greater than the design power input. Selecting an oversized motor to meet the power peak is not an energy efficient solution, as the motor will be underutilized during the normal operation. One way of overcoming the problem is to throttle the suction gas during the pull-down so that the refrigerant mass flow rate is reduced and the motor does not pass through the power peak. In multi-cylinder compressors, some of the cylinders can be unloaded during the pull-down so as to reduce the power requirement.

d) On COP and volume flow rate per unit capacity:

The COP of the system is defined as:

$$\mathbf{COP} = \frac{\mathbf{Q_e}}{\mathbf{W_c}} = \frac{\mathbf{q_e}}{\mathbf{\Delta h_c}} \qquad (19.5)$$

As discussed before, as the evaporator temperature increases the refrigeration effect, q_e increases marginally and the work of compression, Δh_c reduces sharply. As a result the COP of the system increases rapidly as the evaporator temperature increases as shown in Fig.19.5.

The volume flow rate per unit capacity, V is given by:

$$\mathbf{V} = \frac{\eta_{\mathbf{V,cl}} \cdot \dot{\mathbf{V}}_{\mathbf{SW}}}{\mathbf{Q_e}} = \frac{\mathbf{v_e}}{\mathbf{q_e}} \qquad (19.6)$$

As evaporator temperature increases the specific volume of the refrigerant at compressor inlet reduces rapidly and the refrigerant effect increases marginally. Due to the combined effect of these two, the volume flow rate of refrigerant per unit capacity reduces sharply with evaporator temperature as shown in Fig. 19.5. This implies that for a given refrigeration capacity, the required volumetric flow rate and hence the size of the compressor becomes very large at very low evaporator temperatures.

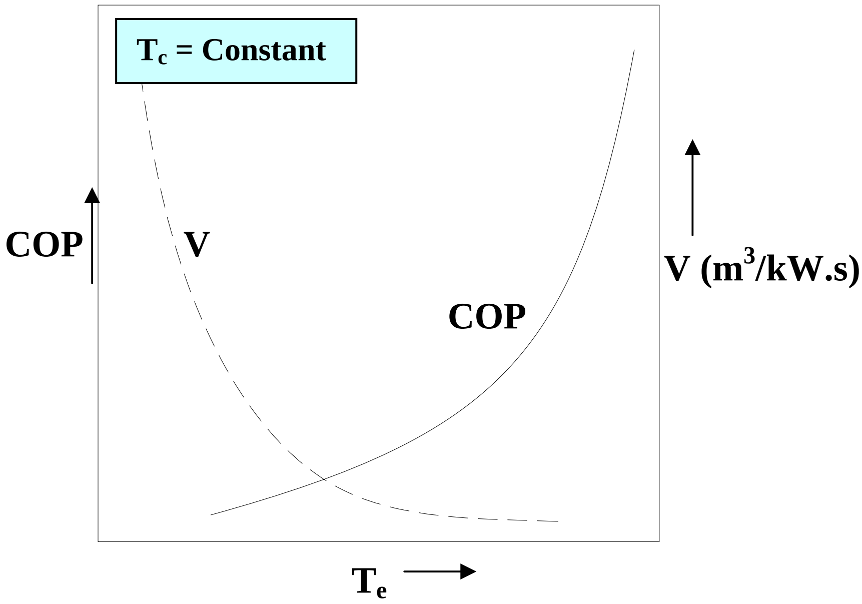

***Fig.19.5**: Effect of evaporator temperature on COP and volume flow rate per unit capacity (V)*

19.1.2. Effect of condenser temperature:

Atmospheric air is the cooling medium for most of the refrigeration systems. Since the ambient temperature at a location can vary over a wide range, the heat rejection temperature (i.e., the condensing temperature) may also vary widely. This affects the performance of the compressor and hence the refrigeration system. The effect of condensing temperature on compressor performance can be studied by keeping evaporator temperature constant.

a) On volumetric efficiency and refrigerant mass flow rate:

Figure 19.6 shows the effect of condensing temperature on clearance volumetric efficiency and mass flow rate of refrigerant. At a constant evaporator temperature as the condensing temperature increases, the pressure ratio increases, hence, both the volumetric efficiency and mass flow rate decrease as shown in the figure. However, the effect of condensing temperature on mass flow rate is not as significant as the evaporator temperature as the specific volume of refrigerant at compressor inlet is independent of condensing temperature.

b) On refrigeration effect and refrigeration capacity:

At a constant evaporator temperature as the condensing temperature increases, then the enthalpy of refrigerant at the inlet to the evaporator increases. Since the evaporator enthalpy remains constant at a constant evaporator temperature, the refrigeration effect decreases with increase in condensing temperature as shown in Fig. 19.7. The refrigeration capacity (Qe) also reduces with increase in condensing temperature as both the mass flow rate and refrigeration effect decrease as shown in Fig.19.7.

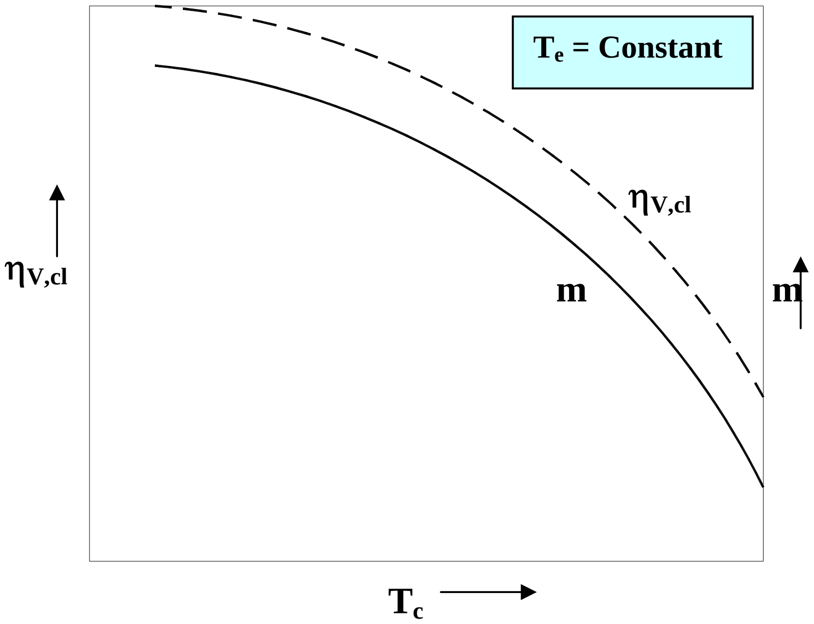

Fig.19.6. *Effect of condenser temperature on clearance volumetric efficiency and mass flow rate of refrigerant*

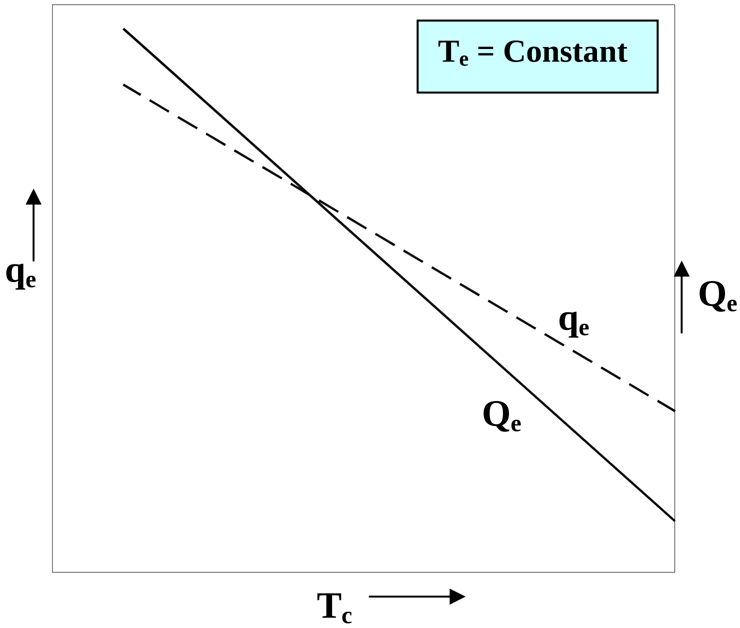

Fig.19.7. *Effect of condenser temperature on refrigeration effect and refrigeration capacity*

c) On work of compression and power requirement:

The work of compression is zero when the condenser temperature is equal to the evaporator temperature, on the other hand at a limiting condensing temperature the mass flow rate of refrigerant becomes zero as the clearance volumetric efficiency becomes zero as explained before. Hence, similar to the effect of evaporator temperature on power curve, the compressor power input increases from zero (work of compression is zero), reaches a peak and then again becomes zero at a high value of condensing temperature as shown in Fig.19.8. However, the peak power in this case is not as critical as with evaporator temperature since the chances of condenser operating at such a high temperatures are rare.

d) On COP and volume flow rate per unit capacity:

As condensing temperature increases the refrigeration effect reduces marginally and work of compression increases, as a result the COP reduces as shown in Fig.19.9. Even though the specific volume at compressor inlet is independent of condensing temperature, since the refrigeration effect decreases with increase in condensing temperature, the volume flow rate of refrigerant per unit capacity increases as condenser temperature increases as shown in Fig.19.9.

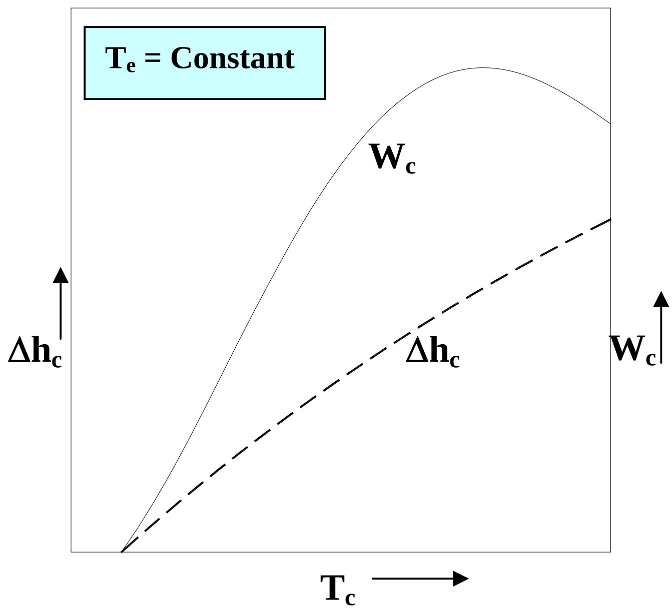

Fig.19.8: *Effect of condenser temperature on work of compression and power input to compressor*

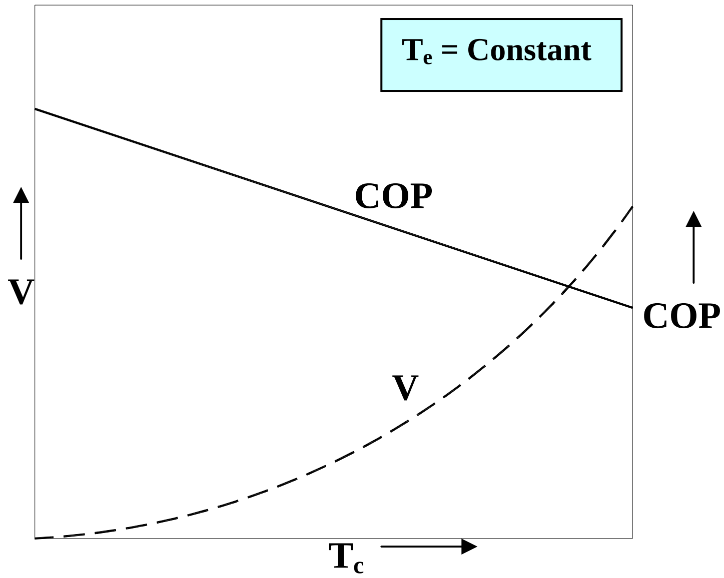

***Fig.19.9**: Effect of condensing temperature on COP and volume flow rate per unit capacity (V)*

The above discussion shows that the performance of the system degrades as the evaporator temperature decreases and condensing temperature increases, i.e., the temperature lift increases. This is in line with the effect of these temperatures on reverse Carnot refrigeration system. It is seen that compared to the condensing temperature, the effect of evaporator temperature is quiet significant. When the heat sink temperature does not vary too much then the effect of condensing temperature may not be significant.

19.1.3. Compressor discharge temperature:

If the compressor discharge temperature is very high then it may result in breakdown of the lubricating oil, causing excessive wear and reduced life of the compressor valves (mainly the discharge valve). In hermetic compressors, the high discharge temperature adversely affects the motor insulation (unless the insulation is designed for high temperatures). When the temperature is high, undesirable chemical reactions may take place inside the compressor, especially in the presence of water. This may ultimately damage the compressor.

If the compression process is assumed to be isentropic and the refrigerant vapour is assumed to be have as a perfect gas, then the following equations apply:

$$\mathbf{Pv^{\gamma} = constant \ \ and \ Pv = RT} \qquad (19.7)$$

Then the discharge temperature, T_d is given by:

$$\mathbf{T_d = T_e \left(\frac{P_c}{P_e}\right)^{\frac{\gamma-1}{\gamma}}} \qquad (19.8)$$

Thus for a given compressor inlet temperature, T_e, the discharge temperature T_d increases as the pressure ratio (P_c/P_e) and specific heat ratio γ increase. Even though refrigerant vapour may not exactly behave as a perfect gas, the trends remain same. Figure 19.10 shows the variation of discharge temperature as a function of pressure ratio for three commonly used refrigerants, ammonia, R 22 and R 12. As shown in the figure since specific heat ratio of ammonia is greater than R 22, which in turn is greater than R 12, at a given pressure ratio, the discharge temperature of ammonia is higher than R 22, which in turn is higher than R 12. Since the high discharge temperature of ammonia may damage the lubricating oil, normally ammonia compressors are cooled externally using water jackets.

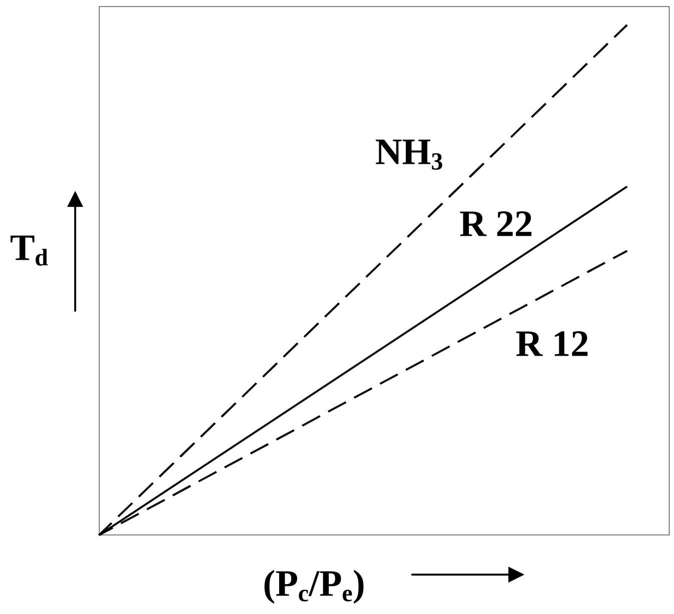

***Fig.19.10**: Variation of compressor discharge temperature with pressure ratio for different refrigerants*

19.2. Actual compression process

Actual compression processes deviate from ideal compression processes due to:

i. Heat transfer between the refrigerant and surroundings during compression and expansion, which makes these processes non-adiabatic
ii. Frictional pressure drops in connecting lines and across suction and discharge valves
iii. Losses due to leakage

19.2.1. Effect of heat transfer:

Heat transfer from the cylinder walls and piston to the refrigerant vapour takes place during the suction stroke and heat transfer from the refrigerant to the surroundings takes place at the end of the compression. In hermetic compressors additional heat transfer from the motor winding to refrigerant takes place. The effect of this heat transfer is to increase the temperature of refrigerant, thereby increasing the specific volume. This in general results in reduced volumetric efficiency and hence reduced refrigerant mass flow rate and refrigeration capacity. The extent of reduction in mass flow rate and refrigeration capacity depends on the pressure ratio, compressor speed and compressor design. As seen before, the discharge temperature and hence the temperature of the cylinder and piston walls increase with pressure ratio. As the compressor speed increases the heat transfer rate from the compressor to the surroundings reduces, which may result in higher refrigerant temperature. Finally, the type of external cooling provided and compressor design also affects the performance as it influences the temperature of the compressor.

Since the compression and expansion processes are accompanied by heat transfer, these processes are not adiabatic in actual compressors. Hence, the index of compression is not isentropic index but a polytropic index. However, depending upon the type of the compressor and the amount of external cooling provided, the compression process may approach an adiabatic process (as in centrifugal compressors) or a reversible polytropic process (as in reciprocating compressors with external cooling). The index of compression may be greater than isentropic index (in case of irreversible adiabatic compression). When the process is not reversible, adiabatic, then the polytropic index of compression 'n' depends on the process and is not a property of the refrigerant. Also the polytropic index of compression may not be equal to the polytropic index of expansion. Since the compression process in general is irreversible, the actual power input to the compressor will be greater than the ideal compression work. Sometimes the isentropic efficiency is used to estimate the actual work of compression. The isentropic efficiency η_{is} for the compressor is defined as:

$$\eta_{is} = \frac{\Delta h_{c,is}}{\Delta h_{c,act}} \qquad (19.9)$$

where $\Delta h_{c,is}$ is the isentropic work of compression and $\Delta h_{c,act}$ is the actual work of compression. It is observed that for a given compressor the isentropic efficiency of the compressor is mainly a function of the pressure ratio. Normally the function varies from compressor to compressor, and is obtained by conducting experimental studies on compressors. The actual work of compression and actual power input can be obtained if the isentropic efficiency of the compressor is known as the isentropic work of compression can be calculated from the operating temperatures.

19.2.2. Effect of pressure drops:

In actual reciprocating compressors, pressure drop takes place due to resistance to fluid flow. Pressure drop across the suction valve is called as "wire drawing". This pressure drop can have adverse effect on compressor performance as the suction pressure at the inlet to the compressor P_s will be lower than the evaporator pressure as shown in Fig.19.11. As a result, the pressure ratio and discharge temperature increases and density of refrigerant decreases. This in turn reduces the volumetric efficiency, refrigerant mass flow rate and increases work of compression. This pressure drop depends on the speed of the compressor and design of the suction valve. The pressure drop increases as piston speed increases.

Even though the pressure drop across the discharge valve is not as critical as the pressure drop across suction valve, it still affects the compressor performance in a negative manner.

The net effect of pressure drops across the valves is to reduce the refrigeration capacity of the system and increase power input. The pressure drops also affect the discharge temperature and compressor cooling in an adverse manner.

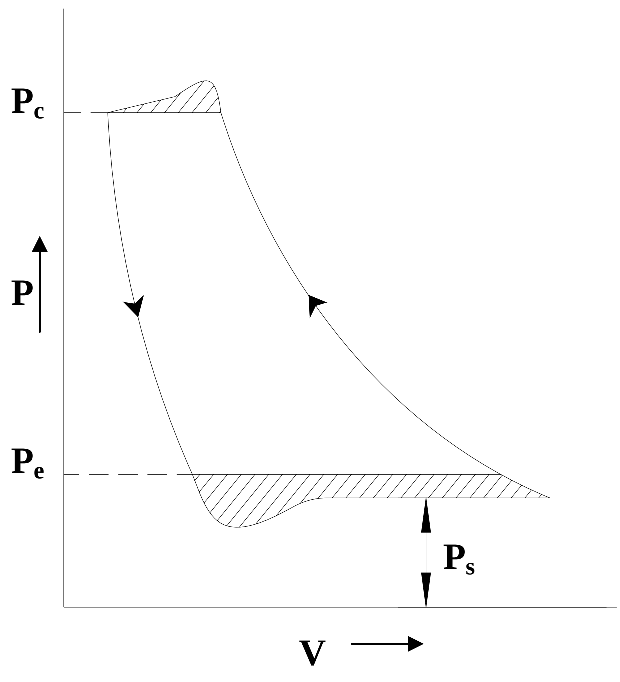

Fig.19.11: *Effects of suction and discharge side pressure drops on P-V diagram of a reciprocating compressor*

19.2.3. Effect of leakage:

In actual compressors, refrigerant leakage losses take place between the cylinder walls and piston, across the suction and discharge valves and across the oil seal in open type of compressors. The magnitude of these losses depends upon the design of the compressor valves, pressure ratio, compressor speed and the life and condition of the compressor. Leakage losses increase as the pressure ratio increases, compressor speed decreases and the life of compressor increases. Due to the leakage, some amount of refrigerant flows out of the suction valves at the beginning of compression stroke and some amount of refrigerant enters the cylinder through the discharge valves at the beginning of suction stroke. The net effect is to reduce the mass flow rate of refrigerant. Even though it is possibly to minimize refrigerant leakage across cylinder walls, eliminating leakages across valves is not possible as it is not possible to close the valves completely during the running of the compressor.

As a result of the above deviations, the actual volumetric efficiency of refrigerant compressors will be lower than the clearance volumetric efficiency. It is difficult to estimate the actual efficiency from theory alone. Normally empirical equations are developed to estimate this parameter. The actual volumetric efficiency can be defined either in terms of volumetric flow rates or in terms of mass flow rates, i.e.,

$$\eta_{V,act} = \frac{\text{actual volumetric flow rate}}{\text{Compressor displacement rate}} = \frac{\text{actual mass flow rate}}{\text{maximum possible mass flow rate}}$$

In general,

$$\eta_{V,act} = \eta_{V,th} \frac{T_s}{T_{sc}} - \xi_L \qquad (19.10)$$

where $\eta_{V,th}$ = Theoretical volumetric efficiency obtained from P-V diagram
T_s = Temperature of vapour at suction flange, K
T_{sc} = Temperature of vapour at the beginning of compression, K
ξ_L = Leakage loss (fraction or percentage)

Several tests on compressors show that the actual volumetric of a given compressor is mainly a function of pressure ratio, and for a given pressure ratio it remains practically constant, irrespective of other operating conditions. Also, compressors with same design characteristics will have approximately the same volumetric efficiency, irrespective of the size. It is shown that for a given compressor, the actual volumetric efficiency can be obtained from the empirical equation:

$$\eta_{V,act} = A - B(r_p)^C \qquad (19.11)$$

where A, B and C are empirical constants to be obtained from actual test data and rp is the pressure ratio.

Depending upon the compressor and operating conditions, the difference between actual and theoretical volumetric efficiency could be anywhere between 4 to 20 percent.

Since heat transfer rate and leakage losses reduce and pressure drops increase with increase in refrigerant velocity, the actual volumetric efficiency reaches a maximum at a certain optimum speed. An approximate relation for optimum speed as suggested by Prof. Gustav Lorentzen is:

$$V_{opt} / \sqrt{M} \approx 420 \text{ m/s} \quad (19.12)$$

where V_{opt} is the optimum velocity of the refrigerant through the valve port in m/s and M is the molecular weight of the refrigerant in kg/kmol. This relation suggests that higher the molecular weight of the refrigerant lower is the optimum refrigerant velocity.

19.3. Capacity control of reciprocating compressors:

Normally refrigerant compressors are designed to take care of the most severe operating conditions, which normally occurs when the cooling load is high and/or the condenser operates at high temperatures due to high heat sink temperatures. However, when the operating conditions are not so severe, i.e., when the cooling load is low and/or the heat sink temperature is low, then the compressor designed for peak load conditions becomes oversized. If no control action is taken, then the compressor adjusts itself by operating at lower evaporator temperature, which may affect the refrigerated space temperature. The temperature of the evaporator during part load conditions reduces as the rate at which the compressor removes refrigerant vapour from the evaporator exceeds the rate of vaporization in the evaporator. As a result the evaporator pressure, and hence the evaporator temperature reduces. Operating at low evaporator temperature may lead to other problems such as low air humidity, frosting of evaporator coils and freezing of the external fluid. To avoid these problems, the capacity of the compressor has to be regulated depending upon the load. Various methods available in practice for controlling the capacity of compressors are:

a) Cycling or on-off control
b) Back pressure regulation by throttling of suction gas
c) Hot gas by-pass
d) Unloading of cylinders in multi-cylinder compressors, and
e) Compressor speed control

The cycling or on-off control is normally used in very small capacity refrigeration systems such as domestic refrigerators, room air conditioners, water coolers etc. The on-off control is achieved with the help of a thermostat, which normally senses the temperature inside the refrigerated space or evaporator temperature. As long as the temperature is greater than a set temperature (cut-out point) the compressor runs, and when the temperature falls below the cut-out temperature the thermostat switches-off the compressor. The temperature at which the compressor is switched-on again is known as cut-in temperature. The difference between the cut-in and cut-out temperatures is called as differential of the thermostat, which can be adjusted internally. The level of temperature at which the thermostat operates is called as the range of the thermostat, which can also be adjusted by the customer by turning a knob. For example, a thermostat may have a cut-in temperature of 10^oC and a cut-out temperature of 9^oC, in which case the differential is 1^oC. By turning the thermostat knob, the same thermostat can be made to operate, say at 7^oC of cut-in temperature and 6^oC of cut-out temperature. In this example, the differential has been kept fixed at 1^oC, while the range has been varied. As mentioned, it is also possible to vary the differential so that the thermostat can operate at a cut-in temperature of 10^oC and a cut-out temperature of 8^oC, with a differential of 2^oC. Thus the temperature in the refrigerated space varies between the cut-out and cut-in values. In stead of a thermostat which takes control action based on temperatures, it is also possible to use a pressure sensing device to initiate on-off control. This type of device is called a pressostat, and is designed to take control action by sensing the evaporator pressure. The on-off control is satisfactory in applications where the fluctuation in product temperatures due to on-off control is acceptable. Thus it is suitable when the thermal capacity of the product or the refrigerated space is large so that small variation in it can give sufficient variation in evaporator temperature. On-off control is not good when the temperature has to be regulated within a small range, in which case the compressor has to start and stop very frequently. Small compressor motors can be cycled for about 10 cycles per hour, whereas large compressor motors are normally not allowed to start and stop for more than one or two times in an hour.

Back-pressure regulation by throttling the suction gas reduces the refrigeration capacity of the compressor. However, this method is not normally used for regular capacity control as it does not reduce the compressor power input proportionately, consequently it is energy inefficient. This method is normally used during the pull-down period so as to avoid the power peak.

Hot gas bypass to suction side is an effective method of controlling the capacity. In this method, when the evaporator pressure falls below a predetermined value, a hot gas bypass valve is opened and hot refrigerant from the discharge side flows back into the suction side of the compressor. A constant pressure expansion valve can be used as a hot gas bypass valve. Though by this method the capacity of the compressor can be regulated quite closely, this method suffers from some disadvantages such as little or no reduction in compressor power consumption at reduced refrigeration capacities, excessive superheating of the suction gas resulting in overheating of the compressors. Hence, this method is normally used in small compressors. However, in conjunction with other efficient methods, hot gas bypass is used when it is required to regulate the capacity down to 0 percent or for unloaded starting. Overheating of the compressor can be reduced by sending the hot bypass gas to the evaporator inlet. This also maintains sufficiently high refrigerant velocity in the evaporator so that oil return to the compressor can be improved during low cooling loads. Figure 19.12 shows the schematic of a refrigeration system with a hot gas bypass arrangement. In the figure, the solid line is for the system in which the by-passed hot gas enters the inlet of the compressor, while the dashed line is for the system in which the by-passed hot gas enters at the inlet to the evaporator.

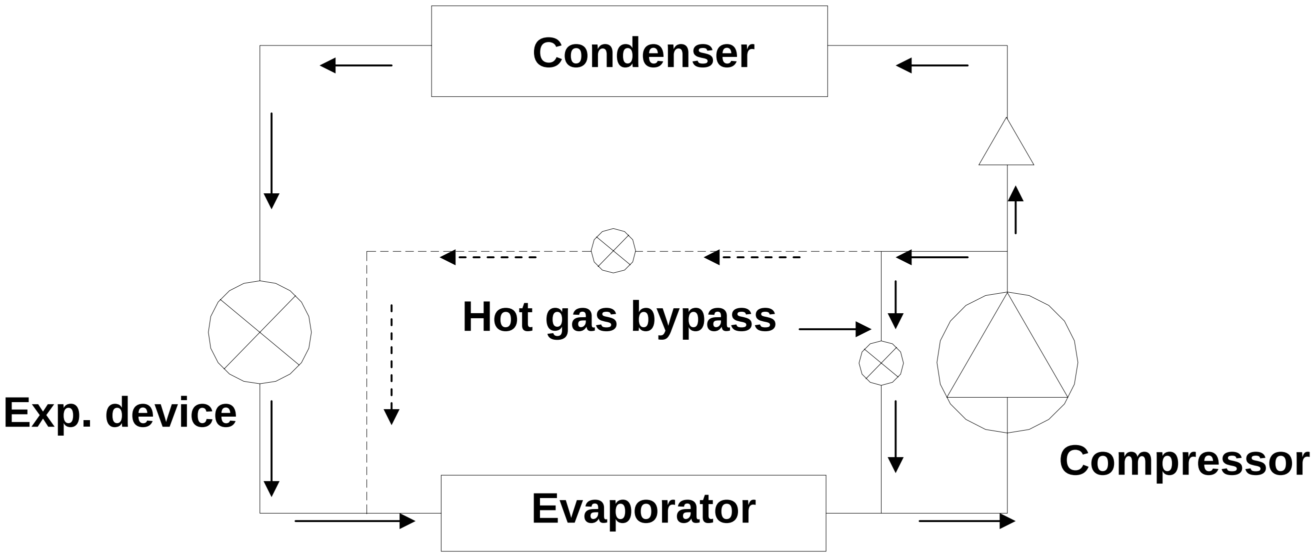

***Fig.19.12**: A vapour compression refrigeration system with hot gas bypass arrangement*

Unloading of cylinders in multi-cylinder compressors is another effective method of regulating compressor capacity. This is achieved usually by keeping the suction valves of some of the cylinders open during the compression stroke. As a result, the suction vapour drawn into these cylinders during suction stroke is returned to the suction line during the compression stroke. This is done with the help of pressure sensing switch, which senses the low pressure in the evaporator and opens some of the suction valves. In addition to capacity regulation, this method is also used during pull-down so that the peak power point can be skipped. This method is efficient as the required power input reduces with reduced cooling load, though not in the same proportion. Hence, this is one of the methods commonly employed in large systems.

Controlling the capacity of the compressor by regulating its speed is one of the most efficient methods as the required power input reduces almost in the same proportion with cooling load. However, for complete control a variable frequency drive may be required, which increases the cost of the system. In addition, reducing the speed too much may effect the compressor cooling and oil return.

19.4. Compressor lubrication:

Reciprocating compressors require lubrication to reduce wear between several parts, which rub against each other during the operation. Normally lubricating oil is used to lubricate the compressors. The lubricating oil usually comes in contact with the refrigerant and mixes with it, hence, it is essential to select a suitable oil in refrigerant compressors. The important properties that must be considered while selecting lubricating oil in refrigerant compressors are:

a) Chemical stability
b) Pour and/or floc points
c) Dielectric strength, and
d) Viscosity

In addition to the above, the nature of the refrigerant used, type and design of the compressor, evaporator and compressor discharge temperatures have to be considered while selecting suitable lubricating oils.

The oil should not undergo any chemical changes for many years of operation. This aspect is especially critical in hermetic compressor where, oil is not supposed to be changed for ten years or more. Since the discharge temperature is normally high in these compressors, the oil should not decompose even under very high temperatures. The chemical stability of the oil is inversely proportional to the number of unsaturated hydrocarbons present in the oil. For refrigerant compressors, oils with low percentage of unsaturated hydrocarbons are desirable.

The pour point of the oil may be defined as the lowest temperature at which the oil can flow or pour, when tested under specific conditions. The pour point is important for systems working at low evaporator temperatures. The pour point depends upon the wax content, higher the wax content, higher will be the pour point. Hence, for low temperature applications oils with low wax content should be used, otherwise the oil may solidify inside the evaporator tubes affecting the system performance and life of the compressor. The temperature at which the wax in the oil begins to precipitate is called as the cloud point. The floc point of the oil is the temperature at which wax will start to precipitate from a mixture of 90% R 12 and 10% oil by volume. In case of refrigerants such as R 12, viscosity of oil is reduced, as the refrigerant is soluble in oil. The floc point of the oil is a measure of the tendency of the oil to separate wax when mixed with an oil-soluble refrigerant. Hence it is an important parameter to be considered while selecting lubricating oils for these refrigerants. Since the tendency for wax to separate increases with amount of oil in refrigerant, the concentration of oil in refrigerant should normally be kept below 10 percent with these refrigerants. Floc point is not important in case of refrigerants that are not soluble in oil (e.g. ammonia).

Dielectric strength of the oil is a measure of its resistance to the flow of electric current. It is normally expressed in terms of the voltage required to cause an electric arc across a gap of 0.1 inch between two poles immersed in oil. Since impurities such as moisture, dissolved solids (metallic) reduce the dielectric strength of oil, a high dielectric strength is an indication of the purity of the oil. This parameter is very important in case of hermetic compressors as an oil with low dielectric strength may lead to shorting of the motor windings.

The viscosity of the oil is an important parameter in any lubricating system. The viscosity of the oil should be maintained within certain range for the lubrication system to operate effectively. If the viscosity is too low then the wear between the rubbing surfaces will be excessive, in addition to this it may not act as a good sealing agent to prevent refrigerant leakage. However, if the viscosity is too high then fluid friction will be very high and the oil may not fill the small gaps between the rubbing surfaces, again leading to excessive wear. The problem is complicated in refrigerant compressors as the viscosity of the oil varies considerably with temperature and refrigerant concentration. The oil viscosity increases as temperature and concentration of refrigerant decrease and vice versa.

Both mineral oils as well as synthetic oils have been used as lubricating oils in refrigeration. The mineral oils have to be refined to improve their chemical stability and reduce their pour and/or floc points. Synthetic oils have been developed to provide high chemical stability, good lubricity, good refrigerant solubility, lower pour/floc points and required viscosity.

19.4.1. Methods of lubrication:

Lubrication can be either splash type or force feed type. Normally small compressors (upto 10 kW input) are splash lubricated. Larger compressors use forced feed type lubrication. In splash type lubrication, the compressor crankcase which acts as an oil sump is filled with oil to a certain level. As the crankshaft rotates, the connecting rod and crankshaft dip into the oil sump causing the oil to be splashed on the rubbing surfaces. In some compressors, small scoops or dippers are attached to the connecting rod, which pick the oil and throws it onto the rubbing surfaces. In small, high-speed compressors, flooded type splash lubrication is used. In these modified type, slinger rings are screws are used for lifting the oil above crankshaft or main bearings, from where the oil floods over the rubbing surfaces. This prevents excessive oil carryover due to violent splashing in high-speed compressors.

In the forced feed method of lubrication an oil pump is used to circulate the oil to various rubbing surfaces under pressure. The oil drains back into the oil sump due to gravity and is circulated again.

If the refrigerants are not soluble in lubricating oil, then there is possibility of oil being carried away from the compressor and deposited elsewhere in the system. To prevent this, oil separators are used on the discharge side of the compressor, from where the oil is separated from the refrigerant vapour and is sent back to the compressor.

Questions and answers:

1. The refrigeration capacity of a reciprocating compressor increases:

a) As the evaporator temperature increases and condenser temperature decreases
b) As the evaporator temperature decreases and condenser temperature increases
c) As the evaporator and condenser temperatures increase
d) As the evaporator and condenser temperatures decrease

Ans. a)

2. For a given refrigeration capacity, the required size of the compressor increases as:

a) As the evaporator temperature increases and condenser temperature decreases
b) As the evaporator temperature decreases and condenser temperature increases
c) As the evaporator and condenser temperatures increase
d) As the evaporator and condenser temperatures decrease

Ans. b)

3. During every pull-down, the reciprocating compressor is likely to be overloaded as:

a) The initial refrigerant mass flow rate is high and work of compression is low
b) The initial refrigerant mass flow rate is low and work of compression is high
c) Both the mass flow rate and work of compression are high in the initial period
d) None of the above

Ans. a)

4. Ammonia compressors normally have water jackets for cooling as:

a) The latent heat of ammonia is high compared to synthetic refrigerants
b) The boiling point of ammonia is high
c) The critical temperature of ammonia is high
d) The index of compression of ammonia is high

Ans. d)

5. The actual volumetric efficiency of a reciprocating compressor is smaller than the clearance volumetric efficiency due to:

a) Pressure drop across suction line and suction valve
b) Pressure drop across discharge line and discharge valve
c) Heat transfer in suction line
d) Leakage of refrigerant across valves
e) All of the above

Ans. e)

6. When the compression process is reversible, polytropic with heat transfer from compressor, then:

a) The index of compression will be smaller than the isentropic index of compression
b) The index of compression will be higher than the isentropic index of compression
c) Power input will be smaller than that of a reversible, isentropic process
d) Discharge temperature will be higher than isentropic discharge temperature

Ans. a) and c)

7. As the speed of the compressor increases:

a) Heat transfer rate from compressor increases
b) Heat transfer rate from compressor decreases
c) Pressure drops increase and leakage losses decrease
d) Pressure drops decrease and leakage losses increase

Ans. b) and c)

8. On-off control is generally used only in small refrigeration capacity systems as:

a) Variation in refrigerated space temperature may be acceptable in smaller systems
b) Frequent start-and-stops can be avoided in small systems
c) It is simple and inexpensive
d) All of the above

Ans. a) and c)

9. Hot gas bypass to compressor inlet:

a) Provides an effective means of capacity control
b) Is an energy efficient method
c) Leads to increased discharge temperature
d) Provides effective cooling in hermetic compressor

Ans. a) and c)

3. A reciprocating compressor is to be designed for a domestic refrigerator of 100 W cooling capacity. The refrigerator operates at an evaporator temperature of –23.3^{o}C and a condensing temperature of 54.4^{o}C. The refrigeration effect at these conditions is 87.4 kJ/kg. At the suction flange the temperature of the refrigerant is 32^{o}C and specific volume is 0.15463 m^3/kg. Due to heat transfer within the compressor the temperature of the refrigerant increases by 15^{o}C. The indicated volumetric efficiency of the compressor is 0.85 and the leakage loss factor is 0.04. The rotational speed of the compressor is 2900 RPM. Find a) The diameter and stroke of the compressor in cms; b) Find the COP of the system if the actual mean effective pressure of the compressor is 5.224 bar.

Given:	Cooling capacity, Qe	= 100 W = 0.1 kW
	Evaporator Temperature, Te	= -23.3^{o}C
	Refrigeration effect, qe	= 87.4 kJ/kg
	Temperature at suction flange, Ts	= 32^{o}C
	Sp. vol. of vapour at flange, v_s	= 0.15463 m^3/kg
	Temperature rise in compressor	= 15^{o}C
	Indicated volumetric efficiency, $\eta_{V,th}$	= 0.85
	Leakage losses, ξ_L	= 0.04
	Mean effective pressure, mep	= 5.224 bar
	Rotational speed of compressor, N	= 2900 rpm

Find: a) Diameter and stroke length of compressor
b) COP

Ans:

a) The mass flow rate of refrigerant, m

m = refrigeration capacity/refrigeration effect
= (0.1/87.4) = **1.1442 X 10^{-3} kg/s**

Volumetric flow rate at suction flange, V_r

V_r = m X v_s = **1.7693 X 10^{-4} m^3/s**

Required compressor displacement rate, $V_{SW} = V_r/\eta_{V,act}$

Actual volumetric efficiency, $\eta_{V,act}$:

$$\eta_{V,act} = \eta_{V,th}\frac{T_s}{T_{sc}} - \xi_L = 0.85\frac{(273.15+32)}{(273.15+32+15)} - 0.04 = 0.77$$

Required compressor displacement rate, $V_{SW} = V_r/\eta_{V,act} = 1.7693 \text{ X } 10^{-4}/0.77$
$= \mathbf{2.298 \text{ X } 10^{-4} \text{ m}^3/\text{s}}$

The compressor displacement rate is equal to:

$$\dot{V}_{SW} = n\left(\frac{\pi D^2 L}{4}\right)\left(\frac{N}{60}\right) = n\left(\frac{\pi D^3 \theta}{4}\right)\left(\frac{N}{60}\right)$$

where n is the number of cylinders and θ is the stroke-to-bore ratio (L/D)

Since the refrigeration capacity is small, we can assume a single cylinder compressor, i.e., **n = 1**

Assuming a stroke-to-bore ratio θ **of 0.8** and substituting the input values in the above expression, we obtain:

Diameter of cylinder, D = 0.01963 m = **1.963 cm,** and
Stroke length, L = 0.8D = **1.5704 cm**

b) COP:

Actual power input to the compressor, W_c

W_c = mep X displacement rate = 5.224X100X**2.298X10^{-4} = 0.12 kW**

Hence, COP = (0.1/0.12) = 0.833