UNIT 5 REFRIGERATION SYSTEMS

Structure

5.1 INTRODUCTION

Refrigeration systems refer to the different physical components that make up the total refrigeration unit. The different stages in the refrigeration cycle are undergone in these physical systems. These systems consist of an evaporator, a condenser, a compressor and an expansion valve.

The evaporator is the space that needs to be cooled by the refrigerant; the compressor compresses the refrigerant from the low pressure of the evaporator to the pressure at the condenser. The heat gained by the refrigerant is rejected at the condenser and the high pressure refrigerant is expanded into the low pressure evaporator by the expansion valve. This is a very general representation of the various units in a refrigeration system.

The refrigeration systems vary according to the purpose and the type of refrigerant used. They are the means by which we can actually carry out the refrigeration process. A better understanding of them is thus, very essential.

Objectives

After studying this unit, you should be able to know

- the various types of refrigeration systems in common use along with their relative advantage and disadvantages, and
- to improve the performances of these systems.

5.2 VAPOUR COMPRESSION SYSTEMS

The challenge in refrigeration and air conditioning is to remove heat from a low temperature source and dump it at a higher temperature sink. Compression refrigeration cycles in general take advantage of the idea that highly compressed fluids at one temperature will tend to get colder when they are allowed to expand. If the pressure change is high enough, then the compressed gas will be hotter than our source of cooling (outside air, for instance) and the expanded gas will be cooler than our desired cold temperature. In this case, we can use it to cool at a low temperature and reject the heat to a high temperature.

Vapour-compression refrigeration cycles specifically have two additional advantages. First, they exploit the large thermal energy required to change a liquid to a vapour so we can remove lots of heat out of our air-conditioned space. Second, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to the temperature of whatever is being cooled. This is a benefit because the closer the working fluid temperature approaches that of the surroundings, the lower the rate of heat transfer. The isothermal process allows the fastest rate of heat transfer

Vapour compression refrigeration is the primary method to provide mechanical cooling. All vapor compression systems consist of the following four basic components alongwith the interconnecting piping. These are the evaporator, condenser, compressor and the expansion valve. Typical vapor compression systems can be represented as shown in figure 5.1a.

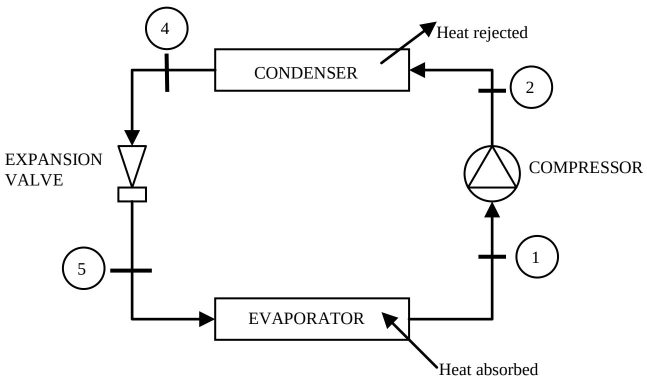


Figure 5.1 (a) Schematic Representation of a Vapour Compression System

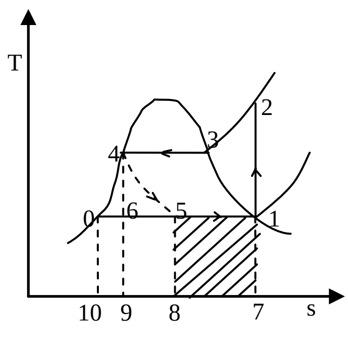


Figure 5.1 (b) T-*S* Diagram

The evaporator and the condenser are heat exchangers that evaporate and condense the refrigerant while absorbing and rejecting the heat. The compressor takes the refrigerant from the evaporator and raises the pressure sufficiently for the vapor to condense in the condenser. The expansion device controls the flow of condensed refrigerant at this higher pressure back into the evaporator. Some typical expansion devices are throttle valves, capillary tubes and thermostatic expansion valves in case of large refrigeration systems.

Figure 5.1 (b) shows the *T-S* plot of the working of such a system. Here, the dry saturated working medium at state 1 is compressed isentropically to state 2. Constant pressure heat transfer occurs from state 2 until the compressed vapor becomes saturated liquid or condensate at state 4. The compressed vapor is next throttled from the high pressure region in the condenser (state 4) to the low pressure region in the evaporator (state 5). Since throttling is an irreversible process, it is represented by a broken line. After throttling to evaporator pressure, the heat transfer in the evaporator causes vaporization of the working medium until state 1 is reached, thus completing the cycle.

Referring to Figure 5.1 (b), considering ideal processes, we can see that:

$q_{1-2} = 0$, being an isentropic process and,

$$q_{2-4} = \int_2^4 \delta q = \int_2^4 dh = h_4 - h_2 = -(h_2 - h_4)$$

The negative sign in the above equation represents heat transfer from the system to the surroundings.

The process *4-5* is assumed to be adiabatic during throttling, an isenthalpic process.

$$q_{4-5} = \int_4^5 dh = 0, i.e., h_4 = h_5$$

$$q_{5-1} = \int_5^1 \delta q = \int_5^1 dh = h_1 - h_5$$

The heat transfer during *5-1* is the required refrigeration effect.

Again, using first law of thermodynamics, we get:

$$w = \oint \delta q = q_{1-2} + q_{2-4} + q_{4-5} + q_{5-1}$$

$$= 0 - (h_2 - h_4) + (h_1 - h_5) = -(h_2 - h_1)$$

where the negative sign indicates that work is done on the system in order to execute the cyclic process. Then,

$$\text{COP} = \frac{\text{net refrigerating effect}}{|\text{work done}|} = (h_1 - h_4)/(h_2 - h_1)$$

In the above equation, the quantities h_1 and h_4 are known for respective pressures p_1 (evaporator pressure) and p_2 (condenser pressure). The state h_2 is found from the intersection of constant entropy line passing through state 1 and pressure line p_2.

5.3 CARNOT VAPOR COMPRESSION SYSTEMS

Here, the compression is imagined to take place in two stages: isentropic compression upto state 2 and isothermal compression from state 2 to 3 as shown in Figure 5.2 (b).

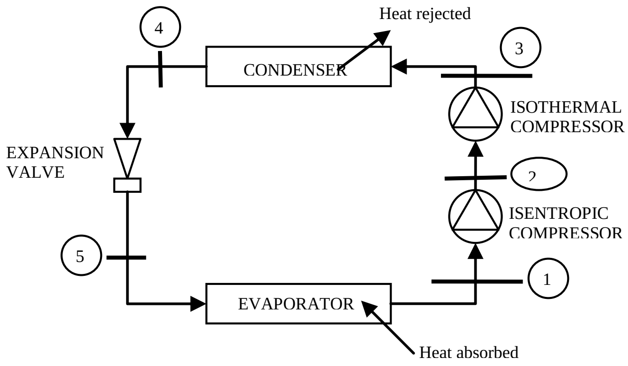


Figure 5.2 (a) Schematic Representation of a Carnot Vapour Compression System

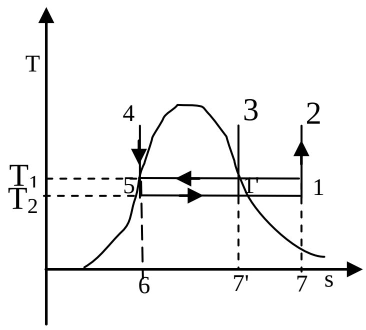


Figure 5.2 (b) T-*S* Diagram

The working medium is condensed in a heat exchanger giving saturated liquid at state 4. The isentropic expansion from state 4 to state 5 gives the refrigeration effect, the area under line 5-1.

Comparing figs. 5.1 and 5.2, we can see that the Carnot vapor compression cycle gives a greater refrigeration effect than the vapor compression cycle.

The COP of a Carnot vapor compression cycle is given by:

$$q_c = \frac{\text{refrigerating effect}}{\text{work input}} = \frac{\dot{Q}_c / \dot{m}}{\dot{W}_{net}} = \frac{\text{area 5-1-7-6}}{\text{area 1-2-3-4-5}}$$

$$\Rightarrow q_c = \frac{T_2 \left(s_1 - s_5 \right)}{\left(T_1 - T_2 \right) \left(s_1 - s_5 \right)}$$

$$\Rightarrow q_c = \frac{T_2}{T_1 - T_2}$$

where, T_1 is the temperature at which heat is rejected in the condenser and T_2 is the temperature maintained in the evaporator. It can be seen that the refrigeration system working on the Carnot vapor compression cycle has the highest COP.

5.4 LIMITATIONS OF CARNOT VAPOR COMPRESSION SYSTEMS WITH VAPOR AS REFRIGERANT

Although in theory, the Carnot vapor compression cycle has the highest COP; it is not suited for use in practical refrigeration systems. This is because it is virtually impossible to compress the refrigerant isothermally from state 2 to state 3 in a finite time interval. To offset this difficulty, we can follow the alternate path 1'-3-4-5. However, this results in other difficulties which are mentioned in detail below:

5.4.1 Dry vs. Wet Compression

If the Carnot vapour cycle follows the path 1-2-3-4, then there is dry compression of the refrigeration vapor since the refrigerant is dry saturated at state 1. This type of compression is desirable in the compressor. But, in this case we see that the refrigerant now has to be compressed isothermally from state 2 to state 3, which is impossible to achieve in practice. The alternate path 1'-3-4-5 involves a wet compression of the vapor from state 1' to state 3. Wet compression is highly undesirable as the compressor now has to deal with two different fluid phases. Besides, the liquid droplets present in the vapor would now react with the lubricant in the compressor which is highly undesirable. Thus, we see that both the paths of the Carnot vapor cycle are not suitable for use in practical refrigeration systems.

5.4.2 Throttling vs. Isentropic Compression

In the Carnot vapour compression cycle, there is isentropic expansion from state 4 to state 5. This is achieved by the use of a turbine. However, in actual cycles, the expansion from saturated liquid at state 4 to liquid-vapor mixture at state 5 produces very little work. A turbine working under such conditions would have very low efficiency which would not justify the cost involved in using a turbine. Also, the refrigeration system would become very bulky and not suitable for domestic use.

In actual practice, an expansion valve is used to achieve the desired expansion from state 4 to state 5. The refrigerant gets throttled in the expansion valve from saturated liquid to liquid-vapor mixture. The expansion no longer remains isentropic. The expansion now becomes an isenthalpic process.

Thus, we see that the Carnot vapour refrigeration cycle is not suitable for use in refrigeration systems. A better ideal cycle is the vapor compression refrigeration cycle.

5.5 STANDARD VAPOUR COMPRESSION CYCLE

The standard vapour compression refrigeration cycle would consist of the following:

(a) Low pressure saturated vapor would enter the compressor.

(b) Compression would be isentropic.

(c) Constant pressure heat rejection would occur at the condenser with the exit state of the refrigerant being saturated liquid.

(d) The throttling process would be adiabatic.

(e) Constant pressure heat transfer would take place in the evaporator.

All these ideal conditions can be shown by the following T-*s* diagram (Figure 5.3)

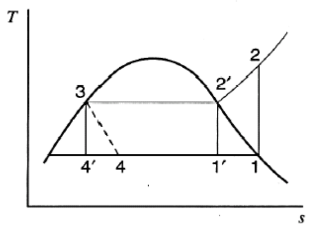


Figure 5.3 T-*S* Diagram of Ideal Vapour Compression Refrigeration Cycle

5.6 PRESSURE ENTHALPY DIAGRAM

The refrigeration industry did not always have the analysis tools that are available today. For many decades, the manufacturers and technicians relied on the graphical and tabulated values of refrigerant properties and expected equipment performance. One of their favourite tools was the pressure-enthalpy diagram which defines the thermodynamic properties for the refrigerant in use and the performance of the equipment.

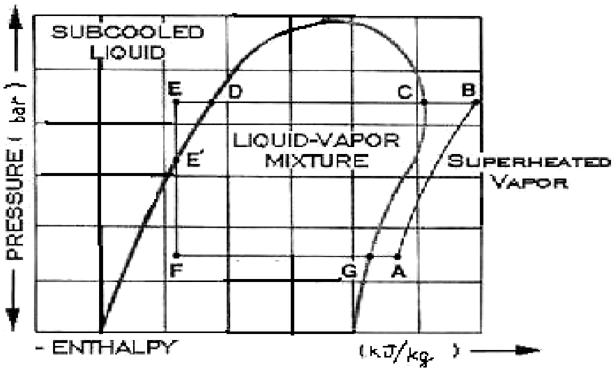


Figure 5.4 Pressure-Enthalpy Diagram along with the Various Components of the Vapour Compression System

This chart (Figure 5.4) shows the pressure expressed in bar along the vertical axis. The energy content or enthalpy of the refrigerant is shown along the horizontal axis in kJ/kg. The above pressure-enthalpy chart is typical of the refrigerant R22, a common refrigerant in small refrigeration systems.

A closer analysis of the chart shows that there are distinct regions separated by three “boundary lines”. The region on the left is subcooled liquid. This is the refrigerant liquid at a temperature lower than the equivalent boiling point for the pressure noted.

The region inside the “dome” is a liquid-vapor mixture. If the liquid is at the boiling point, but just hasn't begun to boil, it is defined as saturated liquid. Adding any heat to this liquid will vaporize a portion of it. Adding more heat to the liquid-vapor mixture eventually evaporates all of the liquid. At some precise point (G), the vapor is fully saturated. Adding any more heat to the vapor will cause it to rise in temperature further; this is referred to as superheated vapor.

It is a general tendency to believe that a superheated vapor is “hot”. This is not always the case. Superheated vapours can be cold. By the term superheated, we simply mean that they are above the corresponding saturated vapor point. Similarly, subcooled liquid can be generally warm. It just means that the liquid is cooler than the saturation line at that pressure. We now present a detailed study of the pressure-enthalpy diagram.

We consider the refrigerant to be initially at point A. To reach this point after leaving the evaporator at G, the refrigerant is heated slightly and crosses the compressor suction valve to point A. The compressor elevates the refrigerant's pressure to a point at which it can push the discharge valve open and flow into the condenser. The refrigerant vapor leaves the compressor at point B, desuperheats to point C, and then begins to condense. After the vapor is completely condensed at point D, it is subcooled a bit further (E), at which time it is still at a much higher pressure than the evaporator.

Controlling the flow to the evaporator and throttling to the pressure of the evaporator is the job performed by the expansion device, a capillary tube or a throttling valve in small refrigeration systems. This pressure reduction step vaporizes a portion of the liquid which cools (called flash gas) the remaining liquid going to point F. The "average" mixture of vapor and liquid crossing the valve doesn't change in energy content. It simply separates into liquid and vapor at the reduced temperature and pressure according to its precise thermodynamic properties. The liquid at point F is then ready to pick up heat in the evaporator and form vapor at point G where the cycle repeats itself.

Example 1.1

An R12 refrigeration system works between pressure limits 1.83 bar and 9.63 bar respectively. The heat transfer from the condenser is found to be 80 kJ/min.The refrigerant vapor leaves the evaporator in the saturated state. The condensate leaves the condenser in just saturated state. The refrigerant flow through the system is found to be 0.6 kg/min. Find a) COP, b) capacity of the plant and c) the energy input to the compresser.

Solution

From the saturation property tables or p-h diagram, the enthalpies are obtained as:

$$h_1 = 181.94\,\text{kJ/kg}, \quad h_4 = 74.77\,\text{kJ/kg}$$

The refrigerating effect is found from:

$$q_c = h_1 - h_4 = 181.94 - 74.77 = 107.17\,\text{kJ/kg}$$

Then,

$$\dot{Q}_c = \dot{m} q_c = 0.6 \times 101.17 = 64.30\,\text{kJ/min}$$

From the first law:

$$\dot{W} = \oint \delta \dot{Q} = \dot{Q}_{evaporator} - \dot{Q}_{condenser} = 64.30 - 80 = -15.70\,\text{kJ/min}$$

The negative sign indicates work input to the compressor.

(a) COP = $\dot{Q}_c / \dot{W} = 64.30/15.70 = 4.09$

(b) Capacity = $\dot{Q}_c / 210 = 64.30/210 = 0.306\,\text{ton}$

(c) Power input = $\dot{W} = 15.70/60 = 262\text{W}$

5.7 EWING'S METHOD FOR SUCTION STATE WITH RESPECT TO MAXIMUM COP

The COP of a simple vapor compression system may be achieved close to that of the ideal value by using (a) an appropriate refrigerant and (b) selecting operating limits as close to each other as possible. The Ewing's method is based on the fact that as the point of compression proceeds away from state 5 (Figure 5.5), the COP increases. If the compression starts right from state 5, the COP would be zero. Referring to the figure, we see that COP is:

$$\text{COP} = (h_1 - h_4)/(h_2 - h_1) = T_1 \delta s / (h_2 - h_4 - T_1 \delta s)$$

$$= T_1 / [(h_2 - h_4)/\delta s - T_1]$$

The slope $(h_2 - h_4)/\delta s$ will be minimum corresponding to the tangent to the condensing pressure line.

$$\mathrm{COP}_{max} = T_1 / [\{(h_2 - h_4)/\delta s\}_{min} - T_1]$$

The quantity $[(h_2 - h_4)/\delta s]_{min}$ is determined by drawing a line from state 5, tangent to constant pressure line $p_{2.}$ in the present case, this minimum value is $[(h_2 - h_4)/\delta s]_{min} = (h_{2'} - h_4)/(s_1 - s_5)'$ Because in wet compression in high speed compressors it would not be possible to evaporate tiny liquid droplets, Ewing's method is no longer used. Hence, the performance not only would not match that stated above, but there would also be serious mechanical difficulties.

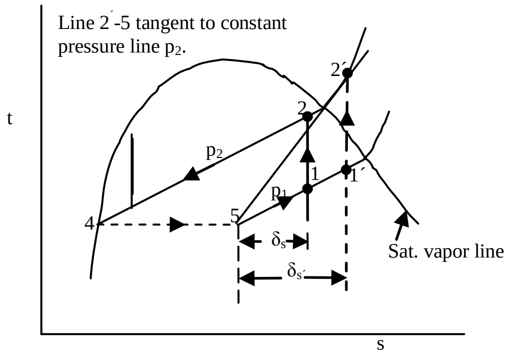


Figure 5.5 Ewing's Diagram

5.8 STANDARD RATING CYCLE AND EFFECT OF OPERATING CONDITIONS

The standard rating cycle corresponds to the ideal refrigeration cycle shown in figure 5.3. However, different operating conditions have different effects on the vapor compression cycle. We study these effects in detail below.

5.8.1 Effect of Evaporator Pressure

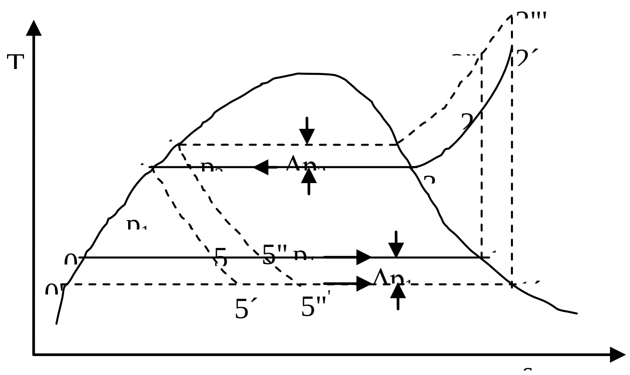


Figure 5.6 Effect of Evaporator and Condenser Pressure Drops

As shown in Figure 5.6, a decrease in evaporator pressure p_1 causes the states 5 and 1 to shift to states 5′ and 1′ respectively. As a result, the work input to the compressor increases by areas 0′-1′-1-0 and 1-1′-2′-2. The COP and capacity of the refrigeration system thus decreases.

5.8.2Effect of Condenser Pressure

In this case, the refrigeration effect is reduced by an area on the T-S diagram (fig 5.6) under the process 5-5″. At the same time, the condenser work increases by an area 2-2"-4'-4 compared to the normal cycle. Thus, the COP and refrigeration effect are reduced compared to the ideal cycle. Further, the work input has increased but is somewhat less than the increase when there is a change in the evaporator pressure.

The effects of reduced evaporator pressure and increased condensers pressure are easily seen in Figure 5.6 b[1] and c on p – h diagrams respectively.

5.8.3Effect of Suction Vapour Superheat

It is a usual practice to admit slightly superheated vapor before the beginning of compression to avoid the possibility of wet compression. Wet compression is undesirable as there may be accumulation of liquid inside the cylinder, which in turn will wash away the lubricant resulting in sever mechanical difficulties. Thus, to avoid this, a 5 to 20 K superheat of the refrigerant is always desirable. The vapor superheat has the following effects on the refrigeration cycle:

(a) It increases the refrigeration effect per unit mass of the refrigerant from $h_1 - h_4$ to $h_1' - h_{4'}$.

(b) The specific volume increases from v_1 to $v_{1'}$. This implies the reduction in mass flow rate for the same displacement volume of the compressor.

(c) The energy for compression of refrigerant vapor will increase due to the diverging nature of the isentropic lines.

The above mentioned effects of superheating of the vapor are shown in the Figure 5.7.

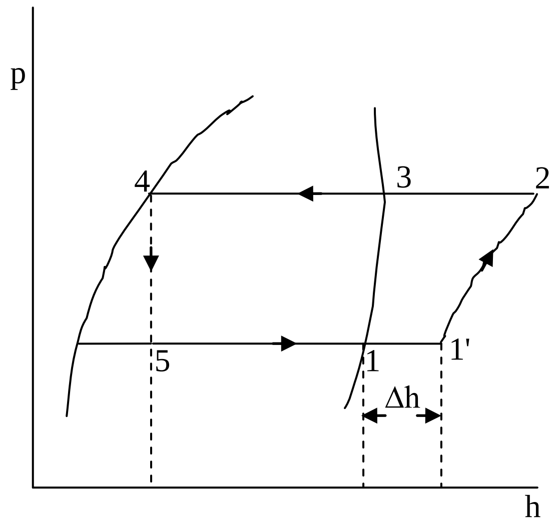


Figure 5.7 Vapour Superheating Before Suction

5.8.4 Effect of Liquid Subcooling

It is always desirable to subcool the refrigerant before throttling. This effect is shown in the following Figure (Figure 5.8(a) and 5.8(b)).

It is seen from these figures that the refrigeration effect is increased by an area under the process 5'-5.

The magnitude of subcooling is usually kept between 5 to 10 K. If there is too much subcooling, then the work input increases considerably, thus offsetting the gains from the subcooling of the refrigerant.

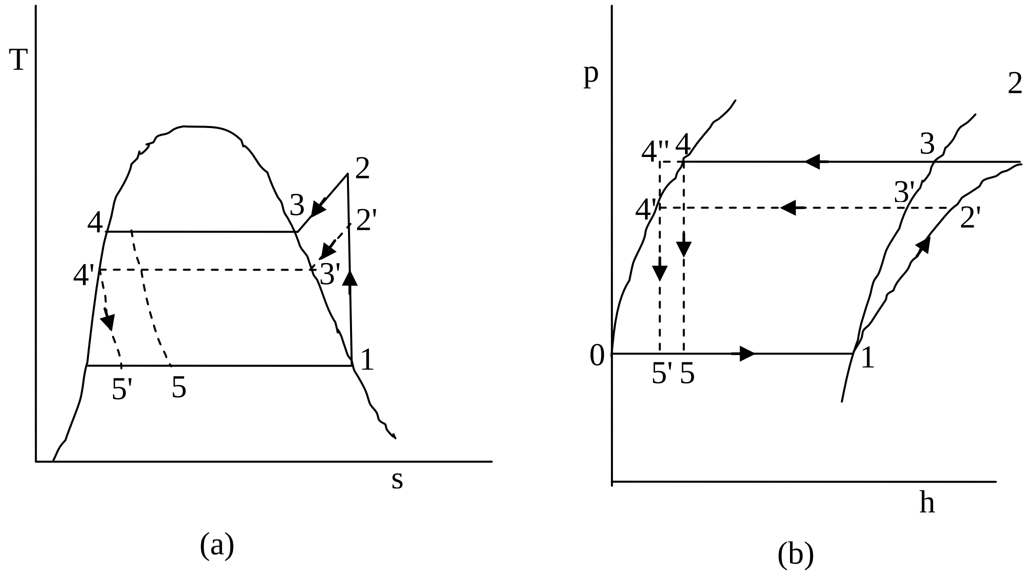


Figure 5.8 Effect of Condensate Subcooling on COP

5.8.5 Using Liquid-Vapour Heat Exchanger

When the vapour leaving the evaporator is heated up by the condensate, the temperature of condensate decreases from T_4 to $T_{4'}$ (figure 5.9). The heat exchanger is essentially a concentric type counter-flow heat exchanger which causes the subcooling of the refrigerant before throttling. There is no change in the refrigerating effect. This is done as the heat transfer from the outside is highly undesirable since it increases the energy input to the compressor and the vapor superheating at the discharge of the compressor.

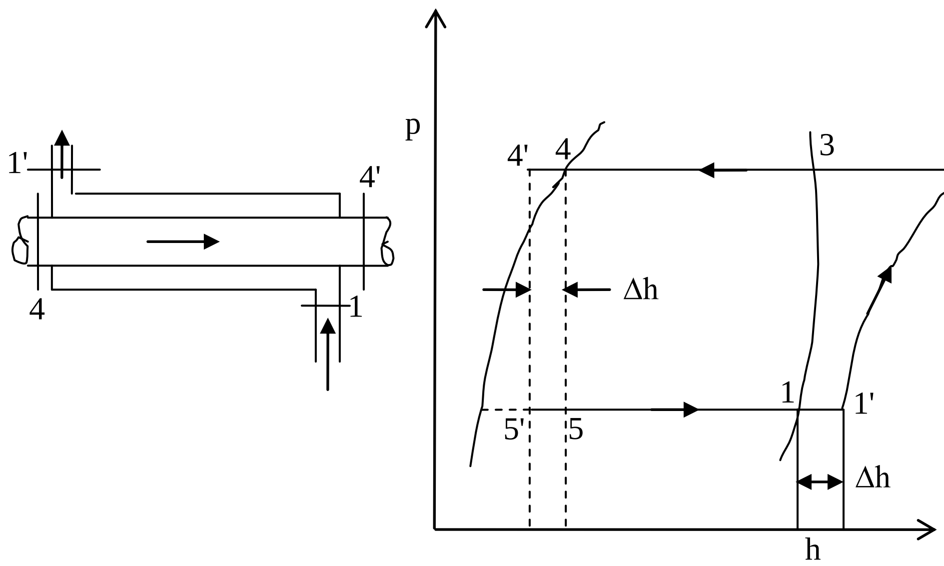


Figure 5.9 Superheating of Vapour in a Counter-Flow Heat Exchanger

5.9 ACTUAL VAPOR COMPRESSION CYCLE

For thermodynamic analysis of the refrigeration systems, we consider the ideal behaviour of the fluid and neglect any extraneous factors. In actual practice however, these factors have significant effect on the COP and refrigerating effect of the refrigeration cycle. In this section, we go for a detailed analysis of these external factors and their influence on the overall system performance.

The actual vapor compression cycle is shown in Figure 5.10. With the help of a T-*s* diagram for the operating pressure limits p_1 and p_2. The actual compression follows 1 – 2′ instead causing considerable difference due to the irreversibilities associated with the compression process 1 – 2. Again, during condensation and subcooling by about 5 to 10 K (2'-4) in the condenser, the pressure before throttling drops about of 2%. The throttled condensate enters the evaporator at a pressure somewhat higher than the evaporator pressure p_1. These irreversibilities cause significant distortion in the ideal cycle, thereby indicating a significant reduction in the COP. Also, if we consider the suction and discharge losses in the compressor, the COP value would decrease further.

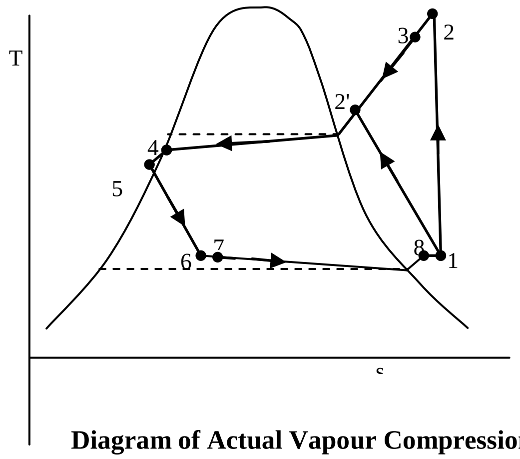


Figure 5.10 T-*S* Diagram of Actual Vapour Compression Refrigeration Cycle

5.10 MULTISTAGE VAPOUR COMPRESSION SYSTEM

Multistage vapor compression refrigeration system can be studied under the following heads:

5.10.1 Multistage Compression System

Whenever a gas is compressed from a given pressure to a higher pressure in one stage, it takes more work for compressing the same mass and needs a larger swept volume of the compressor than when more than one stage is used. If the compression ratio is very large, then the compressor may stop delivery of the fluid. Single stage compressors are therefore designed with the compression ratio around 7 to 9. However, for high capacity refrigeration systems, even at this pressure ratio, multistage compression is used.

Multistage compression results in a higher volumetric efficiency. This means that the compressor capacity has increased as compared to a single stage compressor having the same stroke and clearance volume.

The working of a multistage compression system is discussed with the help of the following T-*s* diagram (fig 5.10 (a) and (b)). The refrigerant from the evaporator at state 1 is compressed by the low pressure (LP) compressor to the flash intercooler. The vapor is desuperheated from state 2 to state 3. It combines with the vapor separated by throttling from state 5-6 and this increased mass of vapor is now delivered to the high pressure (HP) compressor. The compressed vapor from the HP compressor is delivered at state 4. This vapor is now condensed in the condenser before being throttled. The liquid at the intermediate pressure p_i is again throttled to the evaporator pressure. The heat transfer to the evaporator causes the vaporization of the refrigerant, thus completing the cycle.

5.10.2 Multistage Evaporator System

In multistage evaporator system, the refrigerant is used to provide cooling effect in more than one refrigeration space. This is used when we have to maintain two or more refrigerated spaces at different temperatures and the size of the refrigerated space is such that the setting up of individual refrigeration systems for each one of them is not justified.

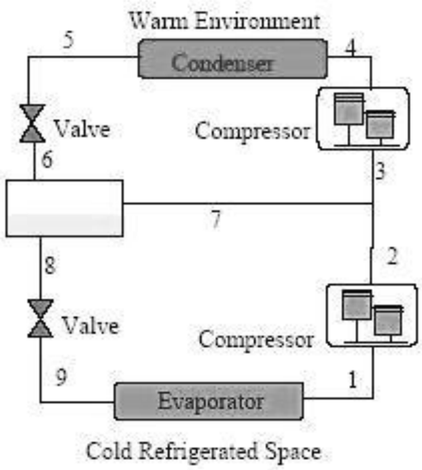


Figure 5.11 (a) Schematic Representation of Multistage Compression System

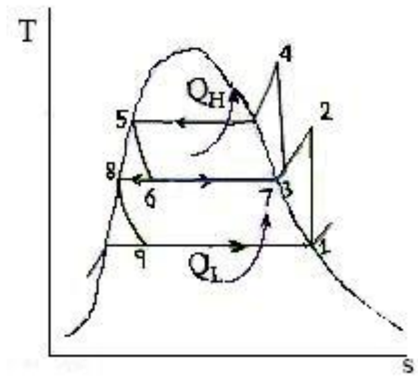


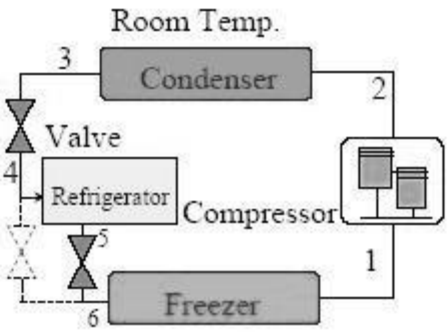
Figure 5.11 (b) T-*s* Plot of Multistage Compression System

The schematic representation of this type of system is given in Figure 5.11(a) alongwith the corresponding T – S diagram in Figure 5.11 (b).

The refrigerant is first throttled from state 3-4 and then passes into the first evaporator space. The refrigerant coming out the evaporator is again throttled to a lower pressure and then enters the next evaporator (freezer in figure). The refrigerant coming out of the freezer is then compressed in the compressor from state 1 to state 2. The rest of the cycle is similar to a simple vapor compression refrigeration cycle.

5.11 CASCADE REFRIGERATION SYSTEM

Figure 5.12 (a). is a schematic representation of a cascade refrigeration system alongwith its T-S diagram (Figure 5.12 (b)). A cascade refrigeration system can be considered to be equivalent to two independent vapor-compression systems linked together in such a way that the evaporator of the high-temperature system becomes the condenser of the low-temperature system. However, the working media of the two systems are separated from each other. This therefore, allows the use of different refrigerants working at different temperature ranges to achieve the desired effect, which would otherwise, need to be achieved by a single refrigerant working at a bigger operating pressure range. Thus, we can make use of suitable refrigerants at the higher and lower pressure ranges to derive maximum benefit.



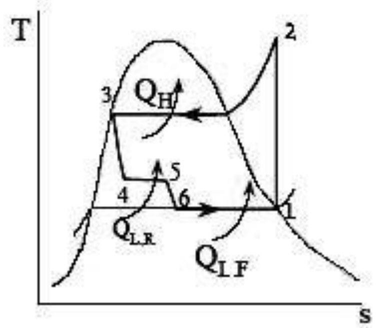


Figure 5.12 (a) Schematic Representation of Multistage Evaporator System

Figure 5.12 (b) T-*s* plot of Multistage Evaporator System

There are many industrial and medical applications which necessitate the use of the cascade refrigeration system. One such example is the storage of blood in blood banks. Blood needs temperatures as low as – 80 ^{0}C for proper storage.

SAQ 1

(a) Describe a Carnot vapour compression cycle and discuss it carnot be used in practice.

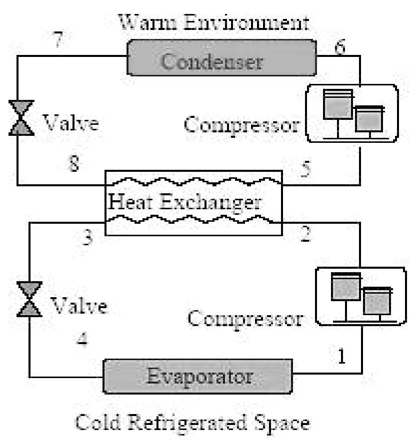
(b) Show a standard vapour compression refrigeration cycle on T – S and p – h diagrams and discuss the effects of decreasing evaporator pressure and increasing condenser pressure separately.

(c) How do the following two factors influence the performance of a vapour – compression refrigeration system?

   (i) Section vapour superheat and

   (ii) Liquid subcooling

(d) What is a casead refrigeration system? Compare this system with multistorage compression and evaporation.



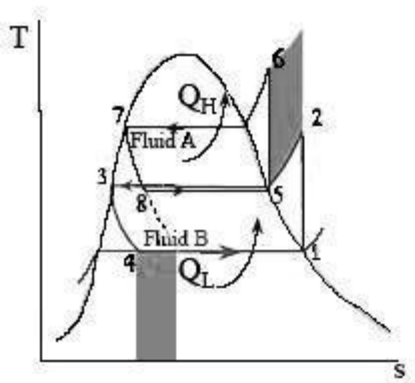


Figure 5.13 (a) Schematic Representation of Cascade Refrigeration System

Figure 5.13 (b) T-*s* plot of Cascade Refrigeration System

5.12 SUMMARY

This chapter has dealt with the study of the vapor compression refrigeration cycle. This cycle is one of the most widely used refrigeration cycle when designing the refrigeration systems.

This unit also details the mathematics involved in the detailed study of the refrigeration systems. The p-h and T-S diagram are the most widely used in the analysis of the refrigeration systems. The T-S diagram is of general importance in the field of thermodynamics. In this chapter, we have studied the p-h diagram in details. The p-h chart is the most widely used chart in the refrigeration industry and a complete understanding of this diagram is therefore, imperative in the study of refrigeration systems. We have discussed this p-h diagram with a few sample problems.

5.13 ANSWERS TO SAQs

Refer the relevant preceding texts in the unit or other useful books on the topic listed in the section “Further Readings” to get the answers of the SAQs.