

Lesson

10

Vapour Compression Refrigeration Systems

The specific objectives of the lesson:

This lesson discusses the most commonly used refrigeration system, i.e. Vapour compression refrigeration system. The following things are emphasized in detail:

1. The Carnot refrigeration cycle & its practical limitations (*Section 10.3*)
2. The Standard Vapour compression Refrigeration System (*Section 10.4*)
3. Analysis of Standard Vapour compression Refrigeration System (*Section 10.5*)

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

1. Analyze and perform cyclic calculations for Carnot refrigeration cycle (*Section 10.3*)
2. State the difficulties with Carnot refrigeration cycle (*Section 10.3*)
3. Analyze and perform cyclic calculations for standard vapour compression refrigeration systems (*Section 10.4*)
4. Perform various cycle calculations for different types of refrigerants (*Section 10.4*)

10.1. Comparison between gas cycles and vapor cycles

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. As mentioned in the previous chapter, in a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

10.2. Vapour Compression Refrigeration Systems

As mentioned, vapour compression refrigeration systems are the most commonly used among all refrigeration systems. As the name implies, these systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems. Vapour compression refrigeration

systems are available to suit almost all applications with the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle. Before the actual cycle is discussed and analysed, it is essential to find the upper limit of performance of vapour compression cycles. This limit is set by a completely reversible cycle.

10.3. The Carnot refrigeration cycle

Carnot refrigeration cycle is a completely reversible cycle, hence is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. It is used as reference against which the real cycles are compared. Figures 10.1 (a) and (b) show the schematic of a Carnot vapour compression refrigeration system and the operating cycle on T-s diagram.

As shown in Fig.10.1(a), the basic Carnot refrigeration system for pure vapour consists of four components: compressor, condenser, turbine and evaporator. Refrigeration effect ($q_{4-1} = q_e$) is obtained at the evaporator as the refrigerant undergoes the process of vaporization (process 4-1) and extracts the latent heat from the low temperature heat source. The low temperature, low pressure vapour is then compressed isentropically in the compressor to the heat sink temperature T_c . The refrigerant pressure increases from P_e to P_c during the compression process (process 1-2) and the exit vapour is saturated. Next the high pressure, high temperature saturated refrigerant undergoes the process of condensation in the condenser (process 2-3) as it rejects the heat of condensation ($q_{2-3} = q_c$) to an external heat sink at T_c . The high pressure saturated liquid then flows through the turbine and undergoes isentropic expansion (process 3-4). During this process, the pressure and temperature fall from P_c, T_c to P_e, T_e . Since a saturated liquid is expanded in the turbine, some amount of liquid flashes into vapour and the exit condition lies in the two-phase region. This low temperature and low pressure liquid-vapour mixture then enters the evaporator completing the cycle. Thus as shown in Fig.10.1(b), the cycle involves two isothermal heat transfer processes (processes 4-1 and 2-3) and two isentropic work transfer processes (processes 1-2 and 3-4). Heat is extracted isothermally at evaporator temperature T_e during process 4-1, heat is rejected isothermally at condenser temperature T_c during process 2-3. Work is supplied to the compressor during the isentropic compression (1-2) of *refrigerant vapour* from evaporator pressure P_e to condenser pressure P_c , and work is produced by the system as *refrigerant liquid* expands isentropically in the turbine from condenser pressure P_c to evaporator pressure P_e . All the processes are both internally as well as externally reversible, i.e., net entropy generation for the system and environment is zero.

Applying first and second laws of thermodynamics to the Carnot refrigeration cycle,

$$\begin{aligned} \oint \delta q &= \oint \delta w \\ \oint \delta q &= q_{4-1} - q_{2-3} = q_e - q_c \\ \oint \delta w &= w_{3-4} - w_{1-2} = w_T - w_C = -w_{\text{net}} \end{aligned} \quad (10.1)$$

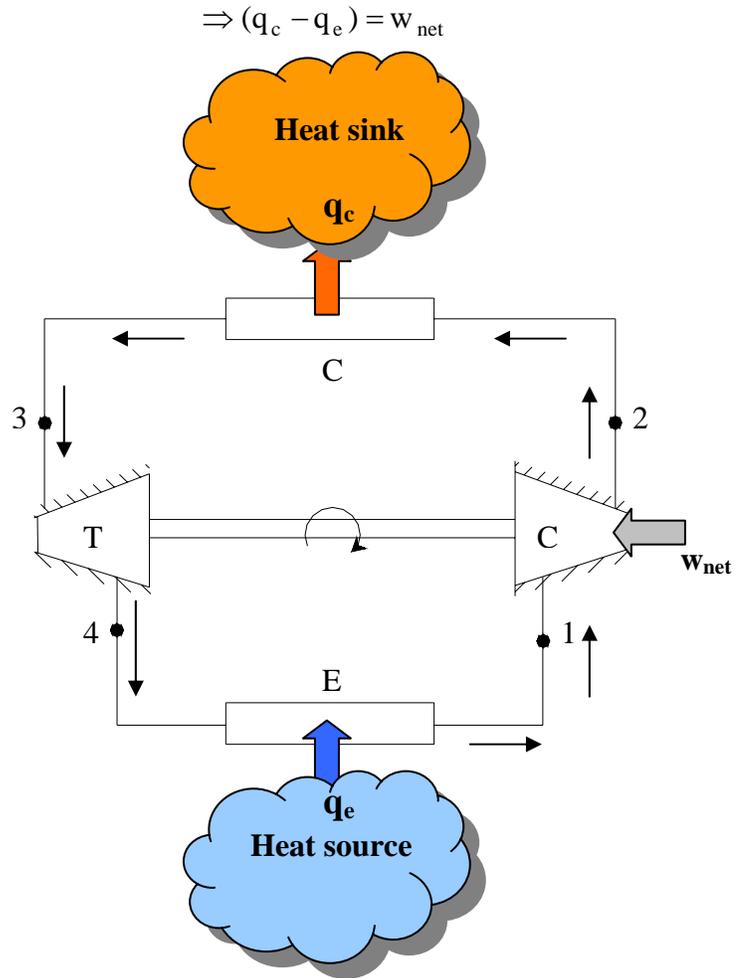


Fig.10.1(a): Schematic of a Carnot refrigeration system

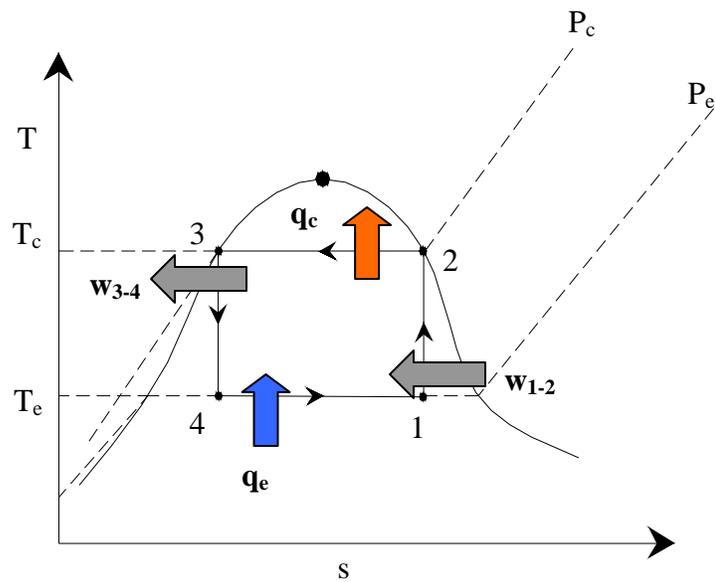


Fig. 10.1(b): Carnot refrigeration cycle on T - s diagram

now for the reversible, isothermal heat transfer processes 2-3 and 4-1, we can write:

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c(s_2 - s_3) \quad (10.2)$$

$$q_e = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4) \quad (10.3)$$

where T_e and T_c are the evaporator and condenser temperatures, respectively, and,

$$s_1 = s_2 \text{ and } s_3 = s_4 \quad (10.4)$$

the Coefficient of Performance (COP) is given by:

$$\text{COP}_{\text{Carnot}} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{w_{\text{net}}} = \frac{T_e(s_1 - s_4)}{T_c(s_2 - s_3) - T_e(s_1 - s_4)} = \left(\frac{T_e}{T_c - T_e} \right) \quad (10.5)$$

thus the COP of Carnot refrigeration cycle is a function of evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle (Lesson 9). The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP.

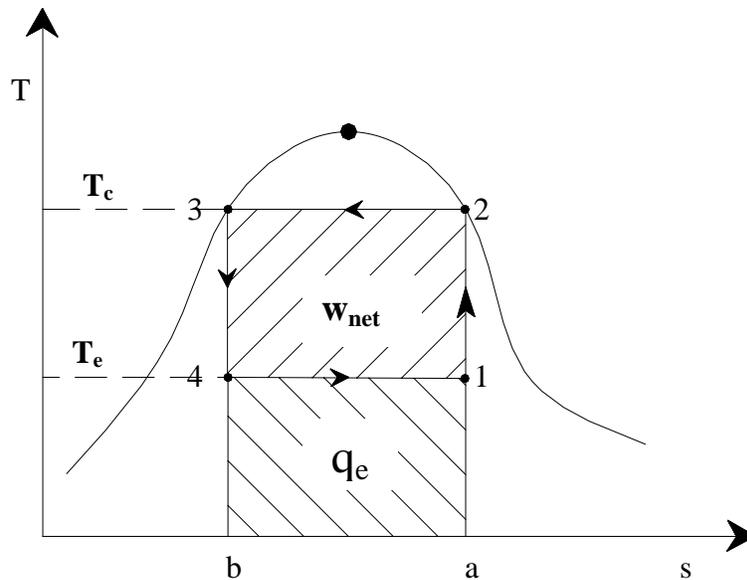


Fig.10.2. Carnot refrigeration cycle represented in T - s plane

It can be seen from the above expression that the COP of a Carnot refrigeration system increases as the evaporator temperature increases and condenser temperature decreases. This can be explained very easily with the help of the T-s diagram (Fig.10.2). As shown in the figure, COP is the ratio of area a-1-4-b to the area 1-2-3-4. For a fixed condenser temperature T_c , as the evaporator temperature T_e increases, area a-1-4-b (q_e) increases and area 1-2-3-4 (w_{net}) decreases as a result, COP increases rapidly. Similarly for a fixed evaporator temperature T_e , as the condensing temperature T_c increases, the net work input (area 1-2-3-4) increases, even though cooling output remains constant, as a result the COP falls. Figure 10.3 shows the variation of Carnot COP with evaporator temperature for different condenser temperatures. It can be seen that the COP increases sharply with evaporator temperatures, particularly at high condensing temperatures. COP reduces as the condenser temperature increases, but the effect becomes marginal at low evaporator temperatures. It will be shown later that actual vapour compression refrigeration systems also behave in a manner similar to that of Carnot refrigeration systems as far as the performance trends are concerned.

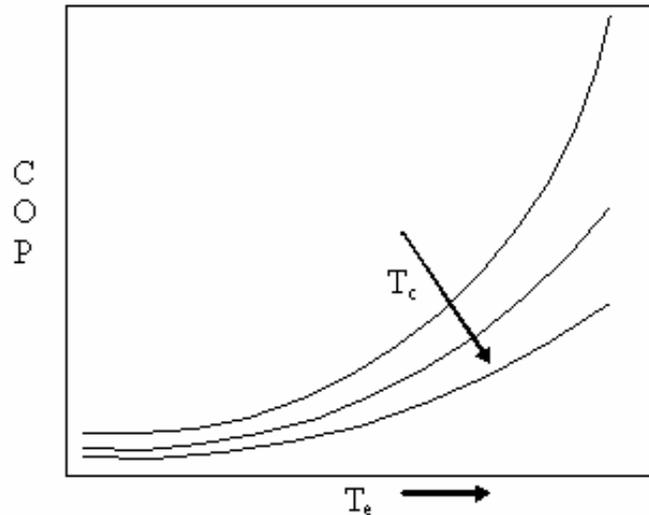


Fig.10.3. Effects of evaporator and condenser temperatures on Carnot COP

Practical difficulties with Carnot refrigeration system:

It is difficult to build and operate a Carnot refrigeration system due to the following practical difficulties:

- i. During process 1-2, a mixture consisting of liquid and vapour have to be compressed isentropically in the compressor. Such a compression is known as *wet compression* due to the presence of liquid. In practice, wet compression is very difficult especially with reciprocating compressors. This problem is particularly severe in case of high speed reciprocating compressors, which get damaged due to the presence of liquid droplets in the vapour. Even though some types of compressors can tolerate the presence of liquid in

vapour, since reciprocating compressors are most widely used in refrigeration, traditionally *dry compression* (compression of vapour only) is preferred to wet compression.

ii. The second practical difficulty with Carnot cycle is that using a turbine and extracting work from the system during the isentropic expansion of liquid refrigerant is not economically feasible, particularly in case of small capacity systems. This is due to the fact that the specific work output (per kilogram of refrigerant) from the turbine is given by:

$$w_{3-4} = \int_{P_e}^{P_c} v \cdot dP \quad (10.6)$$

since the specific volume of liquid is much smaller compared to the specific volume of a vapour/gas, the work output from the turbine in case of the liquid will be small. In addition, if one considers the inefficiencies of the turbine, then the net output will be further reduced. As a result using a turbine for extracting the work from the high pressure liquid is not economically justified in most of the cases¹.

One way of achieving dry compression in Carnot refrigeration cycle is to have two compressors – one isentropic and one isothermal as shown in Fig.10.4.

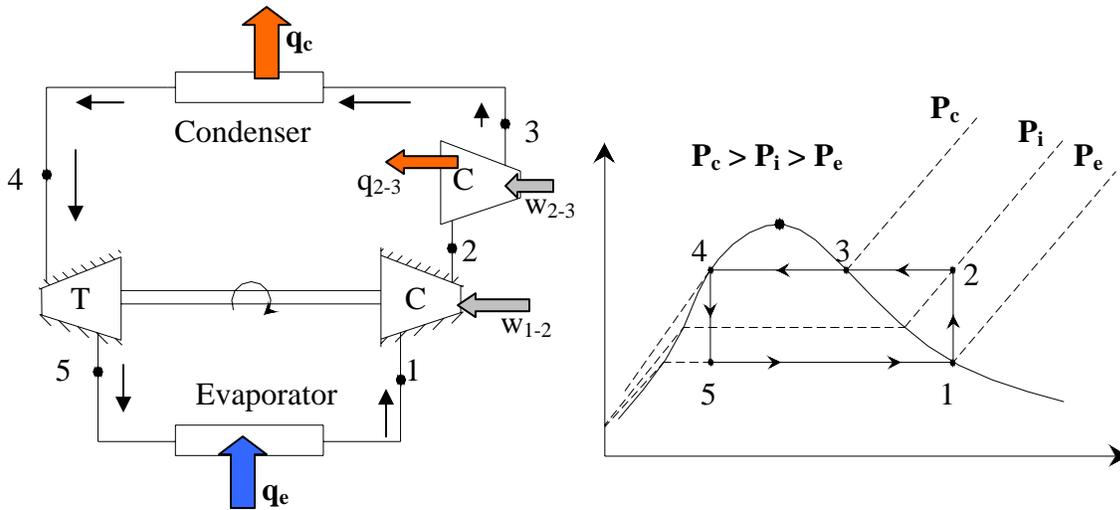


Fig.10.4. Carnot refrigeration system with dry compression

As shown in Fig.10.4, the Carnot refrigeration system with dry compression consists of one isentropic compression process (1-2) from evaporator pressure P_e to an intermediate pressure P_i and temperature T_c , followed by an isothermal compression process (2-3) from the intermediate pressure P_i to the condenser pressure P_c . Though with this modification the problem of wet compression can be avoided, still this modified system is not practical due to the difficulty in achieving true isothermal compression using high-speed compressors. In addition, use of two compressors in place of one is not economically justified.

¹ However, currently efforts are being made to recover this work of expansion in some refrigeration systems to improve the system efficiency.

From the above discussion, it is clear that from practical considerations, the Carnot refrigeration system need to be modified. Dry compression with a single compressor is possible if the isothermal heat rejection process is replaced by isobaric heat rejection process. Similarly, the isentropic expansion process can be replaced by an isenthalpic throttling process. A refrigeration system, which incorporates these two changes is known as Evans-Perkins or reverse Rankine cycle. This is the theoretical cycle on which the actual vapour compression refrigeration systems are based.

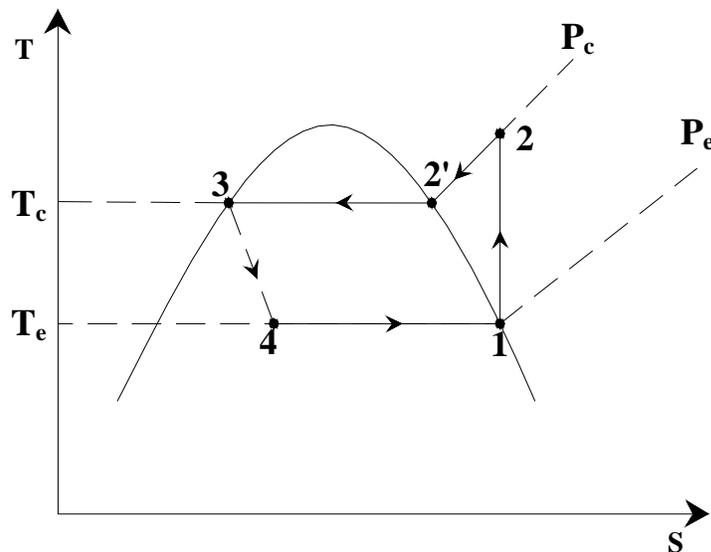
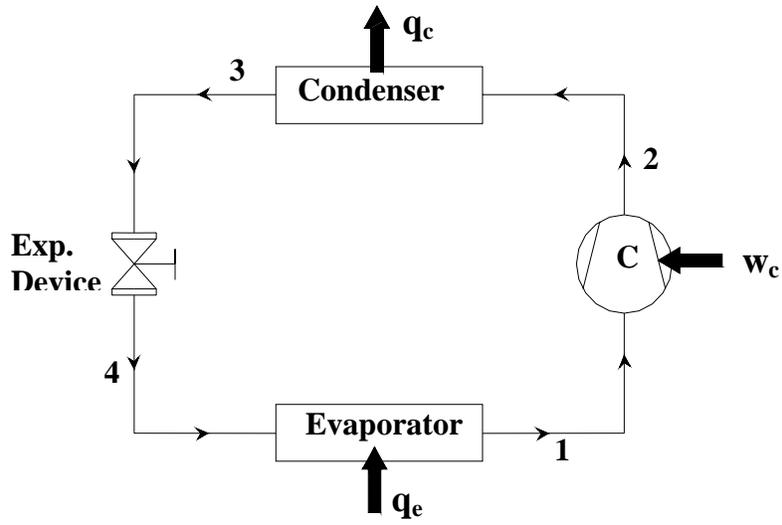


Fig.10.5. Standard Vapour compression refrigeration system

10.4. Standard Vapour Compression Refrigeration System (VCRS)

Figure 10.5 shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a T s diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor

Process 2-3: Isobaric heat rejection in condenser

Process 3-4: Isenthalpic expansion of saturated liquid in expansion device

Process 4-1: Isobaric heat extraction in the evaporator

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non-isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3-4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the cycle diagrams on T s charts. Figure 10.6(a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

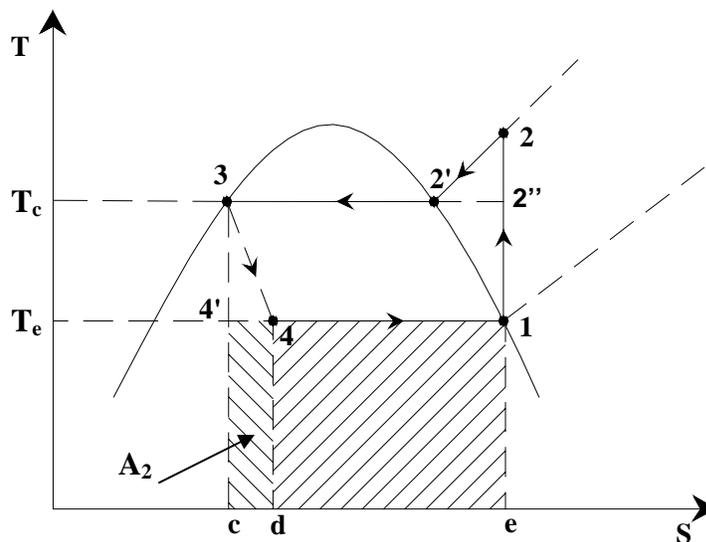


Fig.10.6(a). Comparison between Carnot and standard VCRS

The heat extraction (evaporation) process is reversible for both the Carnot cycle and VCRS cycle. Hence the refrigeration effect is given by:

For Carnot refrigeration cycle (1-2''-3-4'):

$$q_{e,\text{Carnot}} = q_{4'-1} = \int_{4'}^1 T \cdot ds = T_e (s_1 - s_{4'}) = \text{area } e-1-4'-c-e \quad (10.7)$$

For VCRES cycle (1-2-3-4):

$$q_{e,VCRES} = q_{4-1} = \int_4^1 T.ds = T_e (s_1 - s_4) = \text{area } e-1-4-d-e \quad (10.8)$$

thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRES cycle, this reduction is equal to the area $d-4-4'-c-d$ (area A_2) and is known as *throttling loss*. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e.,

$$q_{e,Carnot} - q_{VCRES} = \text{area } d-4-4'-c-d = (h_3 - h_{4'}) = (h_4 - h_{4'}) = \text{area } A_2 \quad (10.9)$$

It is easy to show that the loss in refrigeration effect increases as the evaporator temperature decreases and/or condenser temperature increases. A practical consequence of this is a requirement of higher refrigerant mass flow rate.

The heat rejection in case of VCRES cycle also increases when compared to Carnot cycle.

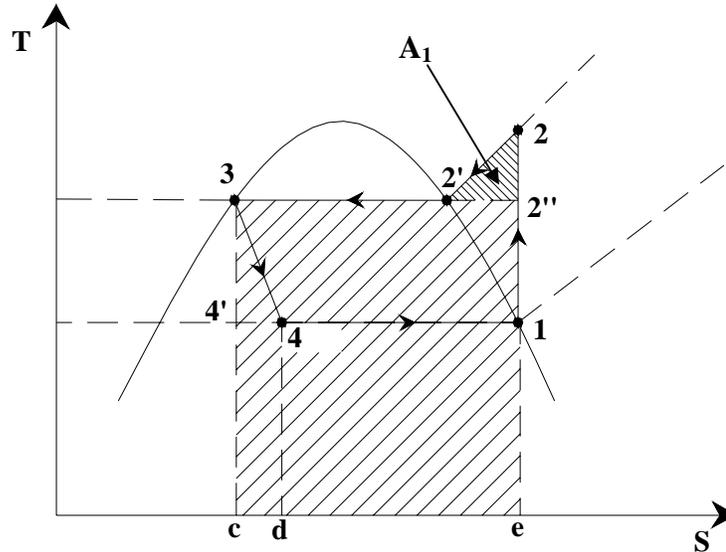


Fig.10.6(b). Comparative evaluation of heat rejection rate of VCRES and Carnot cycle

As shown in Fig.10.6(b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

$$q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^3 T.ds = T_c (s_{2''} - s_3) = \text{area } e-2''-3-c-e \quad (10.10)$$

In case of VCRES cycle, the heat rejection rate is given by:

$$q_{c,VCRES} = -q_{2-3} = -\int_2^3 T.ds = \text{area } e-2-3-c-e \quad (10.11)$$

Hence the increase in heat rejection rate of VCRES compared to Carnot cycle is equal to the area $2''-2-2'$ (area A_1). This region is known as *superheat horn*, and is due to the

replacement of isothermal heat rejection process of Carnot cycle by isobaric heat rejection in case of VCRS.

Since the heat rejection increases and refrigeration effect reduces when the Carnot cycle is modified to standard VCRS cycle, the net work input to the VCRS increases compared to Carnot cycle. The net work input in case of Carnot and VCRS cycles are given by:

$$W_{\text{net,Carnot}} = (q_c - q_e)_{\text{Carnot}} = \text{area } 1-2'-3-4'-1 \quad (10.12)$$

$$W_{\text{net,VCRS}} = (q_c - q_e)_{\text{VCRS}} = \text{area } 1-2-3-4'-c-d-4-1 \quad (10.13)$$

As shown in Fig.10.6(c), the increase in net work input in VCRS cycle is given by:

$$W_{\text{net,VCRS}} - W_{\text{net,Carnot}} = \text{area } 2'-2-2'' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2 \quad (10.14)$$

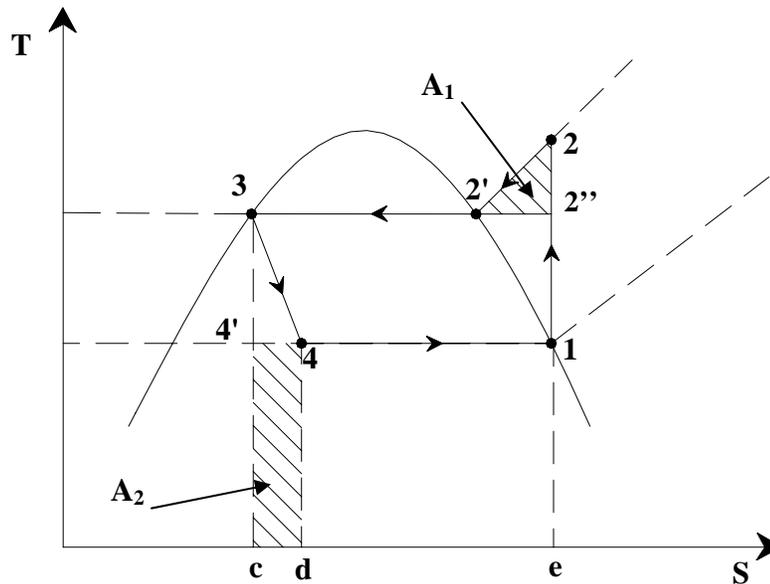


Fig.10.6(c). Figure illustrating the increase in net work input in VCRS cycle

To summarize the refrigeration effect and net work input of VCRS cycle are given by:

$$q_{e,\text{VCRS}} = q_{e,\text{Carnot}} - \text{area } A_2 \quad (10.15)$$

$$W_{\text{net,VCRS}} = W_{\text{net,Carnot}} + \text{area } A_1 + \text{area } A_2 \quad (10.16)$$

The COP of VCRS cycle is given by:

$$\text{COP}_{\text{VCRS}} = \frac{q_{e,\text{VCRS}}}{W_{\text{net,VCRS}}} = \frac{q_{e,\text{Carnot}} - \text{area } A_2}{W_{\text{net,Carnot}} + \text{area } A_1 + \text{area } A_2} \quad (10.17)$$

If we define the cycle efficiency, η_R as the ratio of COP of VCRS cycle to the COP of Carnot cycle, then:

$$\eta_R = \frac{COP_{VCRS}}{COP_{Carnot}} = \left[\frac{1 - \left(\frac{\text{area } A_2}{q_{e,Carnot}} \right)}{1 + \left(\frac{\text{area } A_1 + \text{area } A_2}{W_{net,Carnot}} \right)} \right] \quad (10.18)$$

The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle. Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid.

If we assume that the potential and kinetic energy changes during isentropic compression process 1-2 are negligible, then the work input w_{1-2} is given by:

$$w_{1-2,VCRS} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f) \quad (10.19)$$

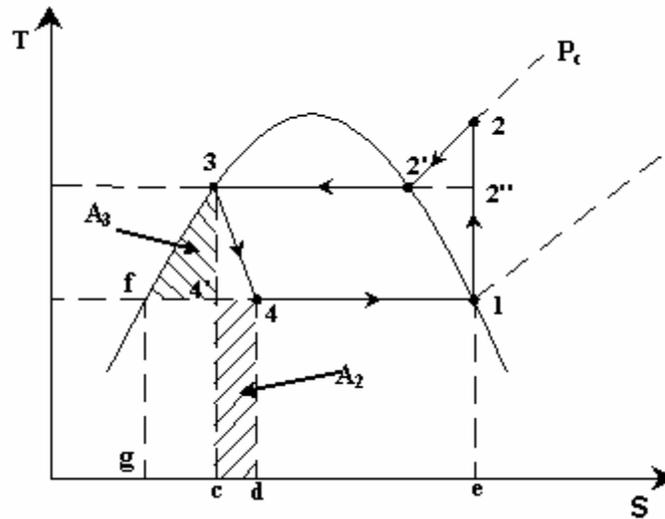


Fig.10.7. Figure showing saturated liquid line 3-f coinciding with the constant pressure line

Now as shown in Fig.10.7, if we further assume that the saturated liquid line 3-f coincides with the constant pressure line P_c in the subcooled region (which is a reasonably good assumption), then from the 2nd Tds relation;

$$\begin{aligned} Tds &= dh - v dP = dh; \text{ when } P \text{ is constant} \\ \therefore (h_2 - h_f) &= \int_2^f Tds = \text{area } e-2-3-f-g-e \end{aligned} \quad (10.20)$$

$$\text{and, } (h_1 - h_f) = \int_1^f T ds = \text{area } e-1-f-g-e \quad (10.21)$$

Substituting these expressions in the expression for net work input, we obtain the compressor work input to be equal to area 1-2-3-f-1. Now comparing this with the earlier expression for work input (area 1-2-3-4'-c-d-4-1), we conclude that area A_2 is equal to area A_3 .

As mentioned before, the losses due to superheat (area A_1) and throttling (area $A_2 \approx A_3$) depend very much on the shape of the vapor dome (saturation liquid and vapour curves) on T s diagram. The shape of the saturation curves depends on the nature of refrigerant. Figure 10.8 shows T s diagrams for three different types of refrigerants.

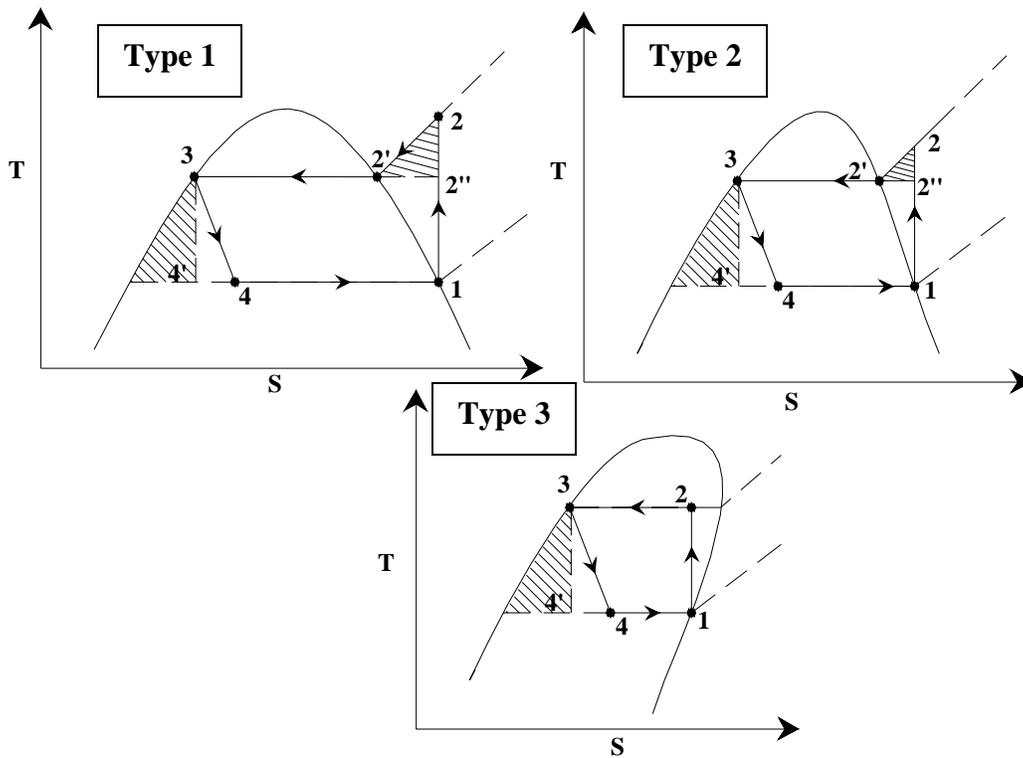


Fig.10.8. T-s diagrams for three different types of refrigerants

Refrigerants such as ammonia, carbon di-oxide and water belong to Type 1. These refrigerants have symmetrical saturation curves (vapour dome), as a result both the superheat and throttling losses (areas A_1 and A_3) are significant. That means deviation of VCRC cycle from Carnot cycle could be significant when these refrigerants are used as working fluids. Refrigerants such as CFC11, CFC12, HFC134a belong to Type 2, these refrigerants have small superheat losses (area A_1) but large throttling losses (area A_3). High molecular weight refrigerants such as CFC113, CFC114, CFC115, iso-butane belonging to Type 3, do not have any superheat losses, i.e., when the compression inlet condition is saturated (point 1), then the exit condition will be in the 2-phase region, as a result it is not necessary to superheat the refrigerant. However, these refrigerants

experience significant throttling losses. Since the compressor exit condition of Type 3 refrigerants may fall in the two-phase region, there is a danger of wet compression leading to compressor damage. Hence for these refrigerants, the compressor inlet condition is chosen such that the exit condition does not fall in the two-phase region. This implies that the refrigerant at the inlet to the compressor should be superheated, the extent of which depends on the refrigerant.

Superheat and throttling losses:

It can be observed from the discussions that the superheat loss is fundamentally different from the throttling loss. The superheat loss increases only the work input to the compressor, it does not effect the refrigeration effect. In heat pumps superheat is not a loss, but a part of the useful heating effect. However, the process of throttling is inherently irreversible, and it increases the work input and also reduces the refrigeration effect.

10.5. Analysis of standard vapour compression refrigeration system

A simple analysis of standard vapour compression refrigeration system can be carried out by assuming a) Steady flow; b) negligible kinetic and potential energy changes across each component, and c) no heat transfer in connecting pipe lines. The steady flow energy equation is applied to each of the four components.

Evaporator: Heat transfer rate at evaporator or *refrigeration capacity*, \dot{Q}_e is given by:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) \quad (10.22)$$

where \dot{m}_r is the refrigerant mass flow rate in kg/s, h_1 and h_4 are the specific enthalpies (kJ/kg) at the exit and inlet to the evaporator, respectively. $(h_1 - h_4)$ is known as specific refrigeration effect or simply *refrigeration effect*, which is equal to the heat transferred at the evaporator per kilogram of refrigerant. The evaporator pressure P_e is the saturation pressure corresponding to evaporator temperature T_e , i.e.,

$$P_e = P_{\text{sat}}(T_e) \quad (10.23)$$

Compressor: Power input to the compressor, \dot{W}_c is given by:

$$\dot{W}_c = \dot{m}_r (h_2 - h_1) \quad (10.24)$$

where h_2 and h_1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. $(h_2 - h_1)$ is known as specific work of compression or simply *work of compression*, which is equal to the work input to the compressor per kilogram of refrigerant.

Condenser: Heat transfer rate at condenser, \dot{Q}_c is given by:

$$\dot{Q}_c = \dot{m}_r (h_2 - h_3) \quad (10.25)$$

where h_3 and h_2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

The condenser pressure P_c is the saturation pressure corresponding to evaporator temperature T_c , i.e.,

$$P_c = P_{\text{sat}}(T_c) \quad (10.26)$$

Expansion device: For the isenthalpic expansion process, the kinetic energy change across the expansion device could be considerable, however, if we take the control volume, well downstream of the expansion device, then the kinetic energy gets dissipated due to viscous effects, and

$$h_3 = h_4 \quad (10.27)$$

The exit condition of the expansion device lies in the two-phase region, hence applying the definition of quality (or dryness fraction), we can write:

$$h_4 = (1 - x_4)h_{f,e} + x_4h_{g,e} = h_f + x_4h_{fg} \quad (10.28)$$

where x_4 is the quality of refrigerant at point 4, $h_{f,e}$, $h_{g,e}$, h_{fg} are the saturated liquid enthalpy, saturated vapour enthalpy and latent heat of vaporization at evaporator pressure, respectively.

The COP of the system is given by:

$$\text{COP} = \left(\frac{\dot{Q}_e}{\dot{W}_c} \right) = \left(\frac{\dot{m}_r (h_1 - h_4)}{\dot{m}_r (h_2 - h_1)} \right) = \frac{(h_1 - h_4)}{(h_2 - h_1)} \quad (10.29)$$

At any point in the cycle, the mass flow rate of refrigerant \dot{m}_r can be written in terms of volumetric flow rate and specific volume at that point, i.e.,

$$\dot{m}_r = \frac{\dot{V}}{v} \quad (10.30)$$

applying this equation to the inlet condition of the compressor,

$$\dot{m}_r = \frac{\dot{V}_1}{v_1} \quad (10.31)$$

where \dot{V}_1 is the volumetric flow rate at compressor inlet and v_1 is the specific volume at compressor inlet. At a given compressor speed, \dot{V}_1 is an indication of the size of the compressor. We can also write, the refrigeration capacity in terms of volumetric flow rate as:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) = \dot{V}_1 \left(\frac{h_1 - h_4}{v_1} \right) \quad (10.32)$$

where $\left(\frac{h_1 - h_4}{v_1} \right)$ is called as *volumetric refrigeration effect* (kJ/m³ of refrigerant).

Generally, the type of refrigerant, required refrigeration capacity, evaporator temperature and condenser temperature are known. Then from the evaporator and condenser temperature one can find the evaporator and condenser pressures and enthalpies at the exit of evaporator and condenser (saturated vapour enthalpy at evaporator pressure and saturated liquid enthalpy at condenser pressure). Since the exit condition of the compressor is in the superheated region, two independent properties are required to fix the state of refrigerant at this point. One of these independent properties could be the condenser pressure, which is already known. Since the compression process is isentropic, the entropy at the exit to the compressor is same as the entropy at the inlet, s_1 which is the saturated vapour entropy at evaporator pressure (known). Thus from the known pressure and entropy the exit state of the compressor could be fixed, i.e.,

$$\begin{aligned} h_2 &= h(P_c, s_2) = h(P_c, s_1) \\ s_1 &= s_2 \end{aligned} \quad (10.33)$$

The quality of refrigerant at the inlet to the evaporator (x_4) could be obtained from the known values of h_3 , $h_{f,e}$ and $h_{g,e}$.

Once all the state points are known, then from the required refrigeration capacity and various enthalpies one can obtain the required refrigerant mass flow rate, volumetric flow rate at compressor inlet, COP, cycle efficiency etc.

Use of Pressure-enthalpy (P-h) charts:

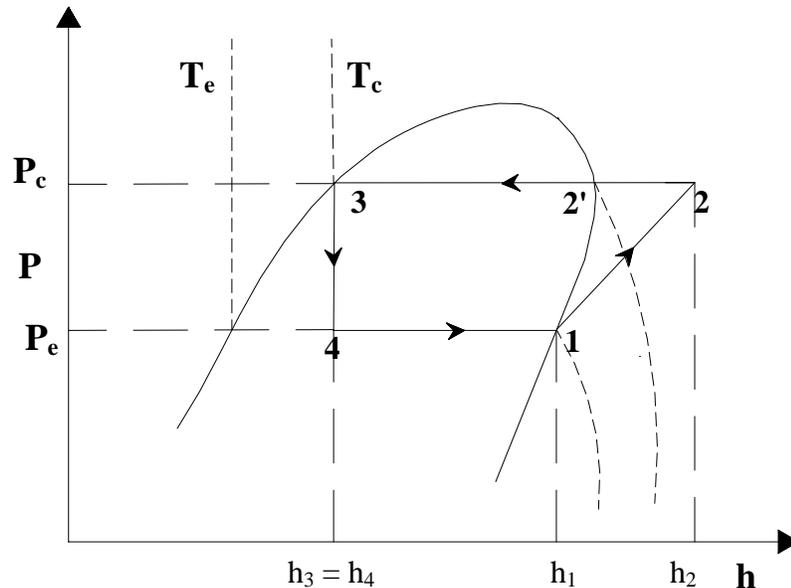


Fig.10.9. Standard vapour compression refrigeration cycle on a P-h chart

Since the various performance parameters are expressed in terms of enthalpies, it is very convenient to use a pressure – enthalpy chart for property evaluation and performance analysis. The use of these charts was first suggested by Richard Mollier. Figure 10.9 shows the standard vapour compression refrigeration cycle on a P-h chart. As discussed before, in a typical P-h chart, enthalpy is on the x-axis and pressure is on y-axis. The isotherms are almost vertical in the subcooled region, horizontal in the two-phase region (for pure refrigerants) and slightly curved in the superheated region at high pressures, and again become almost vertical at low pressures. A typical P-h chart also shows constant specific volume lines (isochors) and constant entropy lines (isentropes) in the superheated region. Using P-h charts one can easily find various performance parameters from known values of evaporator and condenser pressures.

In addition to the P-h and T-s charts one can also use thermodynamic property tables from solving problems related to various refrigeration cycles.

Questions:

1. A Carnot refrigerator using R12 as working fluid operates between 40°C and -30°C. Determine the work of compression and cooling effect produced by the cycle. ([Solution](#))
2. An ideal refrigeration cycle operates with R134a as the working fluid. The temperature of refrigerant in the condenser and evaporator are 40°C and -20°C respectively. The mass flow rate of refrigerant is 0.1 kg/s. Determine the cooling capacity and COP of the plant. ([Solution](#))
3. A R-12 plant has to produce 10 tons of refrigeration. The condenser and evaporator temperatures are 40°C and -10°C respectively. Determine
 - a) Refrigerant flow rate
 - b) Volume flow rate of the compressor
 - c) Operating pressure ratio
 - d) Power required to drive the compressor
 - e) Flash gas percentage after throttling
 - f) COP ([Solution](#))
4. A NH₃ refrigerator produces 100 tons of ice from water at 0°C in a day. The cycle operates between 25°C and -15°C . The vapor is dry saturated at the end of compression. If the COP is 50% of theoretical COP, calculate the power required to drive the compressor. ([Solution](#))
5. In a refrigerator the power rating impressed on the compressor is 1.2 kW. The circulating wire in evaporator is 5 kW and the cooling water took away 10 kW from condenser coil. The operating temperatures range is 18°C and 0°C and their corresponding latent heats are 170 kJ/kg and 230 kJ/kg and the difference between the

liquid energy is 35 kJ/kg. Find the actual COP of the system (2) relative COP, assuming the vapour is just dry and saturated at the end of the compression. ([Solution](#))

6. A water cooler using R12 refrigerant works between 30°C to 9°C. Assuming the volumetric and mechanical efficiency of the compressor to be 80 and 90% respectively, and the mechanical efficiency of motor to be 90% , and 20% of useful cooling is lost into water cooler, find:

- 1) The power requirement of the motor
- 2) Volumetric displacement of the compressor

Given C_p (saturated vapour at 30°C) = 0.7 kJ/kg K ([Solution](#))

The properties of F12 at 30°C and 2°C are:

Temp °C	Pressure (Bar)	Liquid		Vapour		
		h_f (kJ/kg)	S_f (kJ/kg K)	h_g (kJ/kg)	S_g (kJ/kg K)	v_s m ³ /kg
30	7.45	64.6	0.2399	199.6	0.6854	0.0235
5	3.626	40.7	0.1587	189.7	0.6942	0.0475