Vapour Compression Refrigeration Systems

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The specific objectives of the lesson:

This lesson examines the most usually utilized refrigeration system, i.e. Vapour compression refrigeration system. The accompanying things are underscored in point of interest:

1. The Carnot refrigeration cycle & its practical limitations.
2. The Standard Vapour compression Refrigeration System.
3. Analysis of Standard Vapour compression Refrigeration System.

1.1. Comparison between gas cycles and vapour cycles

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. 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 geotropic mixture is utilized as a refrigerant, then there will be a temperature float amid vaporization and build up. Since the refrigeration impact is delivered amid stage change, huge measure of heat (latent heat) can be exchanged every kilogram of refrigerant at a close steady temperature. Thus, the obliged mass stream rates for a given refrigeration limit will be much littler contrasted with a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour retention systems, vapour plane systems and so forth. Among these the vapour compression refrigeration systems are overwhelming.

1.2. Vapour Compression Refrigeration Systems

As specified, vapour compression refrigeration systems are the most ordinarily utilized among all refrigeration systems. As the name suggests, these systems have a place with the general class of vapour cycles, wherein the working liquid (refrigerant) experiences stage change at any rate amid one procedure. In a vapour compression refrigeration system, refrigeration is gotten as the refrigerant dissipates at low temperatures. The info to the system is as mechanical vitality needed to run the compressor. Thus these systems are also called as mechanical refrigeration systems.

Vapour compression refrigeration systems are accessible to suit all applications with the refrigeration limits extending from couple of Watts to couple of megawatts. A wide mixed bag of refrigerants can be utilized as a part of these systems to suit diverse applications, limits and so forth. The real vapour compression cycle is in view of Evans-Perkins cycle, which is additionally called as opposite Rankine cycle. Before the real cycle is examined and dissected, it is key to discover the furthest reaches of execution of vapour compression cycles. This breaking point is situated by a totally reversible cycle.

1.3. The Carnot refrigeration cycle

Carnot refrigeration cycle is a totally reversible cycle, thus is utilized as a model of flawlessness for a refrigeration cycle working between a steady temperature heat source and sink. It is utilized as reference against which the genuine cycles are analyzed. Figures 1.1 (an) and (b) demonstrate the schematic of a Carnot vapour compression refrigeration system and the working cycle on T-s chart.

As shown in Fig.1.1 (a), the basic Carnot refrigeration system for unadulterated vapour comprises of four parts: compressor, condenser, turbine and evaporator. Refrigeration impact (q4- 1 = qe) is gotten at the evaporator as the refrigerant experiences the methodology of vaporization (process 4-1) and concentrates the idle 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 Tc. The refrigerant pressure increments from Pe to Pc amid compression (transform 1-2) and the way out vapour is soaked. Next the high pressure, high temperature soaked refrigerant experiences the methodology of build up in the condenser (process 2-3) as it rejects the heat of build up (q2-3 = qc) to an outside heat sink at Tc. The high pressure immersed fluid then flows through the turbine and experiences isentropic development (transform 3-4).

During this process, the pressure and temperature tumble from Pc,Tc to Pe, Te. Since an immersed fluid is extended in the turbine, some measure of fluid flashes into vapour and the way out condition lies in the two-stage area. This low temperature and low pressure fluid vapour mixture then enters the evaporator finishing the cycle. In this way as indicated in Fig.10.1(b), the cycle includes two isothermal heat exchange (forms 4-1 and 2-3) and two isentropic work exchange (forms 1-2 and 3-4). Heat is separated isothermally at evaporator temperature Te amid procedure 4-1, heat is dismisses isothermally at condenser temperature Tc amid methodology 2-3.

Work is supplied to the compressor amid the isentropic compression (1-2) of refrigerant vapour from evaporator pressure Pe to condenser pressure Pc, and work is created by the system as refrigerant liquid extends isentropically in the turbine from condenser pressure Pc to evaporator pressure Pe. All the courses of action are both inside and additionally remotely reversible, i.e., net entropy era for the system and environment is zero.

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

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = -w_{net}$$

$$\Rightarrow (q_c - q_e) = w_{net}$$

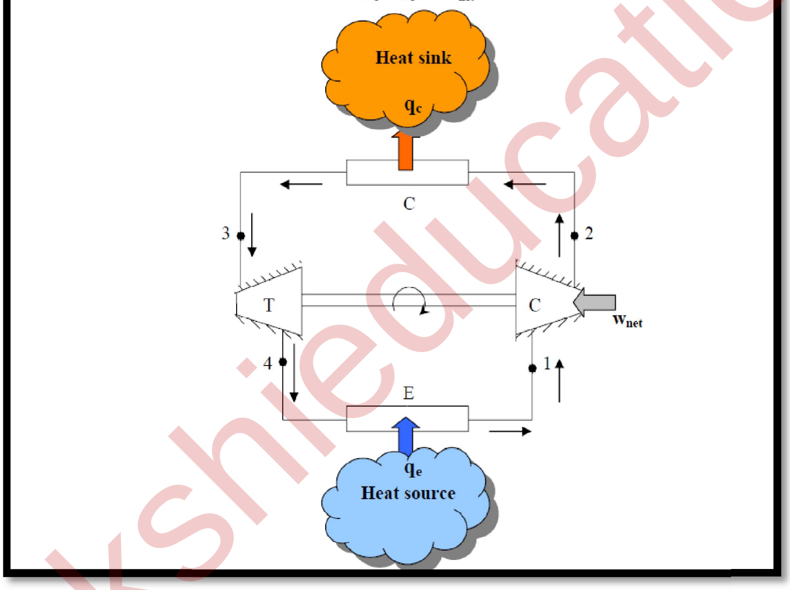


Fig.1.1(a): Schematic of a Carnot refrigeration system

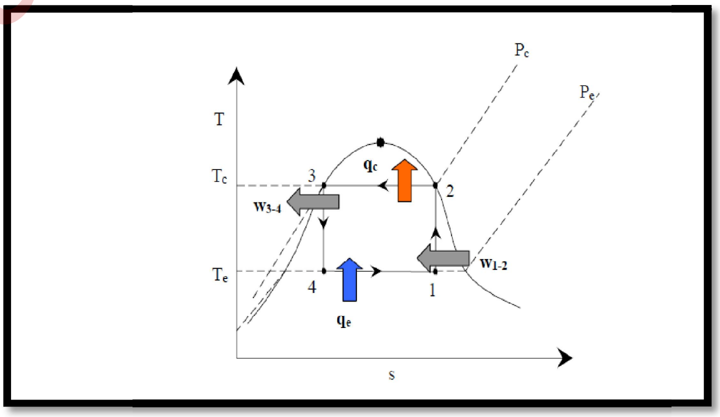


Fig. 1.1(b): Carnot refrigeration cycle on T-s diagram

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

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c(s_2 - s_3)$$
$$q_e = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4)$$

Where Te and Tc are the evaporator and condenser temperatures, respectively, and

$$s_1 = s_2 \text{ and } s_3 = s_4$$

the Coefficient of Performance (COP) is given by:

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

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 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.

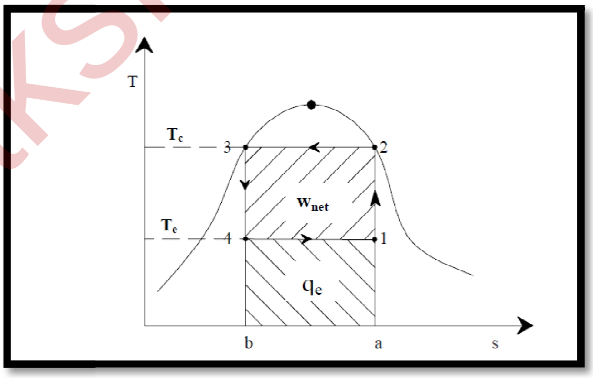


Fig.1.2. Carnot refrigeration cycle represented in T-s diagram

It can be seen from the above interpretation that the COP of a Carnot refrigeration system increments as the evaporator temperature increments and condenser temperature diminishes. This can be clarified effectively with the assistance of the T-s outline (Fig.10.2). As indicated in the figure, COP is the proportion of range a-1-4-b to the territory 1-2-3-4. For a settled condenser temperature Tc, as the evaporator temperature Te expands, range a-1-4-b (qe) increments and region 1-2-3-4 (w_{net}) diminishes accordingly, COP increments rapidly.

Thus for a settled evaporator temperature Te, as the consolidating temperature Tc builds, the net work info (territory 1-2-3-4) expands, despite the fact that cooling yield stays steady, therefore the COP falls. Figure 1.3 demonstrates the variety of Carnot COP with evaporator temperature for diverse condenser temperatures. It can be seen that the COP increments pointedly with evaporator temperatures, especially at high gathering temperatures. COP decreases as the condenser temperature increments, yet the impact gets to be peripheral at low evaporator temperatures. It will be demonstrated later that real vapour compression refrigeration systems additionally act in a way like that of Carnot refrigeration systems the extent that the execution patterns are concerned.

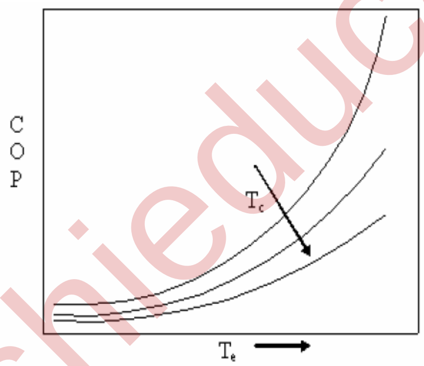


Fig.1.3. Effects of evaporator and condenser temperatures on Carnot COP

Practical difficulties with Carnot refrigeration system:

It is hard to fabricate and work a Carnot refrigeration system because of the following functional challenges:

i. Amid procedure 1-2, a mixture comprising of liquid and vapour must be compressed isentropically in the compressor. Such a compression is known as wet compression because of the pressure of liquid. Practically speaking, wet compression is exceptionally troublesome particularly with responding compressors. This issue is especially serious if there should arise an occurrence of fast responding compressors, which get harmed because of the vicinity of liquid droplets in the vapour. Despite the fact that a few sorts of compressors can endure the vicinity of liquid in vapour, since responding compressors are most broadly is refrigeration, traditionally dry (compression of vapour only) is wanted 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:

$$w_{3-4} = \int_{Pe}^{Pc} v.dP$$

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.

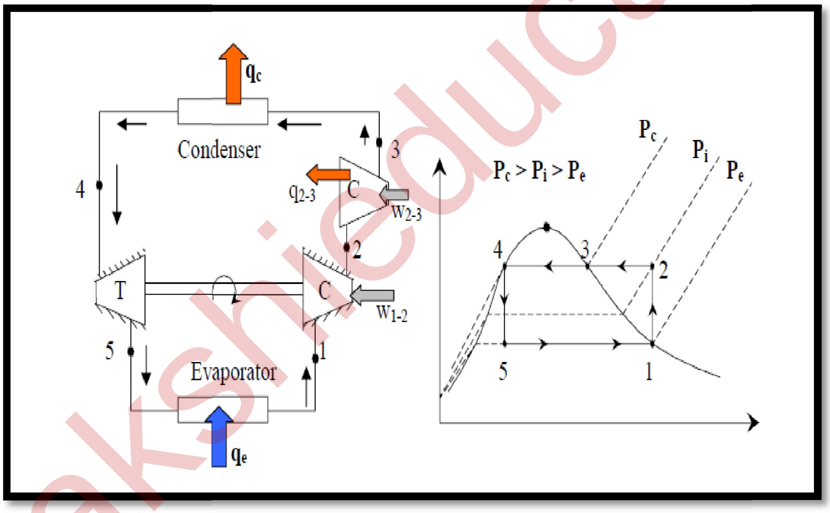
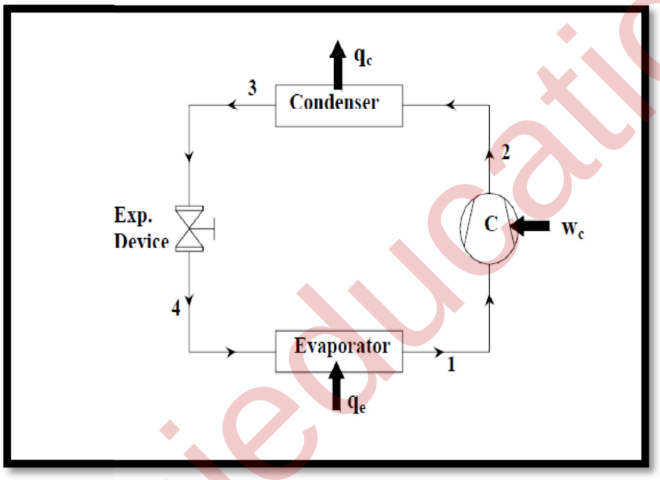


Fig.1.4. Carnot refrigeration system with dry compression

As shown in Fig.1.4, the Carnot refrigeration system with dry compression comprises of one isentropic compression handle (1-2) from evaporator pressure Pe to a middle pressure Pi and temperature Tc, followed by an isothermal compression prepare (2-3) from the transitional pressure Pi to the condenser pressure Pc. Despite the fact that with this alteration the issue of wet compression can be maintained a strategic distance from, still this altered system is not functional because of the trouble in accomplishing genuine isothermal compression utilizing high velocity compressors.

Furthermore, utilization of two compressors set up of one is not monetarily defended. From the above talk, it is pass that from functional contemplations, the Carnot refrigeration system need to be changed. Dry compression with a solitary compressor is conceivable if the isothermal heat dismissal procedure is supplanted by isobaric heat dismissal methodology. Correspondingly, the isentropic development methodology can be supplanted by an isenthalpic throttling procedure. A refrigeration system, which consolidates these two progressions, is known as Evans-Perkins or converse Rankine cycle. This is the hypothetical cycle on which the genuine vapour compression refrigeration systems are based.



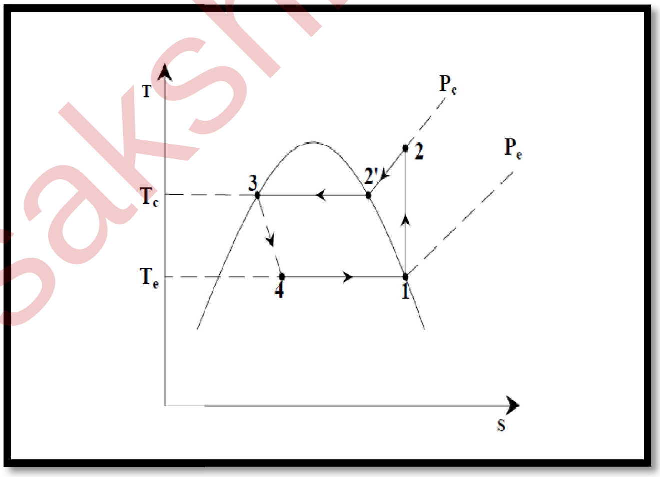


Fig.1.5. Standard Vapour compression refrigeration system

1.4. Standard Vapour Compression Refrigeration System (VCRS)

Figure 1.5 shows the schematic of a standard, soaked, single stage (SSS) vapor compression refrigeration system and the operating cycle on a T s graph. As demonstrated in the figure the standard single stage, soaked vapour compression refrigeration system comprises of the following four techniques:

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 1.6(a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

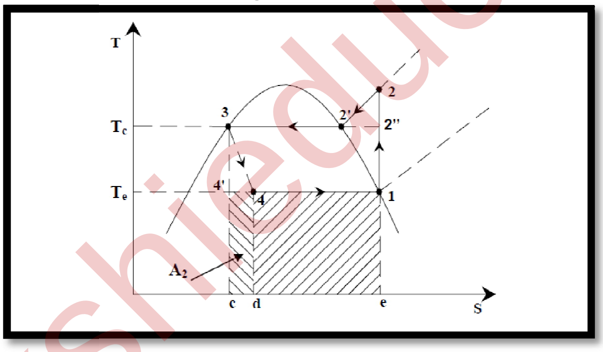


Fig.1.6(a). Comparison between Carnot and standard VCRS

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

<u>For Carnot refrigeration cycle (1-2''-3-4'):</u>

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

<u>For VCRS cycle (1-2-3-4):</u>

$$q_{e,VCRS} = q_{4-1} = \int_{4}^{1} T.ds = T_e(s_1 - s_4) = \text{area } e-1-4-d-e$$

thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRS cycle, this reduction is equal to the area d-4-4'-c-d (area A2) and is known as *throttling loss*. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e,

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

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 VCRS cycle also increases when compared to Carnot cycle.

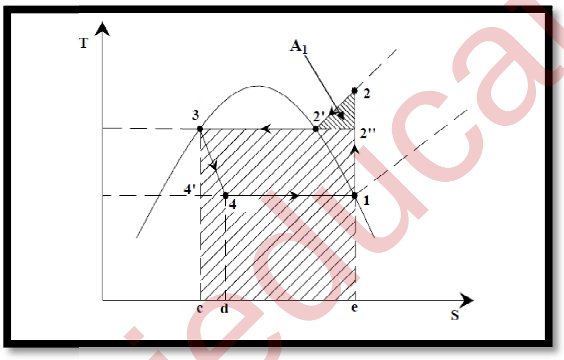


Fig.1.6(b). Comparative evaluation of heat rejection rate of VCRS and Carnot cycle

As shown in Fig.1.6(b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

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

$$q_{c,VCRS} = -q_{2-3} = -\int_{2}^{3} T.ds = \text{area } e-2-3-c-e$$

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

Hence the increase in heat rejection rate of VCRS compared to Carnot cycle is equal to the area 2''-2-2' (area A1). 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:

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

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

As shown in Fig.1.6(c), the increase in net work input in VCRS cycle is given by:

$$w_{net,VCRS} - w_{net,Carnot} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2$$

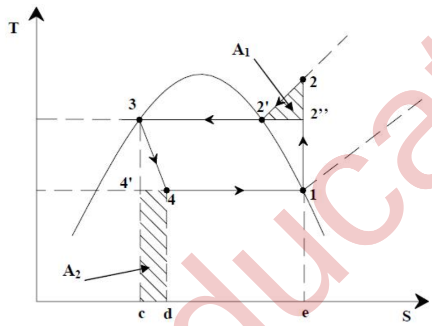


Fig.1.6(c). Figure illustrating the increase in net work input in VCRS cycle

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

$$q_{e,VCRS} = q_{e,Carnot} - \text{area } A_2$$

$$w_{net,VCRS} = w_{net,Carnot} + \text{area } A_1 + \text{area } A_2$$

The COP of VCRS cycle is given by:

$$COP_{VCRS} = \frac{q_{e,VCRS}}{w_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{w_{net,Carnot} + \text{area } A_1 + \text{area } A_2}$$

The COP of VCRS cycle is given by:

$$COP_{VCRS} = \frac{q_{e,VCRS}}{w_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{w_{net,Carnot} + \text{area } A_1 + \text{area } A_2}$$

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

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

The cycle efficiency (additionally called as second law proficiency) is a decent sign of the deviation of the standard VCRS cycle from Carnot cycle. Not at all like Carnot COP, the cycle proficiency depends all that much on the state of T s graph, which thus relies on upon the way of the working liquid.

In the event that we accept that the potential and active vitality changes amid isentropic compression prepare 1-2 are immaterial, then the work info w1-2 is given by:

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

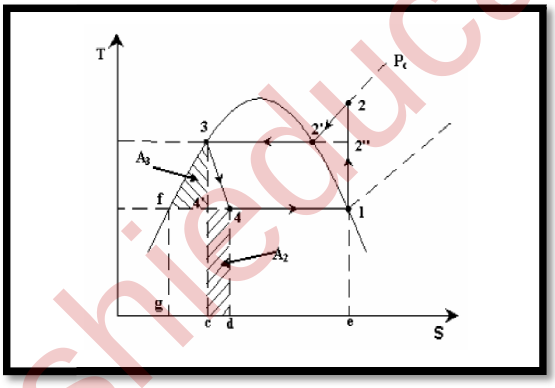


Fig.1.7. Figure showing saturated liquid line 3-f coinciding with the constant pressure line

Now as shown in Fig.1.7, on the off chance that we further accept that the immersed liquid line 3-f corresponds with the steady pressure line Pc in the sub cooled locale (which is a sensibly decent presumption), then from the second Tds connection; Tds =dh - v dP = dh; when P is constant.

$$\therefore (h_2 - h_f) = \int_2^f T ds = \text{area } e-2-3-f-g-e$$

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

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 A2 is equal to area A3.

As mentioned before, the losses due to superheat (area A1) and throttling (area A2 ≈ A3) depend very much on the shape of the vapour 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.

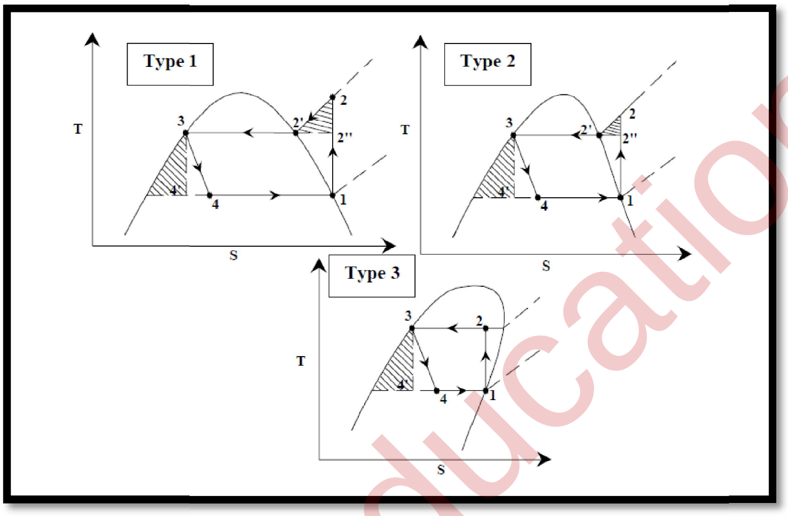


Fig.1.8. T-s diagrams for three different types of refrigerants

Refrigerant, for example, alkali, carbon di-oxide and water fit in with Type 1. These refrigerants have symmetrical saturation curves (vapor vault), subsequently both the superheat and throttling losses (territories A1 and A3) are critical. That implies deviation of VCRS cycle from Carnot cycle could be huge when these refrigerants are utilized as meeting expectations liquids. Refrigerants, for example, CFC11, CFC12, HFC134a fit in with Type 2, these refrigerants have little superheat losses (zone A1) yet vast throttling losses (region A3).

High sub-atomic weight refrigerants, for example, CFC113, CFC114, CFC115, iso-butane fitting in with Type 3, don't have any superheat losses, i.e., when the compression channel condition is immersed (point 1), then the way out condition will be in the 2-stage area, subsequently it is not important 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 affect 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.

1.5 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*, .Q $_e$ is given by:

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

Where mr is the refrigerant mass flow rate in kg/s, h1 and h4 are the specific enthalpies (kJ/kg) At the exit and inlet to the evaporator, respectively. (h1 − h4) 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 Pe is the saturation pressure corresponding to evaporator temperature Te, i.e.,

$$P_e = P_{sat}(T_e)$$

.

Compressor: Power input to the compressor, W c is given by:

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

Where h2 and h1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. (h 2 − h1) 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, Q_c is given by:

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

where h3 and h2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

The condenser pressure Pc is the saturation pressure corresponding to evaporator temperature T , i.e.,

$$P_c = P_{sat}(T_c)$$

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

$$h_3 = h_4$$

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:

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

Where x4 is the quality of refrigerant at point 4, hf,e, hg,e, hfg 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:

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

At any point in the cycle, the mass flow rate of refrigerant m_r can be written in terms of volumetric flow rate and specific volume at that point, i.e.,

$$\dot{m}_r = \dot{V}/v$$

applying this equation to the inlet condition of the compressor,

$$\dot{m}_r = \dot{V}_1 / v_1$$

Where V $_1$ is the volumetric flow rate at compressor inlet and v1 is the specific volume at compressor inlet. At a given compressor speed, V1 is an indication of the size of the compressor. We can also write, the refrigeration capacity in terms of volumetric flow rate as:

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

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

Generally, the sort of refrigerant, obliged refrigeration capacity, evaporator temperature and condenser temperature are known. At that point from the evaporator and condenser temperature one can discover the evaporator and condenser pressures and enthalpies at the way out of evaporator and condenser (saturated vapour enthalpy at evaporator pressure and saturated liquid enthalpy at condenser pressure). Since the way out state of the compressor is in the superheated area, two autonomous properties are obliged to alter the condition of refrigerant right now.

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, s1 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., The quality of refrigerant at the inlet to the evaporator (x4) could be obtained from the known values of h3, hf,e and hg,e.

$$h_2 = h(P_c, s_2) = h(P_c, s_1)$$
$$s_1 = s_2$$

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:

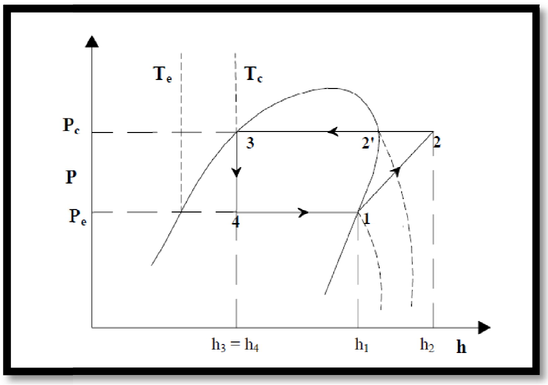


Fig.1.9. Standard vapours compression refrigeration cycle on a P-h chart

Since the different execution parameters are communicated as far as enthalpies, it helpful to utilize a pressure – enthalpy graph for property assessment and execution investigation. The utilization of these graphs was initially proposed by Richard Mollier. Figure 10.9 demonstrates the standard vapour compression refrigeration cycle on a P-h outline. As talked about some time recently, in a normal P-h outline, enthalpy is on the x-pivot and pressure is on y-hub. The isotherms are just about vertical in the sub cooled district, flat in the two-stage locale (for immaculate refrigerants) and somewhat bended in the superheated area at high pressures, and again get to be practically vertical at low pressures.

A run of the mill P-h graph likewise indicates steady particular volume lines (isochors) and consistent entropy lines (isentropes) in the superheated area. Utilizing P-h outlines one can undoubtedly discover different execution parameters from known estimations of evaporator and condenser pressures. Notwithstanding the P-h and T-s diagrams one can likewise utilize thermodynamic property tables from taking care of issues identified with different refrigeration cycle