

Multi-Evaporator and Cascade Systems

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The objectives of this lesson are to:

1. The advantages and applications of multi-evaporator systems compared to single stage systems.
2. Multi-evaporator systems using single compressor and a pressure reducing valve with:
 - a) Individual expansion valves.
 - b) Multiple expansion valves.
3. Multi-evaporator systems with multi-compression, inter cooling and flash gas removal.
4. Multi-evaporator systems with individual compressors and multiple expansion valves.
5. Limitations of multi-stage systems.
6. Cascade systems.
7. The working principle of auto-cascade cycle.

1.1. Introduction

Several applications find refrigeration essential at various temperatures. For example, in a typical food finalizing plant, cold air may be required in -30°C for freezing along with at $+7^{\circ}\text{C}$ for cooling of foods or place cooling. One of the straight forward alternatives is using different refrigeration systems to serve these various loads. However, this is probably not economically viable a result of the high entire initial expense and another alternative would be using a solitary refrigeration method with one particular compressor along with two evaporators both operating in -30°C . The schematic of this type of system along with corresponding functioning cycle in P-h diagram are generally shown throughout Figs. 1. 1(a) along with (b).

As shown inside figure the system is made up of single compressor along with a single condenser but two evaporators. Both evaporators-I along with II operate at very same evaporator temperature (-30°C) one particular evaporator (say Evaporator-I) suits freezing while the other (Evaporator-II) suits product cooling/space treatment at 7°C . It can be seen in which operating the actual evaporator in -30°C as soon as refrigeration is essential at $+7^{\circ}\text{C}$ is thermodynamically inefficient for the reason that system irreversibility's raise with increasing temperature distinction for heat transfer.

The COP of this simple system is given by:

$$\text{COP} = \frac{Q_{e,I} + Q_{e,II}}{W_c} = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

In addition to this there will also be other difficulties such as: evaporator catering to space cooling (7°C) may collect frost leading to blockage of air-flow passages, if a liquid is to chilled then it may freeze on the evaporator and the moisture content of air may become too low leading to water losses in the food products. In such cases multi-stage systems with multiple evaporators can be used. Several multi evaporator combinations are possible in practice. Some of the most common ones are discussed below.

1.2. Individual evaporators and a single compressor with a pressure-reducing valve

1.2.1. Individual expansion valves:

Figures 1.2 (a) and (b) show system schematic and P-h diagram of a multi evaporator system that uses two evaporators at two different temperatures and a single compressor. This system also uses individual expansion valves and a pressure regulating valve (PRV) for reducing the pressure from that corresponding to the high temperature evaporator to the compressor suction pressure. The PRV also maintains the required pressure in high temperature evaporator (Evaporator-II). Compared to the earlier system, this system offers the advantage of higher refrigeration effect at the high temperature evaporator (h_6-h_4) against (h_7-h_5).

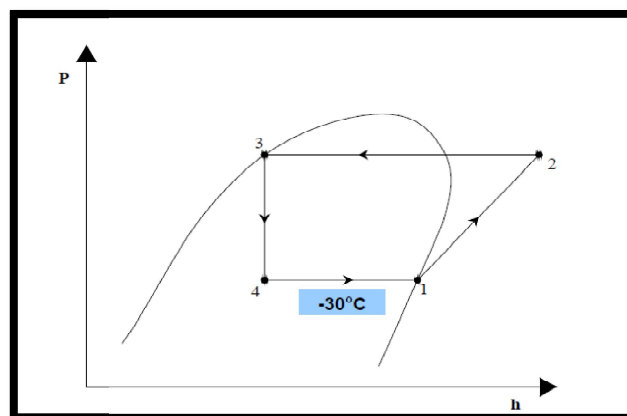
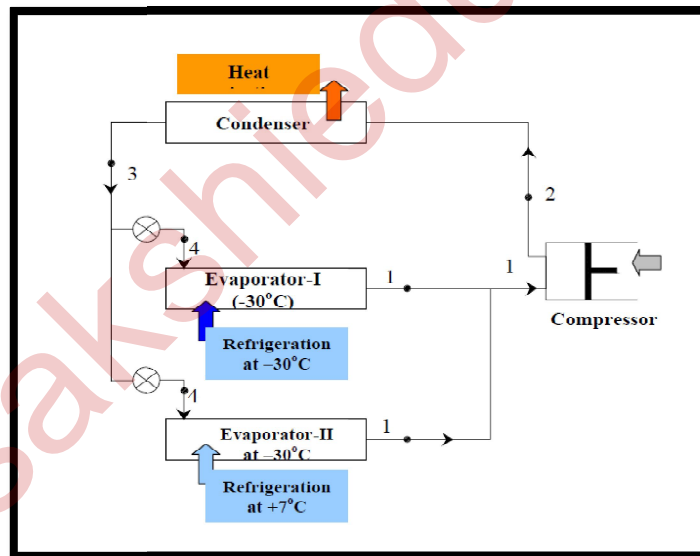


Fig.1.1(a) & (b): A single stage system with two evaporators

However, this advantage is counterbalanced by higher specific work input due to the operation of compressor in superheated region. Thus ultimately there may not be any improvement in system COP due to this arrangement. It is easy to see that this modification does not result in significant improvement in performance due to the fact that the refrigerant vapour at the intermediate pressure is reduced first using the PRV and again increased using compressor. Obviously this is inefficient. However, this system is still preferred to the earlier system due to proper operation of high temperature evaporator.

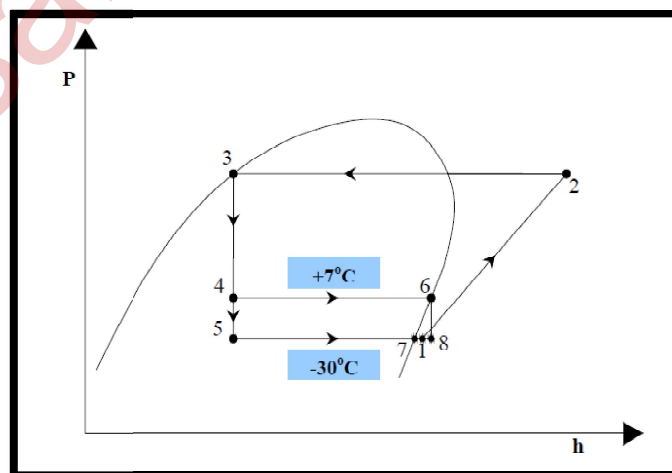
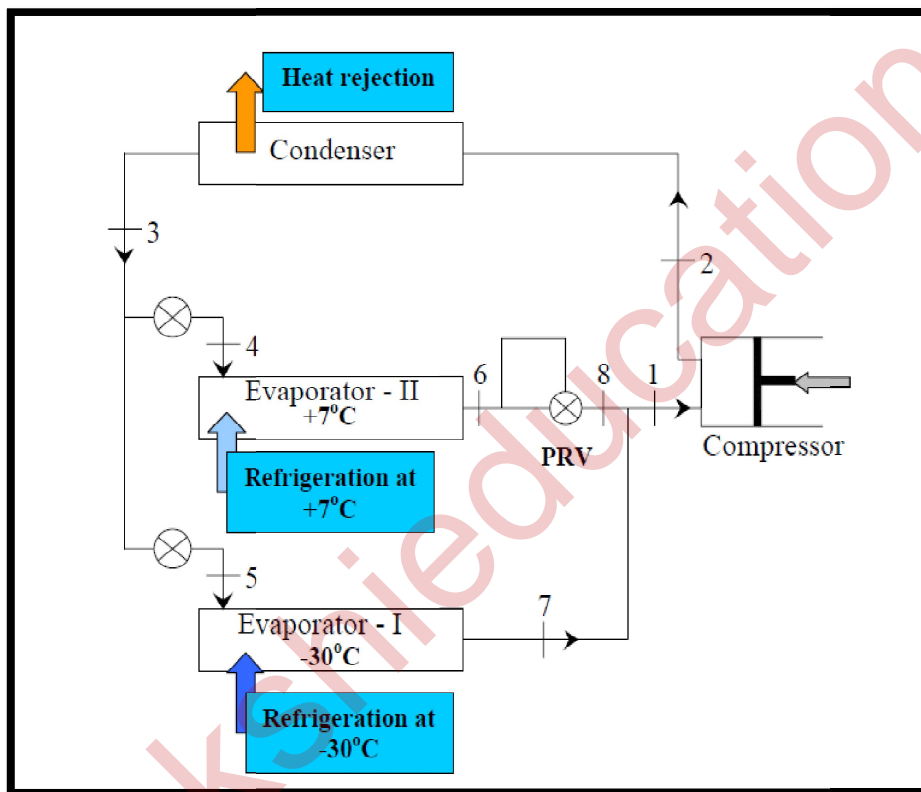


Fig.1.2(a) & (b): Multi-evaporator system with single compressor and individual Expansion valves

The COP of the above system is given by:

$$\text{COP} = \frac{Q_{e,I} + Q_{e,II}}{W_c} = \frac{\dot{m}_I(h_7 - h_5) + \dot{m}_{II}(h_6 - h_4)}{(\dot{m}_I + \dot{m}_{II})(h_2 - h_1)}$$

where \dot{m}_I and \dot{m}_{II} are the refrigerant mass flow rates through evaporator I and II respectively. They are given by:

$$\dot{m}_I = \frac{Q_{e,I}}{(h_7 - h_5)}$$

$$\dot{m}_{II} = \frac{Q_{e,II}}{(h_6 - h_4)}$$

. Enthalpy at point 2 (inlet to compressor) is obtained by applying mass and energy balance to the mixing of two refrigerant streams, i.e.,

$$h_2 = \frac{\dot{m}_I h_7 + \dot{m}_{II} h_8}{\dot{m}_I + \dot{m}_{II}}$$

If the expansion across PRV is isenthalpic, then specific enthalpy h_8 will be equal to h_6 .

1.2.2. Multiple expansion valves:

Figures 1.3 (a) as well as (b) shows system schematic as well as P-h diagram of a multiple evaporator using a single compressor as well as multiple development valves. It is usually noticed from your P-h diagram which the main advantage of this technique compared to the program with individual development valves can be which the refrigeration effect with the small temperature evaporator increases while saturated water makes its way into the lower period development control device. Because adobe flash petrol can be taken out with state 4, the lower temperature evaporator functions more effectively.

The COP of this system is given by:

$$\text{COP} = \frac{Q_{e,I} + Q_{e,II}}{W_c} = \frac{\dot{m}_I(h_8 - h_6) + \dot{m}_{II}(h_7 - h_4)}{(\dot{m}_I + \dot{m}_{II})(h_2 - h_1)}$$

Where \dot{m}_I and \dot{m}_{II} are the refrigerant mass flow rates through evaporator I and II respectively. They are given by:

$$\dot{m}_I = \frac{Q_{e,I}}{(h_8 - h_6)}$$

$$\dot{m}_{II} = \frac{Q_{e,II}}{(h_7 - h_4)}$$

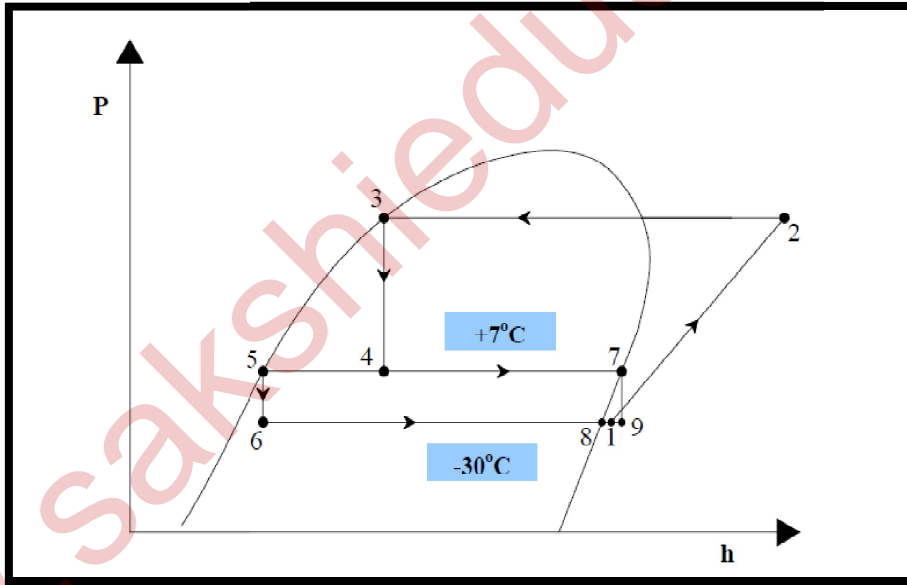
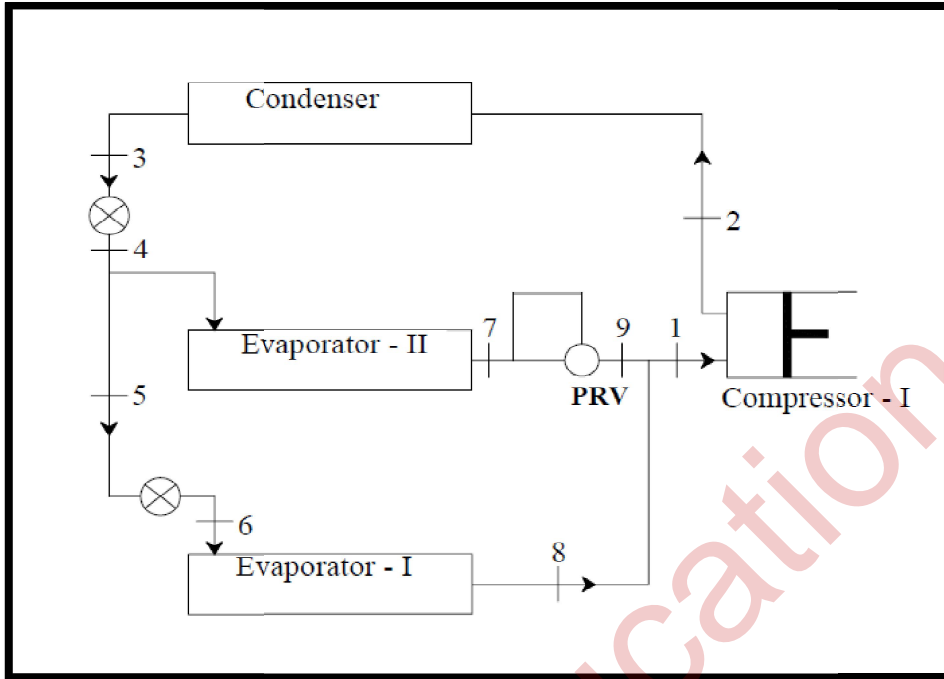


Fig.1.3(a) & (b): Multi-evaporator system with single compressor and multiple Expansion valves

Enthalpy at point 2 (inlet to compressor) is obtained by applying mass and energy balance to the mixing of two refrigerant streams, i.e.,

$$h_2 = \frac{\dot{m}_I h_8 + \dot{m}_{II} h_9}{\dot{m}_I + \dot{m}_{II}}$$

If the expansion across PRV is isenthalpic, then specific enthalpy h_7 will be equal to h_9 .

COP obtained using the above multi-evaporator systems is not much higher compared to single stage system as refrigerant vapour at intermediate pressure is first throttled then compressed, and compressor inlet is in superheated region. Performance can be improved significantly if multiple compressors are used in place of a single compressor.

1.3. Multi-evaporator system with multi-compression, inter cooling and flash gas removal

Figures 1.4(a) as well as (b) demonstrate the schematic as well as P-h diagram of any multi evaporator system which engages multiple compressors, a flash reservoir for flash gas elimination and inter chilling. This system will work for low temperature lift programs with unique refrigeration lots. For example one evaporator running at declares -40°C pertaining to quick cold of foods and additional evaporator running at -25°C pertaining to storage of frozen meal. As shown in the system schematic, the pressure in the high temperature evaporator (Evaporator-II) is just like that of flash reservoir.

Superheated vapour from the low-stage compressor is cooled towards saturation temperature in the flash reservoir. The minimal temperature evaporator operates efficiently since flash gasoline is removed in the flash reservoir. In inclusion the high-stage compressor (Compressor-II) operates efficiently for the reason that suction fumes are unhealthy. Even although high period compressor must handle greater mass stream rate on account of de-superheating of refrigerant in the flash reservoir, still the complete power input towards system might be reduced considerably, especially having refrigerants for example ammonia.

The COP of this system is given by:

$$\text{COP} = \frac{Q_{e,I} + Q_{e,II}}{W_{c,I} + W_{c,II}} = \frac{m_I(h_1 - h_8) + m_{e,II}(h_3 - h_6)}{m_I(h_2 - h_1) + m_{II}(h_4 - h_3)}$$

where m_I and m_{II} are the refrigerant mass flow rates through evaporator I and II respectively. They are given by:

$$m_I = \frac{Q_{e,I}}{(h_8 - h_6)}$$

$$m_{e,II} = \frac{Q_{e,II}}{(h_3 - h_6)}$$

m_{II} is the mass flow rate of refrigerant through the high-stage compressor which can be obtained by taking a control volume which includes the flash tank and high temperature evaporator (as shown by dashed line in the schematic) and applying mass and energy balance:

mass balance:

$$m_5 + m_2 = m_7 + m_3; m_5 = m_{II} = m_3 \text{ \& } m_2 = m_I = m_7$$

energy balance:

$$m_5 h_5 + m_2 h_2 + Q_{e,II} = m_7 h_7 + m_3 h_3$$

from known operating temperatures and evaporator loads ($Q_{e,I}$ and $Q_{e,II}$) one can get the mass flow rate through the high stage compressor and system COP from the above equations.

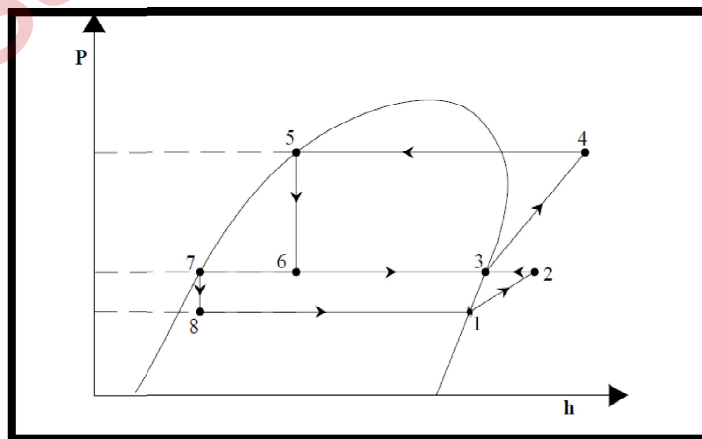
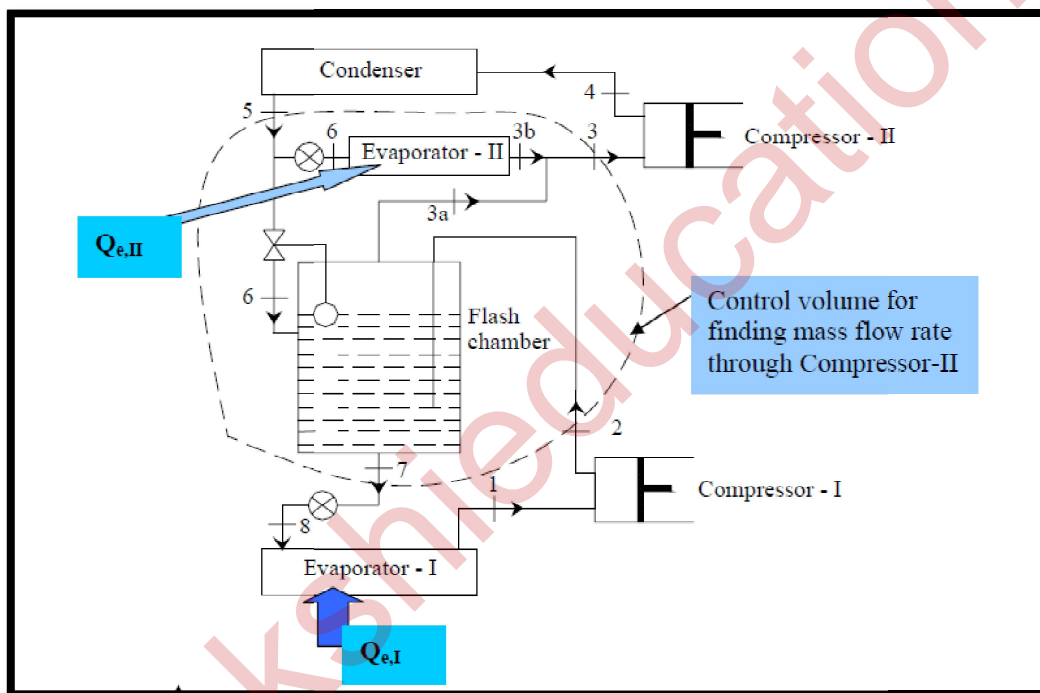


Fig.1.4(a) & (b): Multi-evaporator system with multiple compressors and a flash tank for flash gas removal and inter cooling

1.4. Multi-evaporator system with individual compressors and multiple expansion valves

Figures 1.5(a) and (b) show the schematic and P-h diagram of a multi evaporator system which employs individual compressors and multiple expansion valves.

The COP of this combined system is given by:

$$\text{COP} = \frac{Q_{e,I} + Q_{e,II}}{W_{c,I} + W_{c,II}} = \frac{m_I(h_3 - h_9) + m_{II}(h_1 - h_7)}{m_I(h_4 - h_3) + m_{II}(h_2 - h_1)}$$

Where m_I and m_{II} are the refrigerant mass flow rates through evaporator I and II respectively. They are given by:

$$m_I = \frac{Q_{e,I}}{(h_3 - h_9)}$$

$$m_{II} = \frac{Q_{e,II}}{(h_1 - h_7)}$$

The inlet to the condenser (state 5) is obtained by applying mass and energy balance to the process of mixing of refrigerant vapours from Compressors I and II.

1.5. Limitations of multi-stage systems

Though multi-stage systems have been very successful, they have certain limitations. These are:

- a) Since one refrigerant is used throughout the machine, the refrigerant used should include high critical temperature and low freezing point.
- b) The operating pressures using a single refrigerant may become too much or too low.

Generally solely R12, R22 and NH₃ systems are actually used in multi-stage systems as other conventional working fluids may well operate in vacuum at very low evaporator temperatures. Operation in vacuum leads to leakages into the system and large compressor displacement because of high specific volume.

- c) Possibility of migration of lubricating oil from one compressor to other leading to help compressor break-down.

The above limitations can be overcome by using cascade systems.

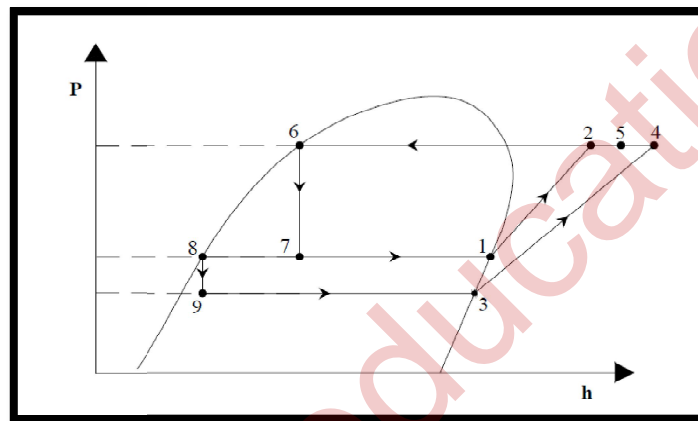
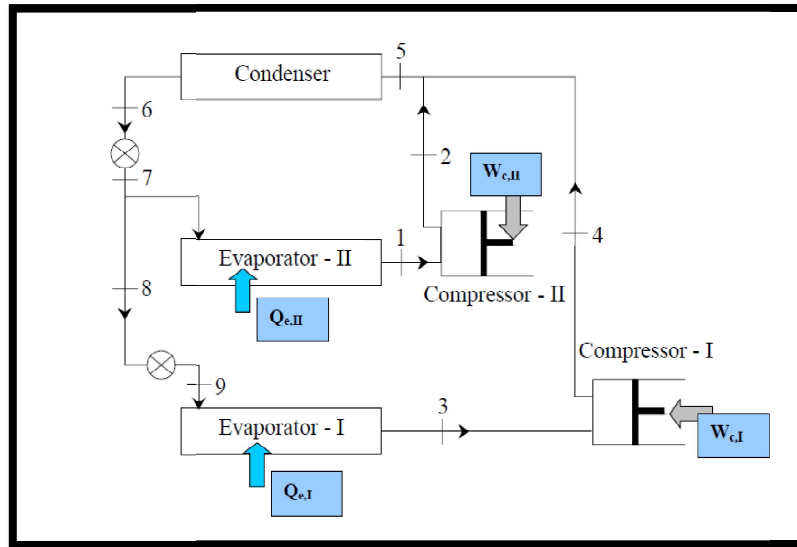


Fig.1.5(a) & (b): Multi-evaporator system with individual compressors and multiple Expansion valves

1.6. Cascade Systems

In the cascade system a few refrigerants with gradually lower boiling points are widely-used in a few single stage devices. The condenser of lower stage process is coupled for the evaporator of the next higher stage system and so forth. The component in which heat of condensation of lower stage refrigerant is supplied for vaporization of next level refrigerant is referred to as cascade condenser. Fig1.6 (a) and (b) present the schematic along with P-h diagrams of a two-stage cascade refrigeration process. As shown, it employs two distinct refrigerants operating inside two individual rounds.

They are thermally coupled inside cascade condenser. The refrigerants selected needs to have suitable pressure-temperature traits. An example of refrigerant combination is the usage of carbon dioxide (NBP = -78.4°C , $T_{cr} = 31.06^{\circ}\text{C}$) in minimal temperature cascade along with ammonia (NBP = -33.33°C , $T_{cr} = 132.25^{\circ}\text{C}$) in high temperature cascade. It is achievable to use greater than two cascade periods, and it can be possible to mix multi-stage systems along with cascade systems.

Applications of cascade systems:

- i. Liquefaction of petroleum vapours
- ii. Liquefaction of industrial gases
- iii. Manufacturing of dry ice
- iv. Deep freezing etc.

Advantages of cascade systems:

- i. Considering that every single cascade uses a distinct refrigerant, you possibly can go with a refrigerant that is most effective for the specific heat range assortment. Quite high or maybe minimal pressures may be eliminated.
- ii. Migration associated with lubricating oil in one compressor towards the other is actually prevented.

Used, complementing associated with a lot within the cascade condenser is actually hard, in particular in the process pull-down. For this reason the particular cascade condensers are normally large. Furthermore, within genuine techniques some sort of heat range difference between condensing in addition to evaporating refrigerants has to be provided within the cascade condenser that leads to be able to loss of productivity.

Furthermore, it really is discovered that at lower temps, superheating (useful or maybe useless) is actually adverse from volumetric refrigeration effect point-of-view, consequently within cascade techniques, the particular superheat needs to be just enough to avoid the particular admittance associated with fluid into compressor, with no much more for all refrigerants.

Optimum cascade temperature:

For a two-stage cascade system working on Carnot cycle, the optimum cascade temperature at which the COP will be maximum, $T_{cc,opt}$ is given by:

$$T_{cc,opt} = \sqrt{T_e \cdot T_c}$$

where T_e and T_c are the evaporator temperature of low temperature cascade and condenser temperature of high temperature cascade, respectively.

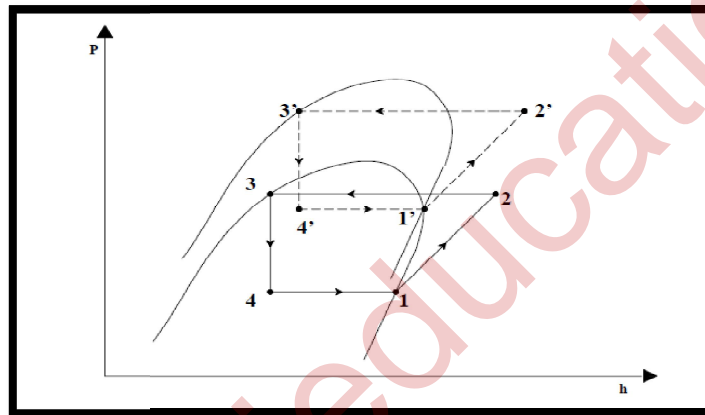
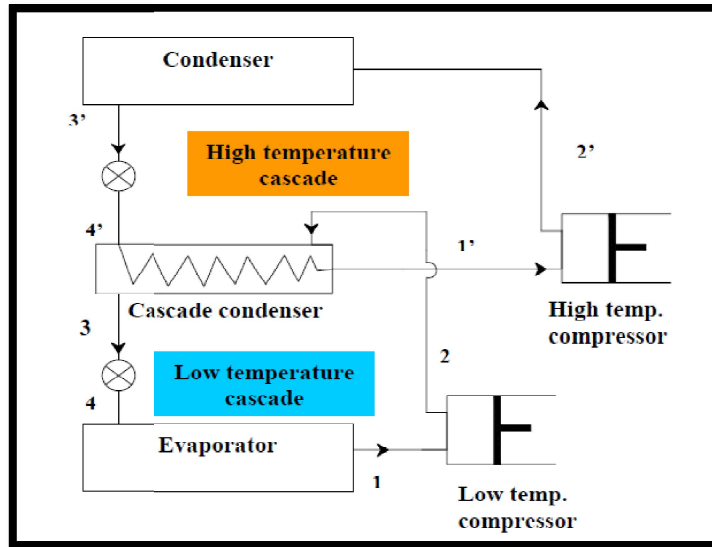


Fig.1.6(a) & (b): A two-stage cascade refrigeration system

For cascade systems employing vapour compression refrigeration cycle, the optimum cascade temperature assuming equal pressure ratios between the stages is given by:

$$T_{cc,opt} = \left(\frac{b_1 + b_2}{\frac{b_2}{T_c} + \frac{b_1}{T_e}} \right)$$

Where b_1 and b_2 are the constants in Clausius-Clayperon equation: $P = a - b/T$ for low and high temperature refrigerants, respectively.

1.7. Auto-cascade systems

An auto-cascade system may be considered as a variation of cascade system, in which a single compressor is used. The concept of auto-cascade system was first proposed by Ruhemann in 1946. Figure 1.7(a) shows the schematic of a two-stage auto-cascade cycle and Fig.1.7(b) shows the vapour pressure curves of the two refrigerants used in the cycle on D~h ring plot.

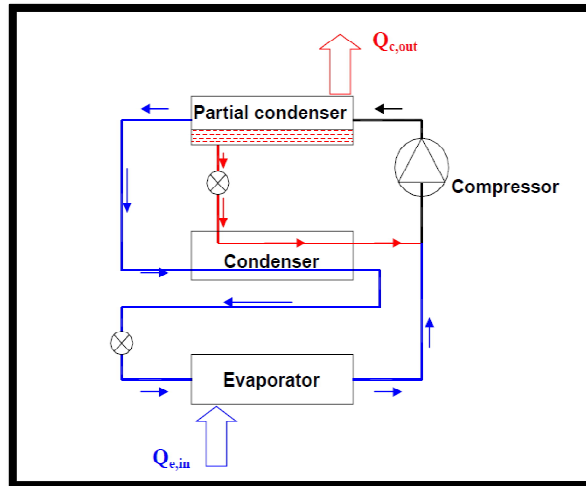


Fig.1.7(a): Schematic of a two-stage auto-cascade system

In a two-stage auto-cascade system two different working fluids; a low boiling point (low temperature) refrigerant and a high boiling point (high temperature) refrigerant are used. The vapour mixture consisting of both these refrigerants is compressed in the compressor to a discharge pressure ($P_{discharge}$). When this high pressure mixture flows through the partial condenser, the high temperature refrigerant can condense by rejecting heat ($Q_{c,out}$) to the external heat sink, if its partial pressure in the mixture is such that the saturation temperature corresponding to the partial pressure is higher than the external heat sink temperature.

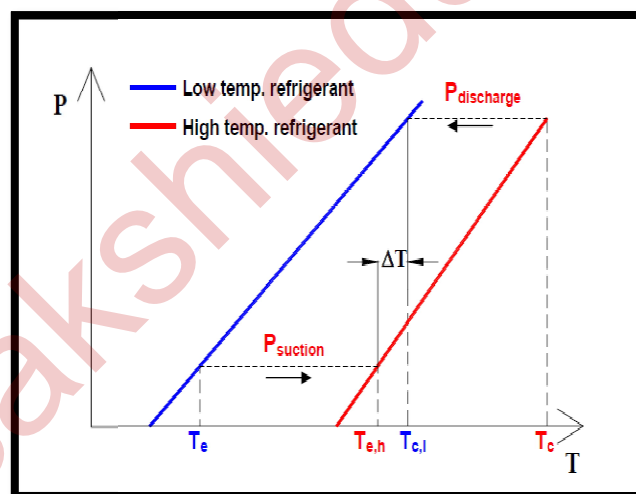


Fig.1.7(b): Schematic illustrating principle of two-stage auto cascade system on Dhiring plot

Since the saturation temperature of the low temperature refrigerant is much lower than the outer warmth sink temperature at its partial weight, it can't consolidate in the partial condenser, thus, stays as vapour. In this way it is conceivable hypothetically to discrete the high temperature refrigerant in fluid structure from the partial condenser. Next this high temperature, high weight fluid is extended through the development valve into the condenser working at weight $P_{suction}$.

Because of the extension of the high temperature refrigerant fluid from $P_{\text{discharge}}$ to P_{suction} , its temperature drops to a sufficiently low esteem ($T_{e,h}$) so that when the low temperature, high weight refrigerant vapour interacts with the high temperature, low weight refrigerant in the condenser it can consolidate at a temperature $T_{c,l}$. This dense, high weight, low temperature refrigerant is then throttled to the suction weight and is then made to move through the evaporator, where it can give the obliged refrigeration impact at a low temperature T_e . Both the high temperature refrigerant from condenser and low temperature refrigerant vapour from evaporator can be blended as they are at the same weight. This mixture is then packed in the compressor to finish the cycle.

Hence utilizing a solitary compressor, it is conceivable to get refrigeration at low temperatures utilizing the auto-course framework. Practically speaking, more than two stages with more than two refrigerants can be utilized to attain to high temperature lifts. On the other hand, in real frameworks, it is unrealistic to partitioned unadulterated refrigerants in the partial condenser as some measure of low temperature refrigerant consolidates in the partial condenser and some measure of high temperature refrigerant leaves the partial condenser in vapour structure. Accordingly all around in the system, one experiences refrigerant mixtures of fluctuating synthesis. These systems are generally utilized as a part of the liquefaction of natural gas.