Lesson
12
Multi-Stage Vapour Compression Refrigeration Systems

The objectives of this lesson are to:

1. Discuss limitations of single stage vapour compression refrigeration systems (*Section 12.1*)
2. Classify multi-stage systems (*Section 12.1*)
3. Discuss the concept of flash gas removal using flash tank (*Section 12.2*)
4. Discuss the concept of intercooling in multi-stage vapour compression refrigeration systems (*Section 12.3*)
5. Discuss multi-stage vapour compression refrigeration systems with flash gas removal and intercooling (*Section 12.4*)
6. Discuss the use of flash tank for flash gas removal only (*Section 12.5*)
7. Discuss the use of flash tank for intercooling only (*Section 12.6*)

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

1. Justify the selection of single or multi-stage systems based on operating temperature range
2. Classify multi-stage systems
3. Applying mass and energy balance equations, evaluate the performance of multi-stage vapour compression refrigeration systems with:
    a) Flash gas removal
    b) Intercooling
    c) Flash gas removal using flash tank and intercooling using flash tank and/or external intercooler
    d) Flash tank for flash gas removal only
    e) Flash tank for intercooling only, and
    f) A combination of any of the above

12.1. Introduction

A single stage vapour compression refrigeration system has one low side pressure (evaporator pressure) and one high side pressure (condenser pressure). The performance of single stage systems shows that these systems are adequate as long as the temperature difference between evaporator and condenser (*temperature lift*) is small. However, there are many applications where the temperature lift can be quite high. The temperature lift can become large either due to the requirement of very low evaporator temperatures and/or due to the requirement of very high condensing temperatures. For example, in frozen food industries the required evaporator can be as low as -40^{o}C, while in chemical industries temperatures as low as -150^{o}C may be required for liquefaction of gases. On the high temperature side the required condensing temperatures can be very high if the refrigeration system is used as a heat pump for heating applications such as process heating, drying etc. However, as the temperature lift increases the single stage systems become inefficient and impractical. For example, Fig. 12.1 shows the effect of decreasing evaporator temperatures on T s and P h diagrams. It can be seen from the T s diagrams that for a given condenser temperature, as evaporator temperature decreases:

i. Throttling losses increase
ii. Superheat losses increase
iii. Compressor discharge temperature increases
iv. Quality of the vapour at the inlet to the evaporator increases
v. Specific volume at the inlet to the compressor increases

As a result of this, the refrigeration effect decreases and work of compression increases as shown in the P h diagram. The volumic refrigeration effect also decreases rapidly as the specific volume increases with decreasing evaporator temperature. Similar effects will occur, though not in the same proportion when the condenser temperature increases for a given evaporator temperature. Due to these drawbacks, single stage systems are not recommended when the evaporator temperature becomes very low and/or when the condenser temperature becomes high. In such cases multi-stage systems are used in practice. Generally, for fluorocarbon and ammonia based refrigeration systems a single stage system is used upto an evaporator temperature of -30^{o}C. A two-stage system is used upto -60^{o}C and a three-stage system is used for temperatures below -60^{o}C.

Apart from high temperature lift applications, multi-stage systems are also used in applications requiring refrigeration at different temperatures. For example, in a dairy plant refrigeration may be required at -30^{o}C for making ice cream and at 2^{o}C for chilling milk. In such cases it may be advantageous to use a multi-evaporator system with the low temperature evaporator operating at -30^{o}C and the high temperature evaporator operating at 2^{o}C

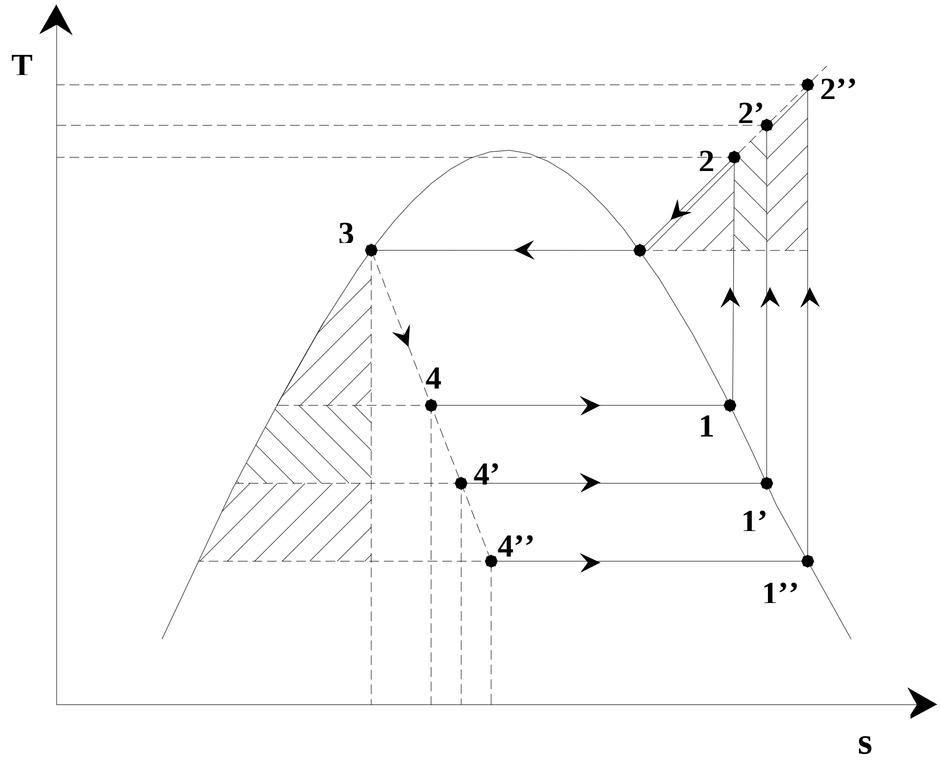

Fig.12.1(a)*: Effect of evaporator temperature on cycle performance (T-s diagram)*

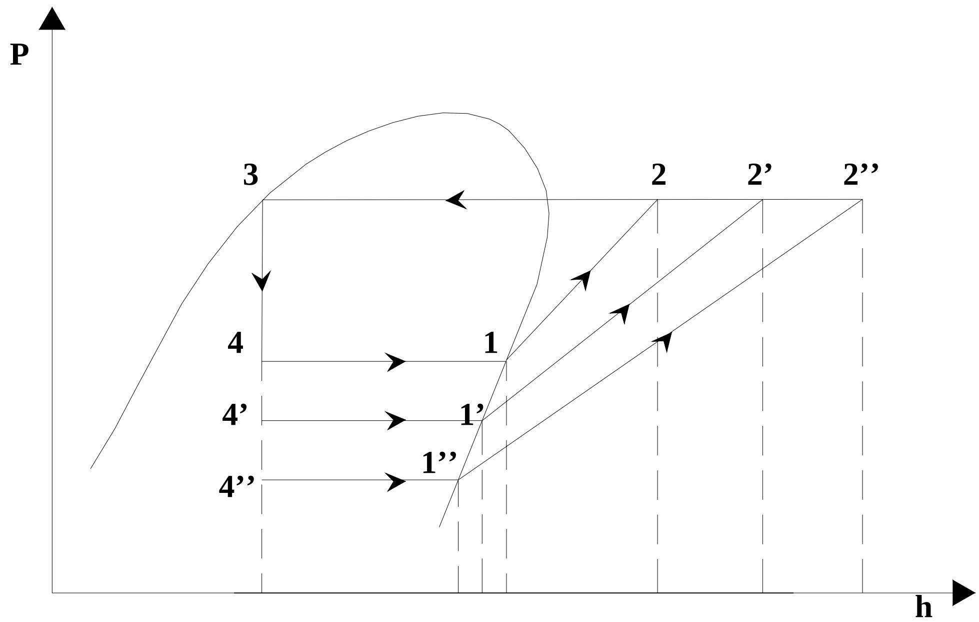

Fig.12.1(b)*: Effect of evaporator temperature on cycle performance (P-h diagram)*

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A multi-stage system is a refrigeration system with two or more low-side pressures. Multi-stage systems can be classified into:

a) Multi-compression systems
b) Multi-evaporator systems
c) Cascade systems, etc.

Two concepts which are normally integral to multi-pressure systems are, i) flash gas removal, and ii) intercooling. Hence these concepts will be discussed first.

12.2. Flash gas removal using flash tank

It is mentioned above that one of the problems with high temperature lift applications is the high quality of vapour at the inlet to the evaporator. This vapour called as *flash gas* develops during the throttling process. The flash gas has to be compressed to condenser pressure, it does not contribute to the refrigeration effect as it is already in the form of vapour, and it increases the pressure drop in the evaporator. It is possible to improve the COP of the system if the flash gas is removed as soon as it is formed and recompressed to condenser pressure. However, continuous removal of flash gas as soon as it is formed and recompressing it immediately is difficult in practice. One way of improving the performance of the system is to remove the flash gas at an intermediate pressure using a *flash tank*. Figure 12.2 shows the schematic of a flash tank and Fig.12.3 shows the expansion process employing flash tank. A flash tank is a pressure vessel, wherein the refrigerant liquid and vapour are separated at an intermediate pressure. The refrigerant from condenser is first expanded to an intermediate pressure corresponding to the pressure of flash tank, P_i using a low side float valve (process 6-7). The float valve also maintains a constant liquid level in the flash tank. In the flash tank, the refrigerant liquid and vapour are separated. The saturated liquid at point 8 is fed to the evaporator after throttling it to the required evaporator pressure, P_e (point 9) using an expansion valve. Depending upon the type of the system, the saturated vapour in the flash tank (point 3) is either compressed to the condenser pressure or throttled to the evaporator pressure. In the absence of flash tank, the refrigerant condition at the inlet to the evaporator would have been point 9’, which has a considerably high vapour quality compared to point 9. As mentioned, the refrigerant liquid and vapour must get separated in the flash tank. This is possible when the upward velocity of the refrigerant vapour in the flash tank is low enough (< 1 m/s) for the refrigerant liquid droplets to fall back into the flash tank due to gravity. Thus the surface area of liquid in the flash tank can be obtained from the volumetric flow rate of refrigerant vapour and the required low refrigerant velocity.

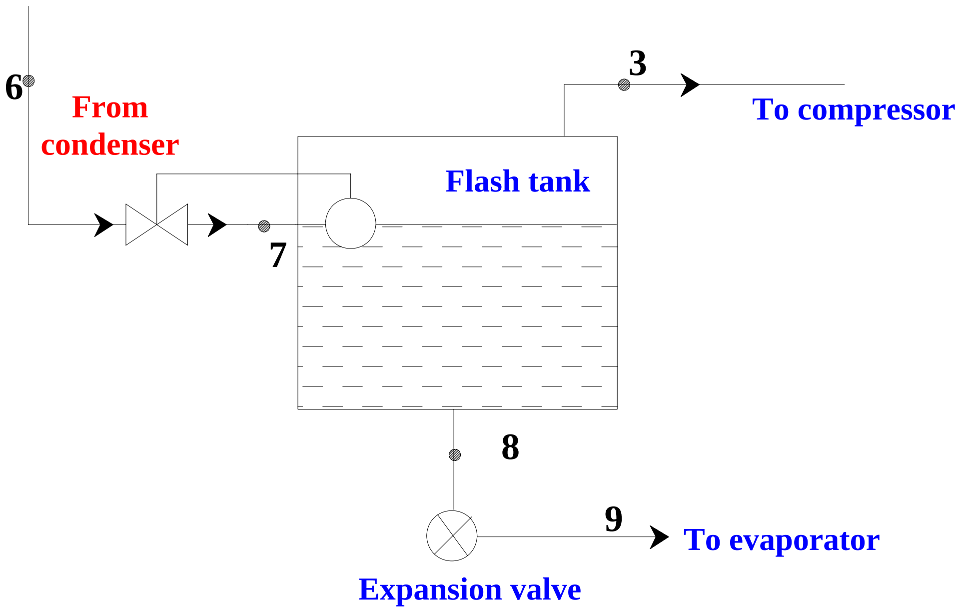


Fig.12.2(a): *Working principle of a flash tank*

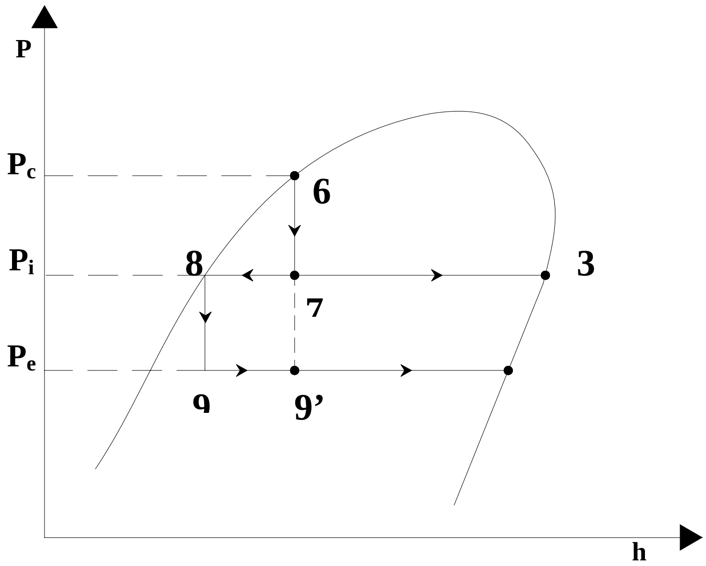


Fig.12.3: *Expansion process using a flash tank on P-h diagram*

12.3. Intercooling in multi-stage compression

The specific work input, w in reversible, polytropic compression of refrigerant vapour is given by:

$$w = -\int_1^2 v.dP = \left(\frac{n}{n-1}\right) P_1 v_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right] \qquad (12.1)$$

where P_1 and P_2 are the inlet and exit pressures of the compressor, v_1 is the specific volume of the refrigerant vapour at the inlet to the compressor and n is the polytropic exponent. From the above expression, it can be seen that specific work input reduces as specific volume, v_1 is reduced. At a given pressure, the specific volume can be reduced by reducing the temperature. This is the principle behind intercooling in multi-stage compression. Figures 12.4 (a) and (b) show the process of intercooling in two-stage compression on Pressure-specific volume (P-v) and P-h diagrams.

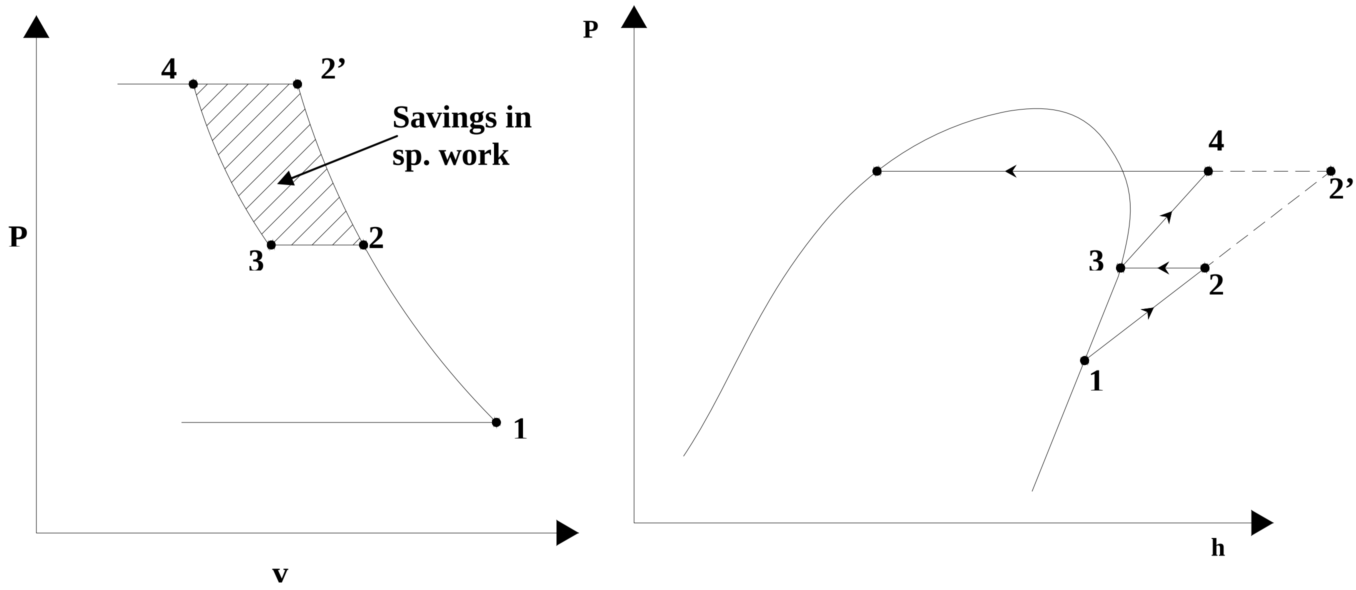


***Fig.12.4(a) & (b)**: Intercooling in two-stage compression*

As shown in the figures, in stead of compressing the vapour in a single stage from state 1 to state 2', if the refrigerant is compressed from state 1 to an intermediate pressure, state 2, intercooled from 2 to 3 and then compressed to the required pressure (state 4), reduction in work input results. If the processes are reversible, then the savings in specific work is given by the shaded area 2-3-4-2' on P-v diagram. The savings in work input can also be verified from the P-h diagram. On P-h diagram, lines 1-2-2' and 3-4 represent isentropes. Since the slope of isentropes on P-h diagram reduces (lines become flatter) as they move away from the saturated vapour line,

$$(h_4-h_3) < (h_{2'}-h_2) \Rightarrow (h_2-h_1)+(h_4-h_3) < (h_{2'}-h_1) \qquad (12.2)$$

Intercooling of the vapour may be achieved by using either a water-cooled heat exchanger or by the refrigerant in the flash tank. Figures 12.5(a) and (b) show these two systems. Intercooling may not be always possible using water-cooled heat exchangers as it depends on the availability of sufficiently cold water to which the refrigerant from low stage compressor can reject heat. Moreover, with water cooling the refrigerant at the inlet to the high stage compressor may not be saturated. Water cooling is commonly used in air compressors. Intercooling not only reduces the work input but also reduces the compressor discharge temperature leading to better lubrication and longer compressor life.

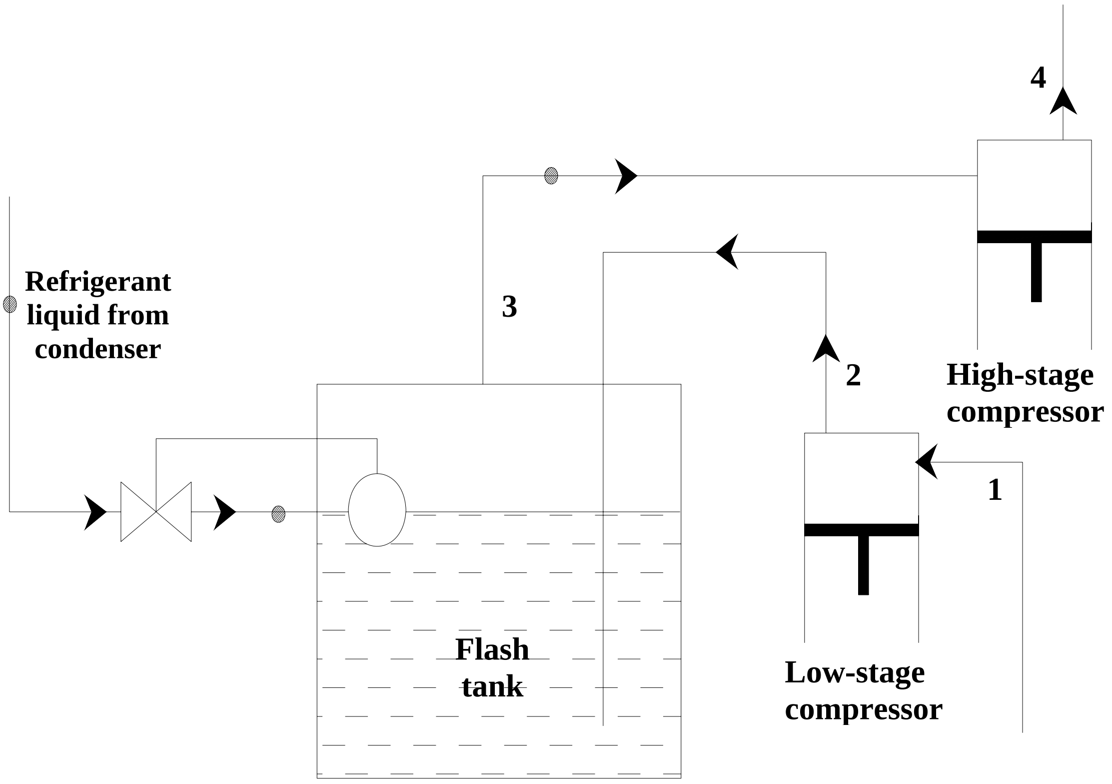


Fig.12.5(a): *Intercooling using liquid refrigerant in flash tank*

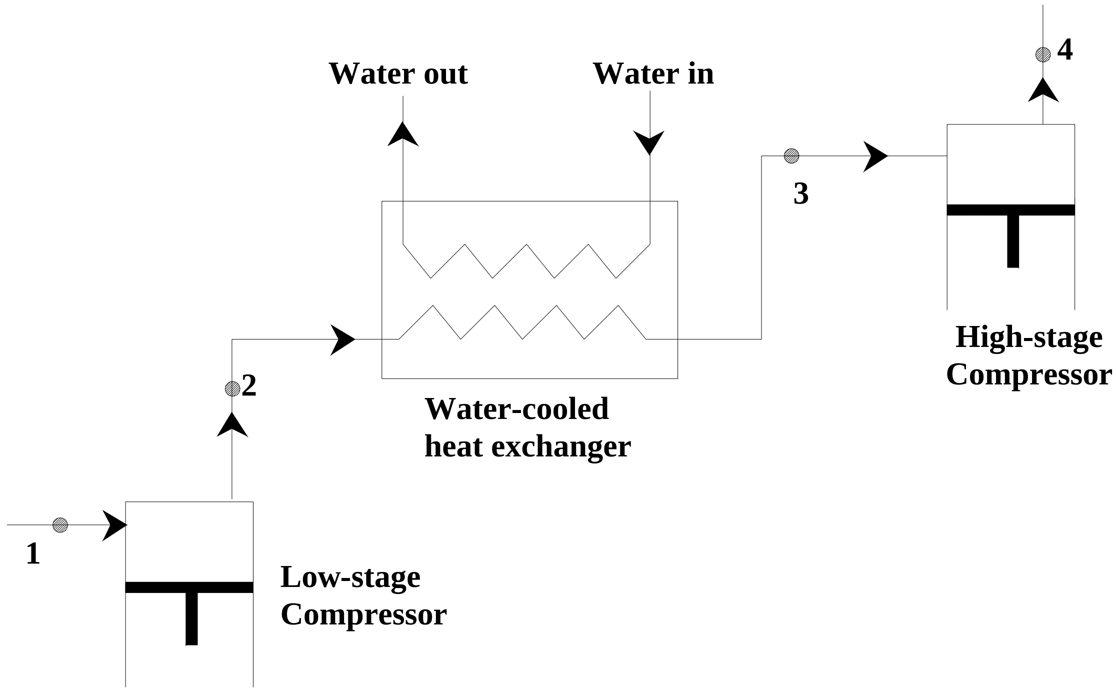


Fig.12.5(b): *Intercooling using external water cooled heat exchanger*

Intercooling using liquid refrigerant from condenser in the flash tank may or may not reduce the power input to the system, as it depends upon the nature of the refrigerant. This is due to the fact that the heat rejected by the refrigerant during intercooling generates additional vapour in the flash tank, which has to be compressed by the high stage compressor. Thus the mass flow rate of refrigerant through the high stage compressor will be more than that of the low stage compressor. Whether total power input to the system decreases or not depends on whether the increased power consumption due to higher mass flow rate is compensated by reduction in specific work of compression or not. For ammonia, the power input usually decreases with intercooling by liquid refrigerant, however, for refrigerants such as R12, R22, the power input marginally increases. Thus intercooling using liquid refrigerant is not effective for R12 and R22. However, as mentioned one benefit of intercooling is the reduction in compressor discharge temperature, which leads to better compressor lubrication and its longer life.

It is also possible to intercool the refrigerant vapour by a combination of water-cooled heat exchanger and the refrigerant liquid in the flash tank. As a result of using both water-cooling and flash-tank, the amount of refrigerant vapour handled by the high-stage compressor reduces leading to lower power consumption. However, the possibility of this again depends on the availability of cooling water at required temperature.

One of the design issues in multi-stage compression is the selection of suitable intermediate pressure. For air compressors with intercooling to the initial temperature, the theoretical work input to the system will be minimum when the pressure ratios are equal for all stages. This also results in equal compressor discharge temperatures for all compressors. Thus for a two-stage air compressor with intercooling, the optimum intermediate pressure, $P_{i,opt}$ is:

$$P_{i,opt} = \sqrt{P_{low}.P_{high}} \tag{12.3}$$

where P_{low} and P_{high} are the inlet pressure to the low-stage compressor and exit pressure from the high-stage compressor, respectively. The above relation is found to hold good for ideal gases. For refrigerants, correction factors to the above equation are suggested, for example one such relation for refrigerants is given by:

$$P_{i,opt} = \sqrt{P_e.P_c \frac{T_c}{T_e}} \tag{12.4}$$

where P_e and P_c are the evaporator and condenser pressures, and T_c and T_e are condenser and evaporator temperatures (in K).

Several combinations of multi-stage systems are used in practice. Some of them are discussed below.

12.4. Multi-stage system with flash gas removal and intercooling

Figures 12.6(a) and (b) show a two-stage vapour compression refrigeration system with flash gas removal using a flash tank, and intercooling of refrigerant vapour by a water-cooled heat exchanger and flash tank. The superheated vapour from the water cooled heat exchanger bubbles through the refrigerant liquid in the flash tank. It is assumed that in this process the superheated refrigerant vapour gets completely de-superheated and emerges out as a saturated vapour at state 4. However, in practice complete de-superheating may not be possible. As mentioned the use of combination of water cooling with flash tank for intercooling reduces the vapour generated in the flash tank. The performance of this system can be obtained easily by applying mass and energy balance equations to the individual components. It is assumed that the flash tank is perfectly insulated and the potential and kinetic energy changes of refrigerant across each component are negligible.

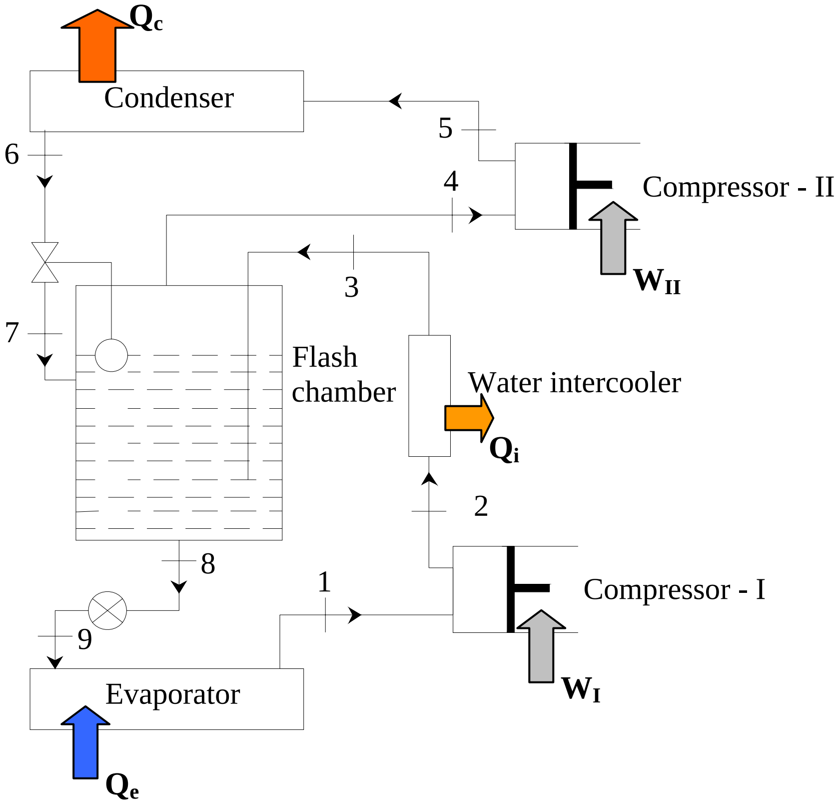


***Fig.126(a)**: Two-stage vapour compression refrigeration system with flash gas removal using a flash tank and intercooling*

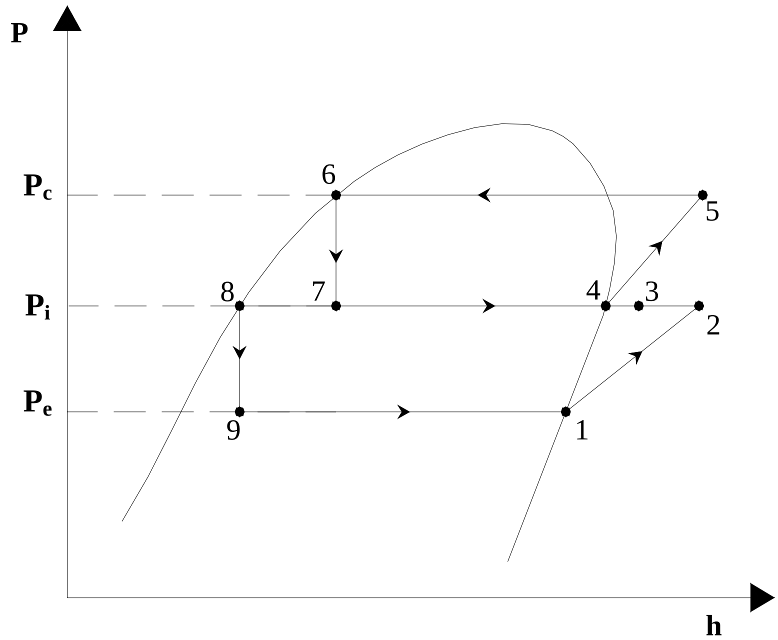


***Fig.126(b)**: Two-stage vapour compression refrigeration system with flash gas removal using a flash tank and intercooling – P-h diagram*

From mass and energy balance of the flash tank:

$$\dot{m}_7 + \dot{m}_3 = \dot{m}_8 + \dot{m}_4 \tag{12.5}$$

$$\dot{m}_7\, h_7 + \dot{m}_3\, h_3 = \dot{m}_8\, h_8 + \dot{m}_4\, h_4 \tag{12.6}$$

From mass and energy balance across expansion valve,

$$\dot{m}_8 = \dot{m}_9 \quad (12.7)$$

$$h_8 = h_9 \quad (12.8)$$

From mass and energy balance across evaporator:

$$\dot{m}_9 = \dot{m}_1 \quad (12.9)$$

$$Q_e = \dot{m}_1 (h_1 - h_9) \quad (12.10)$$

From mass and energy balance across low-stage compressor, Compressor-I:

$$\dot{m}_9 = \dot{m}_1 = \dot{m}_I \quad (12.11)$$

$$W_I = \dot{m}_I (h_2 - h_1) \quad (12.12)$$

where $\dot{m}_I$ is the mass flow rate of refrigerant through Compressor-I.

From mass and energy balance across water-cooled intercooler:

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_I \quad (12.13)$$

$$Q_I = \dot{m}_I (h_2 - h_3) \quad (12.14)$$

where Q_I is the heat transferred by the refrigerant to the cooling water in the intercooler.

From mass and energy balance across high-stage compressor, Compressor-II:

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_{II} \quad (12.15)$$

$$W_{II} = \dot{m}_{II} (h_5 - h_4) \quad (12.16)$$

where $\dot{m}_{II}$ is the mass flow rate of refrigerant through Compressor-II.

Finally, from mass and energy balance across condenser:

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_{II} \quad (12.17)$$

$$Q_c = \dot{m}_{II} (h_5 - h_6) \quad (12.18)$$

Finally, from mass and energy balance across the float valve:

$$\dot{m}_6 = \dot{m}_7 = \dot{m}_{II} \quad (12.19)$$

$$h_6 = h_7 \quad (12.20)$$

From the above set of equations, it can be easily shown that for the flash tank:

$$\dot{m}_7 = \dot{m}_4 = \dot{m}_{II} \quad (12.21)$$

$$\dot{m}_3 = \dot{m}_8 = \dot{m}_I \quad (12.22)$$

$$\dot{m}_{II} = \dot{m}_I \left[\frac{h_3 - h_8}{h_4 - h_7} \right] \quad (12.23)$$

It can be seen from the above expression that the refrigerant flow through the high-stage compression $\dot{m}_{II}$ can be reduced by reducing the enthalpy of refrigerant vapour entering into the flash tank, h_3 from the water-cooled intercooler.

The amount of additional vapour generated due to de-superheating of the refrigerant vapour from the water-cooled intercooler is given by:

$$\dot{m}_{gen} = \dot{m}_I \left[\frac{h_3 - h_4}{h_4 - h_8} \right] \qquad (12.24)$$

Thus the vapour generated $\dot{m}_{gen}$ will be zero, if the refrigerant vapour is completely de-superheated in the water-cooled intercooler itself. However, this may not be possible in practice.

For the above system, the COP is given by:

$$COP = \frac{Q_e}{W_I + W_{II}} = \frac{\dot{m}_I (h_1 - h_9)}{\dot{m}_I (h_2 - h_1) + \dot{m}_{II} (h_5 - h_4)} \qquad (12.25)$$

The above system offers several advantages,

a) Quality of refrigerant entering the evaporator reduces thus giving rise to higher refrigerating effect, lower pressure drop and better heat transfer in the evaporator
b) Throttling losses are reduced as vapour generated during throttling from P_c to P_i is separated in the flash tank and recompressed by Compressor-II.
c) Volumetric efficiency of compressors will be high due to reduced pressure ratios
d) Compressor discharge temperature is reduced considerably.

However, one disadvantage of the above system is that since refrigerant liquid in the flash tank is saturated, there is a possibility of liquid flashing ahead of the expansion valve due to pressure drop or heat transfer in the pipelines connecting the flash tank to the expansion device. Sometimes this problem is tackled by using a system with a *liquid subcooler*. As shown in Fig.12.7, in a liquid subcooler the refrigerant liquid from the condenser is subcooled by exchanging heat with the refrigerant liquid in the flash tank. As a result, a small amount of refrigerant vapour is generated in the flash tank, which needs to be compressed in the high-stage compressor. Compared to the earlier system, the temperature of refrigerant liquid from the subcooler will be higher than the saturated refrigerant temperature in the flash tank due to indirect contact heat transfer. However, since the refrigerant at the inlet to the expansion valve is at high pressure and is subcooled, there is less chance of flashing of liquid ahead of expansion valve.

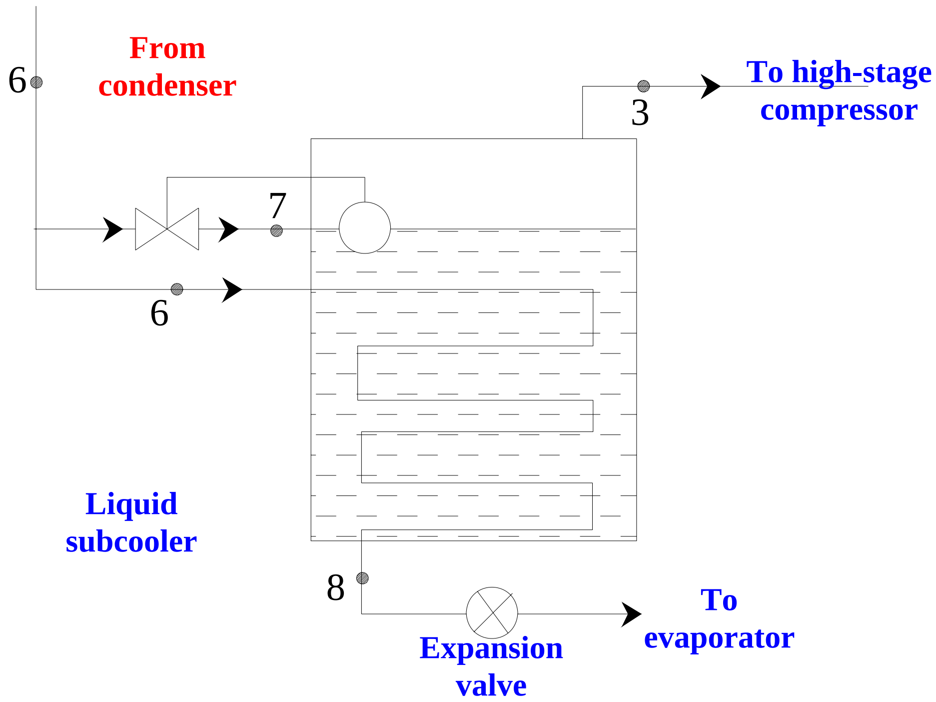
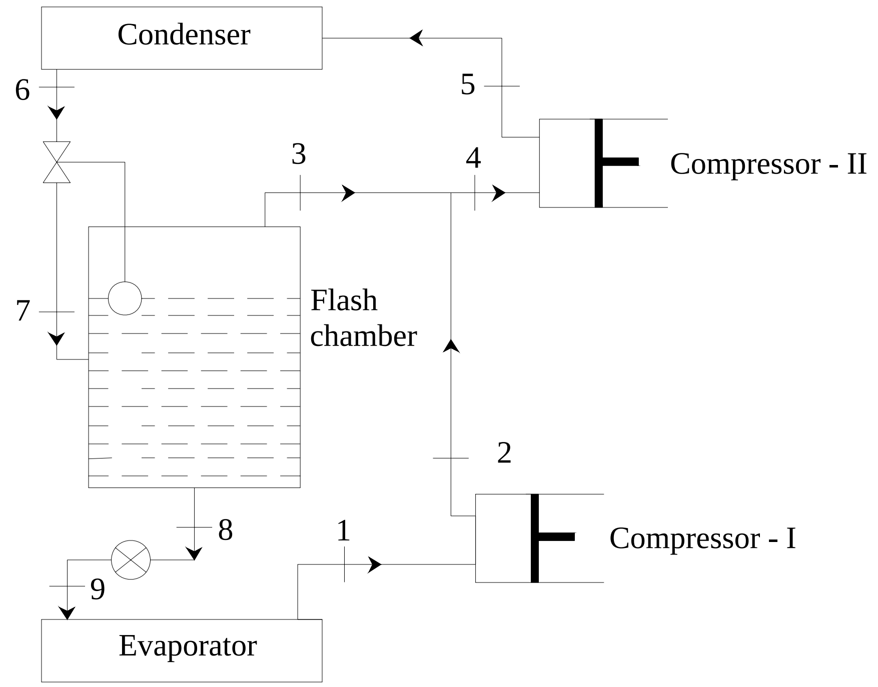


Fig.12.7: *Refrigeration system with liquid subcooler*

12.5. Use of flash tank for flash gas removal

Intercooling of refrigerant vapour using water-cooled heat exchangers is possible in ammonia systems due to high discharge temperature of ammonia. However, this is generally not possible in systems using refrigerants such as R 12 or R 134a due to their low discharge temperatures. In these systems, in stead of passing the refrigerant vapour from the low-stage compressor through the flash tank, vapour from the flash tank is mixed with the vapour coming from the low-stage compressor. As a result, the inlet condition to the high-stage compressor will be slightly superheated. A two-stage compression system with flash tank for flash gas removal for refrigerants such as R 134a is shown in Fig. 12.8 (a). Figure 12.8 (b) shows the corresponding P-h diagram.



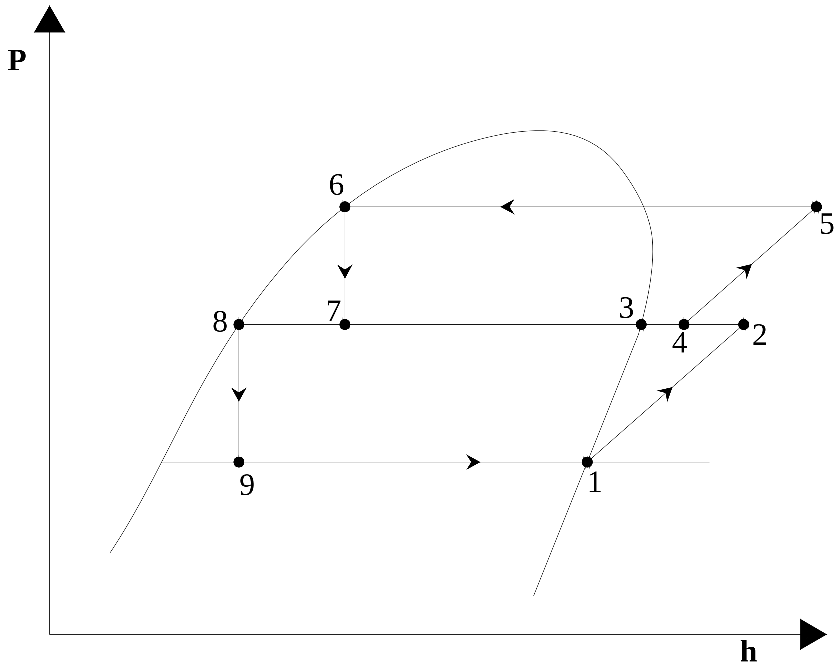


Fig.12.8: *A two-stage compression system with flash tank for flash gas removal only (a) System schematic; (b) Cycle on P-h diagram*

12.6. Use of flash tank for intercooling only

Sometimes the flash tank is used for intercooling of the refrigerant vapour between the low and high-stage compressors. It is not used for flash gas removal. Figures 12.9 (a) and (b) show the system schematic and P-h diagram of a two-stage compression system where the flash tank is used for intercooling only.

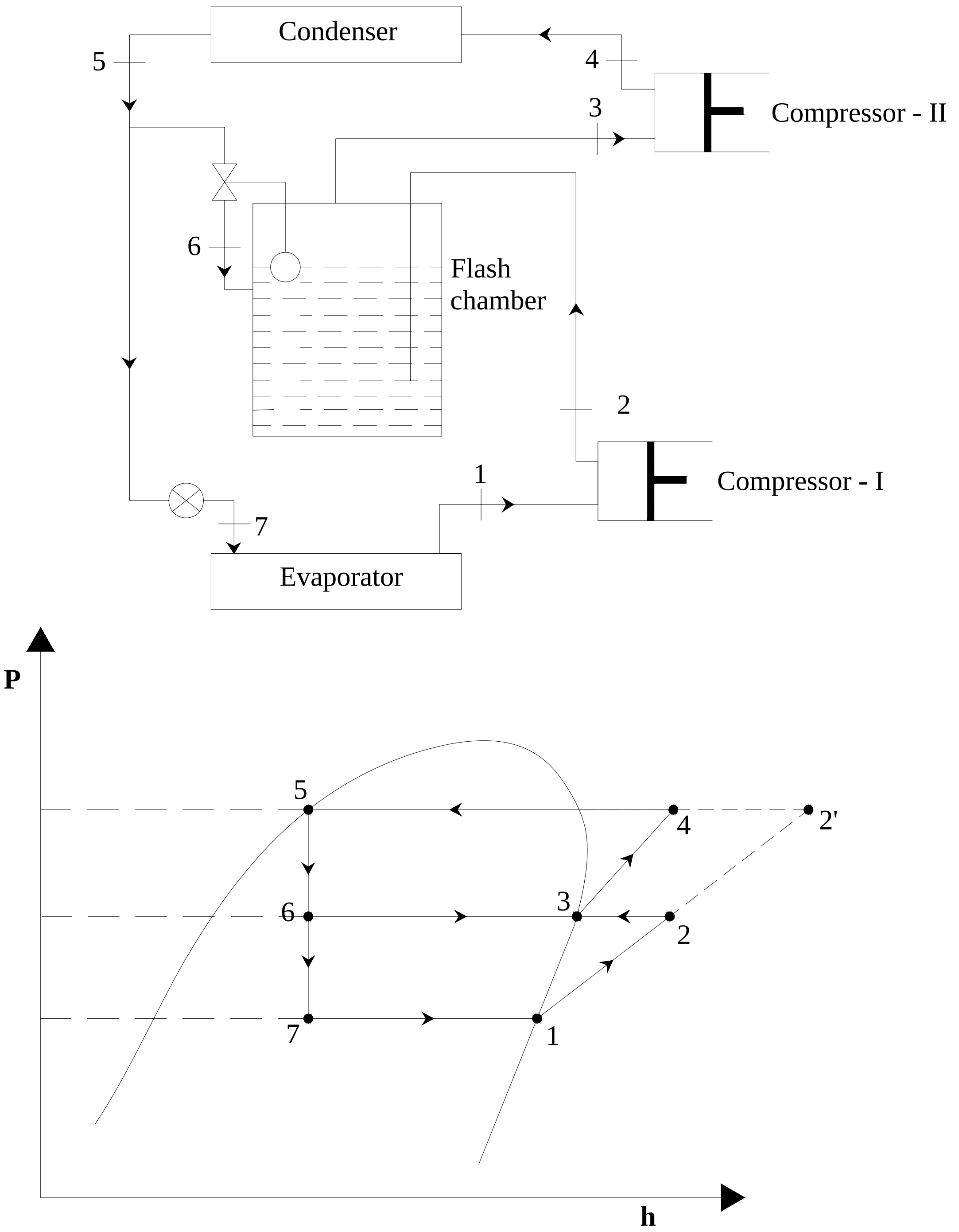


***Fig.12.9**: A two-stage compression system with the flash tank used for intercooling only*
(a) System schematic (b) Cycle on P-h diagram

Questions:

1. When the temperature lift of a single stage vapour compression refrigeration system increases:

a) Refrigeration effect increases
b) Work of compression increases
c) Compressor discharge temperature decreases
d) Volumetric efficiency of compressor increases

Ans.: b)

2. Multi-stage vapour compression refrigeration systems are used when:

a) Required temperature lift increases
b) Required temperature lift decreases
c) Refrigeration is required at different temperatures
d) Required refrigeration capacity is large

Ans.: a) and c)

3. Using a flash tank:

a) Flash gas formed during expansion can be removed at an intermediate pressure
b) Quality of refrigerant at the evaporator inlet can be increased
c) Temperature of refrigerant vapour at the inlet to higher stage compressor can be reduced
d) Pressure drop in evaporator can be reduced

Ans.: a) , c) and d)

4. Using intercooling in multi-stage compression systems:

a) Refrigeration effect can be increased
b) Work of compression in higher stage compressor can be reduced
c) Maximum cycle temperature can be increased
d) All of the above

Ans.: b)

5. External intercooling of refrigerant vapour:

a) Is feasible for ammonia based systems
b) Commonly used in air compressors
c) Commonly used for halocarbon refrigerants
d) Depends on availability of cold external water

Ans.: a) and b)

6. Assuming the refrigerant vapour to behave as an ideal gas and with perfect intercooling, the optimum intermediate pressure of a refrigeration system that operates between 4 bar and 16 bar is equal to:

a) 10 bar
b) 8 bar
c) 6 bar
d) 12 bar

Ans.: b)

7. Refrigeration system with liquid subcooler is used to:

a) Prevent the entry of liquid into compressor
b) Prevent flashing of refrigerant liquid ahead of low stage expansion device
c) Reduce work of compression
d) All of the above

Ans. b)

8. In two-stage compression system with flash gas removal:

a) Refrigerant mass flow rates in both low and high stage compressors are equal
b) Refrigerant mass flow rates in high stage compressors is greater than that in low stage compressor
c) Refrigerant mass flow rates in high stage compressors is smaller than that in low stage compressor
d) Mass flow rates in low and high stage compressors are equal if the pressure ratios are equal

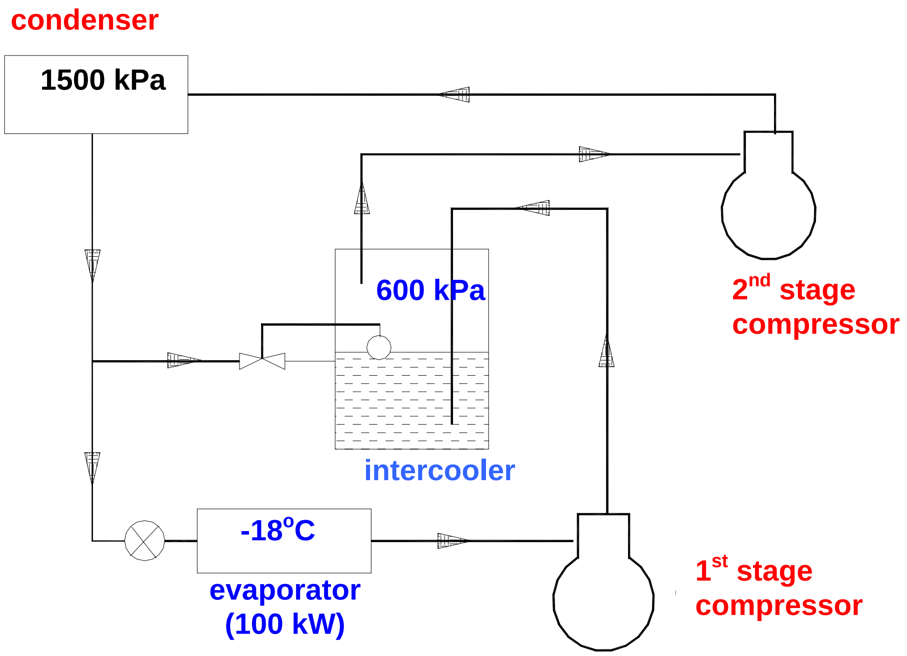
Ans.: b)

9. Use of flash tank for intercooling:

a) Always improves system COP
b) COP increases or decreases depends on the refrigerant used
c) Maximum compressor discharge temperature always decreases
d) Power input to the system always decreases

Ans.: b) and c)

10. The required refrigeration capacity of a vapour compression refrigeration system (with R-22 as refrigerant) is 100 kW at -30^oC evaporator temperature. Initially the system was single-stage with a single compressor compressing the refrigerant vapour from evaporator to a condenser operating at 1500 kPa pressure. Later the system was modified to a two-stage system operating on the cycle shown below. At the intermediate pressure of 600 kPa there is intercooling but no removal of flash gas. Find a) Power requirement of the original single-stage system; b) Total power requirement of the two compressors in the revised two-stage system. Assume that the state of refrigerant at the exit of evaporator, condenser and intercooler is saturated, and the compression processes are isentropic.

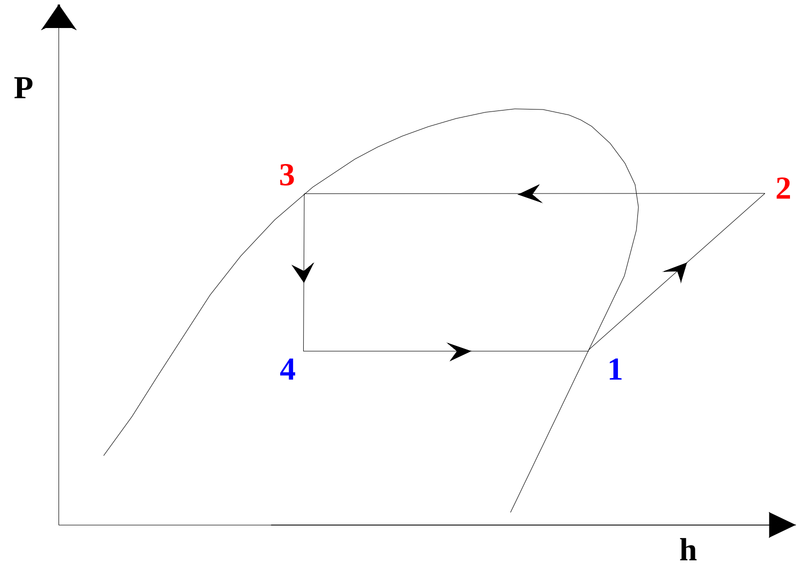


Ans.:

From refrigerant property data, the following values are obtained for R 22:

Point	Temp.,°C	Pressure,kPa	Dryness fraction	Density,kg/m³	Enthalpy, kJ/kg	Entropy, kJ/kg.K
1	-30	163.9	1.0	7.379	392.7	1.802
3	39.1	1500	0.0	-	248.4	-
2	76.93	1500	-	-	449.9	1.802
2'"	53.55	1500	-	-	429.6	1.742
2"	5.86	600	1.0	-	407.2	1.742
2'	28.94	600	-	-	424.4	1.802

a) Single stage system:



Required refrigerant mass flow rate, m_r is given by:

$$m_r = Q_e/(h_1 - h_4) = 100/(392.7 - 248.4) = \mathbf{0.693\ kg/s}$$

Power input to compressor, W_c is given by:

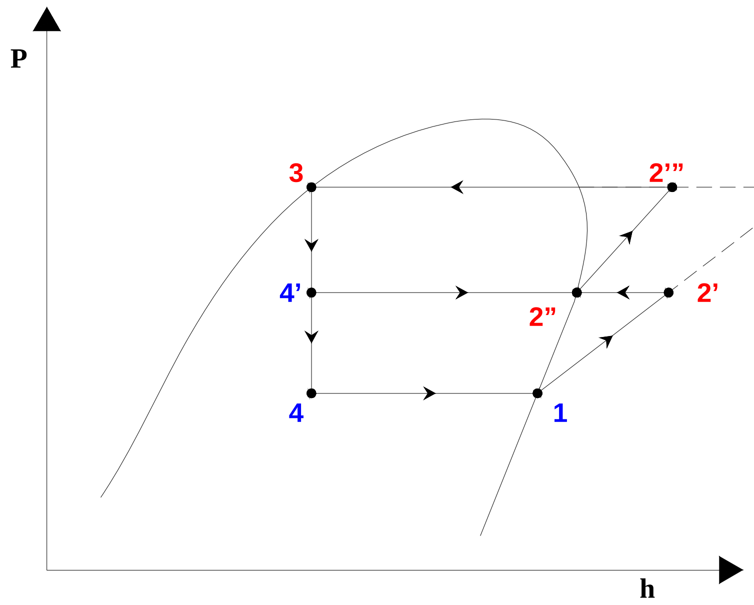
$$W_c = m_r(h_2 - h_1) = 0.693(449.9 - 392.7) = \mathbf{39.64\ kW}$$

COP of the single stage system is given by:

$$COP = Q_e/W_c = 100/39.64 = \mathbf{2.523}$$

Compressor discharge temperature = **76.93 ^{o}C** (from property data)

Two-stage system with flash tank for intercooling only:



Required refrigerant mass flow rate through evaporator and 1st stage compressor (**$m_{r,1}$**) is same as that of single stage system, i.e.,

$$\mathbf{m_{r,1} = 0.693\ kg/s}$$

Power input to 1st stage compressor, $\mathbf{W_{c,1}}$ is given by:

$$W_{c,1} = m_{r,1}(h_{2'} - h_1) = 0.693(424.4 - 392.7) = \mathbf{21.97\ kW}$$

The mass flow rate of refrigerant vapour through 2nd stage compressor ($\mathbf{m_{r,2}}$) is obtained from energy balance across intercooler:

$$m_{r,2}.h_{2''} = m_{r,1}.h_{2'} + (m_{r,2} - m_{r,1}).h_{4'}$$

Substituting the values of enthalpy and mass flow rate through 1st stage compressor:

$$m_{r,2} = \mathbf{0.768\ kg/s}$$

Power input to 2nd stage compressor, $\mathbf{W_{c,2}}$ is given by:

$$W_{c,2} = m_{r,2}(h_{2'''} - h_{2''}) = 0.768(429.6 - 407.2) = \mathbf{17.2\ kW}$$

Therefore, total power input, Wc is given by:

$$W_c = W_{c,1}+W_{c,2} = 21.97+17.2 = \mathbf{39.17\ kW}$$

COP of the two-stage system is given by:

$$COP = Q_e/(W_{c,1}+W_{c,2}) = 100/39.17 = \mathbf{2.553}$$

From property data, the discharge temperatures at the exit of 1st and 2nd stage compressors are given, respectively by:

$$\mathbf{T2' = 28.94^oC}$$
$$\mathbf{T2''' = 53.55^oC}$$

Comments:

It is observed from the above example that for the given input data, though the use of a two-stage system with intercooling in place of a single stage system does not increase the COP significantly (≈ **1.2 %**), there is a significant reduction in the maximum compressor discharge temperature (≈ **24^{o}C).** The results would be different if the operating conditions and/or the refrigerant used is different.