

Lesson

11

Vapour Compression Refrigeration Systems: Performance Aspects And Cycle Modifications

The objectives of this lecture are to discuss

1. Performance aspects of SSS cycle and the effects of evaporator and condensing temperatures on system performance (*Section 11.1*)
2. Modifications to the basic SSS cycle by way of subcooling and superheating and effects of these modifications on system performance (*Section 11.2.1*)
3. Performance aspects of single stage VCRS cycle with Liquid-to-Suction Heat Exchanger and the concept of Grindley's cycle (*Section 11.2.2*)
4. Effect of superheat and criteria for optimum superheat (*Section 11.3*)
5. Actual vapour compression refrigeration systems (*Section 11.4*)
6. Complete vapour compression refrigeration systems (*Section 11.5*)

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

1. Show and discuss qualitatively the effects of evaporator and condensing temperatures on specific and volumic refrigeration effects, on specific and volumic work of compression and on system COP
2. Discuss and evaluate the performance of single stage VCRS with subcooling and superheating from given inputs and known refrigerant property data
3. Evaluate the performance of the system with a LSHX
4. Establish the existence of optimum superheat condition using Ewings-Gosney criteria
5. Evaluate the COP of actual VCRS from condensing and evaporator temperatures, efficiency of motor and compressor
6. Draw an actual VCRS cycle on T-s and P-h diagrams and discuss the effects of various irreversibilities due to pressure drops, heat transfer and non-ideal compression
7. Describe briefly a complete vapour compression refrigeration system

11.1. Performance of SSS cycle

The performance of a standard VCRS cycle can be obtained by varying evaporator and condensing temperatures over the required range. Figure 11.1 shows the effects of evaporator and condensing temperatures on specific and volumic refrigeration effects of a standard VCRS cycle. As shown in the figure, for a given condenser temperature as evaporator temperature increases the specific refrigeration effect increases marginally. It can be seen that for a given evaporator temperature, the refrigeration effect decreases as condenser temperature increases. These trends can be explained easily with the help of the P-h diagram. It can also be observed that the volumetric refrigeration effect increases rapidly with evaporator temperature due to the increase in specific refrigeration effect and decrease in specific volume of refrigerant vapour at the inlet to the compressor. Volumetric refrigeration effect increases marginally as condenser temperature decreases.

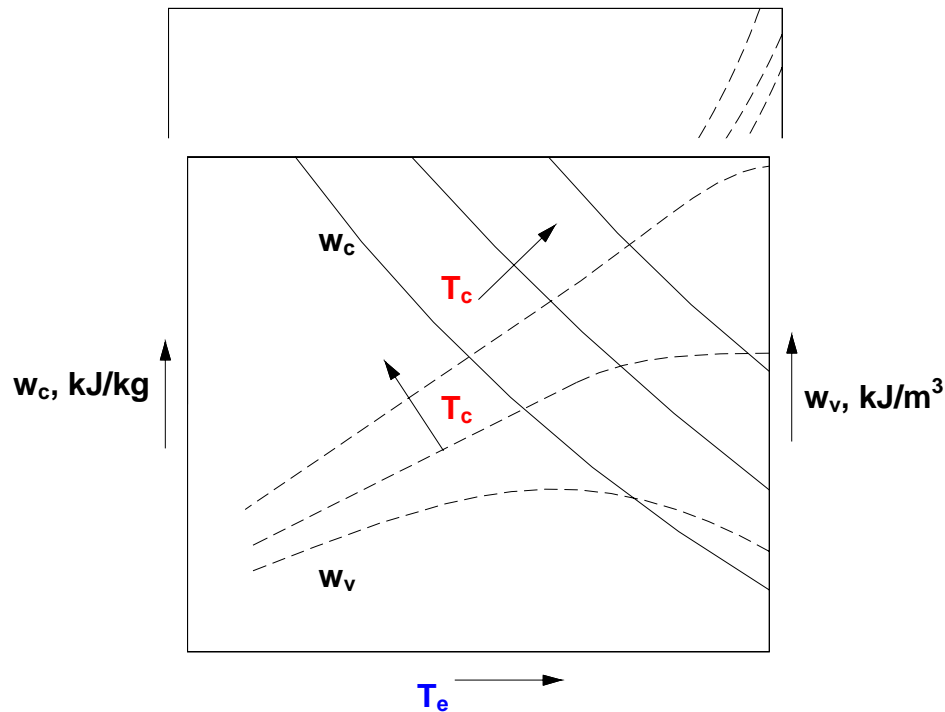


Fig.11.2: Effect of evaporator and condenser temperatures on specific and volumic works of compression of a standard VCRS cycle

Figure 11.2 shows that the specific work of compression decreases rapidly as the evaporator temperature increases and condenser temperature decreases. Once again these effects can be explained using a T s or P h diagram. For a given condenser temperature, the volumic work of compression increases initially, reaches a peak, then starts decreasing. This is due to the fact that as evaporator temperature increases the specific work of compression decreases and the specific volume at the inlet to the compressor also decreases. As a result, an optimum evaporator temperature exists at which the volumic work of compression reaches a maximum. Physically, the volumic work of compression is analogous to mean effective pressure of the compressor, as multiplying this with the volumetric flow rate gives the power input to the compressor. For a given power input, a high volumic work of compression implies smaller volumetric flow rates and hence a smaller compressor.

Figure 11.3 shows the effect of evaporator and condenser temperatures on COP of the SSS cycle. As expected, for a given condenser temperature the COP increases rapidly with evaporator temperature, particularly at low condensing temperatures. For a given evaporator temperature, the COP decreases as condenser temperature increases. However, the effect of condenser temperature becomes marginal at low evaporator temperatures.

The above results show that at very low evaporator temperatures, the COP becomes very low and also the size of the compressor becomes large (due to small volumic refrigeration effect). It can also be shown that the compressor discharge temperatures also increase as the evaporator temperature decreases. Hence, single stage vapour compression refrigeration systems are not viable for very low evaporator temperatures. One has to use multistage or cascade systems for these applications. These systems will be discussed in the next lecture. One can also observe the similarities in performance trends between SSS cycle and Carnot cycle, which is to be expected as the VCRS cycle is obtained by modifying the SSS cycle.

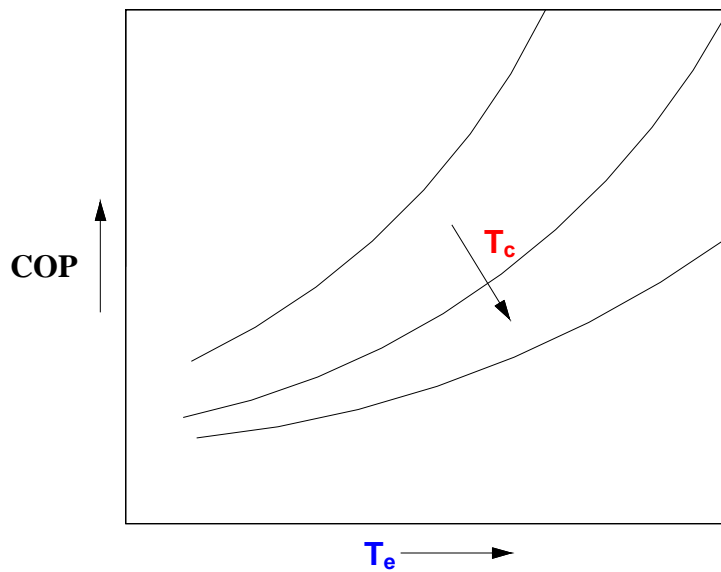


Fig.11.3: Effect of evaporator and condenser temperatures on COP of a standard VCRS cycle

11.2. Modifications to SSS cycle

11.2.1. Subcooling and superheating:

In actual refrigeration cycles, the temperature of the heat sink will be several degrees lower than the condensing temperature to facilitate heat transfer. Hence it is possible to cool the refrigerant liquid in the condenser to a few degrees lower than the condensing temperature by adding extra area for heat transfer. In such a case, the exit condition of the condenser will be in the subcooled liquid region. Hence this process is known as *subcooling*. Similarly, the temperature of heat source will be a few degrees higher than the evaporator temperature, hence the vapour at the exit of the evaporator can be superheated by a few degrees. If the superheating of refrigerant takes place due to heat transfer with the refrigerated space (low temperature heat source) then it is called as *useful superheating* as it increases the refrigeration effect. On the other hand, it is possible for the refrigerant vapour to become superheated by exchanging heat with the surroundings as it flows through the connecting pipelines. Such a superheating is called as useless superheating as it does not increase refrigeration effect.

Subcooling is beneficial as it increases the refrigeration effect by reducing the throttling loss at no additional specific work input. Also subcooling ensures that only liquid enters into the throttling device leading to its efficient operation. Figure 11.4 shows the VCRS cycle without and with subcooling on P-h and T-s coordinates. It can be seen from the T-s diagram that without subcooling the throttling loss is equal to the hatched area **b-4'-4-c**, whereas with subcooling the throttling loss is given by the area **a-4''-4'-b**. Thus the refrigeration effect increases by an amount equal to $(h_4 - h_{4'}) = (h_3 - h_{3'})$. Another practical advantage of subcooling is that there is less vapour at the inlet to the evaporator which leads to lower pressure drop in the evaporator.

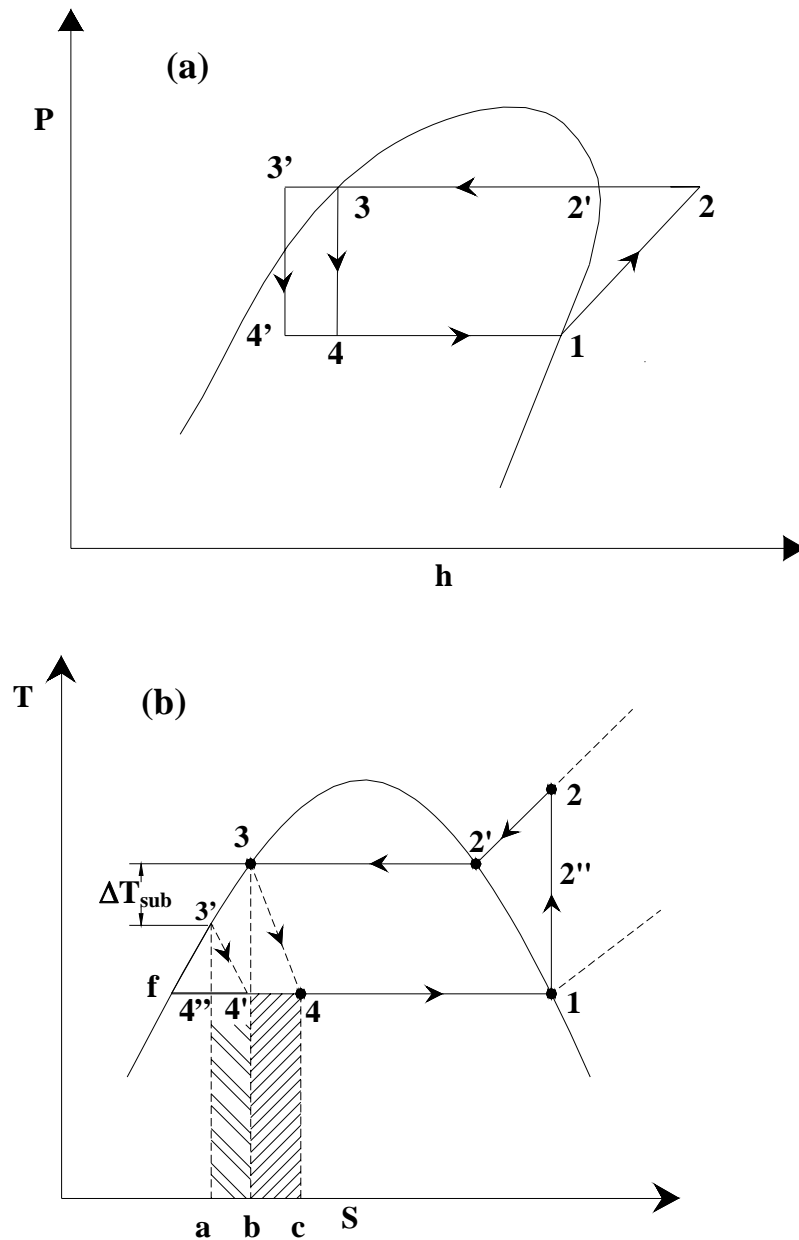


Fig.11.4: Comparison between a VCRES cycle without and with subcooling
(a) on P-h diagram (b) on T-s diagram

Useful superheating increases both the refrigeration effect as well as the work of compression. Hence the COP (ratio of refrigeration effect and work of compression) may or may not increase with superheat, depending mainly upon the nature of the working fluid. Even though useful superheating may or may not increase the COP of the system, a minimum amount of superheat is desirable as it prevents the entry of liquid droplets into the compressor. Figure 11.5 shows the VCRES cycle with superheating on P-h and T-s coordinates. As shown in the figure, with useful superheating, the refrigeration effect, specific volume at the inlet to the compressor and work of compression increase. Whether the volumic refrigeration effect (ratio of refrigeration effect by specific volume at compressor inlet) and COP increase or not depends upon the relative increase in refrigeration effect and work of compression, which in turn depends upon the nature of

the refrigerant used. The temperature of refrigerant at the exit of the compressor increases with superheat as the isentropes in the vapour region gradually diverge.

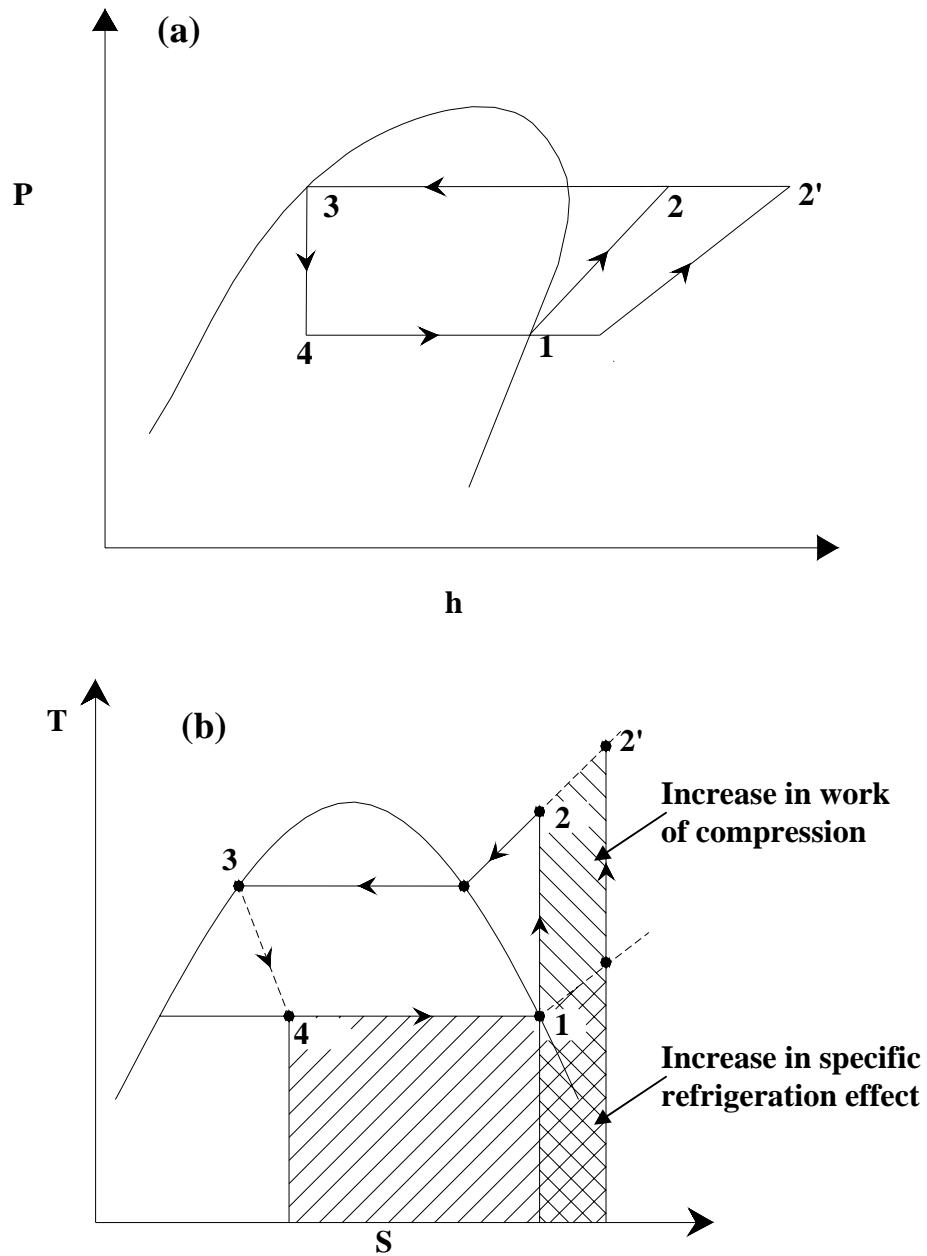


Fig.11.5: Effect of superheat on specific refrigeration effect and work of compression (a) on P-h diagram (b) on T-s diagram

11.2.2. Use of liquid-suction heat exchanger:

Required degree of subcooling and superheating may not be possible, if one were to rely only on heat transfer between the refrigerant and external heat source and sink. Also, if the temperature of refrigerant at the exit of the evaporator is not sufficiently superheated, then it may get superheated by exchanging heat with the surroundings as it flows through the connecting pipelines (useless superheating), which is detrimental to system performance. One way of achieving the required amount of subcooling and superheating is by the use of a liquid-suction heat exchanger (LSHX). A LSHX is a counterflow heat exchanger in which the warm refrigerant liquid from the condenser exchanges heat with the cool refrigerant vapour from the evaporator. Figure 11.6 shows the schematic of a single stage VCRES with a liquid-suction heat exchanger. Figure 11.7 shows the modified cycle on T-s and P-h diagrams. As shown in the T-s diagram, since the temperature of the refrigerant liquid at the exit of condenser is considerably higher than the temperature of refrigerant vapour at the exit of the evaporator, it is possible to subcool the refrigerant liquid and superheat the refrigerant vapour by exchanging heat between them.

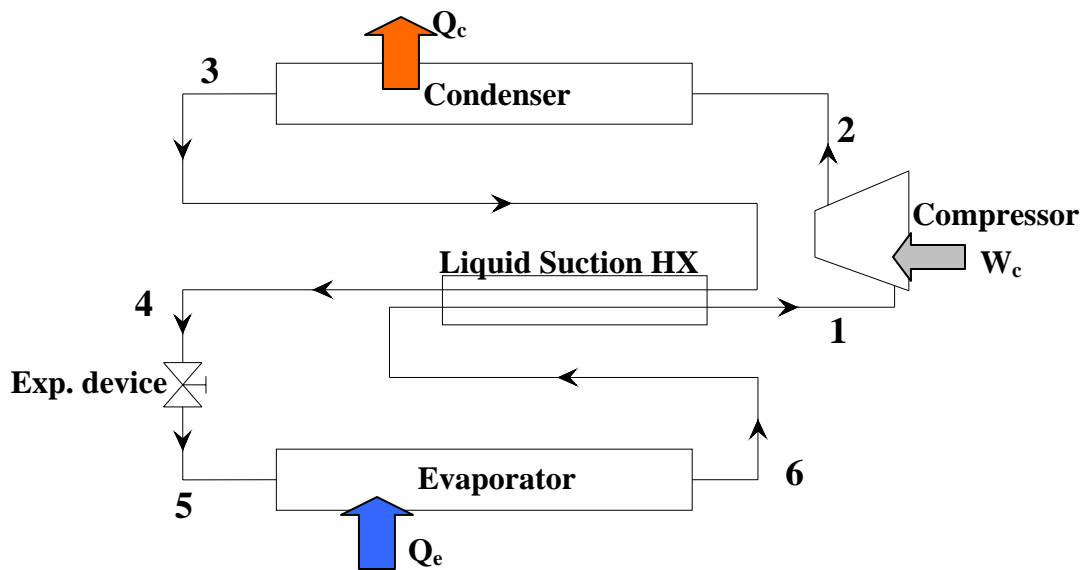


Fig.11.6: A single stage VCRES system with Liquid-to-Suction Heat Exchanger (LSHX)

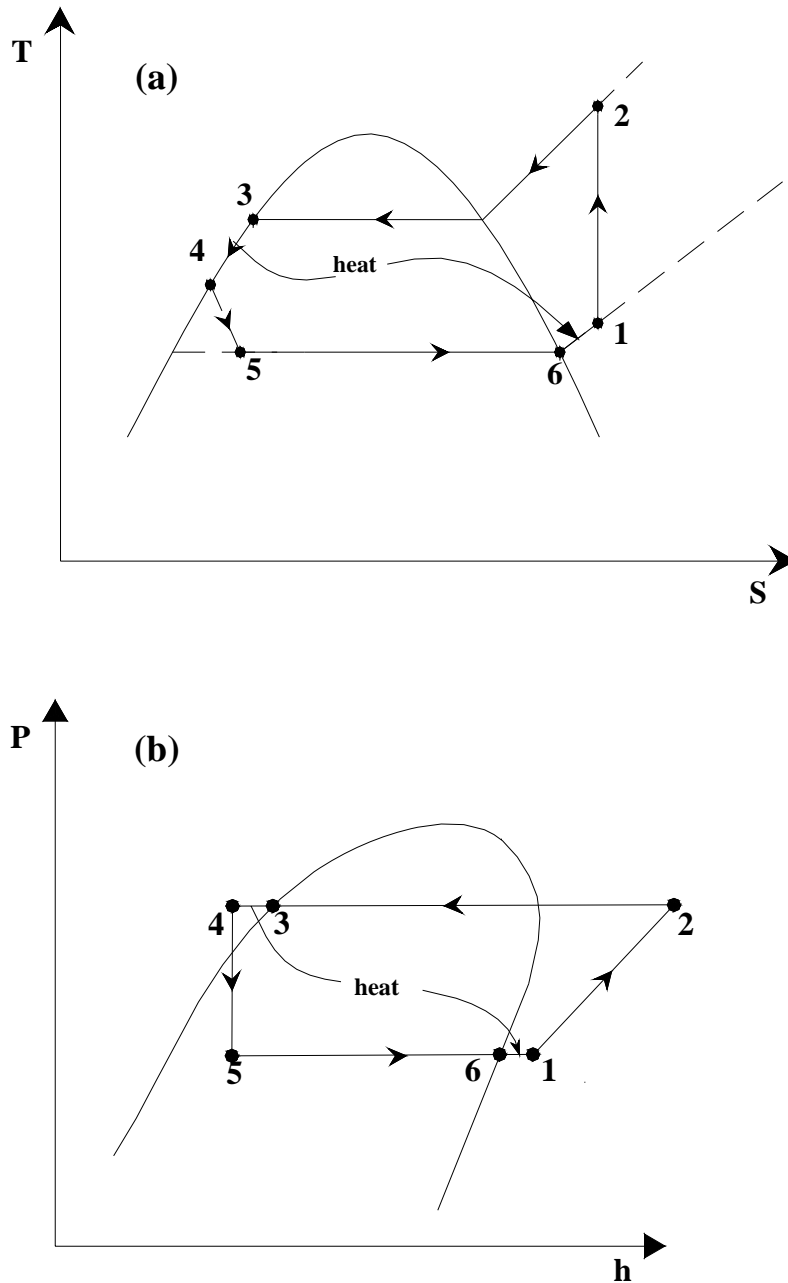


Fig.11.7: Single stage VCRES cycle with LSHX (a) on T-s diagram; (b) on P-h diagram

If we assume that there is no heat exchange between the surroundings and the LSHX and negligible kinetic and potential energy changes across the LSHX, then, the heat transferred between the refrigerant liquid and vapour in the LSHX, Q_{LSHX} is given by:

$$\begin{aligned}\dot{Q}_{\text{LSHX}} &= \dot{m}_r (h_3 - h_4) = \dot{m}_r (h_1 - h_6) \\ \Rightarrow (h_3 - h_4) &= (h_1 - h_6)\end{aligned}\tag{11.1}$$

if we take average values of specific heats for the vapour and liquid, then we can write the above equation as;

$$c_{p,l}(T_3 - T_4) = c_{p,v}(T_1 - T_6)\tag{11.2}$$

since the specific heat of liquid ($c_{p,l}$) is larger than that of vapour ($c_{p,v}$), i.e., $c_{p,l} > c_{p,v}$, we can write:

$$(T_3 - T_4) < (T_1 - T_6)\tag{11.3}$$

This means that, the degree of subcooling ($T_3 - T_4$) will always be less than the degree of superheating, ($T_1 - T_6$). If we define the effectiveness of the LSHX, ϵ_{LSHX} as the ratio of actual heat transfer rate in the LSHX to maximum possible heat transfer rate, then:

$$\epsilon_{\text{LSHX}} = \frac{Q_{\text{act}}}{Q_{\text{max}}} = \frac{\dot{m}_r c_{p,v}(T_1 - T_6)}{\dot{m}_r c_{p,v}(T_3 - T_6)} = \frac{(T_1 - T_6)}{(T_3 - T_6)}\tag{11.4}$$

The maximum possible heat transfer rate is equal to $Q_{\text{max}} = \dot{m}_r c_{p,v}(T_3 - T_6)$, because the vapour has a lower thermal capacity, hence only it can attain the maximum possible temperature difference, which is equal to $(T_3 - T_6)$. If we have a perfect LSHX with 100 percent effectiveness ($\epsilon_{\text{LSHX}} = 1.0$), then from the above discussion it is clear that the temperature of the refrigerant vapour at the exit of LSHX will be equal to the condensing temperature, T_c , i.e., ($T_1 = T_3 = T_c$). This gives rise to the possibility of an interesting cycle called as **Grindley cycle**, wherein the isentropic compression process can be replaced by an isothermal compression leading to improved COP. The Grindley cycle on T-s diagram is shown in Fig.11.8. Though theoretically the Grindley cycle offers higher COP, achieving isothermal compression with modern high-speed reciprocating and centrifugal compressors is difficult in practice. However, this may be possible with screw compressor where the lubricating oil provides large heat transfer rates.

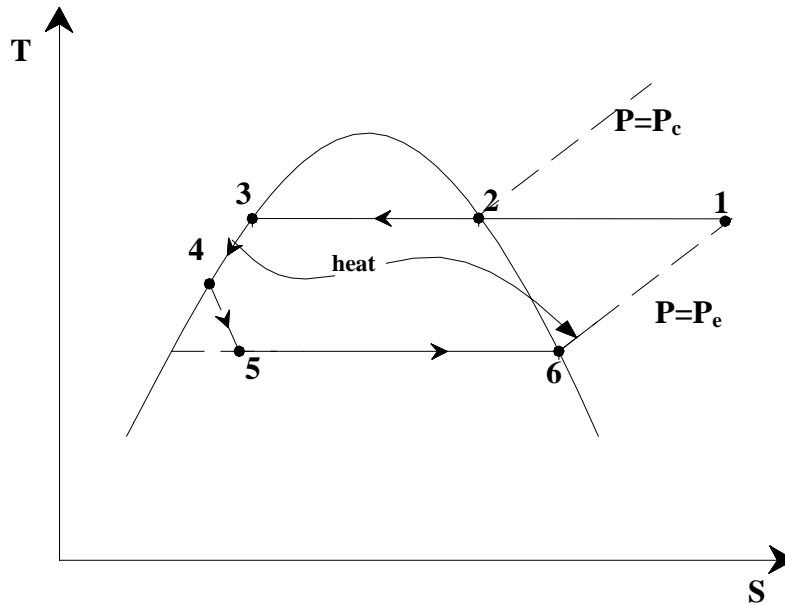


Fig.11.8: Grindley cycle on T-s coordinates (1-2 is isothermal compression)

11.3 Effect of superheat on system COP

As mentioned before, when the refrigerant is superheated usefully (either in the LSHX or the evaporator itself), the refrigeration effect increases. However, at the same time the work of compression also increases, primarily due to increase in specific volume of the refrigerant due to superheat. As a result, the volumic refrigeration effect and COP may increase or decrease with superheating depending on the relative increase in refrigeration effect and specific volume. It is observed that for some refrigerants the COP is maximum when the inlet to the compressor is inside the two-phase region and decreases as the suction condition moves into the superheated region. For other refrigerants the COP does not reach a maximum and increases monotonically with superheat. It was shown by Ewing and Gosney that a maximum COP occurs inside the two-phase region if the following criterion is satisfied:

$$\text{COP}_{\text{sat}} > \frac{T_e}{T_{2,\text{sat}} - T_e} \quad (11.5)$$

where COP_{sat} is the COP of the system with saturated suction condition, T_e is the evaporator temperature and $T_{2,\text{sat}}$ is the compressor discharge temperature when the vapour at suction condition is saturated (see Fig.11.9). For example, at an evaporator temperature of -15°C (258 K) and a condenser temperature of 30°C (303 K), the Table 11.1 shows that for refrigerants such as R11, R22, ammonia the maximum COP occurs inside the two-phase region and superheating reduces the COP and also volumic refrigeration effect, whereas for refrigerants such as R12, carbon dioxide and R502, no maxima exists and the COP and volumic refrigeration effect increase with superheat.

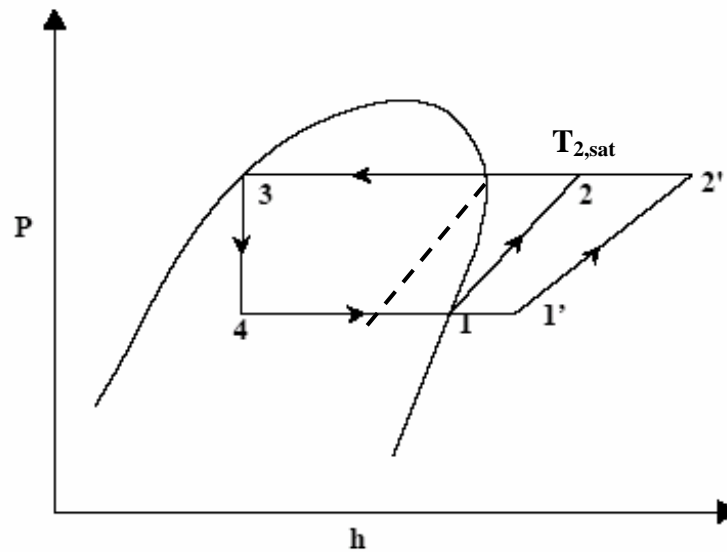


Fig.11.9: Ewing-Gosney criteria for optimum suction condition

Refrigerant	COP_{sat}	$T_{2,sat}$ (K)	$\frac{T_e}{T_{2,sat} - T_e}$	Maximum COP
Ammonia	4.77	372	2.26	Yes
CO ₂	2.72	341	3.11	No
R11	5.03	317	4.38	Yes
R12	4.70	311	4.87	No
R22	4.66	326	3.80	Yes
R502	4.35	310	4.96	No

Table 11.1. Existence of maximum COP, $T_e = 258$ K, $T_c = 303$ K (Gosney)

It should be noted that the above discussion holds under the assumption that the superheat is a useful superheat. Even though superheat appears to be not desirable for refrigerants such as ammonia, still a minimum amount of superheat is provided even for these refrigerants to prevent the entry of refrigerant liquid into the compressor. Also it is observed experimentally that some amount of superheat is good for the volumetric efficiency of the compressor, hence in practice almost all the systems operate with some superheat.

11.4 Actual VCRS systems

The cycles considered so far are internally reversible and no change of refrigerant state takes place in the connecting pipelines. However, in actual VCRS several irreversibilities exist. These are due to:

1. Pressure drops in evaporator, condenser and LSHX
2. Pressure drop across suction and discharge valves of the compressor
3. Heat transfer in compressor
4. Pressure drop and heat transfer in connecting pipe lines

Figure 11.10 shows the actual VCRS cycle on P-h and T-s diagrams indicating various irreversibilities. From performance point of view, the pressure drop in the evaporator, in the suction line and across the suction valve has a significant effect on system performance. This is due to the reason that as suction side pressure drop increases the specific volume at suction, compression ratio (hence volumetric efficiency) and discharge temperature increase. All these effects lead to reduction in system capacity, increase in power input and also affect the life of the compressor due to higher discharge temperature. Hence this pressure drop should be as small as possible for good performance. The pressure drop depends on the refrigerant velocity, length of refrigerant tubing and layout (bends, joints etc.). Pressure drop can be reduced by reducing refrigerant velocity (e.g. by increasing the inner diameter of the refrigerant tubes), however, this affects the heat transfer coefficient in evaporator. More importantly a certain minimum velocity is required to carry the lubricating oil back to the compressor for proper operation of the compressor.

Heat transfer in the suction line is detrimental as it reduces the density of refrigerant vapour and increases the discharge temperature of the compressor. Hence, the suction lines are normally insulated to minimize heat transfer.

In actual systems the compression process involves frictional effects and heat transfer. As a result, it cannot be reversible, adiabatic (eventhough it can be isentropic). In many cases cooling of the compressor is provided deliberately to maintain the maximum compressor temperature within safe limits. This is particularly true in case of refrigerants such as ammonia. Pressure drops across the valves of the compressor increase the work of compression and reduce the volumetric efficiency of the compressor. Hence they should be as small as possible.

Compared to the vapour lines, the system is less sensitive to pressure drop in the condenser and liquid lines. However, this also should be kept as low as possible. Heat transfer in the condenser connecting pipes is not detrimental in case of refrigeration systems. However, heat transfer in the subcooled liquid lines may affect the performance.

In addition to the above, actual systems are also different from the theoretical cycles due to the presence of foreign matter such as lubricating oil, water, air, particulate matter inside the system. The presence of lubricating oil cannot be avoided, however, the system design must ensure that the lubricating oil is carried over properly to the compressor. This depends on the miscibility of refrigerant-lubricating oil. Presence of other foreign materials such as air (non-condensing gas), moisture, particulate matter is detrimental to system performance. Hence systems are designed and operated such that the concentration of these materials is as low as possible.

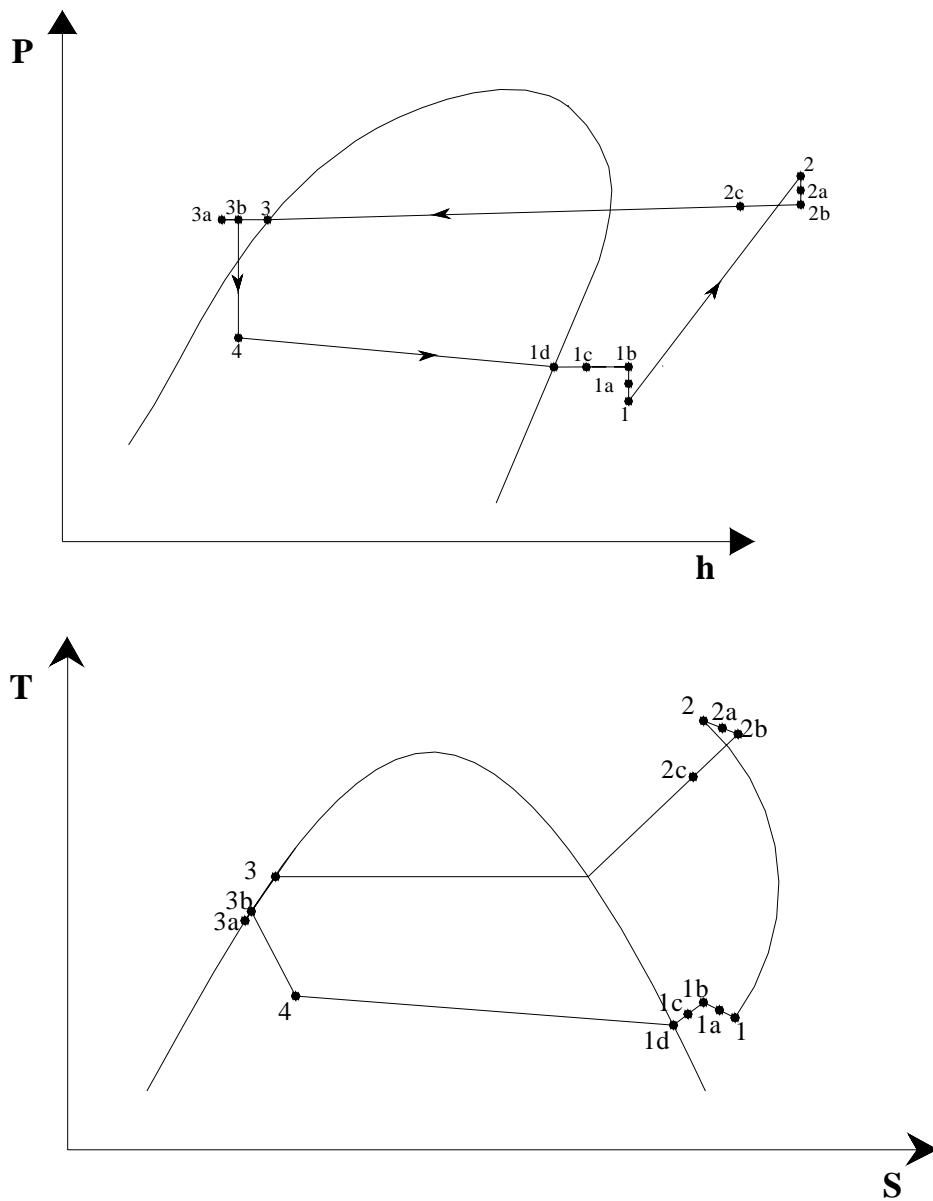


Fig.11.10: Actual VCRES cycle on P-h and T-s diagrams

Process	State
Pressure drop in evaporator	4-1d
Superheat of vapour in evaporator	1d-1c
Useless superheat in suction line	1c-1b
Suction line pressure drop	1b-1a
Pressure drop across suction valve	1a-1
Non-isentropic compression	1-2
Pressure drop across discharge valve	2-2a
Pressure drop in the delivery line	2a-2b
Desuperheating of vapour in delivery pipe	2b-2c
Pressure drop in the condenser	2b-3
Subcooling of liquid refrigerant	3-3a
Heat gain in liquid line	3a-3b

The COP of actual refrigeration systems is sometimes written in terms of the COP of Carnot refrigeration system operating between the condensing and evaporator temperatures ($\text{COP}_{\text{Carnot}}$), cycle efficiency (η_{cyc}), isentropic efficiency of the compressor (η_{is}) and efficiency of the electric motor (η_{motor}), as given by the equation shown below:

$$\text{COP}_{\text{act}} = \eta_{\text{cyc}} \eta_{\text{is}} \eta_{\text{motor}} \text{COP}_{\text{Carnot}} \quad (11.6)$$

An approximate expression for cycle efficiency (η_{cyc}) in the evaporator temperature range of -50°C to $+40^{\circ}\text{C}$ and condensing temperature range of $+10^{\circ}\text{C}$ to $+60^{\circ}\text{C}$ for refrigerants such as ammonia, R 12 and R 22 is suggested by Linge in 1966. This expression for a refrigeration cycle operating without ($\Delta T_{\text{sub}} = 0$) and with subcooling ($\Delta T_{\text{sub}} = T_{\text{c}} - T_{\text{r,exit}} > 0 \text{ K}$) are given in Eqns. (11.7) and (11.8), respectively:

$$\eta_{\text{cyc}} = \left(1 - \frac{T_{\text{c}} - T_{\text{e}}}{265} \right) \text{ without subcooling} \quad (11.7)$$

$$\eta_{\text{cyc}} = \left(1 - \frac{T_{\text{c}} - T_{\text{e}}}{265} \right) \left(1 + \frac{\Delta T_{\text{sub}}}{250} \right) \text{ with subcooling} \quad (11.8)$$

In the above equations T_{c} and T_{e} are condensing and evaporator temperatures, respectively.

The isentropic efficiency of the compressor (η_{is}) depends on several factors such as the compression ratio, design of the compressor, nature of the working fluid etc. However, in practice its value generally lies between 0.5 to 0.8. The motor efficiency (η_{motor}) depends on the size and motor load. Generally the motor efficiency is maximum at full load. At full load its value lies around 0.7 for small motors and about 0.95 for large motors.

11.5 Complete vapour compression refrigeration systems

In addition to the basic components, an actual vapour compression refrigeration consists of several accessories for safe and satisfactory functioning of the system. These include: compressor controls and safety devices such as overload protectors, high and low pressure cutouts, oil separators etc., temperature and flow controls, filters, driers, valves, sight glass etc. Modern refrigeration systems have automatic controls, which do not require continuous manual supervision.

Questions:

1. For the same condensing temperature and refrigeration capacity, a vapour compression refrigeration system operating at a lower evaporator temperature is more expensive than a system operating at a higher evaporator temperature, because at low evaporator temperature:

- a) Volumic refrigeration effect is high, hence the size of the compressor is large
- b) Volumic refrigeration effect is small, hence the size of the compressor is large
- c) Specific refrigeration effect is high, hence size of evaporator is large
- d) All the above

Ans.: b)

2. For a given condensing temperature, the volumic work of compression of a standard VCRS increases initially with evaporator temperature reaches a maximum and then starts decreasing, this is because as evaporator increases:

- a) Both specific volume of refrigerant and work of compression increase
- b) Specific volume of refrigerant increases and work of compression decreases
- c) Both specific volume and work of compression decrease
- d) Specific volume decreases and specific refrigeration effect increases

Ans.: c)

3. Subcooling is beneficial as it:

- a) Increases specific refrigeration effect
- b) Decreases work of compression
- c) Ensures liquid entry into expansion device
- d) All of the above

Ans.: a) and c)

4. Superheating:

- a) Always increases specific refrigeration effect
- b) Always decreases specific work of compression
- c) Always increases specific work of compression
- d) Always increases compressor discharge temperature

Ans.: c) and d)

5. Degree of superheating obtained using a LSHX is:

- a) Always greater than the degree of subcooling
- b) Always less than degree of subcooling
- c) Always equal to degree of subcooling
- d) Depends on the effectiveness of heat exchanger

Ans.: a)

6. Whether the maximum COP occurs when the suction condition is in two-phase region or not depends mainly on:

- a) Properties of the refrigerant
- b) Effectiveness of LSHX
- c) Operating temperatures
- d) All of the above

Ans.: a)

7. In actual VCRS, the system performance is affected mainly by:

- a) Pressure drop and heat transfer in suction line
- b) Pressure drop and heat transfer in discharge line
- c) Heat transfer in compressor
- d) All of the above

Ans.: a)

8. Pressure drop and heat transfer in suction line:

- a) Decrease compression ratio & discharge temperature
- b) Increase compression ratio & discharge temperature
- c) Decreases specific volume of refrigerant at suction
- d) Increases specific volume of refrigerant at suction

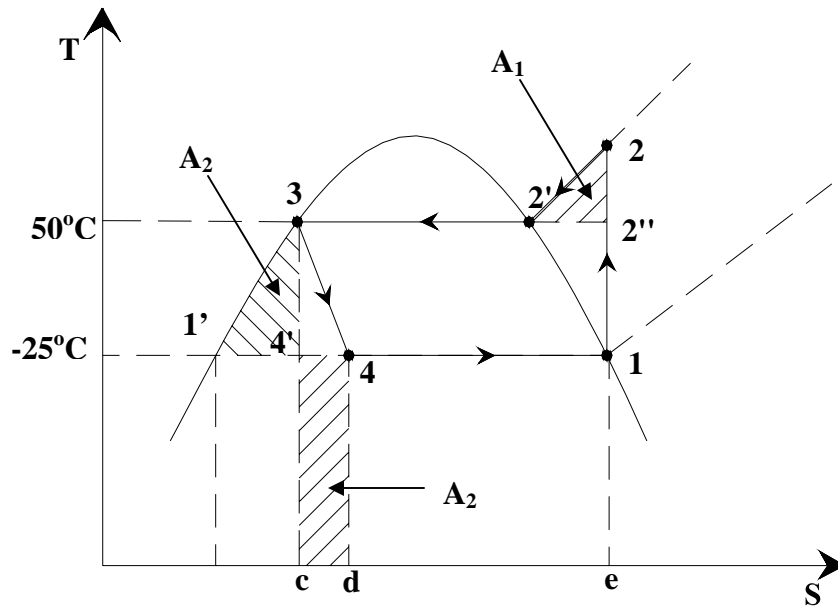
Ans.: b) and d)

9. A SSS vapour compression refrigeration system based on refrigerant R 134a operates between an evaporator temperature of -25°C and a condenser temperature of 50°C . Assuming isentropic compression, find:

- a) COP of the system
- b) Work input to compressor
- c) Area of superheat horn (additional work required due to superheat)

Throttling loss (additional work input due to throttling in place of isentropic expansion) assuming the isobar at condenser pressure to coincide with saturated liquid line.

Ans.: Given: Refrigerant : R 134a
 T_e = -25°C
 T_c = 50°C



Using refrigerant R134a property data, required properties at various state points are:

State Point	T (°C)	P (bar)	h (kJ/kg)	s (kJ/kg.K)	Quality
1	-25.0	1.064	383.4	1.746	1.0
2	60.7	13.18	436.2	1.746	Superheated
3	50.0	13.18	271.6	1.237	0.0
4	-25.0	1.064	271.6	1.295	0.4820
1'	-25.0	1.064	167.2	0.8746	0.0
2'	50.0	13.18	423.4	1.707	1.0
2''	50.0	10.2	430.5	1.746	Superheated
4'	-25.0	1.064	257.1	1.237	0.4158

a) $COP = (h_1 - h_4) / (h_2 - h_1) = 2.1174$

b) Work input to compressor, $W_c = (h_2 - h_1) = 52.8 \text{ kJ/kg}$

c) Superheat horn area, area A_1 :

$$\text{Area } A_1 = \text{Area under } 2-2' - \text{Area under } 2''-2'$$

$$\text{Area under } 2-2': \quad Tds = (dh - v dP) = dh = h_2 - h_2' \quad (dp = 0)$$

$$\Rightarrow \text{Area under } 2-2' = h_2 - h_2' = 12.8 \text{ kJ/kg}$$

$$\text{Area under } 2''-2' = Tds = T_c (s_2'' - s_2') = 12.6 \text{ kJ/kg}$$

$$\text{Superheat horn area} = \text{Area } A_1 = (12.8 - 12.6) = 0.2 \text{ kJ/kg}$$

d) Throttling loss, Area A_2 (assuming the saturated liquid line to coincide with isobar at condenser pressure):

$$\text{Area } A_2 = \text{Area under } 3-1' - \text{Area under } 4'-1' = (h_3 - h_1') - T_e (s_3 - s_1') \quad (s_3 = s_4')$$

$$\text{Throttling area} = (271.6 - 167.2) - 248.15(1.237 - 0.8746) = 14.47 \text{ kJ/kg}$$

Alternatively:

$$\text{Throttling area} = \text{Area under } 4-4' = T_e (s_4 - s_4') = 248.15(1.295 - 1.237) = 14.4 \text{ kJ/kg}$$

Check:

$$W_{\text{SSS}} = W_{\text{Carnot}} + \text{Area } A_1 + \text{Area } A_2$$

$$W_{\text{Carnot}} = (T_c - T_e)(s_1 - s_4') = 75(1.746 - 1.237) = 38.2 \text{ kJ/kg}$$

$$W_{\text{SSS}} = 38.2 + 14.4 + 0.2 = 52.8 \text{ kJ/kg}$$

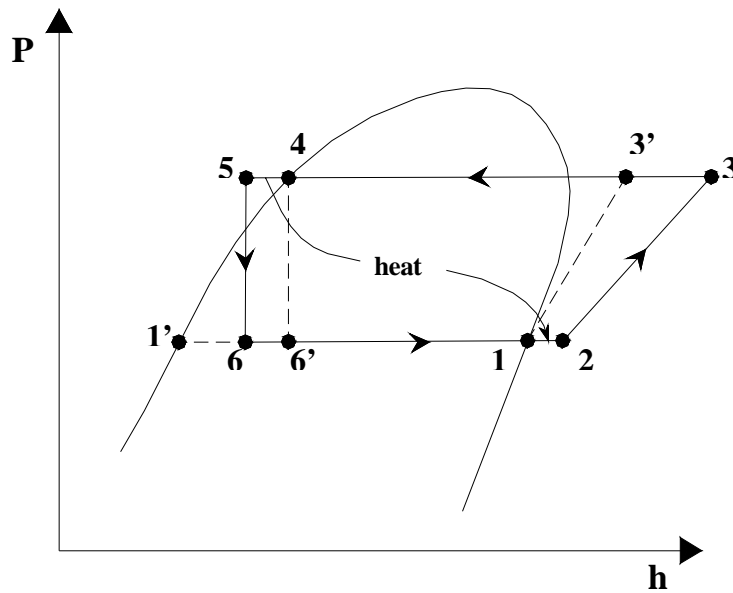
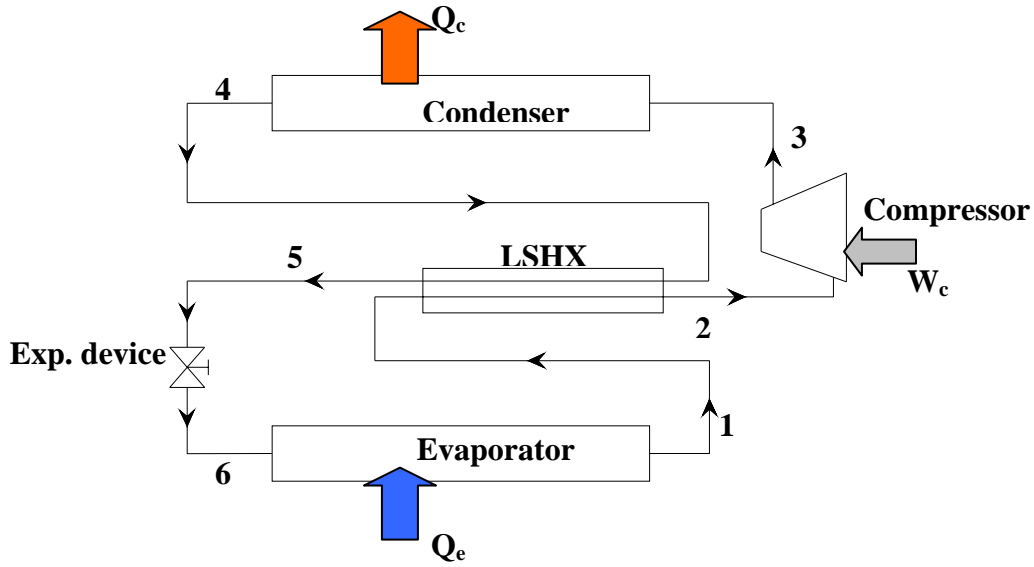
10. In a R22 based refrigeration system, a liquid-to-suction heat exchanger (LSHX) with an effectiveness of 0.65 is used. The evaporating and condensing temperatures are 7.2°C and 54.4°C respectively. Assuming the compression process to be isentropic, find:

- Specific refrigeration effect
- Volumic refrigeration effect
- Specific work of compression
- COP of the system
- Temperature of vapour at the exit of the compressor

Comment on the use of LSHX by comparing the performance of the system with a SSS cycle operating between the same evaporator and condensing temperatures.

Ans.:

Given:	Refrigerant	:	R 22
	T_e	=	7.2°C
	T_c	=	54.4°C
	Effectiveness of LSHX, ϵ_X	=	0.65



Effectiveness of LSHX, $\epsilon_X = (Q_{act}/Q_{max}) = [(mCp)_{min}\Delta T_{act,min}] / [(mCp)_{min}\Delta T_{max}]$
 $= (T_2 - T_1) / (T_4 - T_1)$; $C_{p,vapour} < C_{p,liquid}$

$(T_2 - T_1) / (T_4 - T_1) = 0.65 \Rightarrow T_2 = T_1 + 0.65(T_4 - T_1) = 37.88^\circ\text{C}$

From energy balance across LSHX:

$(h_2 - h_1) = (h_4 - h_5) \Rightarrow h_5 = h_4 - (h_2 - h_1)$

From the above data and using refrigerant property values for R 22 at various state points are:

State Point	T (°C)	P (bar)	h (kJ/kg)	s (kJ/kg.K)	v m ³ /kg	Quality
1	7.2	6.254	407.6	1.741	0.03773	1.0
2	<u>37.88</u>	6.254	430.7	1.819	0.04385	Superheated
3	<u>104.9</u>	21.46	466.8	1.819	-	Superheated
4	54.4	21.46	269.5	1.227	-	0.0
5	<u>37.65</u>	21.46	<u>246.4</u>	1.154	-	Subcooled
6	7.2	6.254	246.4	1.166	-	<u>0.1903</u>
6'	7.2	6.254	269.5	1.248	-	<u>0.3063</u>
3'	<u>74.23</u>	21.46	438.6	1.741	-	Superheated
1'	7.2	6.254	208.5	1.030	-	0.0

With LSHX:

- Refrigeration effect = $(h_1 - h_6) = 161.2 \text{ kJ/kg}$
- Volumic refrigeration effect = $(h_1 - h_6)/v_2 = 3676.2 \text{ kJ/m}^3$
- Work of compression = $(h_3 - h_2) = 36.1 \text{ kJ/kg}$
- COP = $(h_1 - h_6) / (h_3 - h_2) = 4.465$
- Temperature at compressor exit (from P_c and $s_3 = s_2$) = **104.9°C**

Without LSHX:

- Refrigeration effect = $(h_1 - h_{6'}) = 138.1 \text{ kJ/kg}$
- Volumic refrigeration effect = $(h_1 - h_{6'})/v_1 = 3660.2 \text{ kJ/m}^3$
- Work of compression = $(h_{3'} - h_1) = 31.0 \text{ kJ/kg}$
- COP = $(h_1 - h_{6'}) / (h_{3'} - h_1) = 4.455$
- Temperature at compressor exit (from P_c and $s_1 = s_{3'}$) = **74.23°C**

Parameter	With LSHX	Without LSHX
Refrigeration effect, kJ/kg	161.2	138.1
Ref. quality at evaporator inlet	0.1903	0.3063
Vol. Refrigeration effect, kJ/m³	3676.2	3660.2
Work of compression, kJ/kg	36.1	31.0
COP	4.465	4.455
Compressor exit temperature, °C	104.9	74.23

Comments:

- a) There is no appreciable change in COP with the addition of LSHX
- b) Quality of refrigerant at evaporator inlet is significantly lower with LSHX
- c) Discharge temperature is significantly high with LSHX
- d) For refrigerant R-22, use of LSHX does not improve the performance of the system significantly, however, the evaporator with LSHX performs better due to the lower vapour fraction at its inlet