

Absorption system

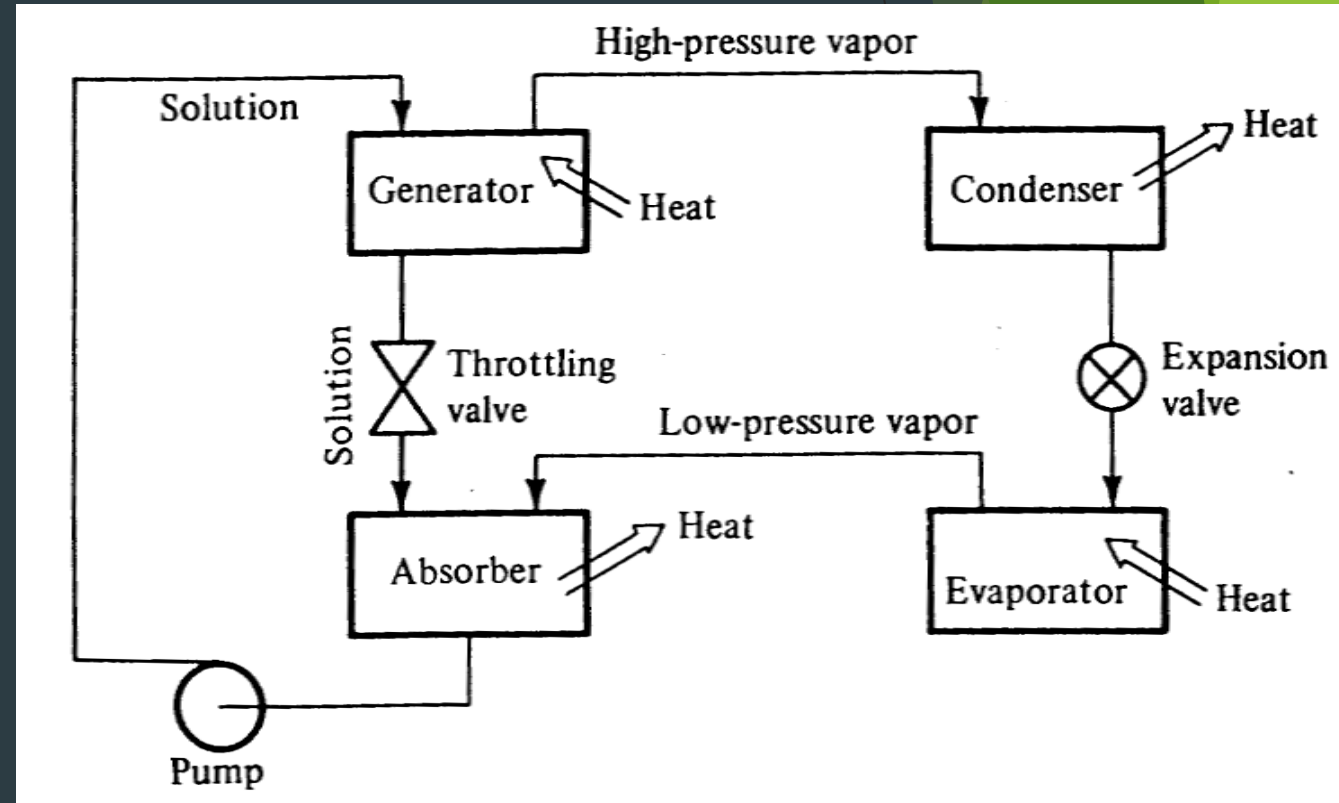
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Relation of the absorption to the vapor-compression cycle

- ▶ Vapor-compression cycle operate with the compressor condenser, expansion valve and evaporator.
- ▶ The vapor-compression system uses a compressor to delivered the vapor from low-pressure section (evaporator) to high-pressure section (condenser).
- ▶ The absorption system replaced compressor by:
 1. Absorb vapor in liquid while removing heat.
 2. Elevate pressure of liquid with pump.
 3. Release vapor by applying heat

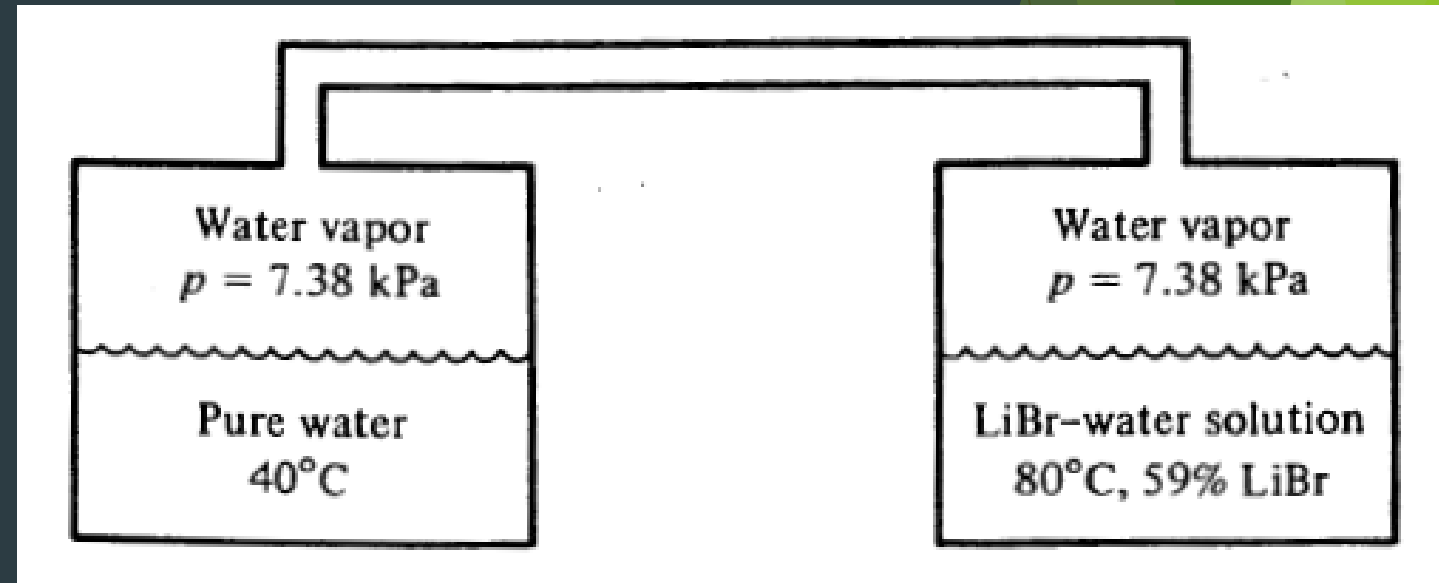
Absorption cycle

- 1- Absorption cycle consist from four main part two in low pressure section (evaporator and absorber) and two in high pressure section (generator and condenser).
- 2- Low-pressure vapor from the evaporator is absorbed by the liquid solution in absorber.
- 3- The absorber is cooled by water or air to reject heat to the atmosphere.
- 4- The pump delivers the solution from the absorber to the generator.
- 5- In the generator, the vapor drives off from the solution by using a high-temperature heat source and the liquid solution returns to the absorber through a throttling valve.
- 6- The vapor is condensed by cooling in the condenser and throttling to the evaporator by expansion valve.
- 7- Condensed liquid is evaporated in the evaporator by adding heat



Temperature-pressure-concentration properties of LiBr-water solutions

- 1- Lithium bromide is a solid salt crystal.
- 2- It has a high ability to absorb the water vapor and become a liquid solution.
- 3- The saturated solution temperature and pressure is a function of the solution concentration.



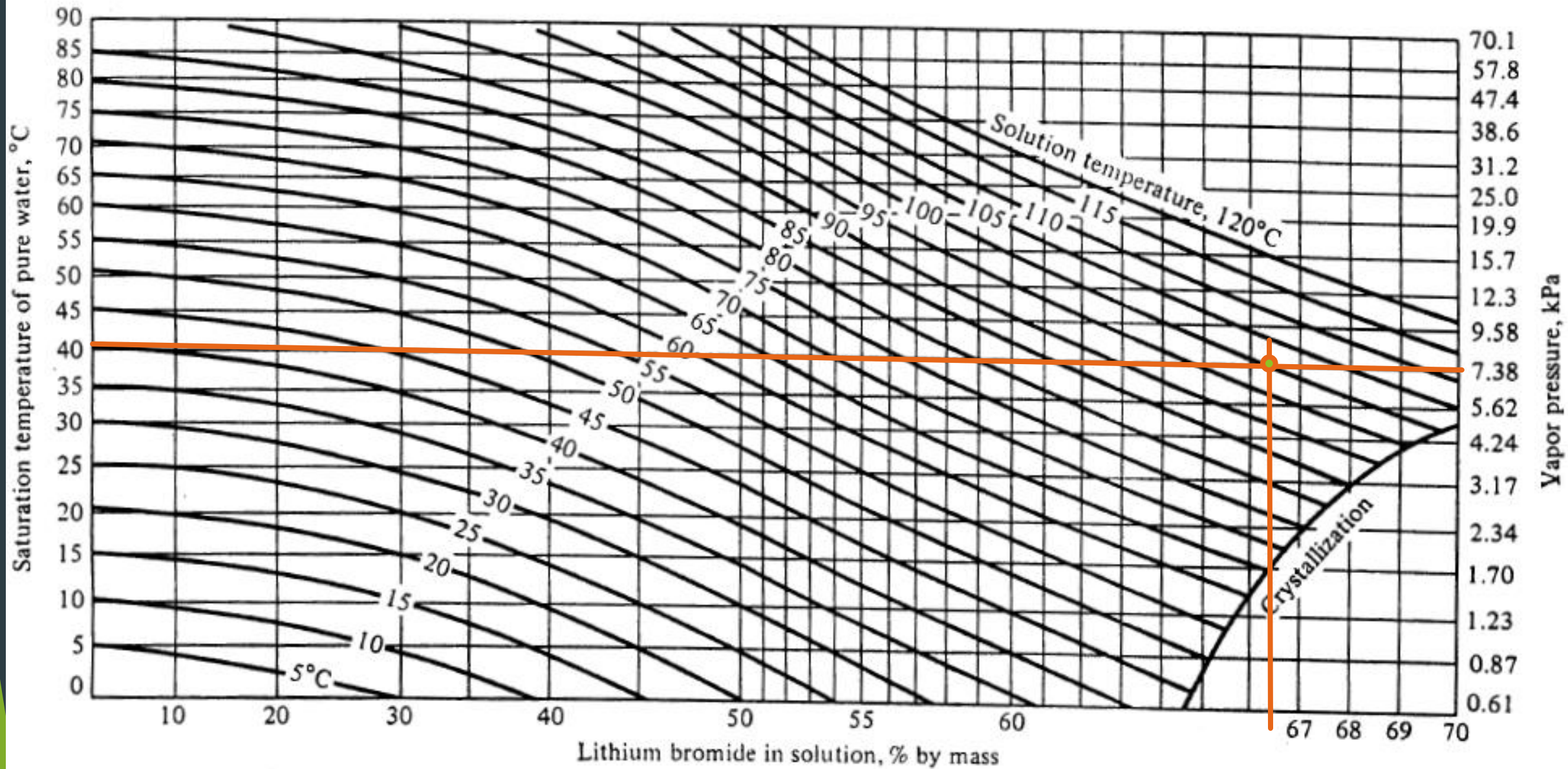


Figure 17-5 Temperature-pressure-concentration diagram of saturated LiBr-water solutions, developed from data in Ref. 1.

Calculation of mass flow rates in the absorption cycle

From a sketch p-x-t diagram:

$$P_{\text{condenser}} = P_{\text{generator}} = 7.38 \text{ kPa}$$

(from saturated temp. of condensor 40 °C)

$$P_{\text{evaporator}} = P_{\text{absorber}} = 1.23 \text{ kPa}$$

(from saturated temp. of evaporator 10 °C)

$$x_2 = 66.4 \% \text{ (by intersection the } P_{\text{cond.}} \text{ \& } T_{\text{generator}})$$

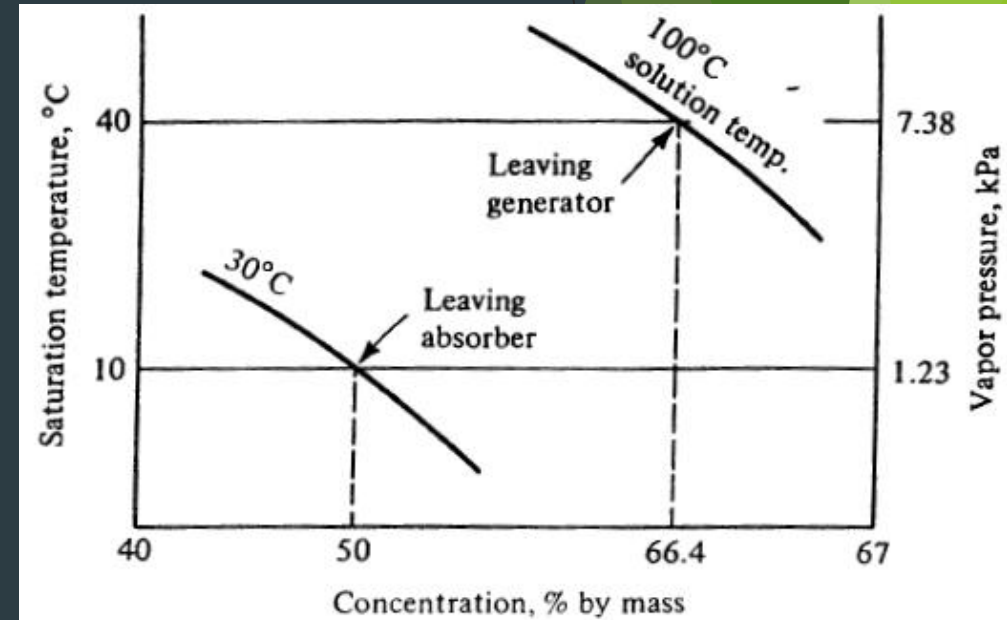
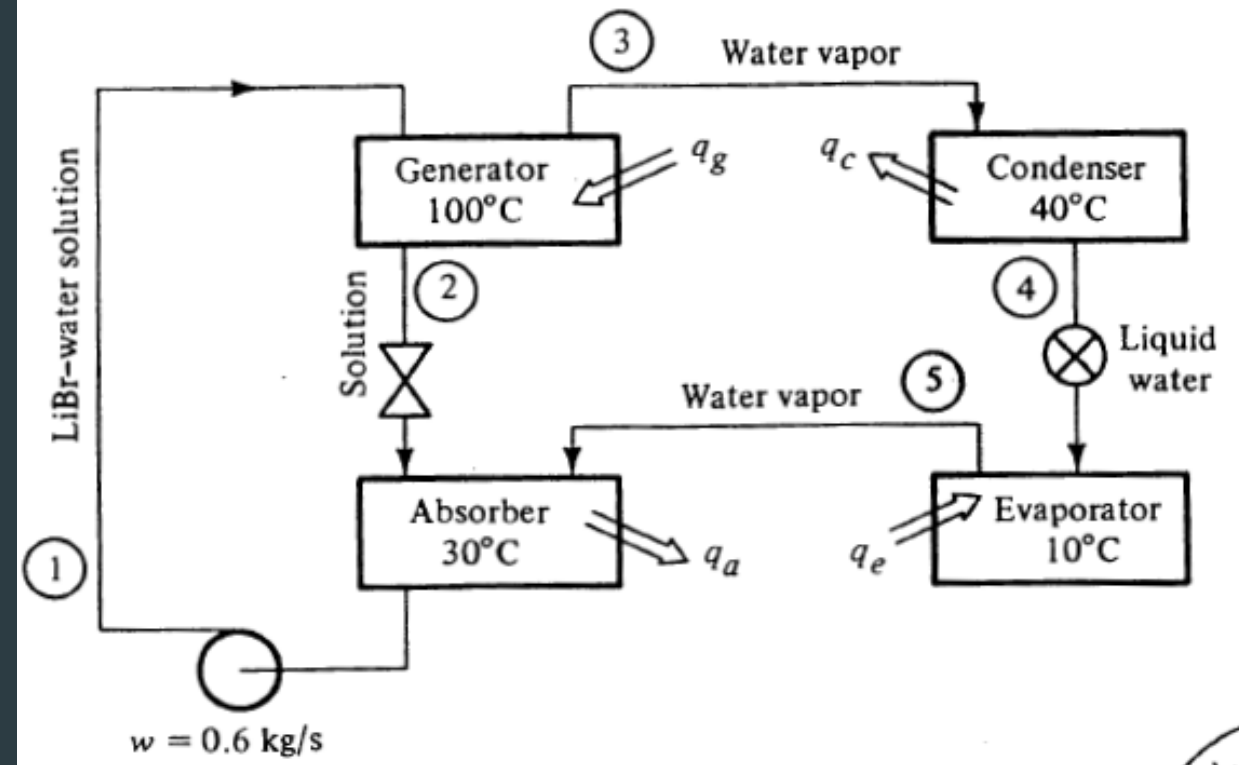
$$x_1 = 50 \% \text{ (by intersection the } P_{\text{evap.}} \text{ \& } T_{\text{absorber}})$$

$$w_2 + w_3 = w_1 = 0.6 \text{ (mass flow balance)}$$

$$w_1 \cdot x_1 = w_2 \cdot x_2 \text{ (LiBr balance)}$$

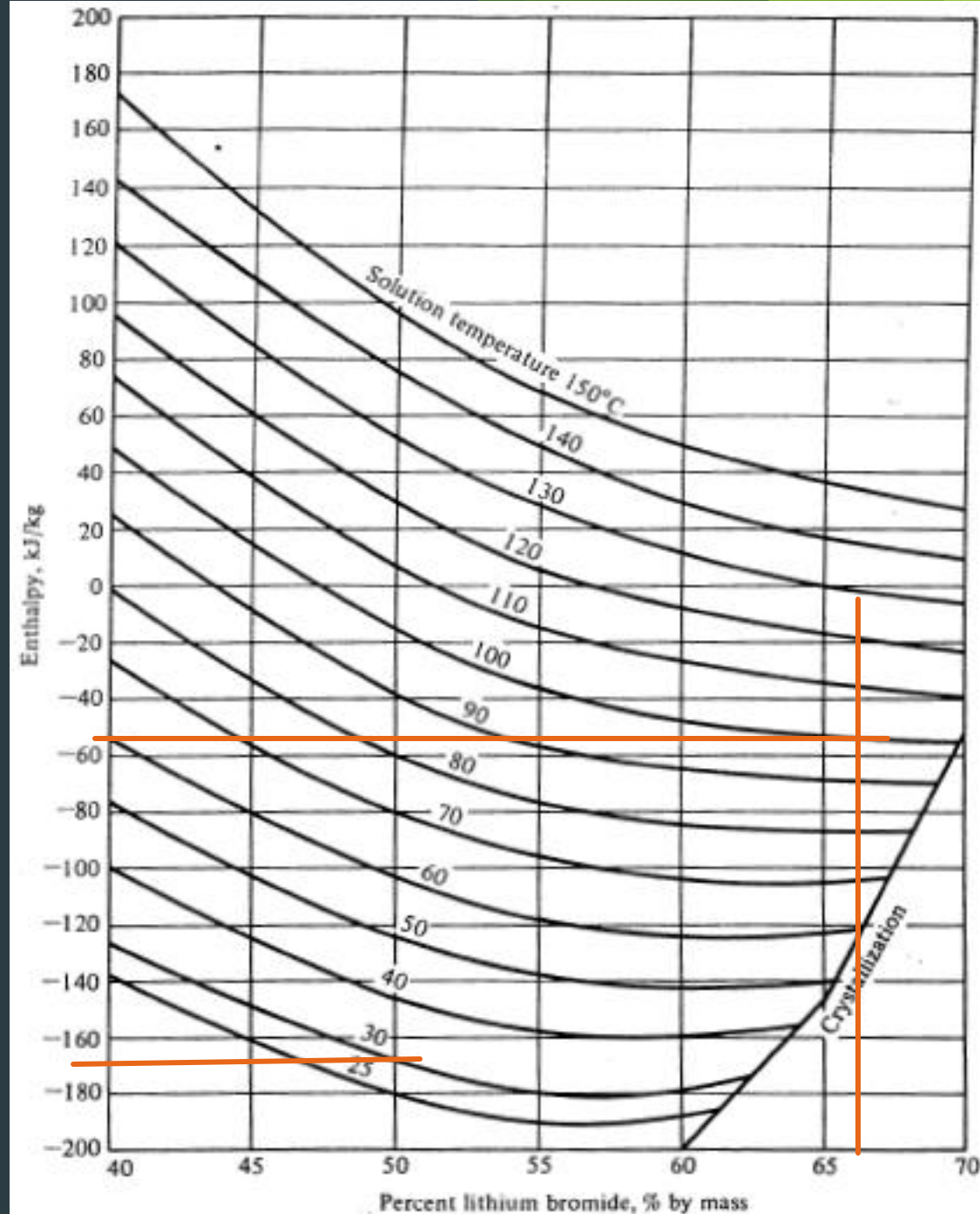
$$0.6 \times 0.5 = w_2 \times 0.664 \implies w_2 = 0.452 \text{ kg/s}$$

$$w_3 = 0.148 \text{ kg/s}$$



Enthalpy of LiBr solutions

- 1- The enthalpy of water in liquid or vapor flows in and out of the condenser and evaporator points can be determined from a table of water properties.
- 2- In the generator and absorber, LiBr-water solutions the enthalpy is a function of both the temperature and concentration by using figure 17-8 we can find the enthalpy data for LiBr-water solutions.



Thermal analysis of simple absorption system

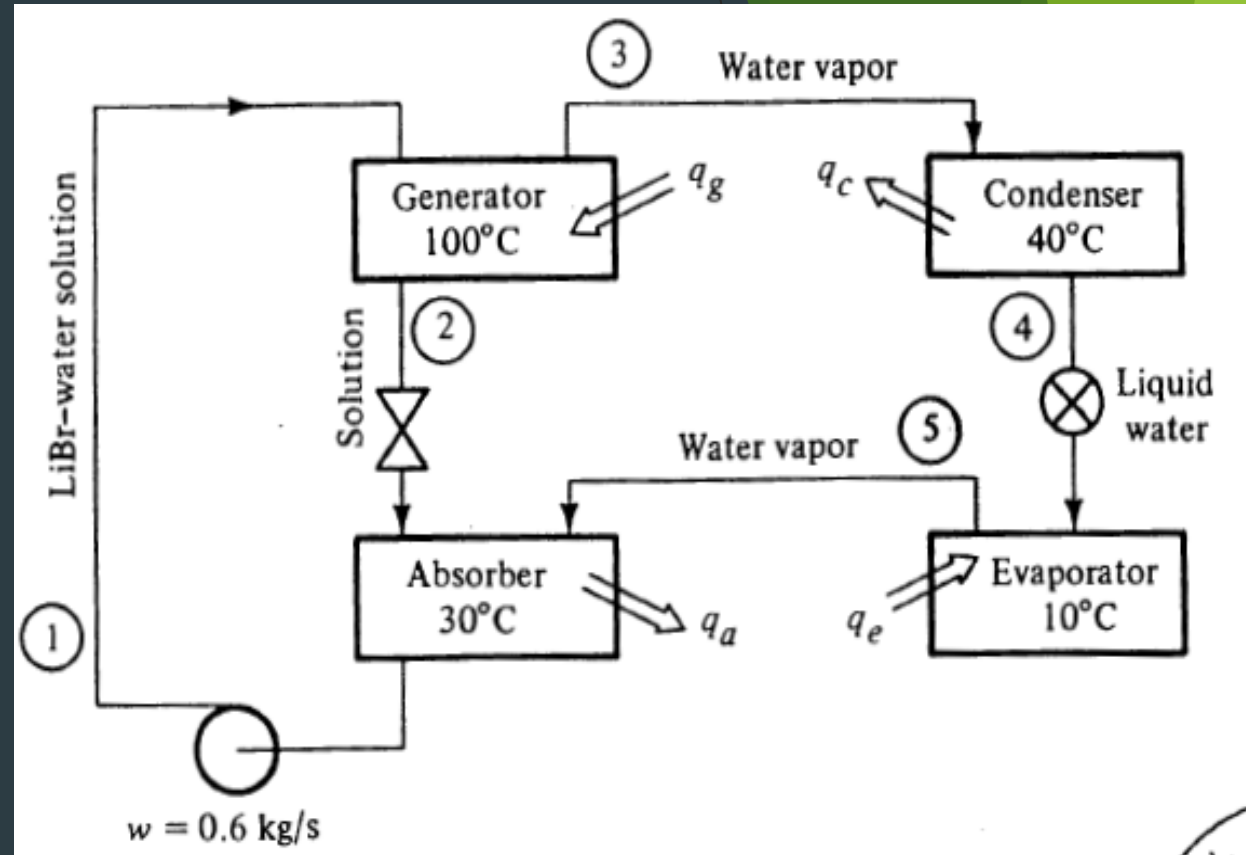
Example 17-3 For the absorption system of Example 17-2 shown in Fig. compute q_g , q_a , q_c , q_e , and the COP.

- ▶ $h_1 = h$ at 30°C & x of 50% = -168 kJ/kg
- ▶ $h_2 = h$ at 100°C & x of 66.4% = -52 kJ/kg
- ▶ The enthalpies of water liquid and vapor are found from Table A-1:

$$h_3 = h \text{ of saturated vapor at } 100^\circ\text{C} = 2676.0 \text{ kJ/kg}$$

$$h_4 = h \text{ of saturated liquid at } 40^\circ\text{C} = 167.5 \text{ kJ/kg}$$

$$h_5 = h \text{ of saturated vapor at } 10^\circ\text{C} = 2520.0 \text{ kJ/kg}$$



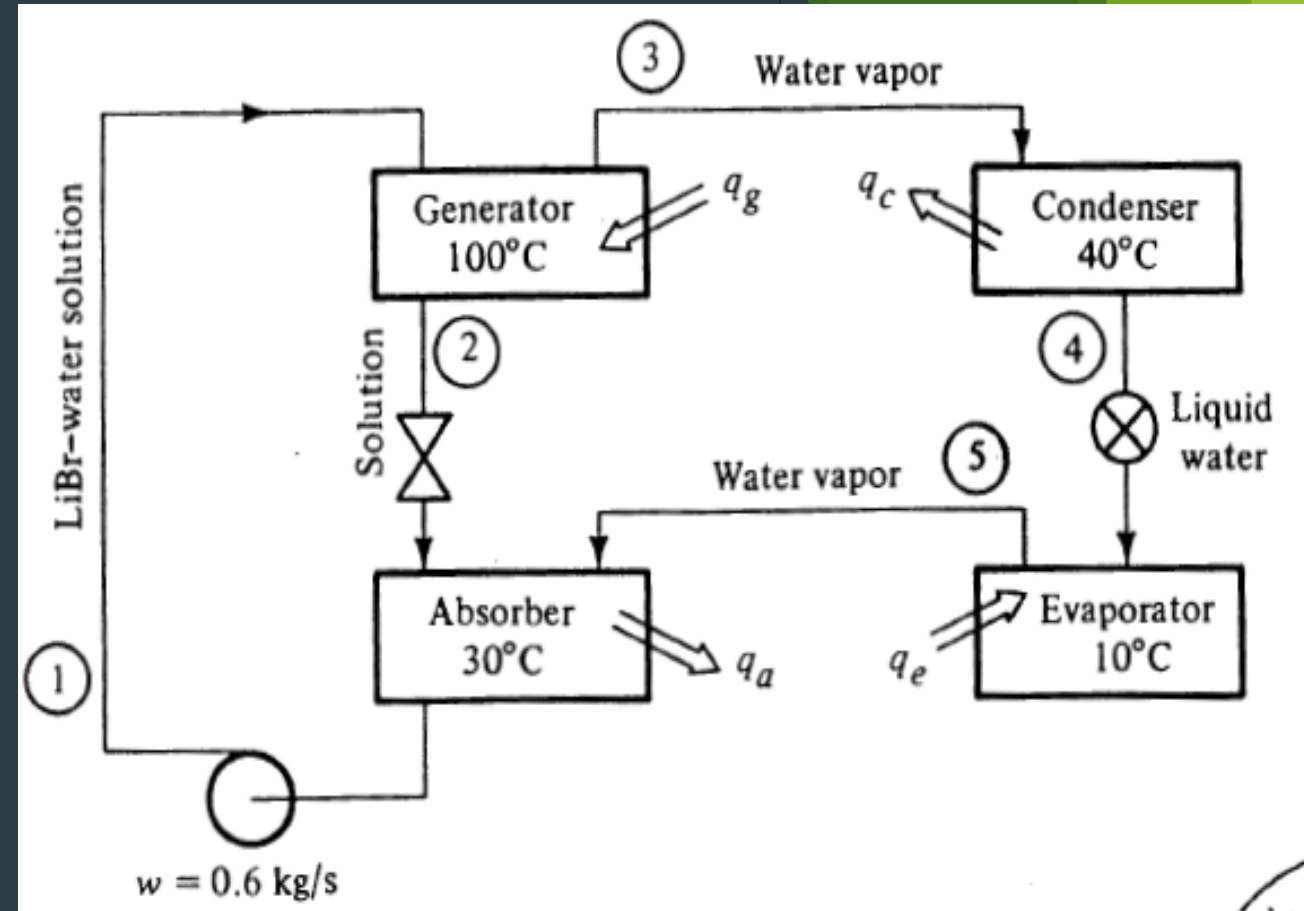
$$\begin{aligned}
 q_g &= w_3 h_3 + w_2 h_2 - w_1 h_1 \\
 &= 0.148(2676) + 0.452(-52) - 0.6(-168) \\
 &= 476.6 \text{ kW}
 \end{aligned}$$

$$\begin{aligned}
 q_c &= w_3 h_3 - w_4 h_4 \\
 &= 0.148(2676 - 167.5) = 371.2 \text{ kW}
 \end{aligned}$$

$$\begin{aligned}
 q_e &= w_5 h_5 - w_4 h_4 = 0.148(2520 - 167.5) \\
 &= 348.2 \text{ kW}
 \end{aligned}$$

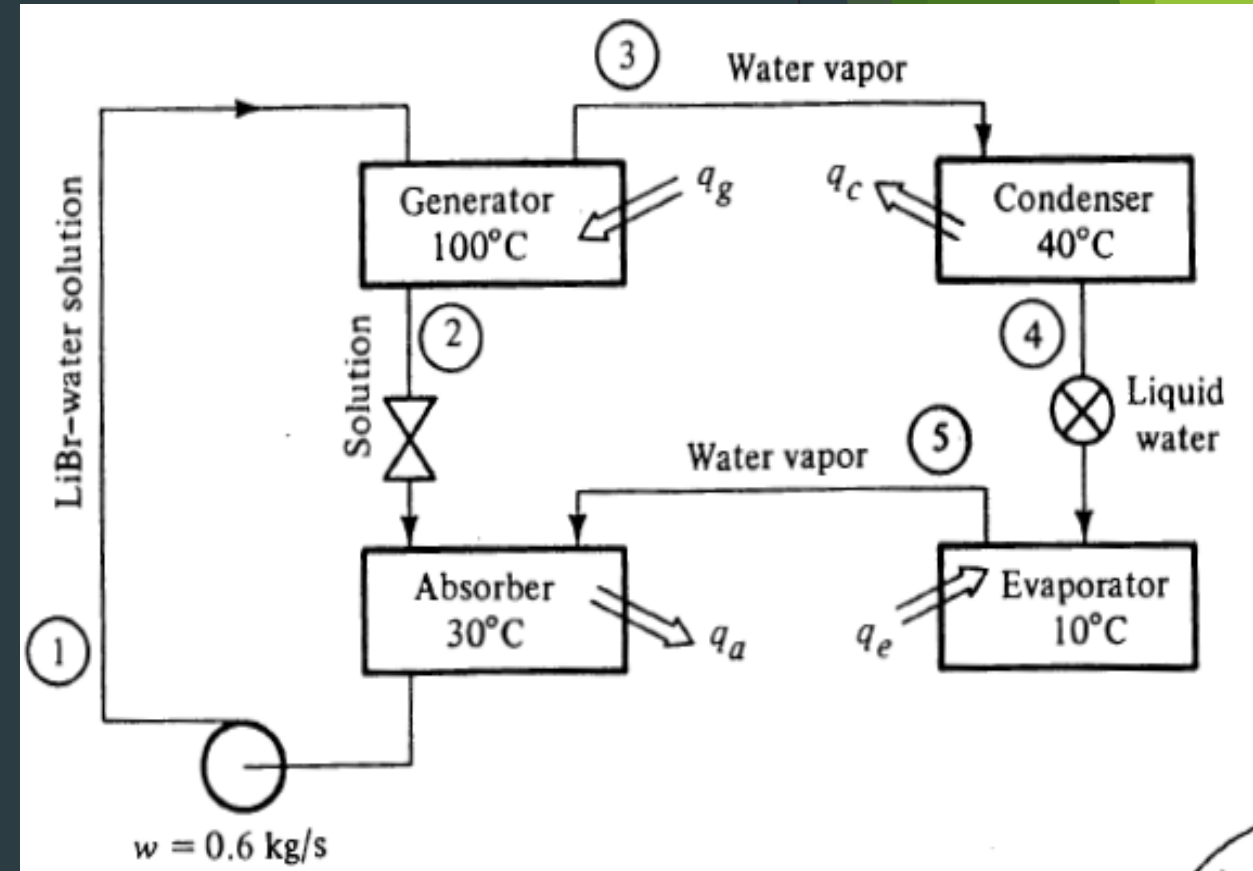
$$\begin{aligned}
 q_a &= w_2 h_2 + w_5 h_5 - w_1 h_1 \\
 &= 0.452(-52) + 0.148(2520) - 0.6(-168) \\
 &= 450.3 \text{ kW}
 \end{aligned}$$

$$COP_{abs} = \frac{q_e}{q_g} = \frac{348.2}{476.6} = 0.736$$



Absorption cycle with heat exchanger

- ▶ The solution at point 1 leaves the absorber at a temperature of 30°C and must be heated to 100°C in the generator.
- ▶ Similarly the solution at point 2 leaves the generator at 100°C and must be cooled to 30°C in the absorber.
- ▶ One of the major operating costs of the system is the heat added in the generator q_g .
- ▶ Therefore by addition to the simple cycle is a heat exchanger as shown in Fig. 17-9 to transfer heat between the two streams of solutions.
- ▶ This heat exchanger heats the solution going to the generator and cools the solution returning to the absorber.



Absorption cycle with heat exchanger

The simple cycle shown in Fig. is modified by the insertion of a heat exchanger, such that. What are the rates of energy transfer at each of the components and the COP_{abs} of cycle?

$$w_1 = w_2 = 0.6 \text{ kg/s}$$

$$w_3 = w_4 = 0.452 \text{ kg/s}$$

$$w_5 = w_6 = w_7 = 0.148 \text{ kg/s}$$

Enthalpies that remain unchanged are:

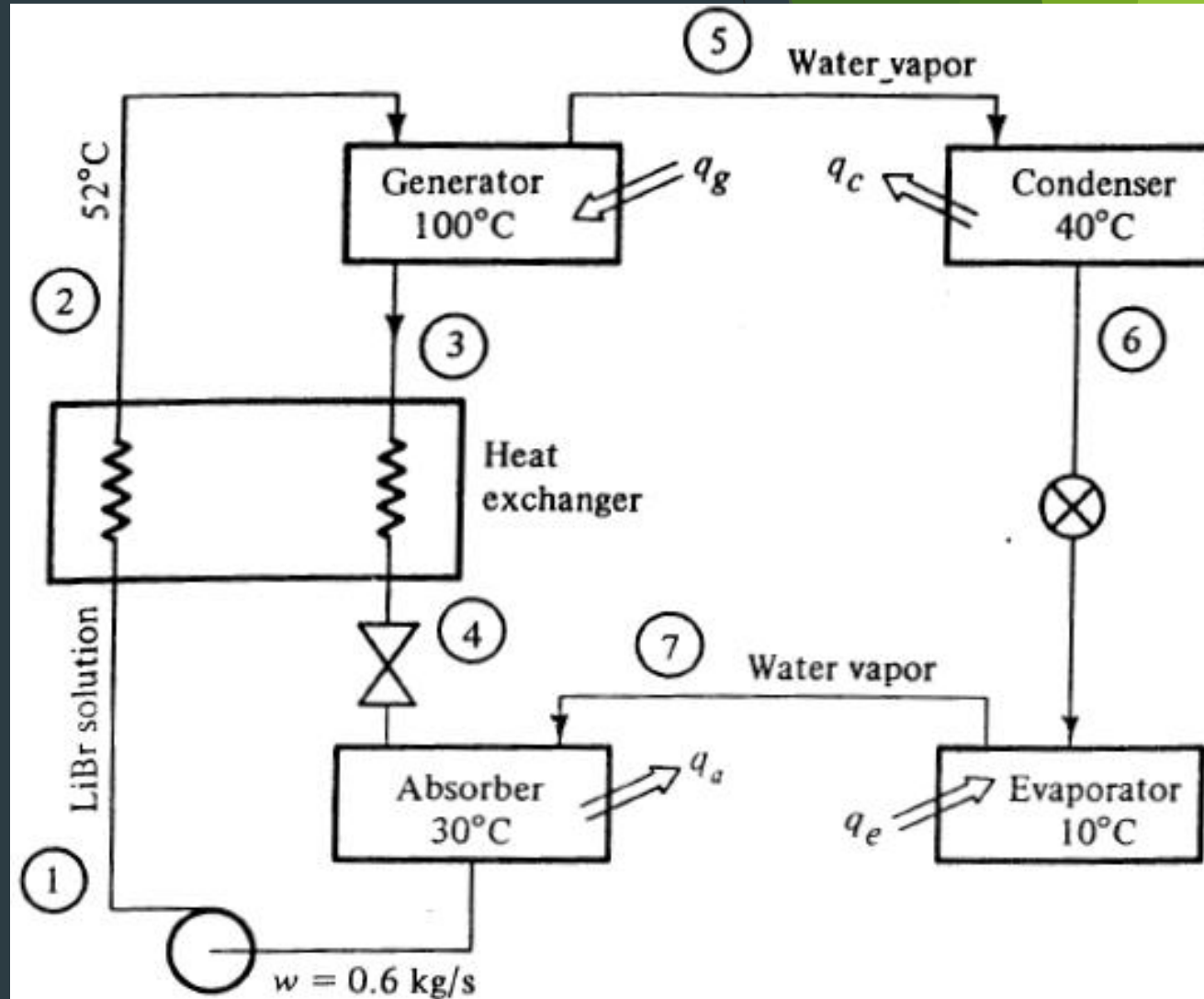
$$h_1 = -168 \text{ kJ/kg}, h_5 = 2676.0 \text{ kJ/kg}$$

$$h_3 = -52 \text{ kJ/kg}, h_6 = 167.5 \text{ kJ/kg}$$

$$h_7 = 2520.0 \text{ kJ/kg}$$

The (q_c & q_e) remain unchanged

$$q_c = 371.2 \text{ kW and } q_e = 348.2 \text{ kW}$$



$h_2 = h$ at 52°C and x of 50% = -120 kJ/kg

The rate of heat absorbed by the solution in heat exchanger:

$$q_{\text{hx}} = w_1 (h_2 - h_1)$$

$$= 0.6[-120 - (-168)] = 28.8 \text{ kW}$$

Also: $q_{\text{hx}} = w_3 (h_3 - h_4)$

$$28.8 = 0.452 (-52 - h_4) \rightarrow h_4 = -116 \text{ kJ/kg}$$

$$q_g = w_5 h_5 + w_3 h_3 - w_2 h_2$$

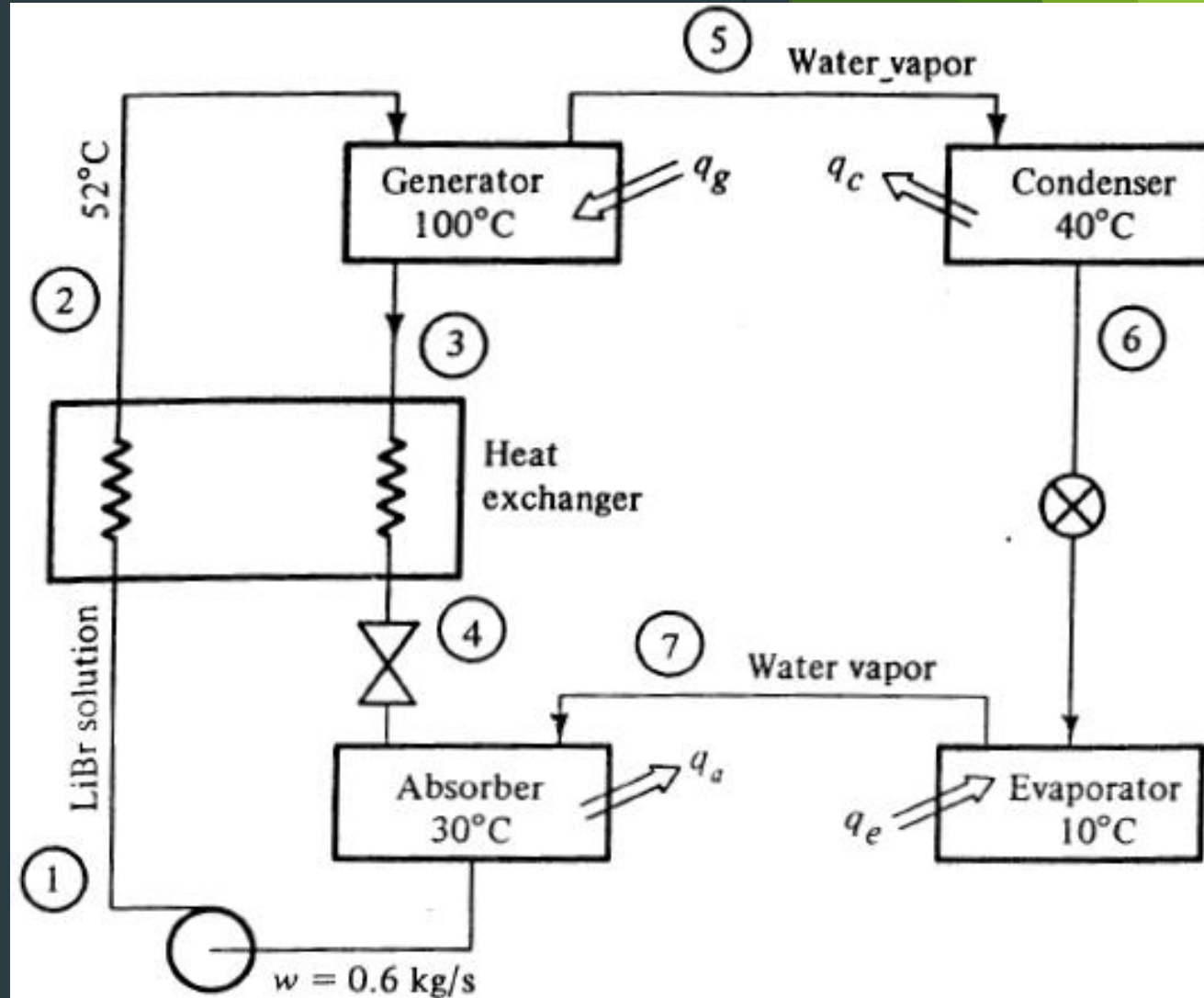
$$= 0.148(2676) + 0.452(-52) - 0.6(-120)$$

$$= 444.5 \text{ kW}$$

$$q_a = w_7 h_7 + w_4 h_4 - w_1 h_1$$

$$= 0.148(2520) + 0.452(-116) - 0.6(-168)$$

$$= 421.3 \text{ kW}$$

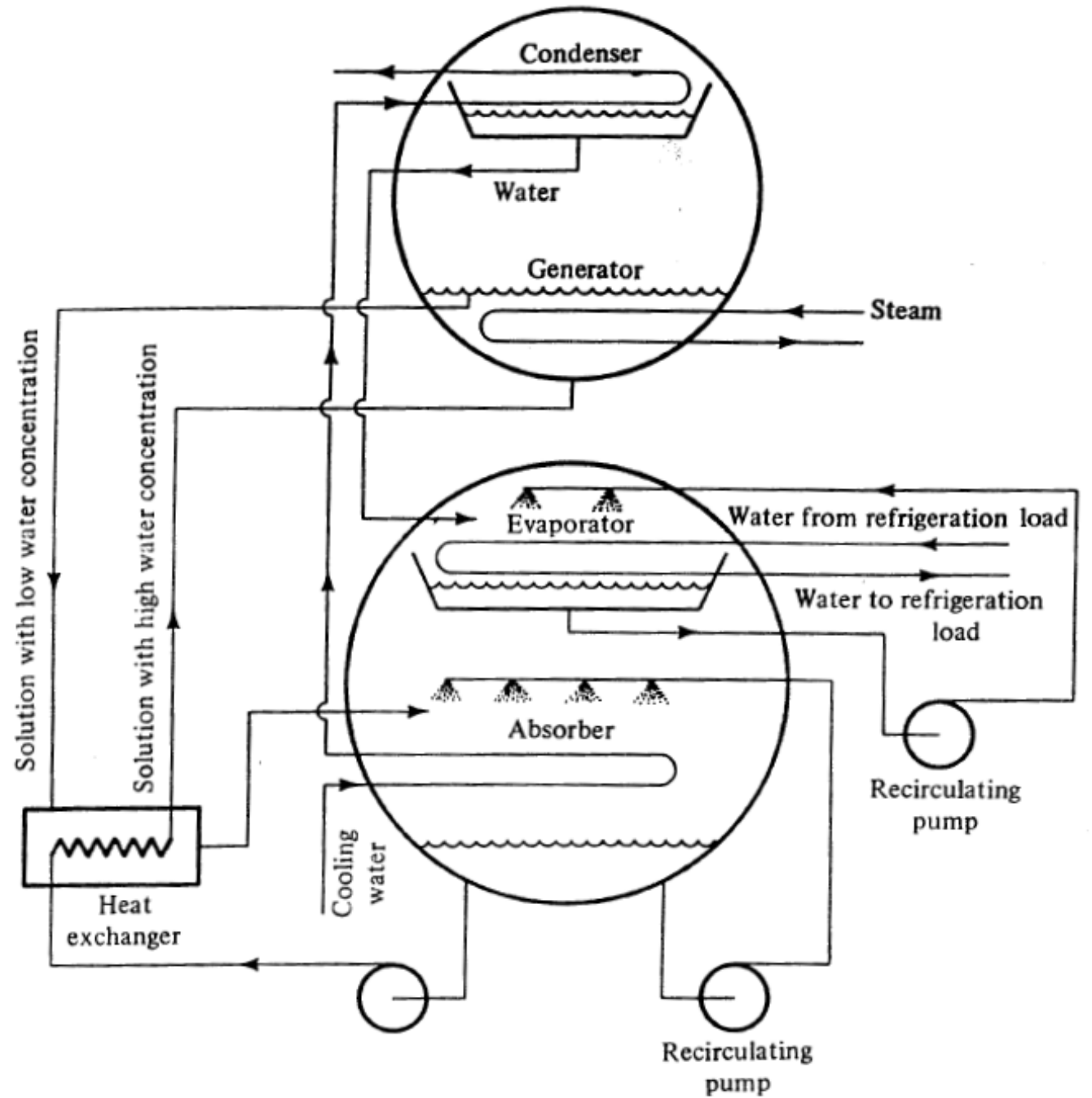


The coefficient of performance of the system that incorporates the heat exchanger is

$$COP_{abs} = \frac{q_e}{q_g} = \frac{348.2}{444.5} = 0.783$$

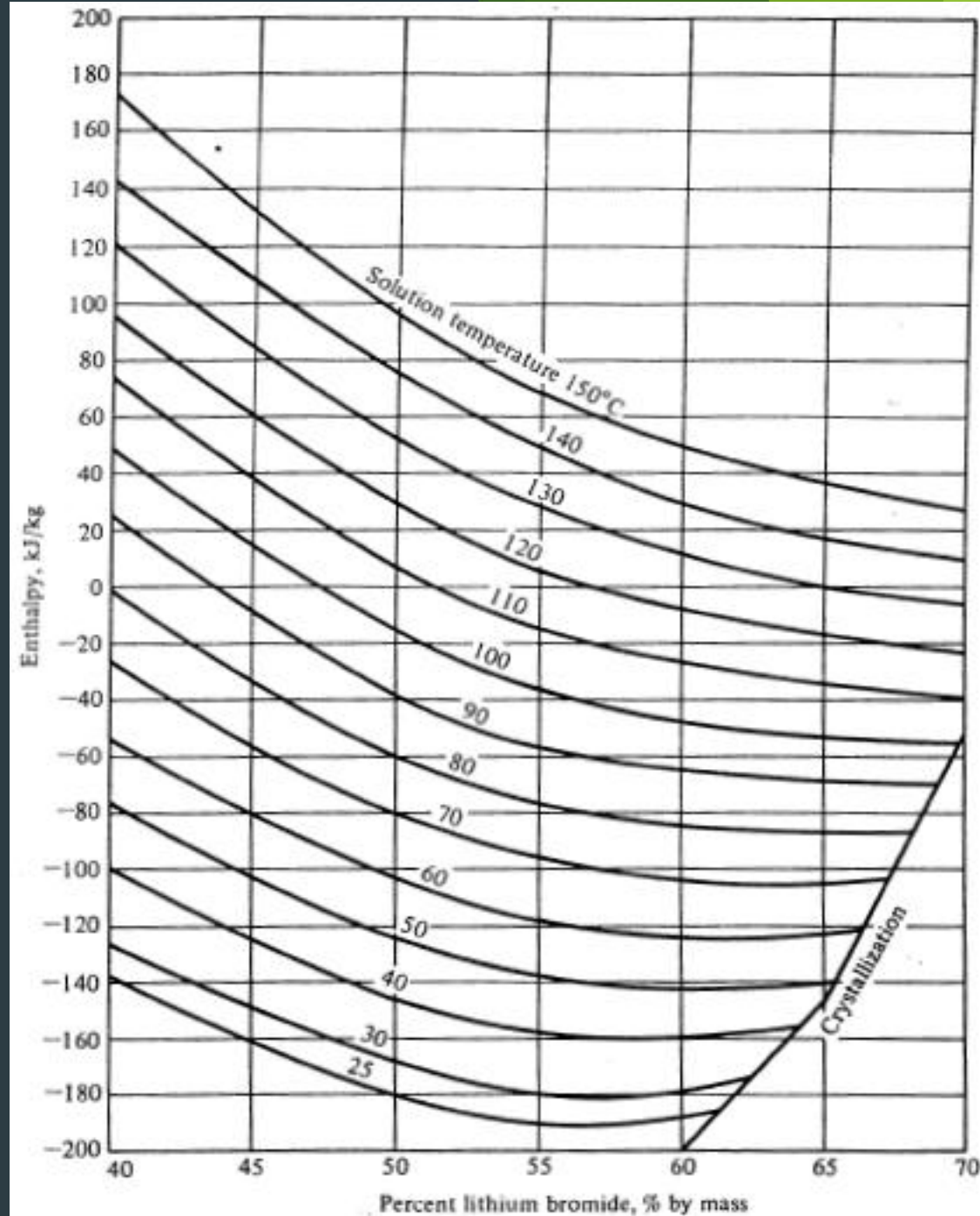
This COP is an improvement over the value of 0.736 applicable to the simple system without a heat exchanger.

commercial absorption units



Crystallization

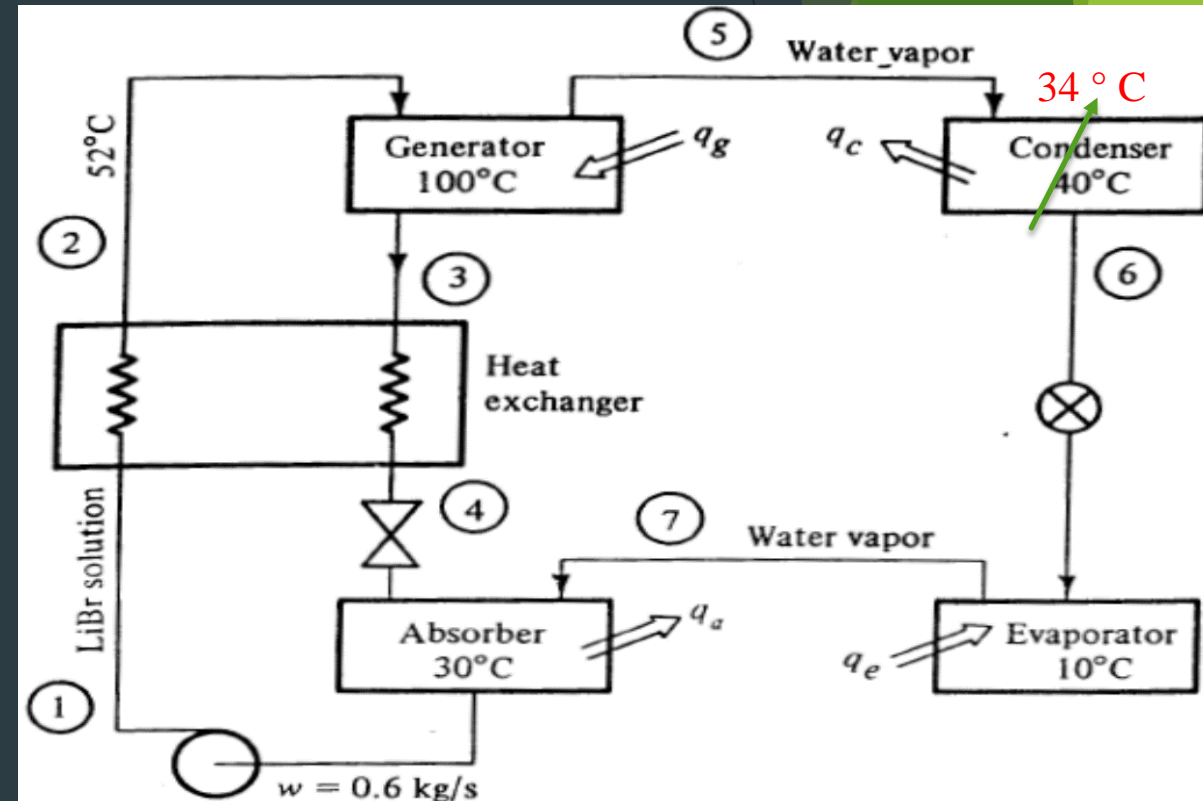
- ▶ crystallization lines appear in the lower right section on the two property charts for LiBr-water solutions those lines indicates a solidification of the LiBr.
- ▶ Dropping into the crystallization region lead to the formation of a slush, which can block the flow in a pipe and cut on the operation of the absorption unit.



Example: In the system shown in Fig. 17-9, the ambient wet-bulb temperature decreases so that the temperature of cooling water drops, which also reduces the condensing temperature to 34°C. All other temperatures specified on Fig. 17-9 remain unchanged. Is there a danger of crystallization?

Solution: The reduction in condensing temperature drops the high-side pressure so that the concentration of solution leaving the generator at point 3 is 69%. The new mass rates of flow apply at points 3 and 5

- ▶ $w_3 \cdot x_3 = w_2 \cdot x_2$
- ▶ $w_3 = 0.6 \times 0.50 / 0.69 = 0.435 \text{ kg/s}$
- ▶ $w_5 = w_2 - w_3 = 0.6 - 0.435 = 0.165 \text{ kg/s}$
- ▶ $h_1 = h$ at 30°C & x of 50% = -168 kJ/kg
- ▶ $h_2 = h$ at 52°C & x of 50% = -120 kJ/kg
- ▶ $h_3 = h$ at 100°C & x of 69% = -54 kJ/kg



An energy balance about the heat exchanger:

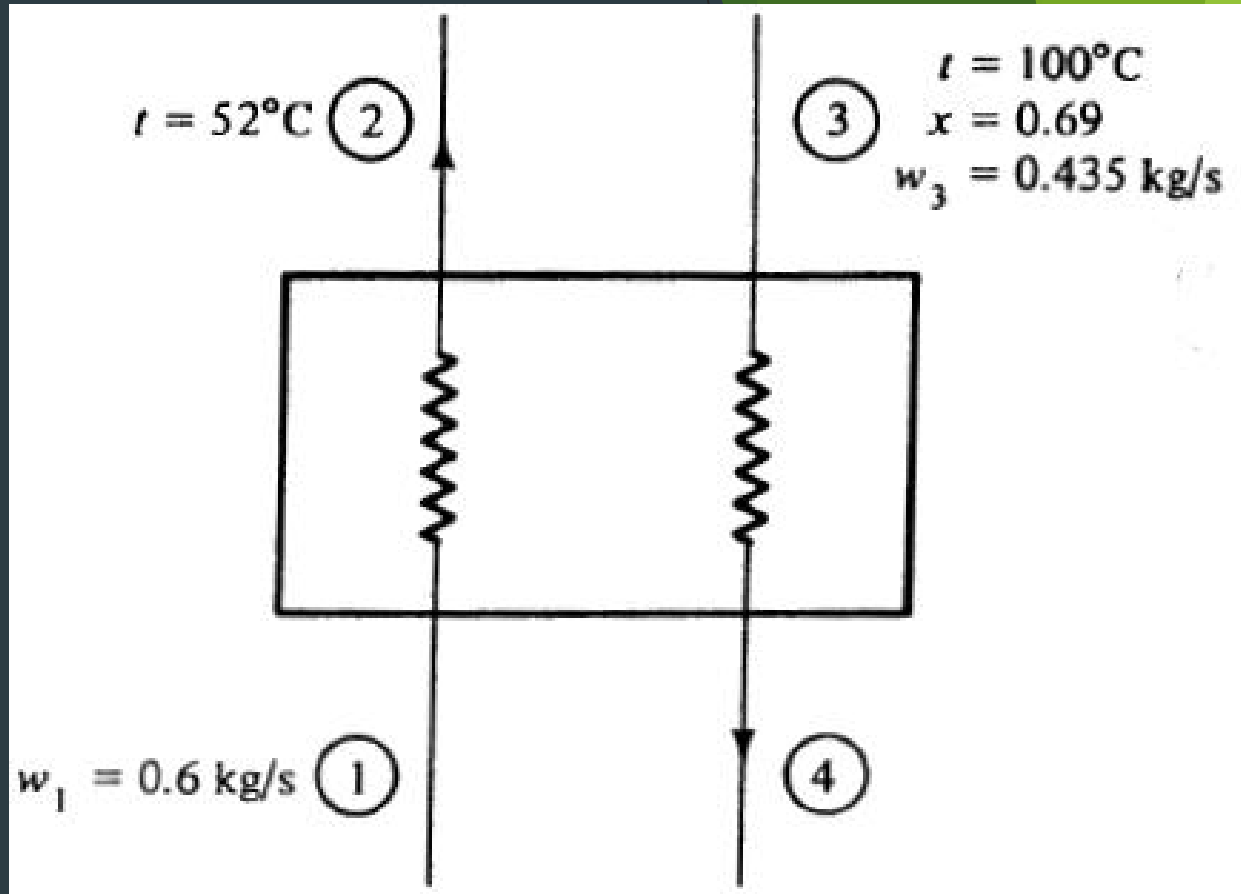
$$h_4 \cdot w_4 + h_2 \cdot w_2 = h_3 \cdot w_4 + h_1 \cdot w_2$$

$$w_4 (h_4 - h_3) = w_2 (h_1 - h_2)$$

$$h_4 = h_3 + w_2 (h_1 - h_2) / w_4$$

$$h_4 = -54 + 0.6 (-168 - (-120)) / 0.435$$

$$h_4 = -120 \text{ kJ/kg}$$



From Fig. 17-8 the condition of solution at point 4, which is $h = -120 \text{ kJ/kg}$, $x = 0.69$, is found to be crystallized.

Capacity control 'capacity reduction'

- ▶ The need for capacity control arises when the refrigeration load drops off or decreasing in the chilled-water temperature returning to the absorption unit (assuming a constant-rate flow of chilled water).
- ▶ With no capacity control the temperature of the chilled water leaving the evaporator would decrease and the pressure on the low pressure side decrease that lead to the refrigerant water would freeze.
- ▶ The most control systems on absorption units attempt to regulate a constant temperature of chilled water leaving the evaporator by reducing in the refrigerant-water flow in the absorption units that can achieve by using three methods:
 1. Reducing the flow rate delivered by the pump at position 1
 2. Reducing the generator temperature
 3. Increasing the condensing temperature

Method I

- ▶ Reducing the flow rate delivered by the pump at position 1 lead to reduce in the rate of refrigerant flow through the condenser and evaporator and then reduction in refrigerating capacity.
- ▶ In this method the proportion of mass flow rate reduction in all component must be same to keep the concentrations of solution remain unchanged.
- ▶ But, the concentrations remain unchanged only if the operating temperatures in the components also remain fixed.
- ▶ Therefore to fixed the operation temperature in the components achieve by Cooperation with method 2 or method 3 to fixed the generator and condenser temperature.

Method 2

Reduction in the generator temperature will reduce the refrigerating capacity and can be achieved by:

- ▶ Throttling the pressure of the steam entering the generator
- ▶ Reducing the flow rate of hot water.

Method 3

Increasing the condensing temperature with reducing the refrigerating capacity can be achieved by:

- ▶ Increasing the temperature of cooling water supplied to the condenser.

Comparing between Aqua-ammonia system and LiBr-water system

- 1- The aqua-ammonia absorption systems was used in years before the LiBr-water combination.
- 2- The aqua-ammonia absorption systems used water as the absorbent and ammonia as the refrigerant.
- 3- The aqua-ammonia system consists of generator, absorber, condenser, evaporator, and solution heat exchanger-plus a rectifier and analyzer.

- 4- The need for a rectifier and analyzer to remove as much water vapor as possible that cause rise in the evaporator temperature.
- 5- The aqua-ammonia system is capable of achieving evaporating temperatures below 0°C , but the LiBr-water system is limited in no lower than about 3°C .
- 6- The LiBr-water system operates at pressures below atmospheric, resulting in unavoidable leakage of air into the system, which must be purged periodically while the aqua-ammonia system operating at pressures above atmospheric.

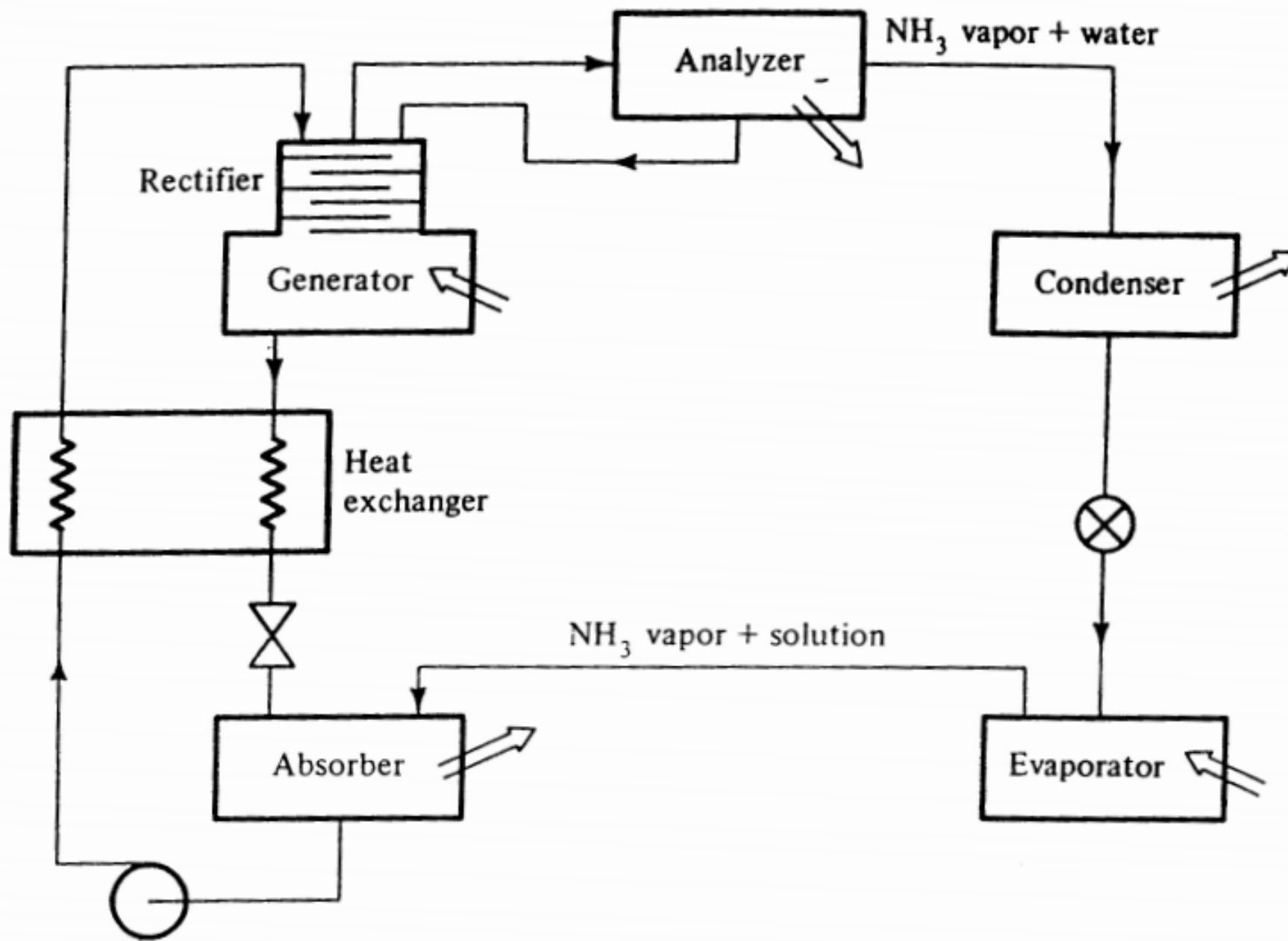


Figure 17-17 Aqua-ammonia absorption system.

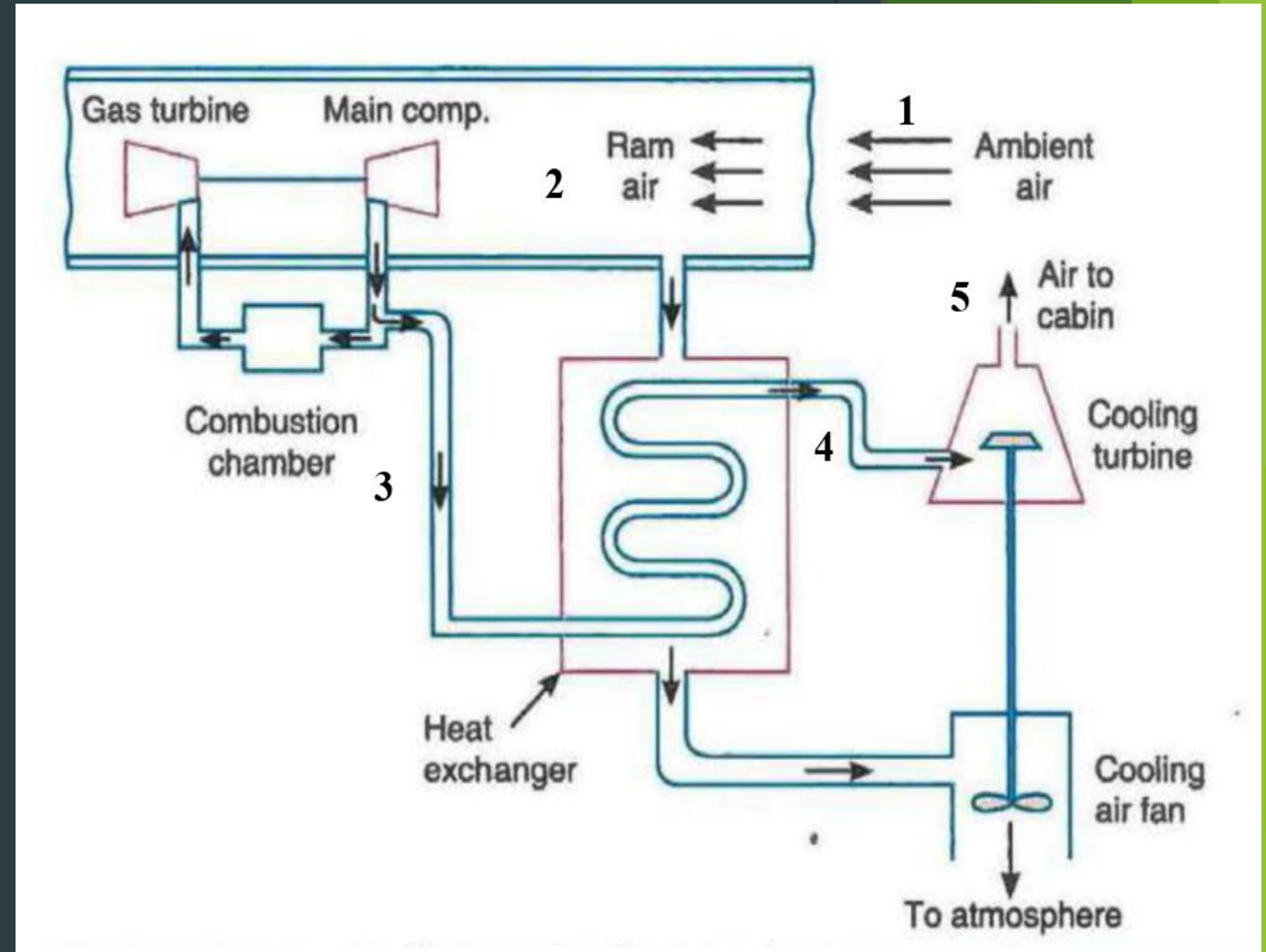
Simple Air Cooling System

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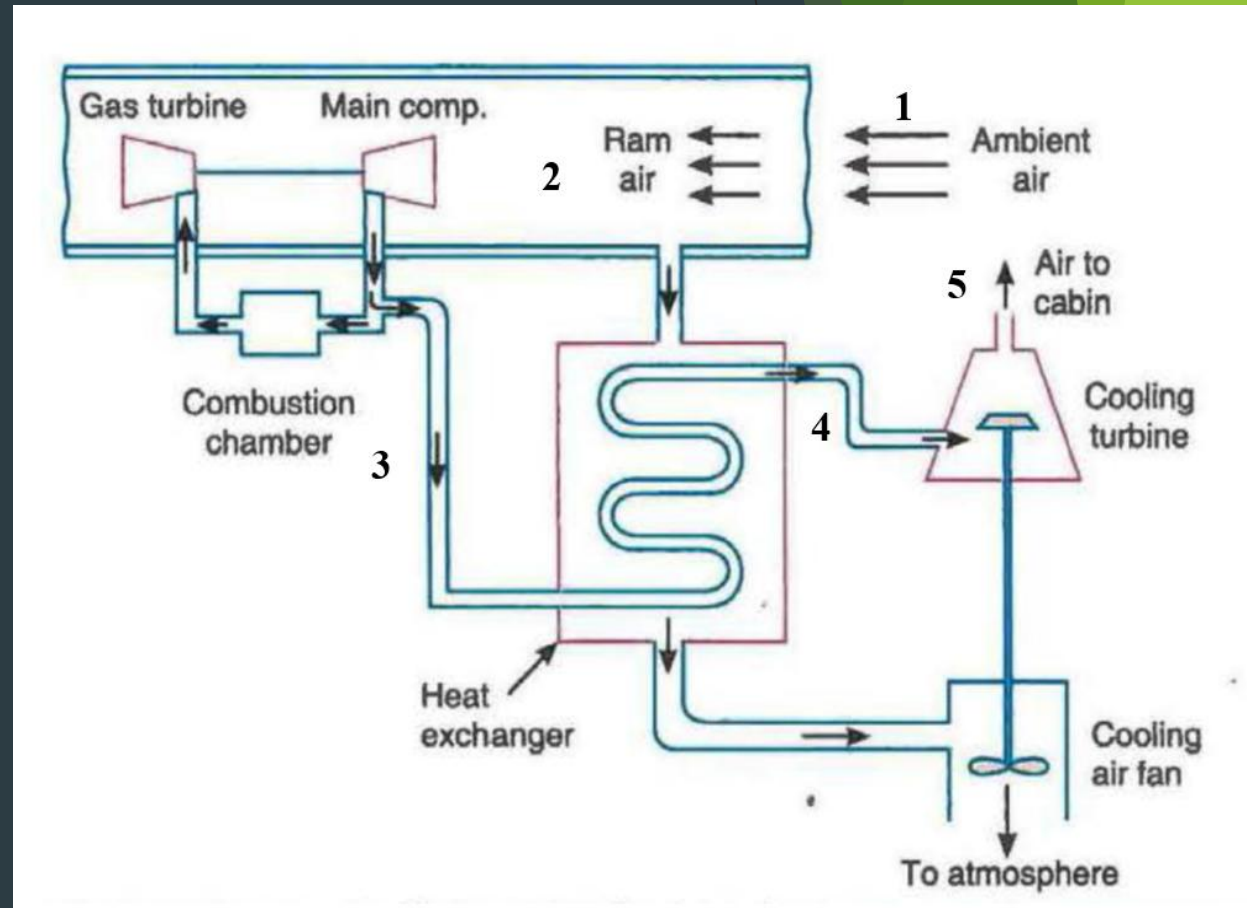
Description

► The main components of the simple air cooling system for aircrafts are:

1. Main compressor driven by a gas turbine.
2. Heat exchanger.
3. Cooling turbine.
4. Cooling air fan.



- ▶ The air required for refrigeration system is bled off from the main compressor.
- ▶ A high pressure and high temperature air is cooled initially in the heat exchanger where ram air is used for cooling.
- ▶ It is further cooled in the cooling turbine by the process of expansion.
- ▶ The work of this turbine is used to drive the cooling fan which draws cooling air through the heat exchanger.



Advantages

- ▶ The air is easily available and there is no cost of the refrigerant.
- ▶ The air is non-toxic and non-inflammable.
- ▶ The leakage of air in small amounts is acceptable.
- ▶ Since the main compressor is employed for the compressed air source, therefore there is no problem of space for extra compressor.
- ▶ The chilled air is directly used for cooling.
- ▶ Since the pressure in the whole system is quite low, therefore the piping, ducting etc. are quite simple to design, fabricate and maintain.

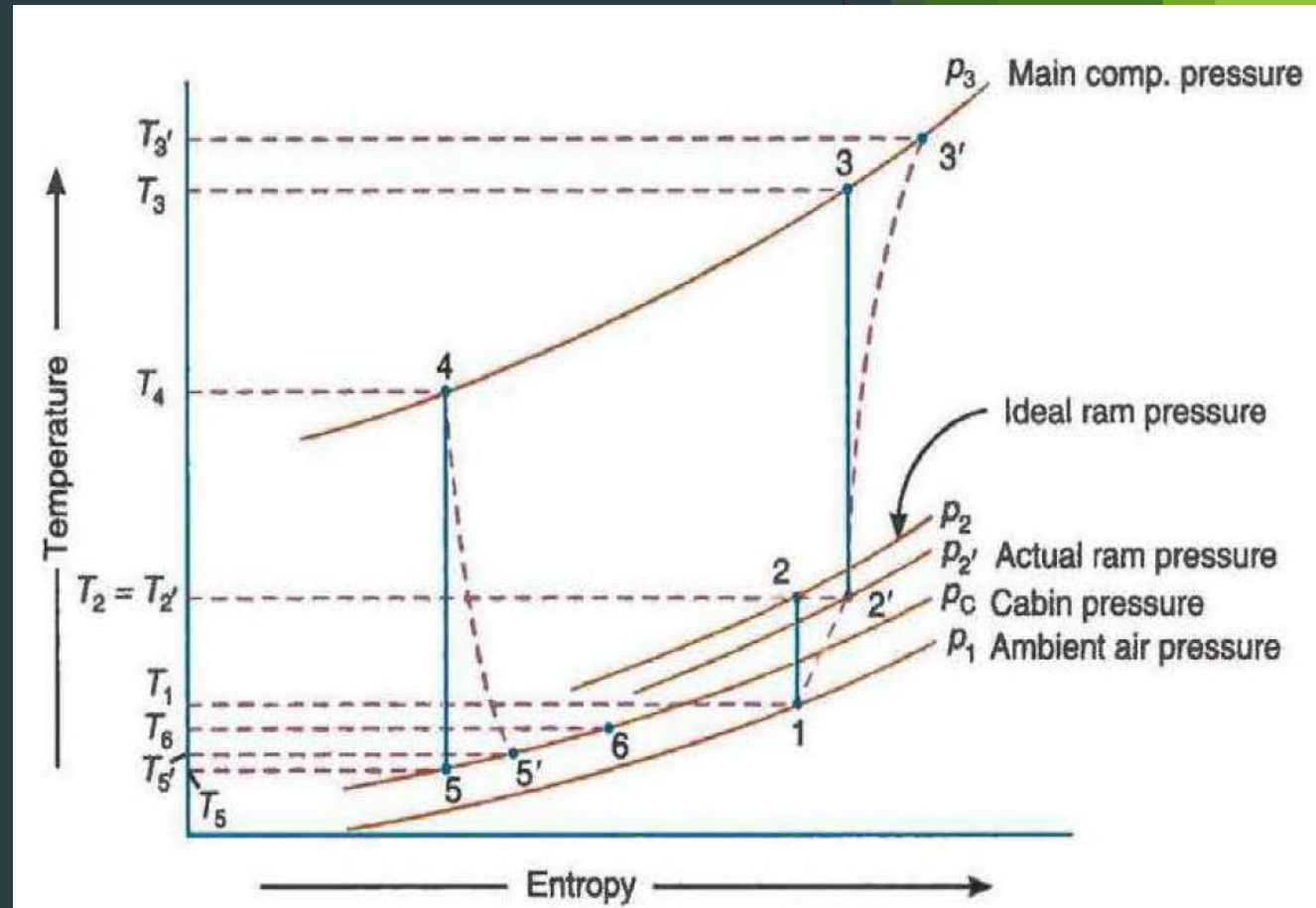
Disadvantages

- ▶ It has low coefficient of performance.
- ▶ The rate of air circulation is relatively large.

A simple air cooling system processes

1. Ramming process:

- ▶ Let the pressure and temperature of ambient air is p_1 and T_1 .
- ▶ The ambient air is rammed isentropically from pressure p_1 and temperature T_1 to the pressure p_2 and temperature T_2 by the vertical line 1-2
- ▶ Because of internal friction due to irreversibility the actual ramming process is shown by the curve 1-2' which is adiabatic but not isentropic due to friction.



Energy equation during ramming

$$h_2 + \frac{V_2^2}{2} = h_1 + \frac{V_1^2}{2}$$

$V_1 =$ velocity of air relative to plane $V_2 = 0$ (stagnation state)

$$h_2 = h_1 + \frac{V_1^2}{2}$$

$$c_p T_2 = c_p T_1 + \frac{V_1^2}{2} \quad \rightarrow \quad T_2 = T_1 + \frac{V_1^2}{2c_p} \quad \rightarrow \quad \frac{T_2}{T_1} = 1 + \frac{V_1^2}{2c_p T_1}$$

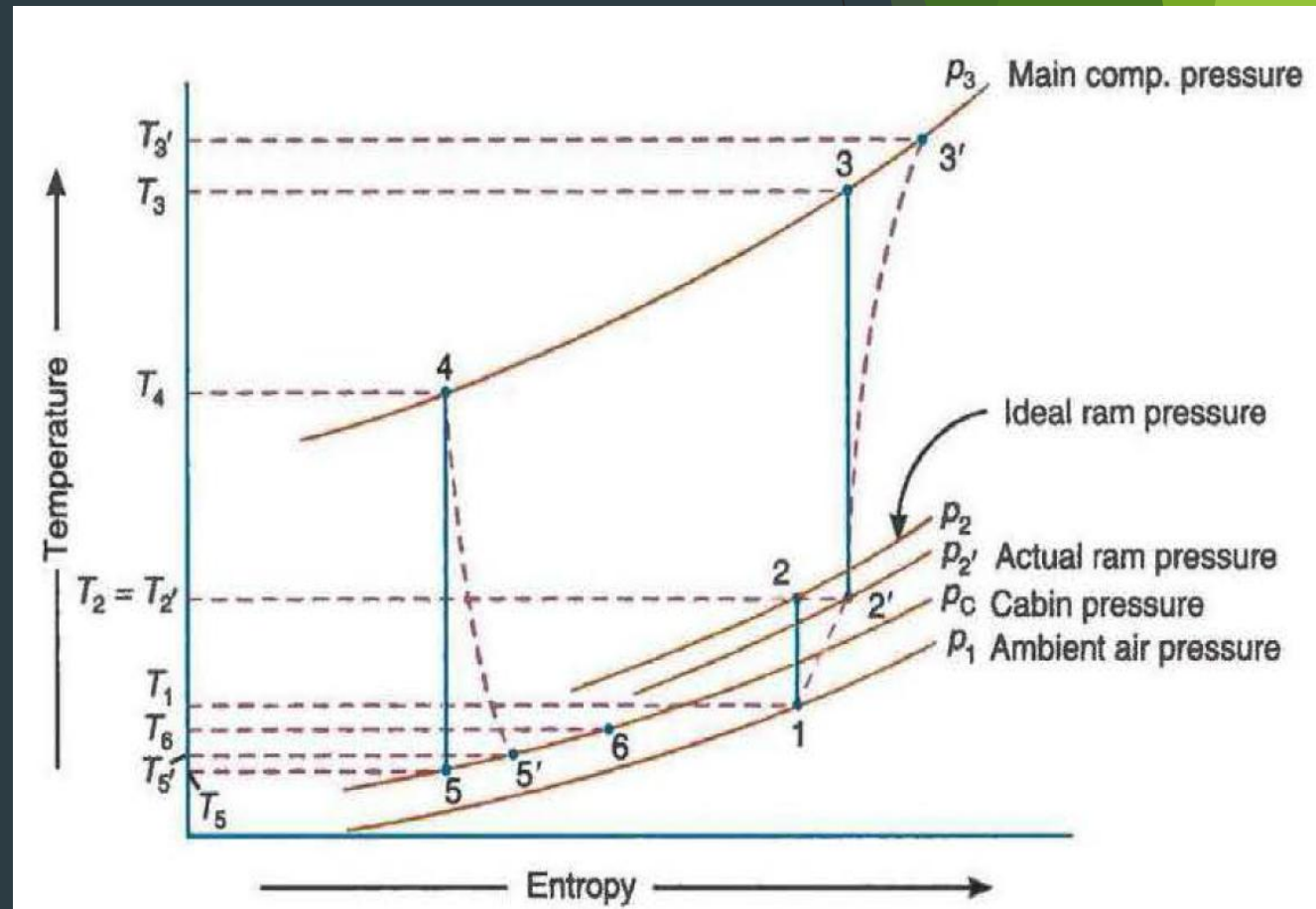
$$c_p = \frac{\gamma}{\gamma - 1} R, \quad a = \sqrt{\gamma R T_1} \quad (\text{local sonic velocity})$$

$$\frac{T_2}{T_1} = 1 + \frac{V_1^2}{2 \left(\frac{\gamma}{\gamma - 1} \right) R T_1} = 1 + \frac{(\gamma - 1) V_1^2}{2 a^2} = 1 + \frac{(\gamma - 1)}{2} M^2$$

$M =$ Mach number (air velocity/local sonic velocity)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (\text{Isentropic compression})$$

$$\eta_{\text{Ram}} = \frac{\text{actual pressure rise}}{\text{reversible adiabatic pressure rise}} = \frac{p_{2'} - p_1}{p_2 - p_1}$$



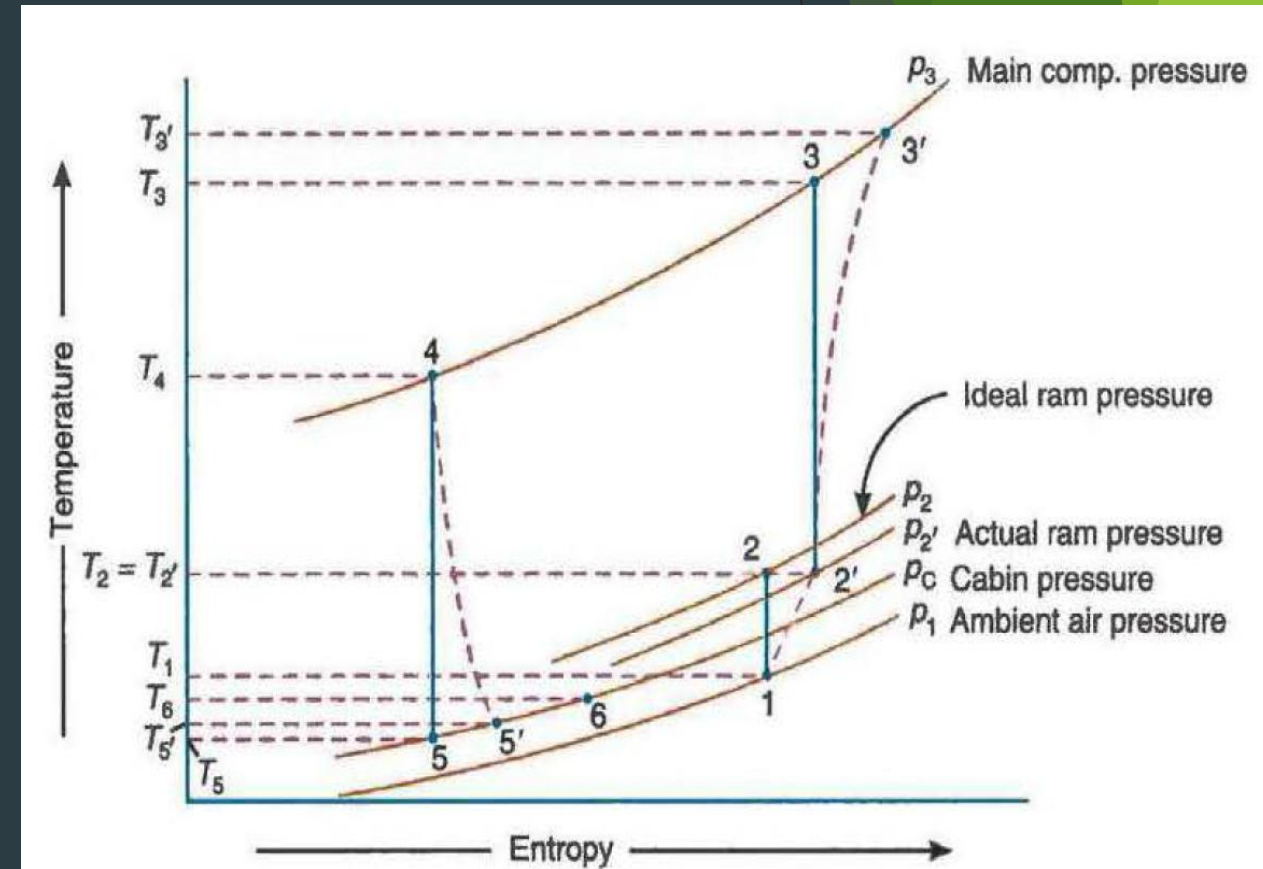
2. Compression process:

- ▶ The isentropic compression of air in the main compressor is represented by the line 2'-3.
- ▶ In actual practice, because of internal friction, due to irreversibility, actual compression is represented by the curve 2'-3'. The work done during this compression process is given by:

$$W_c = m_a c_p (T_{3'} - T_{2'})$$

$$\eta_{comp} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

$$\frac{T_3}{T_{2'}} = \left(\frac{p_3}{p_{2'}} \right)^{\frac{\gamma-1}{\gamma}} \quad (\text{Isentropic compression})$$



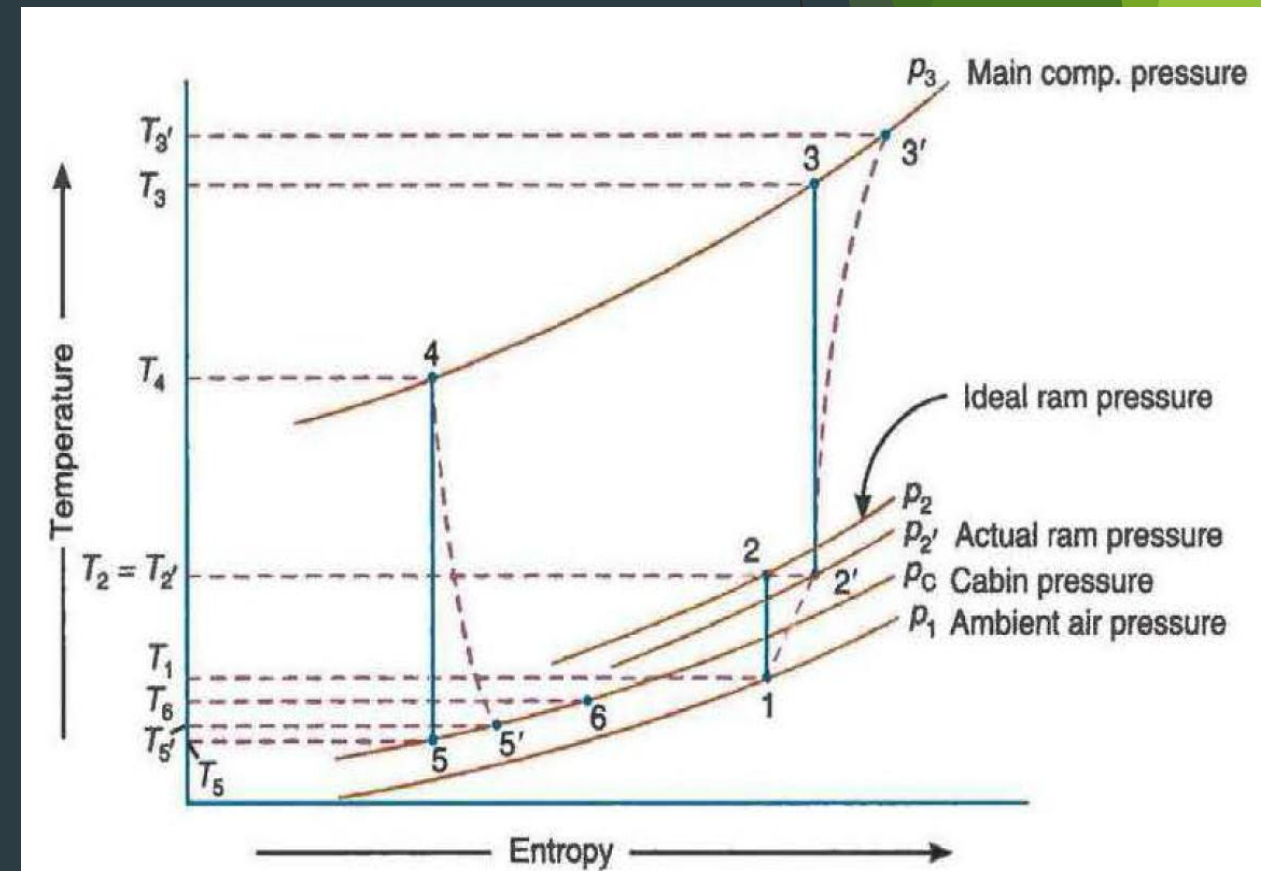
3. Cooling process:

- ▶ The compressed air is cooled by the ram air in the heat exchanger. This process is shown by the curve 3' - 4.
- ▶ In actual practice, there is a pressure drop in the heat exchanger which is not shown in the figure.
- ▶ The temperature of air decreases from $T_{3'}$ to T_4 .
- ▶ The heat rejected in the heat exchanger during the cooling process is given by

$$Q_R = m_a c_p (T_{3'} - T_4)$$

- ▶ The heat exchanger effectiveness

$$\epsilon_{heat\ exchanger} = \frac{T_{3'} - T_4}{T_{3'} - T_{2'}}$$



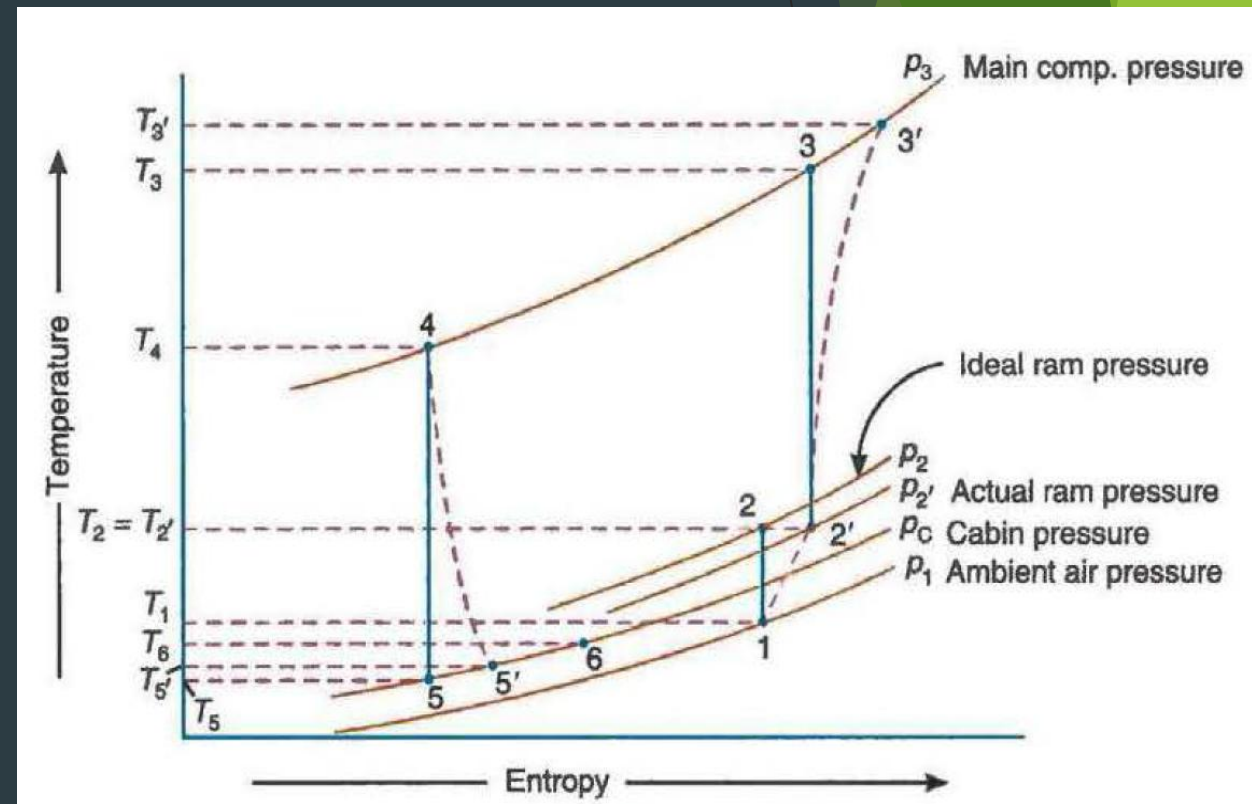
4. Expansion process:

- ▶ The cooled air is now expanded isentropically in the cooling turbine shown by the curve 4 - 5.
- ▶ In actual practice, because of internal friction due to irreversibility, the actual expansion in the cooling turbine is shown by the curve 4 - 5'. The work done by the cooling turbine during this expansion process is given by

$$W_T = m_a c_p (T_4 - T_{5'})$$

$$\eta_T = \frac{T_4 - T_{5'}}{T_4 - T_5}$$

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5} \right)^{\frac{\gamma-1}{\gamma}} \quad (\text{Isentropic compression})$$



5. Refrigeration process:

- ▶ The air from the cooling turbine is sent to cabin and cock pit where it gets heated by the heat of equipment and occupancy.
- ▶ This process is shown by the curve 5' – 6.

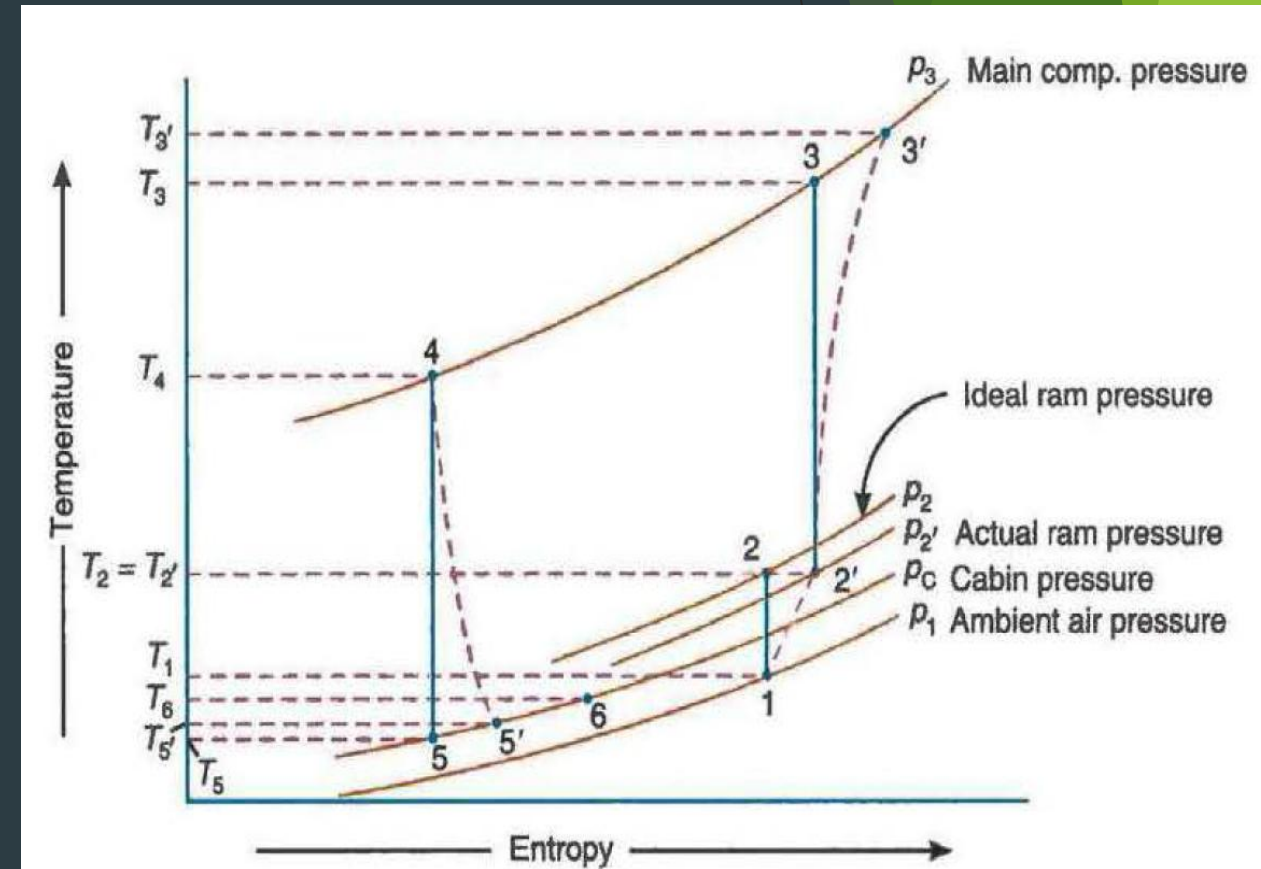
- ▶ The refrigerating effect produced or heat absorbed is given

$$R_E = m_a c_p (T_6 - T_{5'})$$

- ▶ The C.O.P. of the air cycle

$$C.O.P = \frac{\text{Refrigerating effect}}{\text{Work of compressor}}$$

$$= \frac{m_a c_p (T_6 - T_{5'})}{m_a c_p (T_{3'} - T_{2'})}$$



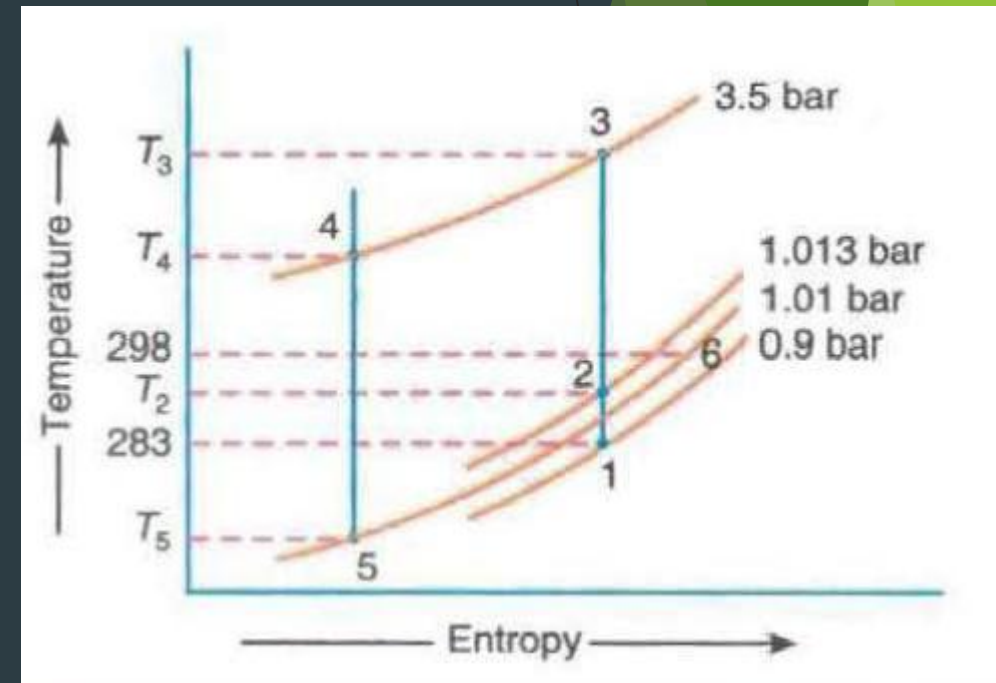
Example 1:

A simple air cooled system is used for an airplane having a load of 10 tonnes. The atmospheric pressure and temperature are 0.9 bar and 10°C respectively. The pressure increases to 1.013 bar due to ramming. The temperature of the air is reduced by 50°C in the heat exchanger. The pressure in the cabin is 1.01 bar and the temperature of air leaving the cabin is 25°C. Assume that all the expansions and compressions are isentropic. The pressure of the compressed air is 3.5 bar. Determine:

- ▶ Power required to take the load of cooling in the cabin.
- ▶ C.O.P. of the system

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.013}{0.9} \right)^{\frac{1.4-1}{1.4}} = (1.125)^{0.286} = 1.034$$

$$T_2 = T_1 \times 1.034 = 283 \times 1.034 = 292.6 \text{ K}$$



$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.5}{1.013} \right)^{\frac{1.4-1}{1.4}} = (3.45)^{0.286} = 1.425$$

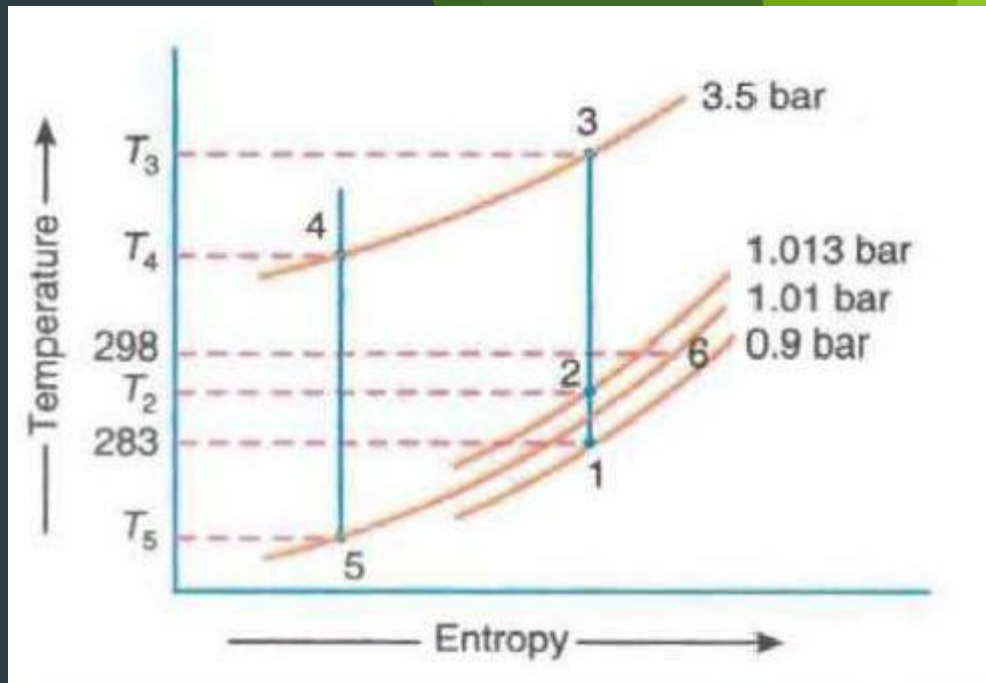
$$T_3 = T_2 \times 1.425 = 292.6 \times 1.425 = 417 \text{ K} = 144^\circ\text{C}$$

- Since the temperature of air is reduced by 50°C in the heat exchanger, therefore temperature

$$T_4 = 144 - 50 = 94^\circ\text{C} = 367 \text{ K}$$

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.01}{3.5} \right)^{\frac{1.4-1}{1.4}} = (0.288)^{0.286} = 0.7$$

$$T_5 = T_4 \times 0.7 = 367 \times 0.7 = 257 \text{ K}$$



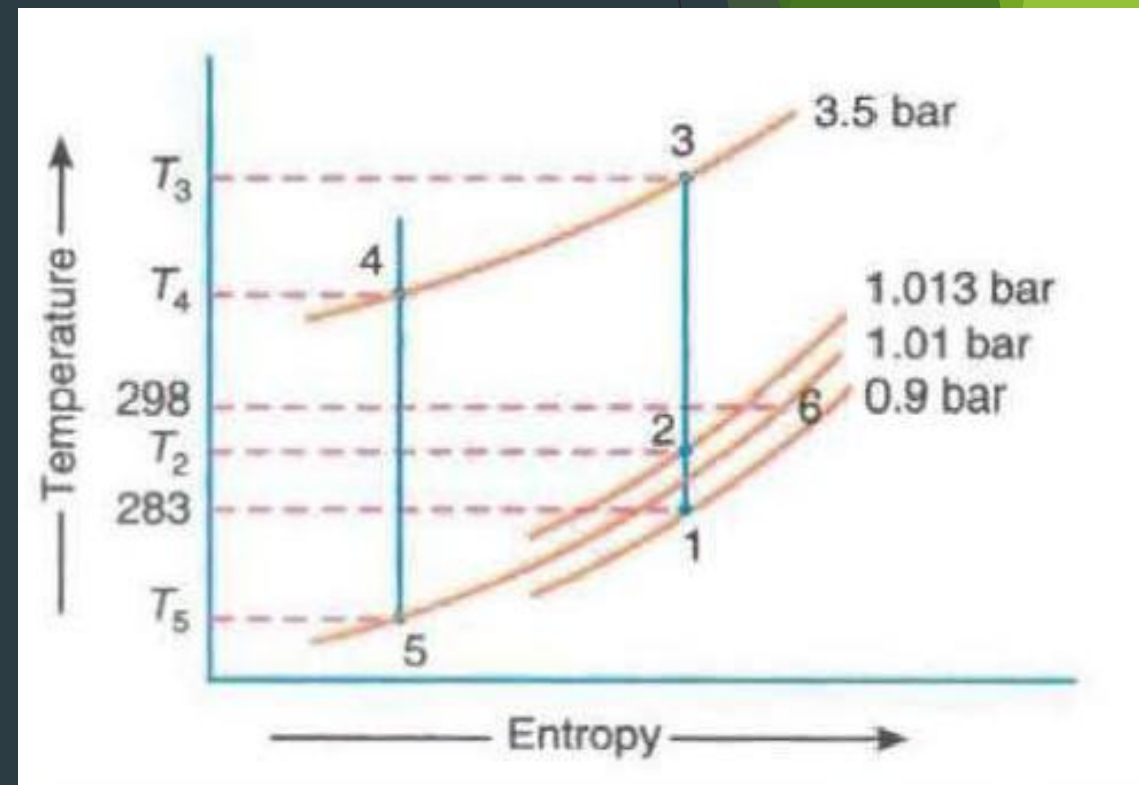
$$m_a = \frac{3.516 \times \text{tonnes}}{c_p \times (T_6 - T_5)}$$

$$m_a = \frac{3.516 \times 10}{1 \times (298 - 257)} = 0.85 \text{ kg/s}$$

$$\text{Power} = m_a c_p \times (T_3 - T_2)$$

$$\text{Power} = 0.85 \times 1 \times (417 - 292.6) = 106 \text{ kW}$$

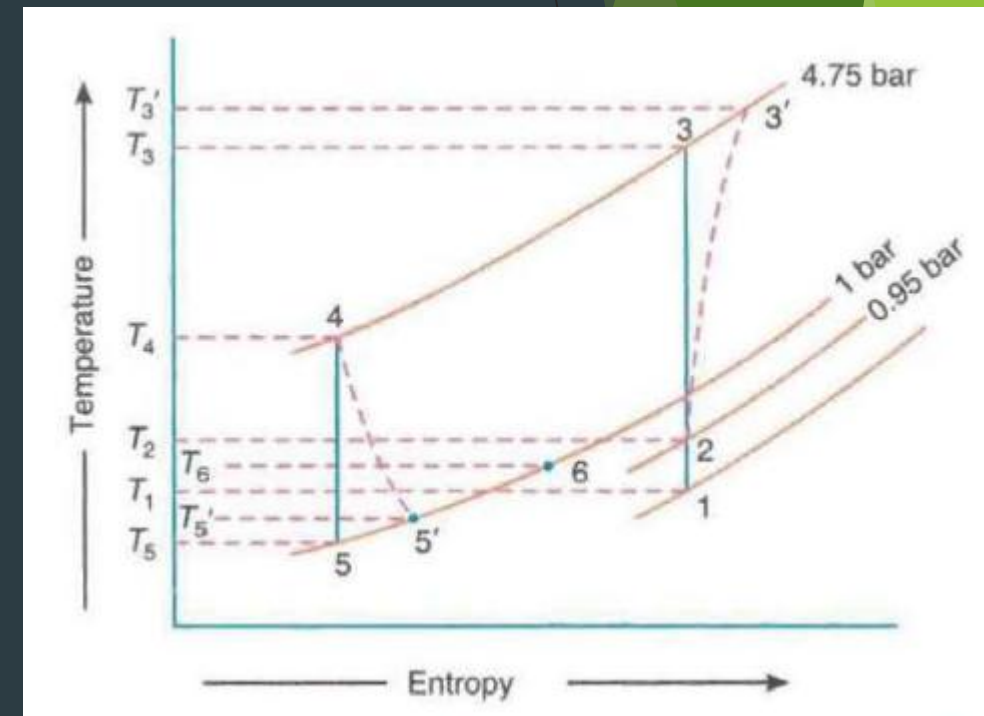
$$\text{COP} = \frac{3.516 \times \text{tonnes}}{\text{Power}} = \frac{3.516 \times 10}{106} = 0.33$$



Example 2:

An aircraft refrigeration plant has to handle a cabin load of 30 tonnes. The atmospheric temperature is 17°C . The atmospheric air is compressed to a pressure of 0.95 bar and temperature of 30°C due to ram action. This air is then further compressed in a compressor to 4.75 bar, cooled in a heat exchanger to 67°C , expanded in a turbine to 1 bar pressure and supplied to the cabin. The air leaves the cabin at a temperature of 27°C . The isentropic efficiencies of both compressor and turbine are 0.9. Calculate:

- ▶ The mass of air circulated per minute.
- ▶ C.O.P. of the system.



for isentropic compression process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.75}{0.95} \right)^{\frac{1.4-1}{1.4}} = (5)^{0.286} = 1.584$$

$$T_3 = T_2 \times 1.584 = 303 \times 1.584 = 480 \text{ K}$$

isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

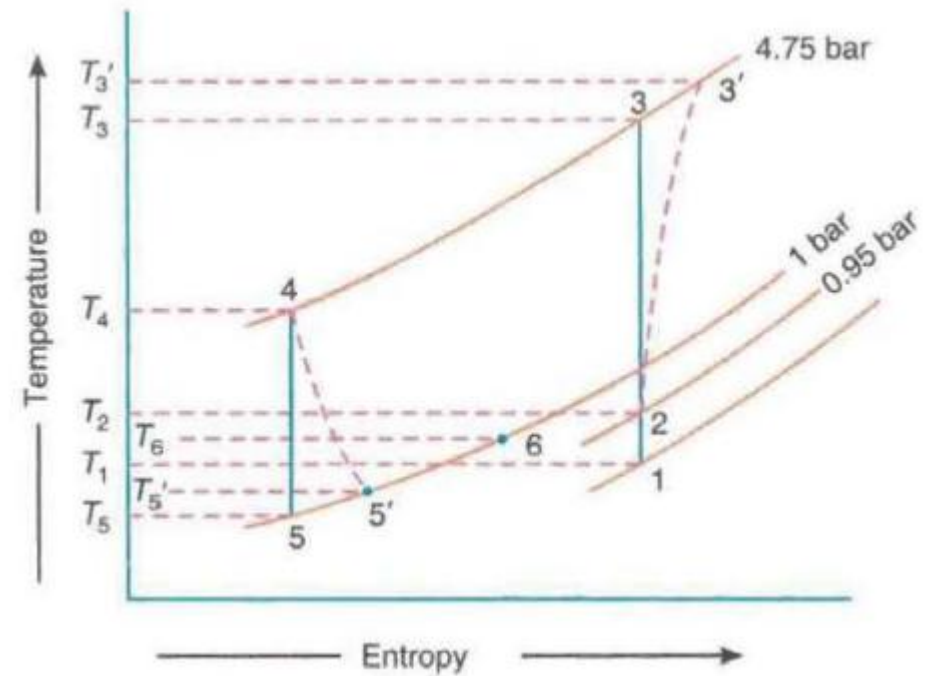
$$0.9 = \frac{480 - 303}{T_{3'} - 303} = \frac{177}{T_{3'} - 303}$$

$$\therefore T_{3'} - 303 = 177/0.9 = 196.7 \text{ or } T_{3'} = 303 + 196.7 = 499.7 \text{ K}$$

the isentropic expansion process 4-5,

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.75}{1} \right)^{\frac{1.4-1}{1.4}} = (4.75)^{0.286} = 1.561$$

$$T_5 = T_4 / 1.561 = 340 / 1.561 = 217.8 \text{ K}$$



isentropic efficiency of the turbine,

$$\eta_T = \frac{\text{Actual increase in temperature}}{\text{Isentropic increase in temperature}} = \frac{T_4 - T_{5'}}{T_4 - T_5}$$

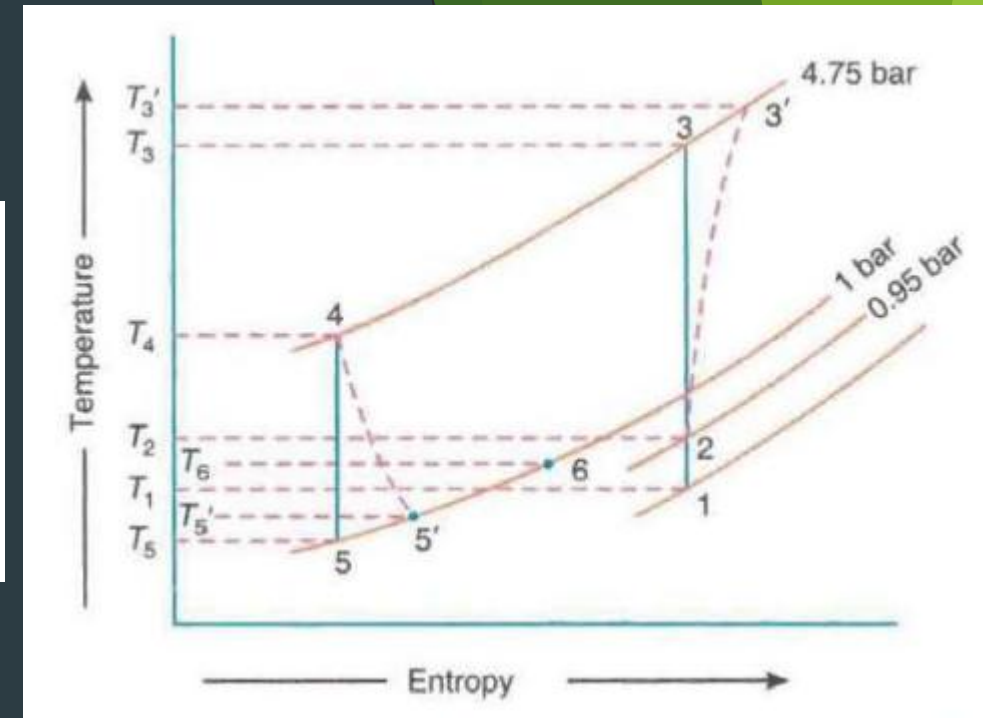
$$0.9 = \frac{340 - T_{5'}}{340 - 217.8} = \frac{340 - T_{5'}}{122.2}$$

$$\therefore T_{5'} = 340 - 0.9 \times 122.2 = 230 \text{ K}$$

$$m_a = \frac{3.516 \times \text{tonnes}}{c_p \times (T_6 - T_{5'})}$$

$$m_a = \frac{3.516 \times 30}{1.005 \times (300 - 230)} = 1.5 \text{ kg/s}$$

$$COP = \frac{3.516 \times \text{tonnes}}{m_a \times c_p \times (T_{3'} - T_2)} = \frac{3.516 \times 30}{1.5 \times 1.005 \times (499.7 - 303)} = 0.356$$

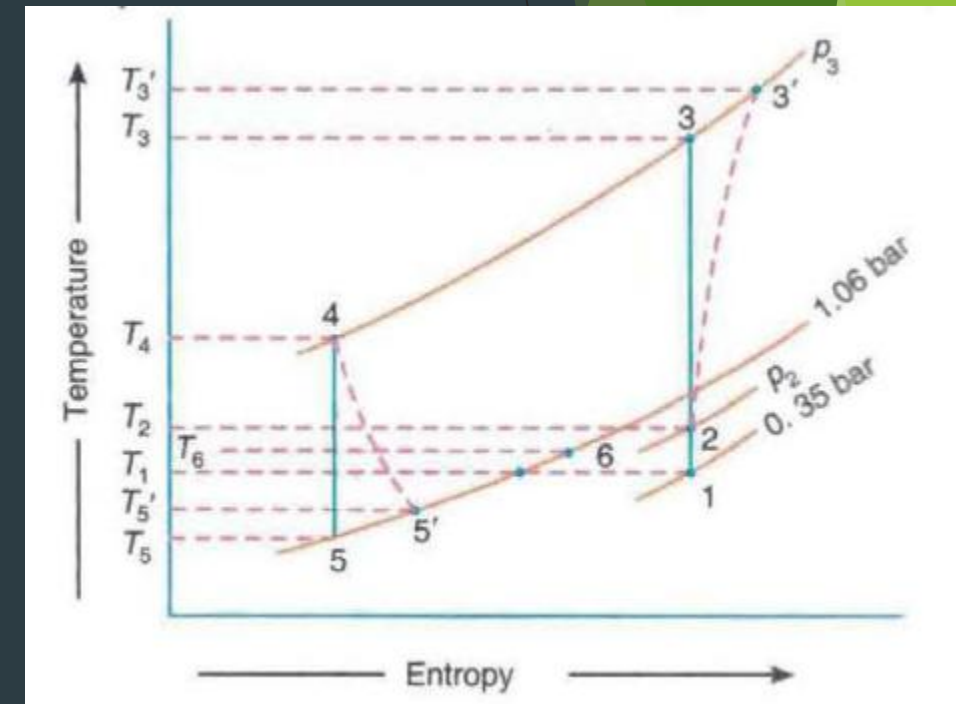


Example 3:

An aircraft moving with speed of 1000 km/h uses simple gas refrigeration cycle for air-conditioning. The ambient pressure and temperature are 0.35 bar and -10°C respectively. The pressure ratio of compressor is 4.5. The heat exchanger effectiveness is 0.95. The isentropic efficiencies of compressor and expander are 0.8. The cabin pressure and temperature are 1.06 bar and 25°C .

Determine:

- ▶ The temperatures and pressures at all points.
- ▶ The volume flow rate through compressor inlet and expander outlet for 100 TR



$$T_2 = T_1 + \frac{V^2}{2000 c_p} = 263 + \frac{(277.8)^2}{2000 \times 1.005}$$

$$= 263 + 38.4 = 301.4 \text{ K Ans.}$$

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1} \right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{301.4}{263} \right)^{\frac{1.4}{1.4-1}} = (1.146)^{3.5} = 1.611$$

$$p_2 = p_1 \times 1.611 = 0.35 \times 1.611 = 0.564 \text{ bar Ans.}$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p}$$

Since $p_3/p_2 = 4.5$ (Given), therefore

$$p_3 = p_2 \times 4.5 = 0.564 \times 4.5 = 2.54 \text{ bar Ans.}$$

We know that for isentropic compression process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = (4.5)^{\frac{1.4-1}{1.4}} = (4.5)^{0.286} = 1.537$$

$$\therefore T_3 = T_2 \times 1.537 = 301.4 \times 1.537 = 463.3 \text{ K}$$

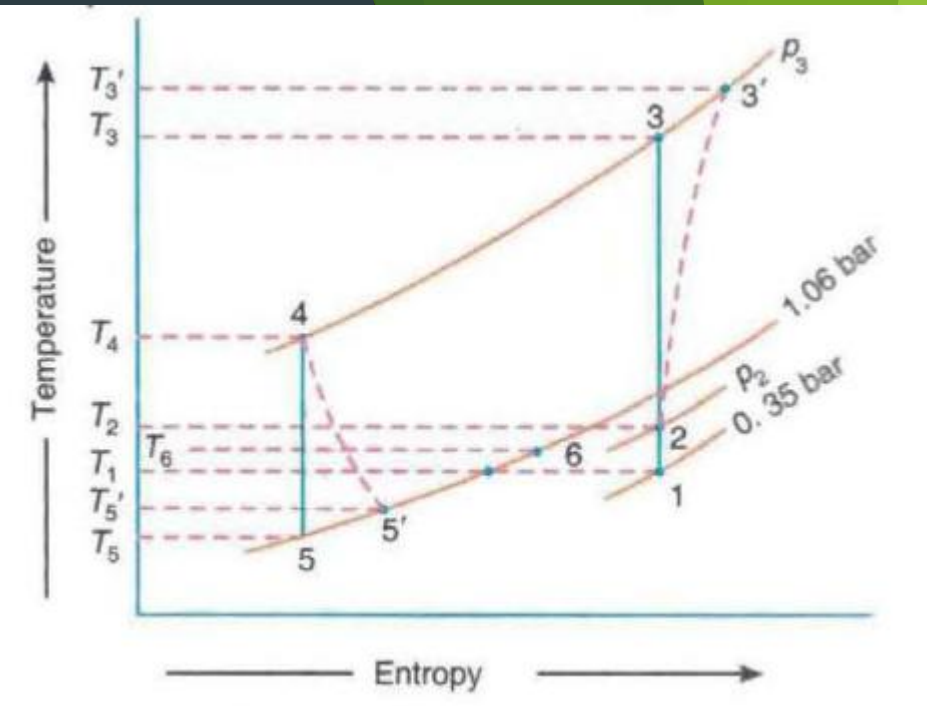
We also know that isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic temperature rise}}{\text{Actual temperature rise}} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

$$0.8 = \frac{463.3 - 301.4}{T_{3'} - 301.4} = \frac{161.9}{T_{3'} - 301.4}$$

$$T_{3'} - 301.4 = 161.9/0.8 = 202.4$$

$$\therefore T_{3'} = 301.4 + 202.4 = 503.8 \text{ K Ans.}$$



Effectiveness of the heat exchanger (η_H),

$$0.95 = \frac{T_{3'} - T_4}{T_{3'} - T_2} = \frac{503.8 - T_4}{503.8 - 301.4} = \frac{503.8 - T_4}{202.4}$$

$$\therefore T_4 = 503.8 - 0.95 \times 202.4 = 311.5 \text{ K Ans.}$$

$$p_4 = p_3 = 2.54 \text{ bar Ans.}$$

We know that for isentropic expansion process 4-5,

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.06}{2.54} \right)^{\frac{1.4-1}{1.4}} = (0.417)^{0.286} = 0.7787$$

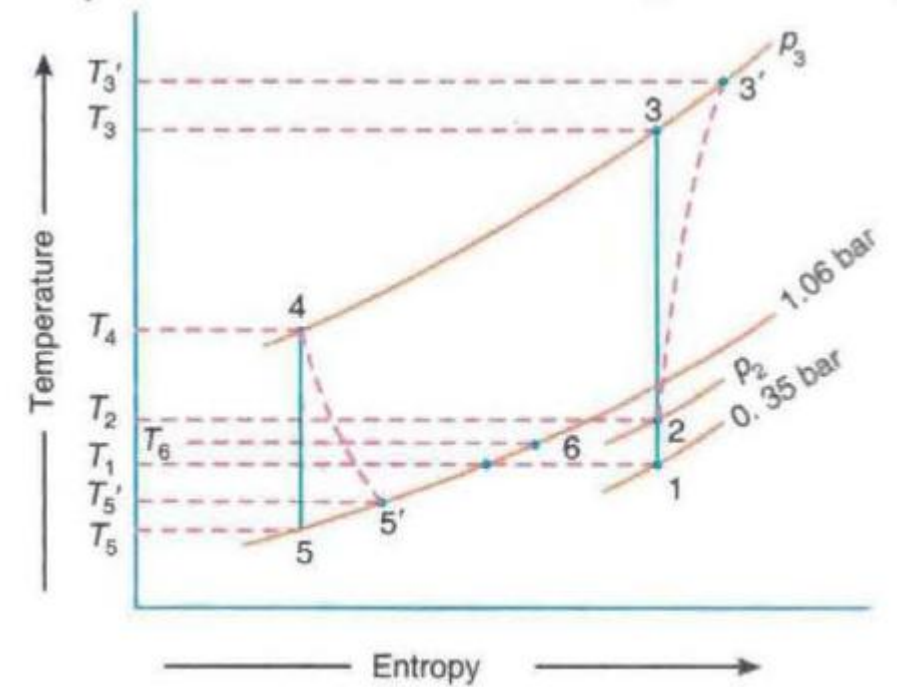
$$\therefore T_5 = T_4 \times 0.7787 = 311.5 \times 0.7787 = 243 \text{ K}$$

isentropic efficiency of the expander,

$$\eta_E = \frac{\text{Actual temperature rise}}{\text{Isentropic temperature rise}} = \frac{T_4 - T_{5'}}{T_4 - T_5}$$

$$0.8 = \frac{311.5 - T_{5'}}{311.5 - 243} = \frac{311.5 - T_{5'}}{68.5}$$

$$T_{5'} = 311.5 - 0.8 \times 68.5 = 256.7 \text{ K Ans.}$$



$$m_a = \frac{3.516 \times \text{tonnes}}{c_p \times (T_6 - T_{5'})}$$

$$m_a = \frac{3.516 \times 100}{1.005 \times (298 - 256.7)} = 8.43 \text{ kg/s}$$

$$p_2 v_2 = m_a R T_2$$

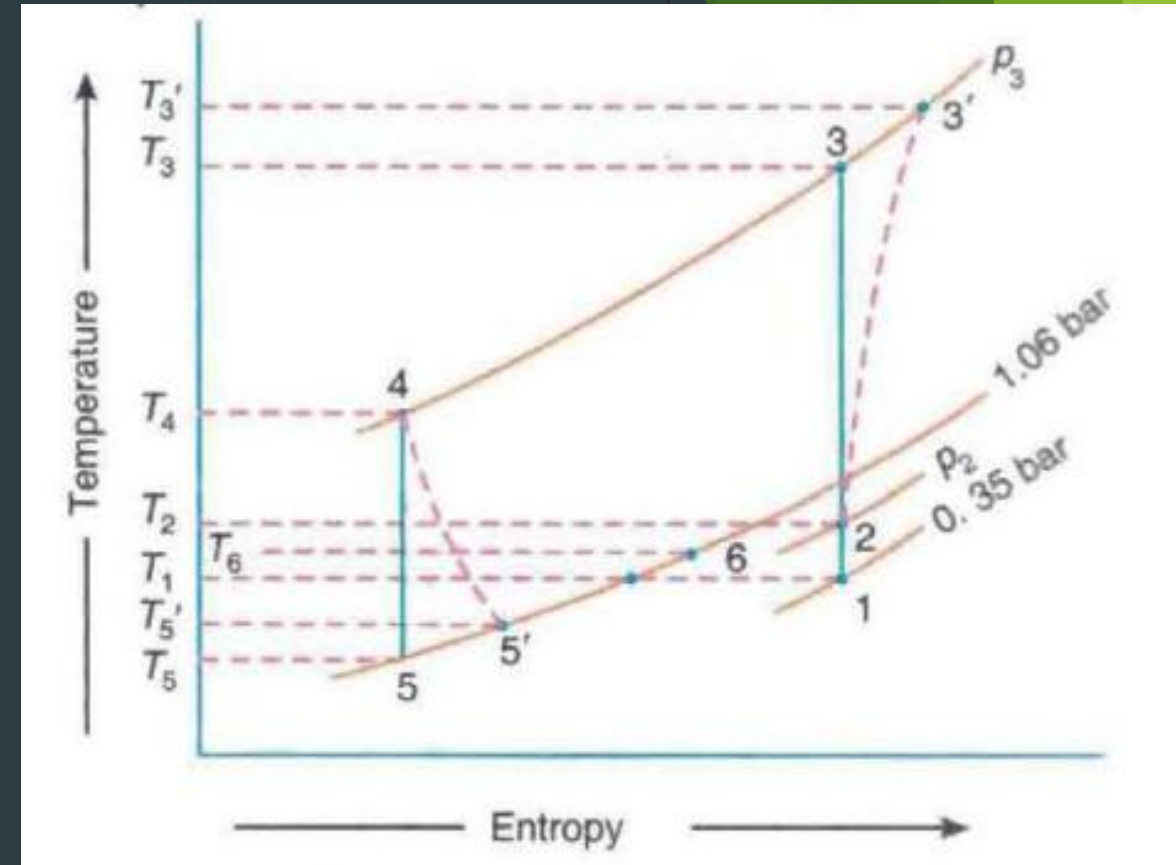
$$0.564 \times 10^5 \times v_2 = 8.43 \times 287 \times 301.4$$

$$v_2 = 12.93 \text{ m}^3/\text{s}$$

$$p_{5'} v_{5'} = m_a R T_{5'}$$

$$1.06 \times 10^5 \times v_{5'} = 8.43 \times 287 \times 256.7$$

$$v_{5'} = 5.86 \text{ m}^3/\text{s}$$



Example 4:

The cock pit of a jet plane flying at a speed of 1200 km/h is to be cooled by a simple air cooling system. The cock pit is to be maintained at 25°C and the pressure in the cock pit is 1 bar. The ambient air pressure and temperature are 0.85 bar and 30°C. The other data available is as follows :

Cock-pit cooling load = 10 TR; Main compressor pressure ratio = 4; Ram efficiency = 90 %; The temperature of air leaving the heat exchanger and entering the cooling turbine = 60°C; pressure drop in the heat exchanger = 0.5 bar; Pressure loss between the cooler turbine and cock pit = 0.2 bar.

Assuming the isentropic efficiencies of main compressor and cooler turbine as 80%. Find:

- ▶ The quantity of air passed through the cooling turbine
- ▶ C.O.P. of the system.

$$T_2 = T_2' = T_1 + \frac{V^2}{2000 c_p} = 303 + \frac{(333.3)^2}{2000 \times 1}$$

$$= 303 + 55.5 = 358.5 \text{ K}$$

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1} \right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{358.5}{303} \right)^{\frac{1.4}{1.4-1}} = (1.183)^{3.5} = 1.8$$

$$p_2 = p_1 \times 1.8 = 0.85 \times 1.8 = 1.53 \text{ bar}$$

ram efficiency,

$$\eta_R = \frac{\text{Actual pressure rise}}{\text{Isentropic pressure rise}} = \frac{p_{2'} - p_1}{p_2 - p_1}$$

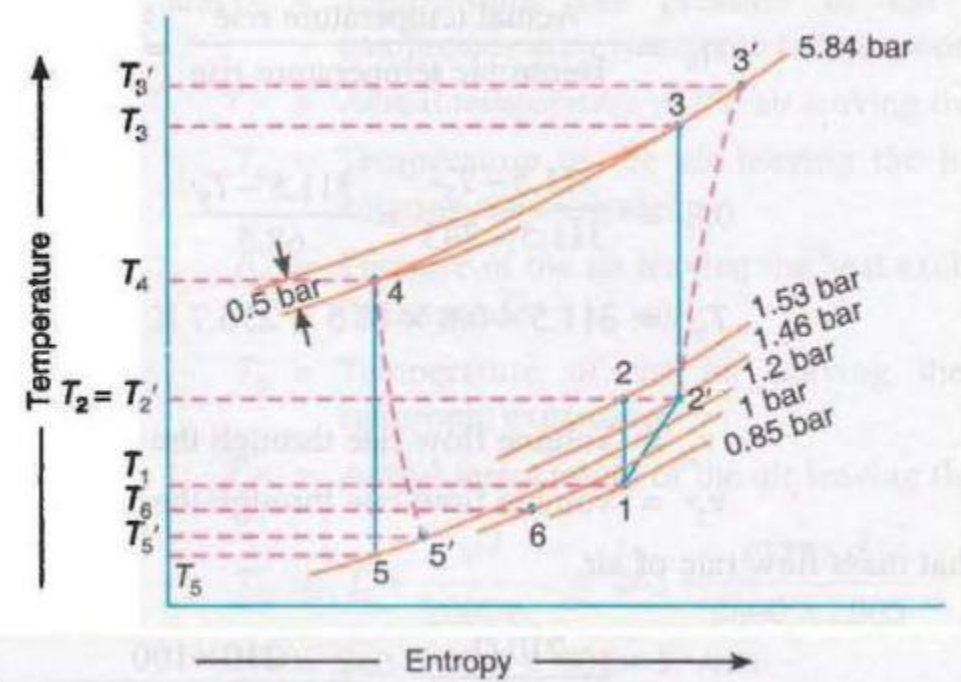
$$0.9 = \frac{p_{2'} - 0.85}{1.53 - 0.85} = \frac{p_{2'} - 0.85}{0.68}$$

$$p_{2'} = 0.9 \times 0.68 + 0.85 = 1.46 \text{ bar}$$

isentropic process 2'-3,

$$\frac{T_3}{T_{2'}} = \left(\frac{p_3}{p_{2'}} \right)^{\frac{\gamma-1}{\gamma}} = (4)^{\frac{1.4-1}{1.4}} = (4)^{0.286} = 1.486$$

$$T_3 = T_{2'} \times 1.486 = 358.5 \times 1.486 = 532.7 \text{ K}$$

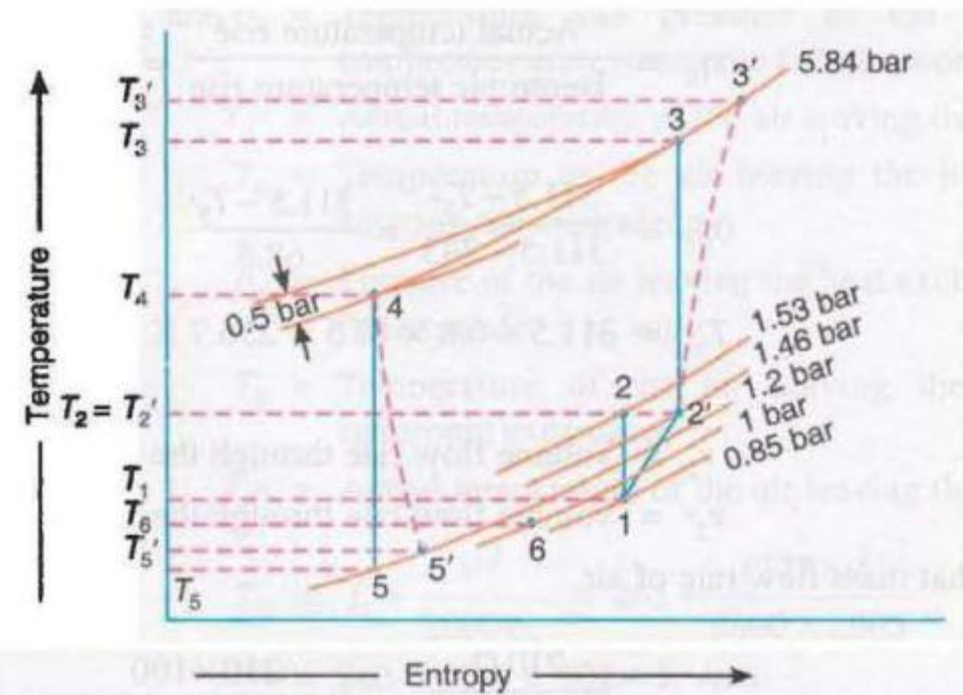


isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic temperature rise}}{\text{Actual temperature rise}} = \frac{T_3 - T_2'}{T_{3'} - T_2'}$$

$$0.8 = \frac{532.7 - 358.5}{T_{3'} - 358.5} = \frac{174.2}{T_{3'} - 358.5}$$

$$\therefore T_{3'} = \frac{174.2}{0.8} + 358.5 = 576 \text{ K}$$



Since the pressure ratio of the main compressor (p_3/p_2') is 4, therefore pressure of air leaving the main compressor,

$$p_3 = p_3' = 4 p_2' = 4 \times 1.46 = 5.84 \text{ bar}$$

Pressure drop in the heat exchanger

$$= 0.5 \text{ bar}$$

\therefore Pressure of air after passing through the heat exchanger or at entrance to the cooling turbine,

$$p_4 = p_3' - 0.5 = 5.84 - 0.5 = 5.34 \text{ bar}$$

Also there is a pressure loss of 0.2 bar between the cooling turbine and the cock pit.

Therefore pressure of air leaving the cooling turbine,

$$p_5 = p_5' = p_6 + 0.2 = 1 + 0.2 = 1.2 \text{ bar}$$

the isentropic process 4-5,

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5.34}{1.2} \right)^{\frac{1.4-1}{1.4}} = (4.45)^{0.286} = 1.53$$

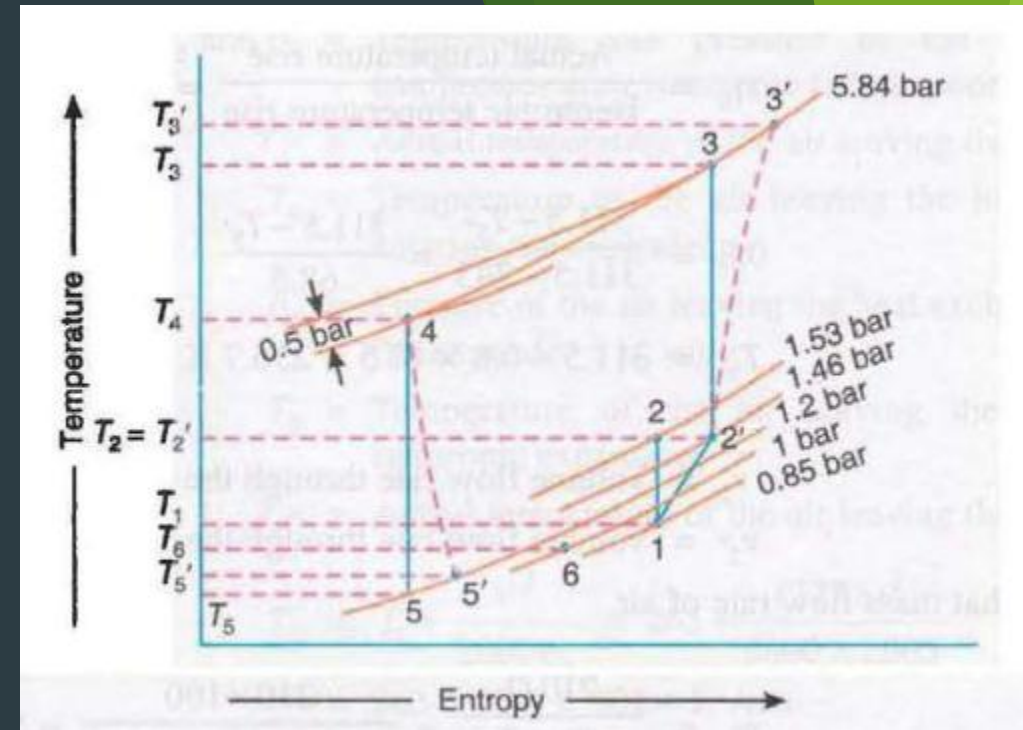
$$T_5 = T_4 / 1.53 = 333 / 1.53 = 217.6 \text{ K}$$

isentropic efficiency of the cooling turbine,

$$\eta_T = \frac{\text{Actual temperature rise}}{\text{Isentropic temperature rise}} = \frac{T_4 - T_{5'}}{T_4 - T_5}$$

$$0.8 = \frac{333 - T_{5'}}{333 - 217.6} = \frac{333 - T_{5'}}{115.4}$$

$$T_{5'} = 333 - 0.8 \times 115.4 = 240.7 \text{ K}$$



$$m_a = \frac{3.516 \times \text{tonnes}}{c_p \times (T_6 - T_{5'})}$$

$$m_a = \frac{3.516 \times 10}{1 \times (298 - 240.7)} = 0.61 \text{ kg/s}$$

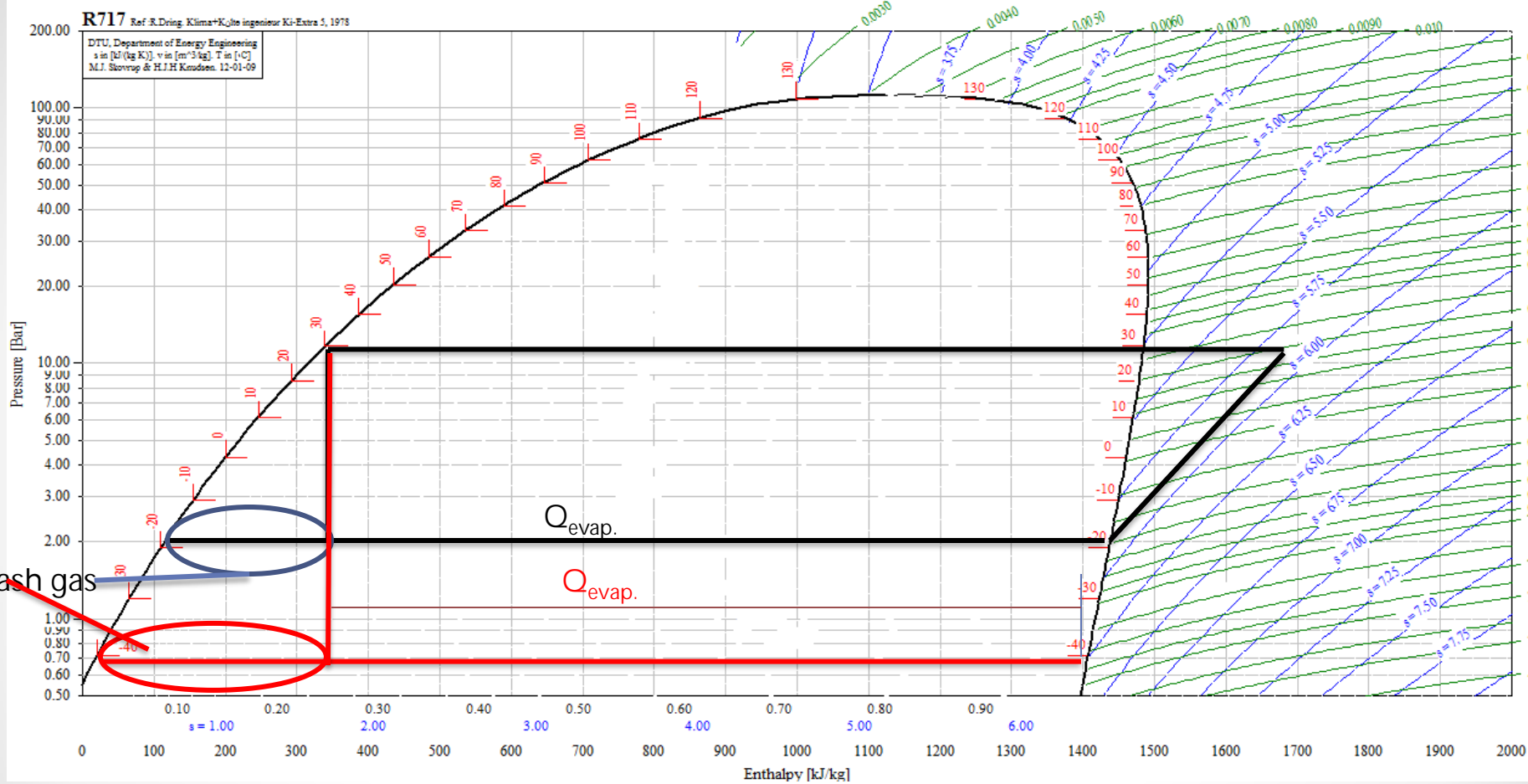
$$COP = \frac{3.516 \times \text{tonnes}}{m_a \times c_p \times (T_{3'} - T_{2'})} = \frac{3.516 \times 10}{0.61 \times 1 \times (576 - 358.5)} = 0.264$$

- Compound Vapour Compression System

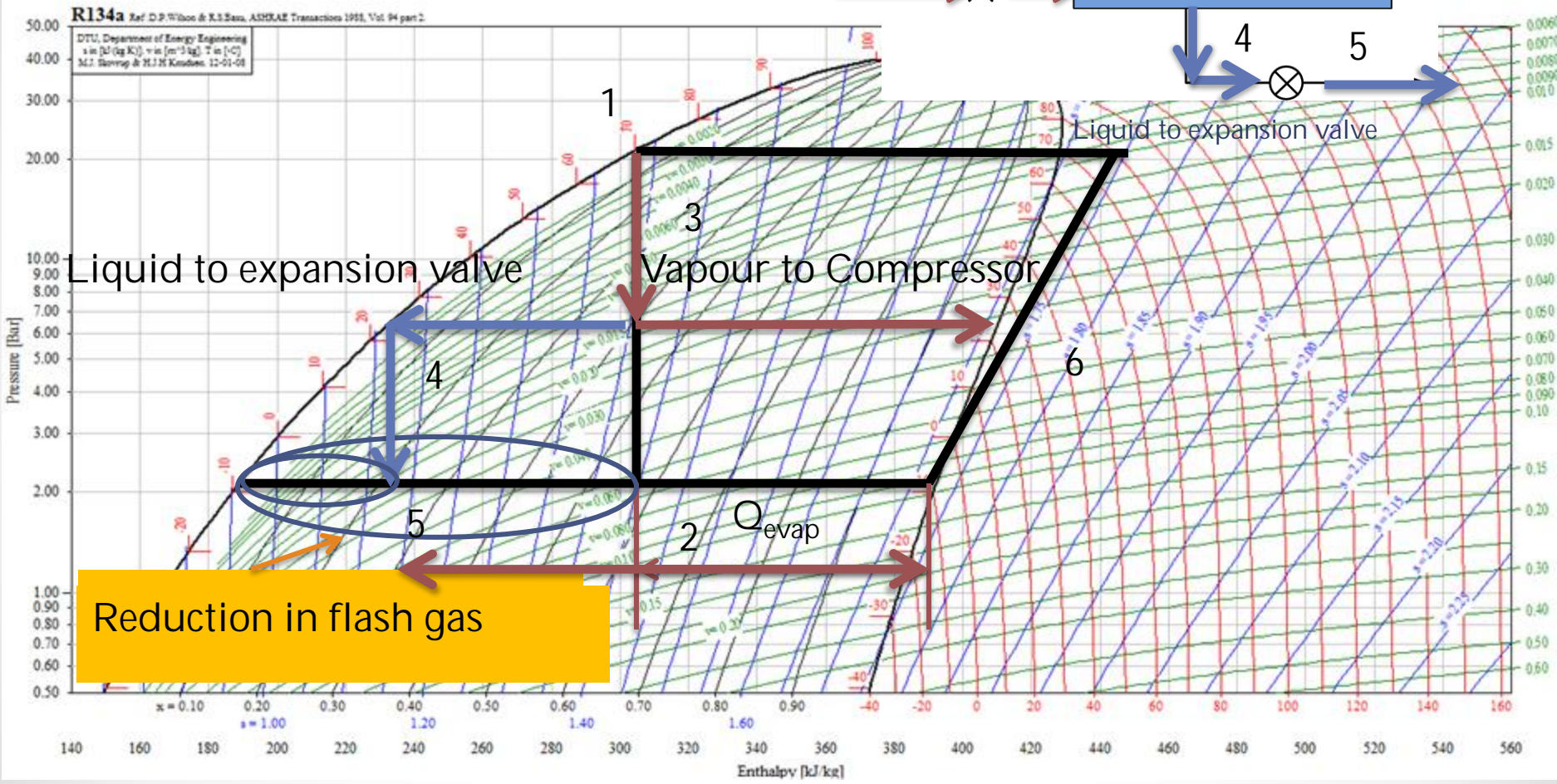
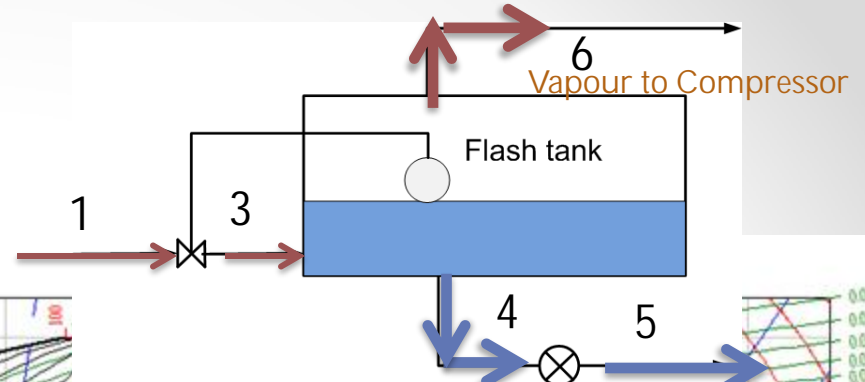
- Removing Of Flash Gas

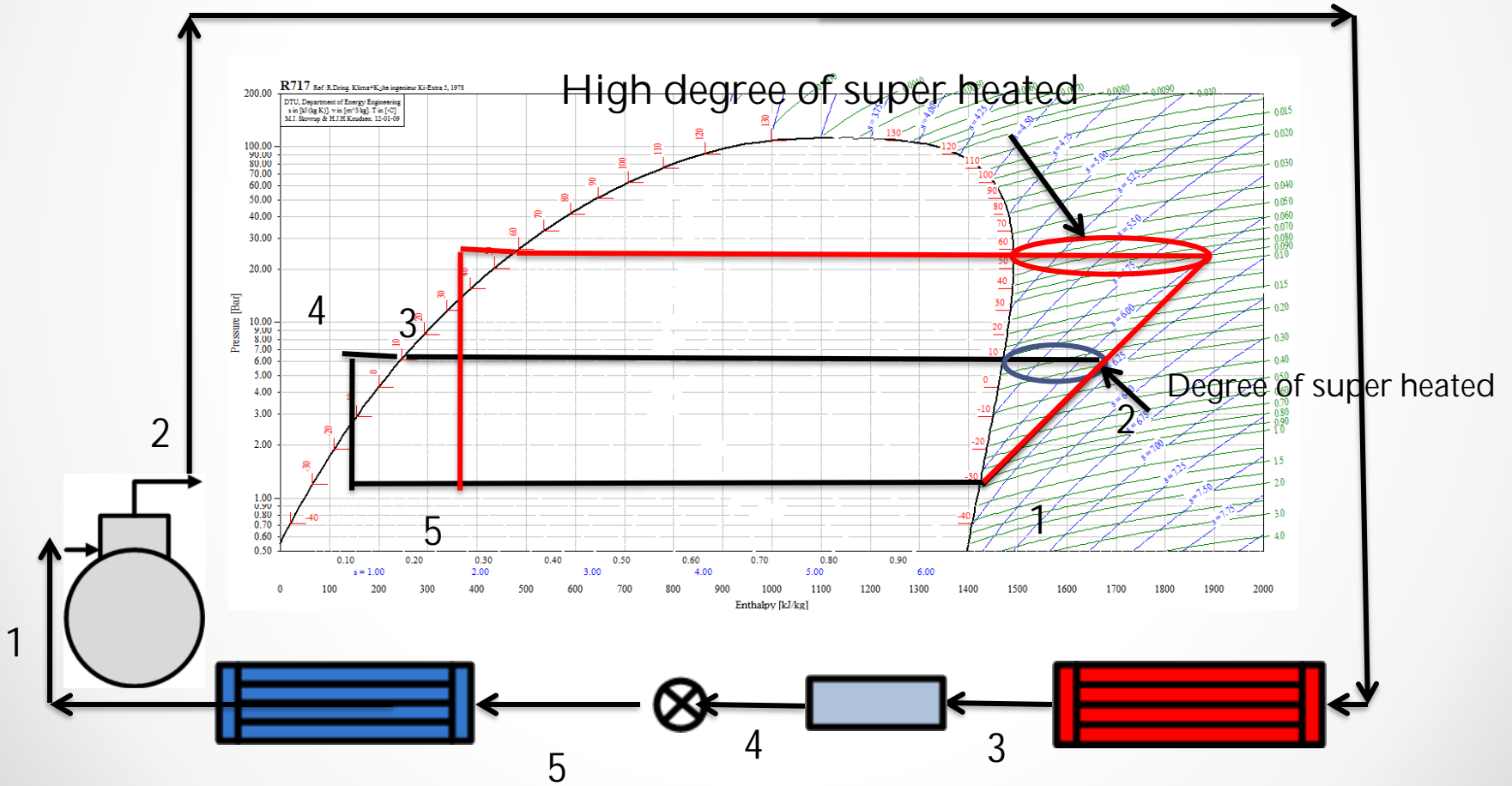
R717 Ref. R.Dring Klima+Kälte ingenieur KI-Extra 5, 1978

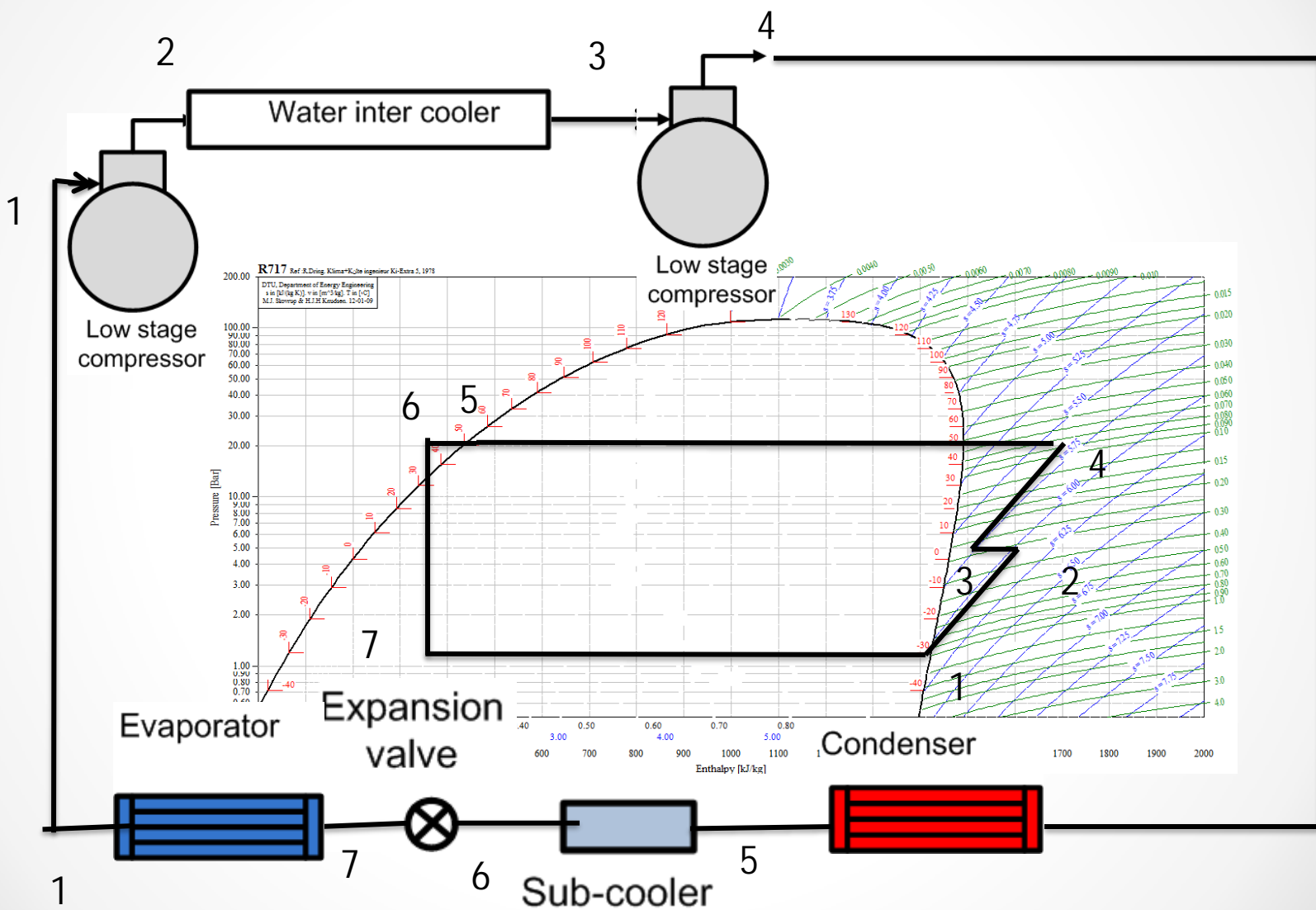
DTU, Department of Energy Engineering
 s in [kJ/(kg K)], v in [m³/kg], T in [°C]
 M.J. Skovrup & H.J.H. Knudsen, 12-01-09



Flash gas





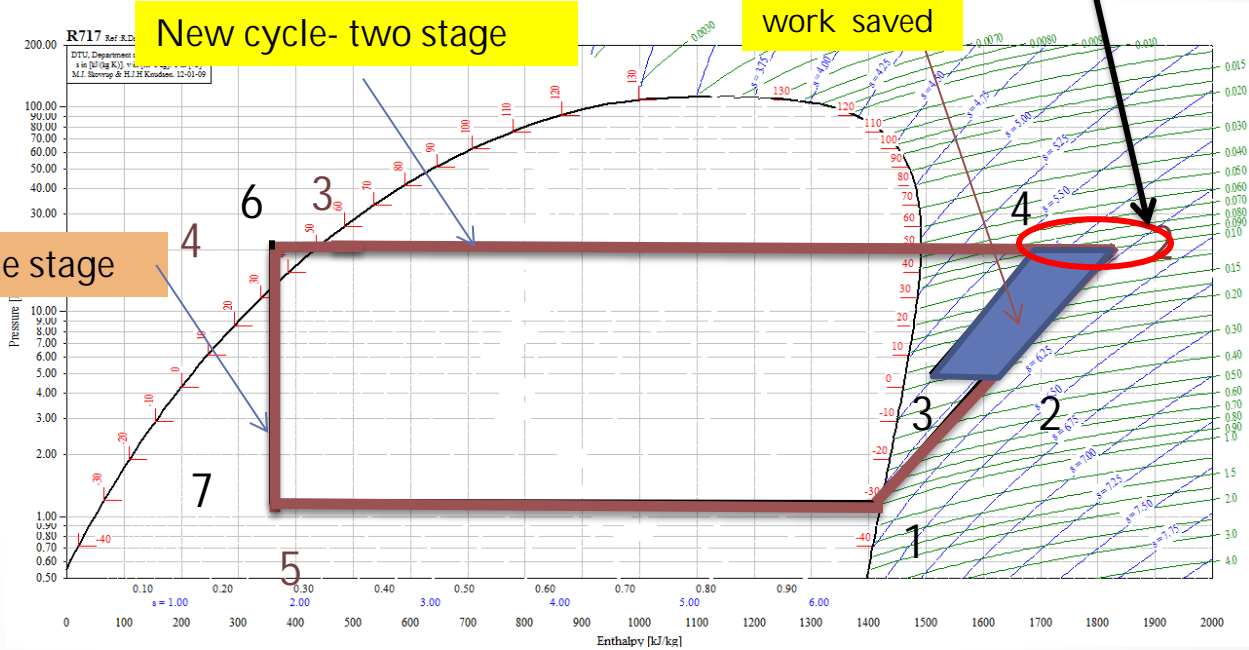


Old cycle- single stage

New cycle- two stage

work saved

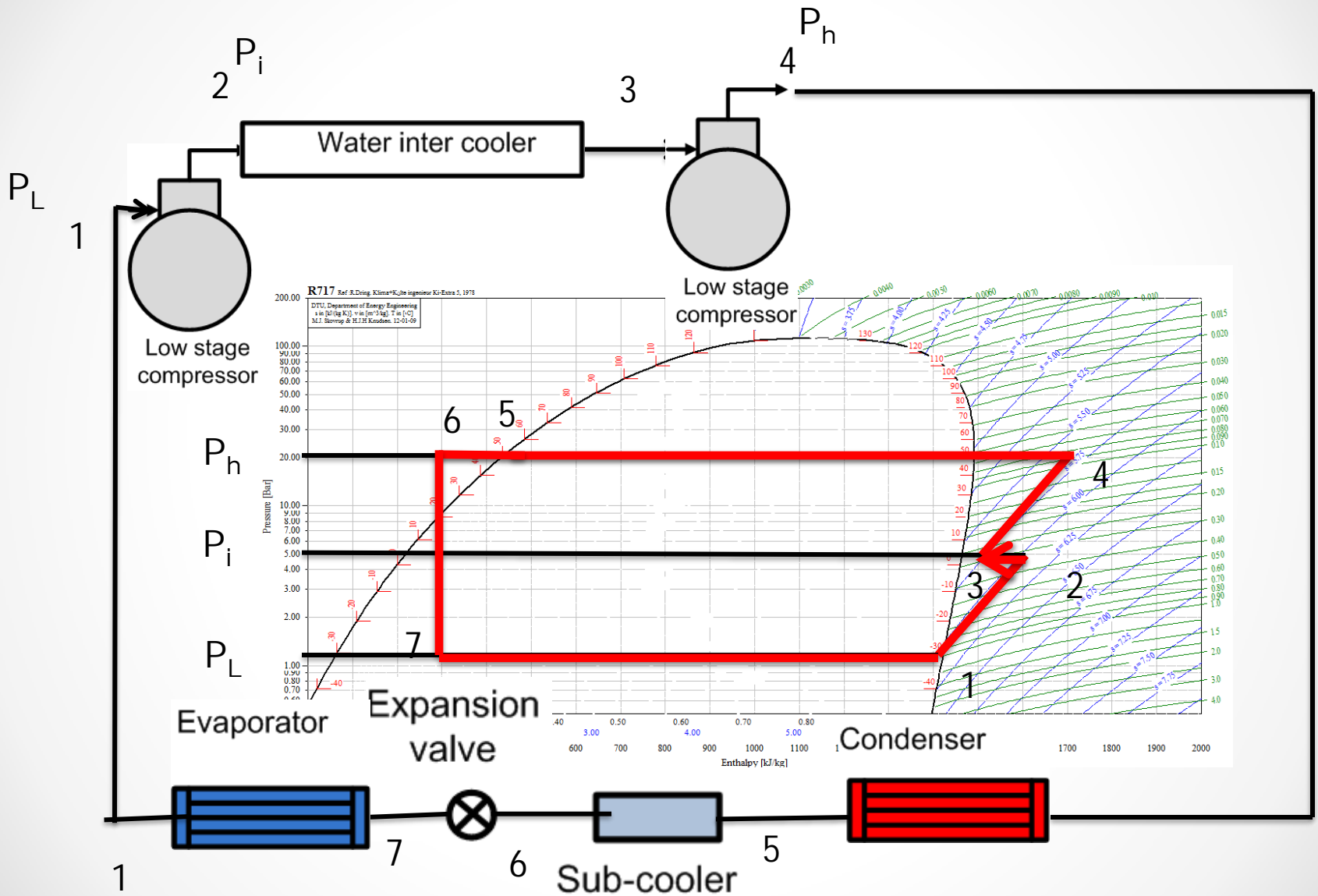
Reduction in super heating



Types of intercoolers

- There are three types of intercoolers as mentioned below.
- 1-water Inter-cooler
- 2-Liquid Refrigerant Inter-cooler
- 3-Flash Gas Inter-cooler

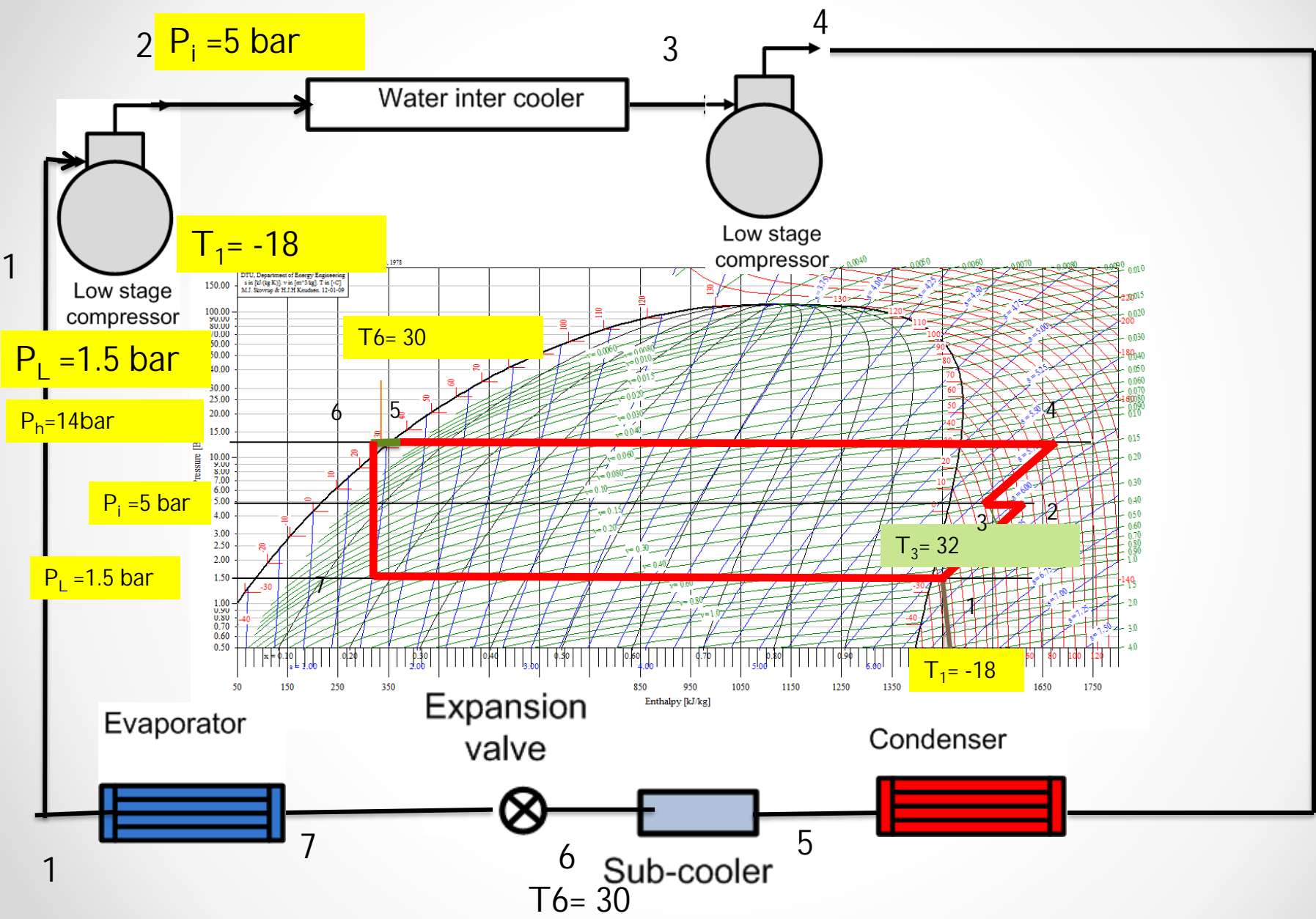
Compound compression with water inter-cooler



Example 1:

- The following data apply to
- 100 ton compound ammonia compression system with water inter-cooler,
- condenser pressure 14 bar,
- evaporator pressure 1.5 bar
- and inter-cooler pressure 5 bar,
- the ammonia is cooled to 32°C in the water inter-cooler
- and sub-cooled to 30 °C in the sub-cooler,.
- The temperature of suction gas is -18 °C;
- find a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d- displacement volume of in both low and high stage and low stage compressor

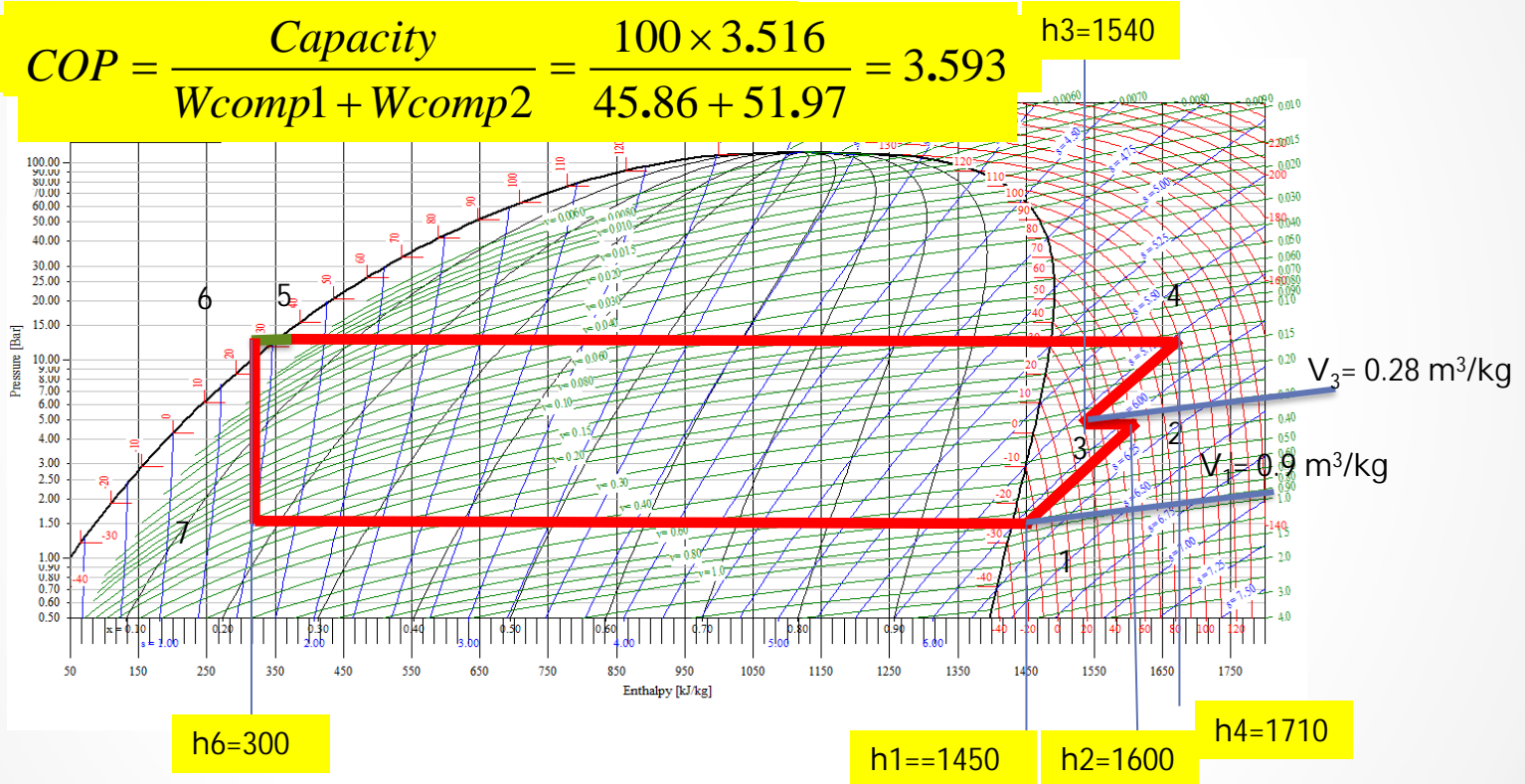
enter and sub-cooled to 30°C in the sub-cooler inter-cooler bar



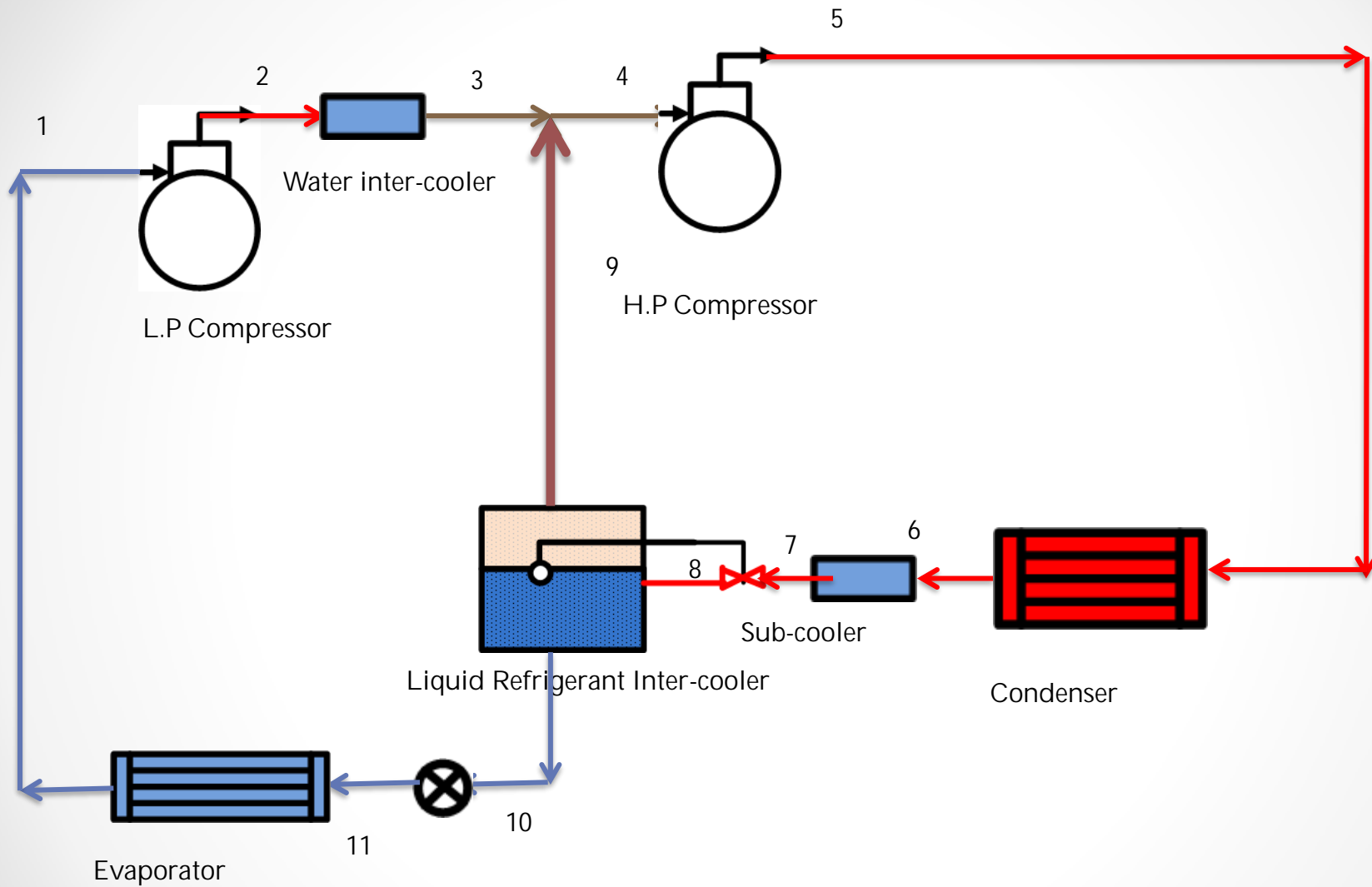
Comp. power L.P.=0.307x(1600-1450)=45.86 kW

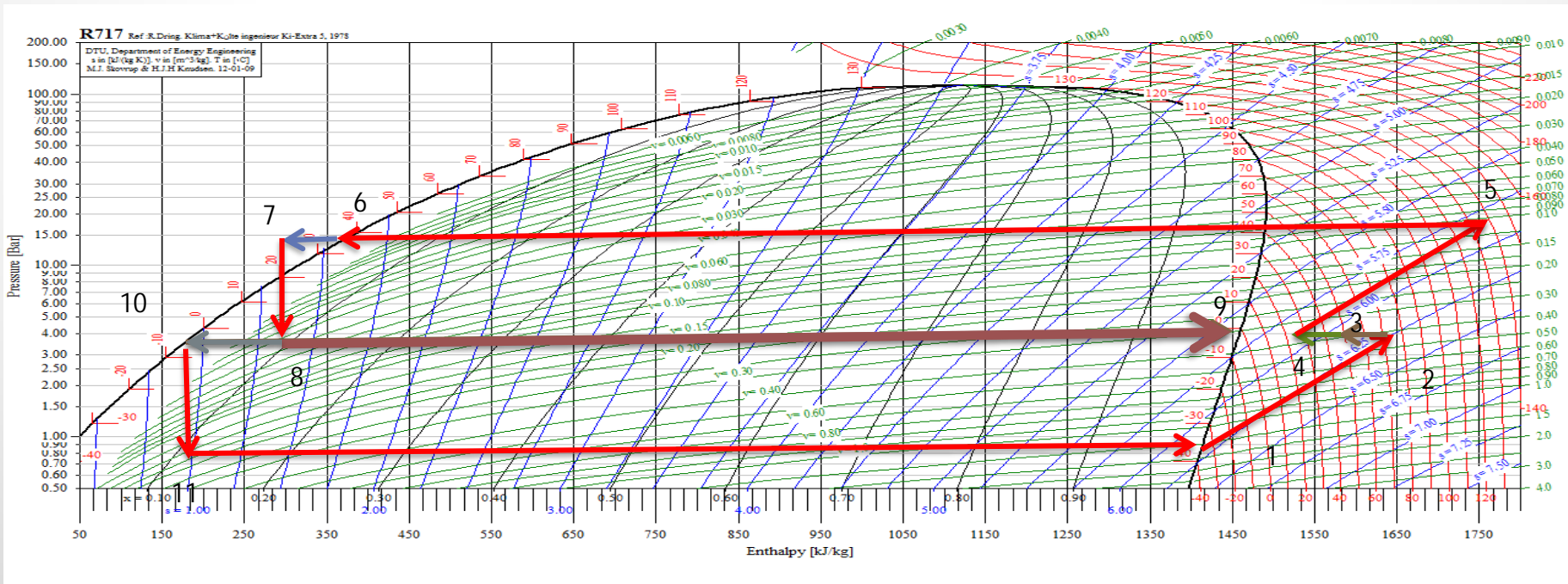
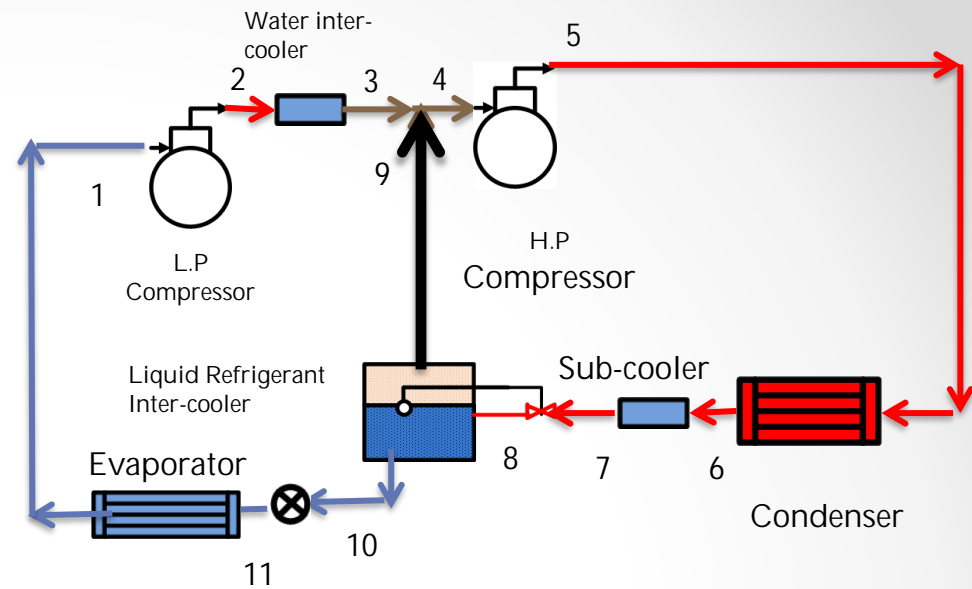
h_1	h_2	h_3	h_4	h_6
1450	1600	1540	1710	300

$$COP = \frac{Capacity}{W_{comp1} + W_{comp2}} = \frac{100 \times 3.516}{45.86 + 51.97} = 3.593$$



- 2-Liquid Refrigerant Inter-cooler

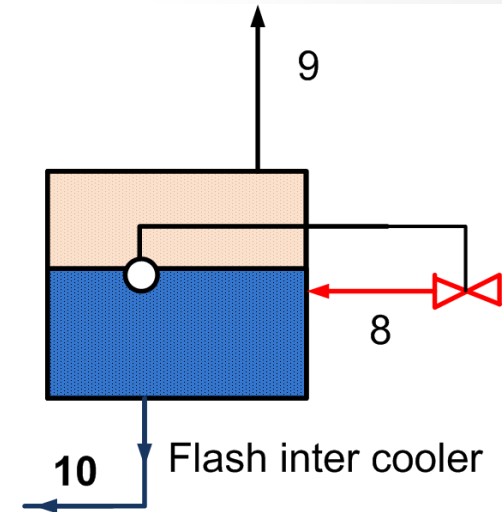




Heat balance of flash inter-cooler

- $m_8 h_8 = m_{10} h_{10} + m_9 h_9$
- $m_8 = m_{10} + m_9$
- $m_9 = m_8 - m_{10}$
- $m_8 h_8 = m_{10} h_{10} + (m_8 - m_{10}) h_9$
- $m_8 (h_8 - h_9) = m_{10} (h_{10} - h_9)$

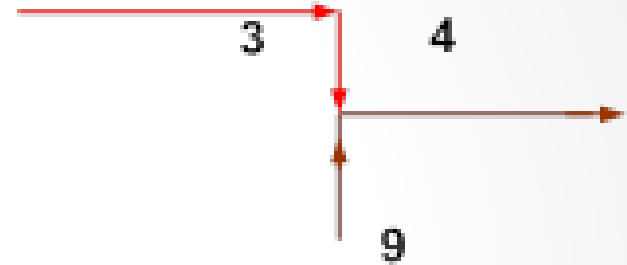
$$m_8 = \frac{m_{10}(h_{10} - h_9)}{h_8 - h_9}$$



Heat balance of mixing process

- Heat balance
- $m_4 \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$
- mass balance
- $m_4 = m_3 + m_9$
- $(m_3 + m_9) \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$

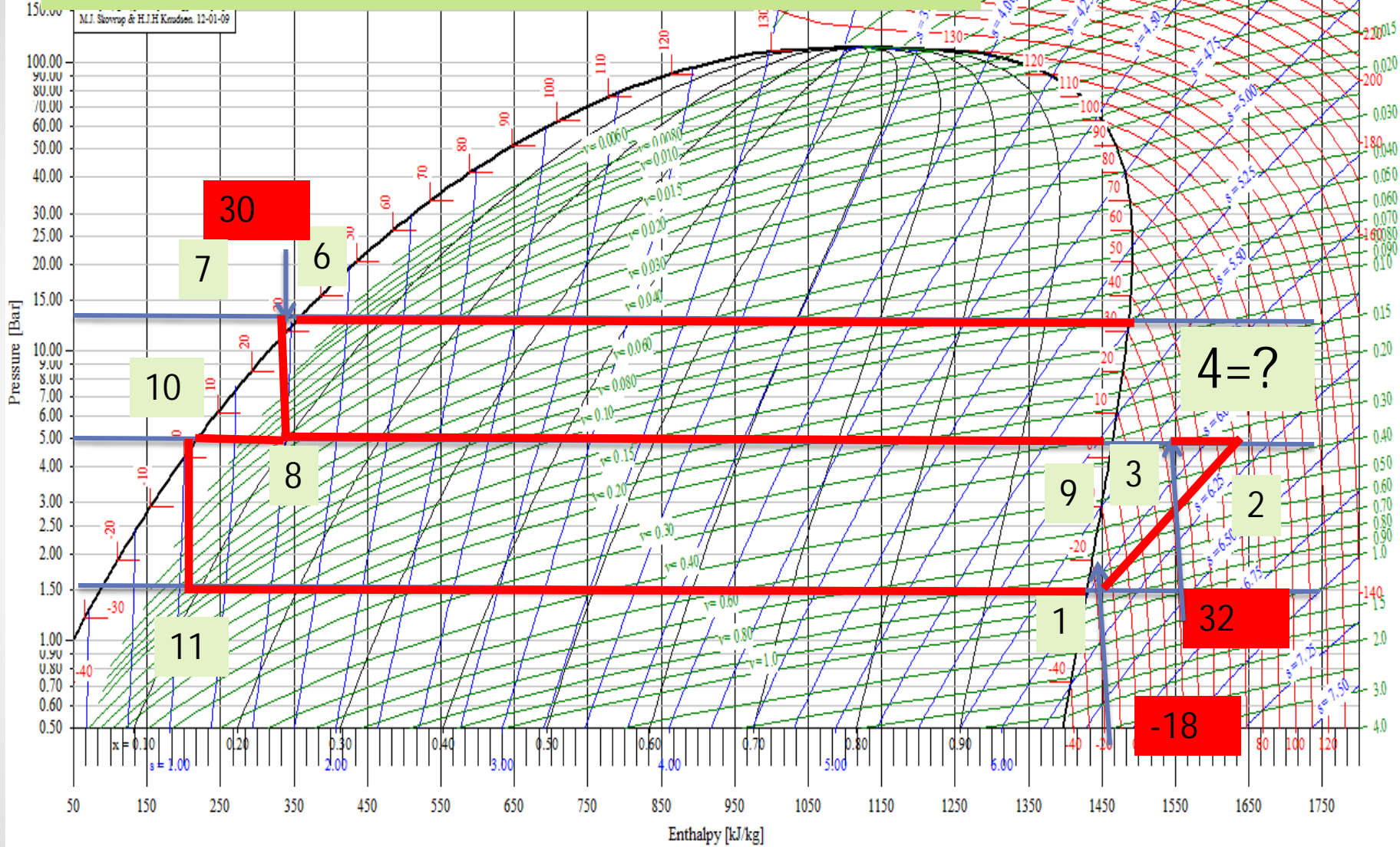
$$h_4 = \frac{m_3 h_3 + m_9 h_9}{m_3 + m_9}$$



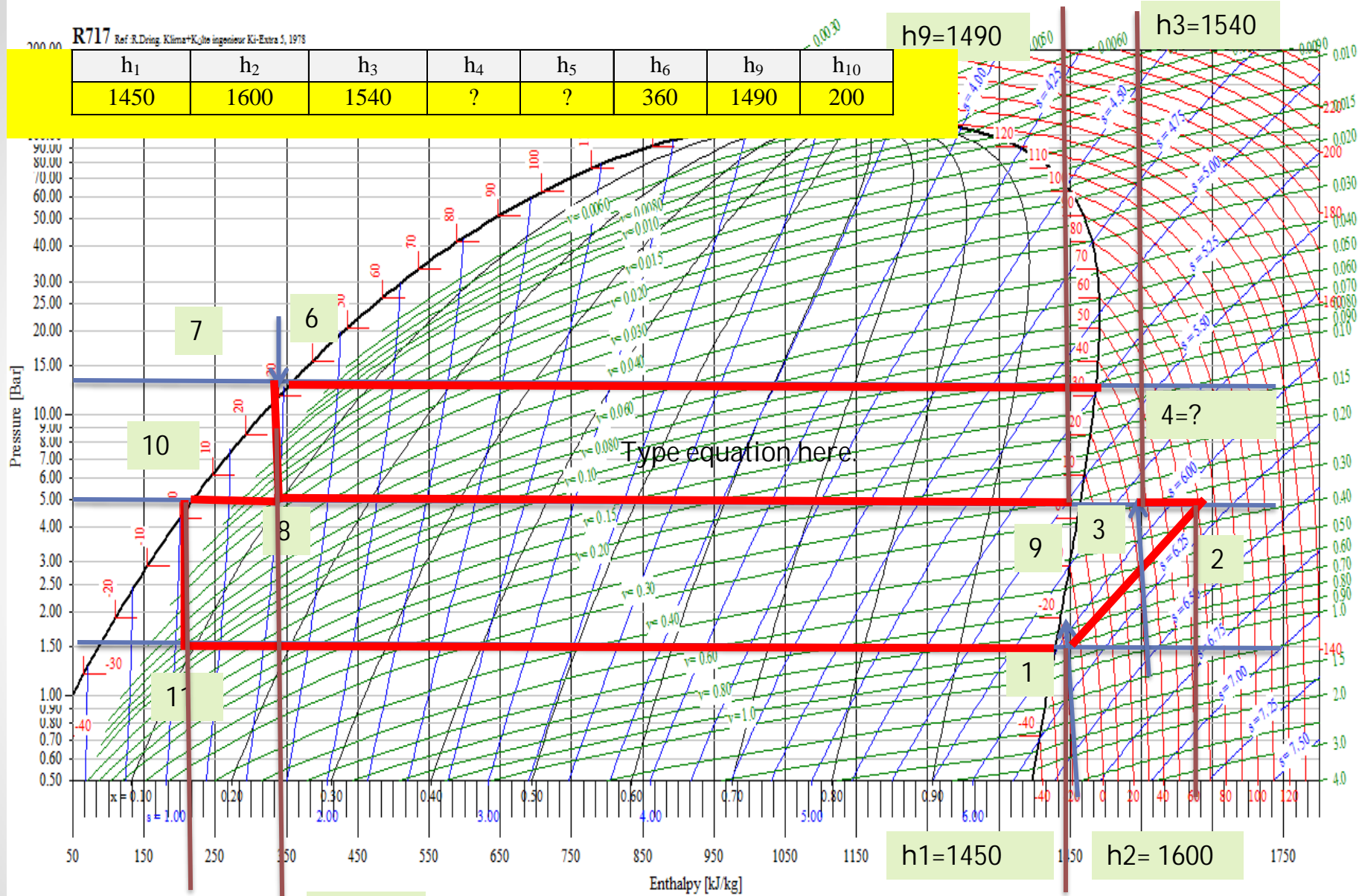
Example 2:

- The following data apply to
- 100 ton compound ammonia compression system with
- water inter-cooler
- and liquid refrigerant inter-cooler,
- condenser pressure 14 bar,
- evaporator pressure 1.5 bar
- and inter-cooler pressure 5 bar,
- the ammonia is cooled to 32°C in the water inter-cooler
- and sub-cooled to 30 °C in the sub-cooler,.
- The temperature of suction gas is -18 °C;
- find a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d-COP

and sub-cooled to 30 °C in the sub-cooler



h_1	h_2	h_3	h_4	h_5	h_6	h_9	h_{10}
1450	1600	1540	?	?	360	1490	200



$h_{10}=200$

$h_6=360$

$h_1=1450$

$h_2=1600$

$h_9=1490$

$h_3=1540$

h9=1490

h3=1540

	h ₁	h ₂	h ₃	h ₄	h ₅	h ₆	h ₉	h ₁₀
1.	1450	1600	1540	?	?	360	1490	200

$$\text{Capacity} = m \cdot (h_1 - h_{11})$$

$$\text{Capacity} = 100 \times 3.516 \text{ kW}$$

$$m_f = \frac{100 \times 3.516}{(1450 - 200)} = 0.28128 = m_{11}$$

$$m_8 h_8 = m_{10} h_{10} + (m_8 - m_{10}) h_9$$

$$m_8 (h_8 - h_9) = m_{10} (h_{10} - h_9)$$

$$m_8 = \frac{m_{10} (h_{10} - h_9)}{h_8 - h_9}$$

Heat balance of flash inter-cooler

$$m_8 h_8 = m_{10} h_{10} + m_9 h_9$$

$$m_8 = \frac{0.281 \times (200 - 1490)}{360 - 1490} = 0.32 \text{ kg/s}$$

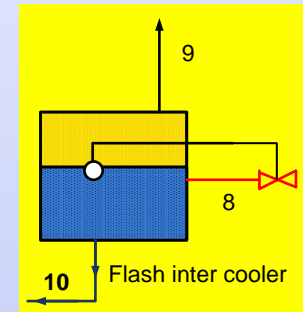
$$m_8 = m_{10} + m_9$$

$$m_8 = 0.32$$

$$m_9 = 0.039$$

$$m_9 = m_8 - m_{10} = 0.32 - 0.281 = 0.039 \text{ kg/s}$$

$$m_9 = m_8 - m_{10}$$



h1=1450

h2=1600

h6=360

h10=200

h ₁	h ₂	h ₃	h ₄	h ₅	h ₆	h ₉	h ₁₀
1450	1600	1540	?	?	360	1490	200

h₉=1490

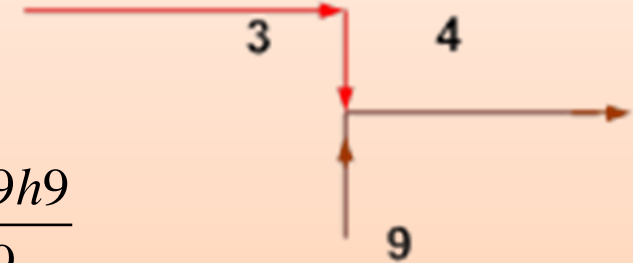
h₃=1540

Heat balance of mixing process

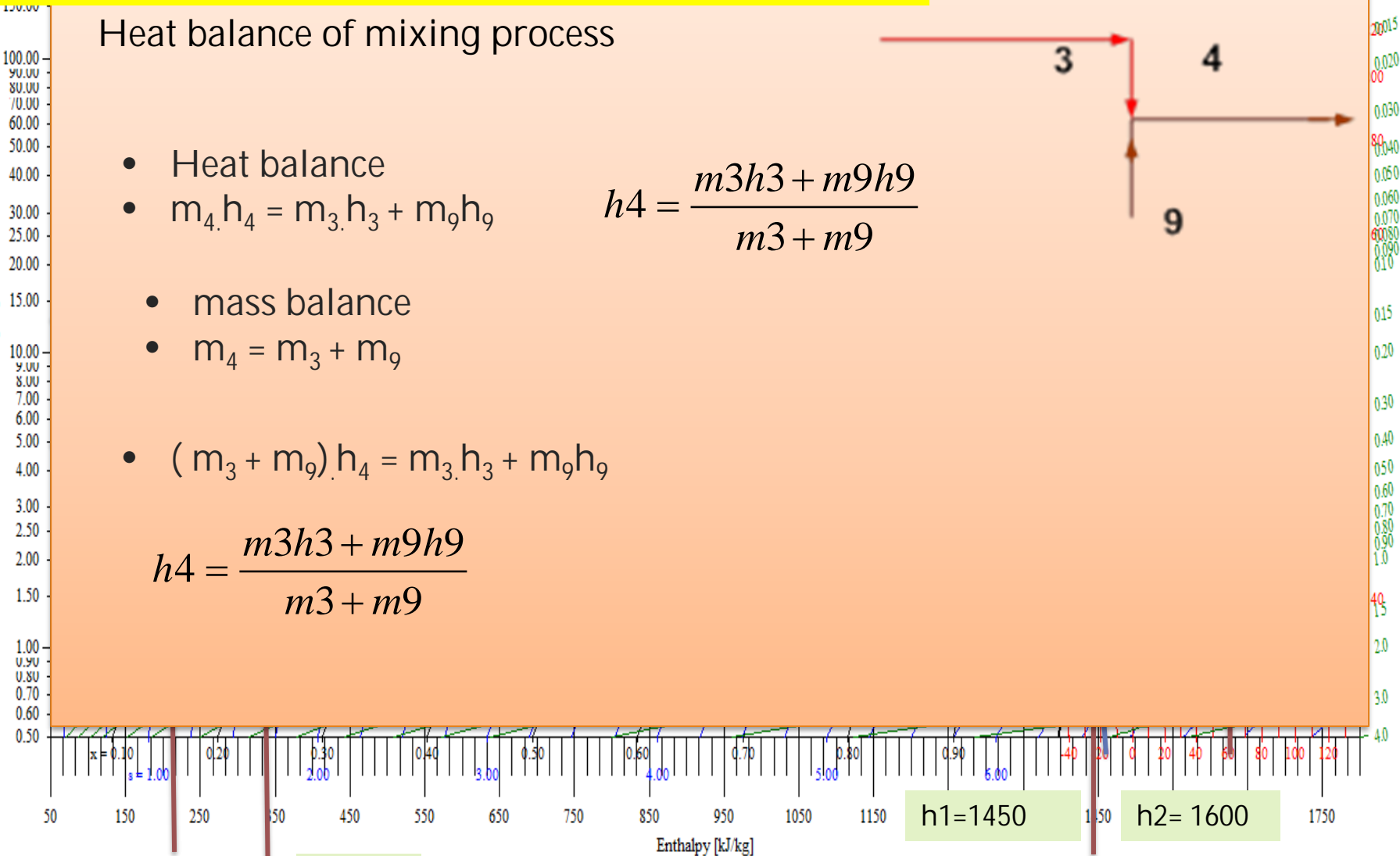
- Heat balance
- $m_4 \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$
- mass balance
- $m_4 = m_3 + m_9$
- $(m_3 + m_9) \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$

$$h_4 = \frac{m_3 h_3 + m_9 h_9}{m_3 + m_9}$$

$$h_4 = \frac{m_3 h_3 + m_9 h_9}{m_3 + m_9}$$



Pressure [Bar]

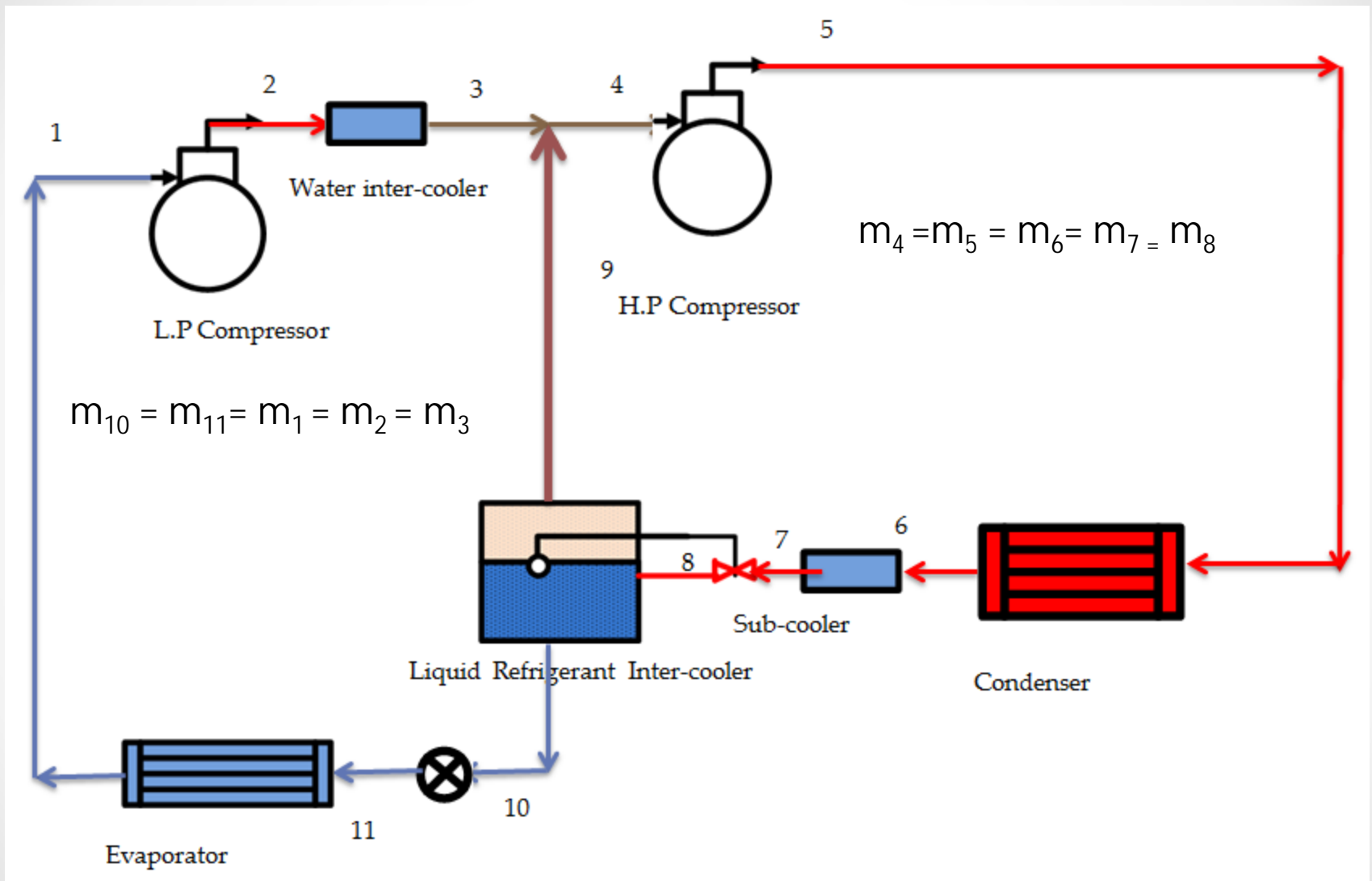


h₁=1450

h₂=1600

h₇=360

h₁₀=200



h_1	h_2	h_3	h_4	h_5	h_6	h_9	h_{10}
1450	1600	1540	?	?	360	1490	200

$h_9=1490$

$h_3=1540$

Heat balance of mixing process

- Heat balance
- $m_4 \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$
- mass balance
- $m_4 = m_3 + m_9$
- $(m_3 + m_9) \cdot h_4 = m_3 \cdot h_3 + m_9 \cdot h_9$

$$h_4 = \frac{m_3 h_3 + m_9 h_9}{m_3 + m_9}$$

$$m_8 = 0.32$$

$$m_9 = 0.039$$

$$m_{11} = 0.281$$

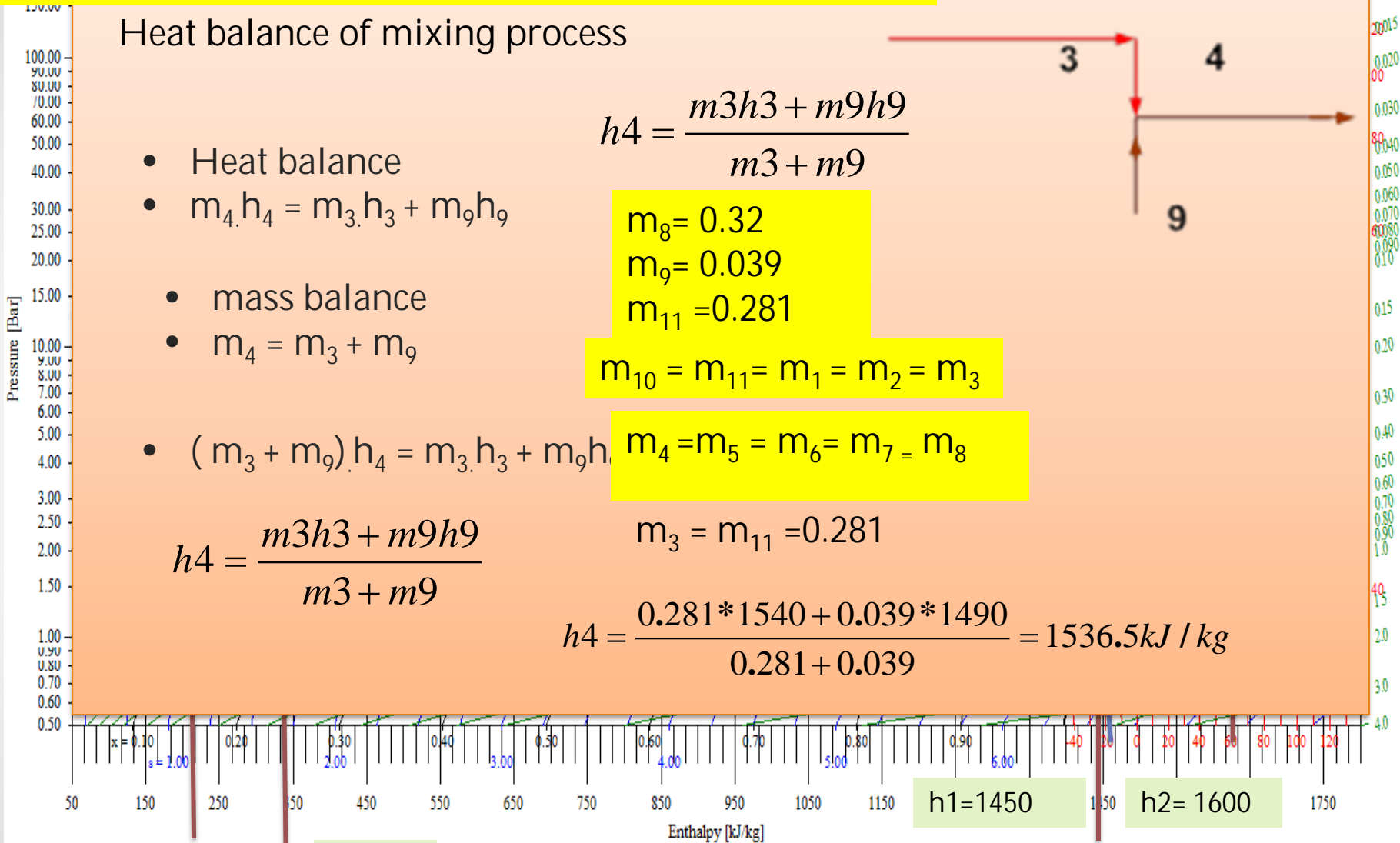
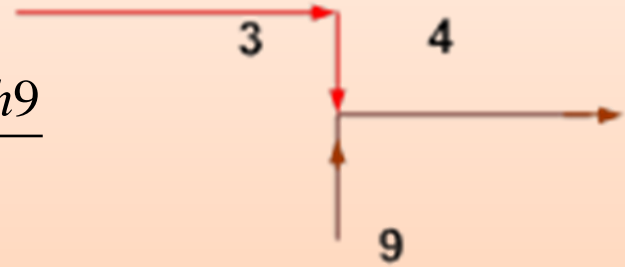
$$m_{10} = m_{11} = m_1 = m_2 = m_3$$

$$m_4 = m_5 = m_6 = m_7 = m_8$$

$$m_3 = m_{11} = 0.281$$

$$h_4 = \frac{m_3 h_3 + m_9 h_9}{m_3 + m_9}$$

$$h_4 = \frac{0.281 \cdot 1540 + 0.039 \cdot 1490}{0.281 + 0.039} = 1536.5 \text{ kJ/kg}$$



$h_1=1450$

$h_2=1600$

$h_7=360$

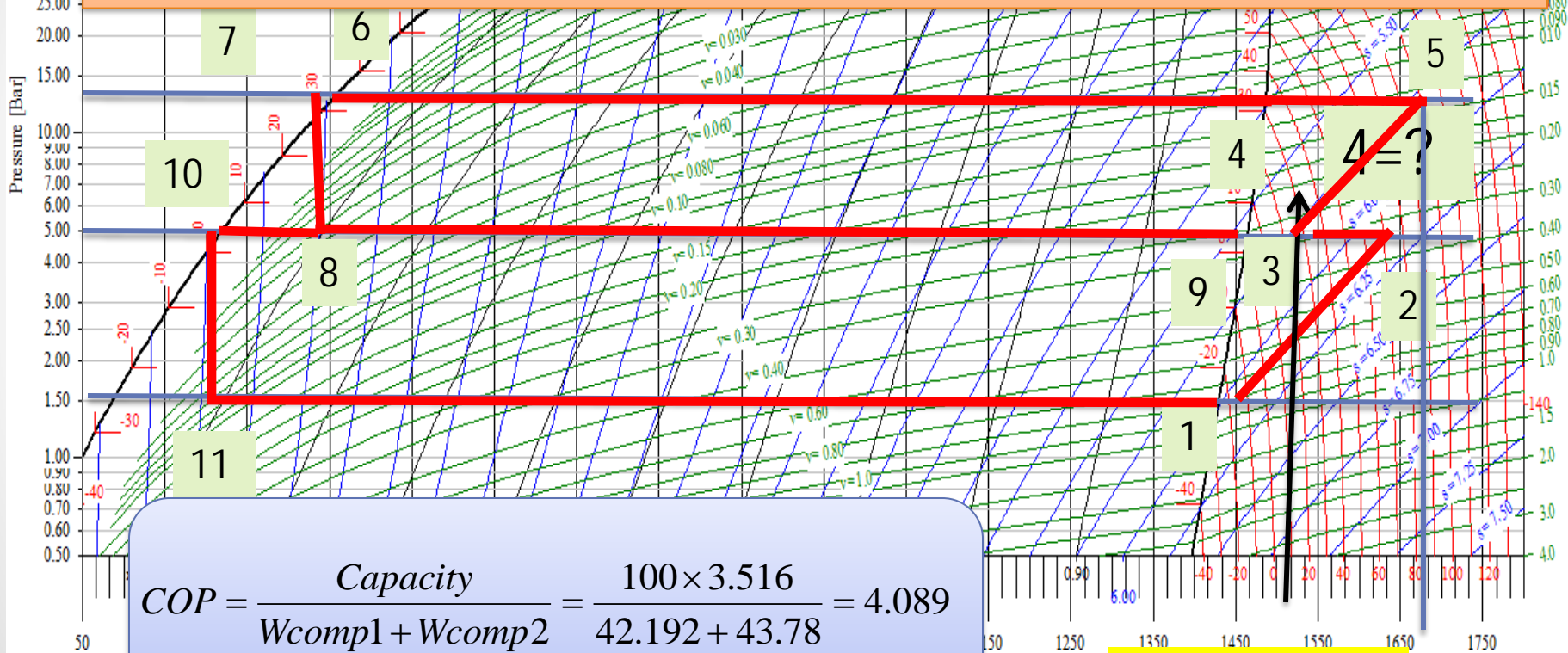
$h_{10}=200$

h_1	h_2	h_3	h_4	h_5	h_6	h_9	h_{10}
1450	1600	1540	1536.5	1680	360	1490	200

Comp. power L.P.= $m_{f1} \cdot (h_2 - h_1) = 0.28128(1600 - 1450) = 42.192 \text{ k w}$

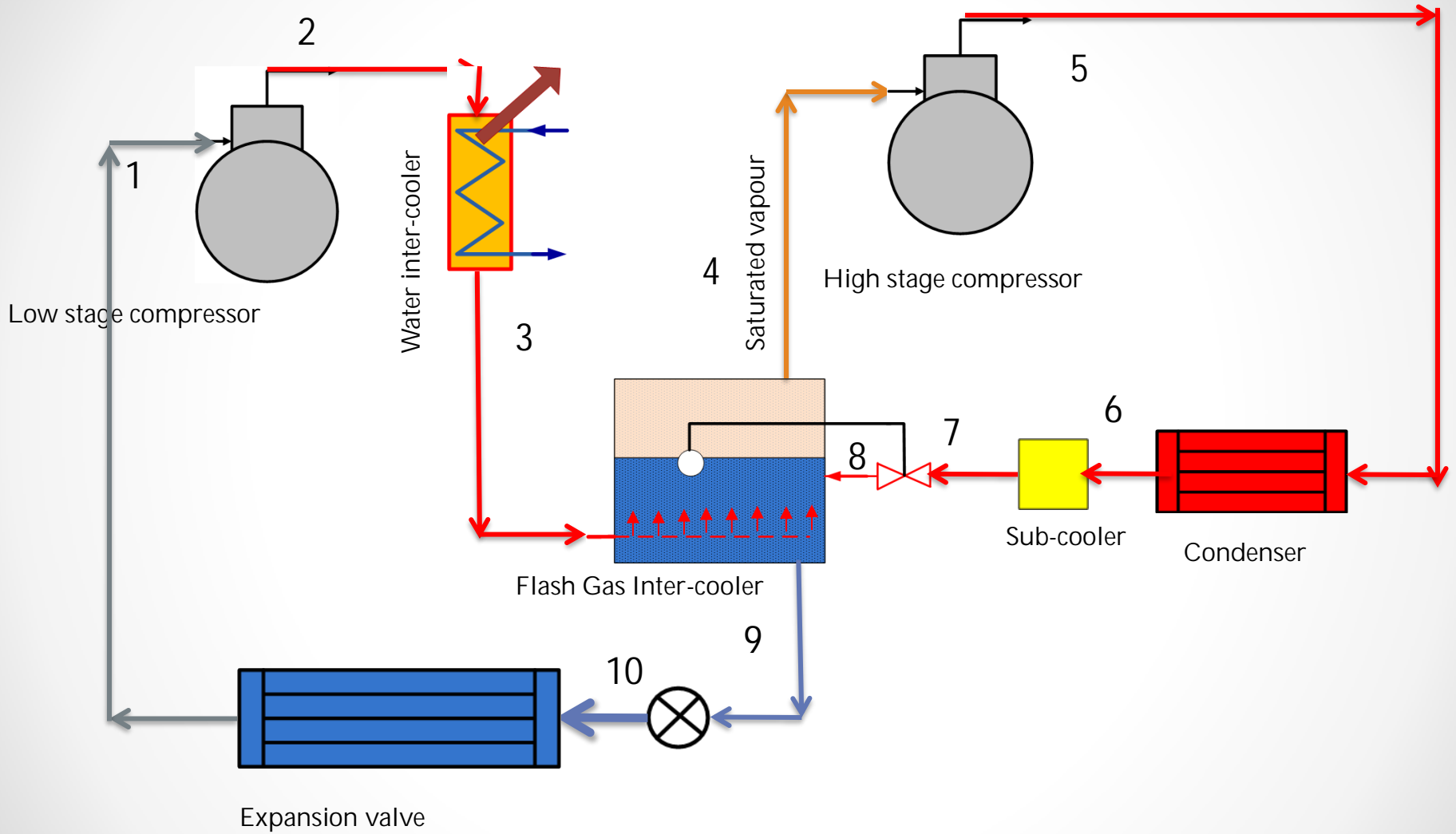
Comp. power H.P.= $m_{f4} \cdot (h_5 - h_4) = 0.3049 \times (1680 - 1536.5) = 43.78 \text{ kW}$

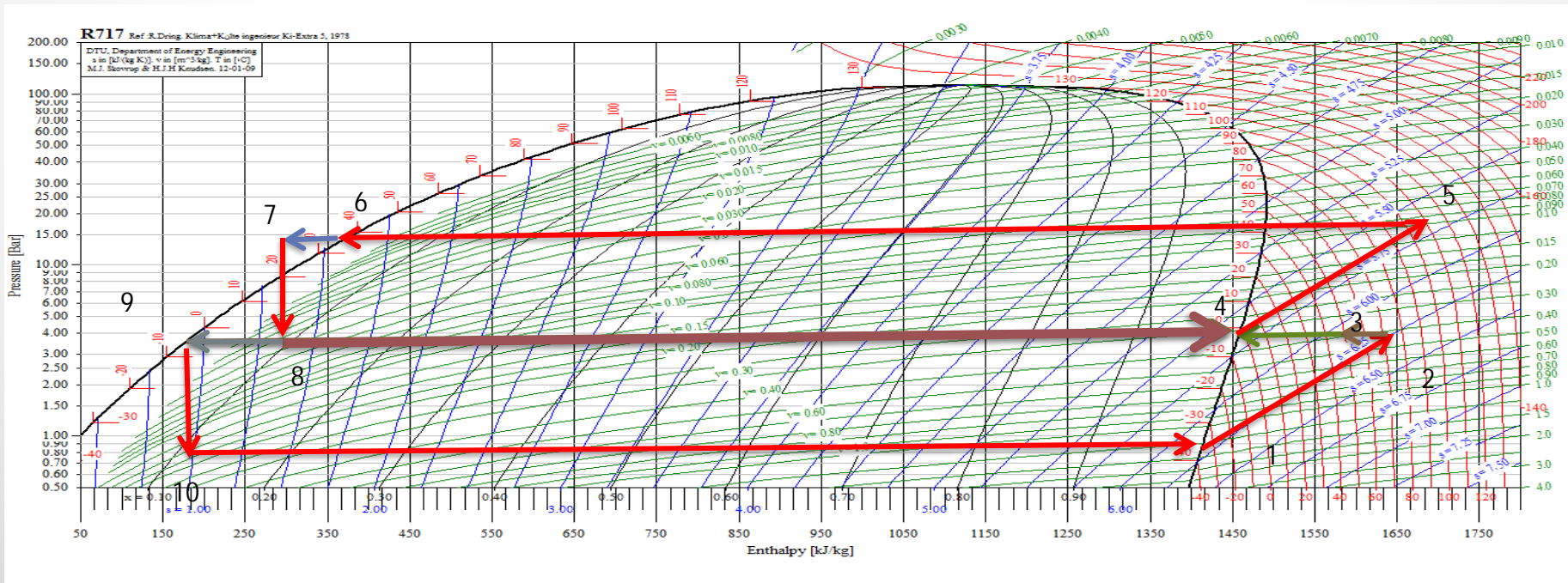
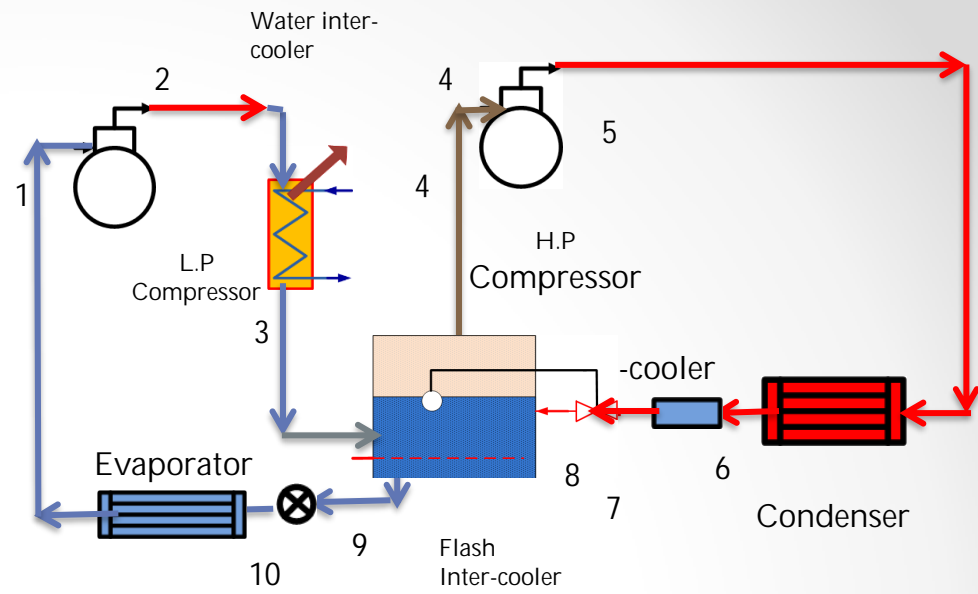
$Q_{\text{inter-cooler}} = m_{f1} \cdot (h_2 - h_3) = 0.28128 \times (1600 - 1540) = 16.87 \text{ kW}$

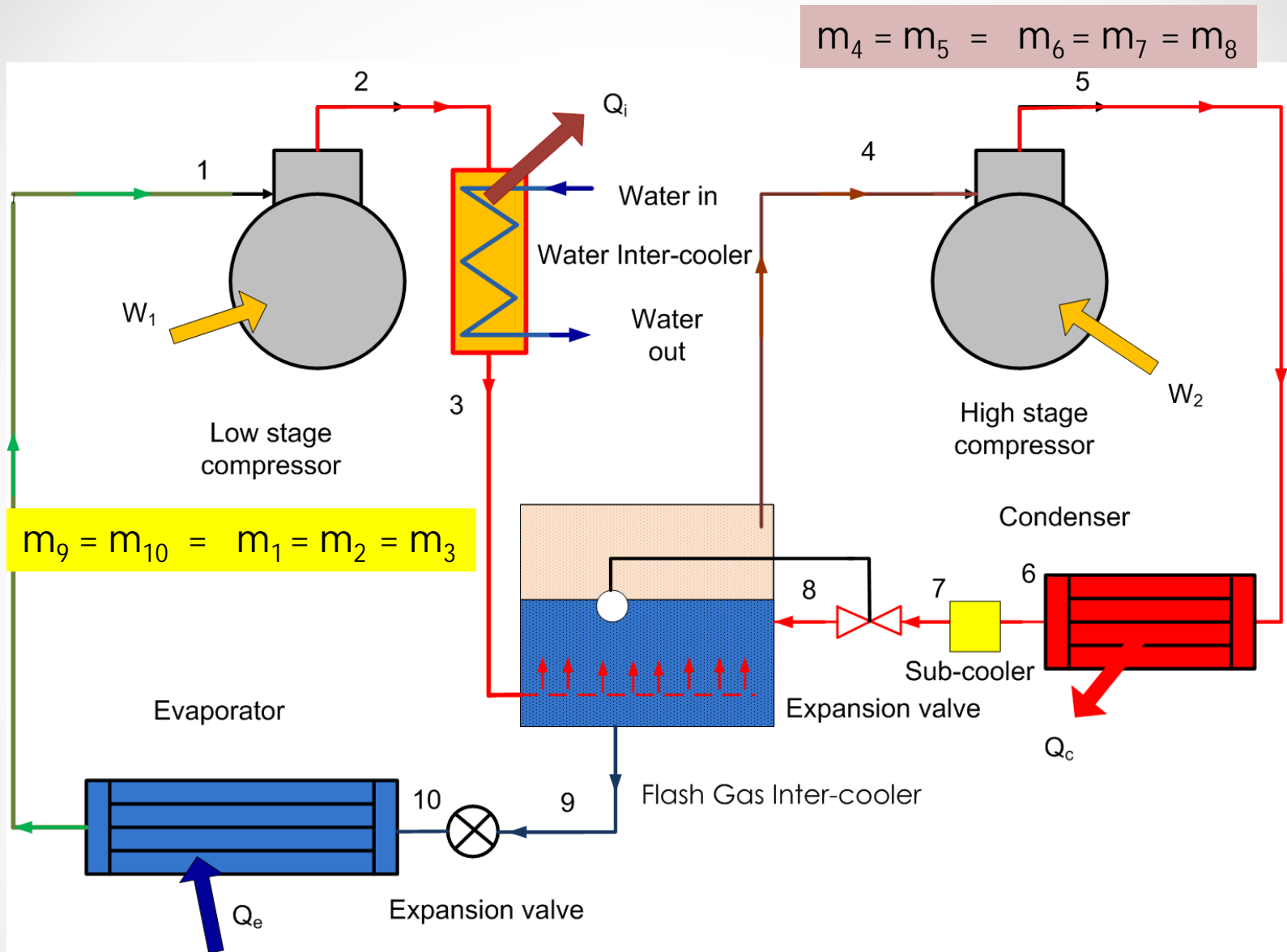


$h_5 = 1680 \text{ kJ/kg g}$

- 3-Flash Gas Inter-cooler







Heat balance inter-cooler

$$m_8 \cdot h_8 + m_3 \cdot h_3 = m_4 \cdot h_4 + m_9 \cdot h_9$$

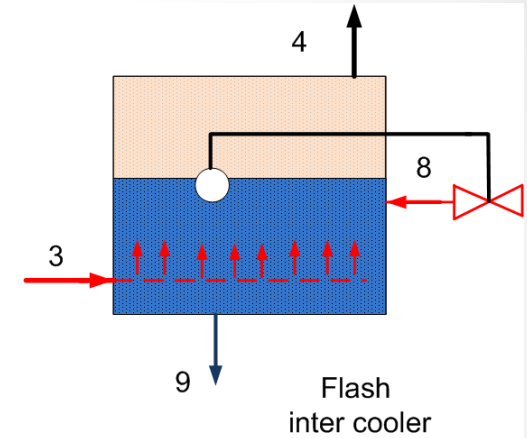
$$m_9 = m_{10} = m_1 = m_2 = m_3$$

$$m_4 = m_5 = m_6 = m_7 = m_8$$

$$m_4 \cdot h_8 + m_3 \cdot h_3 = m_4 \cdot h_4 + m_3 \cdot h_9$$

$$m_4 \cdot (h_8 - h_4) = m_3 \cdot (h_9 - h_3)$$

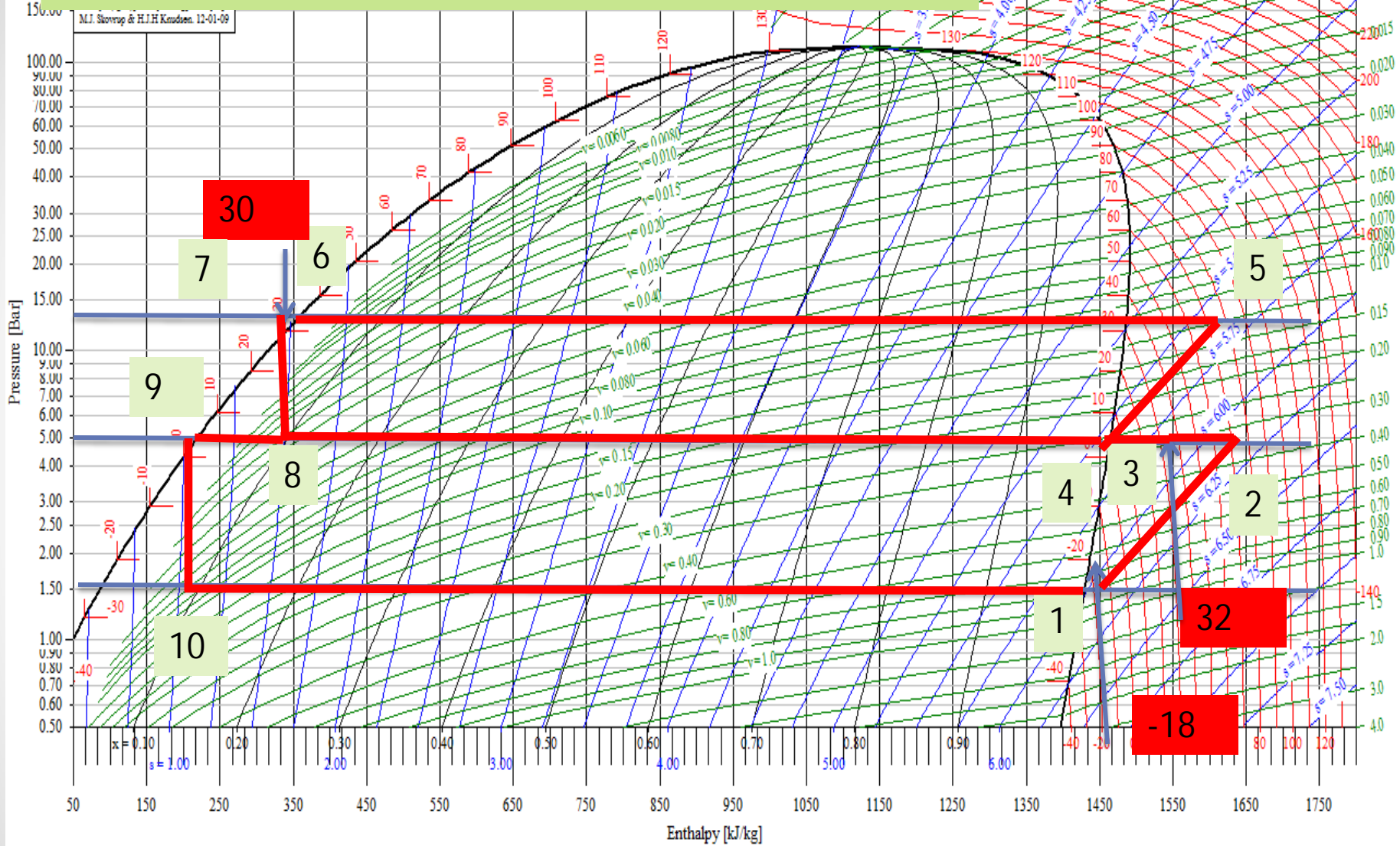
$$m_4 = \frac{m_3(h_9 - h_3)}{h_8 - h_4}$$



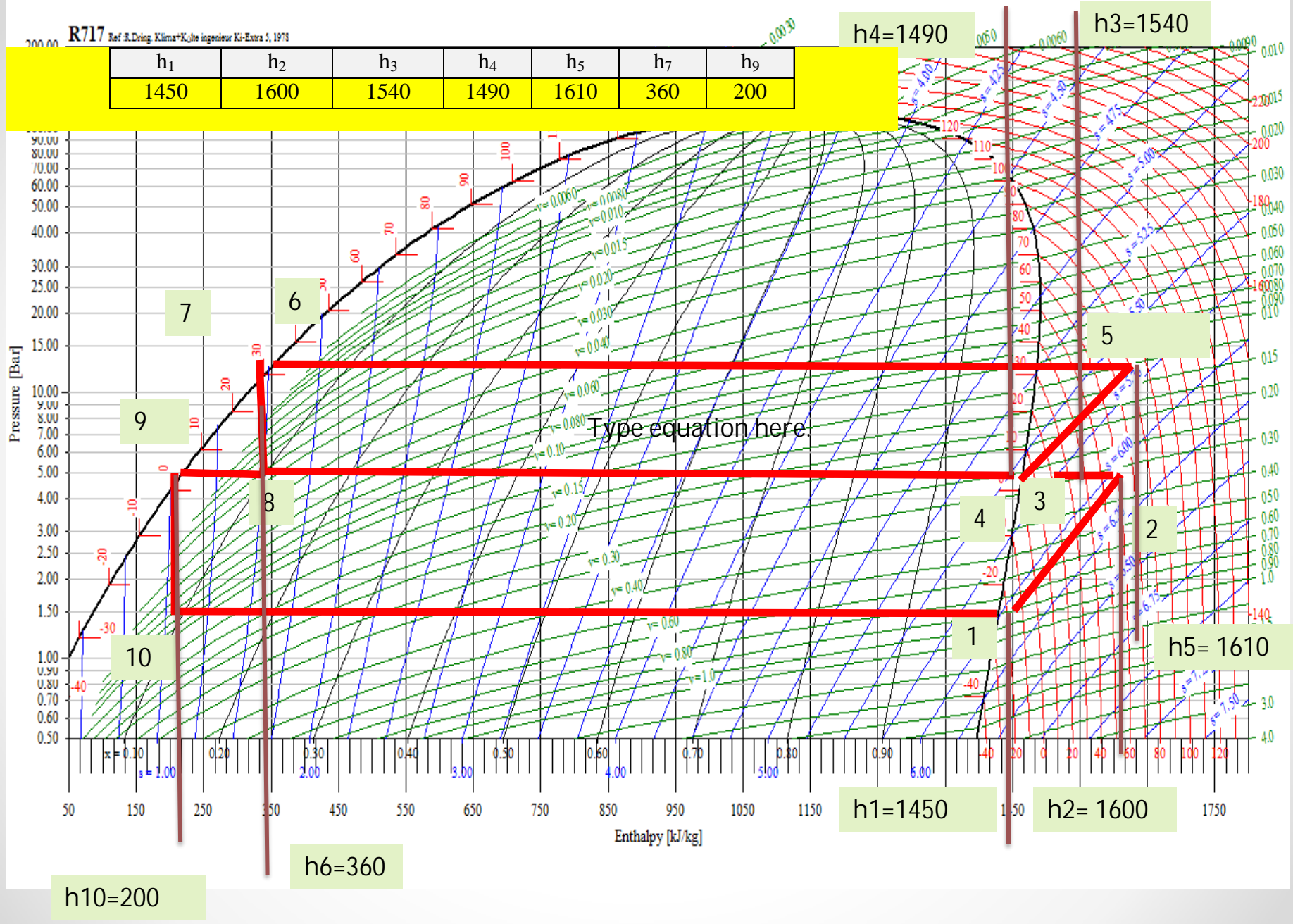
Example 3:

- The following data apply to
- 100 ton compound ammonia compression system with
- water inter-cooler
- and Flash gas inter-cooler,
- condenser pressure 14 bar,
- evaporator pressure 1.5 bar
- and inter-cooler pressure 5 bar,
- the ammonia is cooled to 32°C in the water inter-cooler
- and sub-cooled to 30 °C in the sub-cooler,
- The temperature of suction gas is -18 °C;
- find a- mass flow rate of refrigerant in the cycle b- power consumption in each compressor c- heat rejection in water inter-cooler d-COP

and sub-cooled to 30 °C in the sub-cooler



h_1	h_2	h_3	h_4	h_5	h_7	h_9
1450	1600	1540	1490	1610	360	200

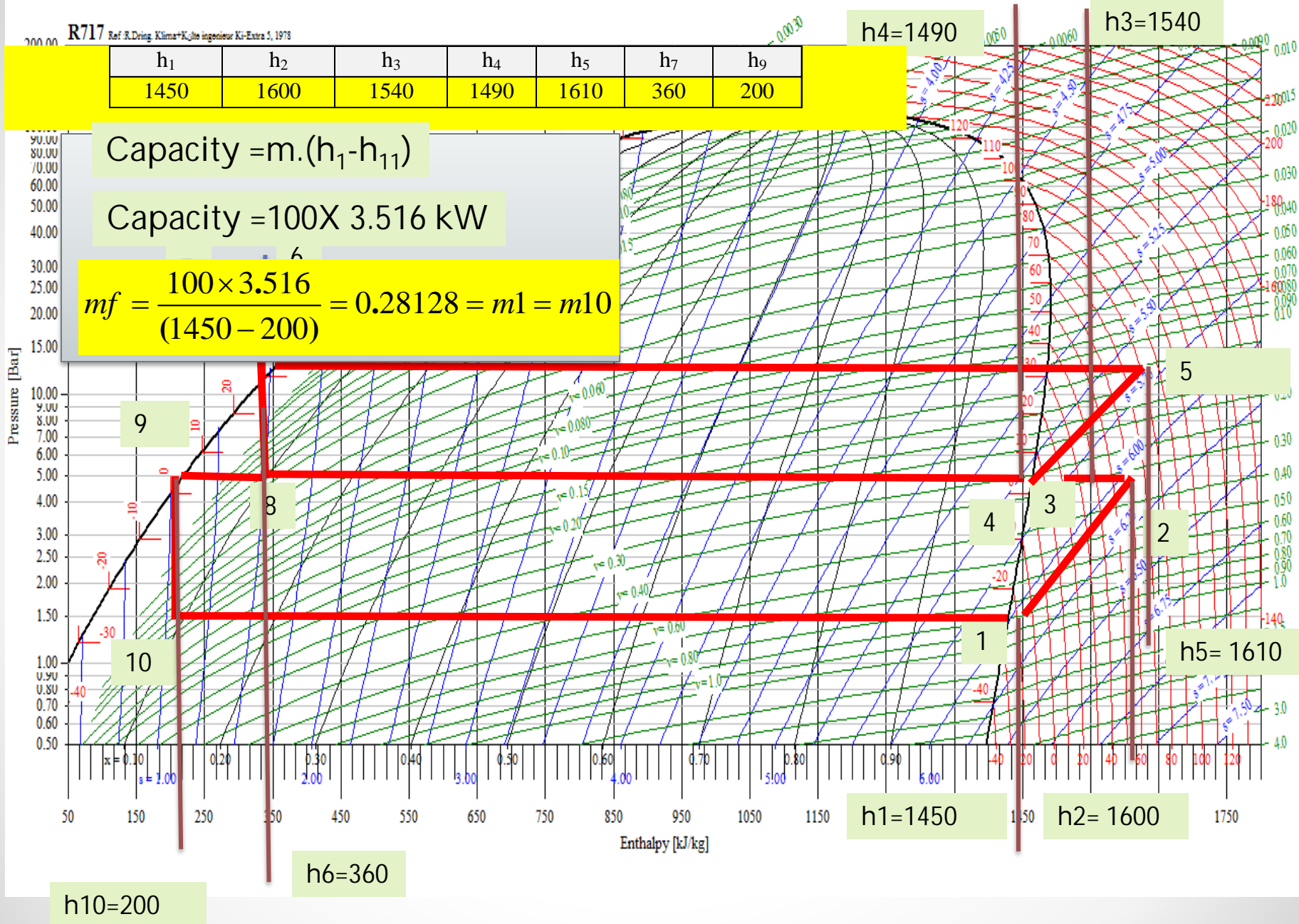


h_1	h_2	h_3	h_4	h_5	h_7	h_9
1450	1600	1540	1490	1610	360	200

Capacity = $m \cdot (h_1 - h_{11})$

Capacity = $100 \times 3.516 \text{ kW}$

$$mf = \frac{100 \times 3.516}{(1450 - 200)} = 0.28128 = m1 = m10$$



Heat balance inter-cooler

h_1	h_2	h_3	h_4	h_5	h_7	h_9
1450	1600	1540	1490	1610	360	200

$$m_8 \cdot h_8 + m_3 \cdot h_3 = m_4 \cdot h_4 + m_9 \cdot h_9$$

$$m_9 = m_{10} = m_1 = m_2 = m_3$$

$$m_4 = m_5 = m_6 = m_7 = m_8$$

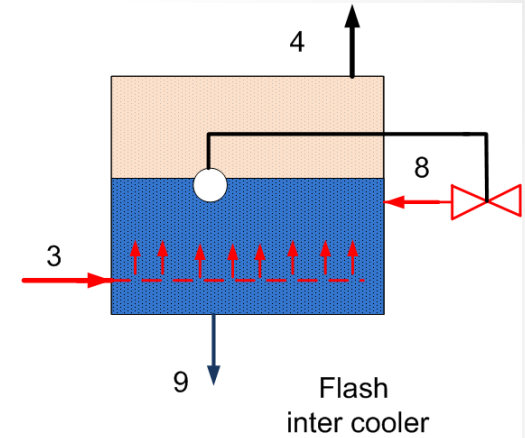
$$m_4 \cdot h_8 + m_3 \cdot h_3 = m_4 \cdot h_4 + m_3 \cdot h_9$$

$$m_4 \cdot (h_8 - h_4) = m_3 \cdot (h_9 - h_3)$$

$$m_1 = 0.28128 \text{ kg/s}$$

$$m_4 = \frac{m_3(h_9 - h_3)}{h_8 - h_4}$$

$$m_4 = \frac{0.2818(200 - 1540)}{360 - 1490} = 0.33 \text{ kg/s}$$

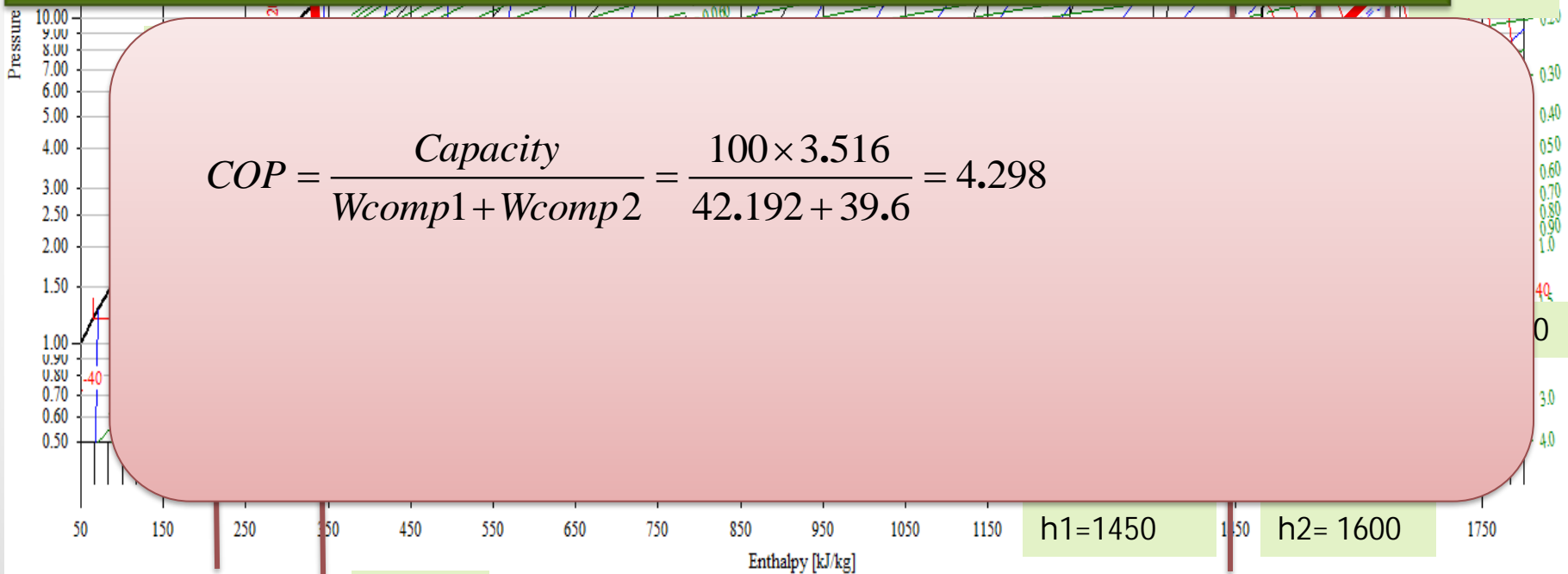


Comp. power L.P.= $m_{f1} \cdot (h_2 - h_1) = 0.28128(1600 - 1450) = 42.192 \text{ kW}$

Comp. power H.P.= $m_{f4} \cdot (h_5 - h_4) = 0.33(1610 - 1490) = 39.6 \text{ kW}$

Qinter-cooler= $m_{f1} \cdot (h_2 - h_3) = 0.28128 \times (1600 - 1540) = 16.87 \text{ kW}$

$$COP = \frac{Capacity}{W_{comp1} + W_{comp2}} = \frac{100 \times 3.516}{42.192 + 39.6} = 4.298$$



h10=200

h6=360

h1=1450

h2=1600

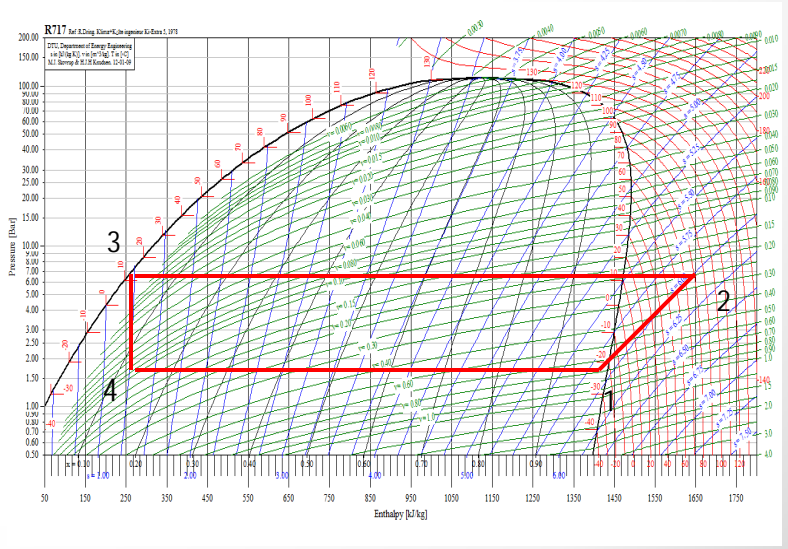
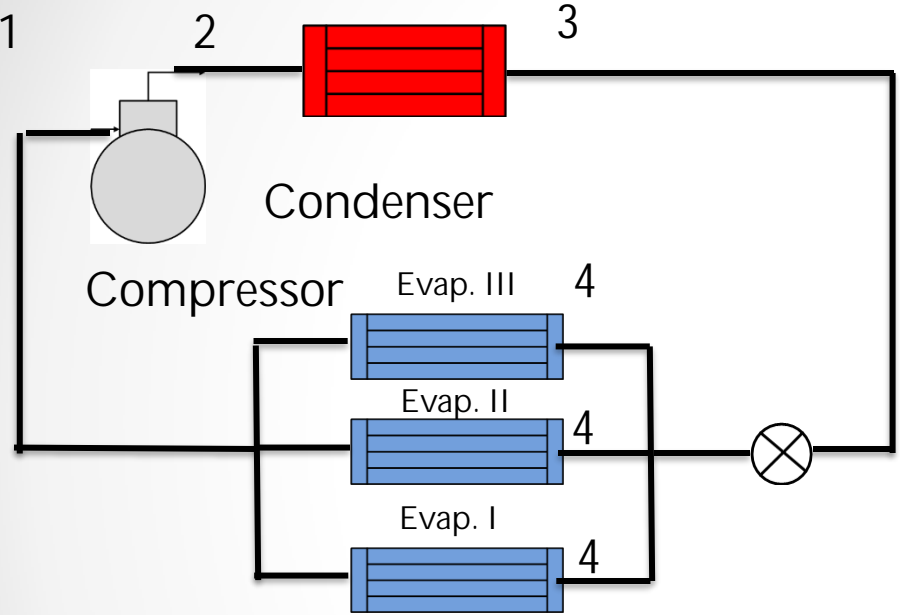
Multiple Evaporator Systems

- In many refrigeration plants, it may be required to have different temperature maintained at various zones or compartments.
- For example in a food market where different food product stored in large quantities, it is imperative that each food product should be stored at different conditions of temperature and humidity.

Type multi evaporator Systems

- 1-All Evaporators Operating at The Same Temperature
- 2- Individual expansion valve and back pressure valves system
- 3- Multiple expansion valves and back pressure valves system

1-All Evaporators Operating at The Same Temperature



Example 4:

- A single compressor system using R-134a has
 - three evaporators of 10, 20 and 30 tons capacities.
 - All evaporators operating at the same temperature of -10°C .
 - The condenser pressure is 10 bar
 - and the liquid is sub cooled in the condenser by 10°C .
 - The discharge from the evaporator is dry saturated vapour
 - and the compression is assumed to be isentropic.
- Determine
- 1-the refrigeration effect 2- rate of flow of the refrigerant in kg/s and 3-the theoretical horse power required.

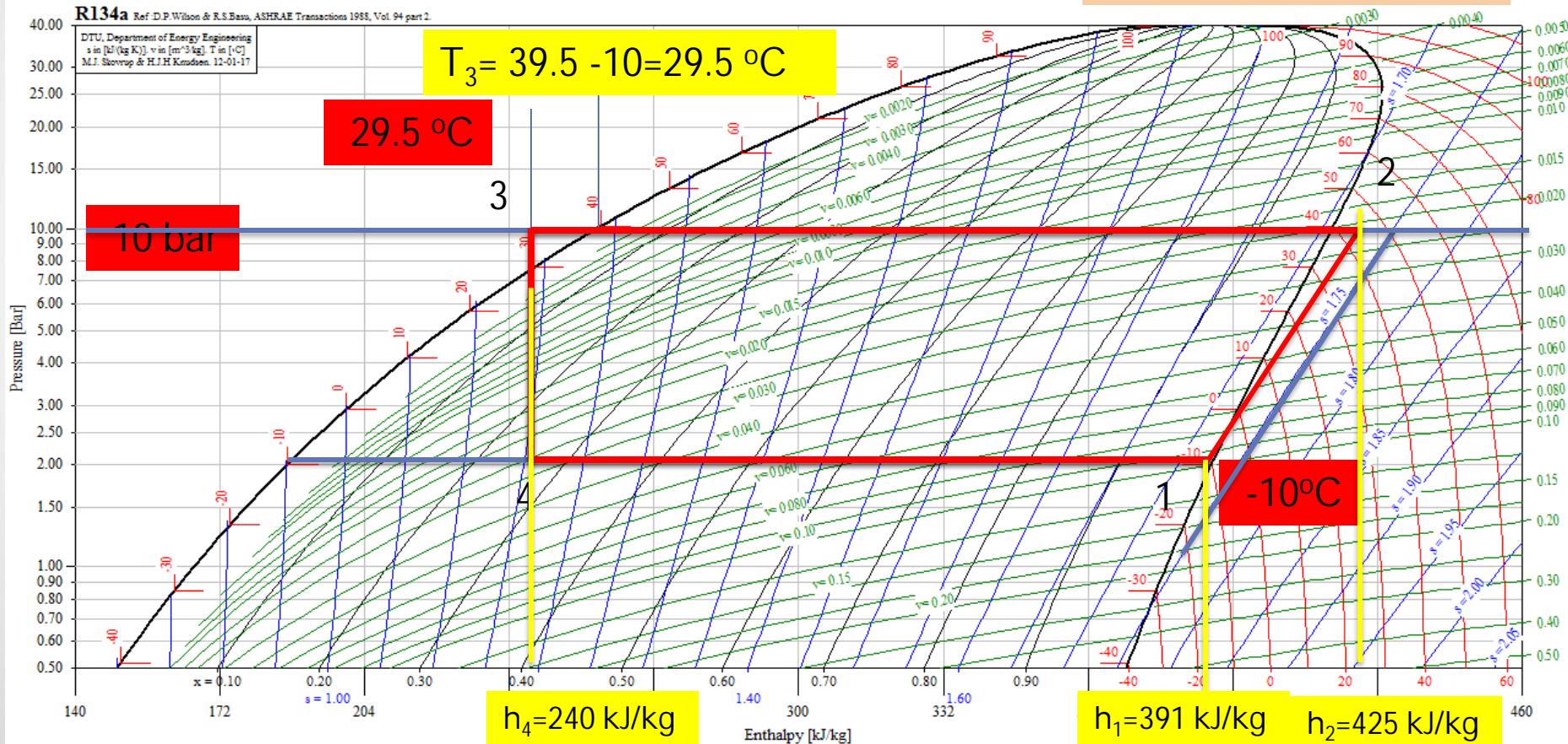
All evaporators at the same temperature of -10°C

The condenser pressure is 10 bar

and the liquid is sub cooled in the condenser by 10°C .

The discharge from the evaporator is dry saturated vapour

h_1	h_2	h_3
391	425	240



Total capacity of the cycle = 10 + 20 + 30 = 60 ton = 60 x 3.516 = 210.84 kW

Refrigeration effect = 391 - 240 = 151 kJ/kg

Capacity = m x (ref. Effect)

$$210.8 = m \times 151$$

$$m = 1.39 \text{ kg/s}$$

h1	h2	h3
391	425	240

Compressor work = 425 - 391 = 34 kJ/kg

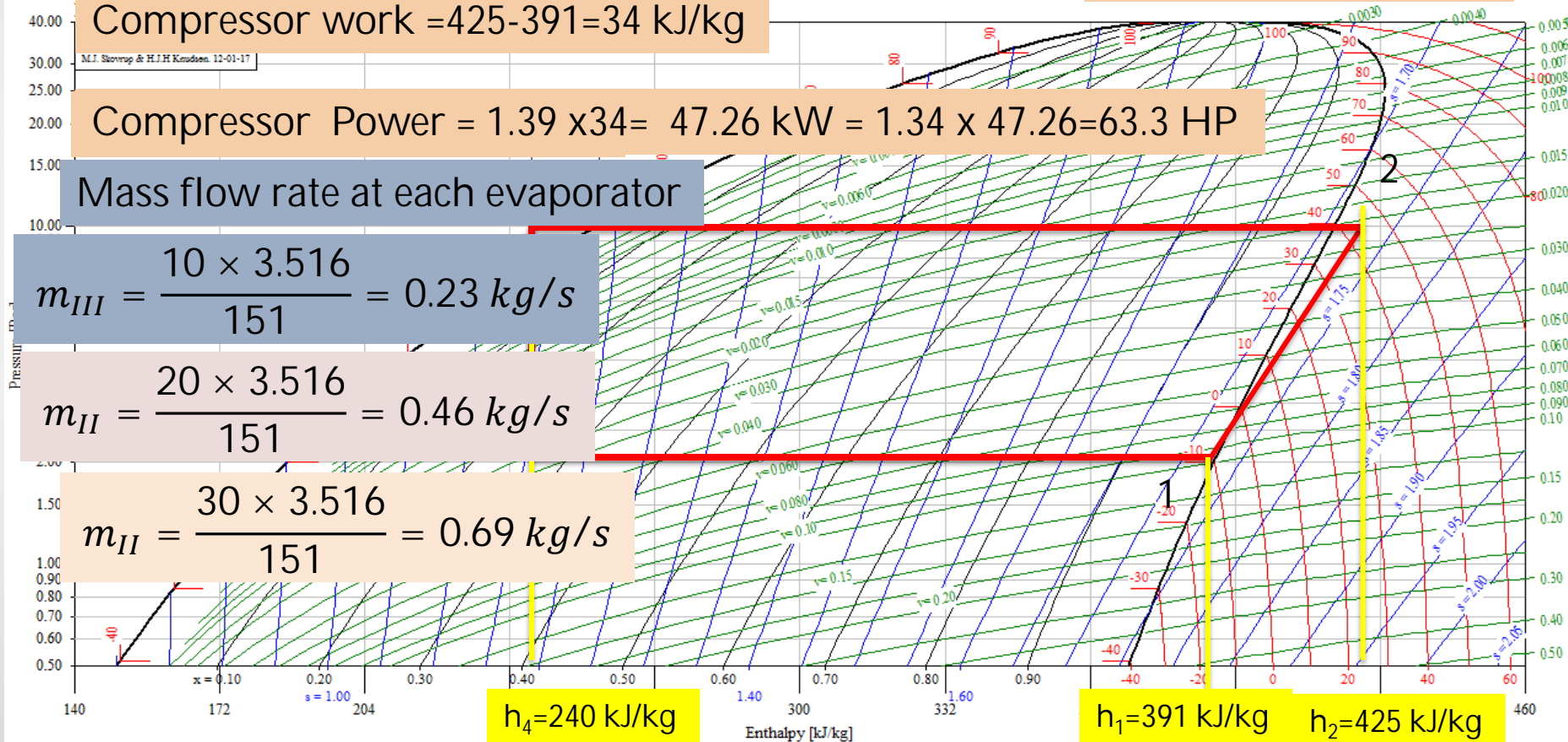
Compressor Power = 1.39 x 34 = 47.26 kW = 1.34 x 47.26 = 63.3 HP

Mass flow rate at each evaporator

$$m_{III} = \frac{10 \times 3.516}{151} = 0.23 \text{ kg/s}$$

$$m_{II} = \frac{20 \times 3.516}{151} = 0.46 \text{ kg/s}$$

$$m_{I} = \frac{30 \times 3.516}{151} = 0.69 \text{ kg/s}$$



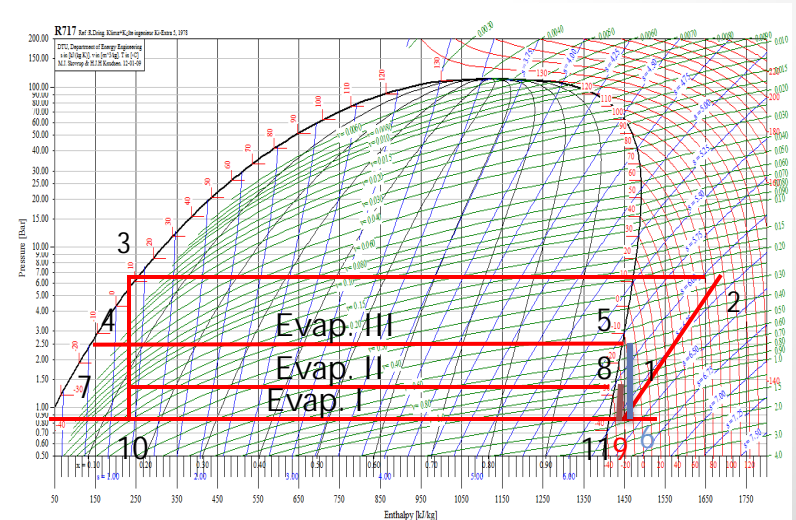
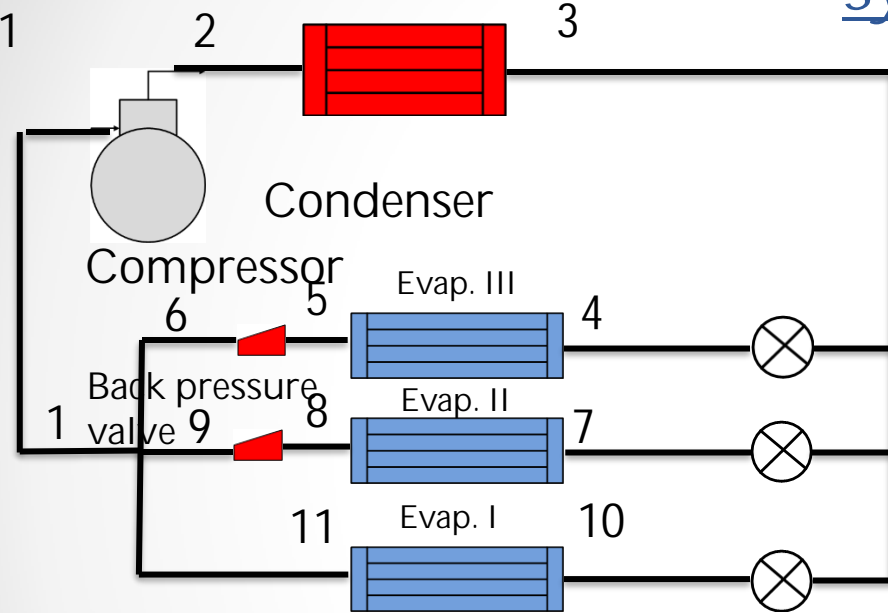
$h_4 = 240 \text{ kJ/kg}$

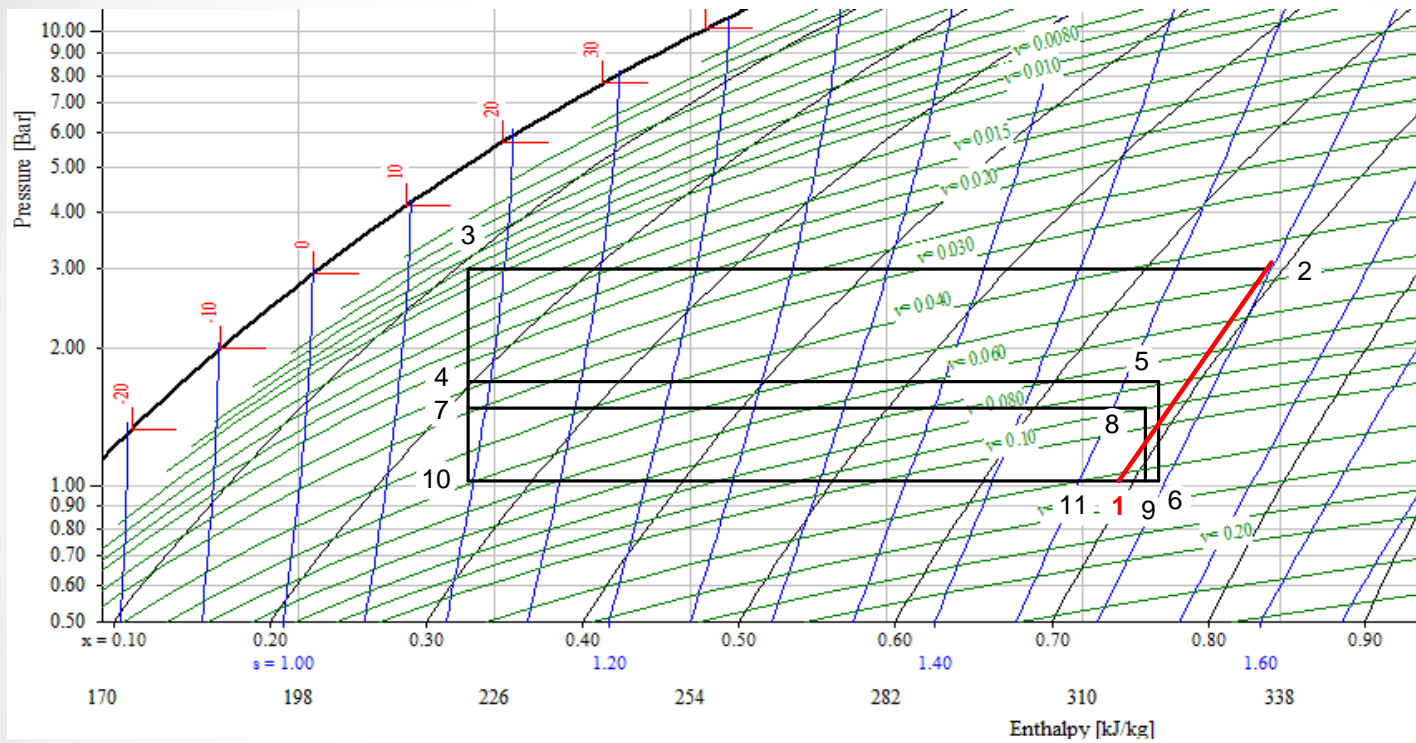
$h_1 = 391 \text{ kJ/kg}$

$h_2 = 425 \text{ kJ/kg}$

Individual expansion valve and back pressure valves

system



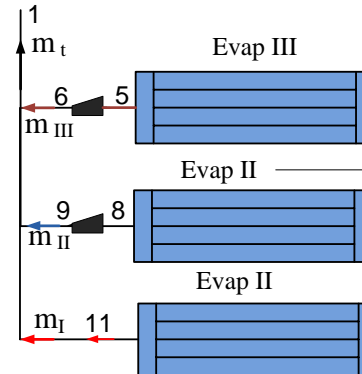


Heat balance of suction line

$$m_t \cdot h_1 = m_{III} \cdot h_6 + m_{II} \cdot h_9 + m_I \cdot h_{11}$$

$$m_t = m_{III} + m_{II} + m_I$$

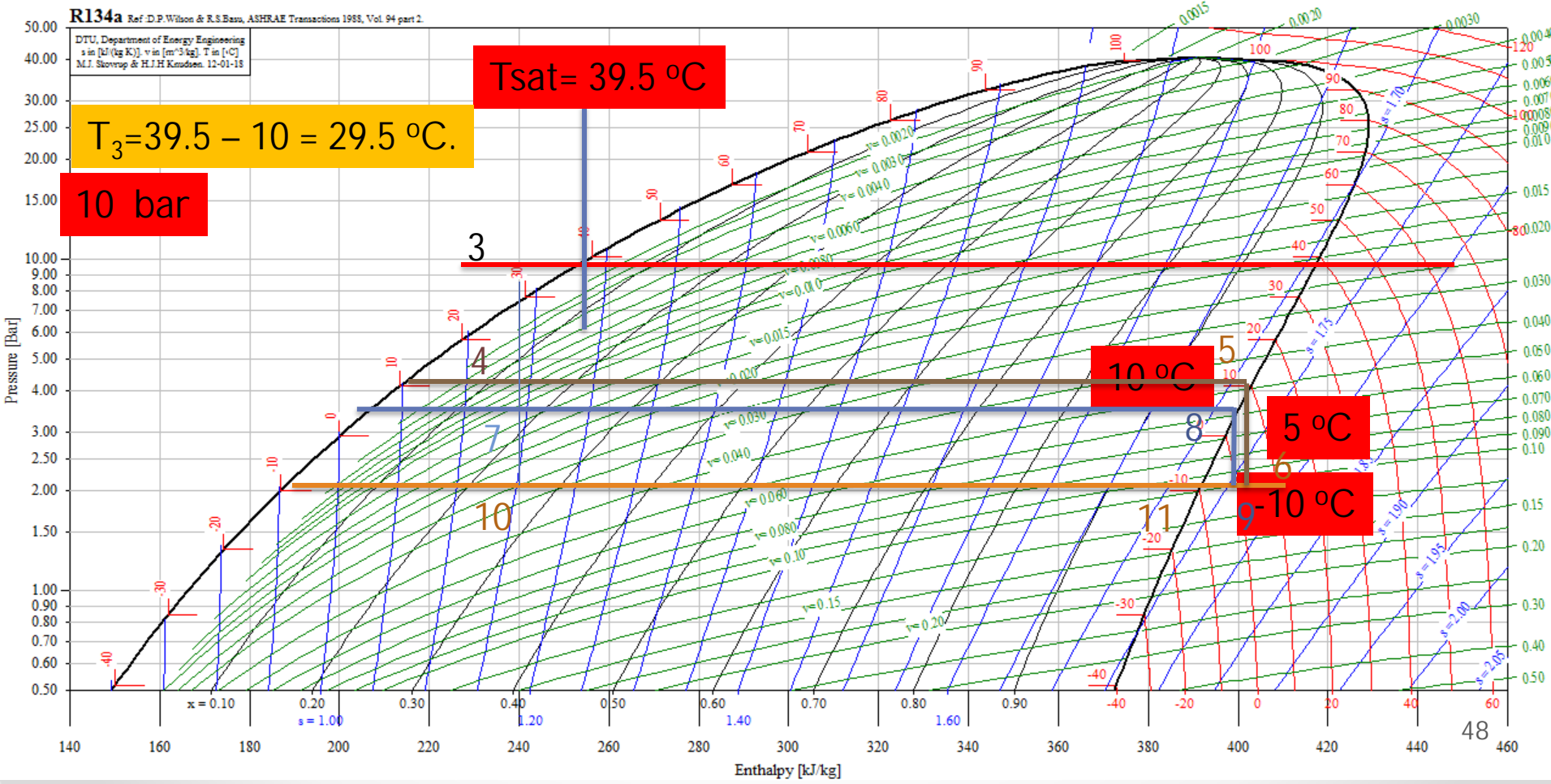
$$h_1 = \frac{m_{III}h_6 + m_{II}h_9 + m_Ih_{11}}{m_{III} + m_{II} + m_I}$$



Example 5

- A refrigeration installation using R-134a
- comprises one compressor,
- one condenser
- and three evaporators of capacities of 10, 20 and 30tons respectively.
- The temperature to be maintained in the evaporators is 10, 5 and -10°C respectively.
- Each evaporator is fitted with an expansion valve.
- The condenser pressure is to be 10 bar.
- The exit condition from evaporator is dry saturated vapour
- and liquid is sub cooled by 10°C in the condenser.
- Determine
- 1-the refrigeration effect
- 2- rate of flow of the refrigerant in kg/s and
- 3-the theoretical horse power required

The exit condition from evaporator is dry saturated vapour

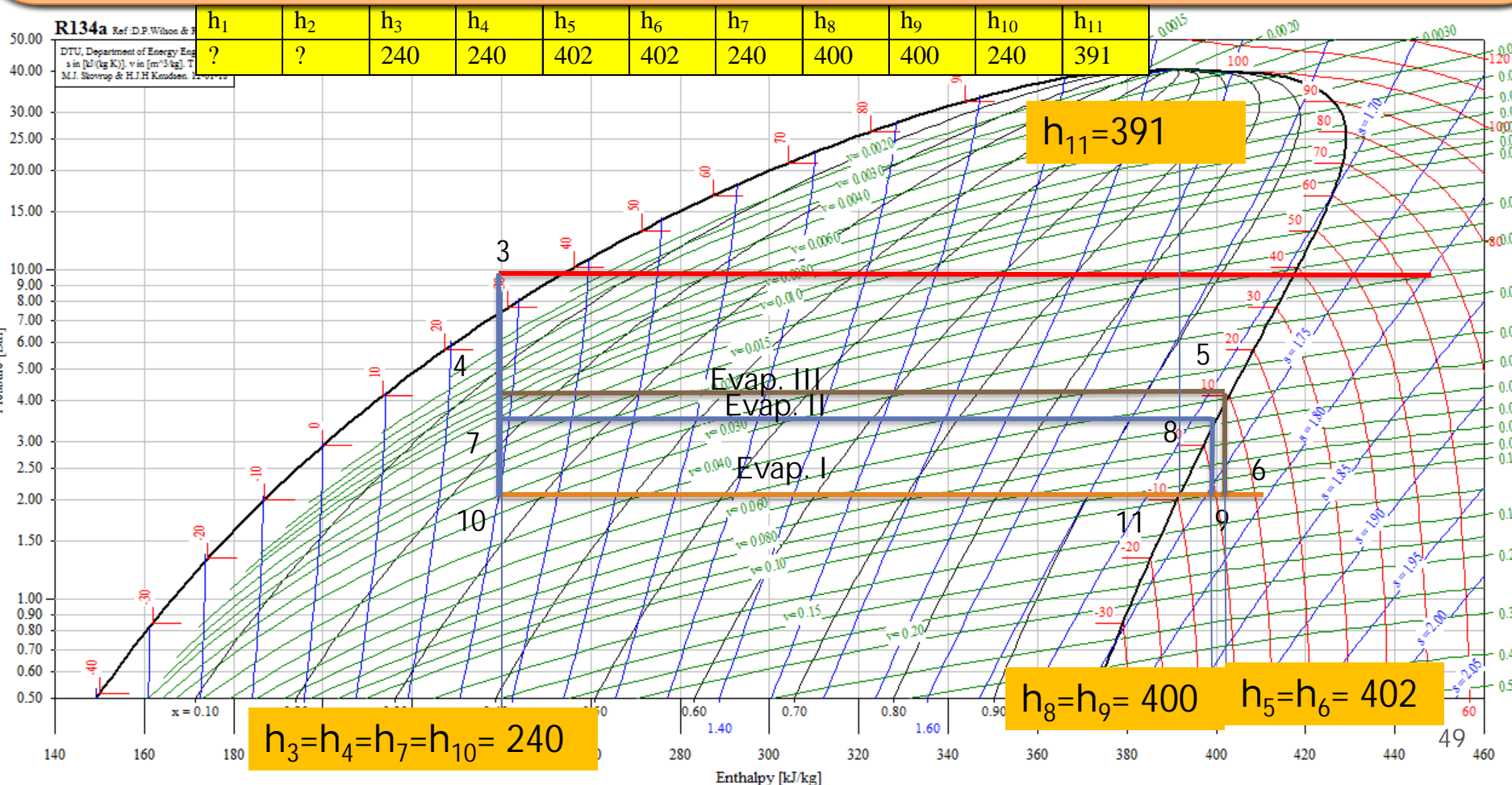
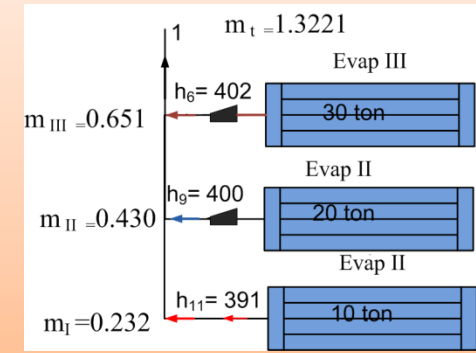


Heat balance of suction line

$$m_t \cdot h_1 = m_{III} \cdot h_6 + m_{II} \cdot h_9 + m_I \cdot h_{11}$$

$$h_1 = \frac{m_{III}h_6 + m_{II}h_9 + m_Ih_{11}}{m_{III} + m_{II} + m_I}$$

$$h_1 = \frac{0.651 \times 402 + 0.439 \times 400 + 0.323 \times 391}{1.413} = 398 \text{ kJ/kg}$$



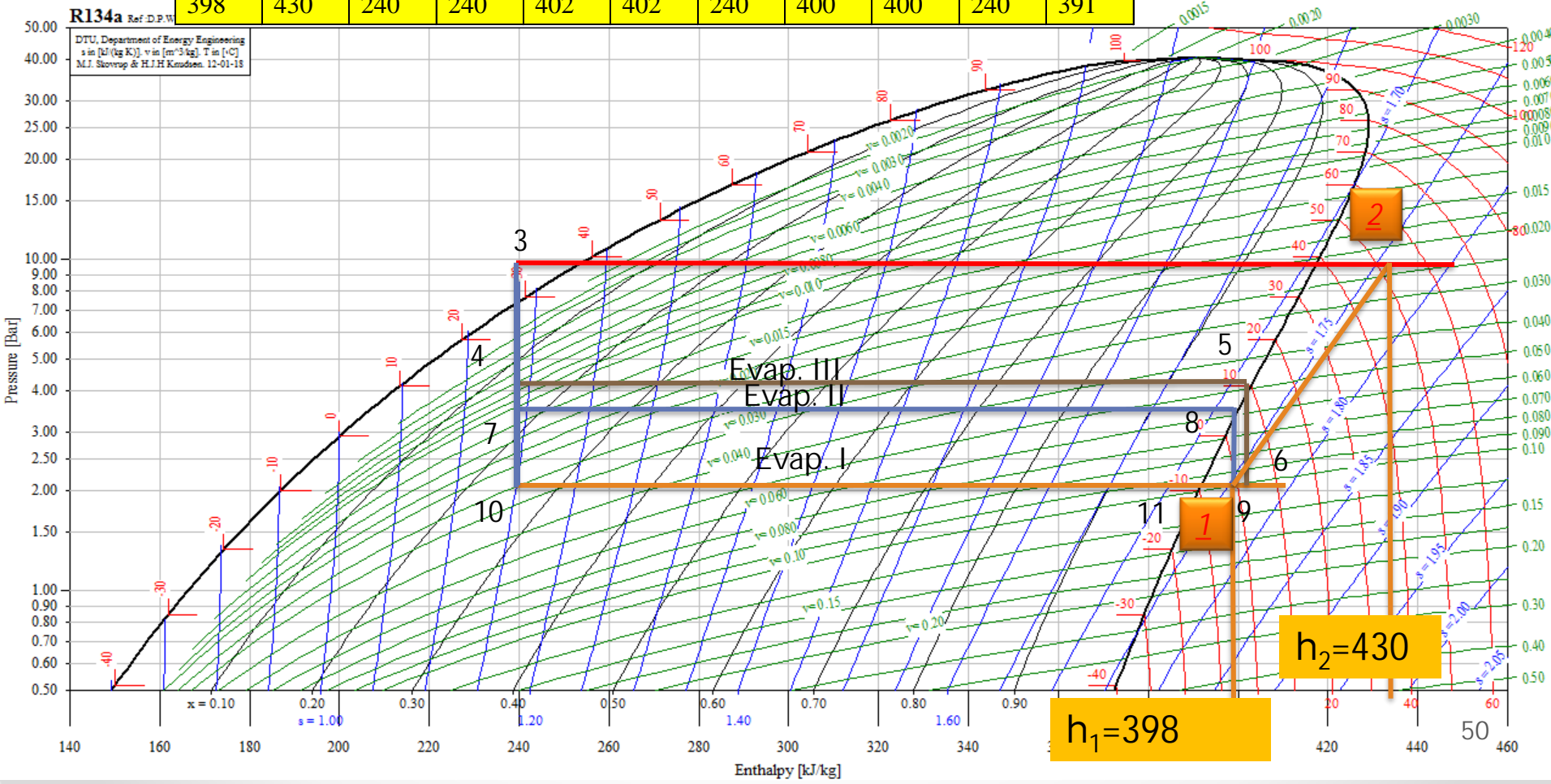
$h_3 = h_4 = h_7 = h_{10} = 240$

$h_8 = h_9 = 400$ $h_5 = h_6 = 402$

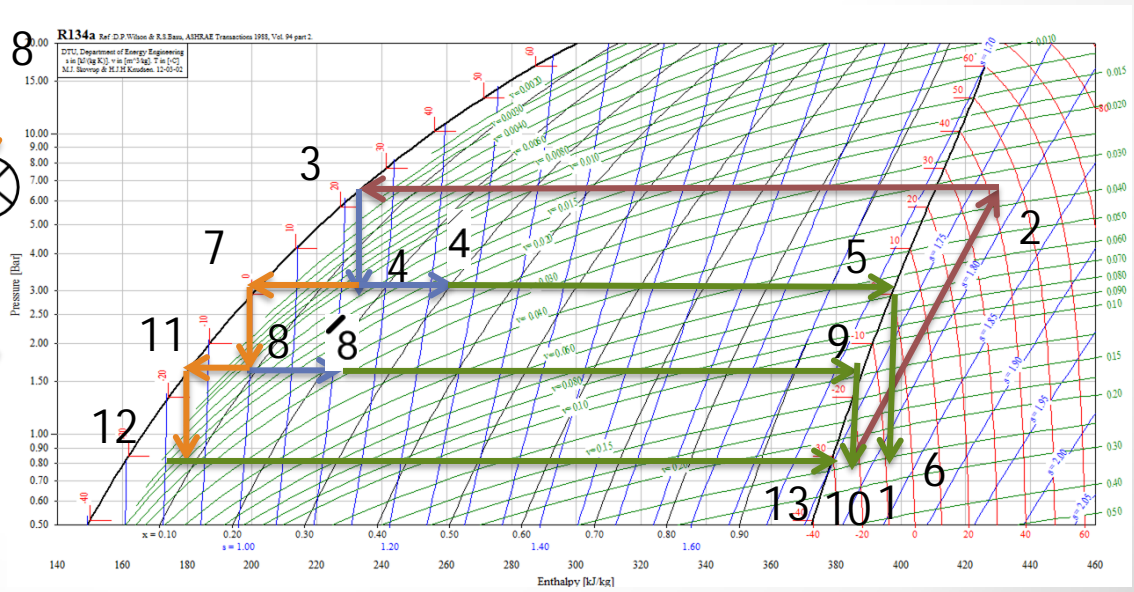
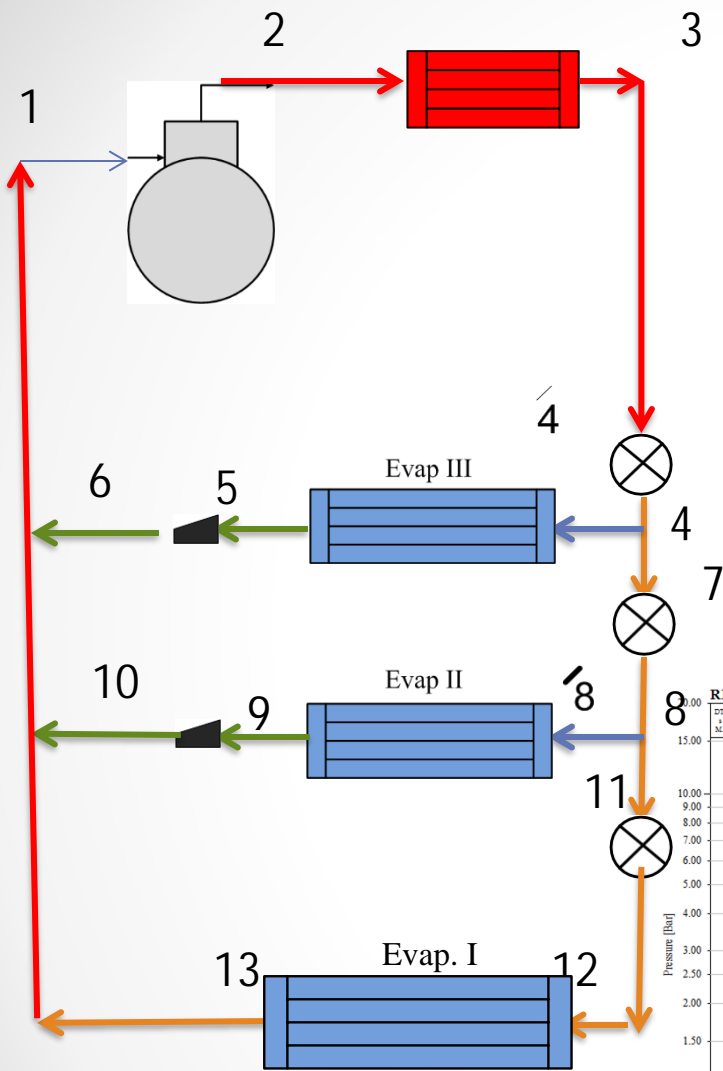
$h_{11} = 391$

comp. Power = $m(h_2 - h_1) = 1.322 \times (430 - 398) = 42.3 \text{ kW} = 42.3 \times 1.34 = 56.6 \text{ HP}$

h_1	h_2	h_3	h_4	h_5	h_6	h_7	h_8	h_9	h_{10}	h_{11}
398	430	240	240	402	402	240	400	400	240	391



- *Multiple expansion valves and back pressure valves system*



Cycle analysis

$$Q_{\text{evap I}} = m_{12} (h_{13} - h_{12})$$

Heat balance

$$m_8 \cdot h_8 = m_{\dot{g}} \cdot h_{\dot{g}} + m_{11} \cdot h_{11}$$

$$m_8 = m_{\dot{g}} + m_{11}$$

$$(m_{\dot{g}} + m_{11}) \cdot h_8 = m_{\dot{g}} \cdot h_{\dot{g}} + m_{11} \cdot h_{11}$$

$$h_{\dot{g}} = \frac{(m_{\dot{g}} + m_{11}) \cdot h_8 - m_{11} \cdot h_{11}}{m_{\dot{g}}}$$

Evap. II

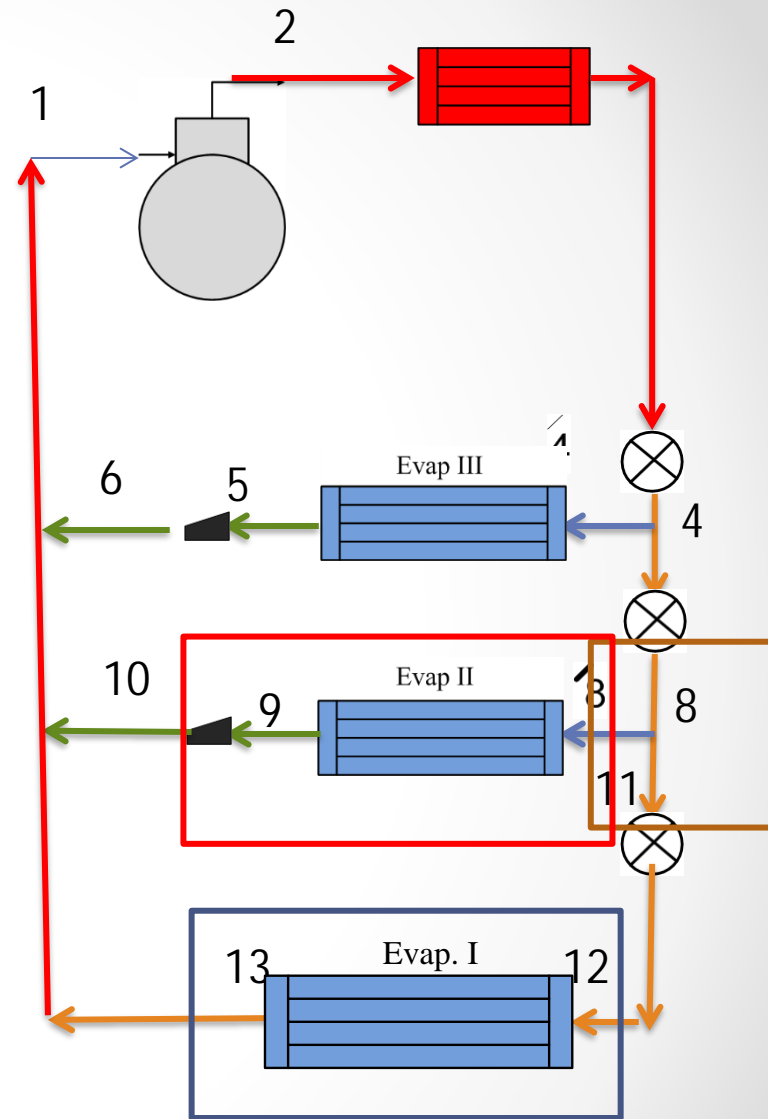
$$Q_{\text{evapII}} = m_{\dot{g}} \cdot (h_9 - h_{\dot{g}})$$

$$= m_{\dot{g}} \cdot \left(h_9 - \frac{(m_{\dot{g}} + m_{11}) \cdot h_8 - m_{11} \cdot h_{11}}{m_{\dot{g}}} \right)$$

$$Q_{\text{evapII}} = m_{\dot{g}} \cdot h_9 - (m_{\dot{g}} + m_{11}) \cdot h_8 + m_{11} \cdot h_{11}$$

$$Q_{\text{evapII}} = m_{\dot{g}} \cdot (h_9 - h_8) - m_{11} \cdot (h_8 - h_{11})$$

$$Q_{\text{evapII}} + m_{11} \cdot (h_8 - h_{11}) = m_{\dot{g}} \cdot (h_9 - h_8)$$

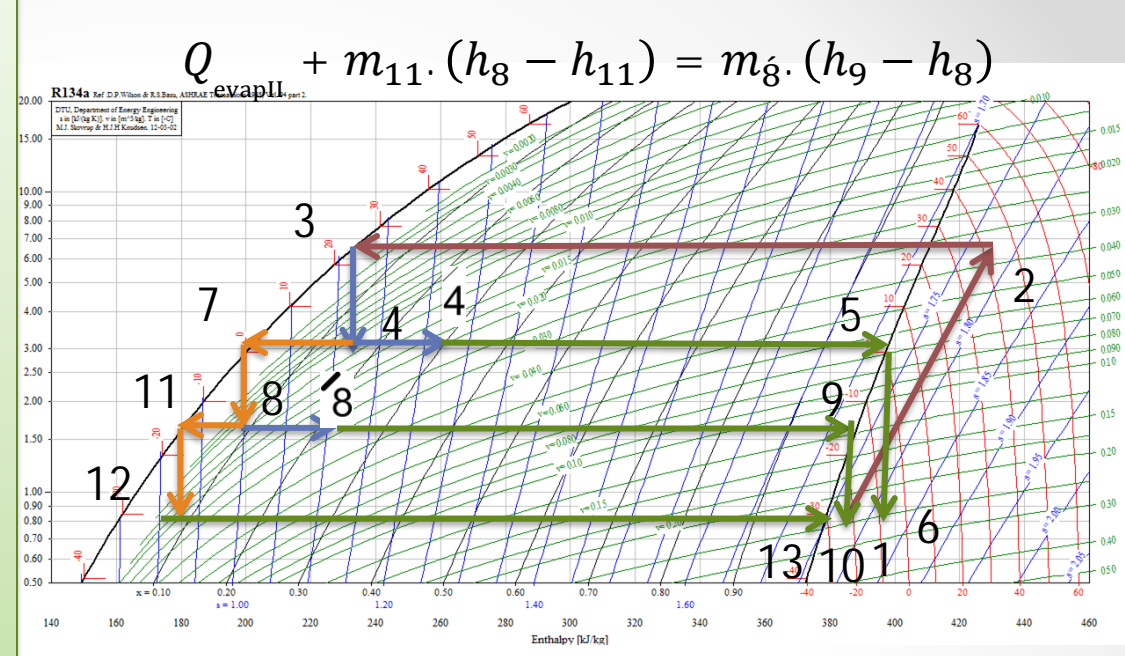


$$m_{\dot{g}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + \frac{m_{11} \cdot (h_8 - h_{11})}{(h_9 - h_8)}$$

$$m_{\dot{g}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + \frac{m_{11} \cdot (h_8 - h_{11})}{(h_9 - h_8)}$$

$$m_{\dot{g}} = \frac{Q_{\text{evapII}}}{(h_9 - h_8)} + m_{11} \frac{x_8}{(1 - x_8)}$$

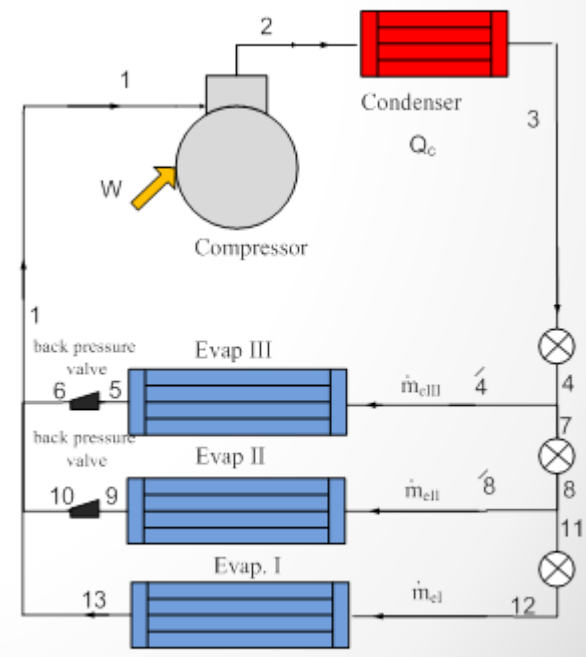
$$m_8 = m_{\dot{g}} + m_{11}$$



Similarly

$$m_{\dot{4}} = \frac{Q_{\text{evapIII}}}{(h_5 - h_4)} + m_7 \frac{x_4}{(1 - x_4)}$$

$$m_4 = m_{\dot{4}} + m_7$$



Heat balance of point 1

$$m_1 \cdot h_1 = m_6 \cdot h_6 + m_{10} \cdot h_{10} + m_{13} \cdot h_{13}$$

$$h_1 = \frac{m_6 \cdot h_6 + m_{10} \cdot h_{10} + m_{13} \cdot h_{13}}{m_1}$$

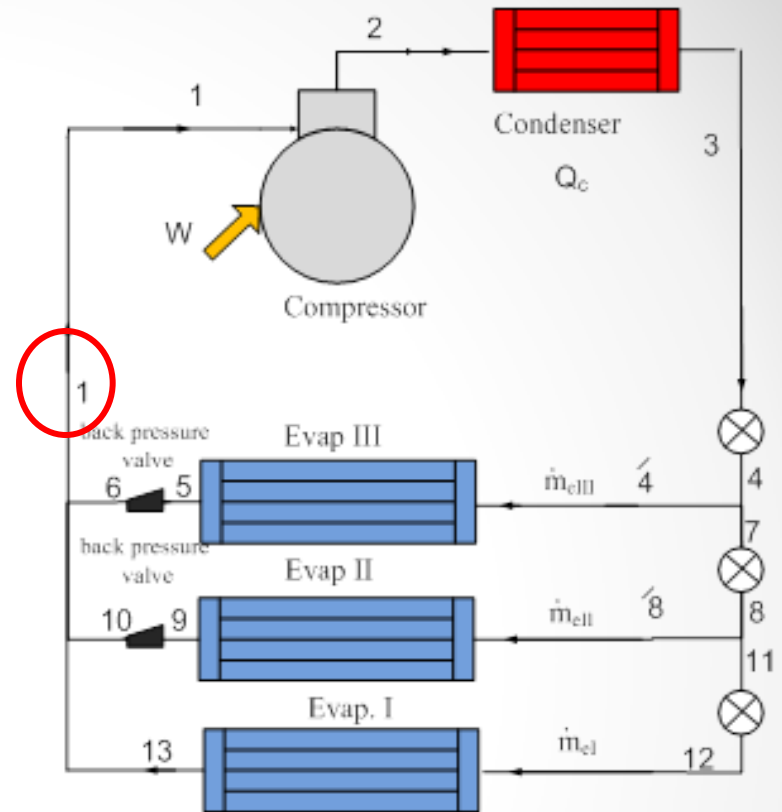
$$m_1 = m_4$$

$$m_4 = m_5 = m_6$$

$$m_8 = m_9 = m_{10}$$

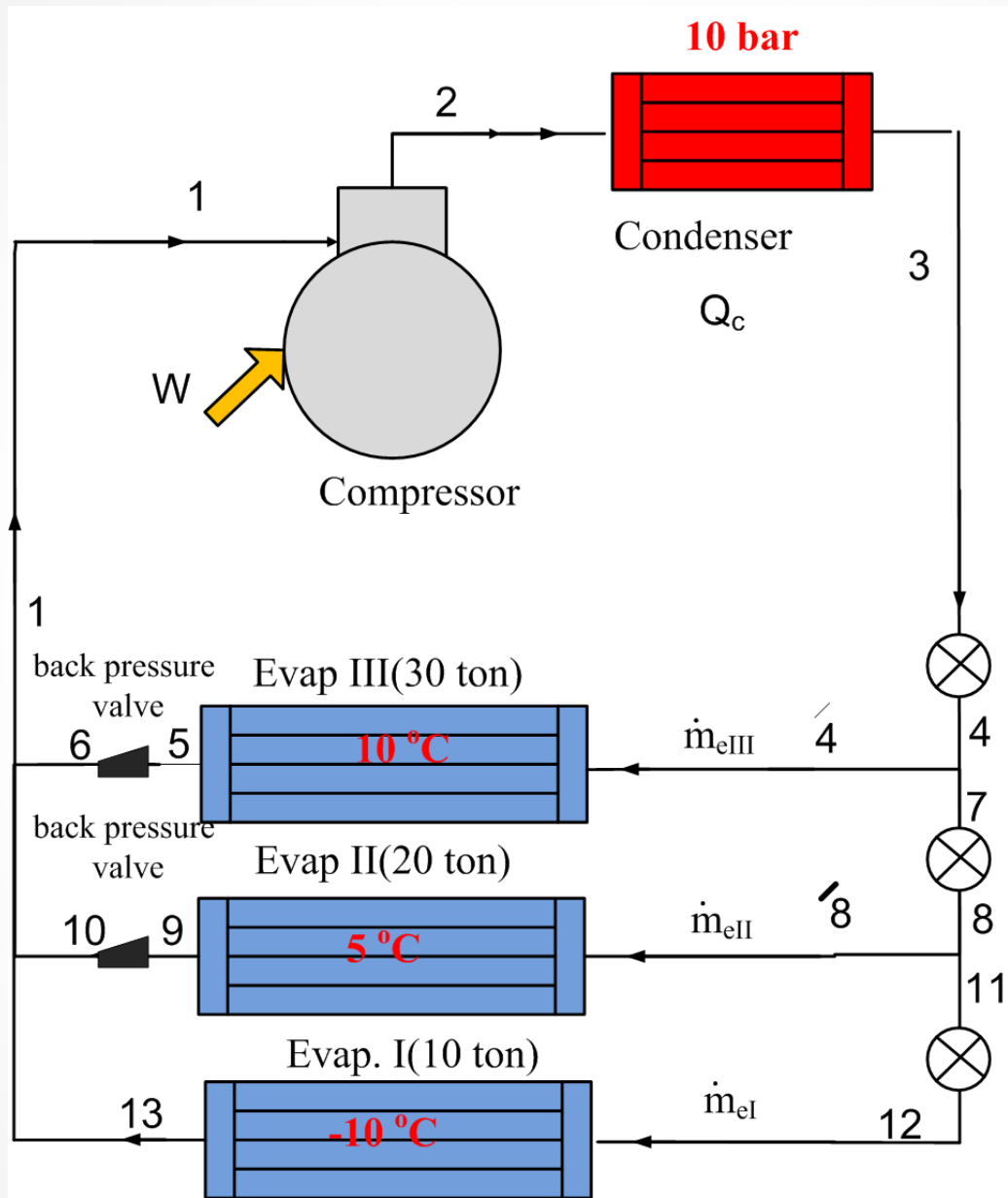
$$m_{12} = m_{13}$$

$$h_1 = \frac{m_4 \cdot h_6 + m_8 \cdot h_{10} + m_{12} \cdot h_{13}}{m_4}$$

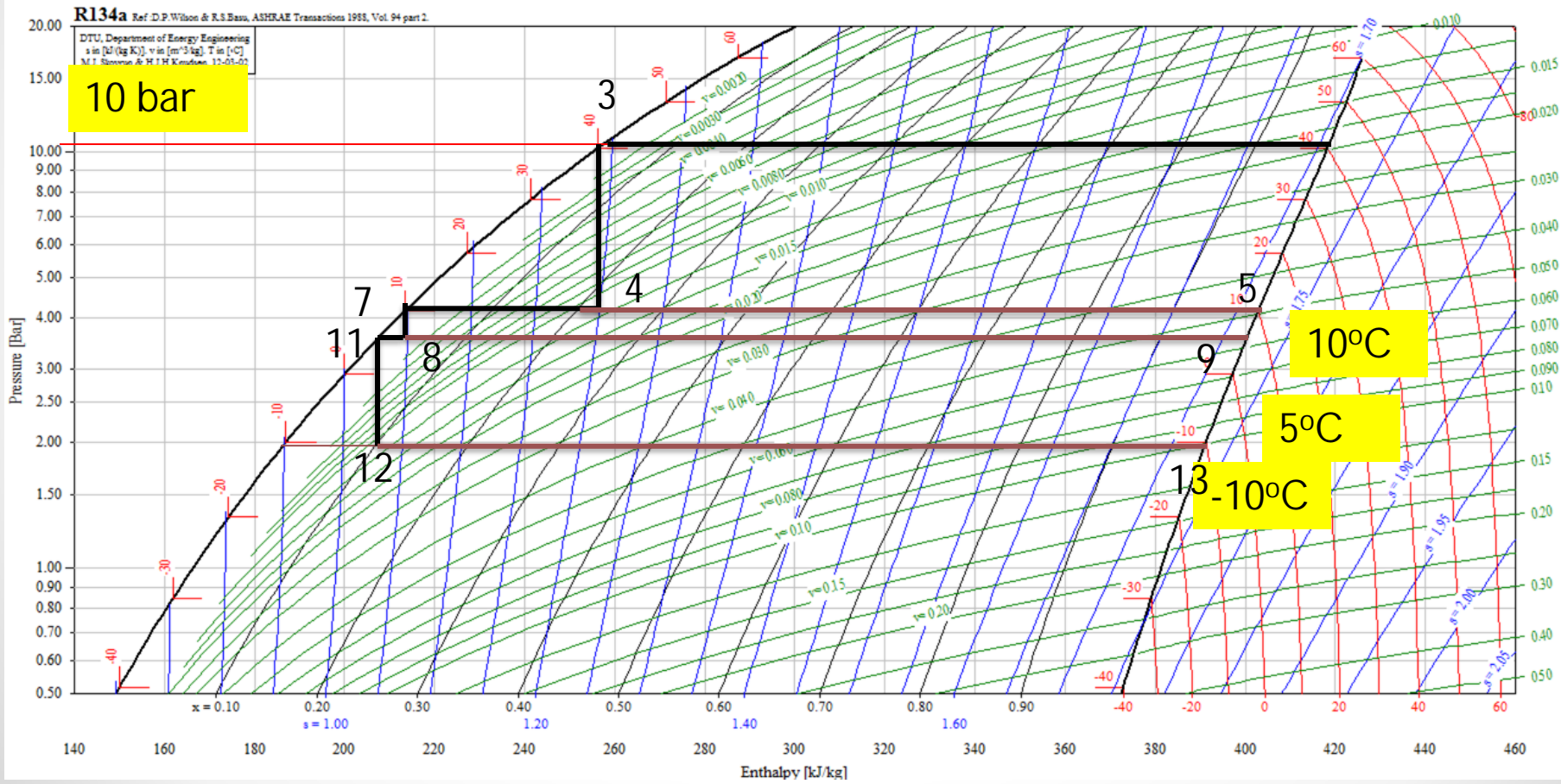


Example 6

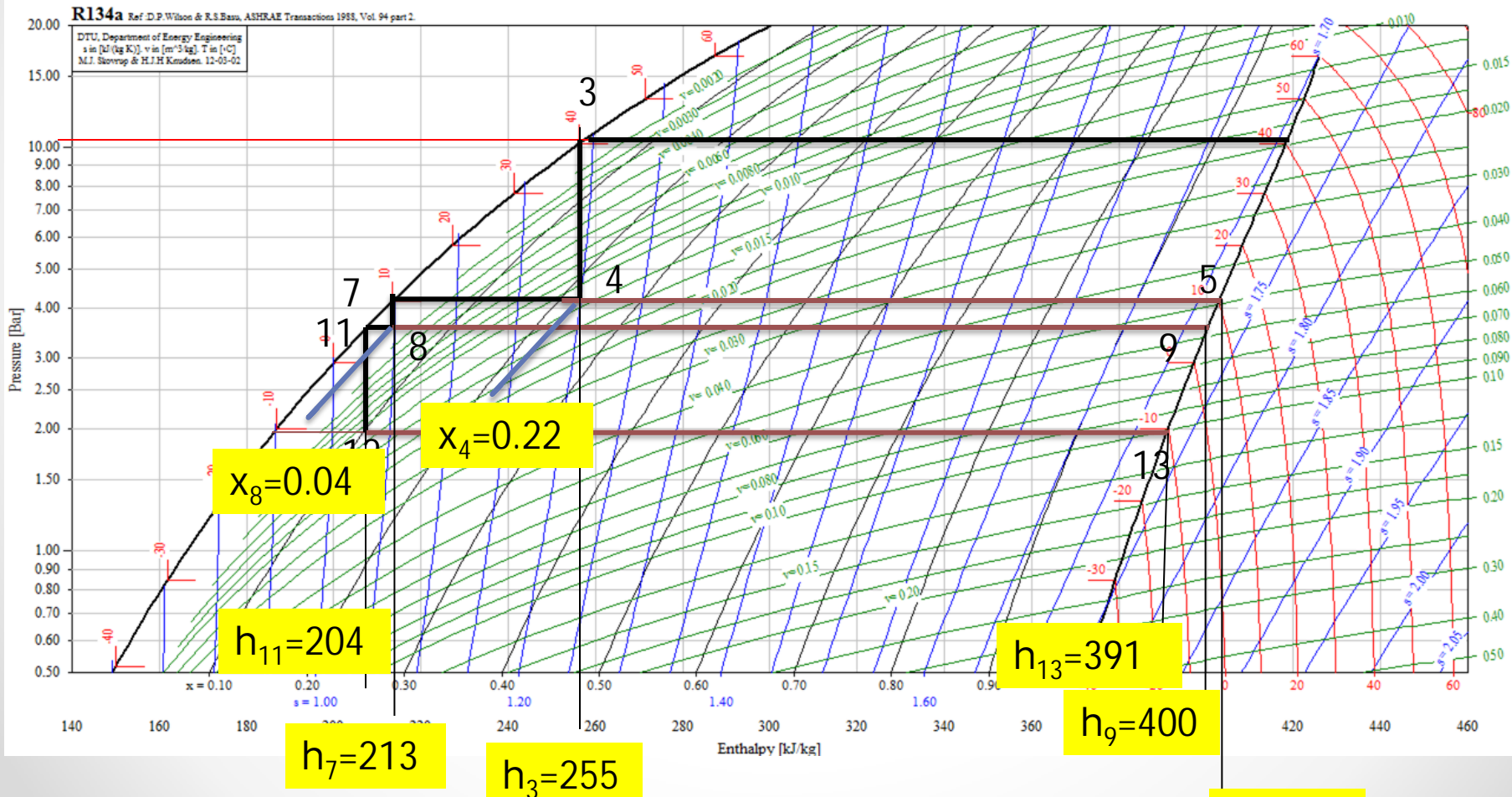
- A refrigeration plant using R-134a, the plant have one compressor, one condenser and three evaporators of 30, 20 and 10 ton capacities, maintained at 10, 5 and -10°C respectively. **The system comprises multiple- expansion valves** and back pressure valves. The condenser pressure is 10 bar. The vapour leaving the evaporators are dry saturated. Assume isentropic compression.
- Determine;
- 1-the refrigeration effect
- 2- rate of flow of the refrigerant in kg/s
- and 3-the theoretical horse power required.



The condenser pressure is 10 bar.



h	3	5	7	9	11	13	1	2	
kJ/kg	255	403	213	400	204	391	?	?	



h	3	5	7	9	11	13	1	2	
kJ/kg	255	403	213	400	204	391	?	?	

Evaporator I

$$Q_{\text{evap I}} = m_{12} (h_{13} - h_{12})$$

$$10 \times 3.516 = m_{12} (391 - 204)$$

$$m_{12} = \frac{10 \times 3.516}{391 - 204} = 0.188 \text{ kg/s}$$

$$x_8 = 0.04$$

$$m_{\dot{8}} = \frac{Q_{\text{evap II}}}{(h_9 - h_8)} + m_{11} \frac{x_8}{(1 - x_8)}$$

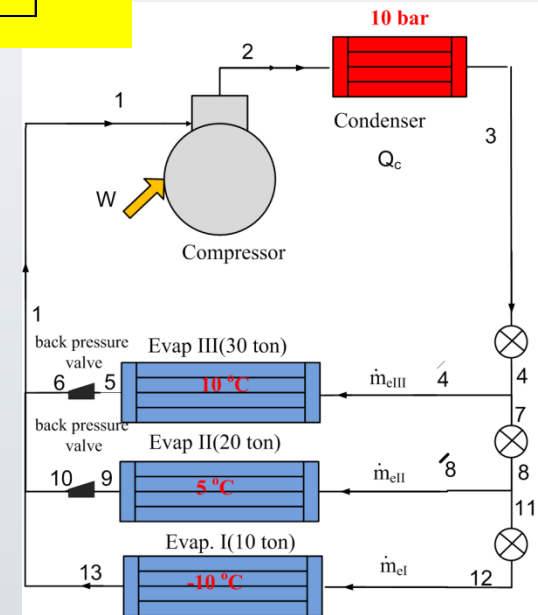
$$= \frac{20 \times 3.516}{(400 - 213)} + 0.188 \frac{0.04}{(1 - 0.04)}$$

$$= 0.376 + 0.0078$$

$$m_{\dot{8}} = 0.383 \text{ kg/s}$$

$$m_8 = m_{\dot{8}} + m_{11} = 0.383 + 0.188$$

$$= 0.571 \frac{\text{kg}}{\text{s}}$$



$$x_4 = 0.22$$

$$m_{\dot{4}} = \frac{Q_{\text{evap III}}}{(h_5 - h_4)} + m_8 \frac{x_4}{(1 - x_4)}$$

$$= \frac{30 \times 3.516}{403 - 255} + 0.571$$

$$\times \frac{0.22}{1 - 0.22} = 0.712 + 0.161$$

h	3	5	7	9	11	13	1	2	
kJ/kg	255	403	213	400	204	391	?	?	

$$m_{\dot{4}} = 0.873 \text{ kg/s}$$

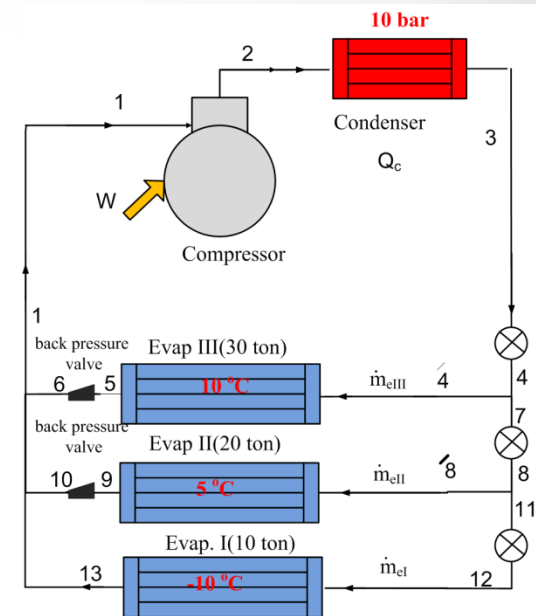
$$m_4 = m_{\dot{4}} + m_7 = 0.873 + 0.571 = 1.44 \text{ kg/s}$$

Heat balance of point 1

$$h_1 = \frac{m_{\dot{4}} \cdot h_6 + m_{\dot{8}} \cdot h_{10} + m_{12} \cdot h_{13}}{m_4}$$

$$= \frac{0.873 \times 403 + 0.383 \times 400 + 0.188 + 391}{1.44}$$

$$h_1 = 401 \text{ kJ/kg}$$

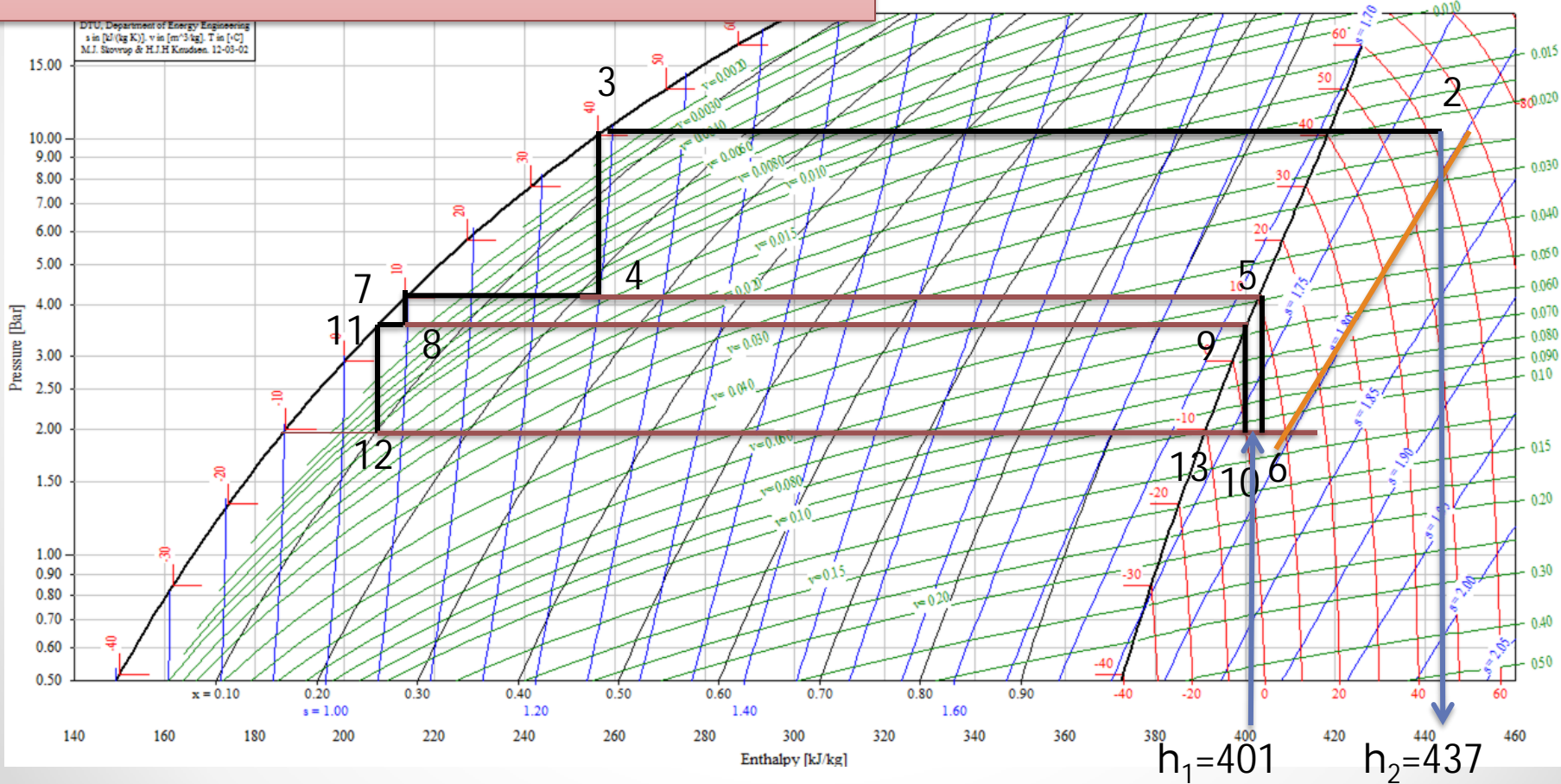


h	3	5	7	9	11	13	1	2	
kJ/kg	255	403	213	400	204	391	?	?	

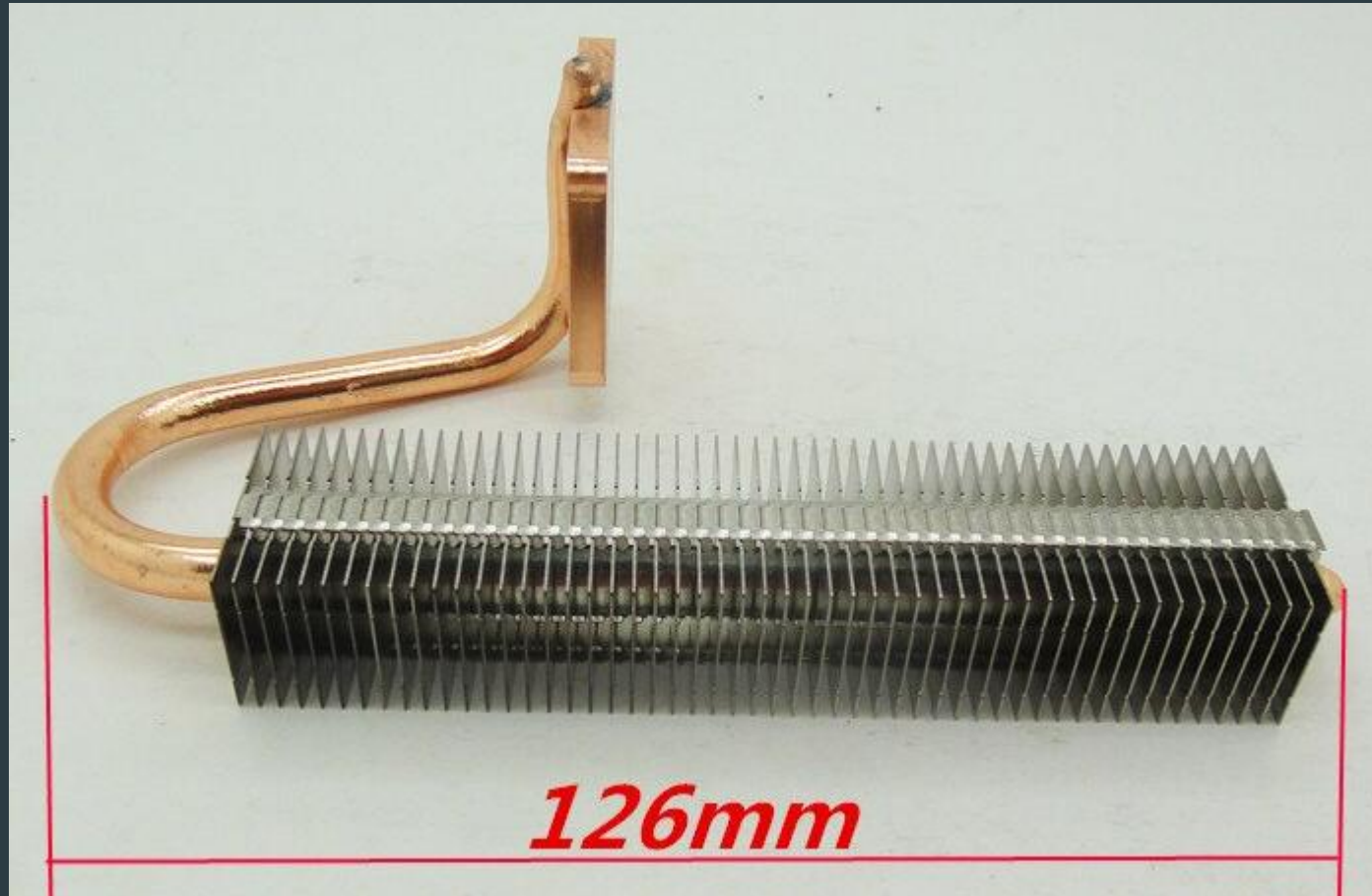
$$\text{Comp. Power} = m_4(h_2 - h_1)$$

$$= 1.44 \times (437 - 401) = 51.84 \text{ kW}$$

$$\text{Comp. HP} = 51.84 \times 1.34 = 69.5 \text{ HP}$$



Heat Pipes

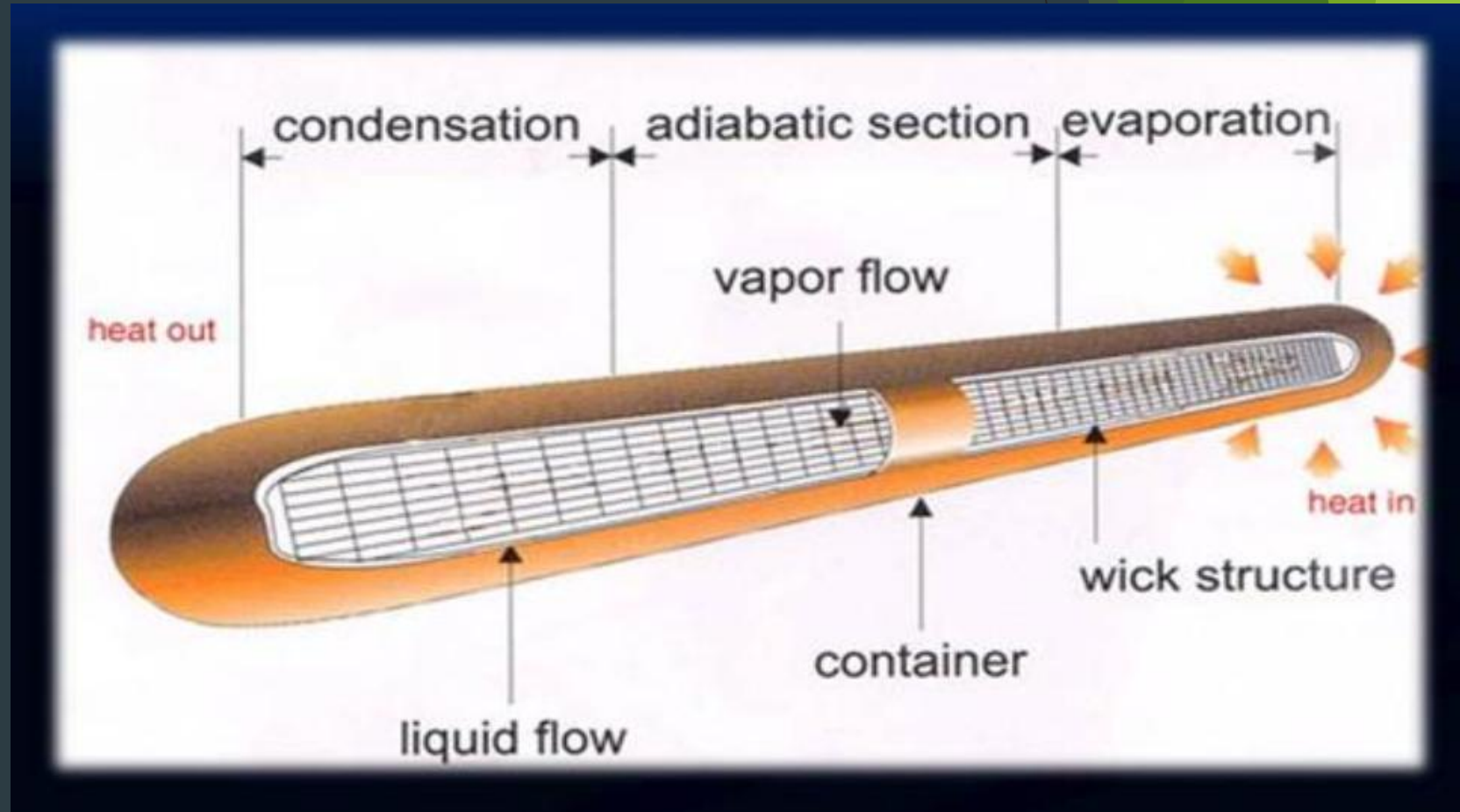


- ▶ A heat pipe is a simple device that can quickly transfer heat from one point to another. They are often referred to as the “superconductors” of heat as it has an extra heat transfer capacity. A heat pipe can transfer up to 1000 times more thermal energy than copper.

Components of a heat pipe

Heat pipe consist of three main components:

1. Container
2. Working fluid
3. Wick or Capillary structure



1- Container:

1. Metal tubing, with high thermal conductivity, usually copper or aluminum.
2. Shape of tubing can be bent or flattened.

2- Working fluid::

- Pure liquids such as helium, water.
- The type of liquid depends on the temperature range of the application.

Properties of working fluid

1. Good thermal conductivity.
2. Wettability of wick and wall material
3. Vapour pressure not too high or low over the operating temperature range
4. High latent heat
5. Low liquid and vapor viscosities
6. High surface tension
7. Acceptable freezing point

3- Wicking structure

The purpose of the wick is to generate capillary pressure to transport the working fluid from the condenser to the evaporator

Purpose of the wick

- Transports working fluid from the condenser to the evaporator.
- Provides liquid flow even against gravity.

How the wick works

- Liquid flows in a wick due to capillary action
- Intermolecular forces between the wick and the fluid are stronger than the forces within the fluid.
- Increase the capillary force and reduce the gravitational effects.



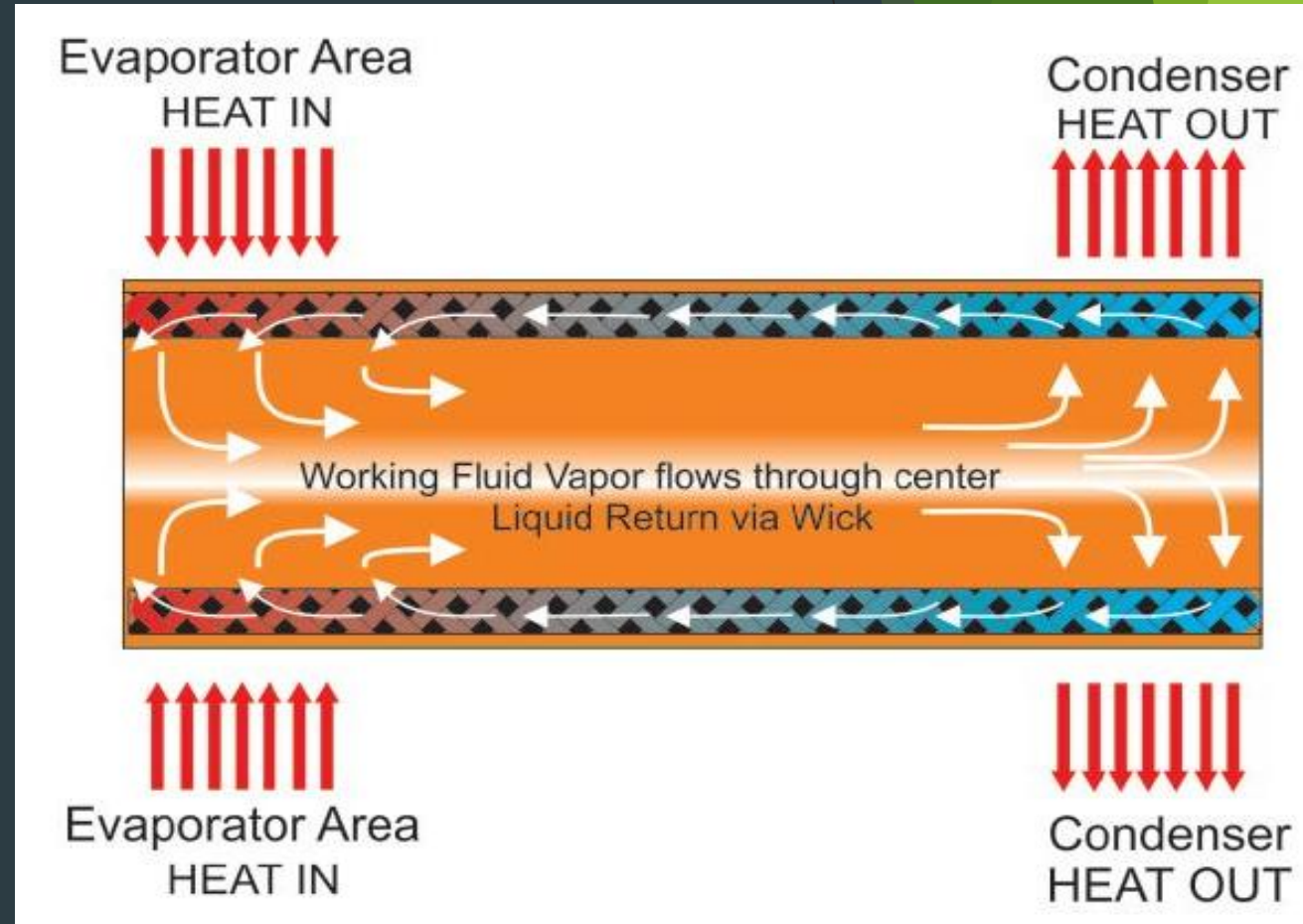
Working

A) Evaporator

- The working fluid is heated to its boiling point and converted into a vapor.
- Pressure and temperature differences forces pay the vapor to flow toward the cooler regions of the heat pipe.

B) Condenser

- Vapor gives up its latent heat of vaporization
- Vapor cools down and returns to its liquid state
- Working fluid then flows back toward the evaporator through the wick.



D) Heat exchanger

- The purpose of the heat exchanger is dissipates heat into environment
- Improve the thermal performance of heat exchangers by:
 1. Increase surface area with more fins.
 2. Using fan.



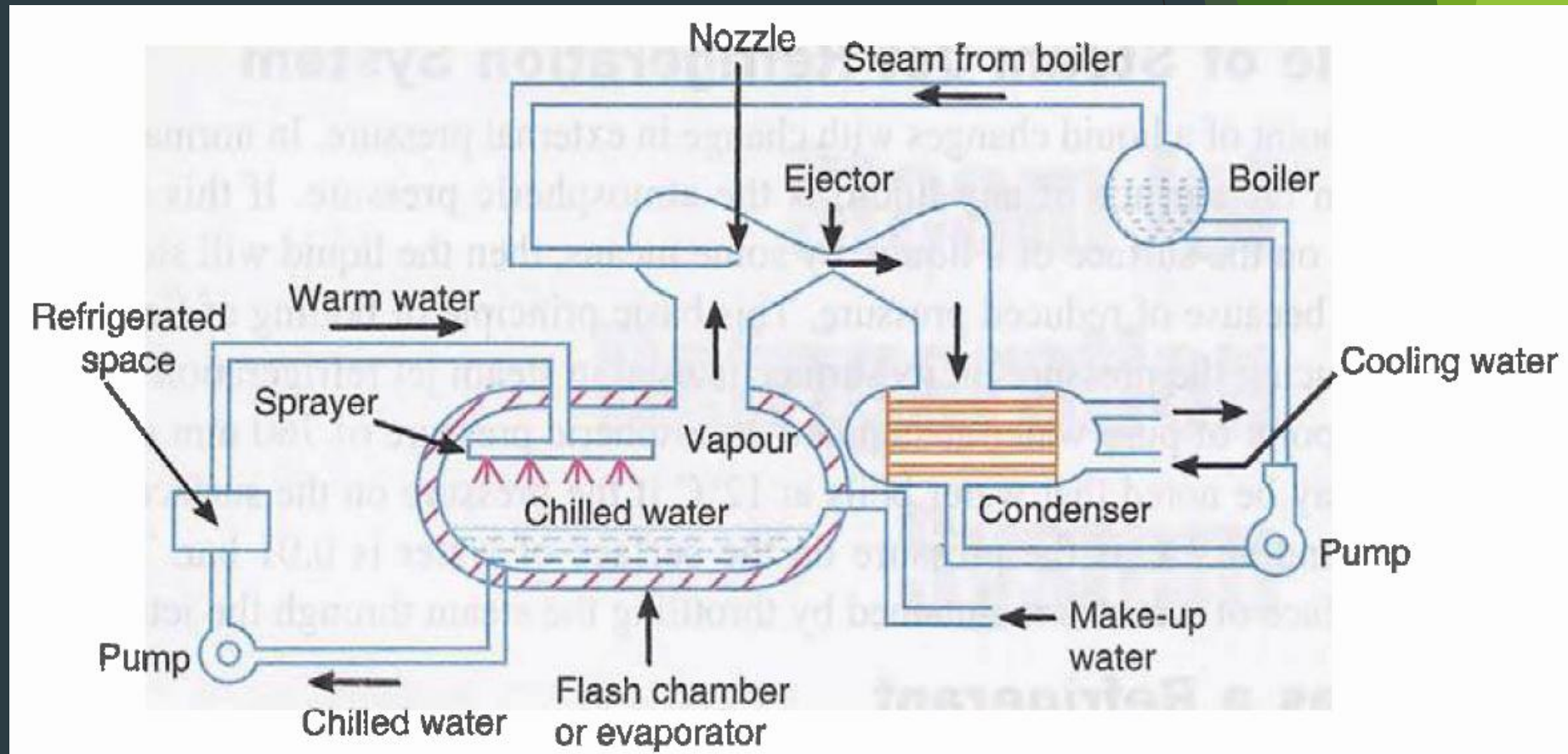
Steam Refrigeration System

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Introduction

- ▶ The steam jet refrigeration system is one of the oldest methods of producing refrigerant effect.
- ▶ In this system, water is used as the refrigerant, therefore cannot be used for applications below 0°C.
- ▶ The steam jet refrigeration system is widely used in food processing plants for precooling of vegetables etc.

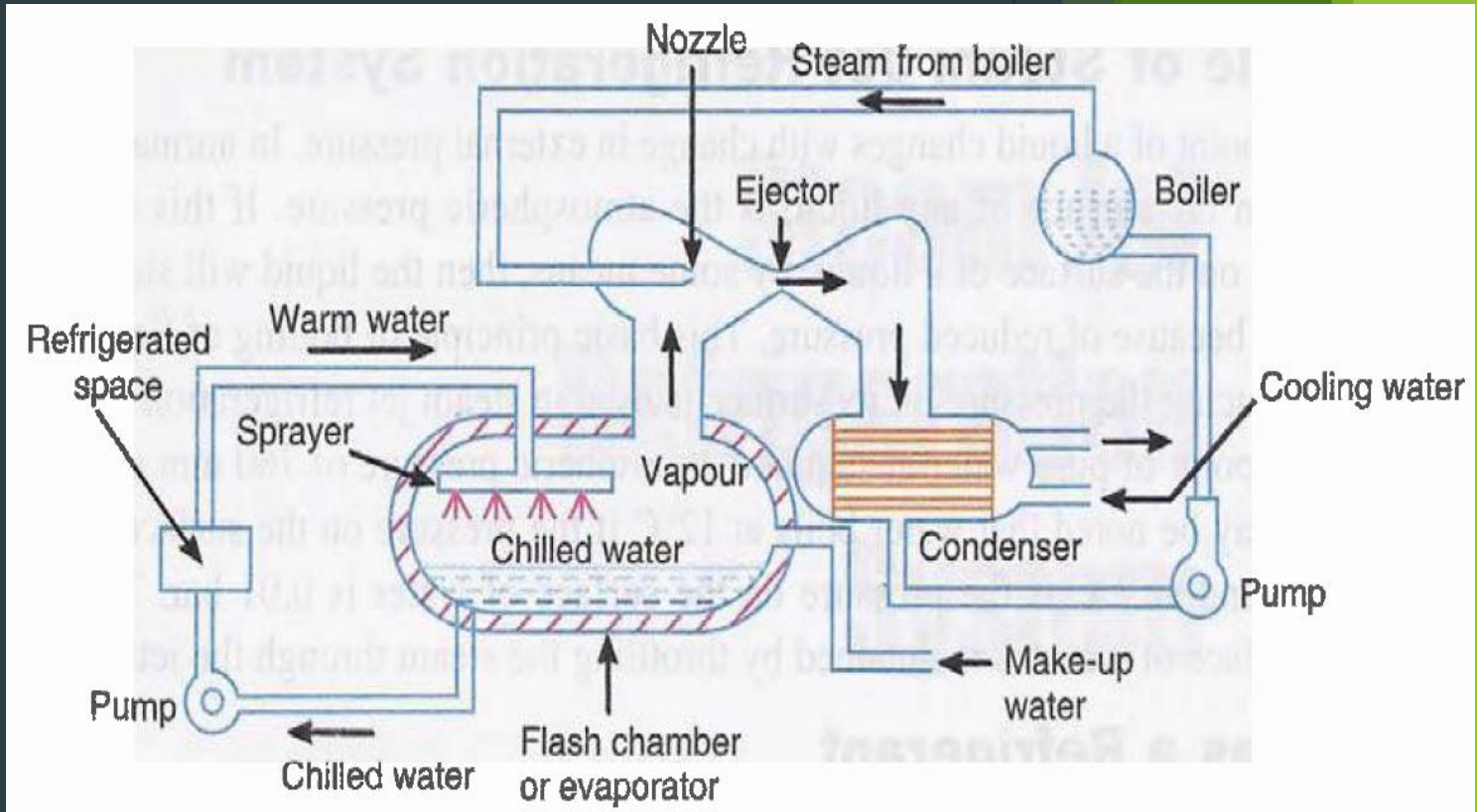
- ▶ The basic components of this system are: an evaporator, a compression device, a condenser, and refrigerant control device.
- ▶ This system employs a steam ejector (instead of mechanical compressor) to compress the refrigerant to the required condenser pressure level.



Principle work of system

The boiling point of pure water at standard atmospheric pressure of 76 cm of Hg (1.013 bar) is 100°C. It may be noted that water boils at 7°C if the pressure on the surface of water is kept at 0.01 bar.

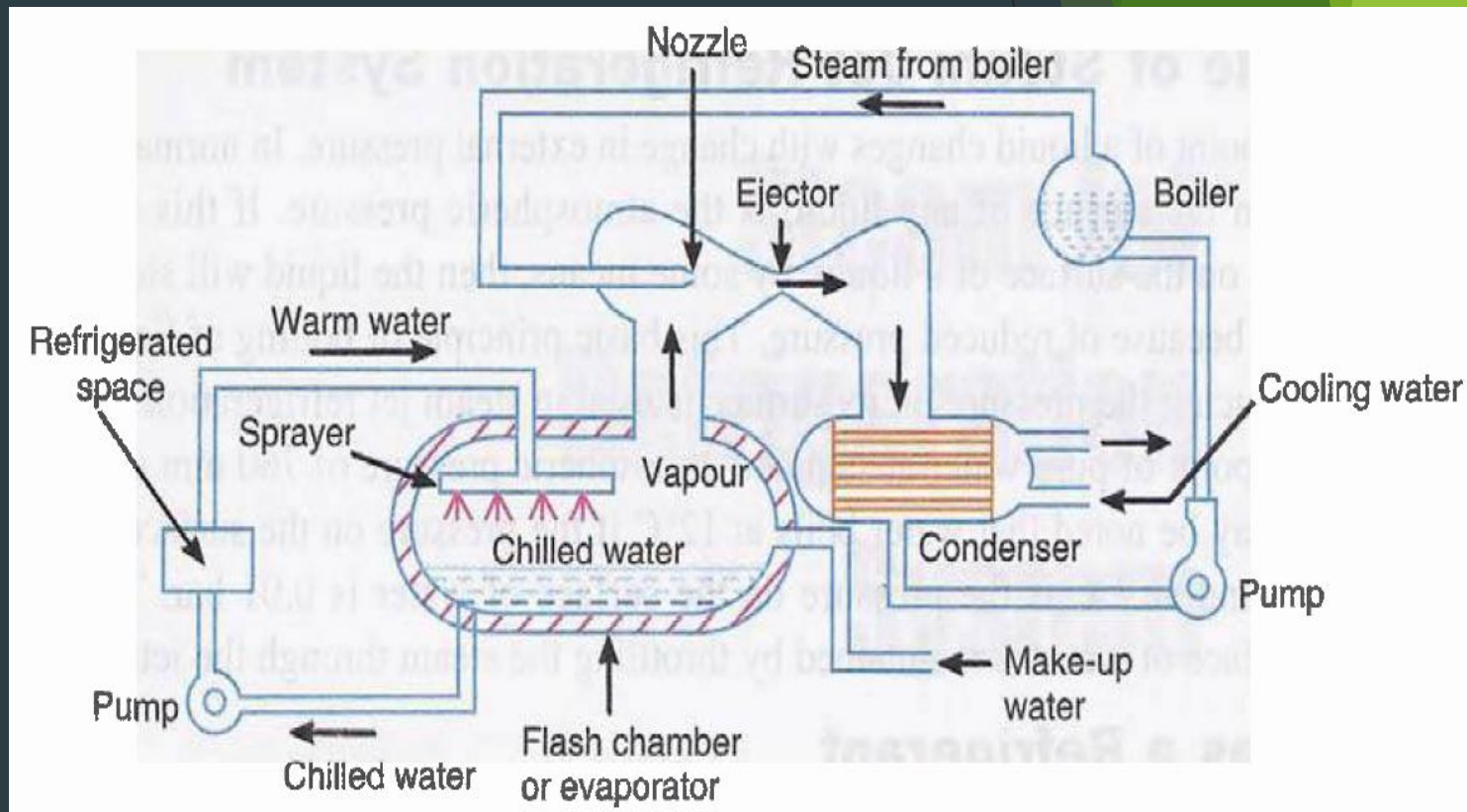
Therefore we can reduce the boiling point temperature of water by reducing the pressure on its surface.



Working of Steam Jet Refrigeration System

The main components of the steam jet refrigeration system are: Boiler, flash chamber or evaporator, steam nozzles, ejector and condenser. The main principles working can be explain by the following steps:

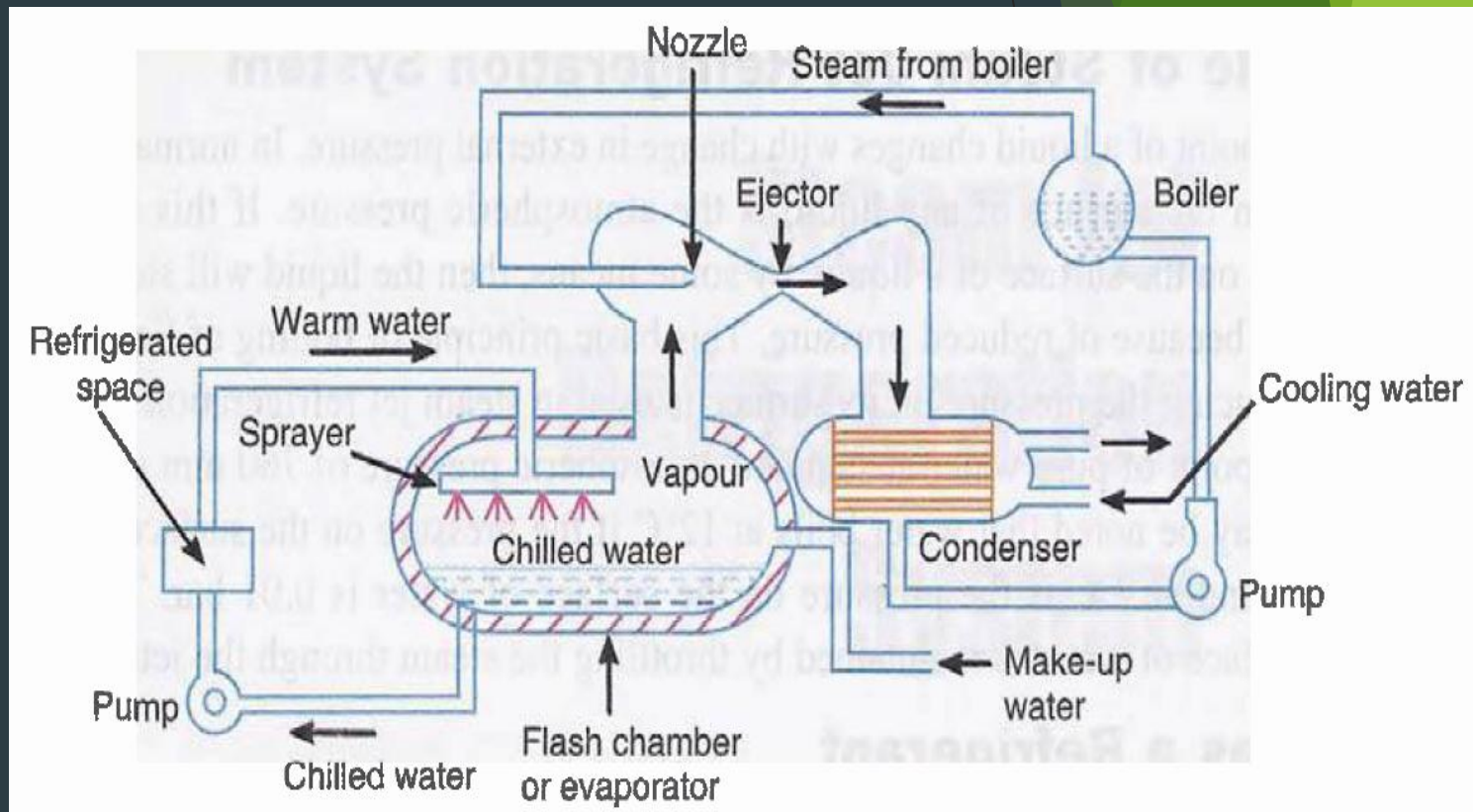
1. The warm water coming out from refrigerated space is sprayed into the flash water chamber where some of water is converted into vapor after absorbing the latent heat.



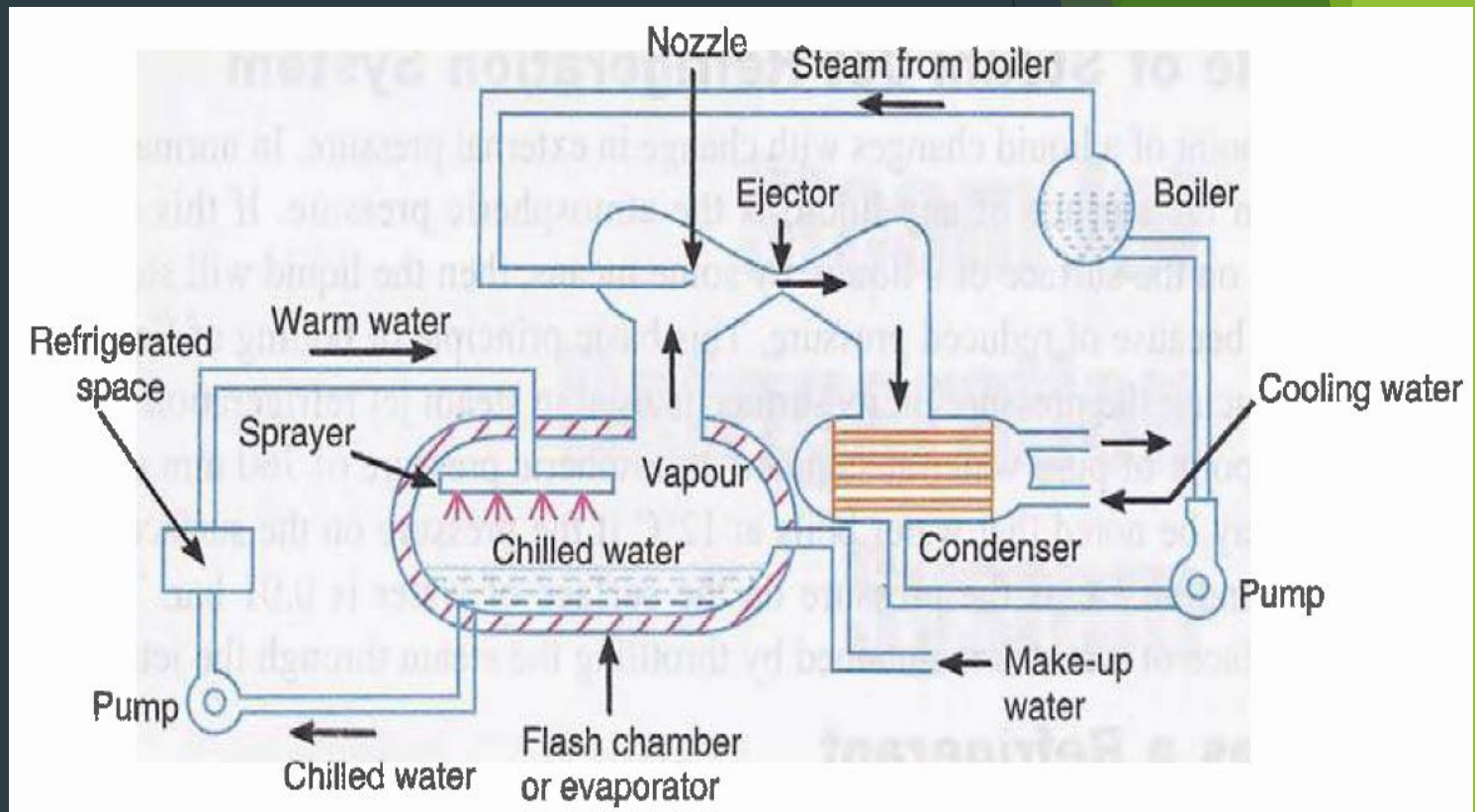
2- The high pressure steam from the boiler is passed through the steam nozzles causing increase in its velocity.

3- The high velocity of steam in the nozzle exit reduce the pressure that lead to entrain the water vapor from the flash chamber.

4- The mixture of steam and water vapor passes through the venturi-tube of the ejector and gets compressed.

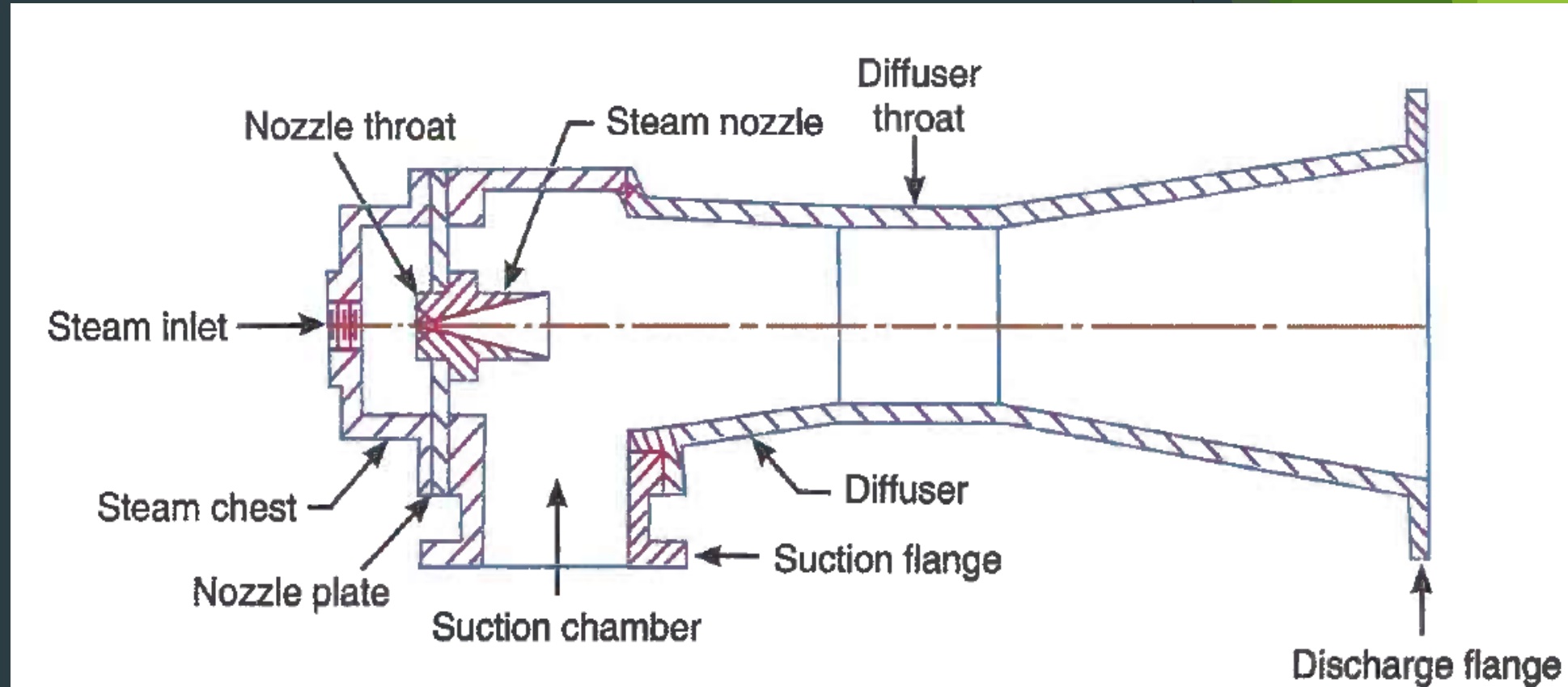


- 5- The temperature and pressure of the mixture rises and fed to the water cooled condenser where it gets condensed.
- 6- The condensate water is again feed to the boiler as feed water.
- 7- A make-up water is feed to the flash chamber to maintain a constant water level due to evaporation.



Steam ejector

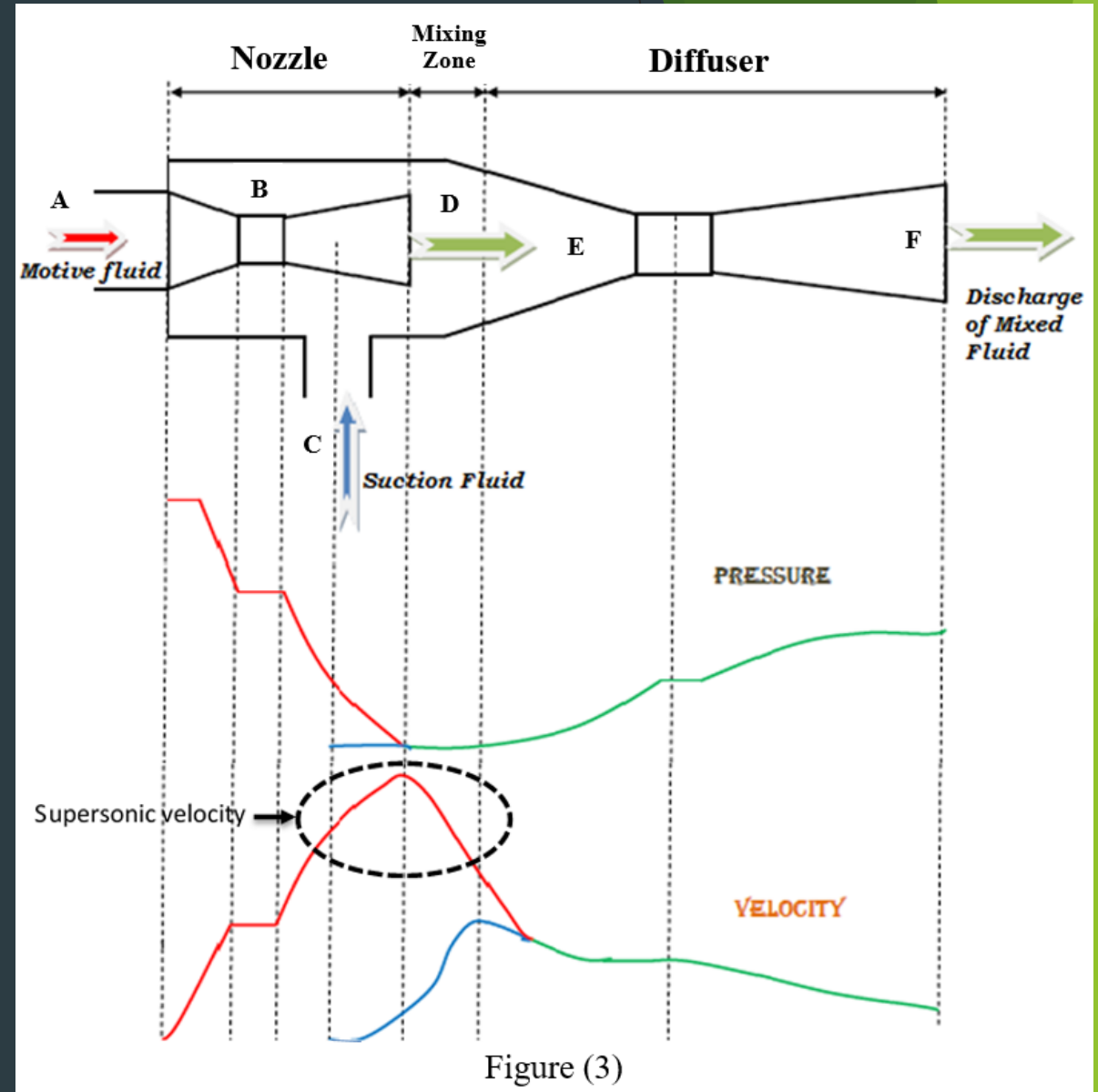
The steam ejector is one of the important components of a steam jet refrigeration system. It used the energy of fast moving jet of steam to entrain the vapors from the flash chamber and then compress it.



Steam ejector working:

1- The high pressure steam from the boiler expands while flowing through the convergent divergent nozzle. The expansion causes a very low pressure and increases the steam velocity.

2- The velocity range between 1000 m/s to 1350 m/s and the nozzle are designed for lowest operating pressure ratio of less than 200.



Steam ejector working:

3- The water vapors from the flash chamber are entrained by the high velocity of steam and both are mixed in the mixing section at constant pressure.

4- The mean velocity of the mixture will be supersonic .

5- The supersonic steam gets a normal shock in the constant area of the diffuser throat, that lead to rise the pressure and reduce the velocity to subsonic.

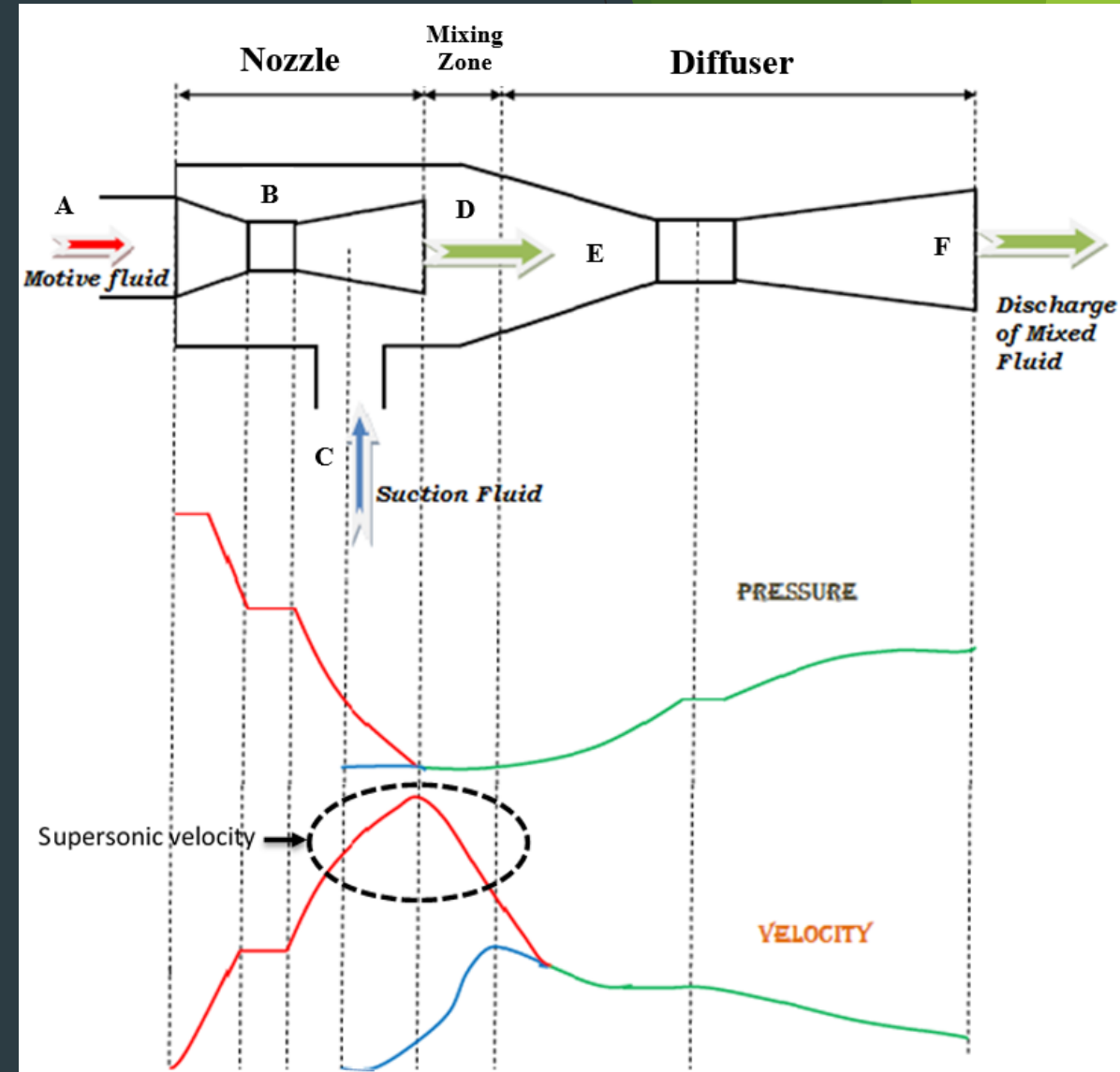
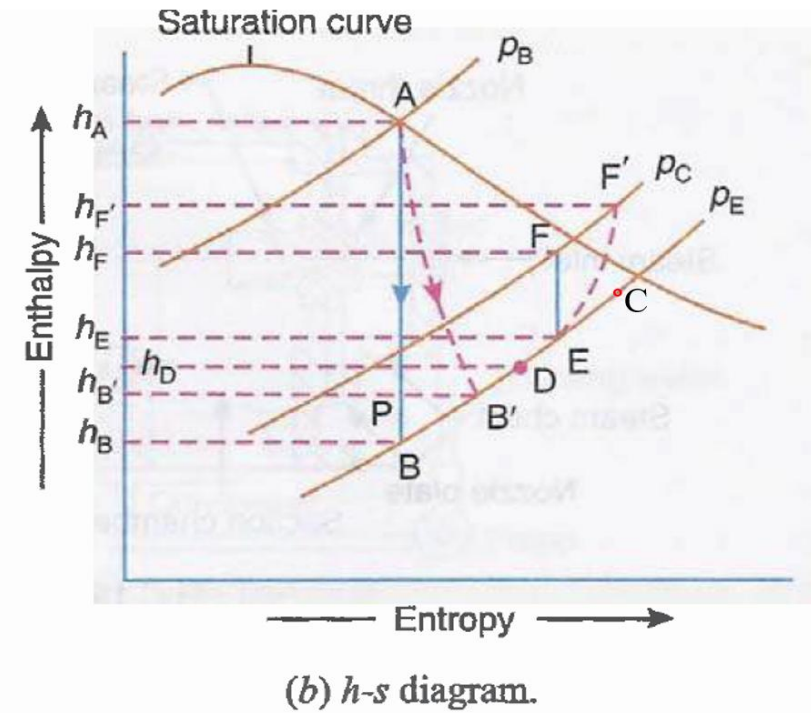
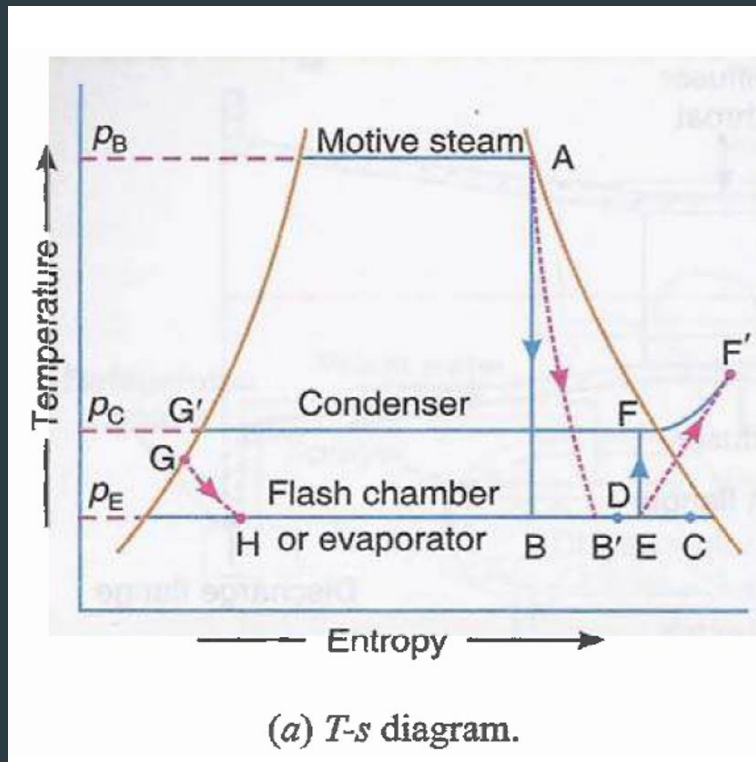


Figure (3)

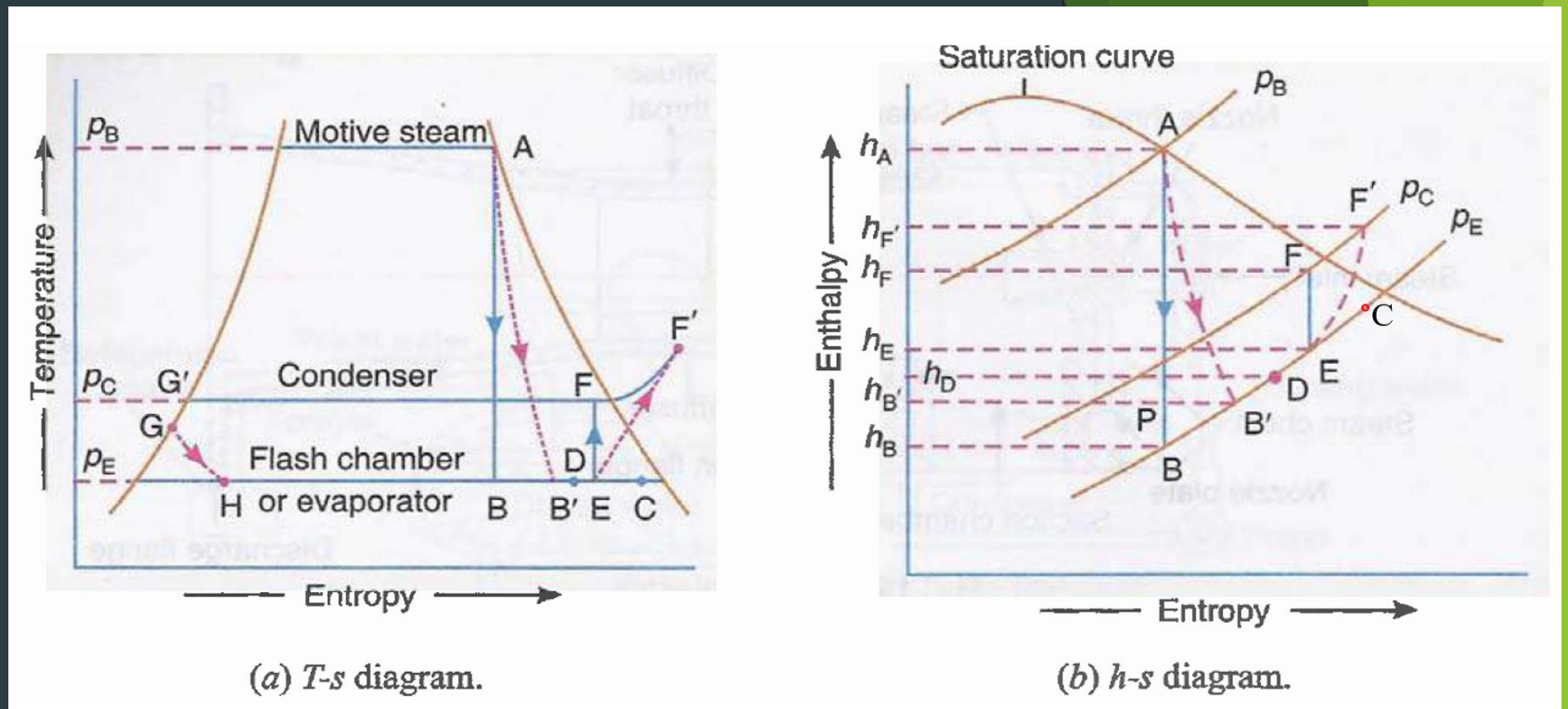
Analysis of system:

➤ The point A represents the initial condition of the motive steam before passing through the nozzle



- the point B is the final condition of the steam after nozzle, assuming isentropic expansion.
- The point C represents the initial condition of the water vapor in the flash chamber or evaporator.

➤ The condition of motive steam just before mixing with the water vapor is shown at point D.



(a) *T-s* diagram.

(b) *h-s* diagram.

➤ the point E is the condition of the mixture of high velocity steam from the nozzle and the entrained water vapor before compression.

➤ Assuming isentropic compression, the final condition of the mixture discharged to the condenser is represented by point F.

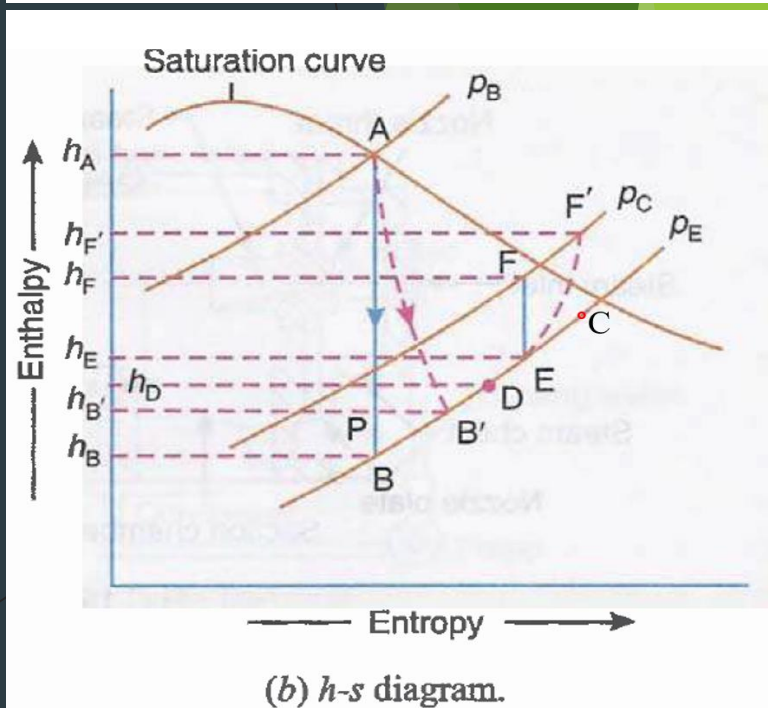
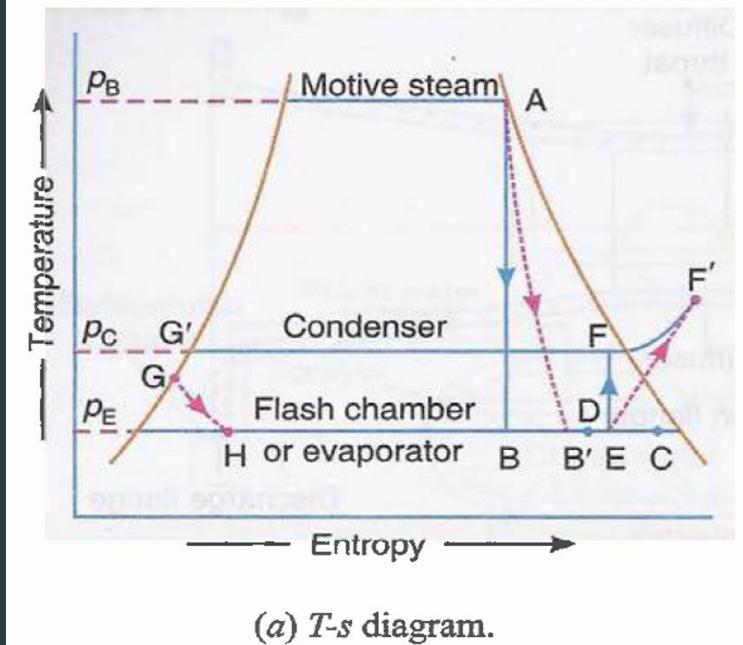
➤ The make-up water is supplied at point G whose temperature is slightly lower than the condenser temperature and is throttled to point H in the flash chamber.

1. Nozzle efficiency: $\eta_N = \frac{\text{actual enthalpy drop}}{\text{isentropic enthalpy drop}} = \frac{h_A - h_{B'}}{h_A - h_B}$

2. Entrainment efficiency: $\eta_E = \frac{h_A - h_D}{h_A - h_{B'}}$

3. Compression eff.: $\eta_C = \frac{\text{isentropic enthalpy increase}}{\text{actual enthalpy increase}} = \frac{h_F - h_E}{h_{F'} - h_E}$

4. Steam velocity from nozzle: $V = 44.72 \sqrt{h_A - h_{B'}}$



5. Mass of mixture for compression = Mass of motive steam + Mass of water vapor

$$m = m_s + m_v$$

6. Energy available for compression = $m_s(h_A - h_D)$

7. Energy required for compression = $m(h_{F'} - h_E)$

$$= (m_s + m_v)(h_{F'} - h_E)$$

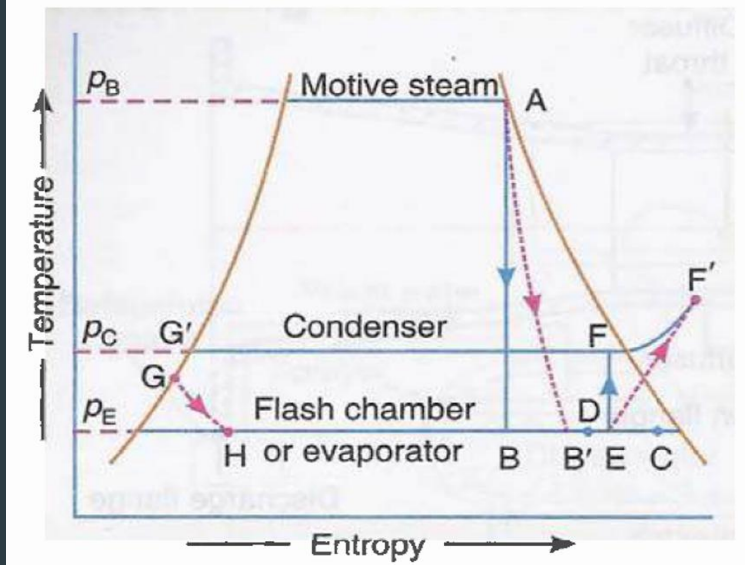
According to law of conservation energy:

$$m_s(h_A - h_D) = (m_s + m_v)(h_{F'} - h_E)$$

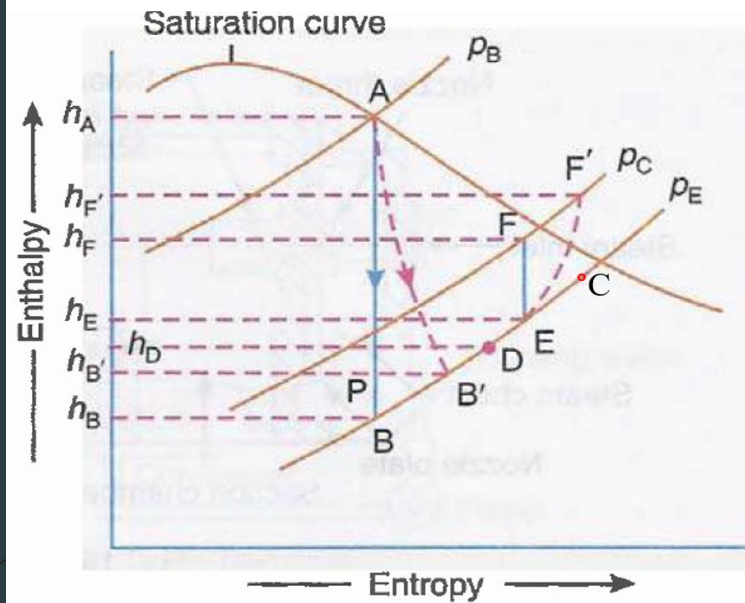
8. Net refrigerant effect: $R_E = m_v(h_C - h_{fG})$

9. The coefficient of performance of the system:

$$C.O.P = \frac{m_v(h_C - h_{fG})}{m_s(h_A - h_{fG})}$$



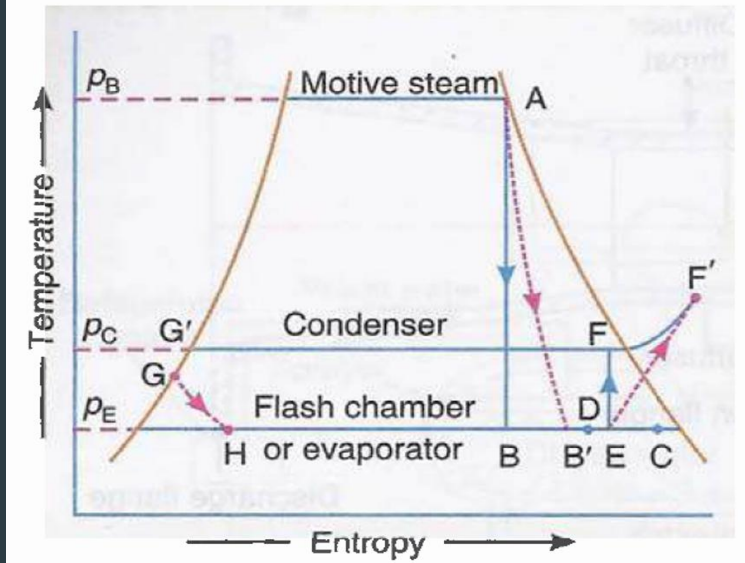
(a) $T-s$ diagram.



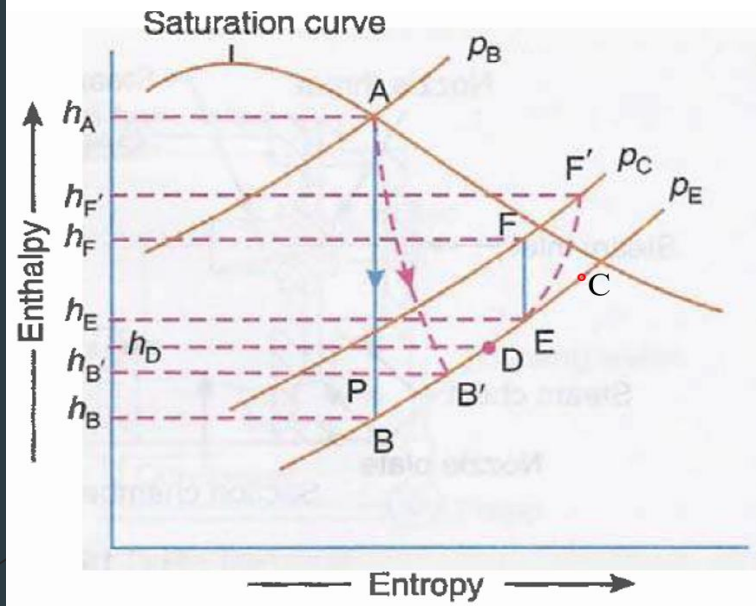
(b) $h-s$ diagram.

10. Mass of motive steam required per kg of water vapor produced in the flash chamber:

$$\frac{m_s}{m_v} = \frac{h_F - h_E}{\eta_N \eta_E \eta_C (h_A - h_B) - (h_F - h_E)}$$



(a) T-s diagram.



(b) h-s diagram.

Entropy before expansion (s_A)

= Entropy after expansion (s_B)

or

$$6.705 = s_{fB} + x_B \times s_{fgB} = 0.0685 + x_B \times 8.9715$$

\therefore

$$x_B = \frac{6.705 - 0.0685}{8.9715} = 0.74$$

and enthalpy at B,

$$h_B = h_{fB} + x_B \times h_{fgB} = 18.9 + 0.74 \times 2490.9 = 1862.16 \text{ kJ/kg}$$

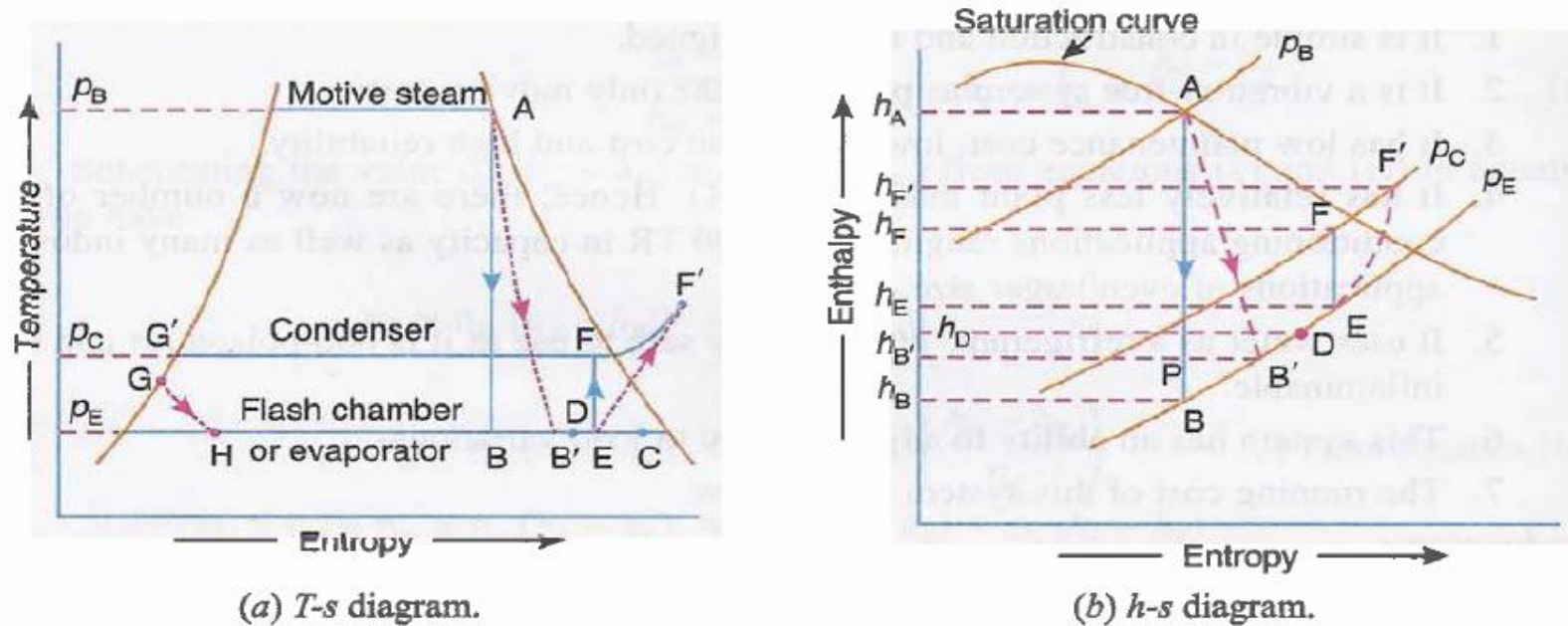


Fig. 15.4

We know that nozzle efficiency (η_N),

$$0.88 = \frac{h_A - h_{B'}}{h_A - h_B} = \frac{2762 - h_{B'}}{2762 - 1862.16}$$

\therefore

$$h_{B'} = 2762 - 0.88 (2762 - 1862.16) = 1970.14 \text{ kJ/kg}$$

and

$$h_{fB} = h_{fB'} = h_{fD} = h_{fE} = 18.9 \text{ kJ/kg}$$

$$h_{fgB} = h_{fgB'} = h_{fgD} = h_{fgE} = 2490.9 \text{ kJ/kg}$$

We know that entrainment efficiency (η_E),

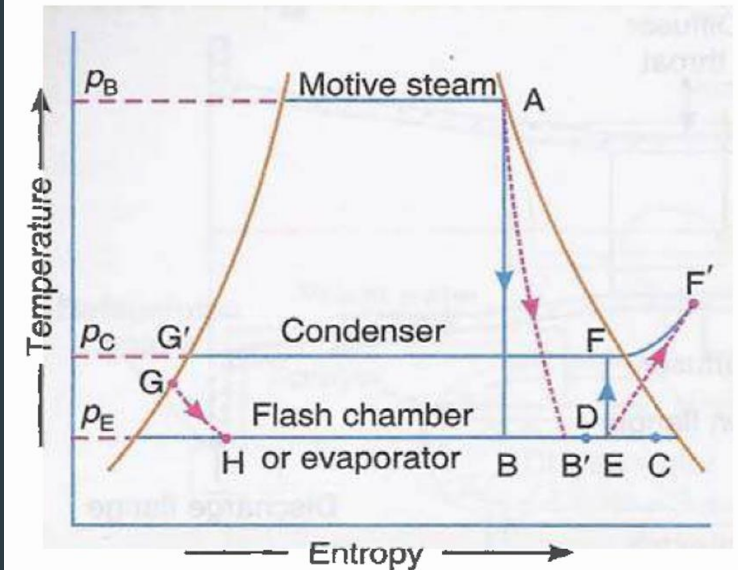
$$0.65 = \frac{h_A - h_D}{h_A - h_{B'}} = \frac{2762 - h_D}{2762 - 1970.14}$$

$$\therefore h_D = 2762 - 0.65 (2762 - 1970.14) = 2247.3 \text{ kJ/kg}$$

Enthalpy at point E,

$$\begin{aligned} h_E &= h_{fE} + x_E \times h_{fgE} = 18.9 + 0.92 \times 2490.9 \\ &= 2310.5 \text{ kJ/kg} \end{aligned}$$

... (\because It is given that $x_E = 0.92$)



(a) T-s diagram.

We know that entropy at point E ,

$$\begin{aligned} s_E &= s_{fE} + x_E \times s_{fgE} = 0.0685 + 0.92 \times 8.9715 \\ &= 8.3223 \text{ kJ/kg K} \quad \dots (\because s_{fE} = s_{fB} \text{ and } s_{fgE} = s_{fgB}) \end{aligned}$$

From steam tables, corresponding to a condenser pressure of 0.058 bar, we find that

$$h_{fF} = 148.86 \text{ kJ/kg}; \quad h_{fgF} = 2417.5 \text{ kJ/kg}$$

$$s_{fF} = 0.512 \text{ kJ/kg K}; \quad s_{fgF} = 7.831 \text{ kJ/kg K}$$

Since the compression of the mixture is isentropic, therefore

Entropy before compression (s_E)

= Entropy after compression (s_F)

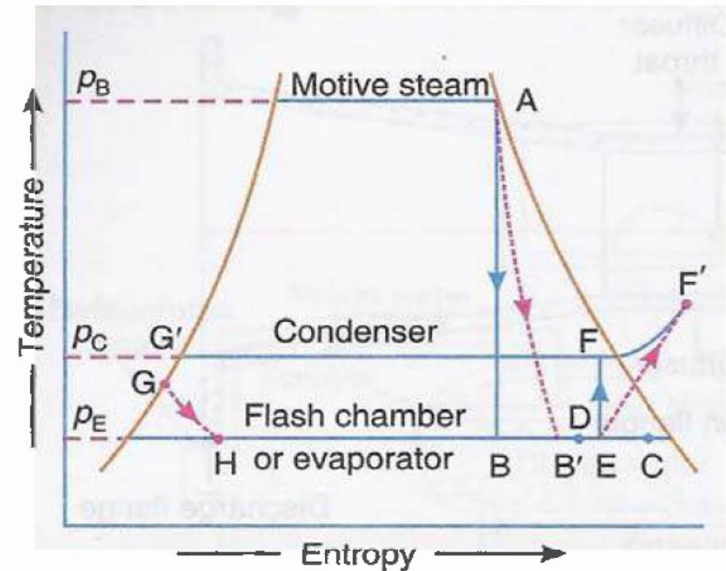
$$8.3223 = s_{fF} + x_F \times s_{fgF} = 0.512 + x_F \times 7.831$$

$$\therefore x_F = \frac{8.3223 - 0.512}{7.831} = 0.997$$

We know that enthalpy at point F ,

$$h_F = h_{fF} + x_F \times h_{fgF} = 148.86 + 0.997 \times 2417.5 = 2559.1 \text{ kJ/kg}$$

We also know that compression efficiency (η_C),



(a) T - s diagram.

$$0.8 = \frac{h_F - h_E}{h_{F'} - h_E} = \frac{2559.1 - 2310.5}{h_{F'} - 2310.5}$$

$$\therefore h_{F'} = \frac{2559.1 - 2310.5}{0.8} + 2310.5 = 2621.2 \text{ kJ/kg}$$

1. Mass of motive steam required per kg of the flash vapour

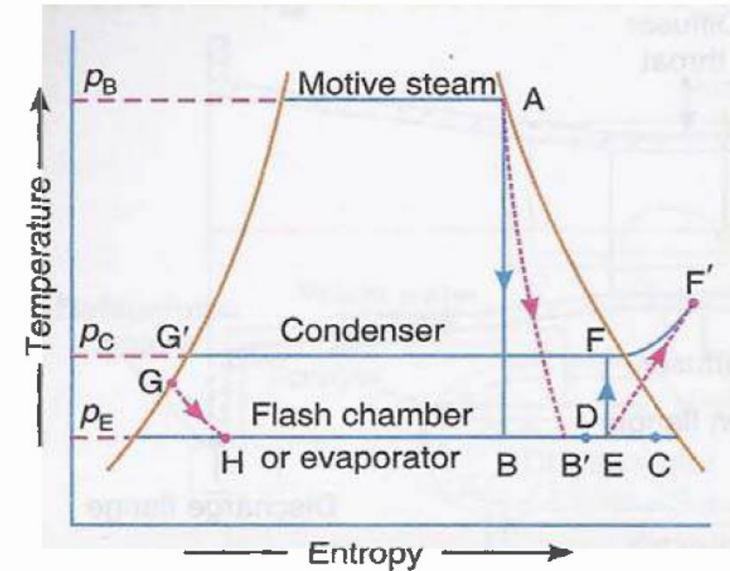
We know that mass of motive steam required per kg of the flash vapour,

$$\begin{aligned} \frac{m_s}{m_v} &= \frac{h_F - h_E}{(h_A - h_B)\eta_N\eta_E\eta_C - (h_F - h_E)} \\ &= \frac{2559.1 - 2310.5}{(2762 - 1862.16)0.88 \times 0.65 \times 0.8 - (2559.1 - 2310.5)} \\ &= \frac{248.6}{411.8 - 248.6} = 1.523 \text{ kg/kg of flash vapour} \end{aligned}$$

2. Quality of vapour flashed from the flash chamber

Let x_C = Dryness fraction of the vapour flashed from the flash chamber.

First of all, let us find the enthalpy at point C. We know that



(a) T-s diagram.

Ans.

$$m_v h_C + m_s h_D = (m_s + m_v) h_E$$

$$h_C + \frac{m_s}{m_v} \times h_D = \left(\frac{m_s}{m_v} + 1 \right) h_E$$

$$h_C + 1.523 \times 2247.3 = (1.523 + 1) 2310.5$$

$$h_C + 3422.6 = 5829.4$$

$$\therefore h_C = 2406.8 \text{ kJ/kg}$$

We also know that enthalpy at point C (h_C),

$$2406.8 = h_{fC} + x_C \times h_{fgB} = 18.9 + x_C \times 2490.9$$

$$\dots (\because h_{fC} = h_{fB} \text{ and } h_{fgC} = h_{fgB})$$

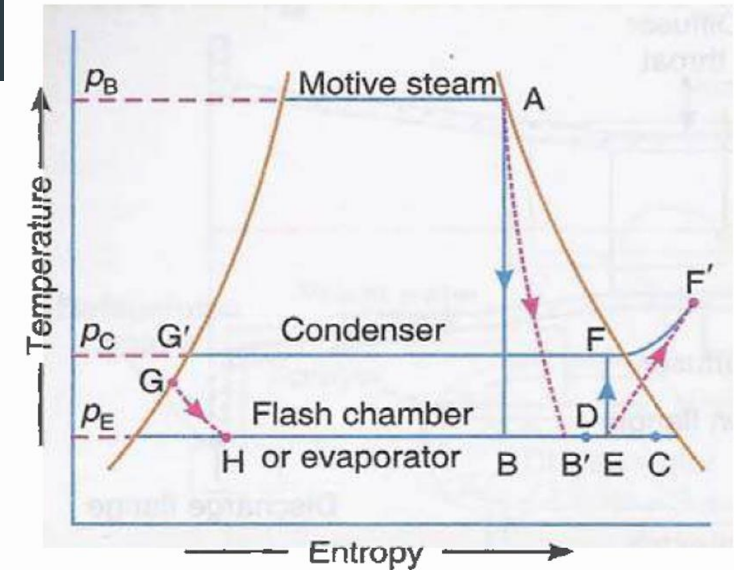
$$\therefore x_C = \frac{2406.8 - 18.9}{2490.9} = 0.96 \text{ Ans.}$$

3. Refrigerating effect per kg of flash vapour

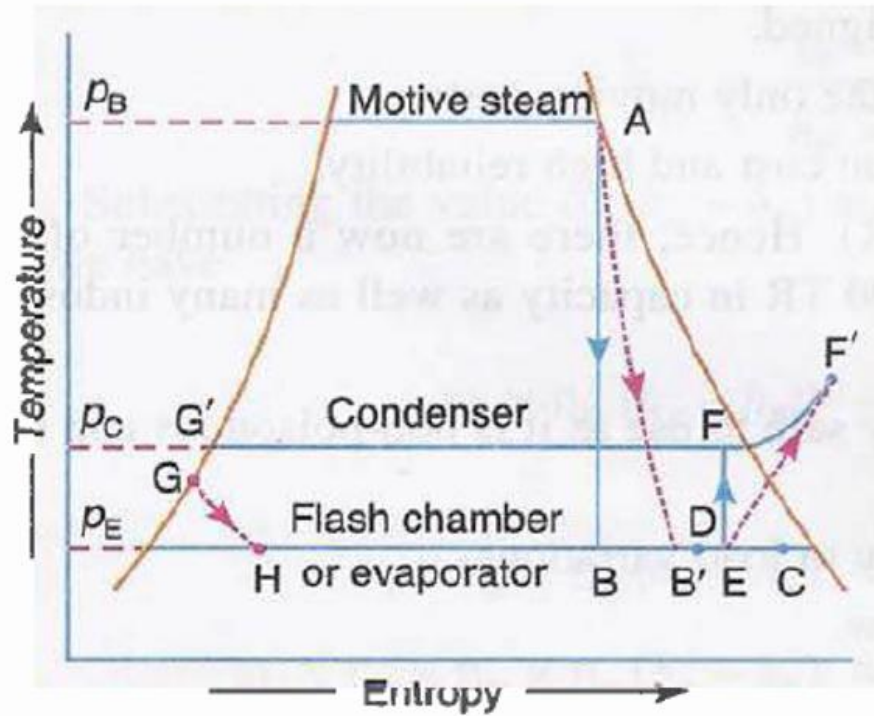
We know that refrigerating effect per kg of flash vapour,

$$R_E = h_C - h_{fG} = 2406.8 - 75.5 = 2331.3 \text{ kJ/kg}$$

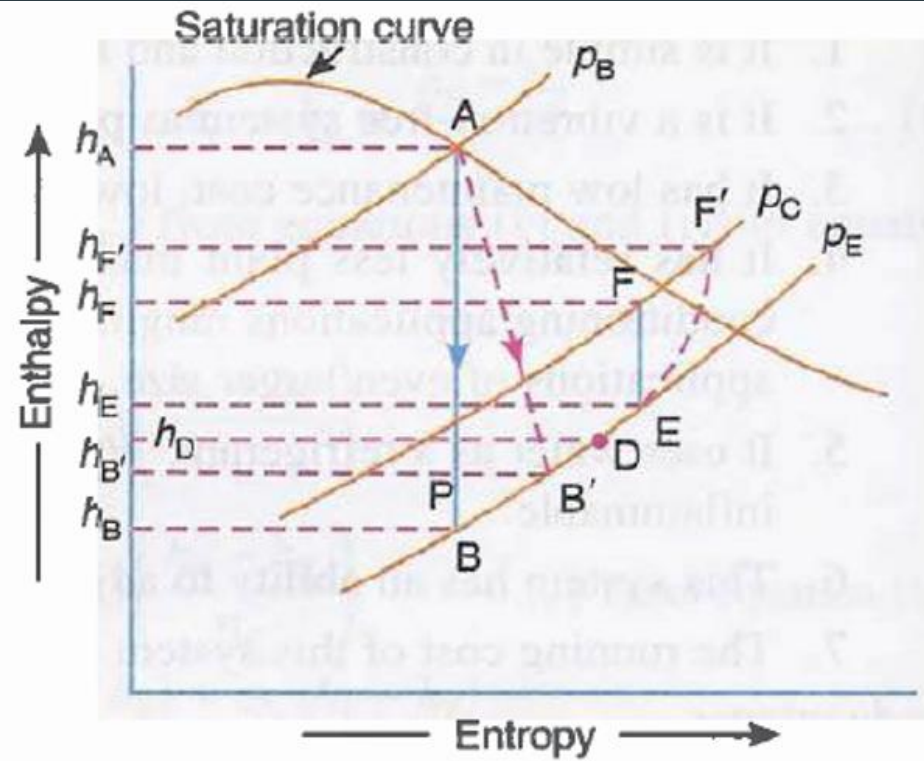
$$\dots (\because \text{From steam tables, } h_{fG} \text{ at } 18^\circ\text{C} = 75.5 \text{ kJ/kg})$$



(a) T - s diagram.



(a) $T-s$ diagram.



(b) $h-s$ diagram.

4. Coefficient of performance of the system

From steam tables, corresponding to a condenser pressure of 0.058 bar, we find that enthalpy of liquid at point G' ,

$$h_{fG'} = 148.8 \text{ kJ/kg}$$

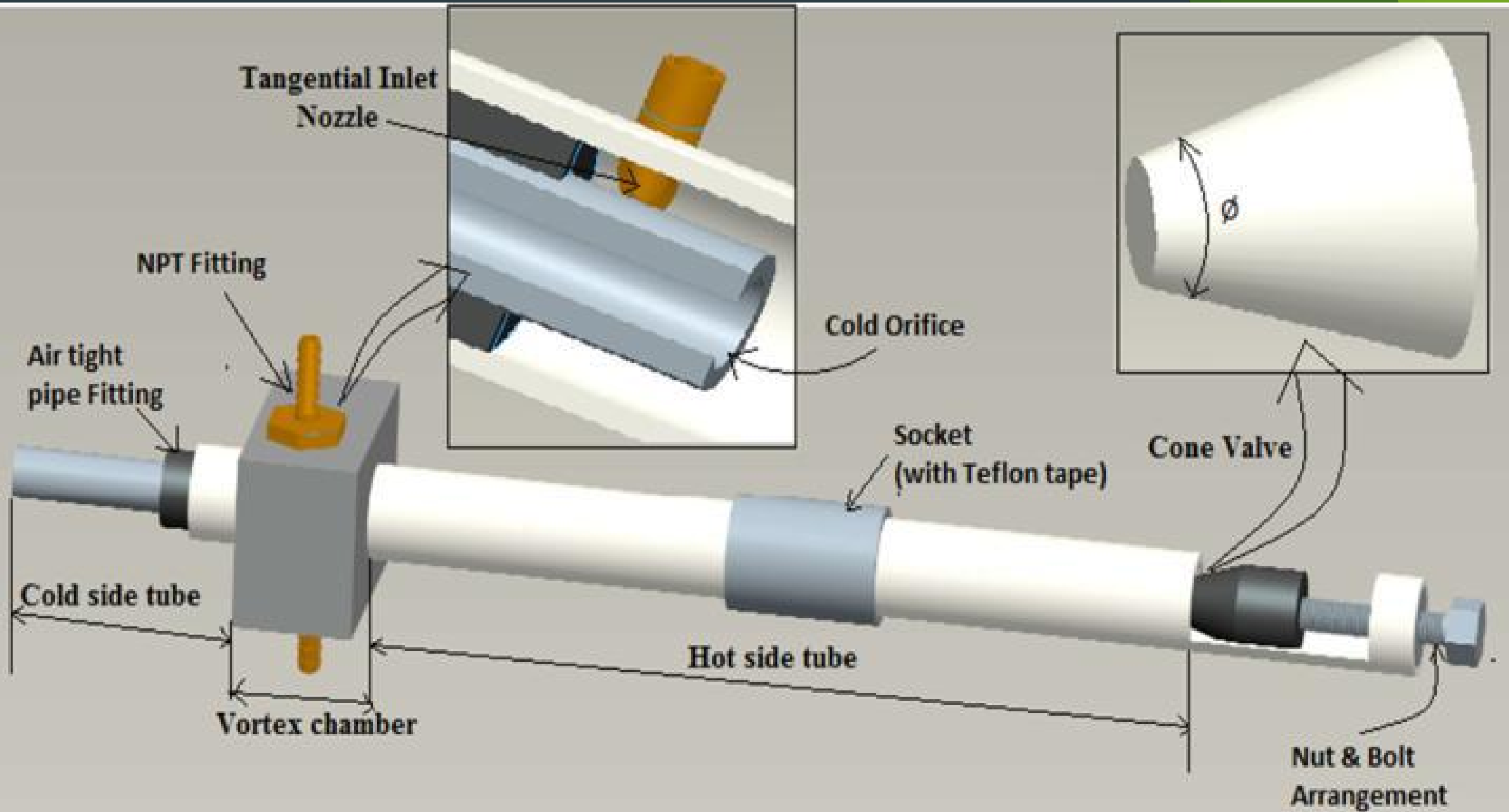
We know that coefficient of performance of the system,

$$\text{C.O.P.} = \frac{m_v (h_C - h_{fG'})}{m_s (h_A - h_{fG'})} = \frac{1 (2406.8 - 75.5)}{1.523 (2762 - 148.8)} = 0.586 \quad \text{Ans.}$$

Vortex Tube Refrigeration

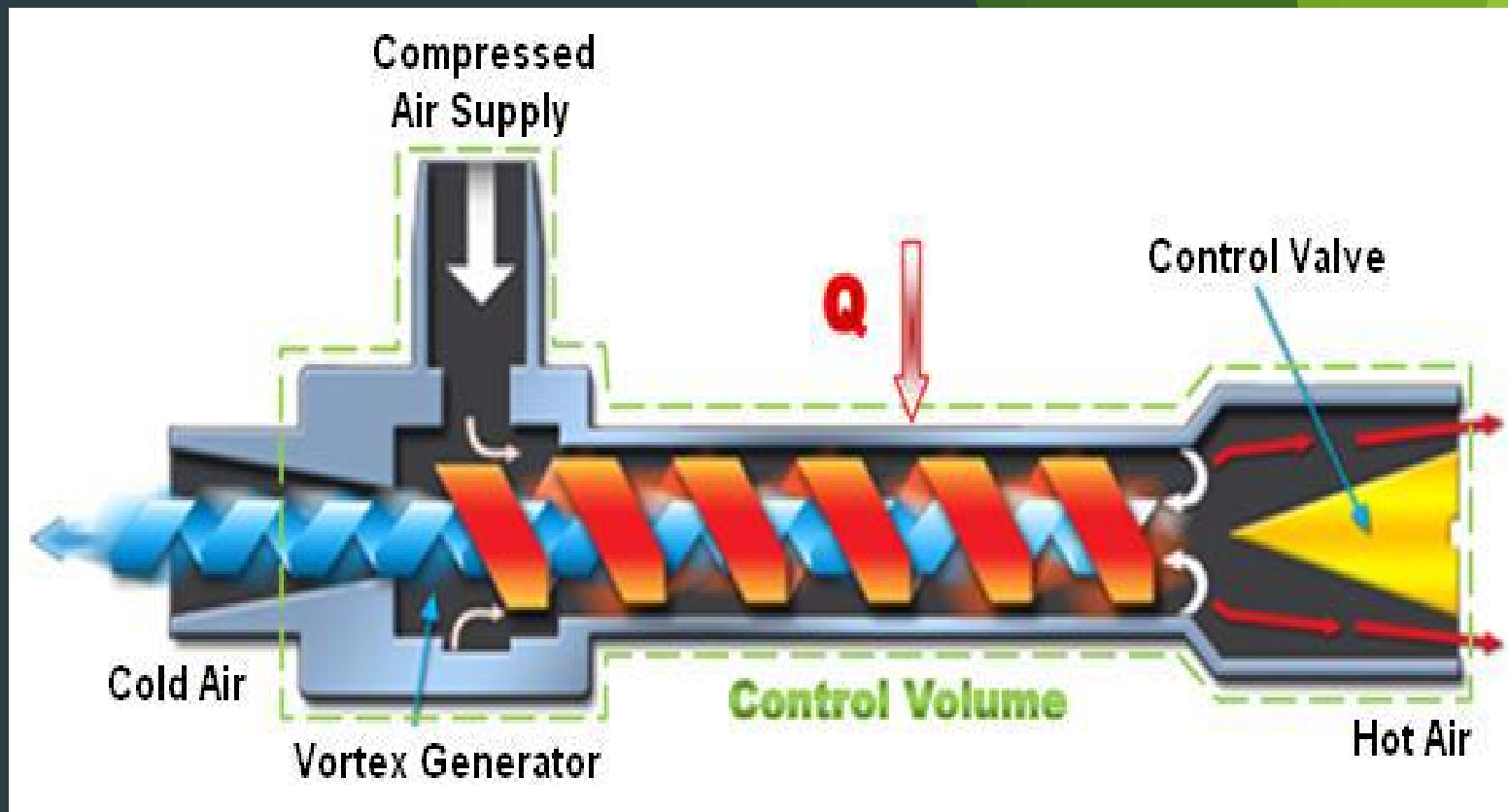
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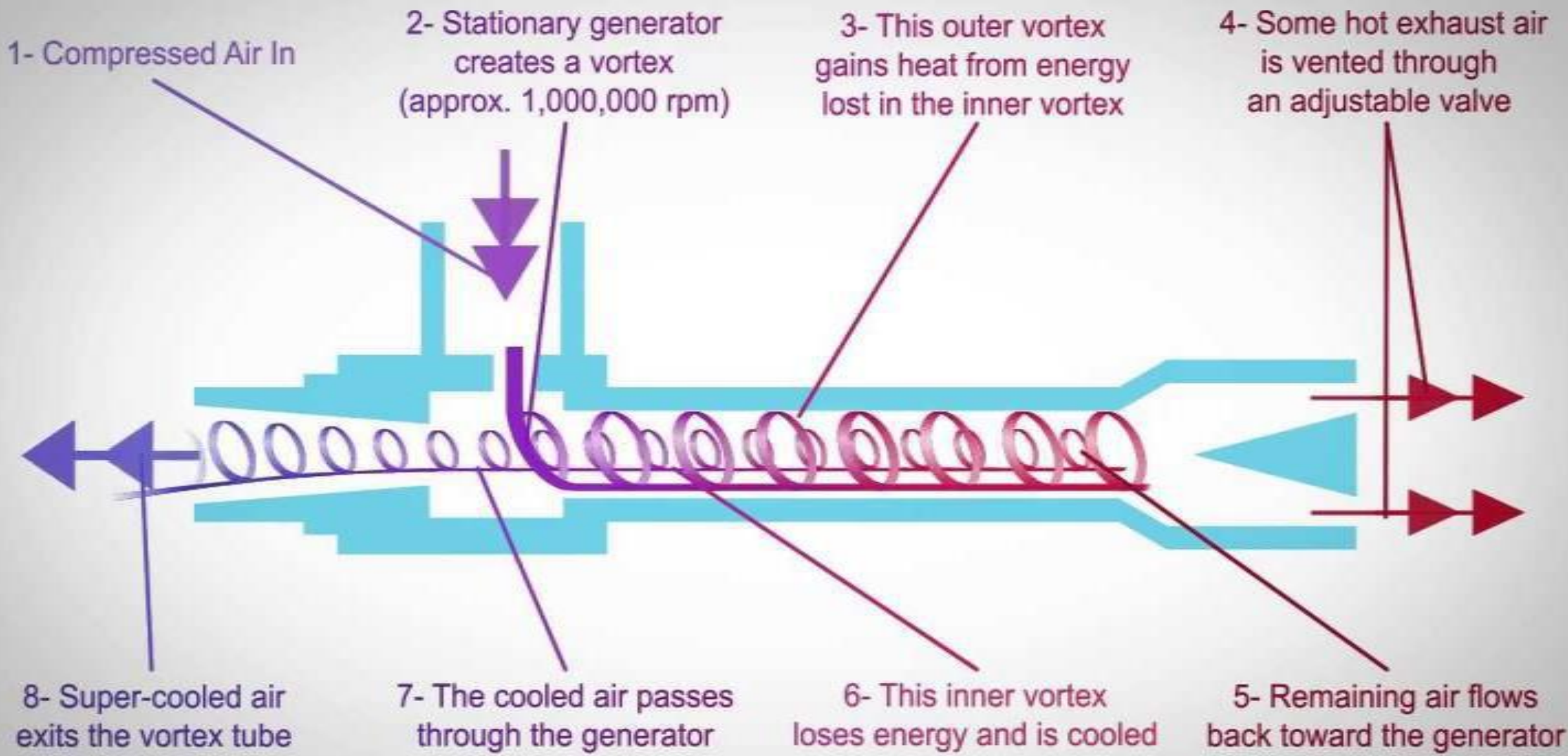
- ▶ It is one of the non-conventional type refrigerating systems.
- ▶ It consists of tangential nozzle, valve, Diaphragm, hot-air side, cold-air side in addition to a compressed air supply.
- ▶ Chamber is a portion that converted the air motion into spiral form.
- ▶ Hot side is cylindrical in cross section and is of different lengths as per design.
- ▶ Arrangement valve to controls the quantity of hot air through vortex tube.
- ▶ Diaphragm is a cylindrical piece of small thickness and having a small diameter at the center which the cold air is passed through.



Working:

1. Compressed air enter the nozzle and expands to high velocity
2. A vortex flow creates in the chamber
3. Air travels in spiral motion along the periphery of the hot side.
4. This flow controls by the valve.
5. Reversed axial flow through the core of the hot side starts from control valve region to the back region.
7. During this process, heat transfer takes place between reversed stream
8. The cold stream is escaped through the diaphragm hole into the cold side





1- Compressed Air In

2- Stationary generator creates a vortex (approx. 1,000,000 rpm)

3- This outer vortex gains heat from energy lost in the inner vortex

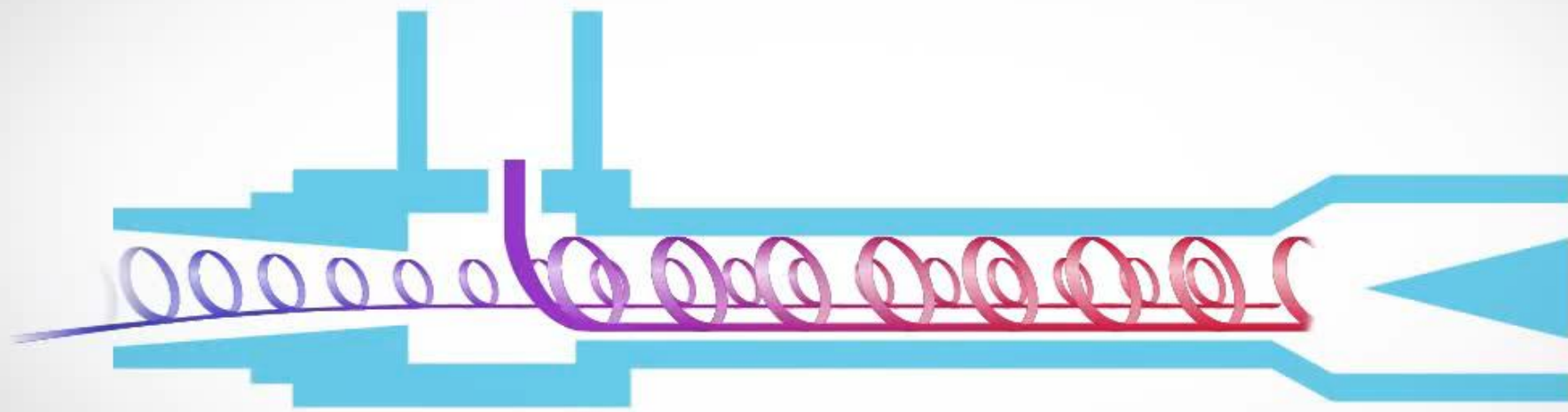
4- Some hot exhaust air is vented through an adjustable valve

8- Super-cooled air exits the vortex tube

7- The cooled air passes through the generator

6- This inner vortex loses energy and is cooled

5- Remaining air flows back toward the generator



Advantages

1. It uses air as refrigerant, so there is no leakage problem.
2. Vortex tube is simple in design and it avoids control systems.
3. There are no moving parts in vortex tube.
4. It is light in weight and requires less space.
5. Initial cost is low and its working cost are also less, where compressed air is available.
6. Temperature as low as $-50\text{ }^{\circ}\text{C}$ can be obtained without any difficulty, so it is very much useful in industries for spot cooling of electronic components.
7. Maintenance is simple.

Disadvantages

1. It has low coefficient of performance COP.
2. It has limited capacity and only small portion of the compressed air appearing as the cold air
3. Only used for spot cooling purpose.

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