Condensers

Introduction: The condenser is an important device used in the high pressure side of a refrigeration system. Its function is to remove heat of the hot vapour refrigerant discharged from the compressor.

The source of heat in the refrigerant cycle come from:
1- Heat absorbed by the evaporator.
2- Heat added by the mechanical energy of the compressor motor.

The heat from the hot vapour refrigerant in a condenser is removed first by transferring it to the walls of the condenser tubes and then from the tubes to the condensing or cooling medium. The cooling medium may be air or water or a combination of
the two.

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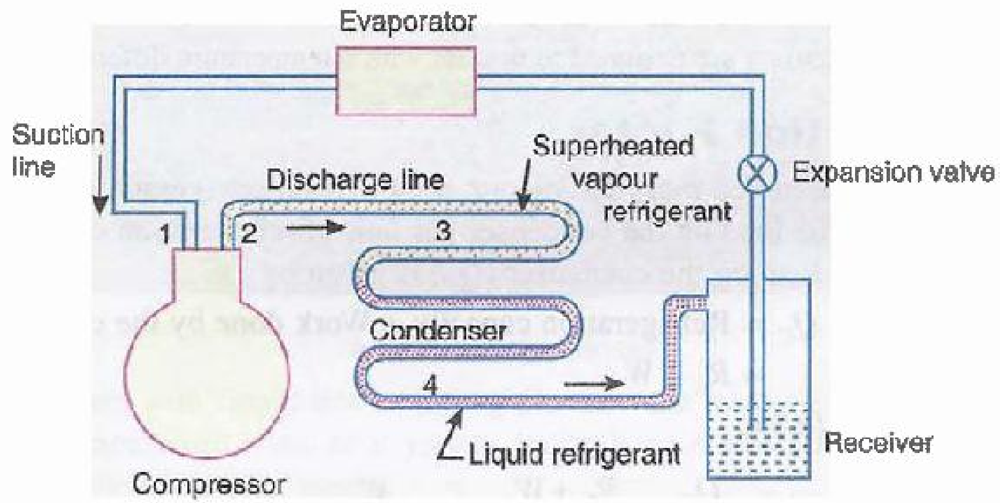
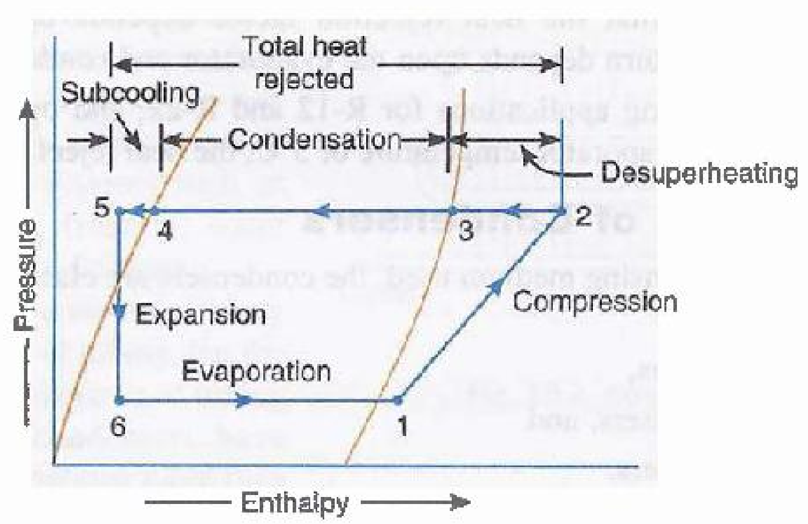
Working of an Evaporator:

1. The compressor draws in the superheated vapour refrigerant that contains the heat it absorbed in the evaporator.
2. The compressor adds more heat.
3. highly superheated vapour from the compressor is pumped to the condenser through the discharge line.
4. The superheated vapour is cooled to saturation temperature (called desuperheating)
5. The saturated vapour refrigerant gives up its latent heat and is condensed to a saturated liquid refrigerant. This process, called condensation.
6. the liquid refrigerant is reduced below its saturation temperature (called sub-cooled).



(a) Schematic diagram of a simple refrigerating system.



(b) p-h diagram of a simple refrigerating system.

Factors Affecting the Condenser Capacity

The heat transfer capacity of a condenser depends upon following factors :

1. Material. It may noted that higher the ability of a material to transfer heat, the smaller will be the size of condenser

2. Amount of contact. The condenser capacity may be varied by the amount contact between the condenser surface and the condensing medium. This can be done by varying the surface area of the condenser and the rate of flow of the condensing medium over condenser surface.

3. Temperature difference. As temperature difference increases, the heat transfer rate increases and therefore the condenser capacity increases.

Note: Most air-cooled condensers are designed to operate with a temperature difference of 14° C.

Classification of Condensers

According to the condensing medium used, the condensers are classified into the following three groups :

1- Air-cooled condensers,

- Natural convection air-cooled conde11sers.
- Forced convection air-cooled conde11sers.

2-Water-cooled condensers

- Tube-in-tube or double-tube condensers: water tube inside a large refrigerant tube.
- Shell and coil co11de11sers: The hot vapour refrigerant enters at the top of the shell and drops to the bottom of the shell while the cool water flow inside the coils.
- Shell and tube condensers: The vapour refrigerant enters at the top of the shell and the cool water flow inside the tubes.

3- Evaporative condensers.

The evaporative condensers use both air and water as condensing mediums to condense the hot vapour refrigerant to liquid refrigerant.

Comparison of Air-Cooled and Water-Cooled Condensers

No.		
1.	The construction of air-cooled condenser is very simple, therefore the initial cost is less.	The construction of water-cooled condenser is complicated, therefore the initial cost is high.
2.	The maintenance cost is low.	The maintenance cost is high.
3.	There is no corrosion, and fouling effect is low	Corrosion occurs inside the tubes carrying the water, and fouling effects are high.
4.	The air-cooled condensers have low heat transfer capacity due to low thermal conductivity of air.	The water-cooled condensers have high heat transfer capacity due to high thermal conductivity of water
5.	These condensers are used for low capacity plants (less than 5 TR).	These condensers are used for large capacity plants.

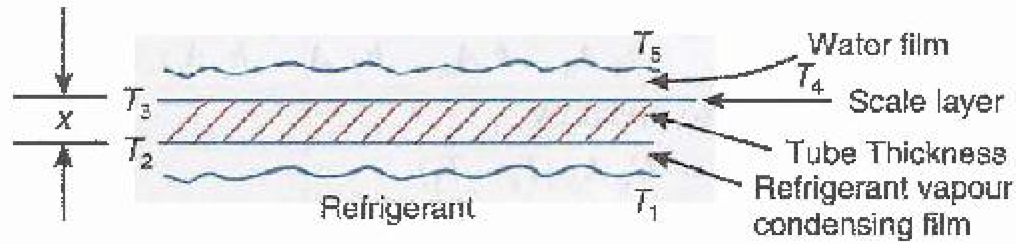
Heat Transfer in Condensers

The heat transfer (Q) in water-cooled condensers is given by

$$Q = UA\,\Delta T = \frac{\Delta T}{R}$$

U = Overall heat transfer coefficient,

A = Surface area of the condenser,



ΔT = Overall temperature difference, and

R = Overall thermal resistance of the condenser = $1/UA$

The heat transfer from the vapour refrigerant to the water in tubes takes place in the following:

1. The heat transfer takes place from the vapour refrigerant to the outside of the tube through the condensing film.

$$Q = h_0 A_0 (T_1 - T_2)$$

$$T_1 - T_2 = \frac{Q}{h_0 A_0}$$

2. The heat transfer takes place from the outside surface to the inside surface of the tube

$$Q = \frac{k A_m (T_2 - T_3)}{x}$$

$$T_2 - T_3 = \frac{Q x}{k A_m}$$

3. The heat transfer takes place through the layer of scale.

$$Q = h_f A_i (T_3 - T_4)$$

$$T_3 - T_4 = \frac{Q}{h_f A_i}$$

4. The heat transfer takes place from the boundary layer film to the water inside the tube

$$Q = h_i A_i (T_4 - T_5)$$

$$T_4 - T_5 = \frac{Q}{h_i A_i}$$

Now, by adding the above equations we get:

$$T_1 - T_5 = \frac{Q}{h_0 A_0} + \frac{Q\,x}{k\,A_m} + \frac{Q}{h_f A_i} + \frac{Q}{h_i A_i}$$

The overall heat transfer is also given by

$$Q = U_0 A_o (T_1 - T_5)$$

or $$T_1 - T_5 = \frac{Q}{U_0 A_0}$$

From equations (*v*) and (*vi*),

$$\frac{1}{U_0 A_0} = \frac{1}{h_0 A_0} + \frac{x}{k\,A_m} + \frac{1}{h_f A_i} + \frac{1}{h_i A_i}$$

Condensing Heat Transfer Coefficient

The average coefficient of heat transfer for vapour condensing outside of horizontal tubes of diameter D is given by

$$h_o = 0.725 \left[\frac{(k_f)^3 (\rho_f)^2 g \times h_{fg}}{N \times D \times \mu_f \times \Delta t} \right]^{\frac{1}{4}} \quad \ldots (ii)$$

k_f = Thermal conductivity of liquid condensate,

ρ_f = Density of liquid condensate,

ρ_g = Density of vapour and is very small as compared to ρ_f,

g = Acceleration due to gravity,

h_{fg} = Latent heat of vaporisation,

μ_f = Viscosity of condensed refrigerant film,

Δt = Difference of temperatures between condensing vapour and outside surface,

N = Number of tubes in a vertical row,

D = Diameter of tube.

N=number of tube/number of vertical row

Water-side Coefficient

The water-side coefficient for forced convection turbulent flow inside the tubes is given by

$$h_i = 0.023 \times \frac{k}{D}\left(\frac{D\,V\,\rho}{\mu}\right)^{0.8}\left(\frac{c\,\mu}{k}\right)^{0.4}$$

D = Inside diameter of tube,

k = Thermal conductivity of water,

V = Average velocity of water,

ρ = Density of water,

μ = Viscosity of water, and

c = Specific heat of water.

Evaporators

Introduction: The evaporator is an important device used in the low pressure side of a refrigeration system. The liquid refrigerant from the expansion device enters the evaporator where it boils and changes into vapor. The function of an evaporator is to absorb heat from the surrounding location or medium which is to be cooled.

Note: The evaporator is also known as a cooling coil, a chilling coil or a freezing coil. The evaporator cools by using the refrigerant's latent heat of vaporization to absorb heat from the medium.

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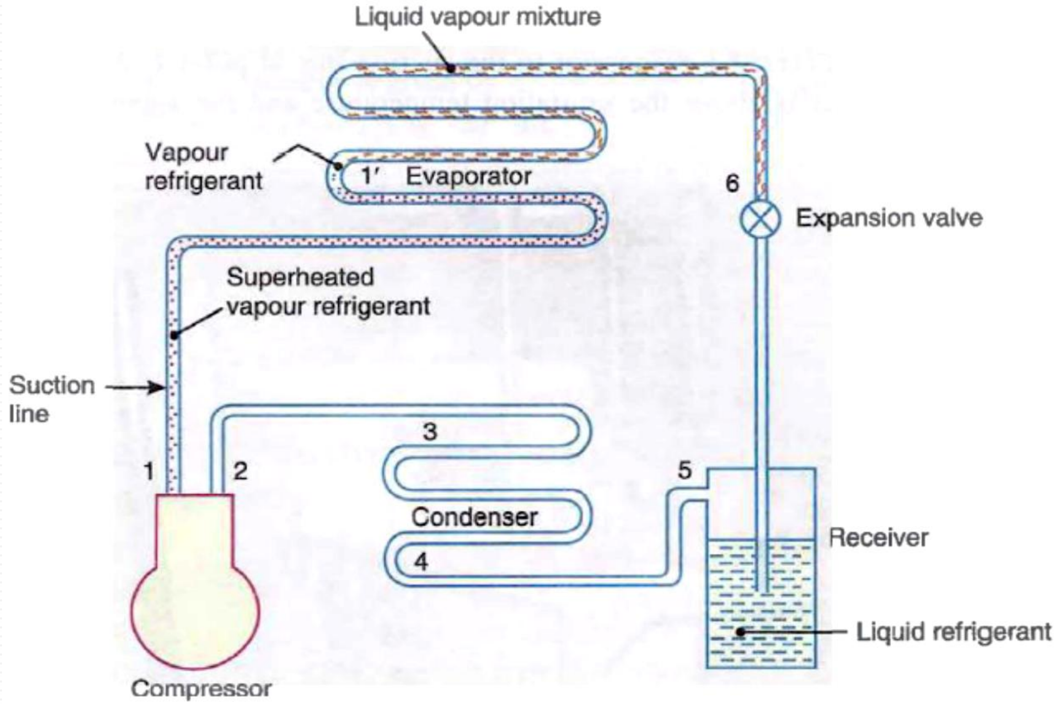
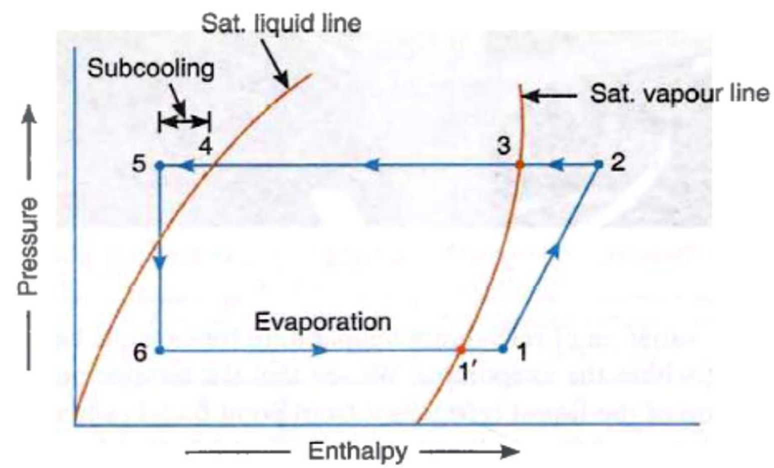
Working of an Evaporator:

The liquid refrigerant enters the evaporator at point 6, passes through the evaporator coil, it continually absorbs heat through the coil walls, from the surrounding. During this, the refrigerant continues to boil and evaporate by absorbed heat to latent heat. Finally at point 1', all the liquid refrigerant become only vapor refrigerant

At point 1' the vapor refrigerant continues to absorb heat from the surrounding. This heat absorption causes an increase in the sensible heat (or temperature) of the vapor refrigerant. The vapor temperature continues to rise until the vapor leaves the evaporator to the suction line at point 1. and the vapor refrigerant is superheated.



(a) Schematic diagram of a simple refrigerating system.



(b) p-h diagram of a simple refrigerating system.

Capacity of an Evaporator

The capacity of an evaporator is defined as the amount of heat absorbed by it over a given period of time. The heat absorbed or heat transfer capacity of an evaporator is given by:

$$Q = U\,A\,(T_2 - T_1)$$

U = Overall heat transfer coefficient in W /m^2 °C,
A = Area of evaporator surface in m^2,
T_2 = Temperature of medium to be cooled (or temperature outside the evaporator) in °C, and
T_1 = Saturation temperature of refrigerant at evaporator pressure (or temperature inside the evaporator) in °C.

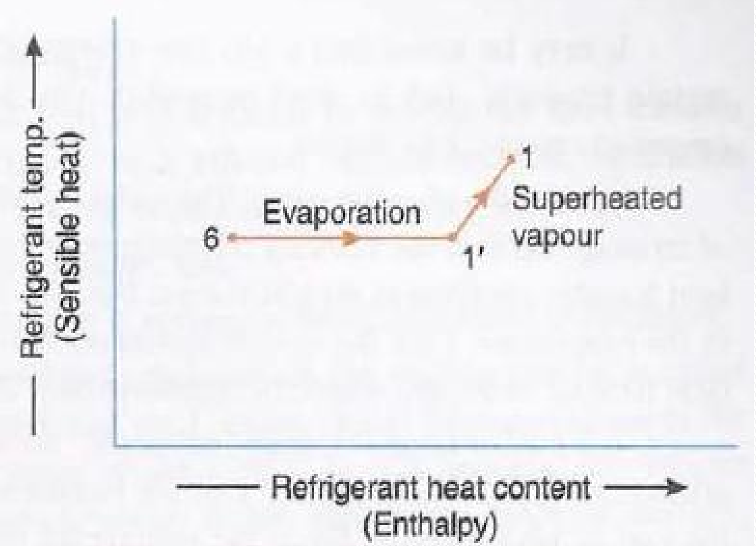


Fig. 11.2. Temperature-enthalpy diagram for an evaporator.

Factors effecting on the Heat Transfer Capacity of an Evaporator

There are many factors which effect on the heat transfer capacity of evaporator.

1- Material

2-Temperature difference. Temperature difference between the refrigerant within the evaporator and the surrounding.

3- Velocity of refrigerant. The velocity of refrigerant also effects on the heat transfer capacity of an evaporator. If the velocity of refrigerant flowing through the evaporator increases, the overall heat transfer coefficient also increases. But this increased velocity will cause greater pressure loss in the evaporator.

4- Thickness of the evaporator coil wall.

5- Contact surface area.

Heat Transfer during Boiling

The boiling occurs in the following two ways:

1- Pool boiling as it occurs in flooded evaporators; and

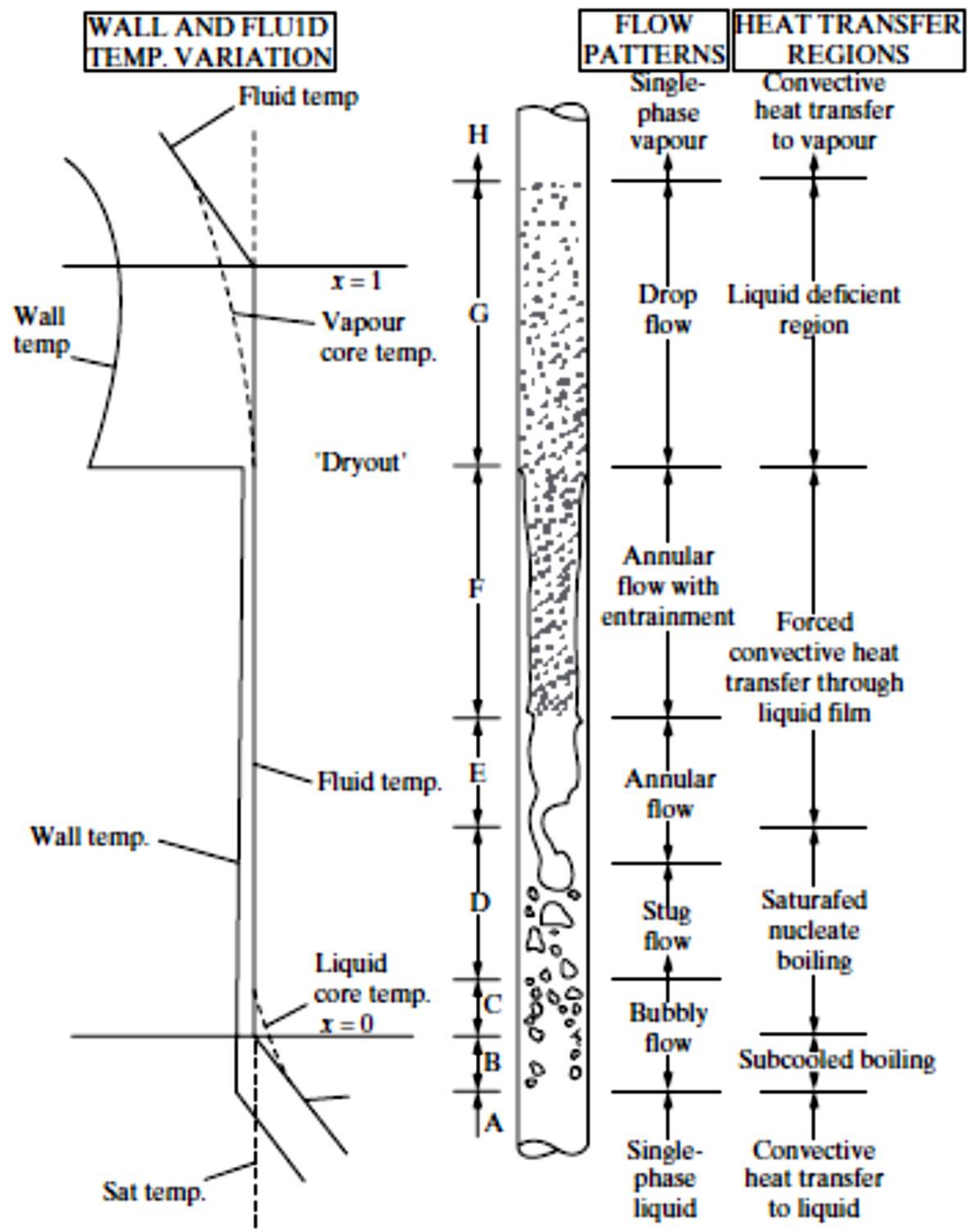
2.Flow boiling or forced convection boiling as it occurs in direct expansion evaporators.

Pool Boiling inside tubes

1- When refrigerant boils inside the tubes, the heat-transfer coefficient changes progressively as the refrigerant flows through the tube.

2- Refrigerant enters the evaporator tube with a low fraction of vapor and the fraction of vapor increases with refrigerant proceeds through the tube and increasing the heat-transfer coefficient

3- When the refrigerant is nearly all vaporized, the coefficient drops off to the magnitude of vapor transferring heat by forced convection



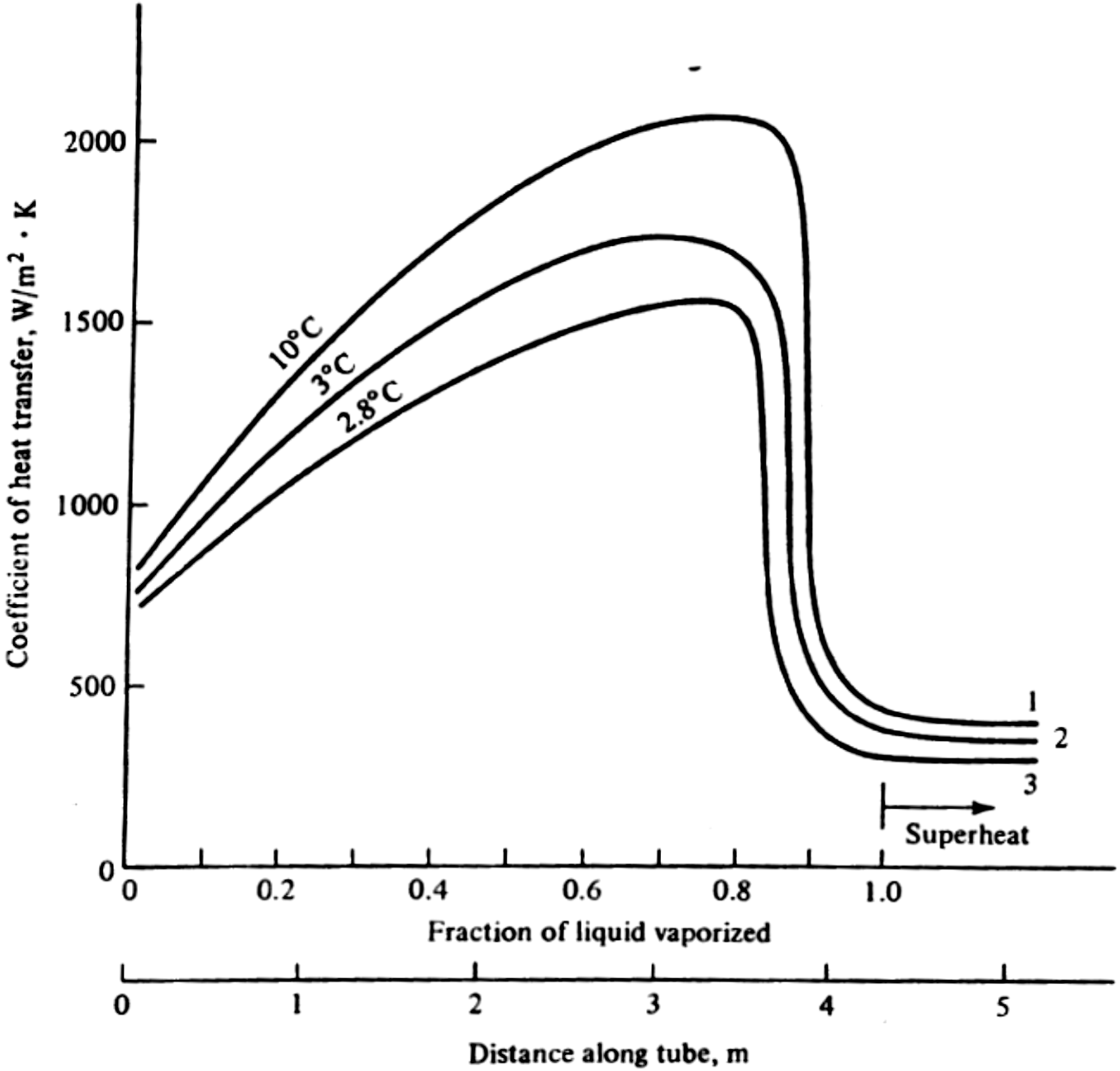


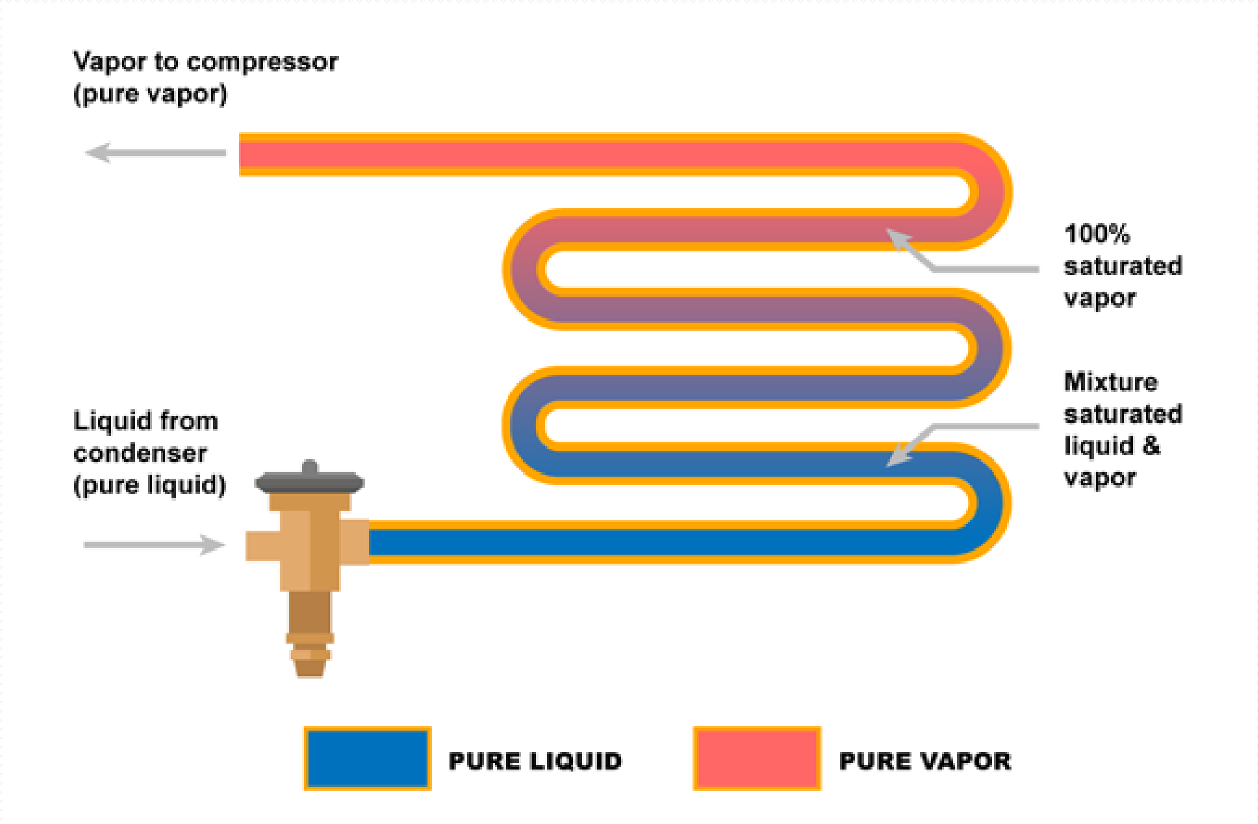
Figure 12-22 Heat-transfer coefficients of refrigerant 22 boiling inside tubes. Curve 1 at 10°C, curve 2 at 3°C, and curve 3 at 2.8°C temperatures of evaporation.[15]

Types of Evaporators

There are many types of evaporators, and can be classified into many group as:

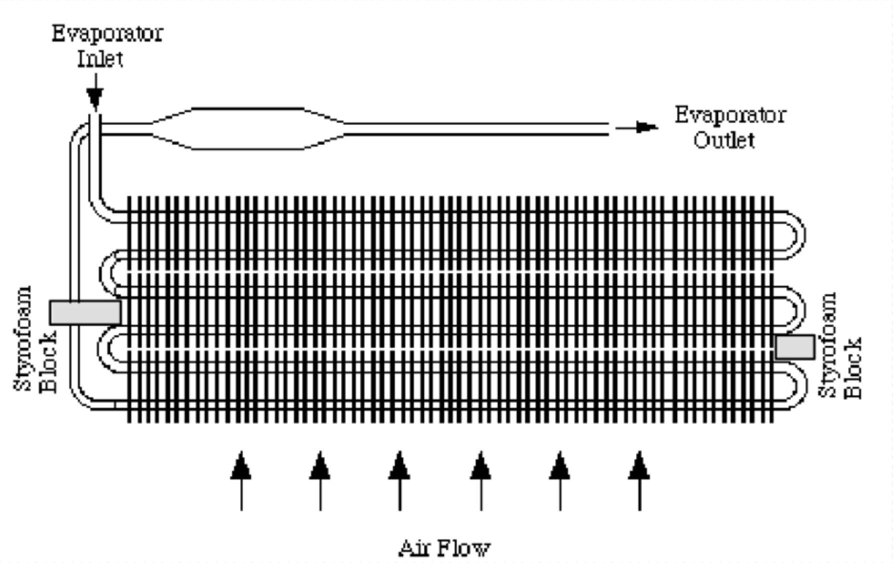
1- According to the type of construction

(a) Bare tube coil evaporator,

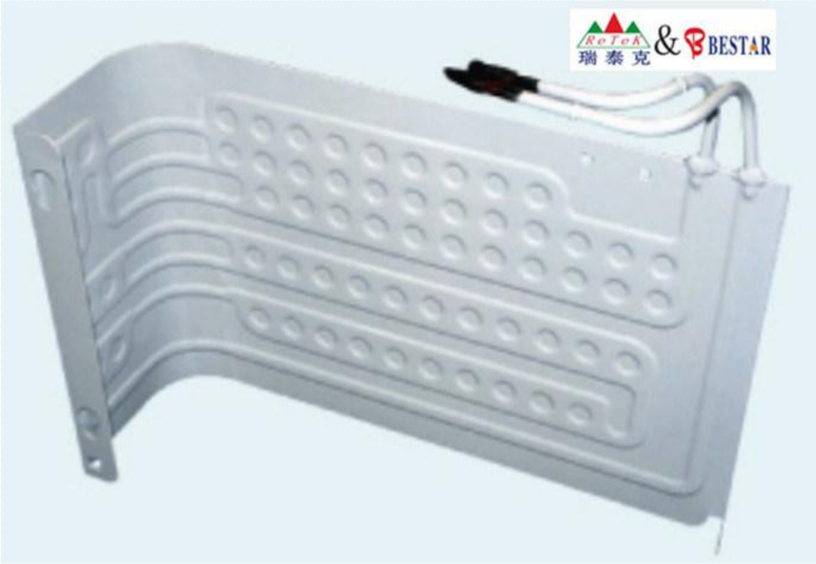


Types of Evaporators

(b) Finned tube evaporator,

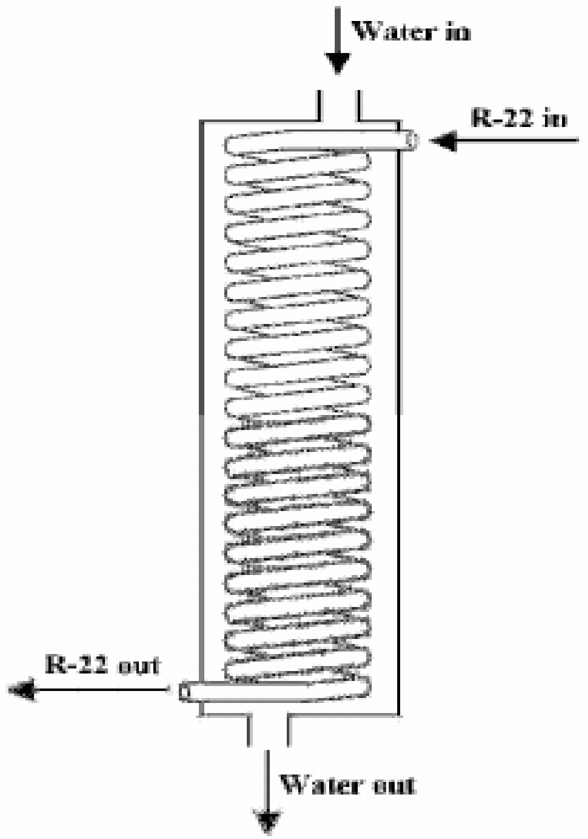


(c) Plate evaporator



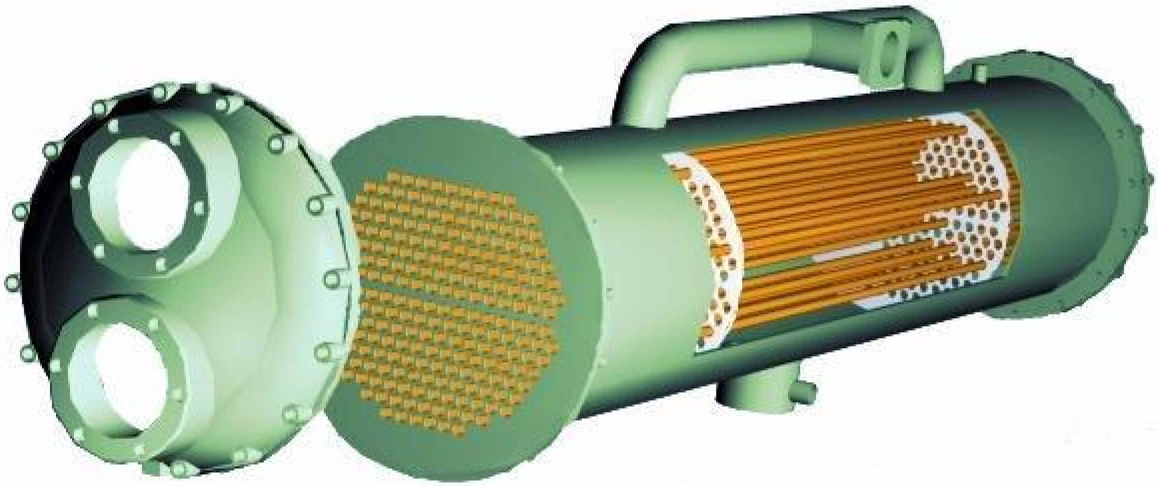
Types of Evaporators

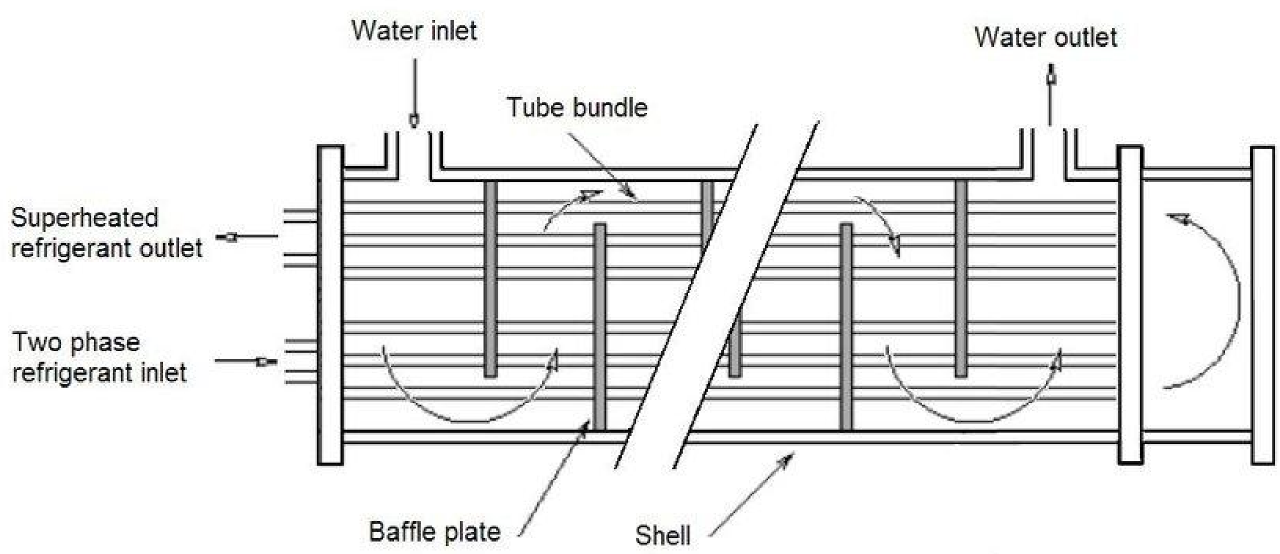
(e) Shell and coil evaporator



Types of Evaporators

(d) Shell and tube evaporator,





Types of Evaporators

(f) Tube-in-tube evaporator

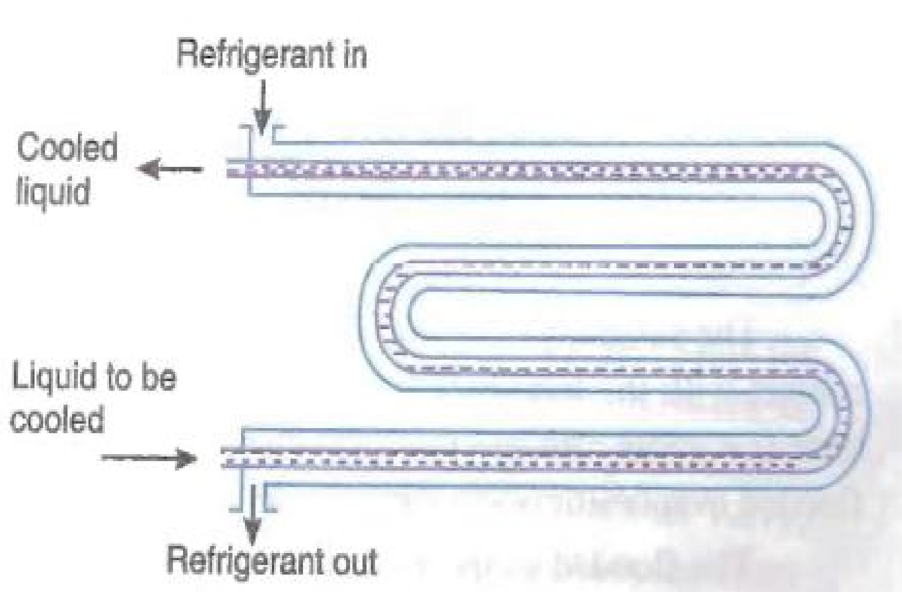


Fig. 11.10. Tube-in-tube or double-tube evaporator.

Types of Evaporators

2. According to the manner in which liquid refrigerant is fed

(a) Flooded evaporator, and

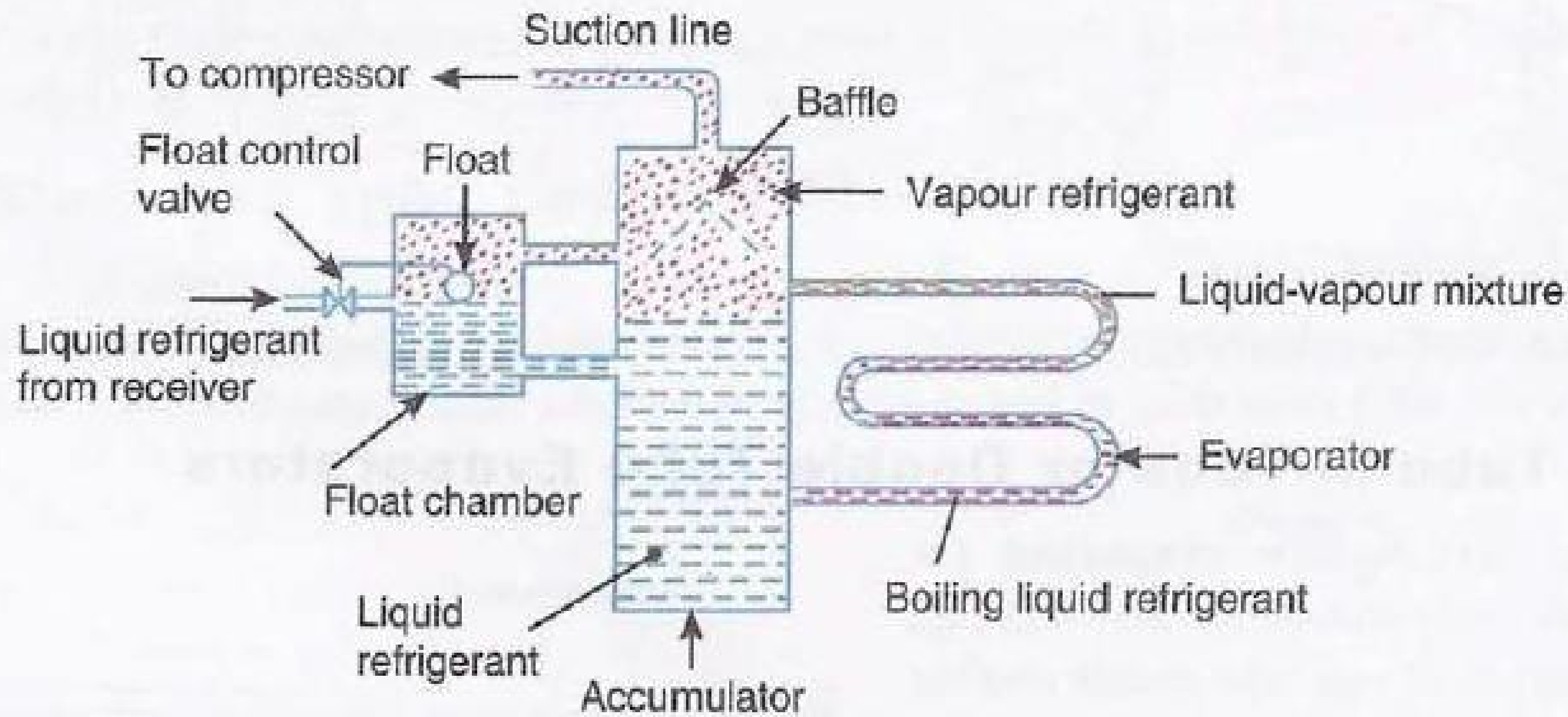


Fig. 11.11. Flooded evaporator.

Types of Evaporators

(b) Dry expansion evaporator

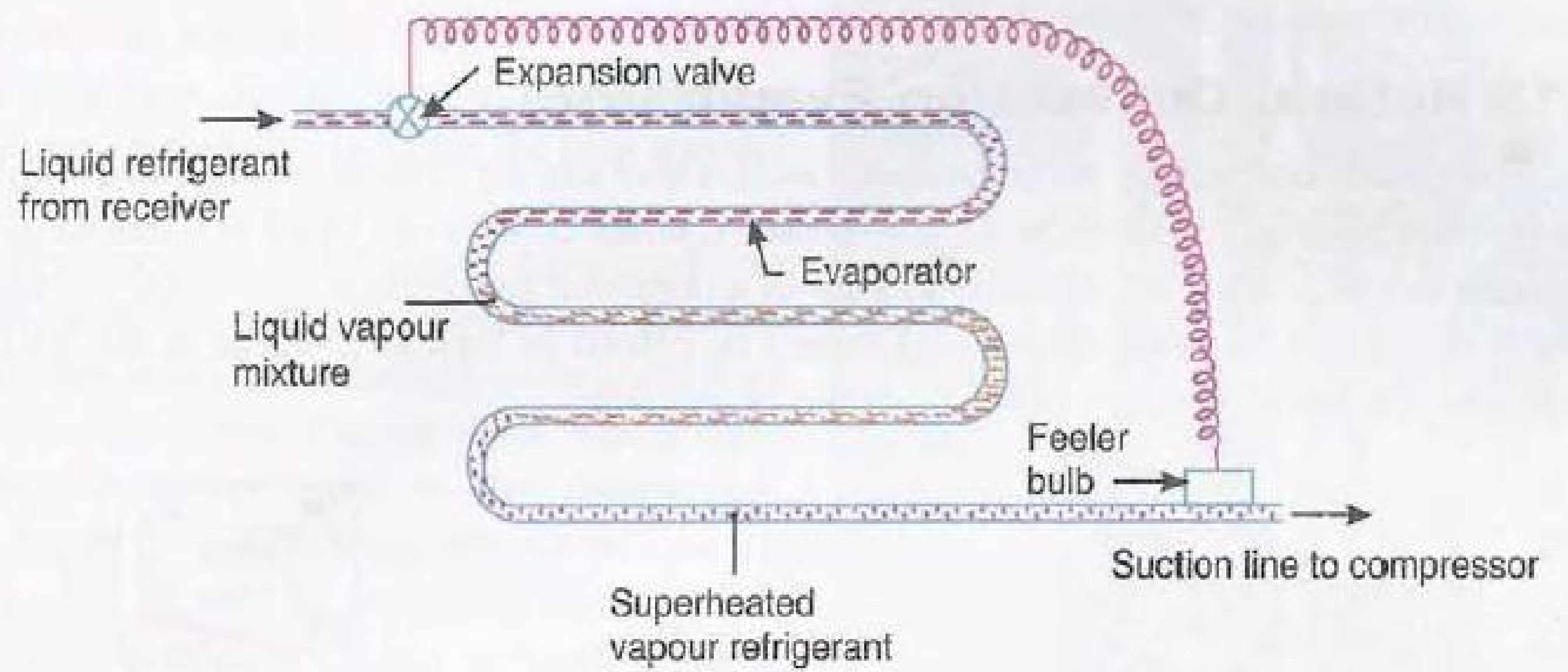


Fig. 11.12. Dry expansion evaporator.

Types of Evaporators

3. According to the mode of heat transfer

(a) Natural convection evaporator,

(b) Forced convection evaporator.

Types of Evaporators

4. According to operating conditions

(a) Frosting evaporator,

The frosting type evaporators always operate at temperatures below 0°C. Therefore, the frost is formed on its surface. The frost comes from the moisture in the air. It may be noted that as the frost grows in thickness, the coil or cooling efficiency decreases.

(c) Non-frosting evaporator,

The non-frosting evaporators operate at a temperature above 0°C at all times. Therefore, the frost does not form on the evaporator.

(d) Defrosting evaporator

A defrosting evaporator is one in which frost accumulates on the coils when the compressor is running and melts after the compressor shuts off with help of one method of defrosting.

Effect of Frosting: Frosting has a undesirable effect on the operation of the refrigeration system for two reasons:

1. Thick layers of frost act as insulation.
2. In forced convection coils the frost reduces the airflow rate. That reduced the rate of *U* value of the coil and the mean temperature difference between the air and refrigerant must increase in order to transfer the same rate of heat flow.

Methods of Defrosting an Evaporator

There are many methods of defrosting are available, and the most popular are:

- In hot-gas defrost, discharge gas from the compressor is sent directly to the evaporator and the evaporator performs temporarily as a condenser. The heat of condensation melts off the frost, which drains away.
- In water defrost, a stream of water is directed over the coil until all the frost is melted.
- Electric Defrosting, in this method the electric resistance heater are built into the evaporator.

Expansion Devices

An expansion device is basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to:

1. Reduce pressure from condenser pressure to evaporator pressure.
2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator

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1- Capillary tubes: Capillary tube serves almost in small refrigeration systems, and its application extends up to refrigerating capacities of 10 ton. A capillary tube is 1 to 6 m long with an inside diameter generally from 0.5 to 2 mm.

The pressure reduction in a capillary tube occurs due to the following two factors:

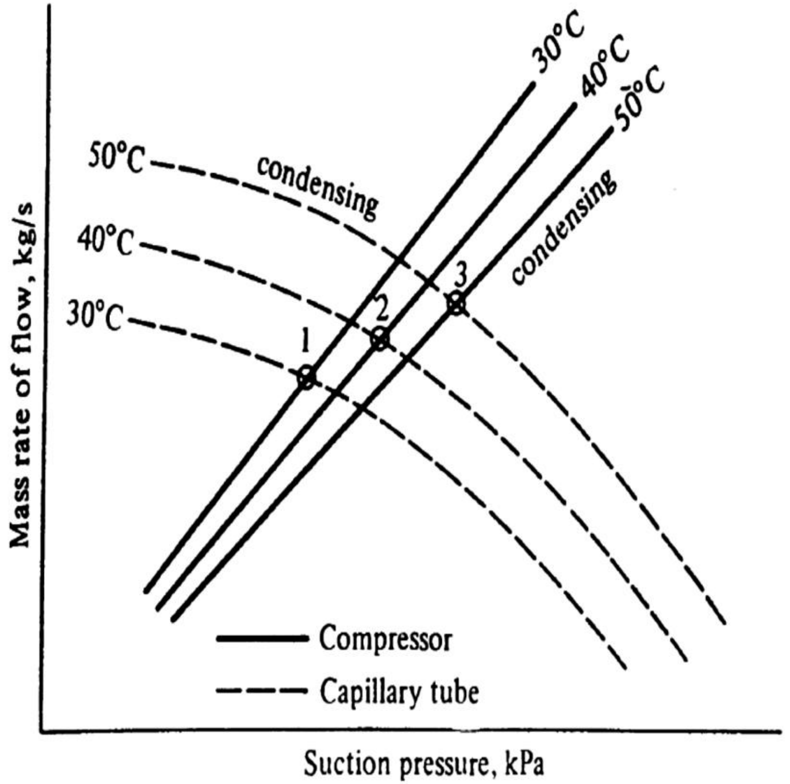
1- The refrigerant has to overcome the frictional resistance offered by tube walls, this leads to some pressure drop.

2- The average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter being constant, this leads to increase in the velocity of refrigerant ($\dot{m} = \rho VA$) the increase in velocity or acceleration of the refrigerant also leads to some pressure drop.

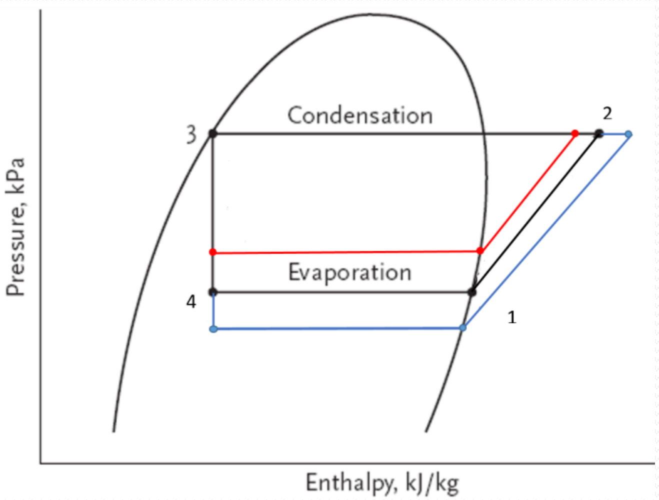
Figure (1) represent the mass rate of flow fed by the capillary tube and the mass rate of flow pumped by the compressor, that can explain as:

1- The compressor pump the refrigerant from the evaporator as the same flow rate of refrigerant that the expansion device feeds to the evaporator this point called balancing point.

2- At high condensing pressures the capillary tube feeds more refrigerant to the evaporator than at low condensing pressures because of the increase in pressure difference across the tube.

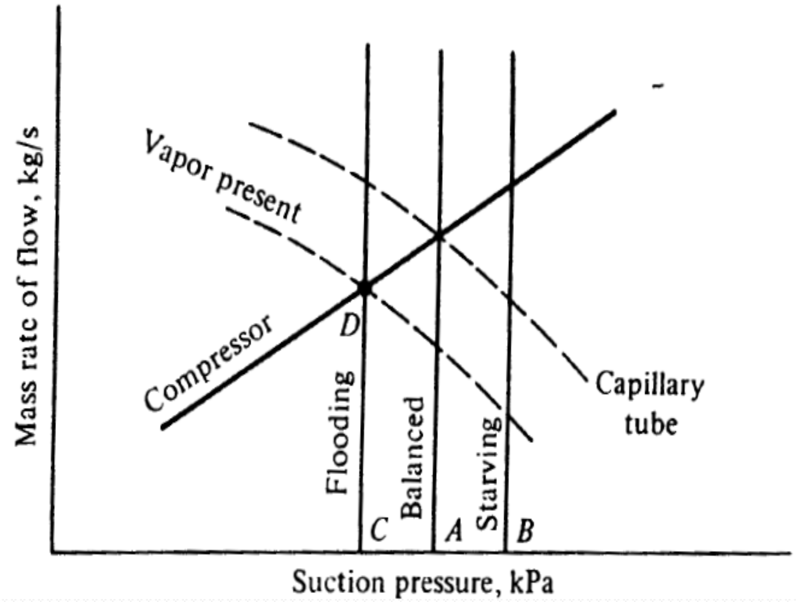


3- The compressor and capillary tube do not have complete control to fix the suction pressure because the heat-transfer relationships of the evaporator must also be satisfied. If the evaporator heat transfer is not satisfied at the compressor-capillary tube balance point, an unbalanced condition results which can starve the evaporator or overfeed the evaporator.

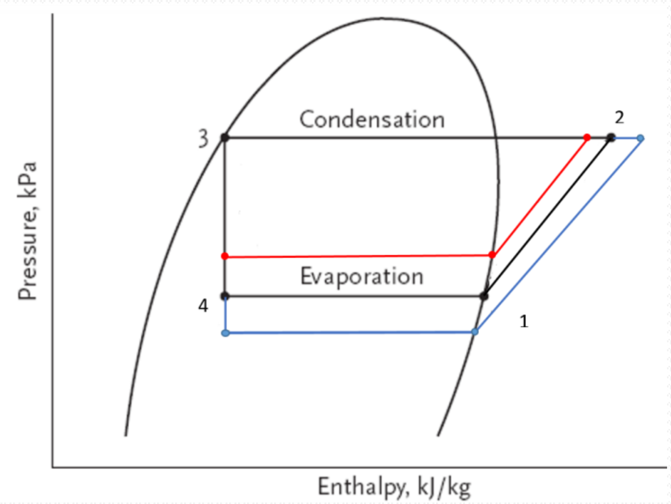


Effect of Load Variation

1- **Starving:** It results when the suction pressure rises and the capillary does not feed sufficient refrigerant to the evaporator while the compressor can draw more refrigerant out of the evaporator that occur at heavy heat load, so the evaporator soon becomes short of refrigerant and the refrigerant accumulates in the condenser. The accumulation of refrigerant in the condenser reduces the effective area of the condenser that is available for heat transfer. That an increase in condenser temperature leads to a decrease in compressor mass flow rate and an increase in capillary mass flow rate. Hence, the system will find a new balance point at higher condenser temperature.



2- **Flooding**: An opposite unbalanced condition results if the refrigeration load decrease less than the refrigeration capacity at the balance point then the suction temperature and pressure drop to point C where the capillary tube can feed more refrigerant to the evaporator than the compressor can draw out, so the evaporator fills with liquid causing flooding of the evaporator. This may lead to dangerous effect if the liquid refrigerant overflows to the compressor causing slugging of the compressor. (liquid can be prevented by limiting the charge of refrigerant in the system by turn off the compressor)



Advantages of Capillary tubes

1. They are simple, have no moving parts, and are inexpensive.
2. They also allow the pressures in the system to equalize during the off cycle that produce low starting torque on the motor driving the compressor.

Disadvantages of Capillary tubes

1. They are not adjustable to changing load conditions.
2. Sensitive to clogging by foreign matter.
3. the capillary tube be used only on hermetically sealed systems, where there is less likelihood of the refrigerant leakage

2- <u>Constant-pressure expansion valve</u>: The valve is used to maintain a constant pressure and then a constant temperature in the evaporator. The schematic diagram of the valve is shown in Figure (3) the valve consists of an adjustable spring that can be adjusted to maintain the required temperature in the evaporator.

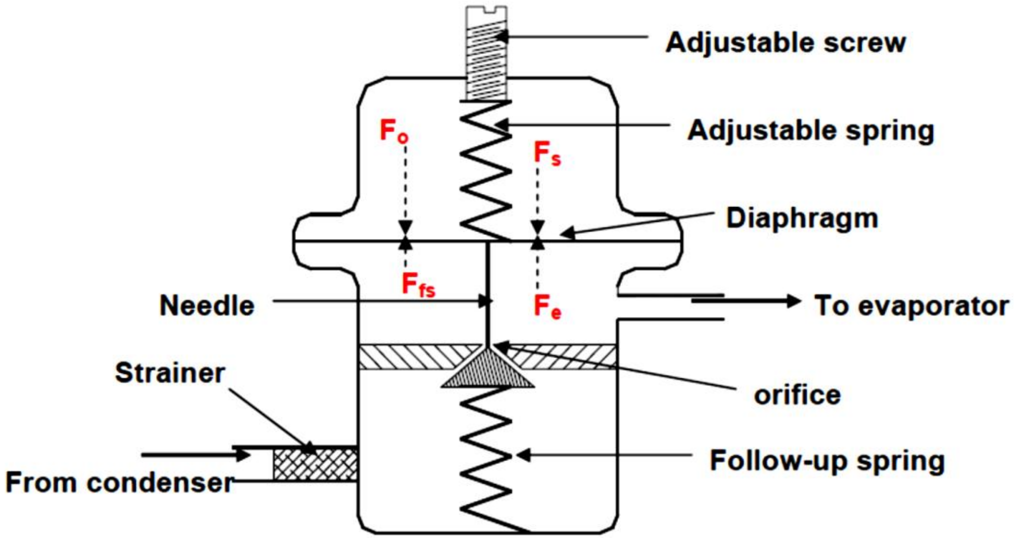
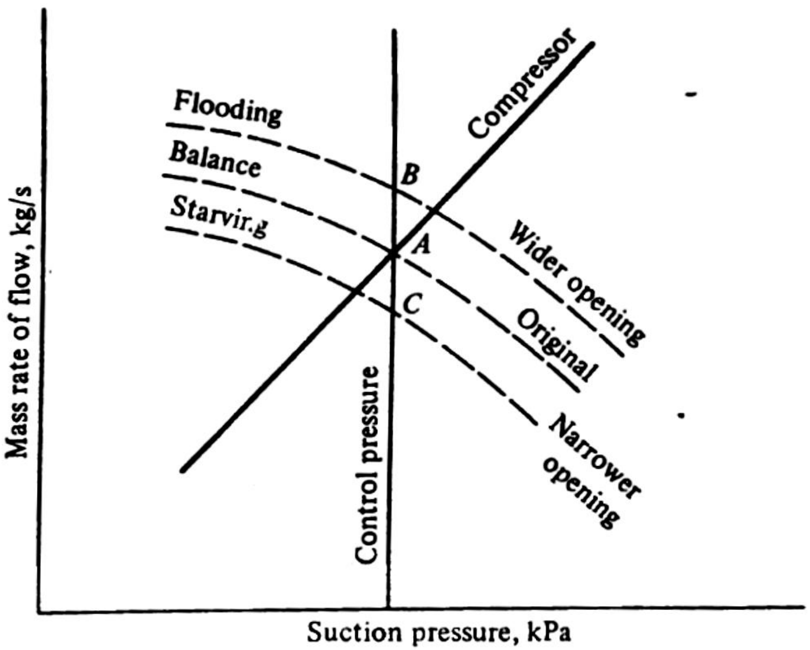


Fig. 3 *: Schematic of an Automatic Expansion Valve*

Effect of Load Variation

1. **Flooding:** If the refrigeration load decreases, the evaporator temperature decrease then the evaporator pressure also decreases. This decreases the force (F_e) that will become less than the adjustable spring force and the needle will be pushed downwards opening the valve wider. This will increase the mass flow rate through the evaporator to keep the evaporator pressure at a constant value. In Figure below, point A is the normal position of the value and B is the position at reduced load and wider opening. This causes accumulation of liquid refrigerant in the evaporator. This is called "flooding" of the evaporator. The liquid refrigerant may fill the evaporator and it may overflow to the compressor causing damage to it.



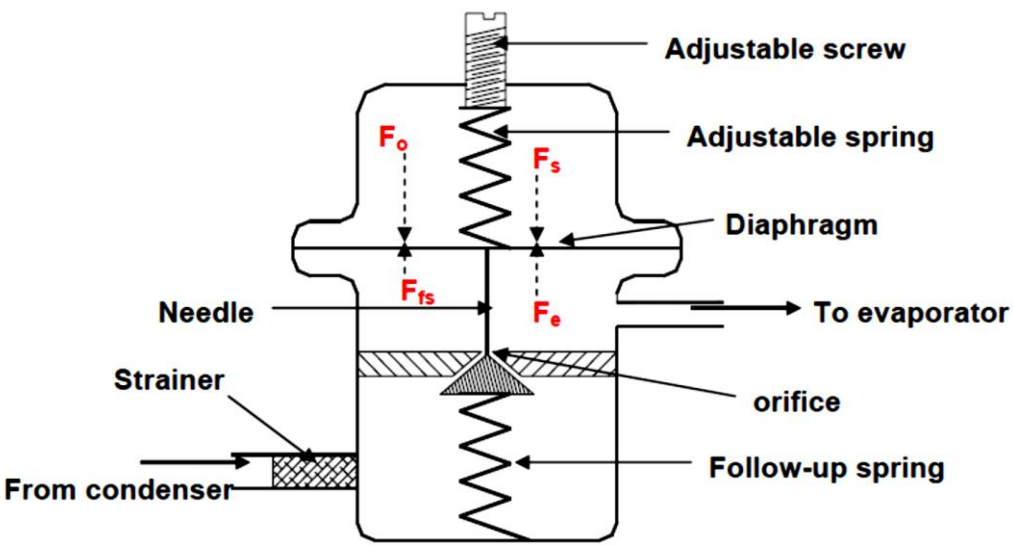
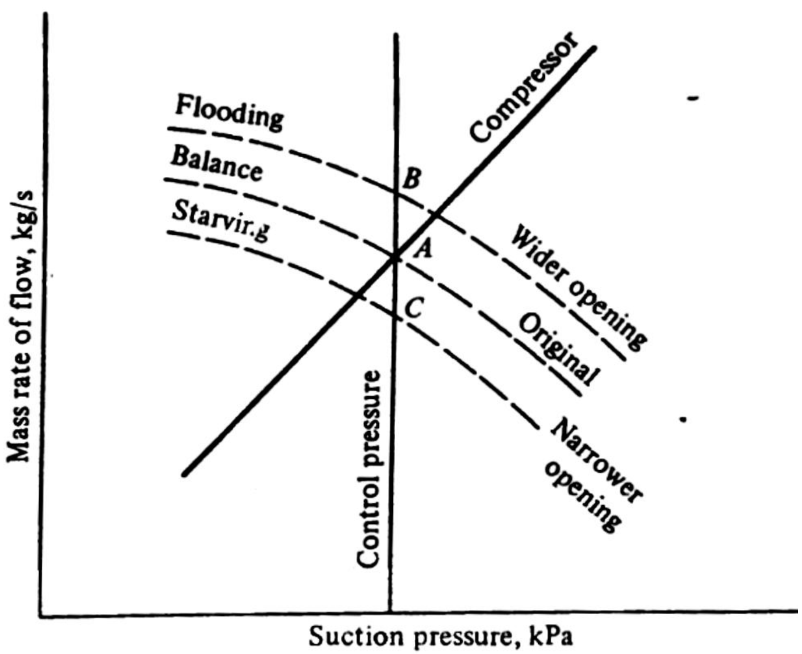


Fig. 3 *: Schematic of an Automatic Expansion Valve*

2. Starving: On the other hand if the refrigeration load increases, the evaporator temperature and pressure will increase. This will increase (F_e) that will tend to move the needle upwards, making the valve opening narrower and decreasing the mass flow rate. This shifts the operating point from A to point C where the compressor draws out more refrigerant than that fed by the expansion valve leading to starving of the evaporator. (When the refrigerant is not fed to evaporator while the compressor removes the refrigerant from the evaporator. The evaporator pressure decreases and the needle stand moves downwards and the valve opens.)



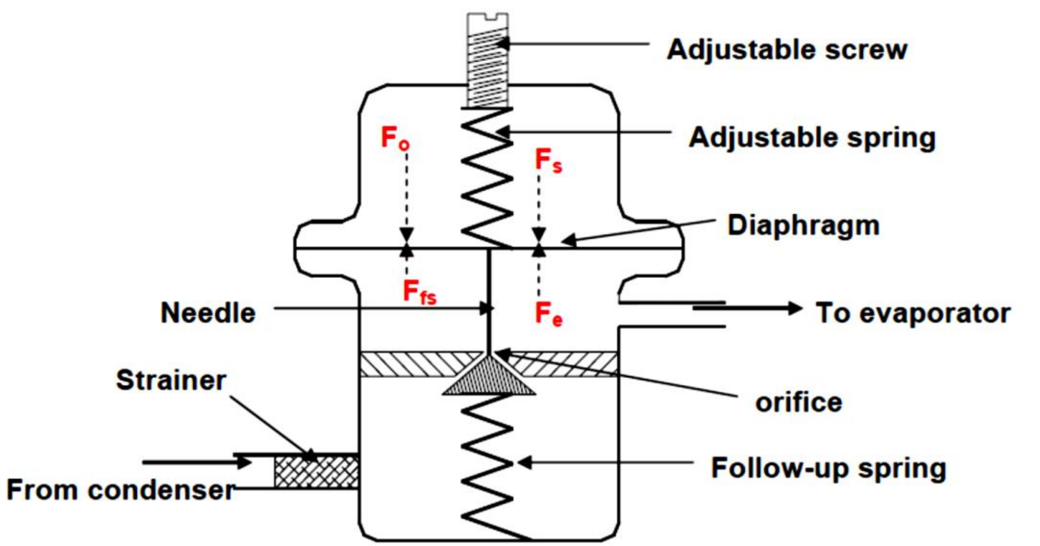


Fig. 3 *: Schematic of an Automatic Expansion Valve*

Applications of automatic expansion valve

The automatic expansion valves are used wherever constant temperature is required, for example, milk chilling units, water coolers where freezing isn't desire. These are also used in small commercial refrigeration systems where hermetic compressors are used. Normally the usage is limited to systems of less than 10 TR capacities.

3- <u>Thermostatic Expansion Valve</u>: Thermostatic expansion valve is the most desire expansion valve and is most commonly used in refrigeration systems. A thermostatic expansion valve maintains a constant degree of superheat at the exit of evaporator; also it is most effective for preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. The schematic diagram of the valve is given in Figure (5). This consists of a feeler bulb that is attached to the evaporator exit tube so that it senses the temperature at the exit of evaporator. The feeler bulb is connected to capillary tube those contain some fluid that is called power fluid. The power fluid may be the same as the refrigerant in the refrigeration system,

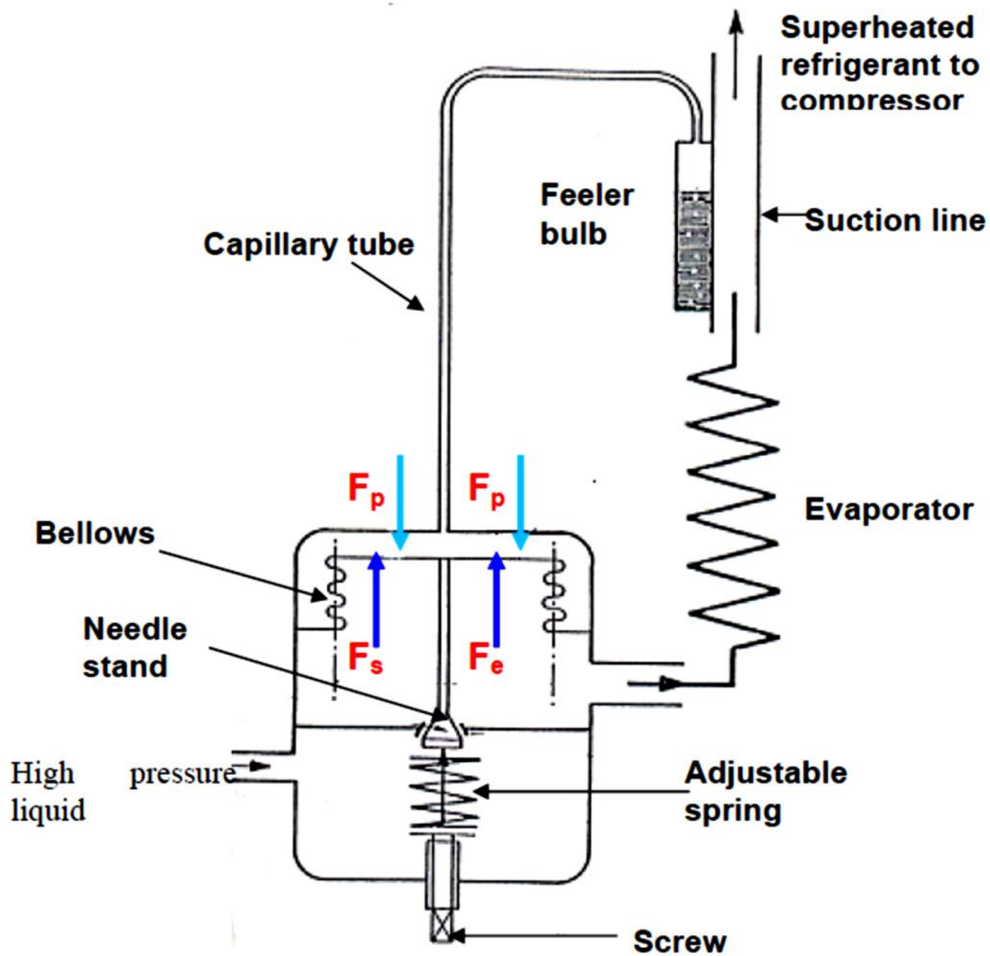


Fig. (5) *Schematic of a Thermostatic Expansion Valve (TEV)*

Effect of Load Variation

If the load on the system increases, the evaporation rate of liquid refrigerant increases. As the degree of superheat increases, pressure of power fluid F_p increases, the needle is pushed down and the mass flow rate of refrigerant increases were the expansion valve is proportional to the load.

On the other hand, if the load on the plant decreases, the evaporation rate of refrigerant decreases, as a result the degree of superheat decreases. The thermostatic expansion valve reduce the mass flow rate through it. Hence, this valve always establishes balanced flow condition of flow between compressor and itself as show in figure (6).

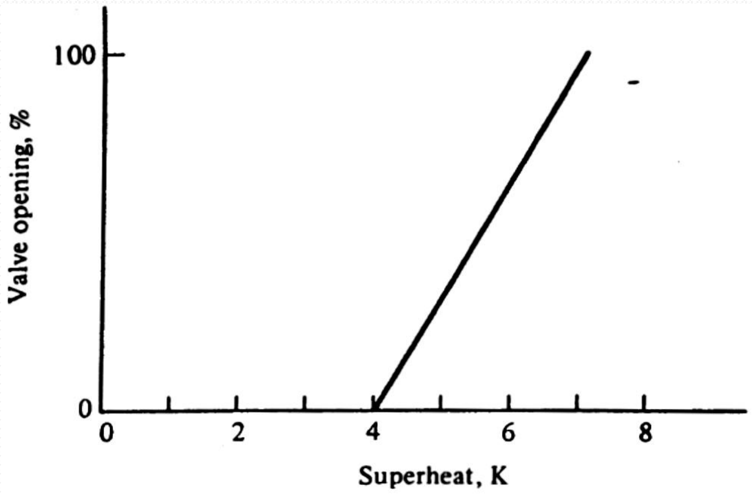


Figure (6) **Throttling range of superheat-controlled valve.**

Thermostatic Expansion Valve (TEV) with cross charge:

Figure (7) shows the saturated vapour line with pressure. The difference between P_p and P_e is proportional to the spring force. The degree of superheat given by a set of P_p and P_e. The figure shows three sets of P_p and P_e for the same spring force at three evaporator temperatures (–40°C, -20°C and 5°C). It is observed that at location A, the degree of superheat is very large whereas at location C the degree of superheat is very small for the same spring force setting proportional to (P_p–P_e).

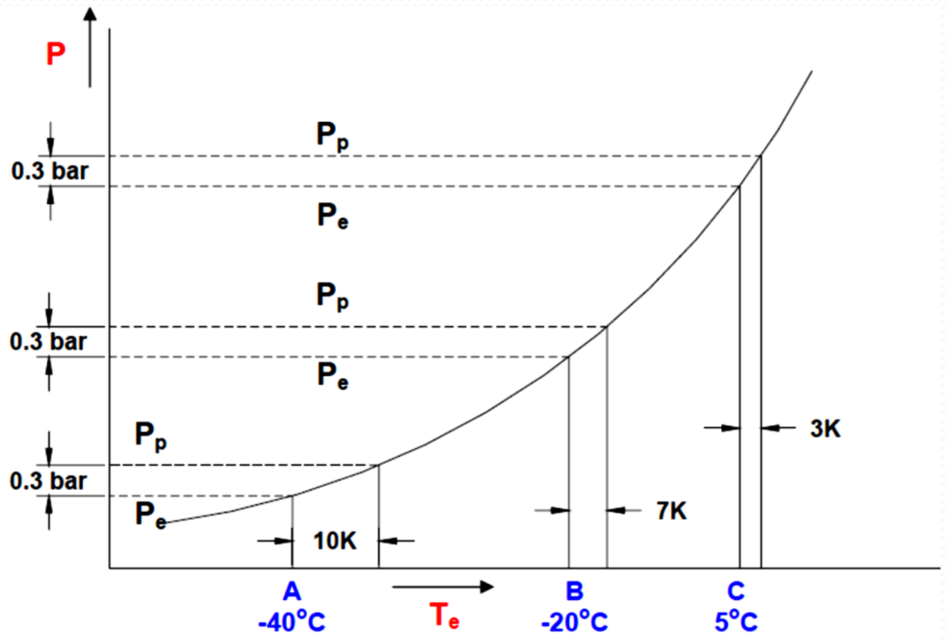


Fig. (7) *Vapour pressure curve of refrigerant and power fluid*

It is observed that if the spring is set for a superheat of (10°C) at (–40°C) evaporator temperature, the degree of superheat will become near to zero at higher temperature. As a result; when the plant is started at warm temperature, there is a possibility of flooding of evaporator. If degree of superheat is set to avoid flooding at say 5°C, then at the design point of say – 40°C, the superheat will be very large and it will starve the evaporator.

This can be corrected if a fluid different from refrigerant is used in the feeler bulb as power fluid. Such a TEV is called ***TEV with cross charge.*** Figure (8) shows the saturated vapour line for the power fluid as well as the refrigerant in the system. The projection for P_p is taken from the saturation line for power fluid and it shows the temperature at the exit of the evaporator. The power fluid is such that at any temperature it has lower saturation pressure than that of the refrigerant in the system, so that as the evaporator temperature increases the degree of superheat increases. Hence cross charge helps in maintaining the same degree of superheat at all evaporator temperatures.

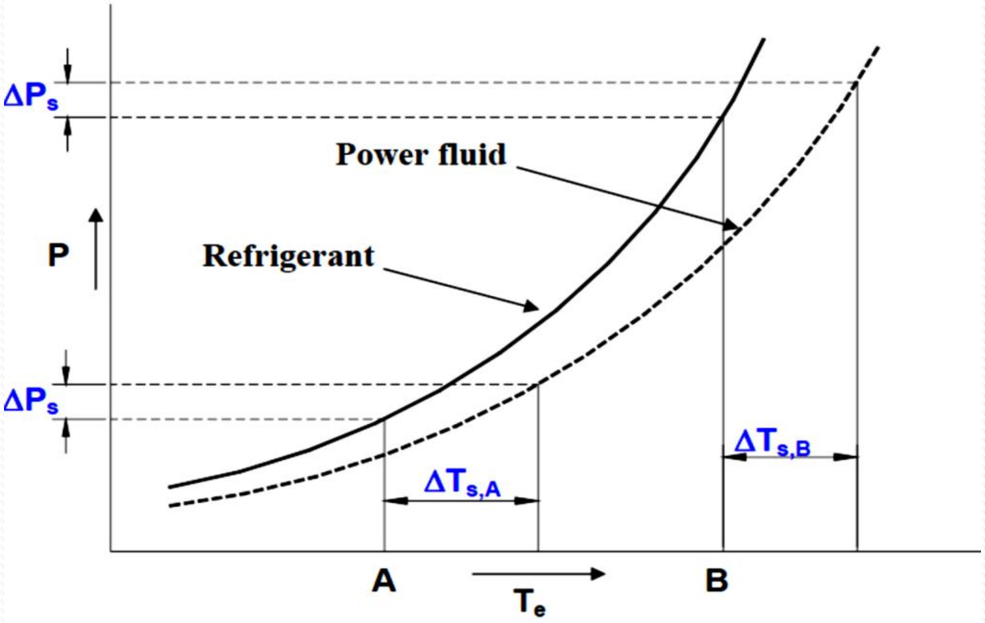


Fig. (8) *Vapour pressure curves of refrigerant and power fluid (cross-charged TEV)*

Advantages and applications of TEV

The advantages of TEV compared to other types of expansion devices are:

1. It provides excellent control of refrigeration capacity as the supply of refrigerant to the evaporator.
2. It ensures that the evaporator operates efficiently by preventing starving under high load conditions.
3. It protects the compressor from slugging by ensuring a minimum degree of superheat under all conditions of load.

4- <u>Float type expansion valves</u>: Float type expansion valves are normally used with flooded evaporators in large capacity refrigeration systems. A float type valve opens or closes depending upon the liquid level. The float valve always maintains a constant liquid level in a chamber called as float chamber.

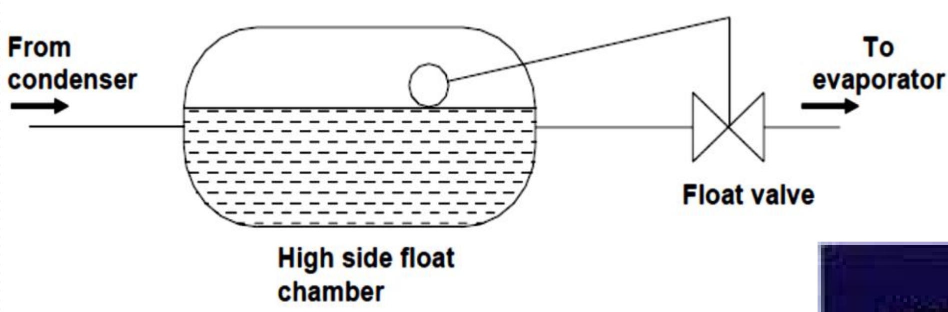


Fig. (9) *Schematic of a high-side float valve*

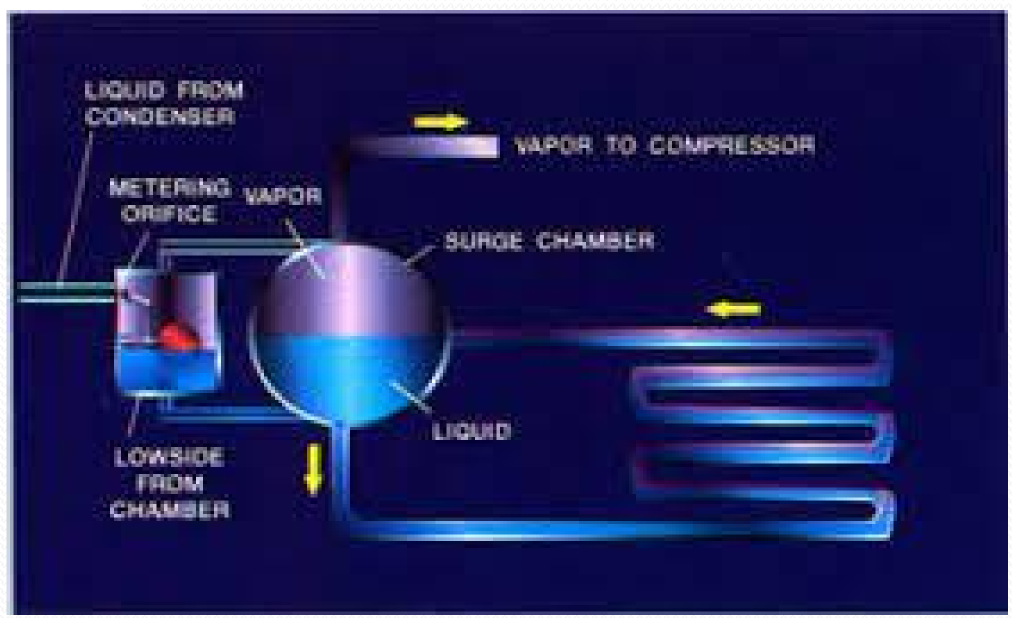


Figure (10) shows an original balance point at *A*. If the refrigeration load should increase, the evaporating temperature and pressure rise, which allows the compressor to pump a greater rate of flow than the valve is feeding. The valve reacts to keep the level constant by widening its average opening. A new balance point occurs at point *B*. If the refrigeration load decreases, the suction pressure drops and the level rises, the valve is closing somewhat and give a balance point at *C*.

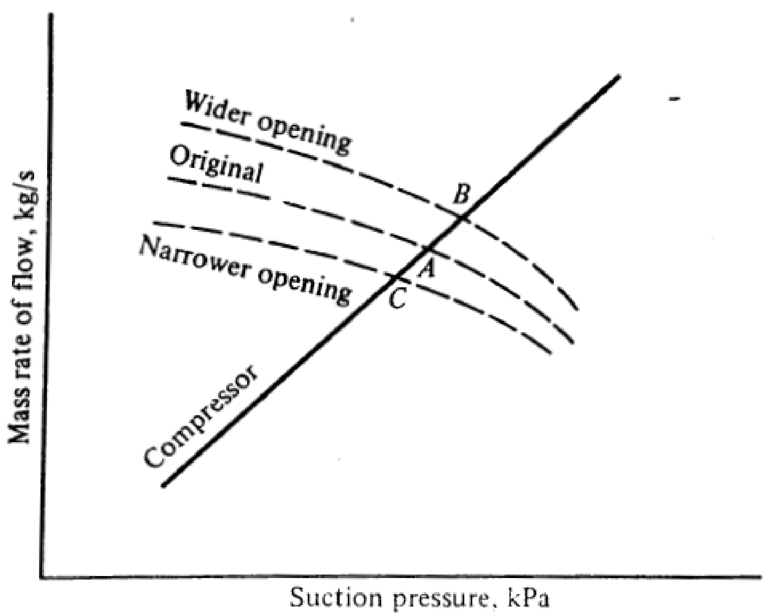
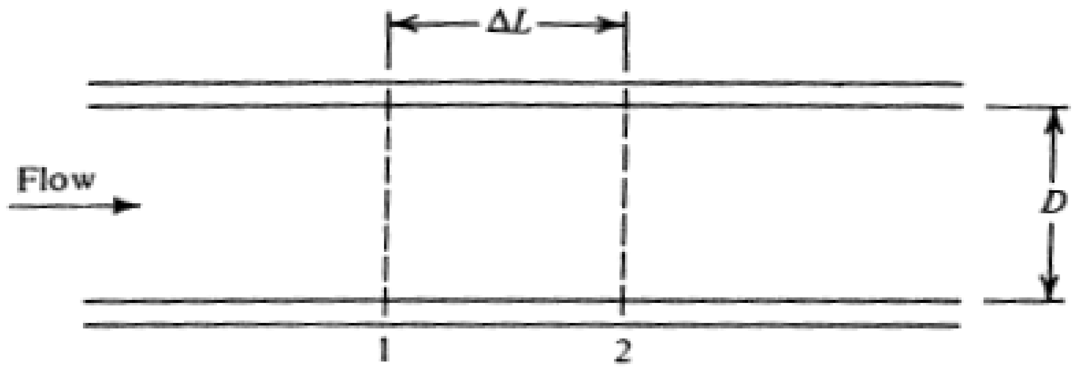


Figure (10) **Balance points with various load conditions using a float valve. The condensing pressure is constant.**

Analytical computation of pressure drop in a capillary tube

The equations relating states and conditions at points I and 2 in a very short length of capillary tube in Fig.



The fundamental equations applied to the control volume between points 1 and 2 in Fig. are: (1) Conservation of mass, (2) Conservation of energy, and (3) Conservation of momentum.

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Technical Engineering college/ Mosul

(1) Conservation of mass: The equation for conservation of mass states is

$$\dot{m}_1 = \dot{m}_2 = \rho_1 \times V_1 \times A_1 = \rho_2 \times V_2 \times A_2$$

Or

$$\frac{V_1 \times A_1}{\vartheta_1} = \frac{V_2 \times A_2}{\vartheta_2} \qquad (A_1 = A_2)$$

$$\frac{V_1}{\vartheta_1} = \frac{V_2}{\vartheta_2} \qquad (1)$$

(2) Conservation of energy: The equation of conservation of energy is

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

(3) Conservation of momentum: The momentum equation is the difference in forces that needed to accelerate the fluid because of drag and pressure difference on opposite ends of the element

$$\left[(p_1 - p_2) - f\frac{\Delta L}{d}\frac{V^2}{2\vartheta}\right] \times A = m^{\cdot}\,(V_2 - V_1) \qquad (3)$$

$$(p_1 - p_2) = f_m\frac{\Delta L}{d}\frac{V_m}{2}\frac{m^{\cdot}}{A} + \frac{m^{\cdot}}{A}\,(V_2 - V_1)$$

$$V_m = \frac{V_2 + V_1}{2}\ , \qquad f_m = \frac{f_2 + f_1}{2}\ , \qquad f = \frac{0.33}{Re^{0.25}} = \frac{0.33}{\left(\frac{V \times d}{\vartheta \times \mu}\right)^{0.25}}$$

$$h = h_f + x(h_g - h_f)$$
$$\vartheta = \vartheta_f + x(\vartheta_g - \vartheta_f)$$
$$\mu = \mu_f + x(\mu_g - \mu_f)$$

Example: What length of capillary tube (ID = 1.63 mm) will drop the pressure from saturated liquid refrigerant 22 at 40°C to the saturation temperature of the evaporator of 5°C? The flow rate is 0.01 kg/s.

Point (1) at (40 °C):

$$x_1 = 0, \qquad h_1 = h_f = 249.9\frac{kJ}{kg}, \qquad \vartheta_1 = \vartheta_f = 0.000886\frac{m^3}{kg},$$

$$p_1 = 1536\ kPa, \qquad \mu_1 = \mu_f = 0.0001394\ Pa.s$$

$$V_1 = \frac{\dot{m} \times \vartheta_1}{A} = \frac{0.01 \times 0.000886}{\frac{\pi}{4}(0.00163)^2} = 4.24\ m/s$$

$$Re_1 = \frac{V_1 \times d}{\vartheta_1 \times \mu_1} = \frac{4.24 \times 0.00163}{0.000886 \times 0.0001394} = 56020$$

$$f_1 = \frac{0.33}{Re_1^{0.25}} = \frac{0.33}{56020^{0.25}} = 0.02145$$

Point (2) at (39 °C):

$p_2 = 1498.8\ kPa$, $h_{f2} = 248.5 \frac{kJ}{kg}$, $h_{g2} = 416.2 \frac{kJ}{kg}$

$\vartheta_{f2} = 0.000882 \frac{m^3}{kg}$, $\vartheta_{g2} = 0.01533 \frac{m^3}{kg}$

$\mu_{f2} = 0.000141\ Pa.s$, $\mu_{g2} = 0.00001346\ Pa.s$

$h_2 = h_1 = h_f + x(h_g - h_f)$

$249.9 = 248.5 + x(416.2 - 248.5) \rightarrow x = 0.0083$

$\vartheta_2 = \vartheta_f + x(\vartheta_g - \vartheta_f)$

$\vartheta_2 = 0.000882 + 0.0083\ (0.01533 - 0.000882) \rightarrow \vartheta_2 = 0.001\ m^3/kg$

$\mu_2 = \mu_f + x(\mu_g - \mu_f)$

$\mu_2 = 0.000141 + 0.0083\ (0.00001346 - 0.000141) \rightarrow \mu_2 = 0.00014\ Pa.s$

$$V_2 = \frac{m^{\cdot} \times \vartheta_2}{A} = \frac{0.01 \times 0.001}{\frac{\pi}{4}(0.00163)^2} = 4.792\, m/s$$

$$Re_2 = \frac{V_2 \times d}{\vartheta_2 \times \mu_2} = \frac{4.792 \times 0.00163}{0.001 \times 0.00014} = 55792$$

$$f_2 = \frac{0.33}{Re_2^{0.25}} = \frac{0.33}{55792^{0.25}} = 0.02147$$

$$V_m = \frac{V_2 + V_1}{2} = \frac{4.24 + 4.792}{2} = 4.516\ m/s$$

$$f_m = \frac{f_2 + f_1}{2} = \frac{0.02145 + 0.02147}{2} = 0.02146$$

$$(p_1 - p_2) = f_m \frac{\Delta L}{d} \frac{V_m}{2} \frac{m^{\cdot}}{A} + \frac{m^{\cdot}}{A} (V_2 - V_1)$$

$$(1536 - 1498.8) \times 10^3 = \frac{0.01}{\frac{\pi}{4}(0.00163)^2} \times \left[0.02146 \times \frac{\Delta L}{0.00163} \frac{4.516}{2} + (4.792 - 4.24)\right]$$

$$\Delta L = 0.242\, m$$

Analysis of complete vapor compression refrigerant system

In this lecture, we shall discuss the analysis of the complete vapor compression refrigerant system and discuss the graphical and mathematical methods with a simple example.

-A complete basic vapor compression refrigeration system consist of as you know:

1- Compressor.

2- Condenser.

3- Expansion device

4- Evaporator.

The performance of each component is influenced by other components. That means they are interdependent (when a change in condition of one component affects performance of other components).

Analysis of the complete system taking into account:

1- The characteristics of individual components.

2- Predict complete system performance.

3- Analyze the effects of external influence on system performance

A conventional method of complete system analysis is by using the concept of balance points that can be obtained ***graphically*** or ***mathematically***.

<u>1- In graphical method</u>: The performance of two interdependent components is plotted for two same variables of common interest.

- **<u>Example</u>**:
- Evaporator temperature and mass flow rate is plotted for both compressor and expansion device at constant condenser temperature.
- The point of intersection of two curves will indicate the conditions at which the mass flow rate and the evaporator temperature will be same for the two components.
- This is the ***balance point*** for compressor and expansion device.
- So we have plotted the new mass flow rate of compressor and capillary tube if the evaporated temperature varied by graphical method in case of compressor and capillary.
- The same method can be extended to other components also for example condenser and compressor condenser and capillary and the evaporator.

2- <u>In mathematical analysis:</u> The performance characteristics of the system analysis can consider more components simultaneously are represented by mathematical equations. So we have equations for individual components and the simultaneous solution (successive substitution) of these equations gives the balance point.

- **<u>For example:</u>**
- The mass flow rate through expansion device and compressor are expressed as algebraic functions of evaporator and condenser temperatures.
- The balance point of the two components is obtained by simultaneous solution of the two algebraic equations.

- **Balance points and system simulation**

- Let the analysis system in steady sate condition .

- The performance data of components is presented in the form of plots or equations.

- The data for this purpose can be obtained from the catalogues of manufacturers of various system components.

Performance of reciprocating compressor

The purpose of balancing what is the refrigeration capacity is required as the function of evaporator and condenser temperature

- The upper set of curves shows the refrigerating capacity were an increase in evaporating temperature or a decrease in condensing temperature results in increased refrigerating capacity.

- The lower set of curves displays the power required by the compressor.

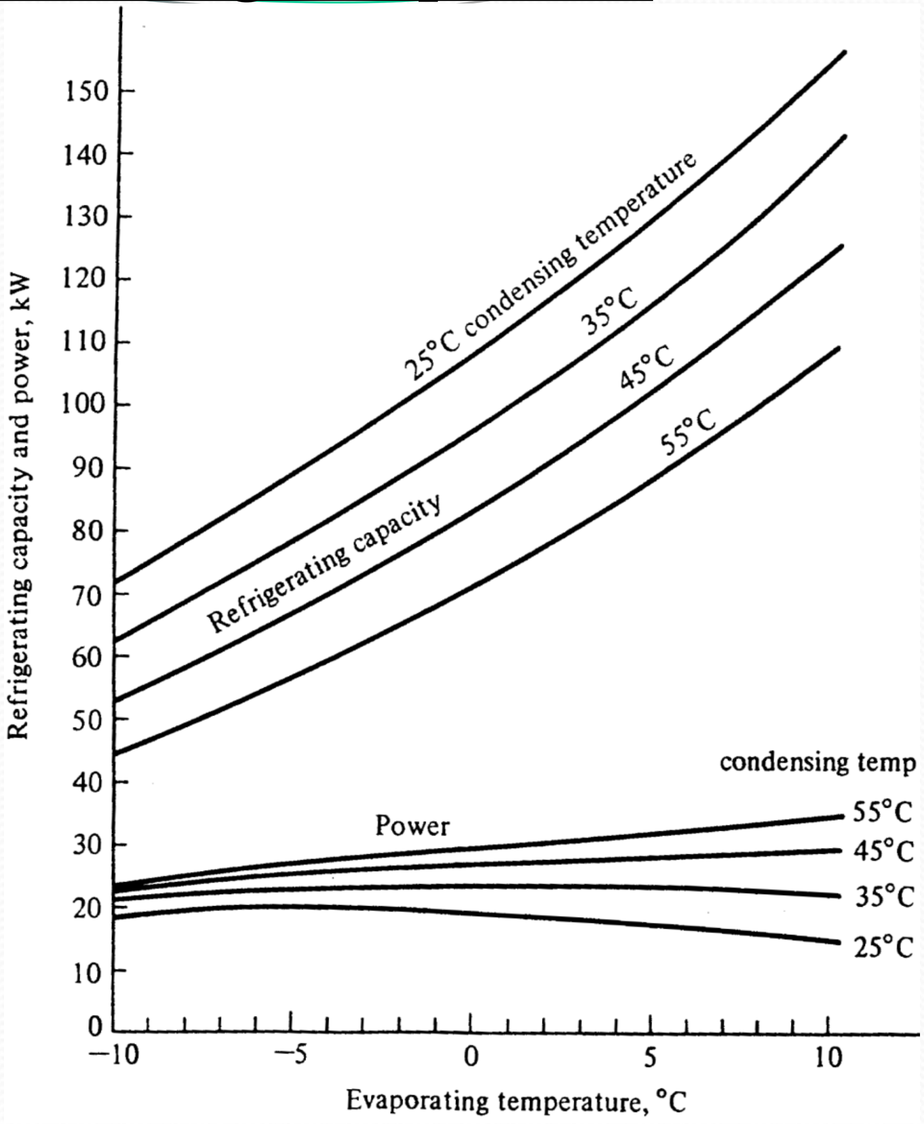


Figure (l) Refrigerating capacity and power requirement of a York (Division of Borg-Warner) hermetic reciprocating H62SP-22E, refrigerant 22, 1750 r/min compressor.

The form of the mathematical equations that represent the performance data of Fig. (1) is:

$$q_e = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_c + c_5 t_c^2 + c_6 t_e t_c + c_7 t_e^2 t_c + c_8 t_e t_c^2 + c_9 t_e^2 t_c^2$$

$$P = d_1 + d_2 t_e + d_3 t_e^2 + d_4 t_c + d_5 t_c^2 + d_6 t_e t_c + d_7 t_e^2 t_c + d_8 t_e t_c^2 + d_9 t_e^2 t_c^2$$

$\boldsymbol{q_c = q_e + p}$ ……(3)

Where:

q_e =Refrigerating capacity, kW

P = power required by compressor, kW

t_e = evaporating temperature, °C

t_c = condensing temperature, °C

q_c= the rate of heat rejection at the condenser.

Table 14-1 Constants in Eqs. (14-1) and (14-2)

c_1 = 137.402	d_1 = 1.00618
c_2 = 4.60437	d_2 = −0.893222
c_3 = 0.061652	d_3 = −0.01426
c_4 = −1.118157	d_4 = 0.870024
c_5 = −0.001525	d_5 = −0.0063397
c_6 = −0.0109119	d_6 = 0.033889
c_7 = −0.00040148	d_7 = −0.00023875
c_8 = −0.00026682	d_8 = −0.00014746
c_9 = 0.000003873	d_9 = 0.0000067962

Condenser performance:

The correct representation of the heat-transfer performance of a condenser can be quite complex, because the refrigerant vapor enters the condenser superheated and then the fraction of liquid and vapor changes constantly through the condenser.

The heat transfer through an air-cooled condenser by assumption of a constant heat-exchanger effectiveness for the condenser,

$$q_c = F(t_c - t_{amb}) \quad \ldots\ldots(4)$$

Where F = capacity per unit temperature difference, kW /K

t_{amb} = ambient temperature, °C

Figure (3) shows the catalog performance for a certain air-cooled condenser for which (F) in Eq. (4) is 9.39 kW/K.

Condensing-unit subsystem; graphic analysis:

The behavior of the condensing unit is influenced by the evaporating temperature (suction pressure) of the vapor received from the evaporator. When t_e changes, the pumping capability and therefore the refrigerating capacity and t_c change.

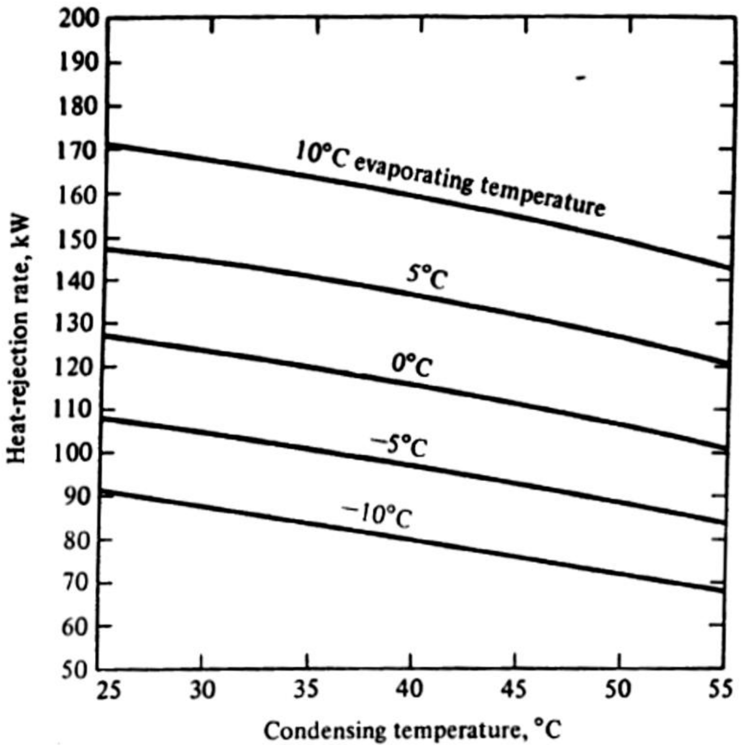


Figure (2) Heat-rejection rate of a York (Division of Borg-Warner) hermetic reciprocating compressor, H62SP-22E, refrigerant 22, 1750 r/min.

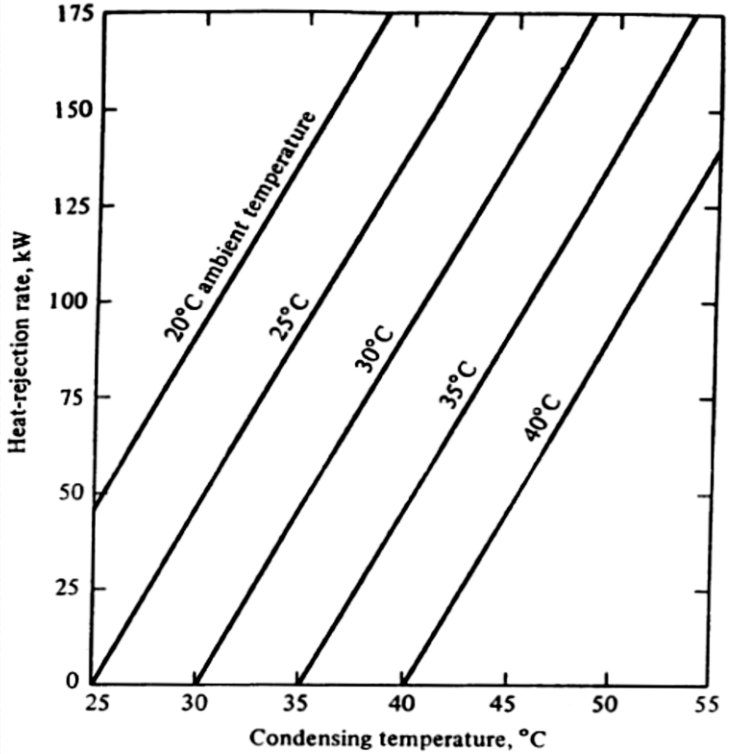


Figure (3) Performance of Bohn Heat Transfer Division air-cooled condenser, refrigerant 22

- Fig. (4), show the behavior of the condensing unit formed by the combination of compressor and condenser of Figs. (2) and (3). For a given ambient temperature, 35°C,
- The refrigerating capacity provided by the condensing unit at various evaporating temperatures can be extracted from a combination of Figs. (4) and (1).

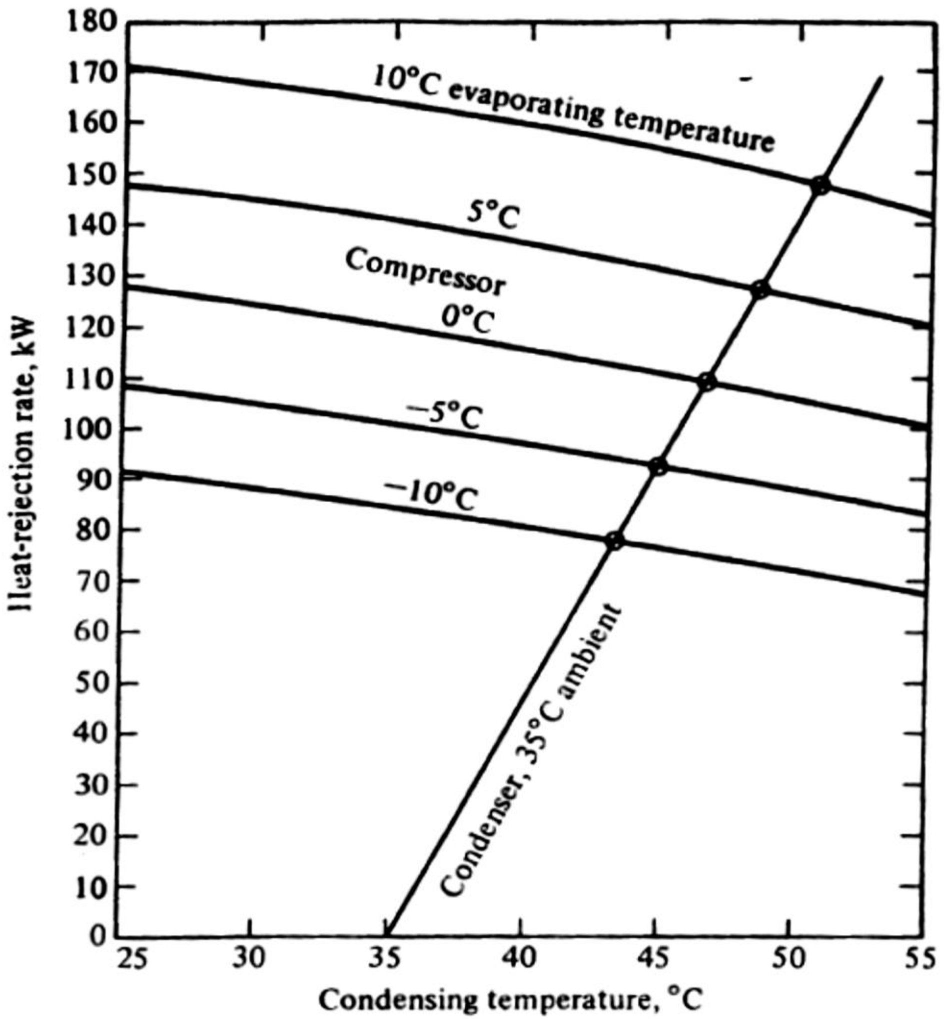


Figure (4) Balancing points of compressor and condenser that indicate performance of condensing unit

Condensing-unit subsystem; mathematical analysis:

The simultaneous solution of equations (1,2,3 and 4) is method of successive substitution, in which the calculation sequence is set up and trial values are introduced for certain variables in order to get started and then the values of the variables are updated each time of calculation through the loop.

A calculation loop or information-flow diagram for simulating the condensing unit is shown in Fig. (5)

1- an ambient temperature of 35°C and an evaporating temperature t_e = 10°C.
2- A trial value for t_c of 50°C is arbitrarily selected and entered into the calculation in order to get started.
3- The values of q_e, P, and q_c are calculated and then from Eq. (4)
5- A new value of t_c is computed which replaces the trial value of 50° C for the next loop.

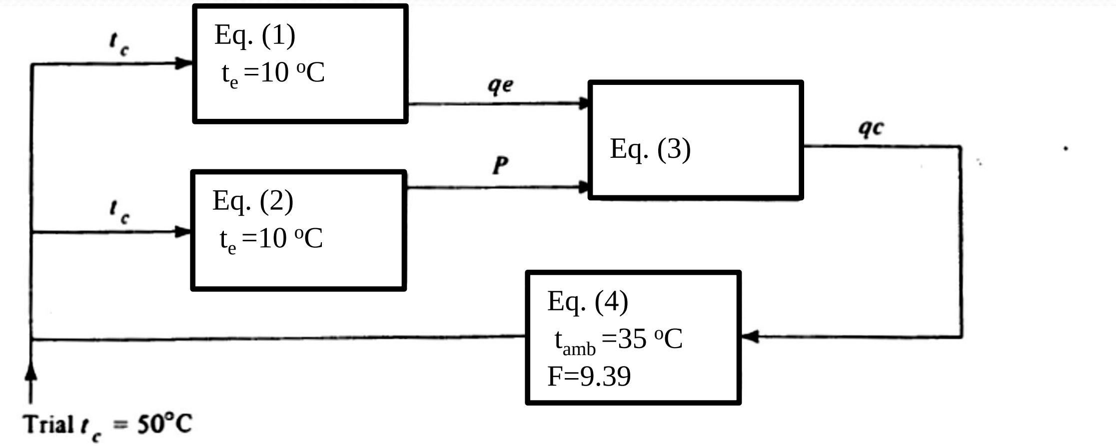


Figure (5) Information-flow diagram for condensing unit with tamb = 35°C and *te* = I0°C.

Table (2) Calculations through information-flow diagram of Fig.(5)

Cycle	q_e	P	q_c	t_c
1	116.71	32.06	148.77	50.84
2	115.32	32.46	147.77	50.74
3	115.49	32.41	147.90	50.75
4	115.47	32.41	147.88	50.75

Evaporator performance

For system simulation we are concerned with the overall performance of the evaporator as shown in the figure (6):

- The capacity increases with a reduction in evaporating temperature and/or an increase in the temperature of entering water.
- The capacity is reduced when the rate of water flow is decreased at a given inlet temperature.

$q_e = G(t_{wi} - t_e)$

Where

t_{wi} = temperature of entering water, °C

G = proportionality factor, kW/K

For the evaporator in Fig. (6) with a water flow rate of 2 kg/s

$G = 6.0 [1 + 0.046(t_{wi} - t_e)]$

Thus

$q_e = 6.0[1 + 0.046(t_{wi} - t_e)] (t_{wi} - t_e)$ …(5)

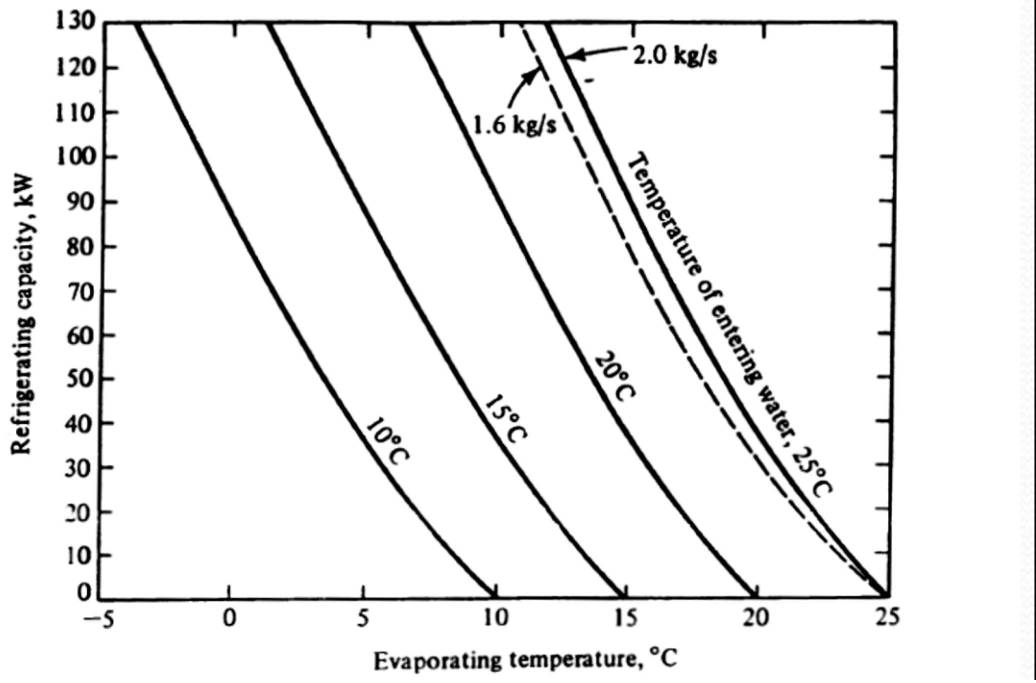


Figure (6) Refrigerating capacity of a Dunham-Bush, refrigerant 22, direct-expansion, and inner-fin liquid chiller CH660B. The solid lines show performance with 2 kg/s water flow and the dashed line with 1.6 kg/ s.

Performance of complete system; graphic analysis:

The complete system consists of the compressor, condenser, and evaporator. Figure (7) shows this combination and the balance points of the system that occur at various temperatures of the return chilled water. In summary, a graphic simulation of the vapor-compression system can be performed by:

1- Establishing the balance points for the condensing unit.

2- Combining the evaporator performance with the condensing-unit performance to find the balance points of the complete system

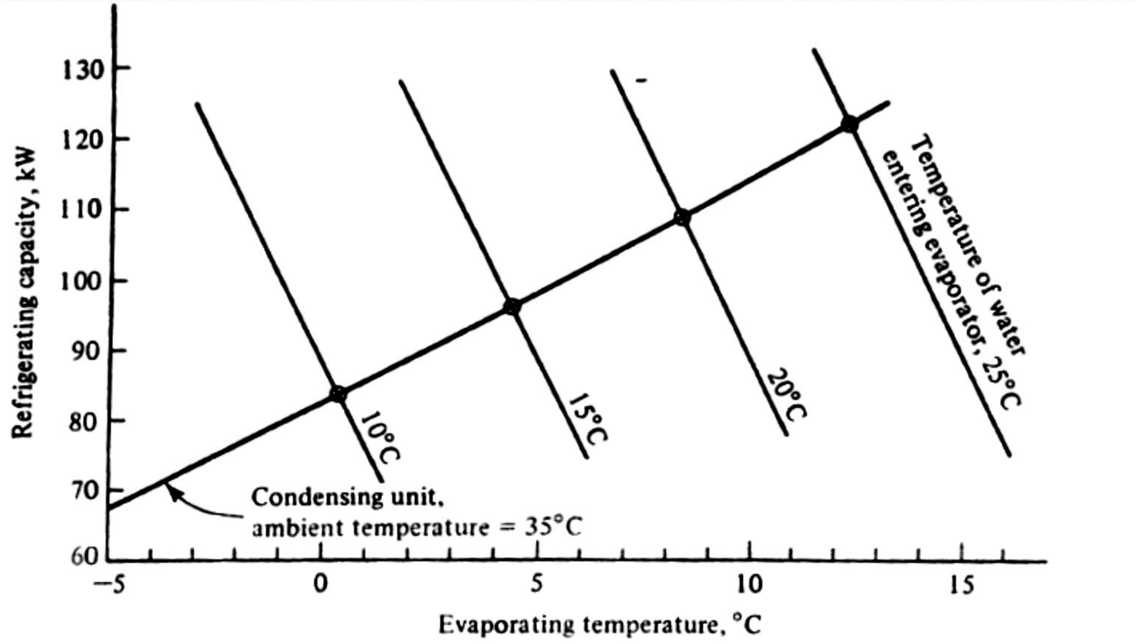


Figure (7) Performance of complete system found by determining the balance points of a condensing unit and an evaporator at various temperatures of entering water to be chilled and 35°C ambient temperature.

Simulation of complete system; mathematical analysis:

In the mathematical simulation it is not necessary to combine the components in pairs; instead the three components can be simulated simultaneously. The sequence of the calculation is shown by the information-flow diagram in Fig. (8).

1- Let the entering-water temperature is 20°C and the ambient temperature is 35° C.

2- Starting with trial values of te = l5°C and tc = 40°C.

3- Table (3) shows the values of the operating variables through the calculation loops proceed.

4- The converged values of capacity and te are (109.0 kW) and (8.2°C) respectively.

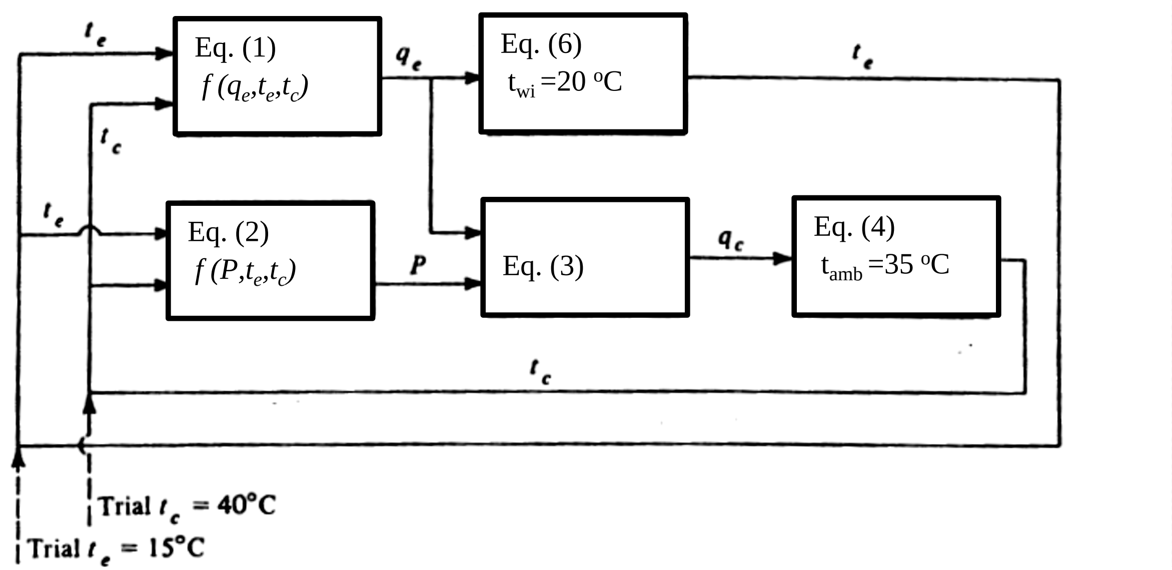


Figure (8) Information flow diagram for simulating the complete vapor-compression system with t_{wi} =20°c and tamb = 35°C.ambient temperature.

Table (3) Simulation using information-flow diagram of Fig. (8) for ambient temperature of 35° C and entering water temperature of 20° C

t_e, °C	t_c, °C	q_e, kW	P, kW	q_c, kW
15.0†	40.0†	158.0	26.2	184.2
4.6	54.6	87.6	31.8	119.4
10.0	47.7	120.4	31.0	151.4
7.3	51.1	103.5	31.8	135.3
8.7	49.4	111.8	31.5	143.3
8.0	50.3	107.7	31.6	140.3
8.3	49.8	109.7	31.6	140.8
8.2	50.0	108.7	31.6	140.3
8.2	49.9	109.2	31.6	140.8
8.2	50.0	109.0	31.6	140.6
8.2	50.0	109.1	31.6	140.7
8.2	50.0	109.0	31.6	140.6

From Table (3) check with the balance point for 20°C entering-water temperature from Fig. (7). In addition, the mathematical simulation shows tc = 50.0°C.

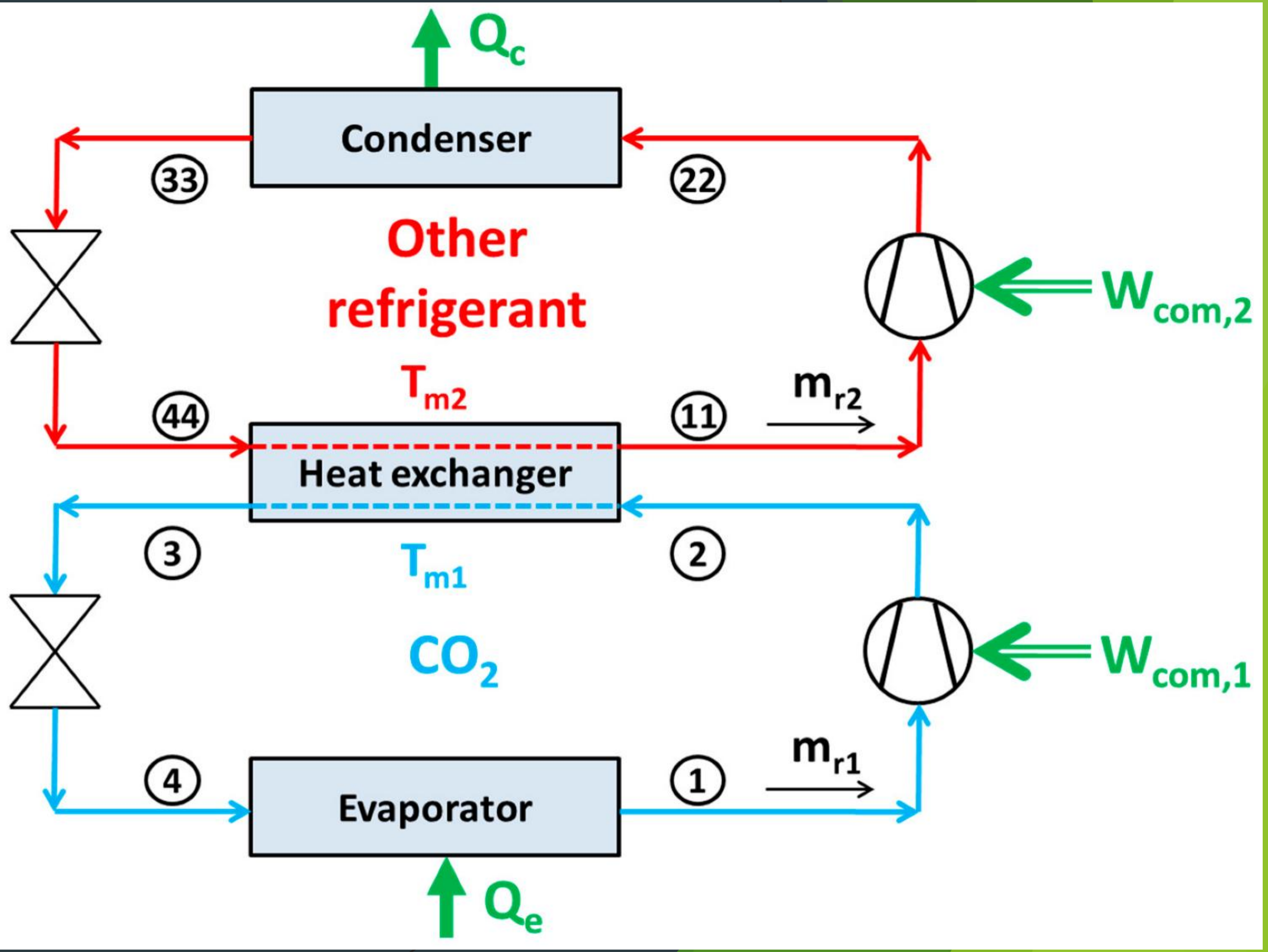
Cascade Refrigeration System

Cascade refrigeration system consist from two simple-vapor compression systems combined together.

The evaporator of the high-temperature system becomes the condenser of the low-temperature system.

The working media of the two systems are separated from each other.

The intermediate heat exchanger is also called a cascade heat exchanger or condenser.



- There are one or more number of intermediate cascades like the multi-stage refrigeration systems).
- There are many industrial, medical applications where very low temperatures are required such as:
- liquefaction of petroleum vapors
- liquefaction of atmospheric gases
- dry ice manufacture, etc.
- For example, the blood storage needs as low as - 80°C and precipitation hardening of special alloy steels needs around - 90°C.

Thermodynamic analysis of the cycle gives the mass flow rate through the high-temperature side by energy balance about heat exchanger:

$$0 = m_2.h_2 + m_8.h_8 - m_5.h_5 - m_3.h_3$$

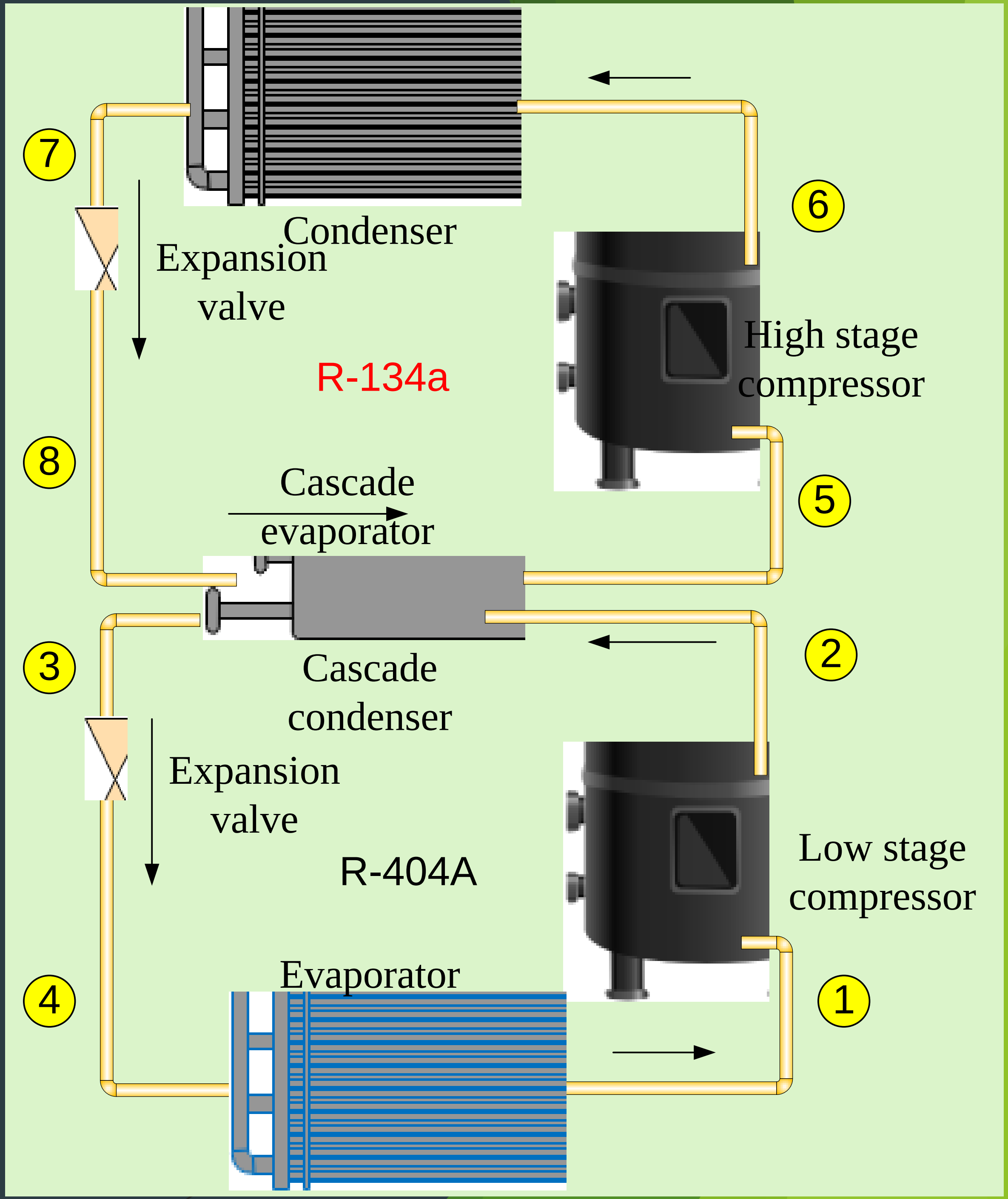
$$m_2 = m_3 = 1 \text{ and } m_5 = m_8$$

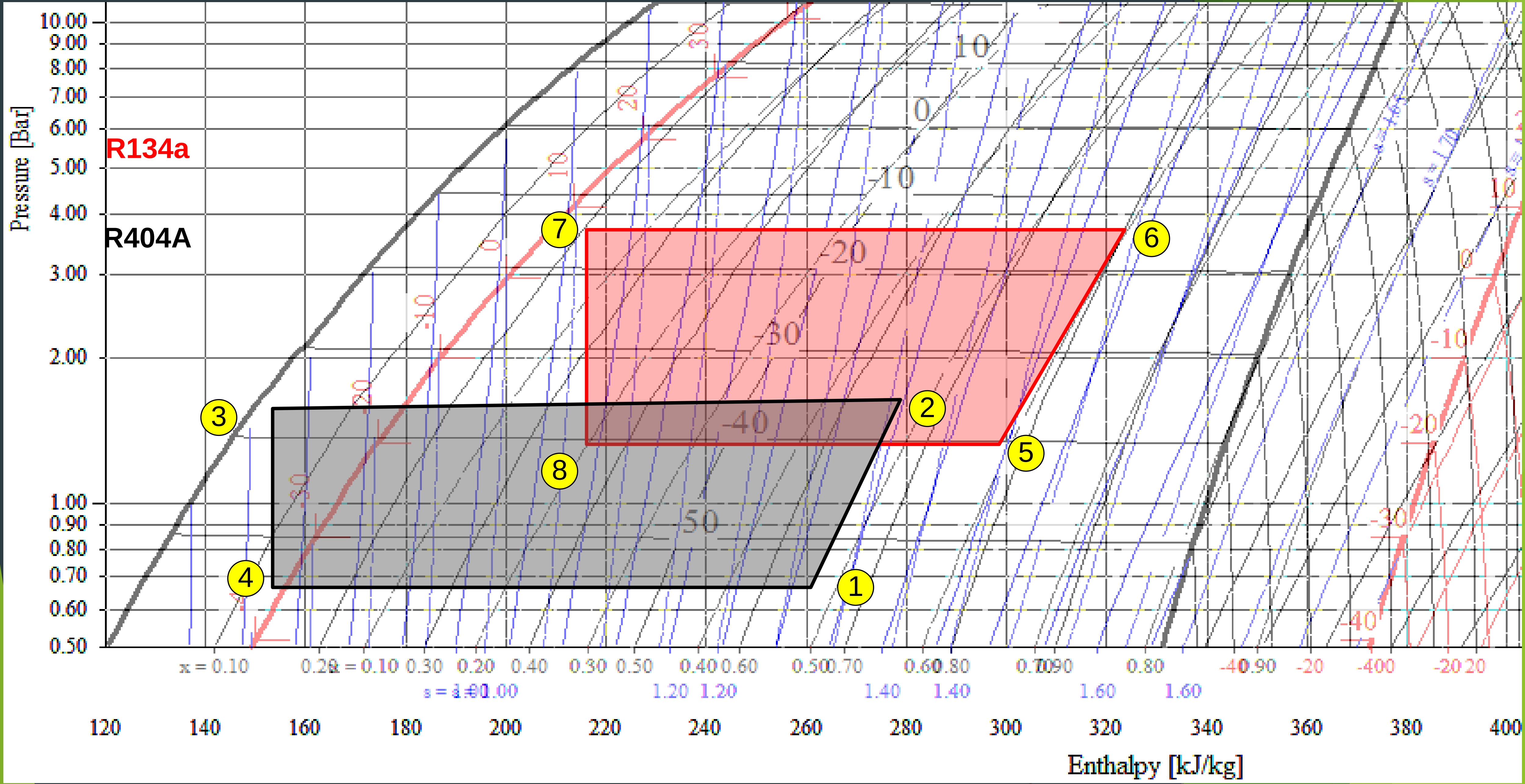
$$m_5 = \frac{(h_2 - h_3)}{(h_5 - h_8)}$$

The work input is found as:

$$w = m_1.(h_2 - h_1) + m_5.(h_6 - h_5)$$

$$COP = \frac{m_1.(h_1 - h_4)}{m_1.(h_2 - h_1) + m_5.(h_6 - h_5)}$$



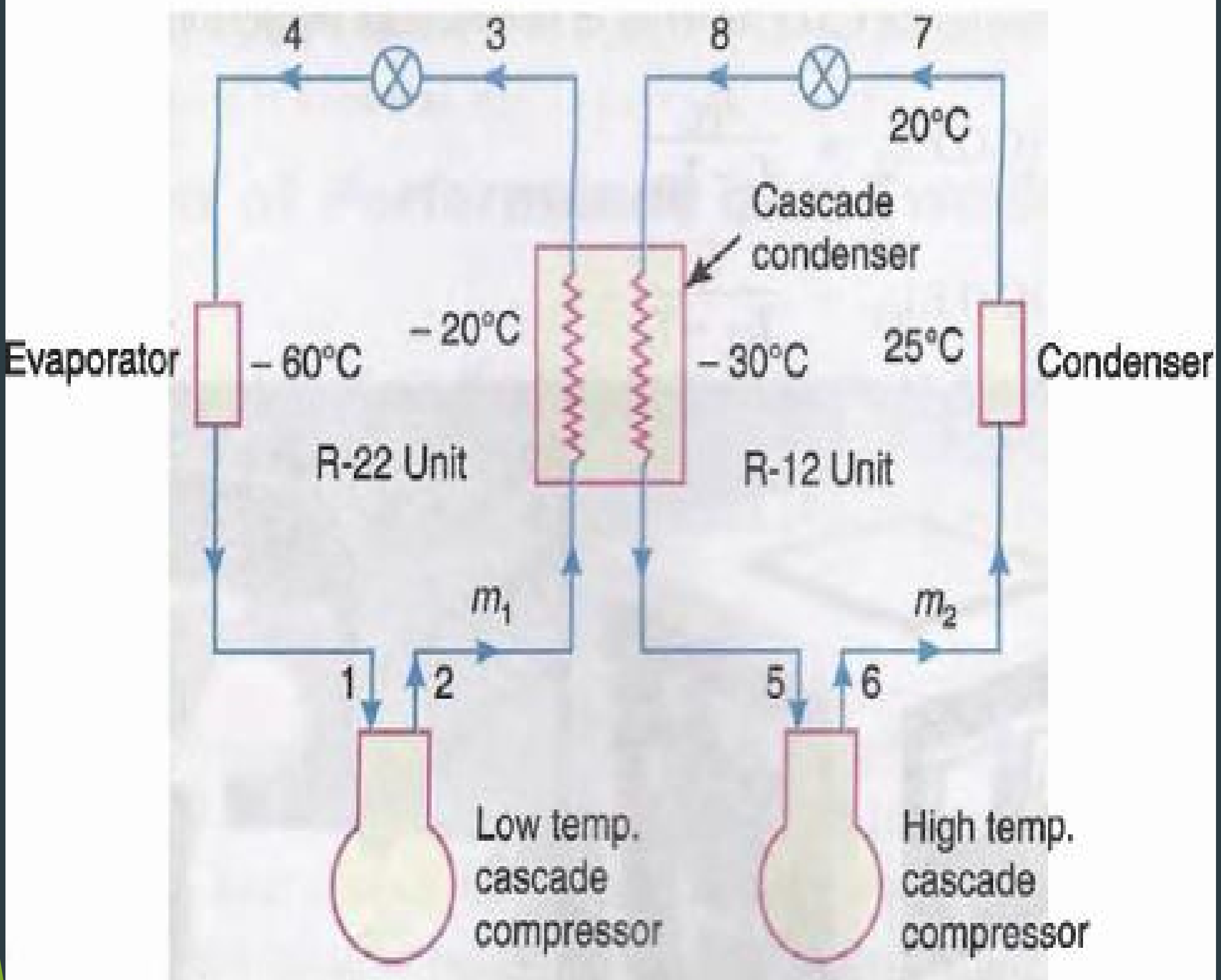


R134a
R404A
Pressure [Bar]
Enthalpy [kJ/kg]
1
2
3
4
5
6
7
8

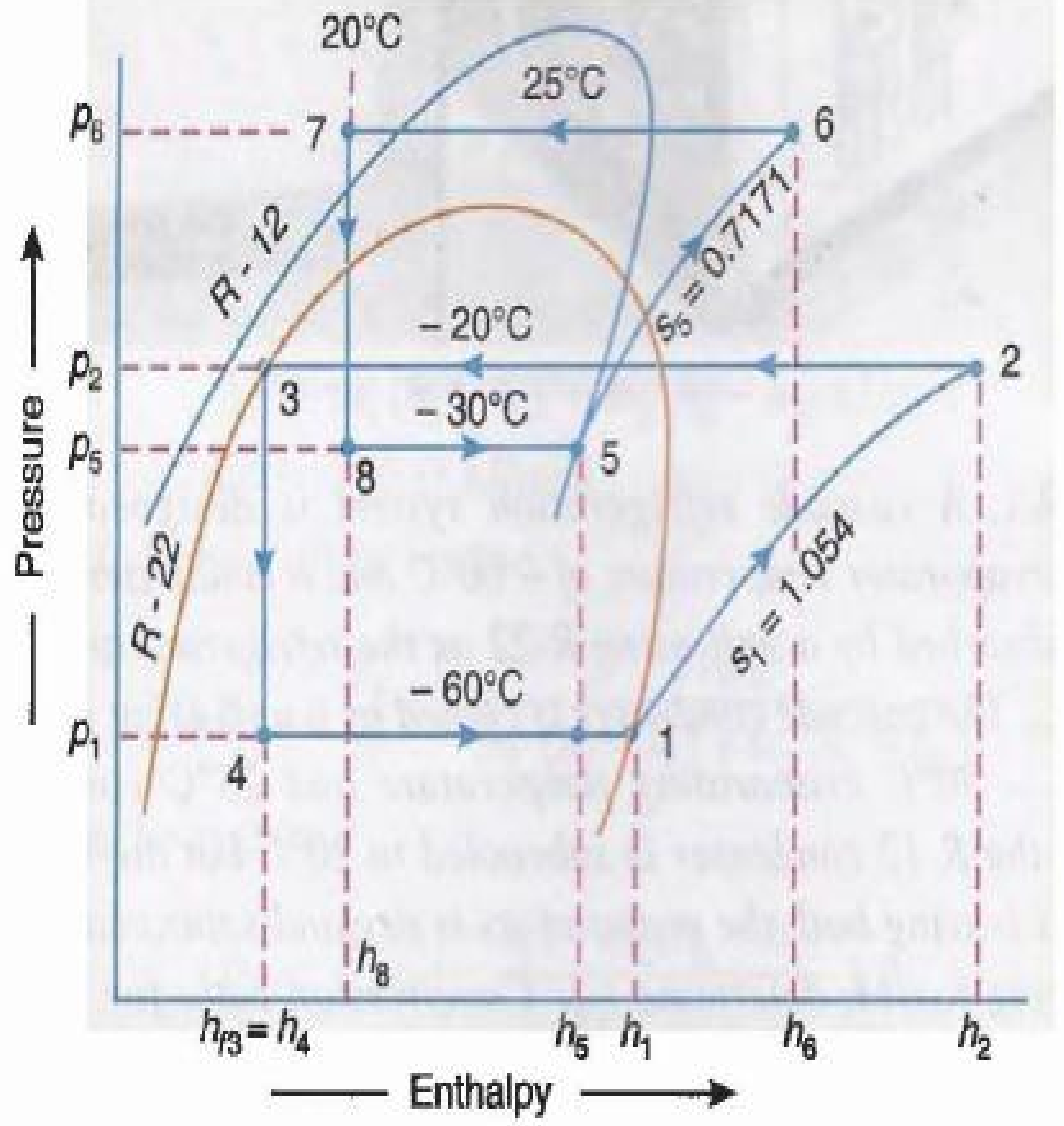
Example 14.1. (Page 427) A cascade refrigeration system is designed to supply 10 tons of refrigeration at an evaporator temperature of- 60°C and a condenser temperature of 25°C. The load at - 60°C is absorbed by a unit using R-22 as the refrigerant and is rejected to a cascade condenser at - 20°C. The cascade condenser is cooled by a unit using R-12 as the refrigerant and operating between - 30°C evaporating temperature and 25°C condenser temperature. The refrigerant leaving the R-12 condenser is subcooled to 20°C but there is no sub-cooling of R-22 refrigerant. The gas leaving both the evaporators is dry and saturated and the compressions are isentropic. Neglecting losses,

Determine:

1. Compression ratio for each unit; 2. Quantity of refrigerant circulated per minute for each unit; 3. C. O.P. for each unit; 4. C. O.P. of the whole system; and 5. Theoretical power required to run the system



(a) Schematic diagram of a two-stage cascade refrigerating system.



(b) p-h diagram.

Solution:

For the R-22 cycle

From the R-22 table the enthalpy of saturated vapor refrigerant at point 1,

$$h_1 = 378.59\ kJ/kg$$

$$s_1 = s_2 = 1.877\ kJ/kg°K$$

To find the enthalpy at point 2 we apply,

$$s_2 = s_{2,g} + cp_{2,g} \times \ln \frac{T_2}{T_{2,g}}$$

$$1.877 = 1.7826 + 0.665 \times \ln \frac{T_2}{(-20 + 273)} \rightarrow T_2 = 291.5°K = 18.5°C$$

$$h_2 = h_{2,g} + cp_{2,g} \times (T_2 - T_{2,g})$$

$$h_2 = 397.06 + 0.665 \times \left(18.5 - (-20)\right) \rightarrow h_2 = 422.6\ kJ/kg$$

From the R-22 table the enthalpy of saturated liquid refrigerant at point 3,

$$h_3 = h_4 = 177.04\ kJ/kg$$

For the R-12 cycle

Now from the R-12 table the enthalpy of saturated vapor refrigerant at point 5,

$$h_5 = 338.94\ \mathrm{kJ/kg}$$

$$s_5 = s_6 = 1.5784\ \mathrm{kJ/kg°K}$$

To find the enthalpy at point 6 we apply,

$$s_6 = s_{6,g} + cp_{6,g} \times \ln\frac{T_6}{T_{6,g}}$$

$$1.5784 = 1.5506 + 0.7 \times \ln\frac{T_6}{(25+273)} \rightarrow T_6 = 310\ °K = 37\ ℃$$

$$h_6 = h_{6,g} + cp_{6,g} \times (T_6 - T_{6,g})$$

$$h_6 = 363.3 + 0.7 \times (37 - 25) \rightarrow h_6 = 371.7\ \mathrm{kJ/kg}$$

To find the enthalpy at point 7 we apply,

$$h_7 = h_{7,f} + cp_{7,f} \times (T_7 - T_{7,f})$$

$$h_7 = 224.06 + 0.9885 \times (20 - 25) \rightarrow h_7 = 219.12\,\frac{\mathrm{kJ}}{\mathrm{kg}} = h_8$$

1. Quantity of refrigerant circulated per minute for each unit

$$\text{For R} - 22 \text{ unit} \rightarrow \dot{m_1} = \frac{10 \times 3.516}{h_1 - h_4} = \frac{35.16}{378.59 - 177.04}$$

$$= 0.1744 \frac{\text{kg}}{\text{s}} \text{ or } 10.46 \text{ kg/min}$$

$$\text{For R} - 12 \text{ unit} \rightarrow \dot{m_2}(h_5 - h_8) = \dot{m_1}(h_2 - h_3)$$

$$\dot{m_2}(338.94 - 219.12) = 0.1744(422.6 - 177.04) \rightarrow \dot{m_2}$$

$$= 0.357 \frac{\text{kg}}{\text{s}} \text{ or } 21.4 \text{ kg/min}$$

2. COP for each unit

$$\text{For } R-22 \text{ unit COP} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{378.59 - 177.04}{422.6 - 378.59} = 4.58$$

$$\text{For } R-12 \text{ unit COP} = \frac{h_5 - h_8}{h_6 - h_5} = \frac{338.94 - 219.12}{371.7 - 338.94} = 3.65$$

3. COP of the whole system

$$COP_{\text{Total}} = \frac{\dot{m}_1 (h_1 - h_4)}{\dot{m}_1 (h_2 - h_1) + \dot{m}_2 (h_6 - h_5)}$$

$$COP_{\text{Total}} = \frac{0.1744 \times (378.59 - 177.04)}{0.1744 \times (422.6 - 378.59) + 0.357 \times (371.7 - 338.94)} = 1.815$$

4. Theoretical power required to run the system

$$\text{power} = \dot{m}_1 \times (h_2 - h_1) + \dot{m}_2 \times (h_6 - h_5)$$
$$= 0.1744 \times (422.6 - 378.59) + 0.357 \times (371.7 - 338.94)$$
$$= 7.67 + 11.69 = 19.36\ \text{kW}$$

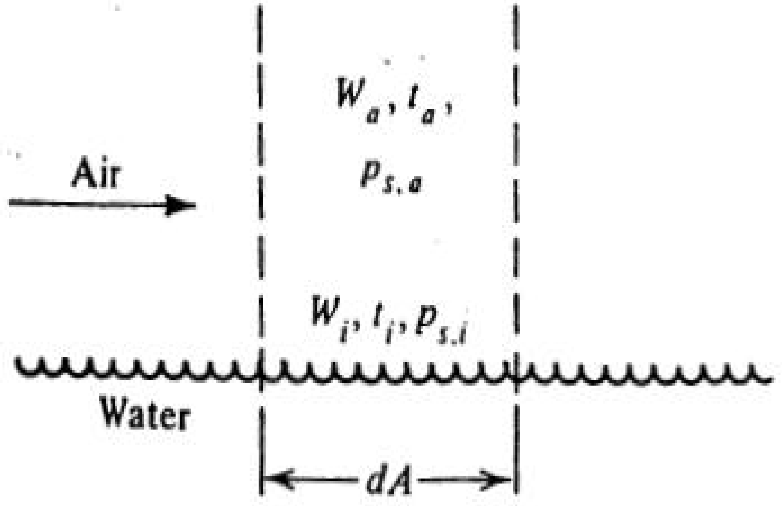
Cooling Tower

Transfer of sensible and latent heat with a wetted surface:

When air flows on a wetted surface, as shown in Figure, there are a transfer of both sensible and latent heat.

If there is a difference in temperature between the air t_a and the wetted surface t_i, heat will be transferred as sensible heat.

If there is a difference in the partial pressure of water vapor in the air $P_{s,a}$ and that of the water $P_{s,i}$ there will be a transfer of mass (water).



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The rate of sensible-heat transfer from the water surface to the air q_s can be calculated by the convection equation

$$dq_s = h_c \, dA \, (t_i - t_a) \qquad (1)$$

Where:

q_s = rate of sensible-heat transfer, W

h_c= convection coefficient, W/m2·K

A= area m^2

The rate of latent-heat transfer from the water surface to the air q_L can be calculated by the convection equation

$$dq_L = h_D \, dA \, (W_i - W_a) h_{fg} \qquad (2)$$

Where

h_D = proportionality constant, kg/m^2 (This proportionality is expressed by $h_D = h_c / c_{pm}$)

W_i = humidity ratio of saturated air at wetted-surface temperature

q_L = rate of latent-heat transfer, W

h_{fg} = latent heat of water at t_i J /kg

c_{pm} = is the specific heat of moist air, J /kg.^{o}K.

The total heat transfer of (sensible plus latent)

$$dq_t = dq_s + dq_L = h_c \, dA \, (t_i - t_a) + h_D \, dA \, (W_i - W_a) h_{fg}$$

$$dq_t = \frac{h_c \, dA}{c_{pm}} (h_i - h_a) \qquad (3)$$

Where

h_c= convection coefficient, kW/m^2 ^{o}K

h_i= enthalpy of saturated air at the water temperature, kJ/(kg dry air)

h_a =enthalpy of air, kJ/(kg dry air)

c_{pm} =specific heat of moist air, kJ/kg ^{o}K

Notes:

1. The value of $(\frac{h_c\, dA}{c_{pm}})$ is a function of the dynamics airflow in the cooling tower, and the magnitude remains essentially constant for a given cooling tower with a constant airflow rate and water flow rate.

2. When the value of $(\frac{h_c\, dA}{c_{pm}})$ is known and the entering air and water flow rates and conditions are known, it should be possible to predict the ***outlet water temperature***.

3. Thus new characterizes of the cooling tower can be predicted at other inlet water temperatures and other inlet wet-bulb temperatures.

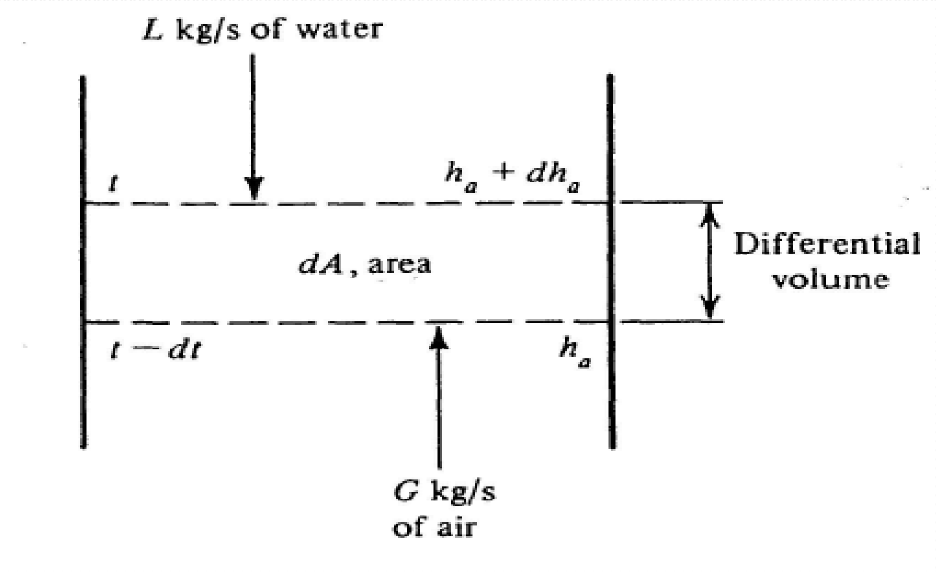
$\frac{h_c\, dA}{c_{pm}}$: هي كمية الطاقة التي يفقدها الماء او برج التبريد لكل Δh في الهواء

Analysis of a counter flow cooling tower

The rate of heat removed from the water d_q is equal to the rate gained by the air:

$$dq = G\, dh_a = L(4.19)dt \qquad (4)$$

Also the rate of heat transfer through the cooling tower dq can be found from Eq. (3)



To find the rate of heat transferred by the entire cooling tower, Eqs. (3 and 4) must be integrated. Combining Eqs. (3) and (4), rearranging, and-integrating gives:

$$4.19\,L \int_{t_{out}}^{t_{in}} \frac{dt}{h_i - h_a} = \int_0^A \frac{h_c dA}{c_{pm}} = \frac{h_c A}{c_{pm}} \qquad (5)$$

Where t_{in} and t_{out} are the water temperatures entering and leaving the tower, respectively, and the integration of Eq. (5) is a numerical process indicated by:

$$\frac{h_c A}{c_{pm}} = 4.19\,L \;\; \Delta t \; \sum \frac{1}{(h_i - h_a)_m} \qquad (6)$$

Where $(h_i - h_a)_m$ is the arithmetic-mean enthalpy difference for an increment of volume. The procedure will be illustrated in the following Example

Example 19-1 A counter flow cooling tower operates with a water flow rate of 18.8 kg/s and an airflow rate of 15.6 kg/s. When the wet-bulb temperature of entering air is 25°C and the entering water temperature is 34°C, the leaving water temperature *is* 29°C. Calculate $\frac{h_c\,dA}{c_{pm}}$ for this cooling tower.

L=18.8 kg/s (masss flow rate of water)

G=15.6 kg/s (masss flow rate of air)

t_a=25 ℃ (wet–bulb temperature of enterning air)

$t_{i,in}$=34 ℃ (temperature of enterning water)

$t_{i,out}$=29 ℃ (temperature of leaving water)

Solution:

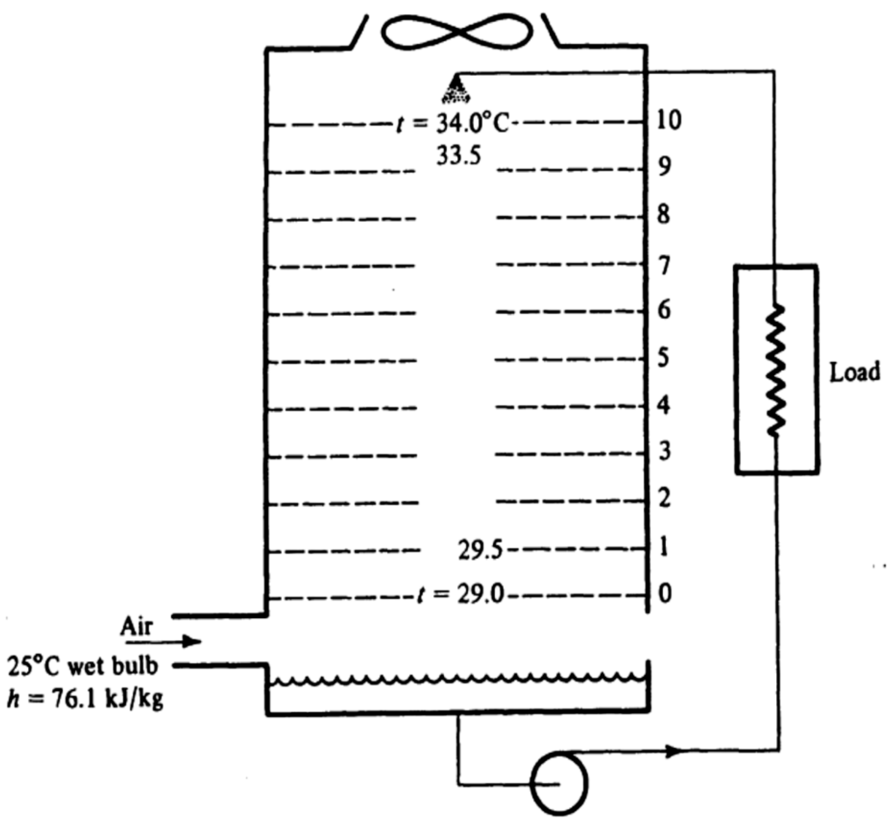
- The cooling tower will be imagined to be divided into 10 sections, as shown in Figure.
- Therefore, the water temperature dropping 0.5 ^oC in each section.
- Find the enthalpy of the inlet air $h_{a,0}$ at its properties (if you have only wet bulb temperature let the relative humidity of inlet air equal to 50 percent) ($h_{a,0}$ = 76.1 kJ/kg)
- Apply the equation (4) to find the enthalpy of air $h_{a,1}$ at section (1).

$$h_{a,1} - h_{a,0} = \frac{L}{G}(4.19)\,(0.5) = 2.53$$

$$h_{a,1} = 76.1 + 2.53 = 78.63\,\frac{kJ}{kg}$$

- The average enthalpy at section (0-1)

$$h_{0-1} = \frac{76.1 + 78.63}{2} = 77.36\,\frac{kJ}{kg}$$



- Then the average enthalpy at section (1-2) can determine as:

h_{1-2} = 77.35 + 2.53 = 79.89 kJ/kg

(As same section 1-2 find average enthalpies for the other remaining section as shown in table)

- Find the average enthalpies of air h_i at average water temperatures for each section from Table A-2 thermodynamic properties of saturated air.

- Find the value of ($h_i - h_a$) in each sections.

- The value of $\frac{h_c A}{c_{pm}}$ can now be calculated from Eq. (6) as

$$\frac{h_c A}{c_{pm}} = 18.8 \times 4.19 \times 0.5 \times 0.5097$$

$$= 20.08\ kW/(\frac{kJ}{kg}\ of\ enthalpy\ difference)$$

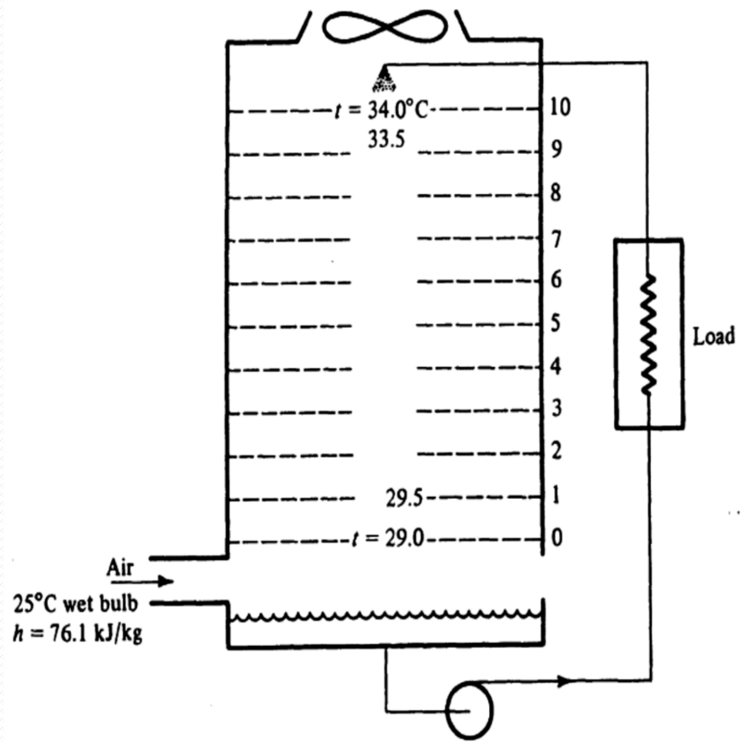


Table 19-1 Stepwise integration for solving Example 19-1

Section	Mean water temperature, °C	Average h_a, kJ/kg	Average h_i, kJ/kg	$(h_i - h_a)_m$	$\frac{1}{(h_i - h_a)_m}$
0-1	29.25	77.35	96.13	18.77	0.05328
1-2	29.75	79.89	98.70	18.81	0.05316
2-3	30.25	82.42	101.32	18.90	0.05291
3-4	30.75	84.95	104.00	19.05	0.05249
4-5	31.25	87.48	106.74	19.26	0.05192
5-6	31.75	90.01	109.54	19.53	0.05120
6-7	32.25	92.54	112.41	19.87	0.05033
7-8	32.75	95.07	115.35	20.28	0.04931
8-9	33.25	97.60	118.36	20.76	0.04817
9-10	33.75	100.13	121.43	21.30	0.04695
					0.5097