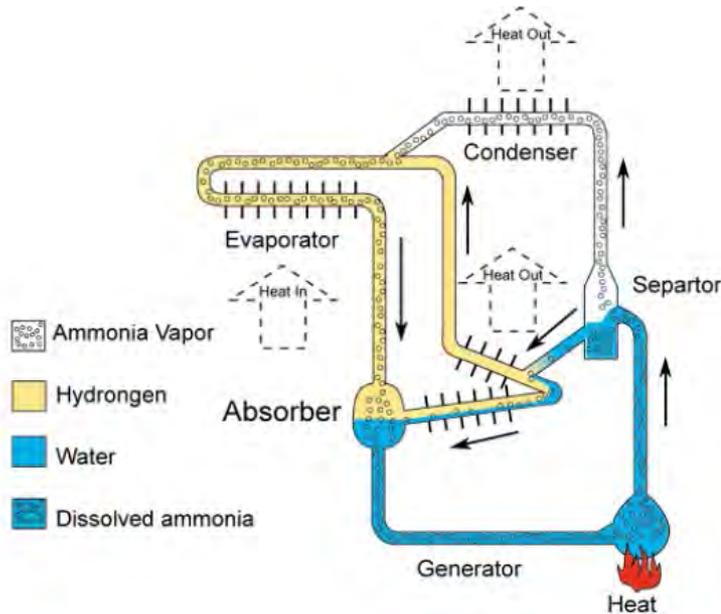


# REFRIGERATION



**MR. RAJKISHOR SINGH**  
**ASSISTANT PROFESSOR**  
**SHOBHIT INSTITUTE OF ENGINEERING AND TECHNOLOGY**  
**DEEMED TO BE UNIVERSITY, MEERUT - 250110**



# INTRODUCTION

Refrigeration may be defined as the process of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature. One of the most important applications of refrigeration has been the preservation of perishable food products by storing them at low temperatures. Refrigeration systems are also used extensively for providing thermal comfort to human beings by means of air conditioning. Air Conditioning refers to the treatment of air so as to simultaneously control its temperature, moisture content, cleanliness, odour and circulation, as required by occupants, a process, or products in the space.



# NATURAL REFRIGERATION

In olden days refrigeration was achieved by natural means such as the use of ice or evaporative cooling. In earlier times, ice was either:

- Transported from colder regions,
- Harvested in winter and stored in ice houses for summer use or,
- Made during night by cooling of water by *radiation to stratosphere*.

In Europe, America and Iran a number of icehouses were built to store ice. Materials like sawdust or wood shavings were used as insulating materials in these icehouses. Later on, cork was used as insulating material. Literature reveals that ice has always been available to aristocracy who could afford it. In India, the Mogul emperors were very fond of ice during the harsh summer in Delhi and Agra, and it appears that the ice used to be made by nocturnal cooling.



# ART OF ICE MAKING BY NOCTURNAL COOLING

The art of making ice by nocturnal cooling was perfected in India. In this method ice was made by keeping a thin layer of water in a shallow earthen tray, and then exposing the tray to the night sky. Compacted hay of about 0.3 m thickness was used as insulation. The water loses heat by radiation to the stratosphere, which is at around  $-55^{\circ}\text{C}$  and by early morning hours the water in the trays freezes to ice. This method of ice production was very popular in India.



# EVAPORATIVE COOLING

As the name indicates, evaporative cooling is the process of reducing the temperature of a system by evaporation of water. Human beings perspire and dissipate their metabolic heat by evaporative cooling if the ambient temperature is more than skin temperature. Animals such as the hippopotamus and buffalo coat themselves with mud for evaporative cooling. Evaporative cooling has been used in India for centuries to obtain cold water in summer by storing the water in earthen pots. The water permeates through the pores of earthen vessel to its outer surface where it evaporates to the surrounding, absorbing its latent heat in part from the vessel, which cools the water. It is said that Patliputra University situated on the bank of river Ganges used to induce the evaporative-cooled air from the river. Suitably located chimneys in the rooms augmented the upward flow of warm air, which was replaced by cool air. Evaporative cooling by placing wet straw mats on the windows is also very common in India. The straw mat made from “khus” adds its inherent perfume also to the air. Now-a-days desert coolers are being used in hot and dry areas to provide cooling in summer.



# COOLING BY SALT SOLUTIONS

Certain substances such as common salt, when added to water dissolve in water and absorb its heat of solution from water (endothermic process). This reduces the temperature of the solution (water+salt). Sodium Chloride salt ( $\text{NaCl}$ ) can yield temperatures up to  $-20^{\circ}\text{C}$  and Calcium Chloride ( $\text{CaCl}_2$ ) up to  $-50^{\circ}\text{C}$  in properly insulated containers. However, as it is this process has limited application, as the dissolved salt has to be recovered from its solution by heating.

**Q1.** The disadvantages of natural refrigeration methods are:

- a) They are expensive
- b) They are uncertain
- c) They are not environment friendly
- d) They are dependent on local conditions

**Q2.** Evaporative cooling systems are ideal for:

- a) Hot and dry conditions
- b) Hot and humid conditions
- c) Cold and humid conditions
- d) Moderately hot but humid conditions



# ARTIFICIAL REFRIGERATION

Refrigeration as it is known these days is produced by artificial means. Though it is very difficult to make a clear demarcation between natural and artificial refrigeration, it is generally agreed that the history of artificial refrigeration began in the year 1755, when the Scottish professor William Cullen made the first refrigerating machine, which could produce a small quantity of ice in the laboratory. Based on the working principle, refrigeration systems can be classified as vapour compression systems, vapour absorption systems, gas cycle systems etc.



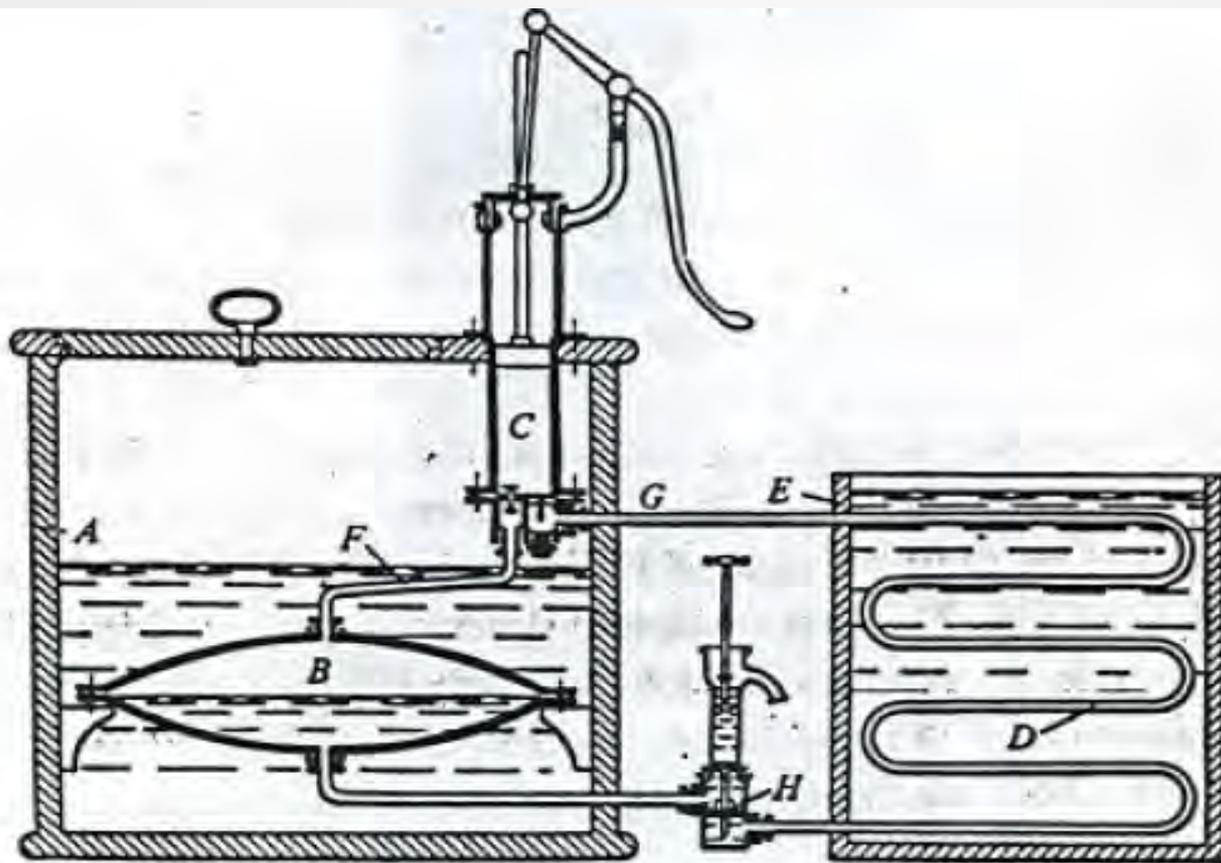
# VAPOUR COMPRESSION REFRIGERATION SYSTEMS

If this process of cooling is to be made continuous the vapors have to be recycled by condensation to the liquid state. The condensation process requires heat rejection to the surroundings. It can be condensed at atmospheric temperature by increasing its pressure. The process of condensation was learned in the second half of eighteenth century. U.F. Clouet and G. Monge liquefied  $\text{SO}_2$  in 1780 while van Marum and Van Troostwijk liquefied  $\text{NH}_3$  in 1787. Hence, a compressor is required to maintain a high pressure so that the evaporating vapours can condense at a temperature greater than that of the surroundings.

Oliver Evans in his book “Abortion of a young Steam Engineer’s Guide” published in Philadelphia in 1805 described a closed refrigeration cycle to produce ice by ether under vacuum. Jacob Perkins, an American living in London actually designed such a system in 1835. The apparatus described by Jacob Perkins in his patent specifications of 1834 is shown in Fig. In his patent he stated “I am enabled to use volatile fluids for the purpose of producing the cooling or freezing of fluids, and yet at the same time constantly condensing such volatile fluids, and bringing them again into operation without waste”.



# VAPOUR COMPRESSION REFRIGERATION SYSTEMS



*Apparatus described by Jacob Perkins in his patent specification of 1834. The refrigerant (ether or other volatile fluid) boils in evaporator B taking heat from surrounding water in container A. The pump C draws vapour away and compresses it to higher pressure at which it can condense to liquids in tubes D, giving out heat to water in vessel E. Condensed liquid flows through the weight loaded valve H, which maintains the difference of pressure between the condenser and evaporator. The small pump above H is used for charging the apparatus with refrigerant.*

# VAPOUR COMPRESSION REFRIGERATION SYSTEMS

John Hague made Perkins's design into working model with some modifications. This Perkins machine is shown in Fig. The earliest vapour compression system used either sulphuric (ethyl) or methyl ether. The American engineer Alexander Twining (1801-1884) received a British patent in 1850 for a vapour compression system by use of ether,  $\text{NH}_3$  and  $\text{CO}_2$ .

The man responsible for making a practical vapor compression refrigeration system was James Harrison who took a patent in 1856 for a vapour compression system using ether, alcohol or ammonia. Charles Tellier of France patented in 1864, a refrigeration system using dimethyl ether which has a normal boiling point of  $-23.6^\circ\text{C}$ .

# VAPOUR COMPRESSION REFRIGERATION SYSTEMS



*Perkins machine built by John Hague*



# DOMESTIC REFRIGERATION SYSTEMS

The domestic refrigerator using natural ice (domestic ice box) was invented in 1803 and was used for almost 150 years without much alteration. The domestic ice box used to be made of wood with suitable insulation. Ice used to be kept at the top of the box, and low temperatures are produced in the box due to heat transfer from ice by natural convection. A drip pan is used to collect the water formed due to the melting of ice. The box has to be replenished with fresh ice once all the ice melts. Though the concept is quite simple, the domestic ice box suffered from several disadvantages. The user has to replenish the ice as soon as it is consumed, and the lowest temperatures that could be produced inside the compartment are limited. In addition, it appears that warm winters caused severe shortage of natural ice in USA.

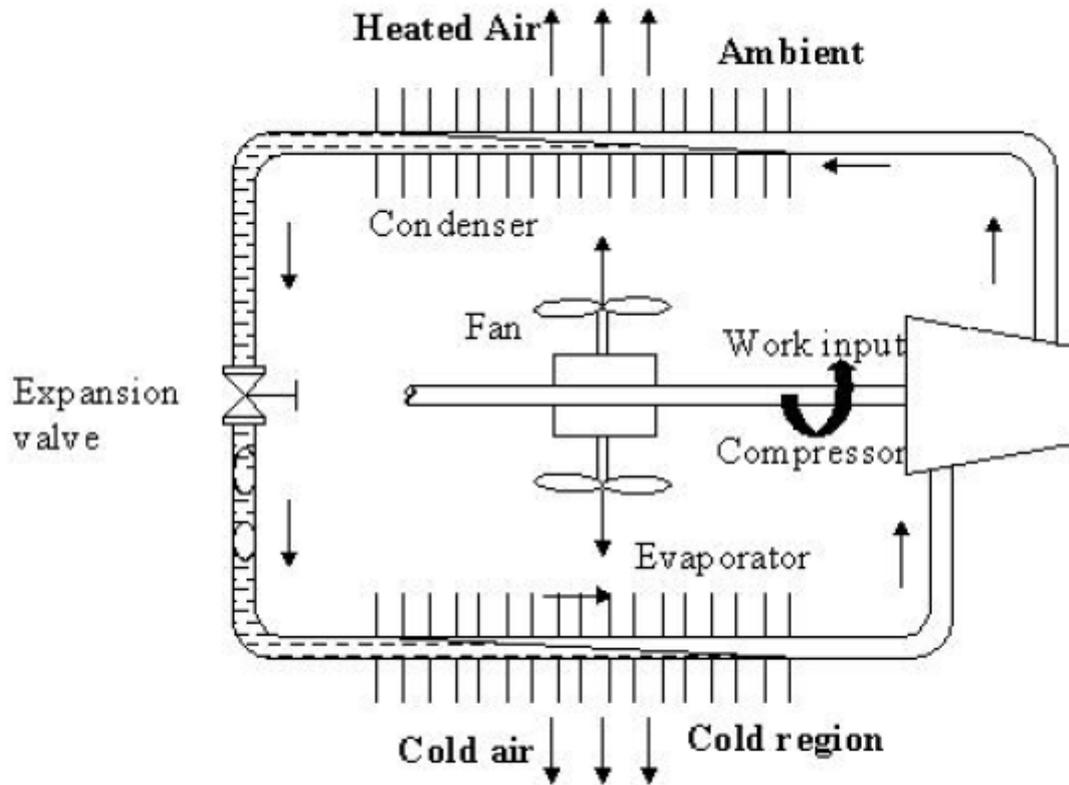


# AIR CONDITIONING SYSTEMS

Refrigeration systems are also used for providing cooling and dehumidification in summer for personal comfort (air conditioning). The first air conditioning systems were used for industrial as well as comfort air conditioning. Eastman Kodak installed the first air conditioning system in 1891 in Rochester, New York for the storage of photographic films. An air conditioning system was installed in a printing press in 1902 and in a telephone exchange in Hamburg in 1904. Many systems were installed in tobacco and textile factories around 1900. The first domestic air conditioning system was installed in a house in Frankfurt in 1894. A private library in St Louis, USA was air conditioned in 1895, and a casino was air conditioned in Monte Carlo in 1901. Efforts have also been made to air condition passenger rail coaches using ice. The widespread development of air conditioning is attributed to the American scientist and industrialist Willis Carrier. Carrier studied the control of humidity in 1902 and designed a central air conditioning plant using air washer in 1904. Due to the pioneering efforts of Carrier and also due to simultaneous development of different components and controls, air conditioning quickly became very popular, especially after 1923. At present comfort air conditioning is widely used in residences, offices, commercial buildings, air ports, hospitals and in mobile applications such as rail coaches, automobiles, aircrafts etc.



# AIR CONDITIONING SYSTEMS



Vapour compression system



# VAPOUR ABSORPTION REFRIGERATION SYSTEMS

John Leslie in 1810 kept  $\text{H}_2\text{SO}_4$  and water in two separate jars connected together.  $\text{H}_2\text{SO}_4$  has very high affinity for water. It absorbs water vapour and this becomes the principle of removing the evaporated water vapour requiring no compressor or pump.  $\text{H}_2\text{SO}_4$  is an absorbent in this system that has to be recycled by heating to get rid of the absorbed water vapour, for continuous operation. Windhausen in 1878 used this principle for absorption refrigeration system, which worked on  $\text{H}_2\text{SO}_4$ . Ferdinand Carre invented aqua-ammonia absorption system in 1860. Water is a strong absorbent of  $\text{NH}_3$ . If  $\text{NH}_3$  is kept in a vessel that is exposed to another vessel containing water, the strong absorption potential of water will cause evaporation of  $\text{NH}_3$  requiring no compressor to drive the vapours. A liquid pump is used to increase the pressure of strong solution. The strong solution is then heated in a generator and passed through a rectification column to separate the water from ammonia. The ammonia vapour is then condensed and recycled.

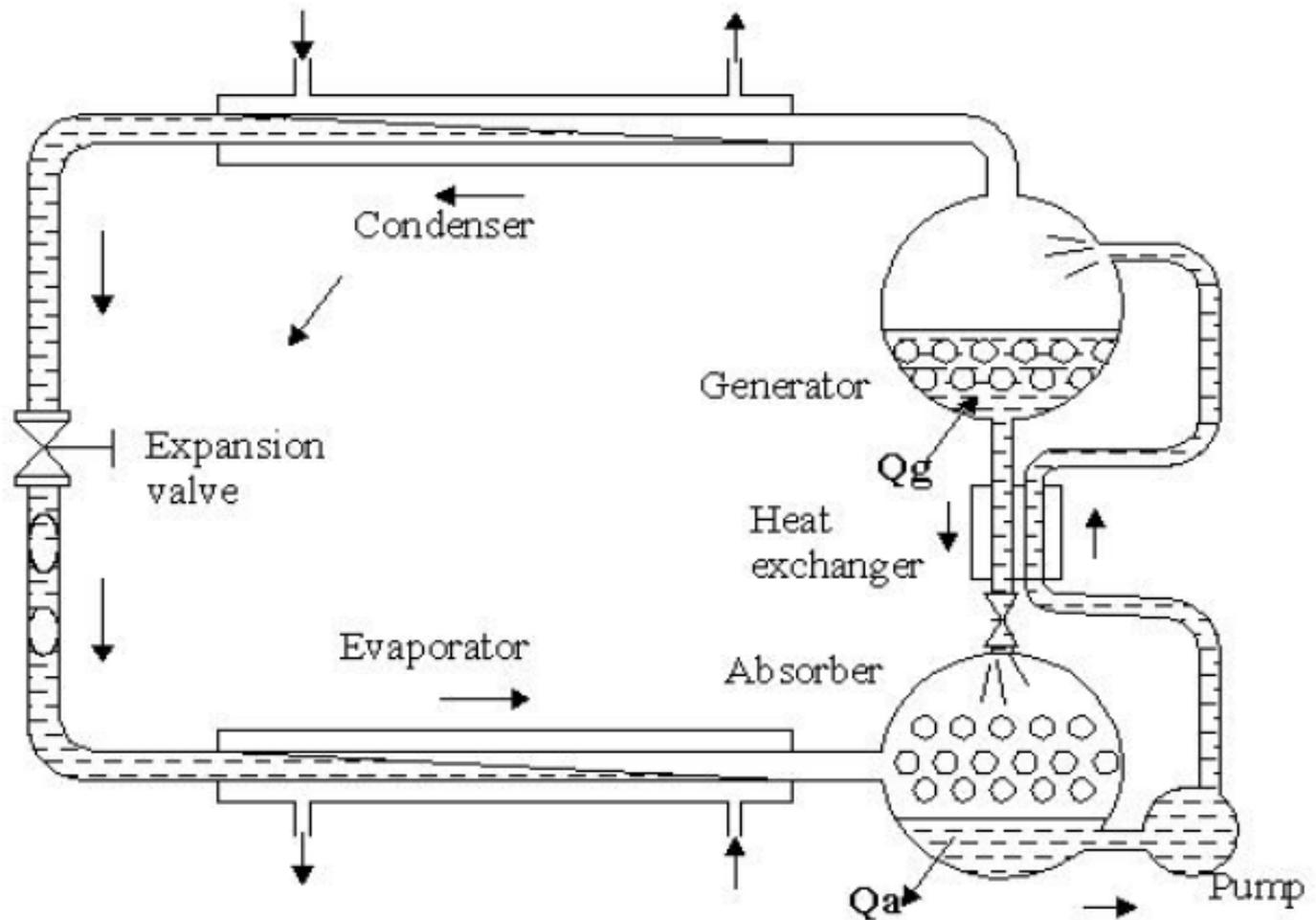


# VAPOUR ABSORPTION REFRIGERATION SYSTEMS

The pump power is negligible hence; the system runs virtually on low-grade energy used for heating the strong solution to separate the water from ammonia. These systems were initially run on steam. Later on oil and natural gas based systems were introduced. Figure shows the essential components of a vapour absorption refrigeration system. In 1922, Balzar von Platen and Carl Munters, two students at Royal Institute of Technology, Stockholm invented a three fluid system that did not require a pump. A heating based bubble pump was used for circulation of strong and weak solutions and hydrogen was used as a non-condensable gas to reduce the partial pressure of  $\text{NH}_3$  in the evaporator. Geppert in 1899 gave this original idea but he was not successful since he was using air as non-condensable gas. The Platen-Munters refrigeration systems are still widely used in certain niche applications such as hotel rooms etc. Figure shows the schematic of the triple fluid vapour absorption refrigeration system.



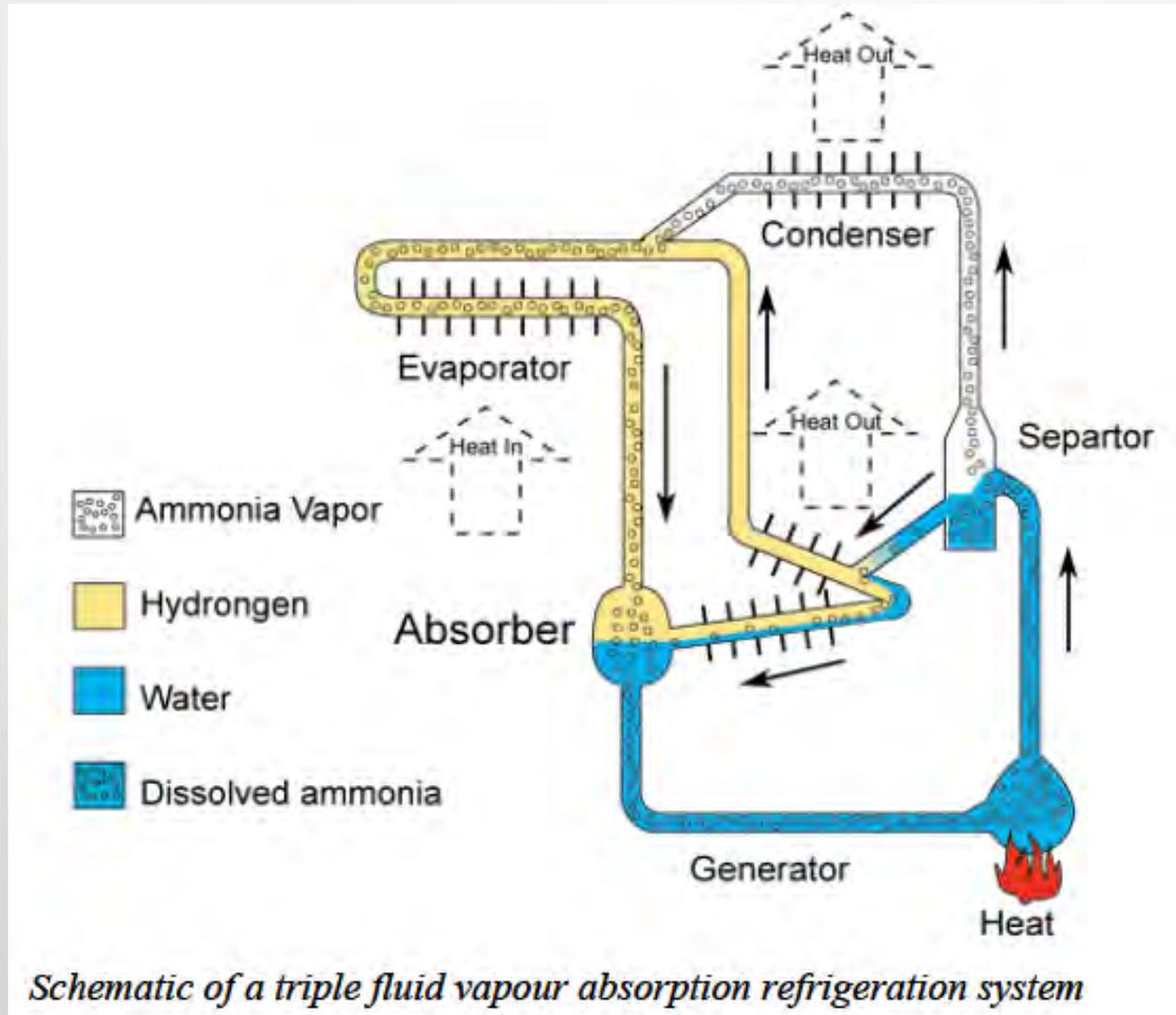
# VAPOUR ABSORPTION REFRIGERATION SYSTEMS



Vapour absorption system



# VAPOUR ABSORPTION REFRIGERATION SYSTEMS





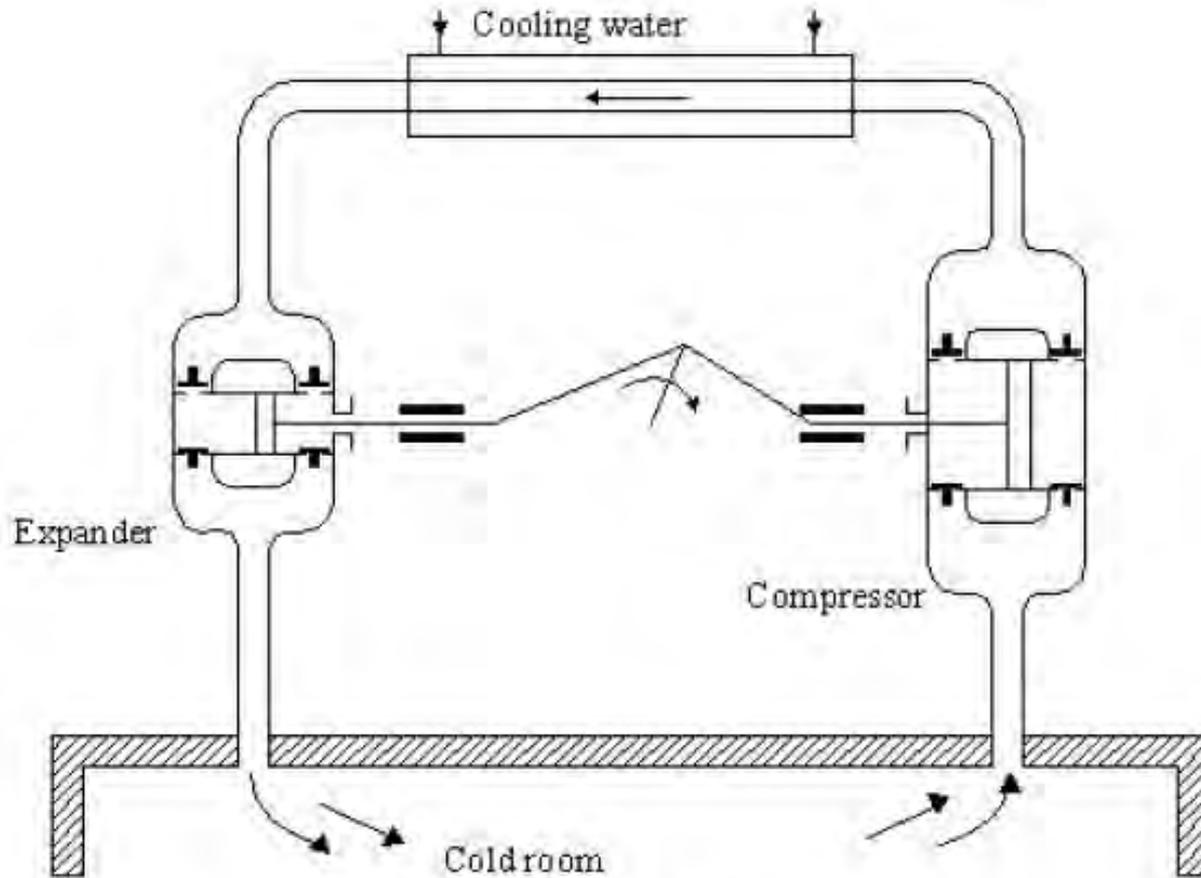
# GAS CYCLE REFRIGERATION

If air at high pressure expands and does work (say moves a piston or rotates a turbine), its temperature will decrease. This fact is known to man as early as the 18<sup>th</sup> century. Dalton and Gay Lusaac studied this in 1807. Sadi Carnot mentioned this as a well-known phenomenon in 1824. However, Dr. John Gorrie a physician in Florida developed one such machine in 1844 to produce ice for the relief of his patients suffering from fever. This machine used compressed air at 2 atm. pressure and produced brine at a temperature of  $-7^{\circ}\text{C}$ , which was then used to produce ice. Alexander Carnegie Kirk in 1862 made an air cycle cooling machine. This system used steam engine to run its compressor. Using a compression ratio of 6 to 8, Kirk could produce temperatures as low as  $40^{\circ}\text{C}$ . Paul Gifford in 1875 perfected the open type of machine.

# GAS CYCLE REFRIGERATION

This machine was further improved by T B Lightfoot, A Haslam, Henry Bell and James Coleman. This was the main method of marine refrigeration for quite some time. Frank Allen in New York developed a closed cycle machine employing high pressures to reduce the volume flow rates. This was named dense air machine. These days air cycle refrigeration is used only in aircrafts whose turbo compressor can handle large volume flow rates. Figure shows the schematic of an open type air cycle refrigeration system. The basic system shown here consists of a compressor, an expander and a heat exchanger. Air from the cold room is compressed in the compressor. The hot and high pressure air rejects heat to the heat sink (cooling water) in the heat exchanger. The warm but high pressure air expands in the expander. The cold air after expansion is sent to the cold room for providing cooling. The work of expansion partly compensates the work of compression; hence both the expander and the compressor are mounted on a common shaft.

# GAS CYCLE REFRIGERATION



**Schematic diagram of the cold air system**

*Schematic of a basic, open type air cycle refrigeration system*



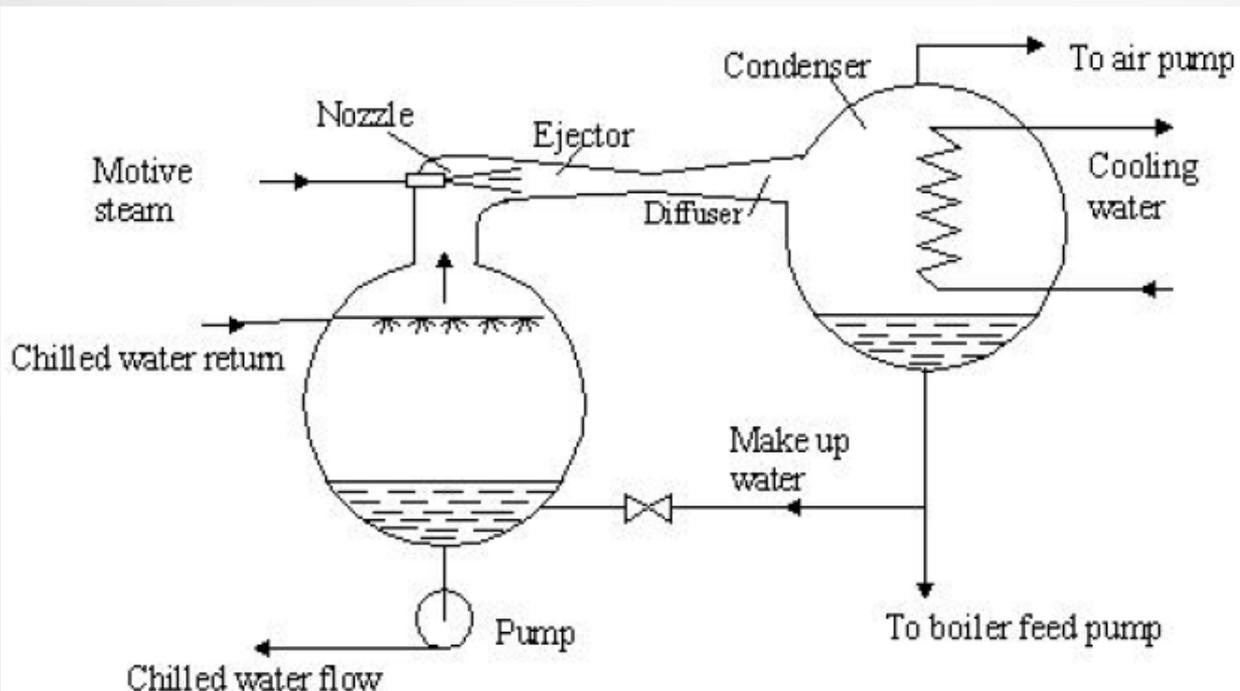
## STEAM JET REFRIGERATION SYSTEM

If water is sprayed into a chamber where a low pressure is maintained, a part of the water will evaporate. The enthalpy of evaporation will cool the remaining water to its saturation temperature at the pressure in the chamber. Obviously lower temperature will require lower pressure. Water freezes at  $0^{\circ}\text{C}$  hence temperature lower than  $4^{\circ}\text{C}$  cannot be obtained with water. In this system, high velocity steam is used to entrain the evaporating water vapour. High-pressure motive steam passes through either convergent or convergent-divergent nozzle where it acquires either sonic or supersonic velocity and low pressure of the order of  $0.009\text{ kPa}$  corresponding to an evaporator temperature of  $4^{\circ}\text{C}$ . The high momentum of motive steam entrains or carries along with it the water vapour evaporating from the flash chamber. Because of its high velocity it moves the vapours against the pressure gradient up to the condenser where the pressure is  $5.6\text{-}7.4\text{ kPa}$  corresponding to condenser temperature of  $35\text{-}45^{\circ}\text{C}$ .



# STEAM JET REFRIGERATION SYSTEM

The motive vapour and the evaporated vapour both are condensed and recycled. This system is known as steam jet refrigeration system. Figure shows a schematic of the system. It can be seen that this system requires a good vacuum to be maintained. Sometimes, booster ejector is used for this purpose. This system is driven by low- grade energy that is process steam in chemical plants or a boiler.



Steam jet refrigeration system



# THERMOELECTRIC REFRIGERATION SYSTEMS

In 1821 the German physicist T.J. Seebeck reported that when two junctions of dissimilar metals are kept at two different temperatures, an electro motive force (emf) is developed, resulting in flow of electric current. The emf produced is found to be proportional to temperature difference. In 1834, a Frenchmen, J. Peltier observed the reverse effect, i.e., cooling and heating of two junctions of dissimilar materials when direct current is passed through them, the heat transfer rate being proportional to the current. In 1838, H.F.E. Lenz froze a drop of water by the Peltier effect using antimony and bismuth (it was later found that Lenz could freeze water as the materials used were not pure metals but had some impurities in them). In 1857, William Thomson (Lord Kelvin) proved by thermodynamic analysis that Seebeck effect and Peltier effect are related and he discovered another effect called Thomson effect after his name. According to this when current flows through a conductor of a thermocouple that has an initial temperature gradient in it, then heat transfer rate per unit length is proportional to the product of current and the temperature



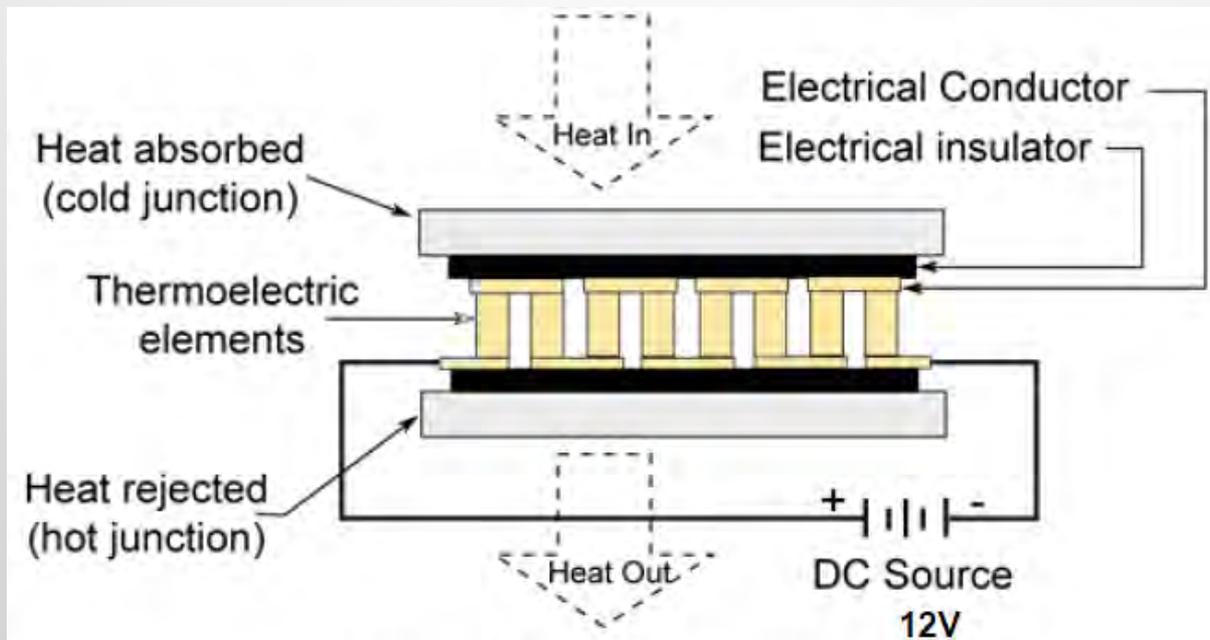
# THERMOELECTRIC REFRIGERATION SYSTEMS

As the current flow through thermoelectric material it gets heated due to its electrical resistance. This is called the Joulean effect, further, conduction heat transfer from the hot junction to the cold junction transfers heat. Both these heat transfer rates have to be compensated by the Peltier Effect for some useful cooling to be produced. For a long time, thermoelectric cooling based on the Peltier effect remained a laboratory curiosity as the temperature difference that could be obtained using pure metals was too small to be of any practical use. Insulating materials give poor thermoelectric performance because of their small electrical conductivity while metals are not good because of their large thermal conductivity. However, with the discovery of semiconductor materials in 1949-50, the available temperature drop could be increased considerably, giving rise to commercialization of thermoelectric refrigeration systems



# THERMOELECTRIC REFRIGERATION SYSTEMS

Figure shows the schematic of the thermoelectric refrigeration system based on semiconductor materials. The Russian scientist, A. F. Ioffe is one of the pioneers in the area of thermoelectric refrigeration systems using semiconductors. Several domestic refrigerators based on thermoelectric effect were made in USSR as early as 1949. However, since 1960s these systems are used mainly used for storing medicines, vaccines etc. and in electronic cooling. Development also took place in many other countries. In USA domestic refrigerators, air conditioners, water coolers, air conditioned diving suits etc. were made



*Schematic of a thermoelectric refrigeration system*



# THERMOELECTRIC REFRIGERATION SYSTEMS

using these effects. System capacities were typically small due to poor efficiency. However some large refrigeration capacity systems such as a 3000 kcal/h air conditioner and a 6 tonne capacity cold storage were also developed. By using multistaging temperatures as low as  $-145^{\circ}\text{C}$  were obtained. These systems due to their limited performance (limited by the materials) are now used only in certain niche applications such as electronic cooling, mobile coolers etc. Efforts have also been made to club thermoelectric systems with photovoltaic cells with a view to develop solar thermoelectric refrigerators.



# QUESTIONS

**Q1.** In an air cycle refrigeration system, low temperatures are produced due to:

- a) Evaporation of liquid air
- b) Throttling of air
- c) Expansion of air in turbine
- d) None of the above

**Q2.** Air cycle refrigeration systems are most commonly used in:

- a) Domestic refrigerators
- b) Aircraft air conditioning systems
- c) Cold storages
- d) Car air conditioning systems

**Q3.** The required input to the steam jet refrigeration systems is in the form of:

- a) Mechanical energy
- b) Thermal energy
- c) High pressure, motive steam
- d) Both mechanical and thermal energy

**Q4.** A nozzle is used in steam jet refrigeration systems to:

- a) To convert the high pressure motive steam into high velocity steam
- b) To reduce energy consumption
- c) To improve safety aspects
- d) All of the above

**Q5.** The materials used in thermoelectric refrigeration systems should have:

- a) High electrical and thermal conductivity
- b) High electrical conductivity and low thermal conductivity
- c) Low electrical conductivity and high thermal conductivity
- c) Low electrical and thermal conductivity

# QUESTIONS

**Q6.** A thermoelectric refrigeration systems requires:

- a) A high voltage AC (alternating current) input
- b) A low voltage AC input
- c) A high voltage DC (direct current) input
- d) A low voltage DC input

**Q7.** When you add sufficient amount of glucose to a glass of water, the water becomes cold. Is it an example of refrigeration, if it is, can this method be used for devising a refrigeration system?

**Q8.** Explain why ice making using nocturnal cooling is difficult on nights when the sky is cloudy?

**Q9.** To what do you attribute the rapid growth of refrigeration technology over the last century?



# INTRODUCTION

Air cycle refrigeration systems belong to the general class of gas cycle refrigeration systems, in which a gas is used as the working fluid. The gas does not undergo any phase change during the cycle, consequently, all the internal heat transfer processes are sensible heat transfer processes. Gas cycle refrigeration systems find applications in air craft cabin cooling and also in the liquefaction of various gases. In the present chapter gas cycle refrigeration systems based on air are discussed.



# AIR STANDARD CYCLE ANALYSIS

Air cycle refrigeration system analysis is considerably simplified if one makes the following assumptions:

- The working fluid is a fixed mass of air that behaves as an ideal gas
- The cycle is assumed to be a closed loop cycle with all inlet and exhaust processes of open loop cycles being replaced by heat transfer processes to or from the environment
- All the processes within the cycle are reversible, i.e., the cycle is internally reversible
- The specific heat of air remains constant throughout the cycle

An analysis with the above assumptions is called as cold Air Standard Cycle (ASC) analysis. This analysis yields reasonably accurate results for most of the cycles and processes encountered in air cycle refrigeration systems. However, the analysis fails when one considers a cycle consisting of a throttling process, as the temperature drop during throttling is zero for an ideal gas, whereas the actual cycles depend exclusively on the real gas behavior to produce refrigeration during throttling.



## BASIC CONCEPTS

The temperature of an ideal gas can be reduced either by making the gas to do work in an isentropic process or by sensible heat exchange with a cooler environment. When the gas does adiabatic work in a closed system by say, expanding against a piston, its internal energy drops. Since the internal energy of the ideal gas depends only on its temperature, the temperature of the gas also drops during the process, i.e.,

$$W = m(u_1 - u_2) = mc_v(T_1 - T_2)$$

where  $m$  is the mass of the gas,  $u_1$  and  $u_2$  are the initial and final internal energies of the gas,  $T_1$  and  $T_2$  are the initial and final temperatures and  $c_v$  is the specific heat at constant volume. If the expansion is reversible and adiabatic, by using the ideal gas equation  $Pv = RT$  and the equation for isentropic process  $P_1v_1^\gamma = P_2v_2^\gamma$  the final temperature ( $T_2$ ) is related to the initial temperature ( $T_1$ ) and initial and final pressures ( $P_1$  and  $P_2$ ) by the equation:

$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

where  $\gamma$  is the coefficient of isentropic expansion given by:

$$\gamma = \left( \frac{c_p}{c_v} \right)$$



## BASIC CONCEPTS

Isentropic expansion of the gas can also be carried out in a steady flow in a turbine which gives a net work output. Neglecting potential and kinetic energy changes, the work output of the turbine is given by:

$$W = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$

The final temperature is related to the initial temperature and initial and final pressures



# REVERSED CARNOT CYCLE EMPLOYING A GAS

Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure shows the schematic of a reversed Carnot refrigeration system using a gas as the working fluid along with the cycle diagram on T-s and P-v coordinates. As shown, the cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor

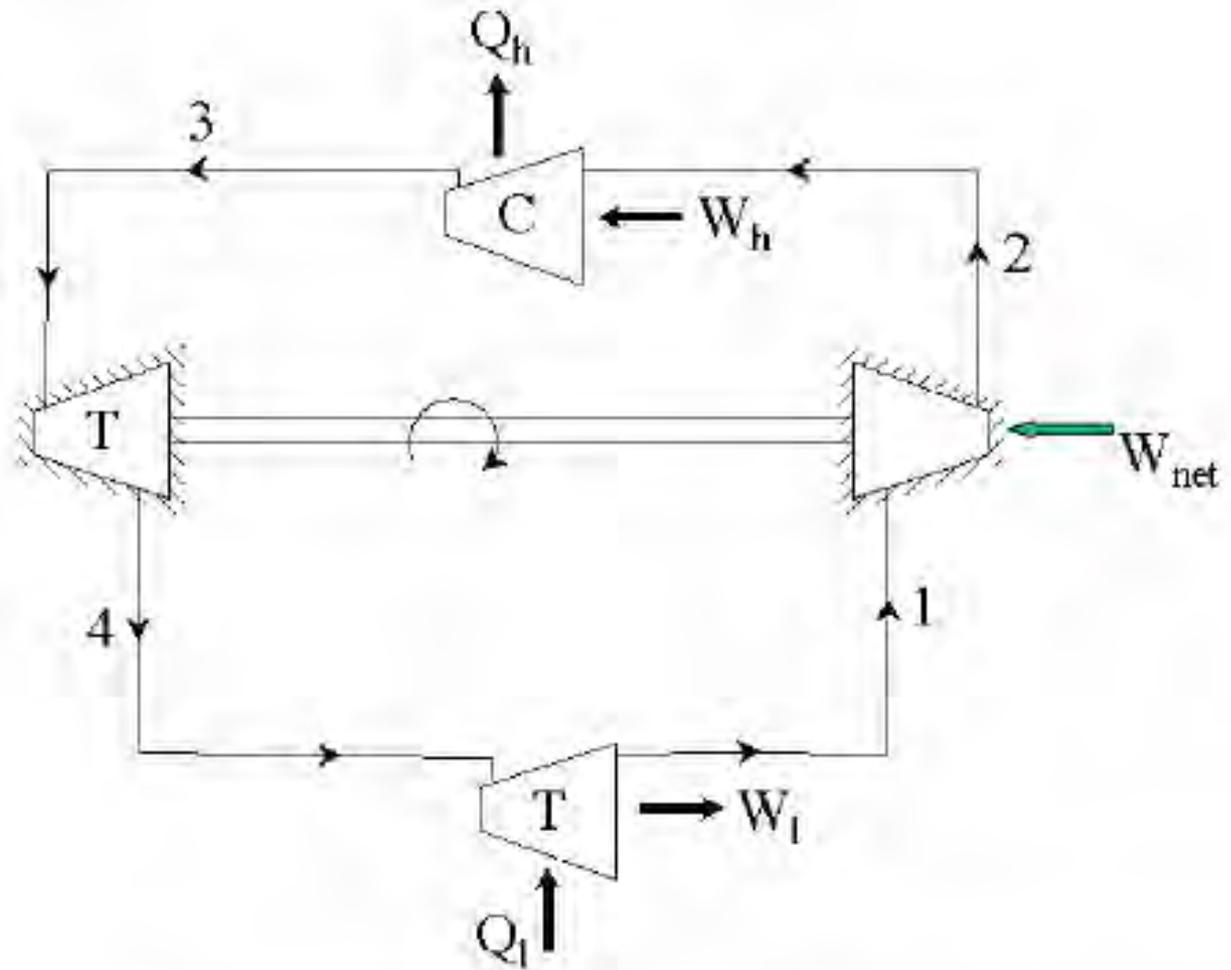
Process 2-3: Reversible, isothermal heat rejection in a compressor

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isothermal heat absorption in a turbine



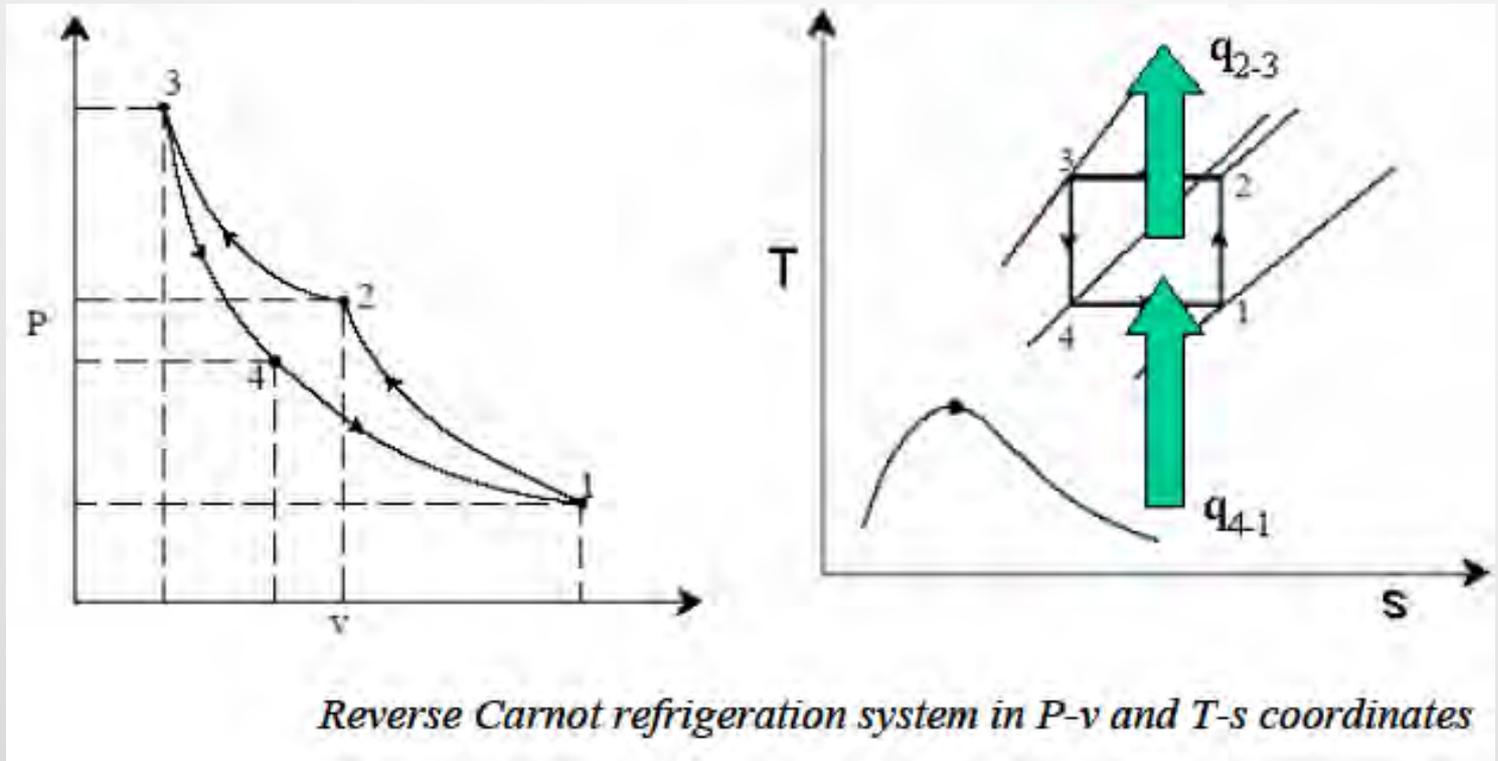
# REVERSED CARNOT CYCLE EMPLOYING A GAS



*Schematic of a reverse Carnot refrigeration system*



# REVERSED CARNOT CYCLE EMPLOYING A GAS



# REVERSED CARNOT CYCLE EMPLOYING A GAS

The heat transferred during isothermal processes 2-3 and 4-1 are given by:

$$q_{2-3} = \int_2^3 T \cdot ds = T_h (s_3 - s_2)$$

$$q_{4-1} = \int_4^1 T \cdot ds = T_l (s_1 - s_4)$$

$$s_1 = s_2 \quad \text{and} \quad s_3 = s_4, \quad \text{hence} \quad s_2 - s_3 = s_1 - s_4$$

Applying first law of thermodynamics to the closed cycle,

$$\oint \delta q = (q_{4-1} + q_{2-3}) = \oint \delta w = (w_{2-3} - w_{4-1}) = -W_{\text{net}}$$

# REVERSED CARNOT CYCLE EMPLOYING A GAS

the work of isentropic expansion,  $w_{3,4}$  exactly matches the work of isentropic compression  $w_{1,2}$ .

the COP of the Carnot system is given by:

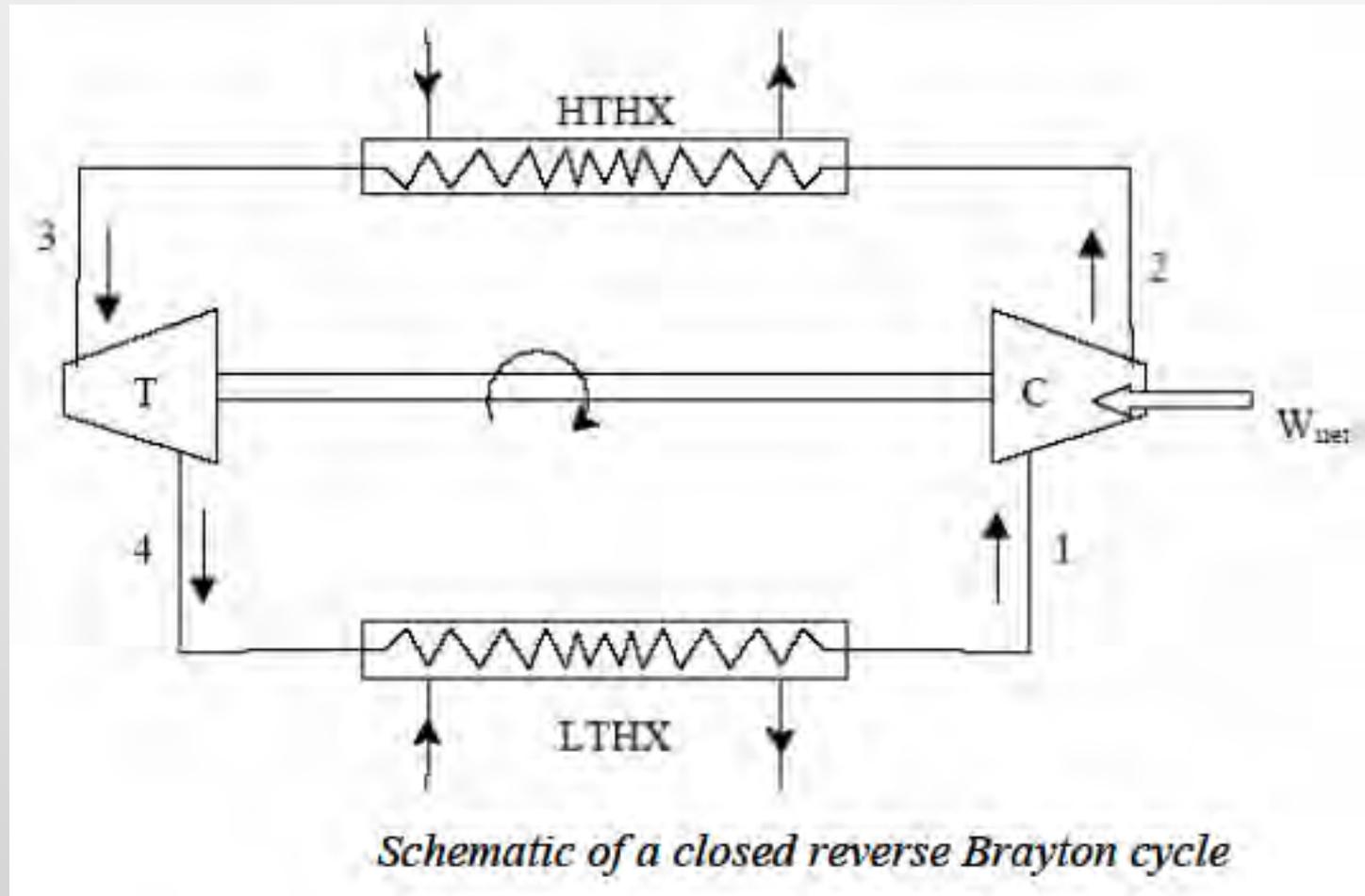
$$\text{COP}_{\text{Carnot}} = \frac{|q_{4-1}|}{|w_{\text{net}}|} = \left( \frac{T_1}{T_h - T_1} \right)$$

Thus the COP of the Carnot system depends only on the refrigeration ( $T_1$ ) and heat rejection ( $T_h$ ) temperatures only.

## Limitations of Carnot cycle:

Carnot cycle is an idealization and it suffers from several practical limitations. One of the main difficulties with Carnot cycle employing a gas is the difficulty of achieving isothermal heat transfer during processes 2-3 and 4-1. For a gas to have heat transfer isothermally, it is essential to carry out work transfer from or to the system when heat is transferred to the system (process 4-1) or from the system (process 2-3). This is difficult to achieve in practice. In addition, the volumetric refrigeration capacity of the Carnot system is very small leading to large compressor displacement, which gives rise to large frictional effects. All actual processes are irreversible, hence completely reversible cycles are idealizations only.

# IDEAL REVERSE BRAYTON CYCLE



# IDEAL REVERSE BRAYTON CYCLE

This is an important cycle frequently employed in gas cycle refrigeration systems. This may be thought of as a modification of reversed Carnot cycle, as the two isothermal processes of Carnot cycle are replaced by two isobaric heat transfer processes. This cycle is also called as Joule or Bell-Coleman cycle. Figure shows the schematic of a closed, reverse Brayton cycle and also the cycle on T-s diagram. As shown in the figure, the ideal cycle consists of the following four processes:

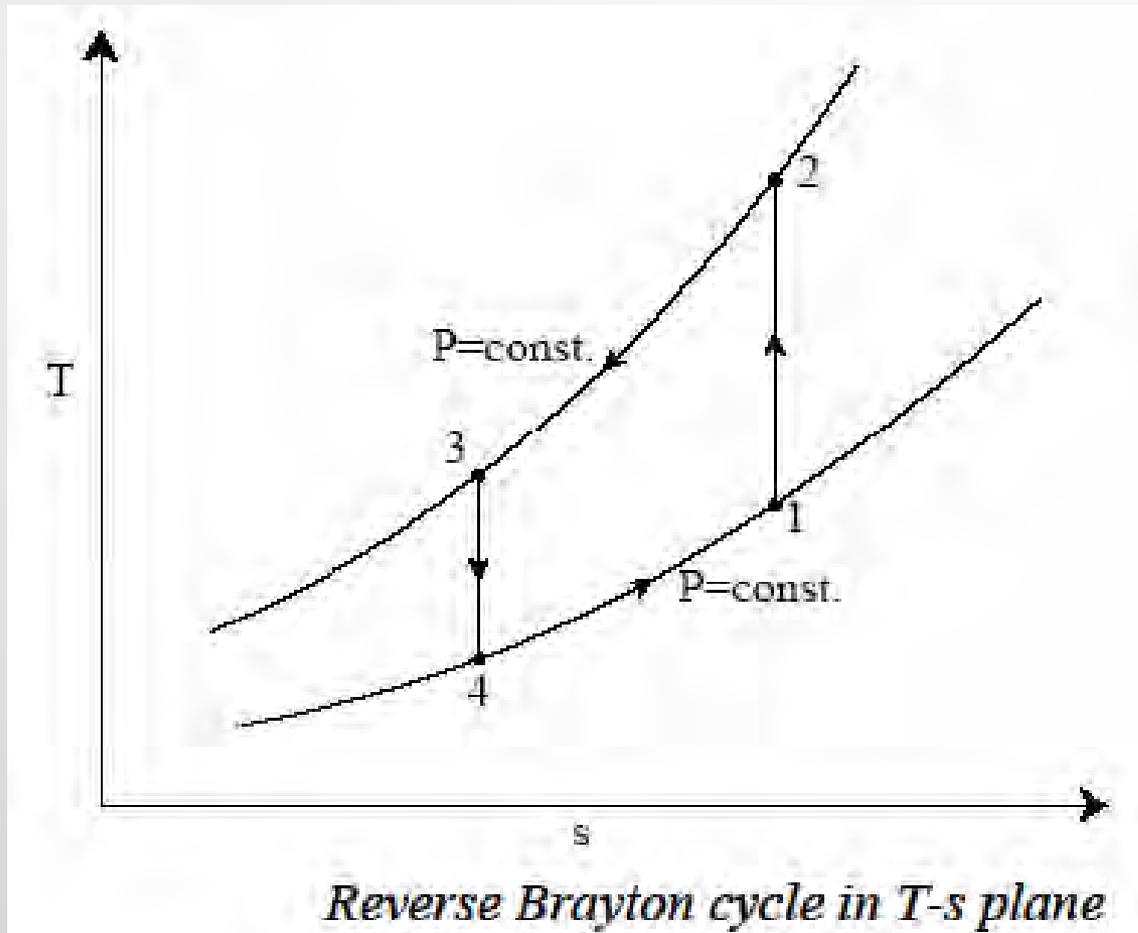
Process 1-2: Reversible, adiabatic compression in a compressor

Process 2-3: Reversible, isobaric heat rejection in a heat exchanger

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isobaric heat absorption in a heat exchanger

# IDEAL REVERSE BRAYTON CYCLE



# IDEAL REVERSE BRAYTON CYCLE

Process 1-2: Gas at low pressure is compressed isentropically from state 1 to state 2. Applying steady flow energy equation and neglecting changes in kinetic and potential energy, we can write:

$$W_{1-2} = \dot{m}(h_2 - h_1) = \dot{m} c_p (T_2 - T_1)$$

$$s_2 = s_1$$

$$\text{and } T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = T_1 r_p^{\frac{\gamma-1}{\gamma}}$$

where  $r_p = (P_2/P_1) =$  pressure ratio

Process 2-3: Hot and high pressure gas flows through a heat exchanger and rejects heat sensibly and isobarically to a heat sink. The enthalpy and temperature of the gas drop during the process due to heat exchange, no work transfer takes place and the entropy of the gas decreases. Again applying steady flow energy equation and second T ds equation:

$$Q_{2-3} = \dot{m}(h_2 - h_3) = \dot{m} c_p (T_2 - T_3)$$

$$s_2 - s_3 = c_p \ln \frac{T_2}{T_3}$$

$$P_2 = P_3$$

# IDEAL REVERSE BRAYTON CYCLE

Process 3-4: High pressure gas from the heat exchanger flows through a turbine, undergoes isentropic expansion and delivers net work output. The temperature of the gas drops during the process from  $T_3$  to  $T_4$ . From steady flow energy equation:

$$W_{3-4} = m(h_3 - h_4) = m c_p (T_3 - T_4)$$

$$s_3 = s_4$$

$$\text{and } T_3 = T_4 \left( \frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = T_4 r_p^{\frac{\gamma-1}{\gamma}}$$

where  $r_p = (P_3/P_4)$  = pressure ratio

Process 4-1: Cold and low pressure gas from turbine flows through the low temperature heat exchanger and extracts heat sensibly and isobarically from a heat source, providing a useful refrigeration effect. The enthalpy and temperature of the gas rise during the process due to heat exchange, no work transfer takes place and the entropy of the gas increases. Again applying steady flow energy equation and second T ds equation:

$$Q_{4-1} = m(h_1 - h_4) = m c_p (T_1 - T_4)$$

$$s_4 - s_1 = c_p \ln \frac{T_4}{T_1}$$

$$P_4 = P_1$$

# IDEAL REVERSE BRAYTON CYCLE

From the above equations, it can be easily shown that:

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{T_3}{T_4}\right)$$

Applying 1<sup>st</sup> law of thermodynamics to the entire cycle:

$$\oint \delta q = (q_{4-1} - q_{2-3}) = \oint \delta w = (w_{3-4} - w_{1-2}) = -w_{\text{net}}$$

The COP of the reverse Brayton cycle is given by:

$$\text{COP} = \left| \frac{q_{4-1}}{w_{\text{net}}} \right| = \left( \frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right)$$

using the relation between temperatures and pressures, the COP can also be written as:

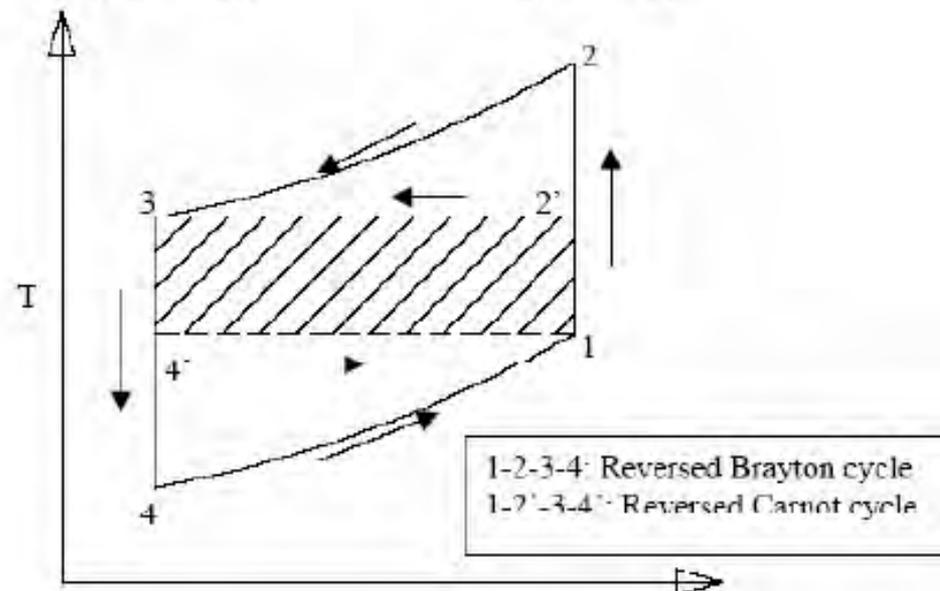
$$\text{COP} = \left( \frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right) = \left( \frac{T_4}{T_3 - T_4} \right) = \left( \frac{(T_1 - T_4)}{(T_1 - T_4)(r_p^{\frac{\gamma-1}{\gamma}} - 1)} \right) = (r_p^{\frac{\gamma-1}{\gamma}} - 1)^{-1}$$

From the above expression for COP, the following observations can be made:

# IDEAL REVERSE BRAYTON CYCLE

- a) For fixed heat rejection temperature ( $T_3$ ) and fixed refrigeration temperature ( $T_1$ ), the COP of reverse Brayton cycle is always lower than the COP of reverse Carnot cycle that is

$$\text{COP}_{\text{Brayton}} = \left( \frac{T_4}{T_3 - T_4} \right) < \text{COP}_{\text{Carnot}} = \left( \frac{T_1}{T_3 - T_1} \right)$$



*Comparison of reverse Carnot and reverse Brayton cycle in T-s plane*

- b) COP of Brayton cycle approaches COP of Carnot cycle as  $T_1$  approaches  $T_4$  (thin cycle), however, the specific refrigeration effect [ $c_p(T_1 - T_4)$ ] also reduces simultaneously.

# AIRCRAFT COOLING SYSTEMS

In an aircraft, cooling systems are required to keep the cabin temperatures at a comfortable level. Even though the outside temperatures are very low at high altitudes, still cooling of cabin is required due to:

- Large internal heat generation due to occupants, equipment etc.
- Heat generation due to skin friction caused by the fast moving aircraft
- At high altitudes, the outside pressure will be sub-atmospheric. When air at this low pressure is compressed and supplied to the cabin at pressures close to atmospheric, the temperature increases significantly. For example, when outside air at a pressure of 0.2 bar and temperature of 223 K (at 10000 m altitude) is compressed to 1 bar, its temperature increases to about 353 K. If the cabin is maintained at 0.8 bar, the temperature will be about 332 K. This effect is called as ram effect. This effect adds heat to the cabin, which needs to be taken out by the cooling system.
- Solar radiation

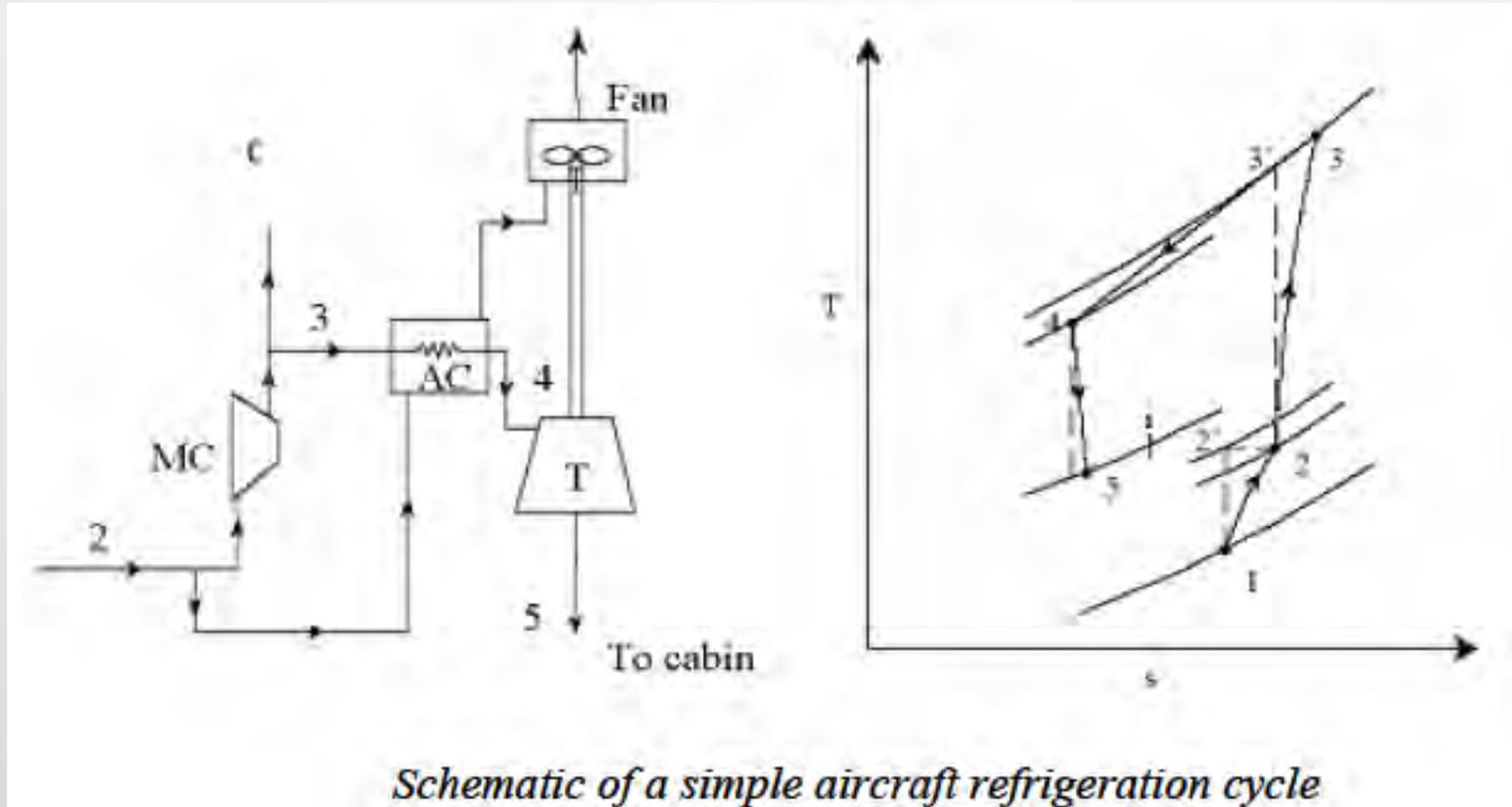
For low speed aircraft flying at low altitudes, cooling system may not be required, however, for high speed aircraft flying at high altitudes, a cooling system is a must.

# AIRCRAFT COOLING SYSTEMS

Even though the COP of air cycle refrigeration is very low compared to vapour compression refrigeration systems, it is still found to be most suitable for aircraft refrigeration systems as:

- Air is cheap, safe, non-toxic and non-flammable. Leakage of air is not a problem
- Cold air can directly be used for cooling thus eliminating the low temperature heat exchanger (open systems) leading to lower weight
- The aircraft engine already consists of a high speed turbo-compressor, hence separate compressor for cooling system is not required. This reduces the weight per kW cooling considerably. Typically, less than 50% of an equivalent vapour compression system
- Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

# SIMPLE AIRCRAFT REFRIGERATION CYCLE



# SIMPLE AIRCRAFT REFRIGERATION CYCLE

Figure shows the schematic of a simple aircraft refrigeration system and the operating cycle on T-s diagram. This is an open system. As shown in the T-s diagram, the outside low pressure and low temperature air (state 1) is compressed due to ram effect to ram pressure (state 2). During this process its temperature increases from 1 to 2. This air is compressed in the main compressor to state 3, and is cooled to state 4 in the air cooler. Its pressure is reduced to cabin pressure in the turbine (state 5), as a result its temperature drops from 4 to 5. The cold air at state 5 is supplied to the cabin. It picks up heat as it flows through the cabin providing useful cooling effect. The power output of the turbine is used to drive the fan, which maintains the required air flow over the air cooler. This simple system is good for ground cooling (when the aircraft is not moving) as fan can continue to maintain airflow over the air cooler.

# SIMPLE AIRCRAFT REFRIGERATION CYCLE

By applying steady flow energy equation to the ramming process, the temperature rise at the end of the ram effect can be shown to be:

$$\frac{T_{2'}}{T_1} = 1 + \frac{\gamma - 1}{2} M^2$$

where  $M$  is the Mach number, which is the ratio of velocity of the aircraft ( $C$ ) to the sonic velocity  $a$  ( $a = \sqrt{\gamma RT_1}$ ), i.e.,

$$M = \frac{C}{a} = \frac{C}{\sqrt{\gamma RT_1}}$$

Due to irreversibilities, the actual pressure at the end of ramming will be less than the pressure resulting from isentropic compression. The ratio of actual pressure rise to the isentropic pressure rise is called as ram efficiency,  $\eta_{\text{Ram}}$ , i.e.,

# SIMPLE AIRCRAFT REFRIGERATION CYCLE

$$\eta_{\text{Ram}} = \frac{(P_2 - P_1)}{(P_{2'} - P_1)}$$

The refrigeration capacity of the simple aircraft cycle discussed,  $\dot{Q}$  is given by:

$$\dot{Q} = \dot{m} c_p (T_i - T_5)$$

where  $\dot{m}$  is the mass flow rate of air through the turbine.

## BOOTSTRAP SYSTEM

Figure shows the schematic of a bootstrap system, which is a modification of the simple system. As shown in the figure, this system consists of two heat exchangers (air cooler and aftercooler), in stead of one air cooler of the simple system. It also incorporates a secondary compressor, which is driven by the turbine of the cooling system. This system is suitable for high speed aircraft, where in the velocity of the aircraft provides the necessary airflow for the heat exchangers, as a result a separate fan is not required. As shown in the cycle diagram, ambient air state 1 is pressurized to state 2 due to the ram effect. This air is further compressed to state 3 in the main compressor. The air is then cooled to state 4 in the air cooler. The heat rejected in the air cooler is absorbed by the ram air at state 2.

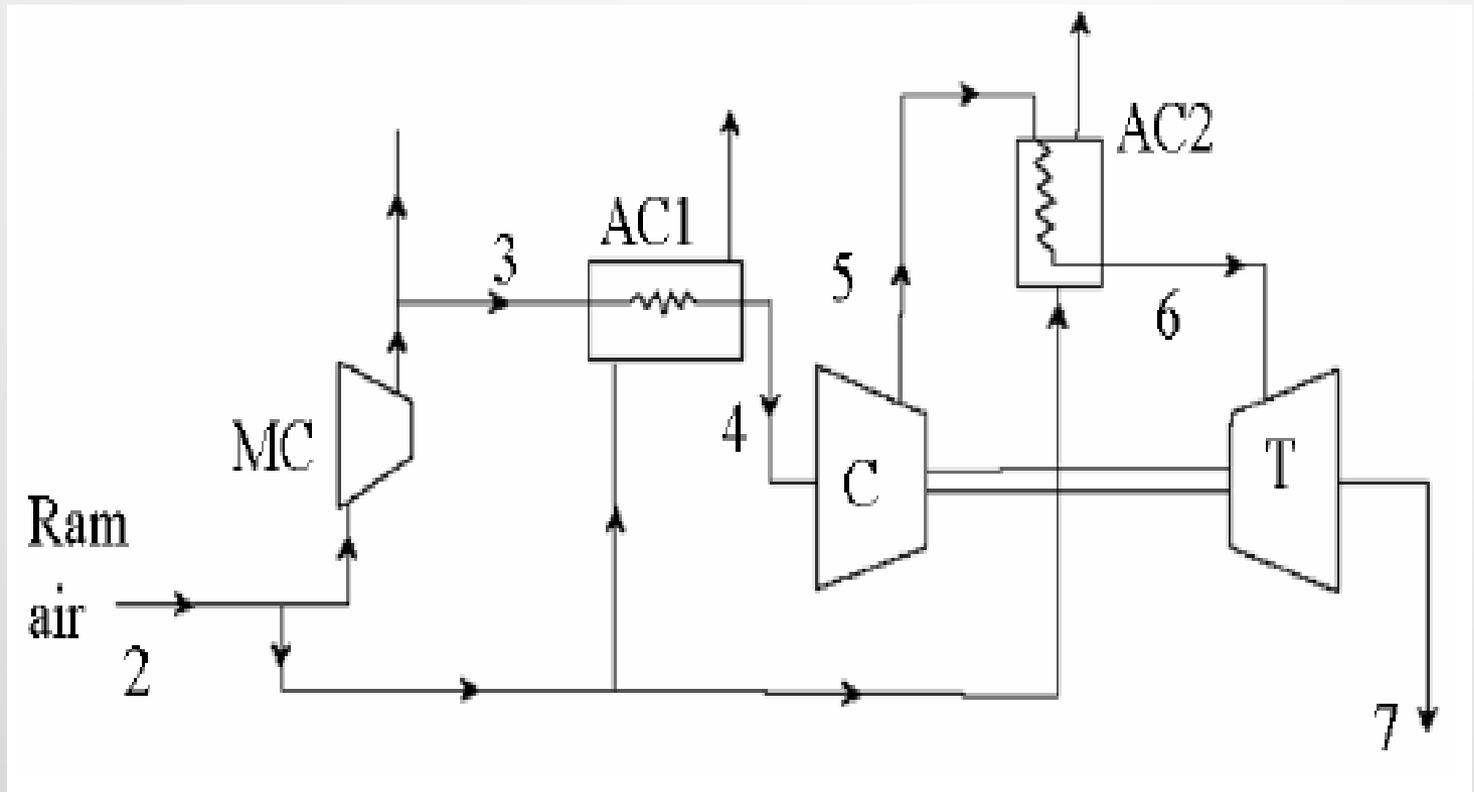
## BOOTSTRAP SYSTEM

The air from the air cooler is further compressed from state 4 to state 5 in the secondary compressor. It is then cooled to state 6 in the after cooler, expanded to cabin pressure in the cooling turbine and is supplied to the cabin at a low temperature  $T_7$ . Since the system does not consist of a separate fan for driving the air through the heat exchangers, it is not suitable for ground cooling. However, in general ground cooling is normally done by an external air conditioning system as it is not efficient to run the aircraft engine just to provide cooling when it is grounded.

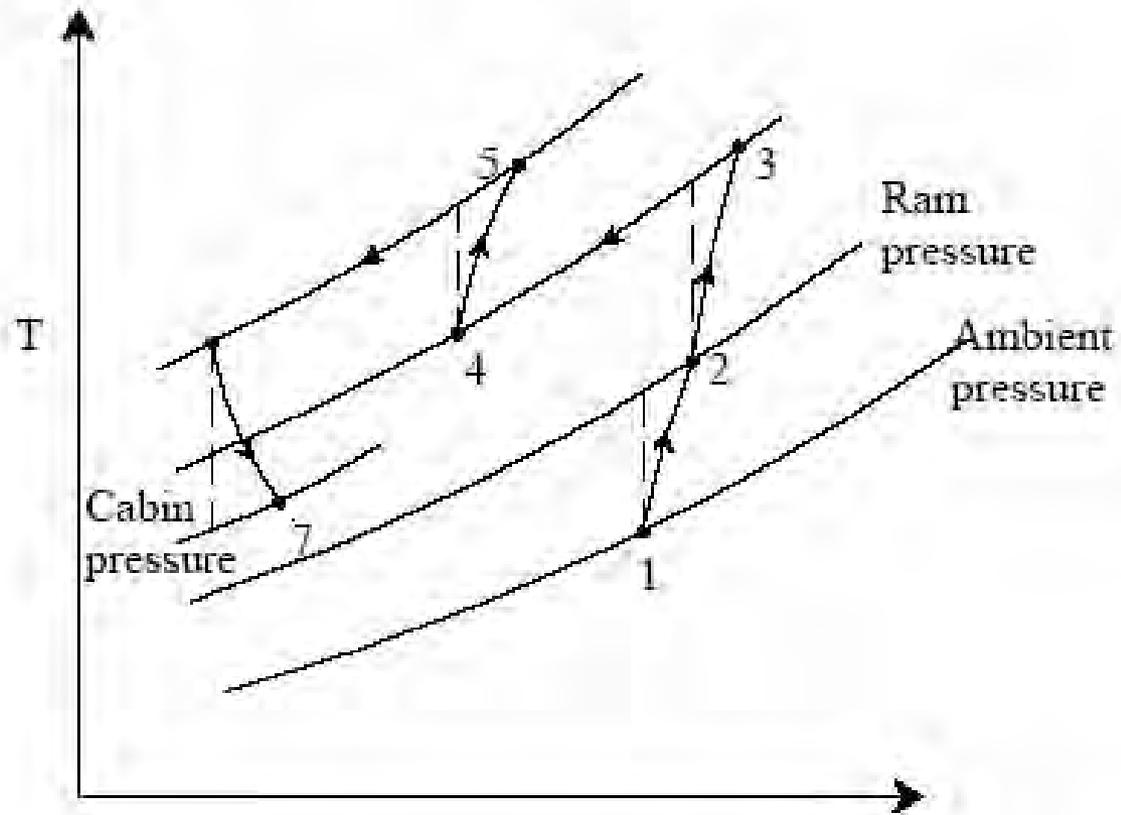
## BOOTSTRAP SYSTEM

Other modifications over the simple system are: regenerative system and reduced ambient system. In a regenerative system, a part of the cold air from the cooling turbine is used for precooling the air entering the turbine. As a result much lower temperatures are obtained at the exit of the cooling turbine, however, this is at the expense of additional weight and design complexity. The cooling turbine drives a fan similar to the simple system. The regenerative system is good for both ground cooling as well as high speed aircrafts. The reduced ambient system is well-suited for supersonic aircrafts and rockets.

# BOOTSTRAP SYSTEM



# BOOTSTRAP SYSTEM



*Schematic of a bootstrap system*

## DRY AIR RATED TEMPERATURE (DART)

The concept of Dry Air Rated Temperature is used to compare different aircraft refrigeration cycles. Dry Air Rated Temperature is defined as the temperature of the air at the exit of the cooling turbine in the absence of moisture condensation. For condensation not to occur during expansion in turbine, the dew point temperature and hence moisture content of the air should be very low, i.e., the air should be very dry. The aircraft refrigeration systems are rated based on the mass flow rate of air at the design DART. The cooling capacity is then given by:

$$\dot{Q} = \dot{m} c_p (T_i - T_{\text{DART}})$$

# DRY AIR RATED TEMPERATURE (DART)

where  $\dot{m}$  is the mass flow rate of air,  $T_{DART}$  and  $T_i$  are the dry air rated temperature and cabin temperature, respectively.

A comparison between different aircraft refrigeration systems based on DART at different Mach numbers shows that:

- DART increases monotonically with Mach number for all the systems except the reduced ambient system
- The simple system is adequate at low Mach numbers
- At high Mach numbers either bootstrap system or regenerative system should be used
- Reduced ambient temperature system is best suited for very high Mach number, supersonic aircrafts

# QUESTIONS

1. A refrigerator working on Bell-Coleman cycle (Reverse brayton cycle) operates between 1 bar and 10 bar. Air is drawn from cold chamber at  $-10^{\circ}\text{C}$ . Air coming out of compressor is cooled to  $50^{\circ}\text{C}$  before entering the expansion cylinder. Polytropic law  $P.V^{1.3} = \text{constant}$  is followed during expansion and compression. Find theoretical C.O.P of the origin. Take  $\gamma = 1.4$  and  $C_p = 1.00 \text{ kJ/kg } ^{\circ}\text{C}$  for air.

2. An air refrigerator working on the principle of Bell-Coleman cycle. The air into the compressor is at 1 atm at  $-10^{\circ}\text{C}$ . It is compressed to 10 atm and cooled to  $40^{\circ}\text{C}$  at the same pressure. It is then expanded to 1 atm and discharged to take cooling load. The air circulation is 1 kg/s.

The isentropic efficiency of the compressor = 80%

The isentropic efficiency of the expander = 90%

Find the following:

- i) Refrigeration capacity of the system
- ii) C.O.P of the system

Take  $\gamma = 1.4$ ,  $C_p = 1.00 \text{ kJ/kg } ^{\circ}\text{C}$

3. A Carnot refrigerator extracts 150 kJ of heat per minute from a space which is maintained at  $-20^{\circ}\text{C}$  and is discharged to atmosphere at  $45^{\circ}\text{C}$ . Find the work required to run the unit.

# QUESTIONS

4. A boot strap cooling system of 10 tons is used in an aeroplane. The temperature and pressure conditions of atmosphere are  $20^{\circ}\text{C}$  and  $0.9\text{ atm}$ . The pressure of air is increased from  $0.9\text{ atm}$  to  $1.1\text{ atm}$  due to ramming. The pressures of air leaving the main and auxiliary compressor are  $3\text{ atm}$  and  $4\text{ atm}$  respectively. Isentropic efficiency of compressors and turbine are  $0.85$  and  $0.8$  respectively.  $50\%$  of the total heat of air leaving the main compressor is removed in the first heat exchanger and  $30\%$  of their total heat of air leaving the auxiliary compressor is removed in the second heat exchanger using removed air. Find:
- Power required to take cabin load
  - COP of the system

The cabin pressure is  $1.02\text{ atm}$  and temperature of air leaving the cabin should be greater than  $25^{\circ}\text{C}$ . Assume ramming action to be isentropic.

5. A simple air cooled system is used for an aeroplane to take a load of 10 tons. Atmospheric temperature and pressure is  $25^{\circ}\text{C}$  and  $0.9\text{ atm}$  respectively. Due to ramming the pressure of air is increased from  $0.9\text{ atm}$ , to  $1\text{ atm}$ . The pressure of air leaving the main compressor is  $3.5\text{ atm}$  and its  $50\%$  heat is removed in the air-cooled heat exchanger and then it is passed through a evaporator for future cooling. The temperature of air is reduced by  $10^{\circ}\text{C}$  in the evaporator. Lastly the air is passed through cooling turbine and is supplied to the cooling cabin where the pressure is  $1.03\text{ atm}$ . Assuming isentropic efficiency of the compressor and turbine are  $75\%$  and  $70\%$ , find
- Power required to take the load in the cooling cabin
  - COP of the system.

The temperature of air leaving the cabin should not exceed  $25^{\circ}\text{C}$ .

## COMPARISON BETWEEN GAS CYCLES AND VAPOR CYCLES

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. As mentioned in the previous chapter, in a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

# VAPOUR COMPRESSION REFRIGERATION SYSTEMS

As mentioned, vapour compression refrigeration systems are the most commonly used among all refrigeration systems. As the name implies, these systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems.

# VAPOUR COMPRESSION REFRIGERATION SYSTEMS

Vapour compression refrigeration systems are available to suit almost all applications with the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle. Before the actual cycle is discussed and analysed, it is essential to find the upper limit of performance of vapour compression cycles. This limit is set by a completely reversible cycle.

# THE CARNOT REFRIGERATION CYCLE

Carnot refrigeration cycle is a completely reversible cycle, hence is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. It is used as reference against which the real cycles are compared. Figures (a) and (b) show the schematic of a Carnot vapour compression refrigeration system and the operating cycle on T-s diagram.

As shown in Fig.(a), the basic Carnot refrigeration system for pure vapour consists of four components: compressor, condenser, turbine and evaporator. Refrigeration effect ( $q_{4-1} = q_e$ ) is obtained at the evaporator as the refrigerant undergoes the process of vaporization (process 4-1) and extracts the latent heat from the low temperature heat source. The low temperature, low pressure vapour is then compressed isentropically in the compressor to the heat sink temperature  $T_c$ . The refrigerant pressure increases from  $P_e$  to  $P_c$  during the compression process (process 1-2) and the exit vapour is saturated. Next the high pressure, high temperature saturated refrigerant undergoes the process of condensation in the condenser (process 2-3) as it rejects the heat of condensation ( $q_{2-3} = q_c$ ) to an external heat sink at  $T_c$ . The high pressure saturated liquid then flows through the turbine and undergoes isentropic expansion (process 3-4).

# THE CARNOT REFRIGERATION CYCLE

During this process, the pressure and temperature fall from  $P_c, T_c$  to  $P_e, T_e$ . Since a saturated liquid is expanded in the turbine, some amount of liquid flashes into vapour and the exit condition lies in the two-phase region. This low temperature and low pressure liquid-vapour mixture then enters the evaporator completing the cycle. Thus as shown in Fig.(b), the cycle involves two isothermal heat transfer processes (processes 4-1 and 2-3) and two isentropic work transfer processes (processes 1-2 and 3-4). Heat is extracted isothermally at evaporator temperature  $T_e$  during process 4-1, heat is rejected isothermally at condenser temperature  $T_c$  during process 2-3. Work is supplied to the compressor during the isentropic compression (1-2) of *refrigerant vapour* from evaporator pressure  $P_e$  to condenser pressure  $P_c$ , and work is produced by the system as *refrigerant liquid* expands isentropically in the turbine from condenser pressure  $P_c$  to evaporator pressure  $P_e$ . All the processes are both internally as well as externally reversible, i.e., net entropy generation for the system and environment is zero.

# THE CARNOT REFRIGERATION CYCLE

Applying first and second laws of thermodynamics to the Carnot refrigeration cycle,

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = -w_{\text{net}}$$

$$\Rightarrow (q_c - q_e) = w_{\text{net}}$$

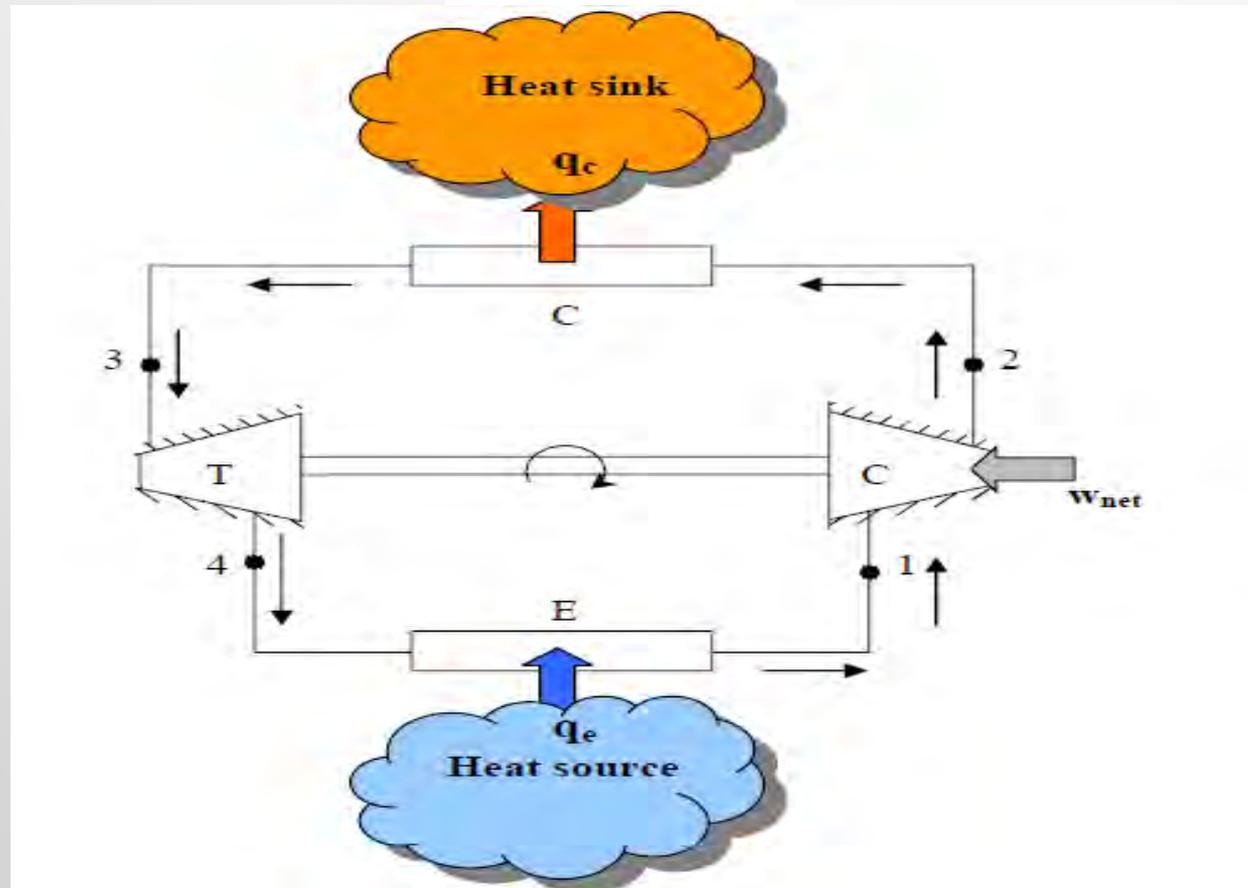


Fig. (a)



# THE CARNOT REFRIGERATION CYCLE

now for the reversible, isothermal heat transfer processes 2-3 and 4-1, we can write:

$$q_c = -q_{2-3} = -\int_2^3 T \cdot ds = T_c (s_2 - s_3)$$

$$q_e = q_{4-1} = \int_4^1 T \cdot ds = T_e (s_1 - s_4)$$

where  $T_e$  and  $T_c$  are the evaporator and condenser temperatures, respectively, and,

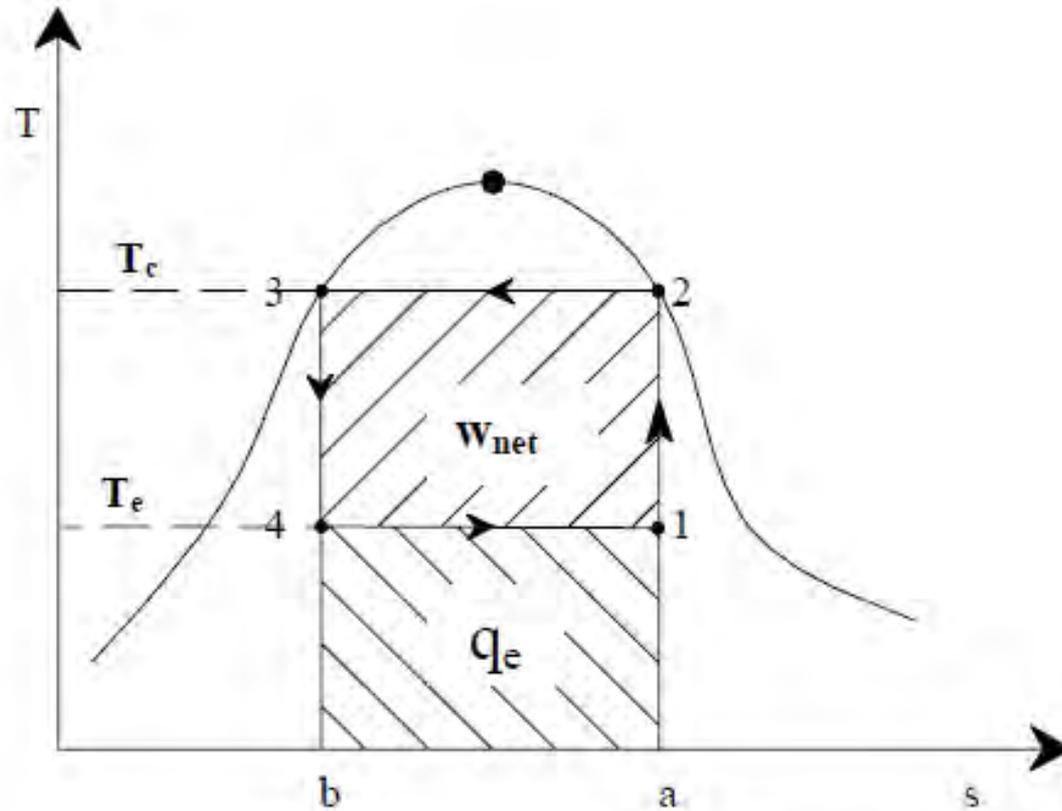
$$s_1 = s_2 \text{ and } s_3 = s_4$$

the Coefficient of Performance (COP) is given by:

$$\text{COP}_{\text{Carnot}} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{W_{\text{net}}} = \frac{T_e (s_1 - s_4)}{T_c (s_2 - s_3) - T_e (s_1 - s_4)} = \left( \frac{T_e}{T_c - T_e} \right)$$

thus the COP of Carnot refrigeration cycle is a function of evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle (Lesson 9). The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP.

# THE CARNOT REFRIGERATION CYCLE

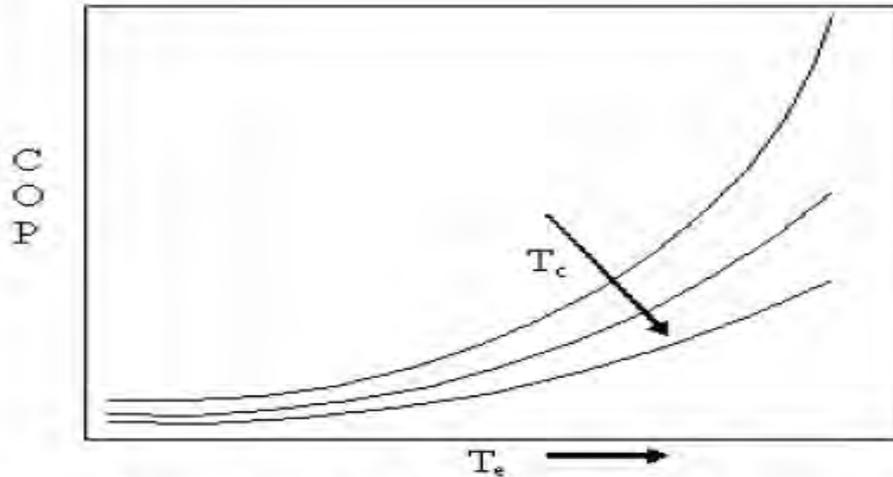


*Carnot refrigeration cycle represented in T-s plane*

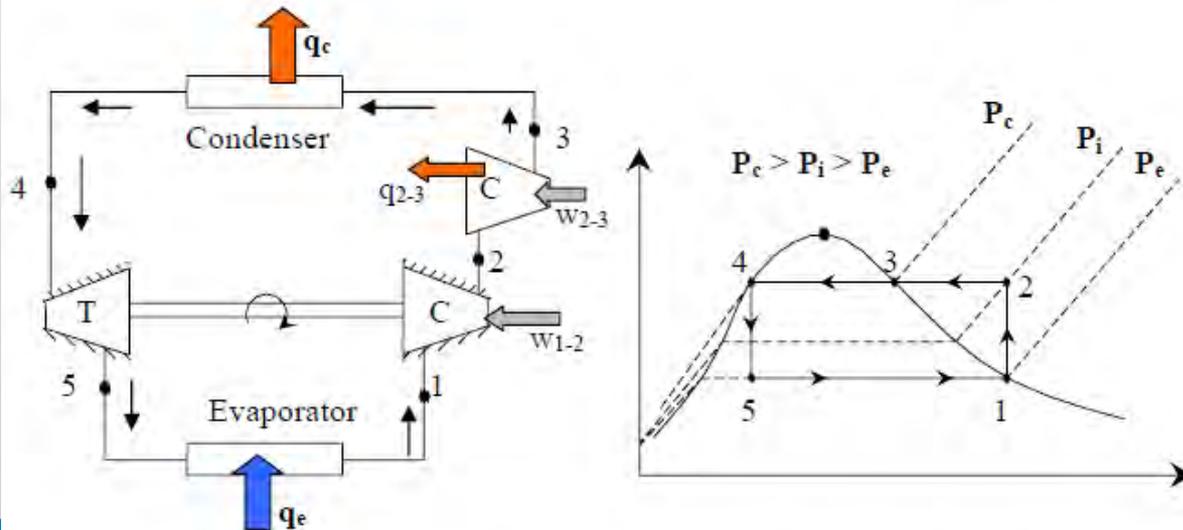
# THE CARNOT REFRIGERATION CYCLE

It can be seen from the above expression that the COP of a Carnot refrigeration system increases as the evaporator temperature increases and condenser temperature decreases. This can be explained very easily with the help of the T-s diagram. As shown in the figure, COP is the ratio of area a-1-4-b to the area 1-2-3-4. For a fixed condenser temperature  $T^c$ , as the evaporator temperature  $T^e$  increases, area a-1-4-b ( $q_e$ ) increases and area 1-2-3-4 ( $w_{net}$ ) decreases as a result, COP increases rapidly. Similarly for a fixed evaporator temperature  $T_e$ , as the condensing temperature  $T_c$  increases, the net work input (area 1-2-3-4) increases, even though cooling output remains constant, as a result the COP falls. Figure shows the variation of Carnot COP with evaporator temperature for different condenser temperatures. It can be seen that the COP increases sharply with evaporator temperatures, particularly at high condensing temperatures. COP reduces as the condenser temperature increases, but the effect becomes marginal at low evaporator temperatures. It will be shown later that actual vapour compression refrigeration systems also behave in a manner similar to that of Carnot refrigeration systems as far as the performance trends are concerned.

# THE CARNOT REFRIGERATION CYCLE

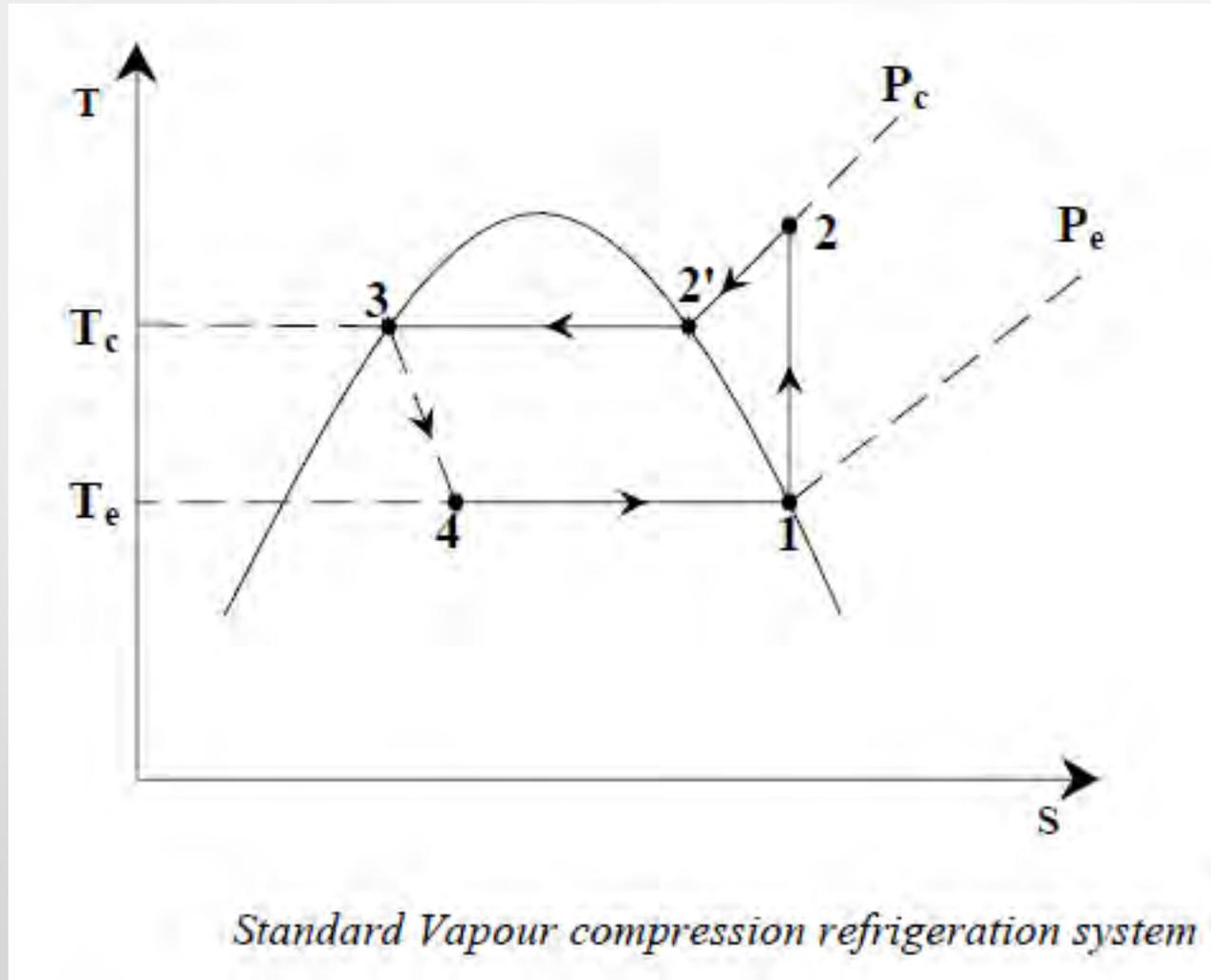


*Effects of evaporator and condenser temperatures on Carnot COP*



*Carnot refrigeration system with dry compression*

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

Figure shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a  $T-s$  diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor

Process 2-3: Isobaric heat rejection in condenser

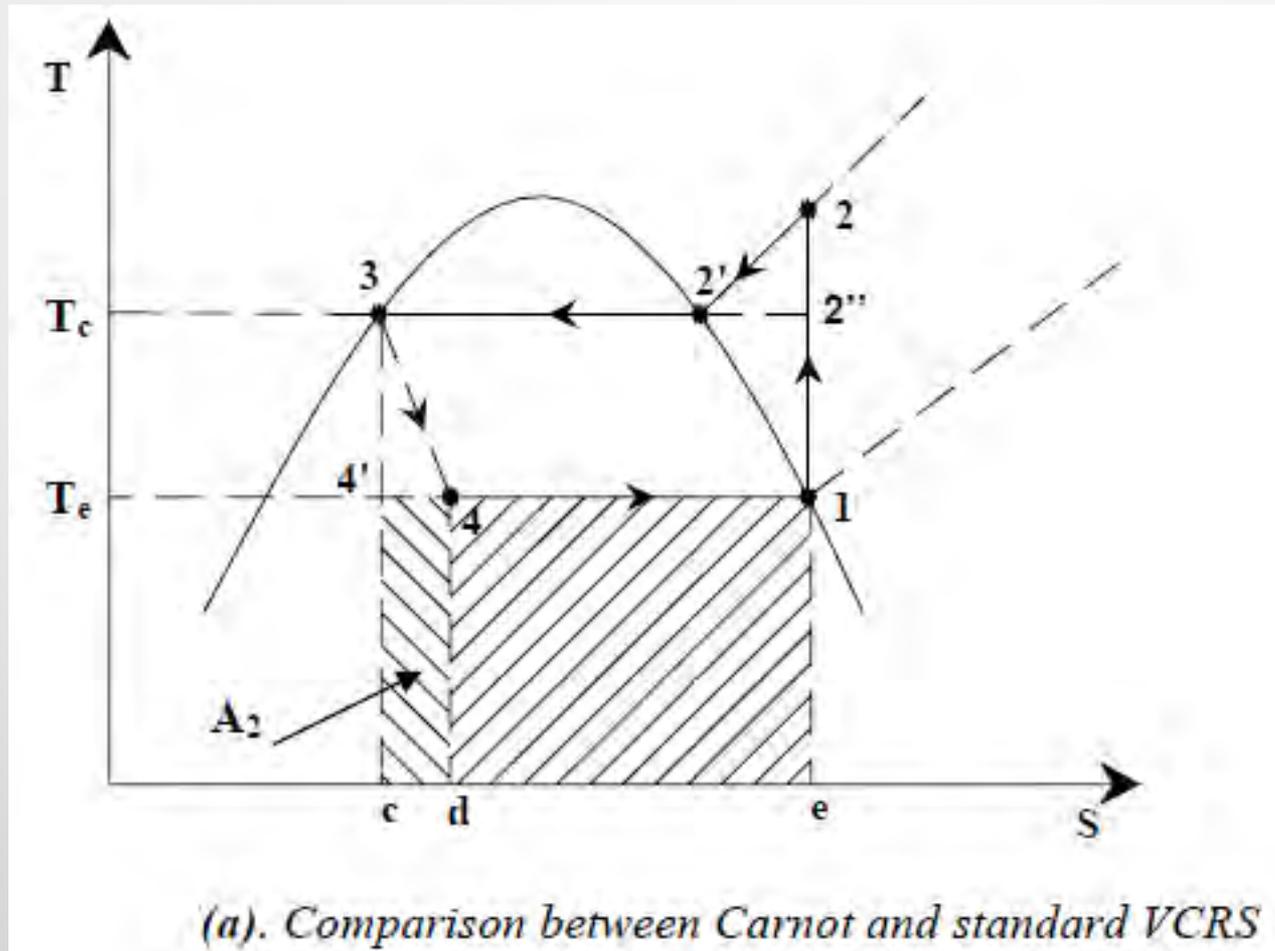
Process 3-4: Isenthalpic expansion of saturated liquid in expansion device

Process 4-1: Isobaric heat extraction in the evaporator

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non-isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3-4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the cycle diagrams on T s charts. Figure (a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

The heat extraction (evaporation) process is reversible for both the Carnot cycle and VCRS cycle. Hence the refrigeration effect is given by:

For Carnot refrigeration cycle (1-2''-3-4'):

$$q_{e,Carnot} = q_{4'-1} = \int_{4'}^1 T \cdot ds = T_e (s_1 - s_{4'}) = \text{area } e-1-4'-c-e$$

For VCRS cycle (1-2-3-4):

$$q_{e,VCRS} = q_{4-1} = \int_4^1 T \cdot ds = T_e (s_1 - s_4) = \text{area } e-1-4-d-e$$

thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRS cycle, this reduction is equal to the area d-4-4'-c-d (area  $A_2$ ) and is known as *throttling loss*. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e.,

$$q_{e,Carnot} - q_{VCRS} = \text{area } d-4-4'-c-d = (h_3 - h_{4'}) = (h_4 - h_{4'}) = \text{area } A_2$$

It is easy to show that the loss in refrigeration effect increases as the evaporator temperature decreases and/or condenser temperature increases. A practical consequence of this is a requirement of higher refrigerant mass flow rate.

The heat rejection in case of VCRS cycle also increases when compared to Carnot cycle.



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

As shown in Fig. (b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

$$q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^3 T.ds = T_c (s_{2''} - s_3) = \text{area } e-2''-3-c-e$$

In case of VCRS cycle, the heat rejection rate is given by:

$$q_{c,VCRS} = -q_{2-3} = -\int_2^3 T.ds = \text{area } e-2-3-c-e$$

Hence the increase in heat rejection rate of VCRS compared to Carnot cycle is equal to the area 2''-2-2' (area  $A_1$ ). This region is known as *superheat horn*, and is due to the replacement of isothermal heat rejection process of Carnot cycle by isobaric heat rejection in case of VCRS.

Since the heat rejection increases and refrigeration effect reduces when the Carnot cycle is modified to standard VCRS cycle, the net work input to the VCRS increases compared to Carnot cycle. The net work input in case of Carnot and VCRS cycles are given by:

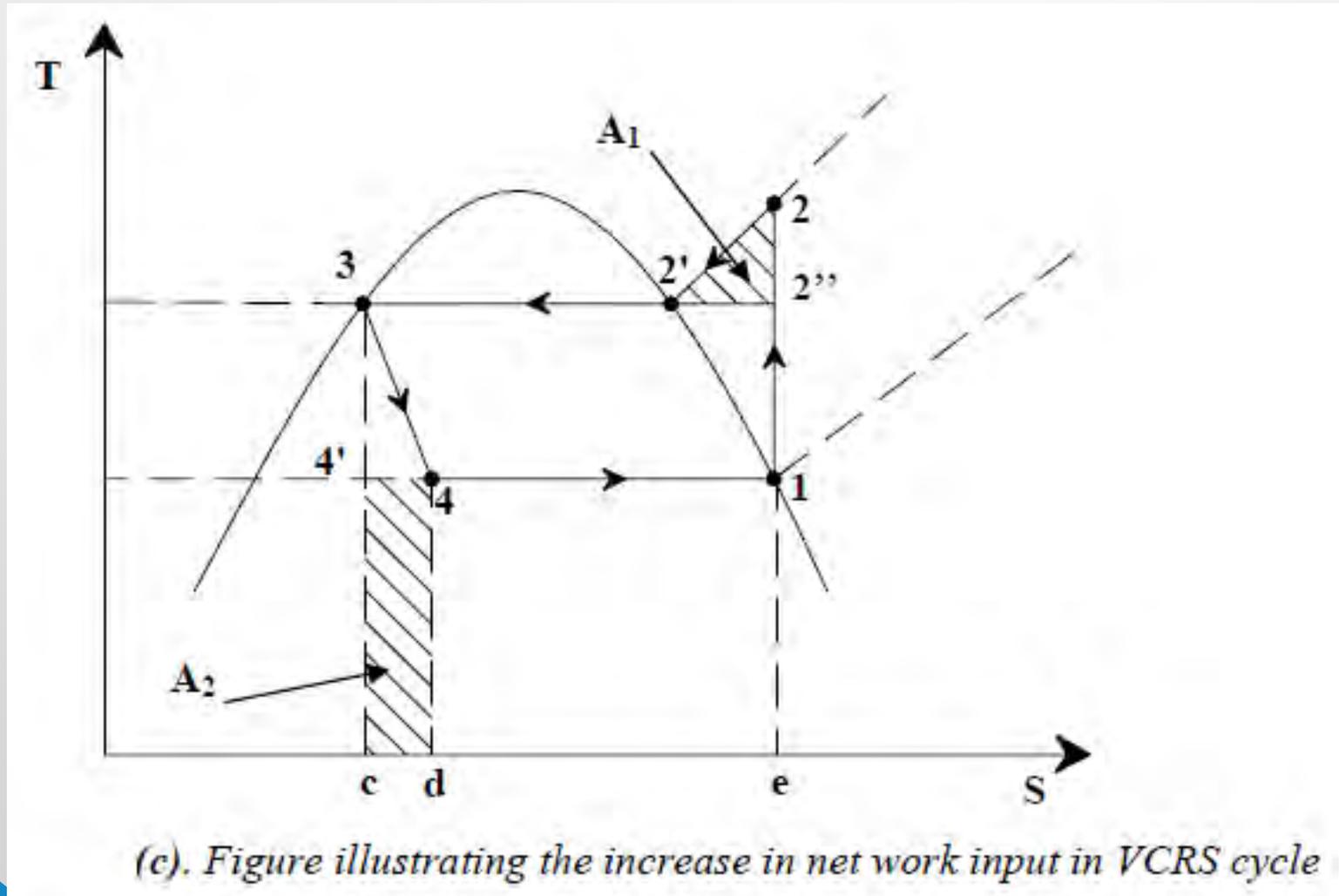
$$W_{net,Carnot} = (q_c - q_e)_{Carnot} = \text{area } 1-2''-3-4'-1$$

$$W_{net,VCRS} = (q_c - q_e)_{VCRS} = \text{area } 1-2-3-4'-c-d-4-1$$

As shown in Fig (c), the increase in net work input in VCRS cycle is given by:

$$W_{net,VCRS} - W_{net,Carnot} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2$$

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

To summarize the refrigeration effect and net work input of VCRS cycle are given by:

$$q_{e,VCRS} = q_{e,Carnot} - \text{area } A_2$$

$$W_{net,VCRS} = W_{net,Carnot} + \text{area } A_1 + \text{area } A_2$$

The COP of VCRS cycle is given by:

$$COP_{VCRS} = \frac{q_{e,VCRS}}{W_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{W_{net,Carnot} + \text{area } A_1 + \text{area } A_2}$$

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

If we define the cycle efficiency,  $\eta_R$  as the ratio of COP of VCRS cycle to the COP of Carnot cycle, then:

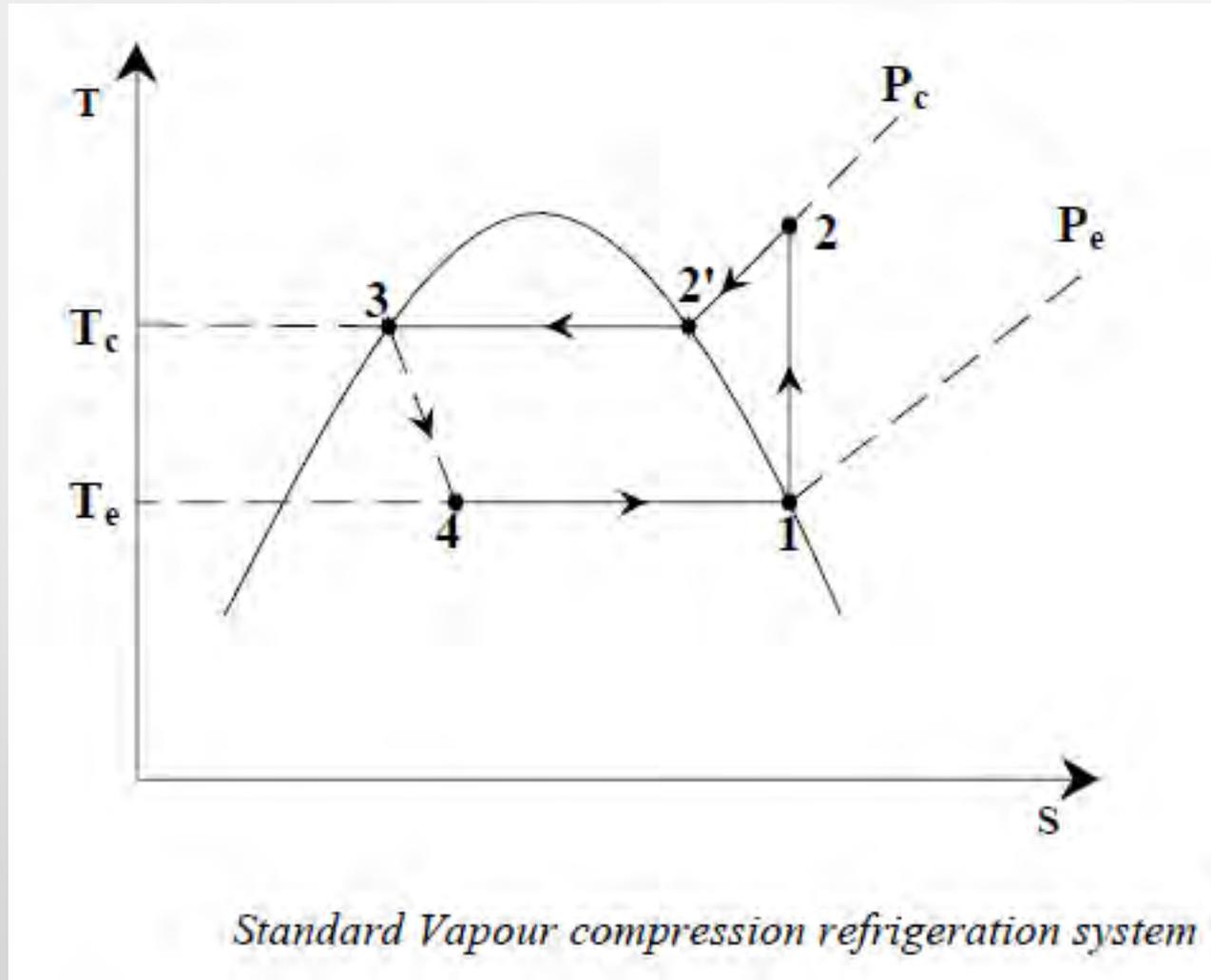
$$\eta_R = \frac{\text{COP}_{\text{VCRS}}}{\text{COP}_{\text{Carnot}}} = \left[ \frac{1 - \left( \frac{\text{area } A_2}{Q_{e,\text{Carnot}}} \right)}{1 + \left( \frac{\text{area } A_1 + \text{area } A_2}{W_{\text{net,Carnot}}} \right)} \right]$$

The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle. Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid.

If we assume that the potential and kinetic energy changes during isentropic compression process 1-2 are negligible, then the work input  $w_{1-2}$  is given by:

$$w_{1-2,\text{VCRS}} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

Figure shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a  $T-s$  diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor

Process 2-3: Isobaric heat rejection in condenser

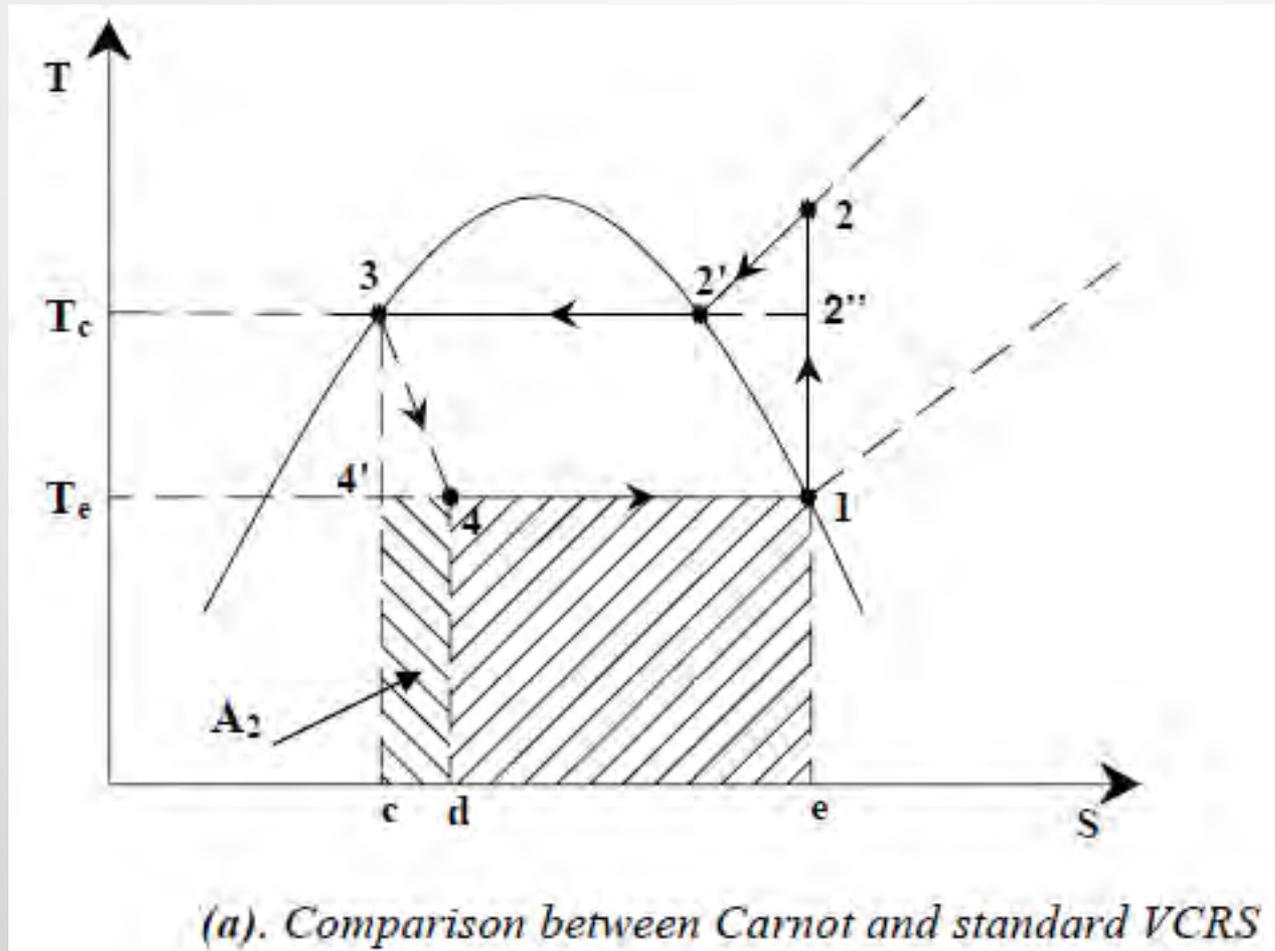
Process 3-4: Isenthalpic expansion of saturated liquid in expansion device

Process 4-1: Isobaric heat extraction in the evaporator

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non-isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3-4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the cycle diagrams on T s charts. Figure (a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

The heat extraction (evaporation) process is reversible for both the Carnot cycle and VCRS cycle. Hence the refrigeration effect is given by:

For Carnot refrigeration cycle (1-2''-3-4'):

$$q_{e,Carnot} = q_{4'-1} = \int_{4'}^1 T \cdot ds = T_e (s_1 - s_{4'}) = \text{area } e-1-4'-c-e$$

For VCRS cycle (1-2-3-4):

$$q_{e,VCRS} = q_{4-1} = \int_4^1 T \cdot ds = T_e (s_1 - s_4) = \text{area } e-1-4-d-e$$

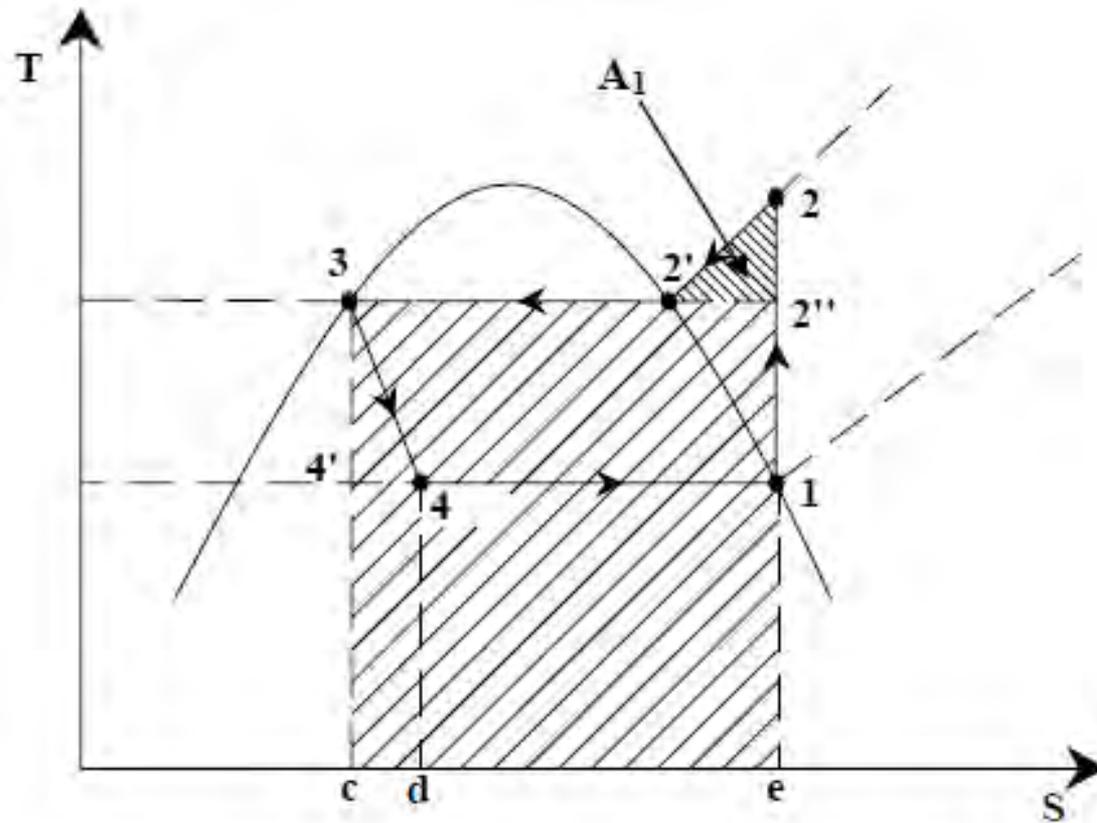
thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRS cycle, this reduction is equal to the area d-4-4'-c-d (area  $A_2$ ) and is known as *throttling loss*. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e.,

$$q_{e,Carnot} - q_{VCRS} = \text{area } d-4-4'-c-d = (h_3 - h_{4'}) = (h_4 - h_{4'}) = \text{area } A_2$$

It is easy to show that the loss in refrigeration effect increases as the evaporator temperature decreases and/or condenser temperature increases. A practical consequence of this is a requirement of higher refrigerant mass flow rate.

The heat rejection in case of VCRS cycle also increases when compared to Carnot cycle.

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)



(b). Comparative evaluation of heat rejection rate of VCRS and Carnot cycle

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

As shown in Fig. (b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

$$q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^3 T.ds = T_c (s_{2''} - s_3) = \text{area } e-2''-3-c-e$$

In case of VCRS cycle, the heat rejection rate is given by:

$$q_{c,VCRS} = -q_{2-3} = -\int_2^3 T.ds = \text{area } e-2-3-c-e$$

Hence the increase in heat rejection rate of VCRS compared to Carnot cycle is equal to the area 2''-2-2' (area  $A_1$ ). This region is known as *superheat horn*, and is due to the replacement of isothermal heat rejection process of Carnot cycle by isobaric heat rejection in case of VCRS.

Since the heat rejection increases and refrigeration effect reduces when the Carnot cycle is modified to standard VCRS cycle, the net work input to the VCRS increases compared to Carnot cycle. The net work input in case of Carnot and VCRS cycles are given by:

$$W_{net,Carnot} = (q_c - q_e)_{Carnot} = \text{area } 1-2''-3-4'-1$$

$$W_{net,VCRS} = (q_c - q_e)_{VCRS} = \text{area } 1-2-3-4'-c-d-4-1$$

As shown in Fig (c), the increase in net work input in VCRS cycle is given by:

$$W_{net,VCRS} - W_{net,Carnot} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2$$



# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

To summarize the refrigeration effect and net work input of VCRS cycle are given by:

$$q_{e,VCRS} = q_{e,Carnot} - \text{area } A_2$$

$$W_{net,VCRS} = W_{net,Carnot} + \text{area } A_1 + \text{area } A_2$$

The COP of VCRS cycle is given by:

$$COP_{VCRS} = \frac{q_{e,VCRS}}{W_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{W_{net,Carnot} + \text{area } A_1 + \text{area } A_2}$$

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

If we define the cycle efficiency,  $\eta_R$  as the ratio of COP of VCRS cycle to the COP of Carnot cycle, then:

$$\eta_R = \frac{\text{COP}_{\text{VCRS}}}{\text{COP}_{\text{Carnot}}} = \left[ \frac{1 - \left( \frac{\text{area } A_2}{Q_{e,\text{Carnot}}} \right)}{1 + \left( \frac{\text{area } A_1 + \text{area } A_2}{W_{\text{net,Carnot}}} \right)} \right]$$

The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle. Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid.

If we assume that the potential and kinetic energy changes during isentropic compression process 1-2 are negligible, then the work input  $w_{1-2}$  is given by:

$$w_{1-2,\text{VCRS}} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

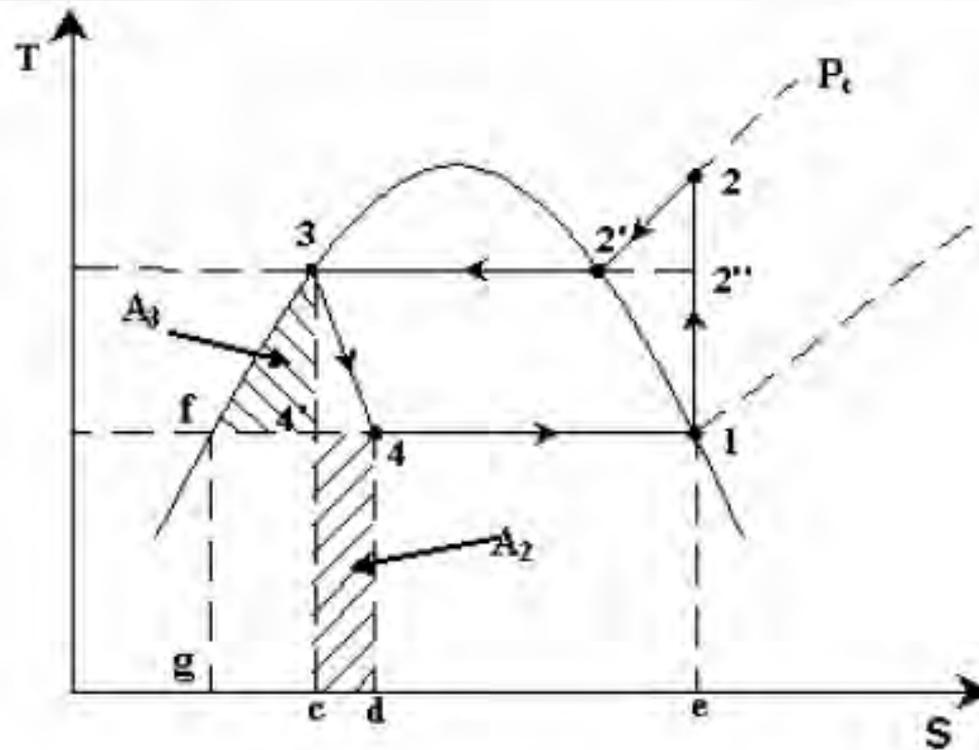


Figure showing saturated liquid line 3-f coinciding with the constant pressure line

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

Now as shown in Fig. if we further assume that the saturated liquid line 3-f coincides with the constant pressure line  $P_c$  in the subcooled region (which is a reasonably good assumption), then from the 2<sup>nd</sup> Tds relation;

$$Tds = dh - v dP = dh; \text{ when } P \text{ is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e$$

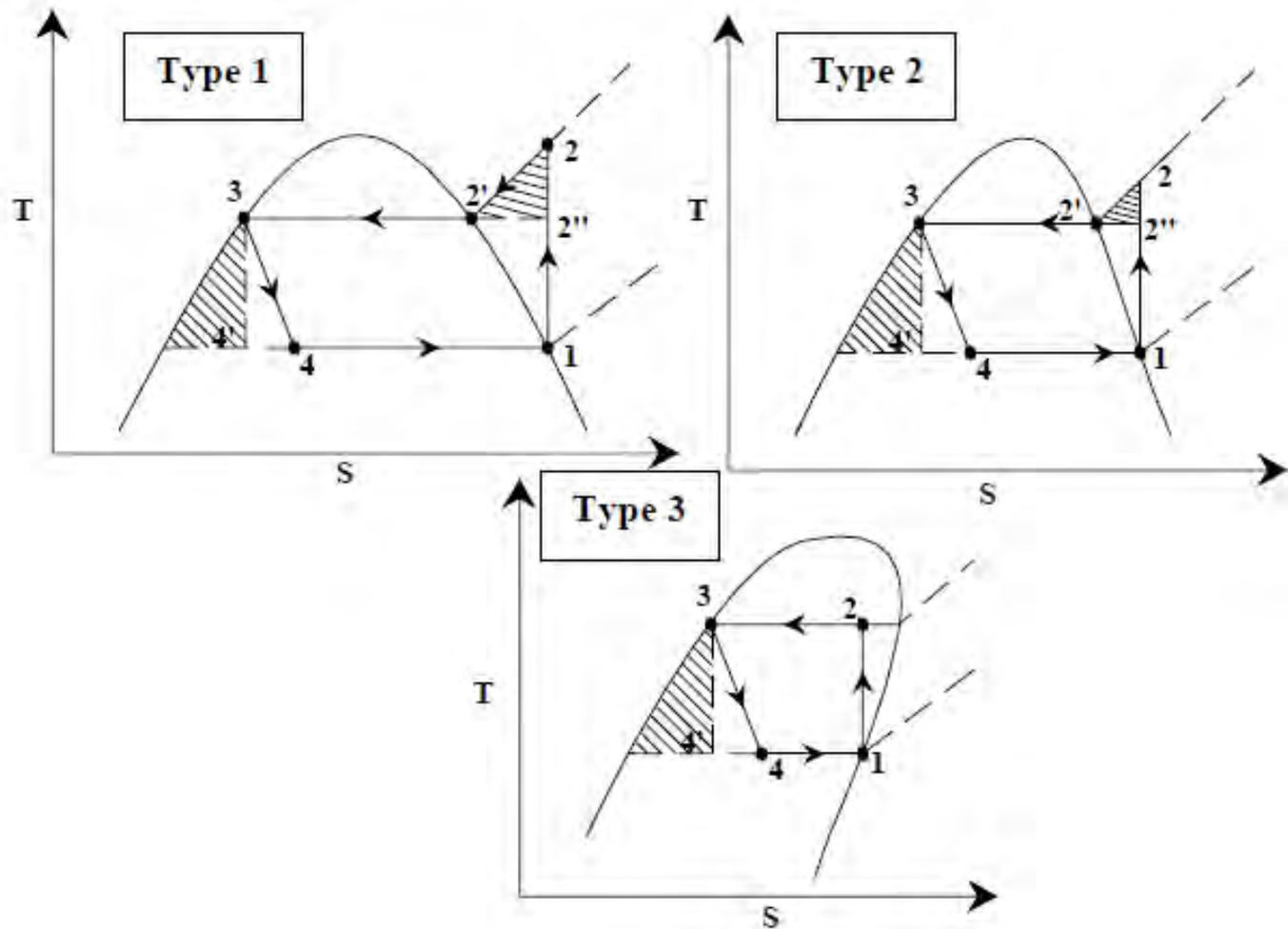
$$\text{and, } (h_1 - h_f) = \int_1^f Tds = \text{area } e-1-f-g-e$$

Substituting these expressions in the expression for net work input, we obtain the compressor work input to be equal to area 1-2-3-f-1. Now comparing this with the earlier expression for work input (area 1-2-3-4'-c-d-4-1), we conclude that area  $A_2$  is equal to area  $A_3$ .

As mentioned before, the losses due to superheat (area  $A_1$ ) and throttling (area  $A_2 \approx A_3$ ) depend very much on the shape of the vapor dome (saturation liquid and vapour curves) on T s diagram. The shape of the saturation curves depends on the nature of refrigerant.

# STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

Figure shows T-s diagrams for three different types of refrigerants.



*T-s diagrams for three different types of refrigerants*

# VAPOUR COMPRESSION REFRIGERATION SYSTEMS

A vapour compression refrigeration system is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant, is used. It condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The refrigerants, usually, used for this purpose are ammonia ( $\text{NH}_3$ ), carbon dioxide ( $\text{CO}_2$ ) and sulphur dioxide ( $\text{SO}_2$ ). The refrigerant used, does not leave the system, but is circulated throughout the system alternately condensing and evaporating. In evaporating, the refrigerant absorbs its latent heat from the brine (salt water) which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapour compression refrigeration system is, therefore a latent heat pump, as it pumps its latent heat from the brine and delivers it to the cooler.

The vapour compression refrigeration system is now-a-days used for all purpose refrigeration. It is generally used for all industrial purposes from a small domestic refrigerator to a big air conditioning plant.

# ADVANTAGES AND DISADVANTAGES OF VAPOUR COMPRESSION REFRIGERATION SYSTEM OVER AIR REFRIGERATION SYSTEM

Following are the advantages and disadvantages of the vapour compression refrigeration system over air refrigeration system:

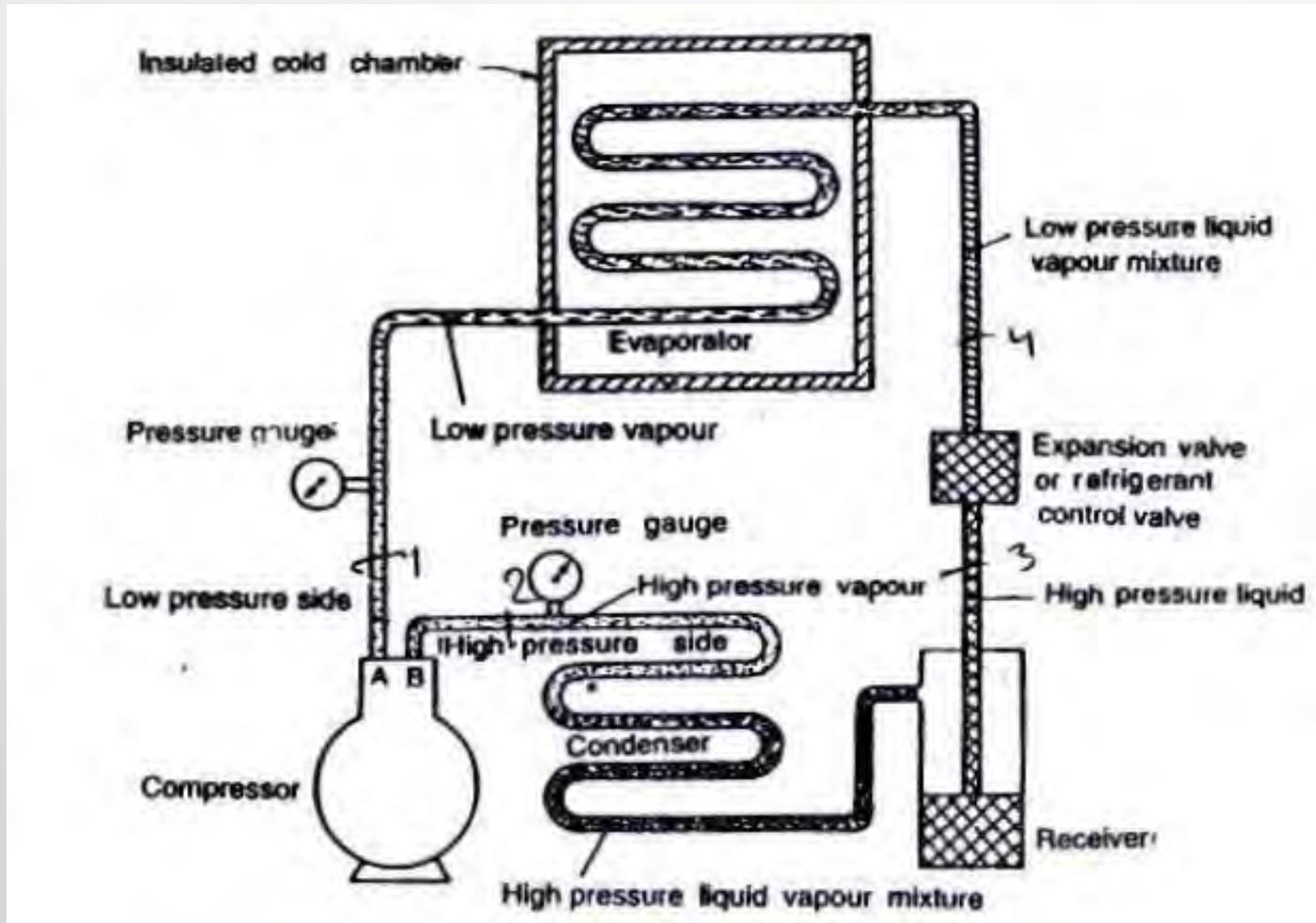
## *Advantages*

1. It has smaller size for the given capacity of refrigeration.
2. It has less running cost.
3. It can be employed over a large range of temperatures.
4. The coefficient of performance is quite high.

## *Disadvantages*

1. The initial cost is high
2. The prevention of leakage of the refrigerant is the major problem in vapour compression system

# MECHANISM OF A SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM



# MECHANISM OF A SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

It consists of the following five essential parts :

1. *Compressor.* The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged to the condenser through the delivery or discharge valve B.
2. *Condenser.* The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.
3. *Receiver.* The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.

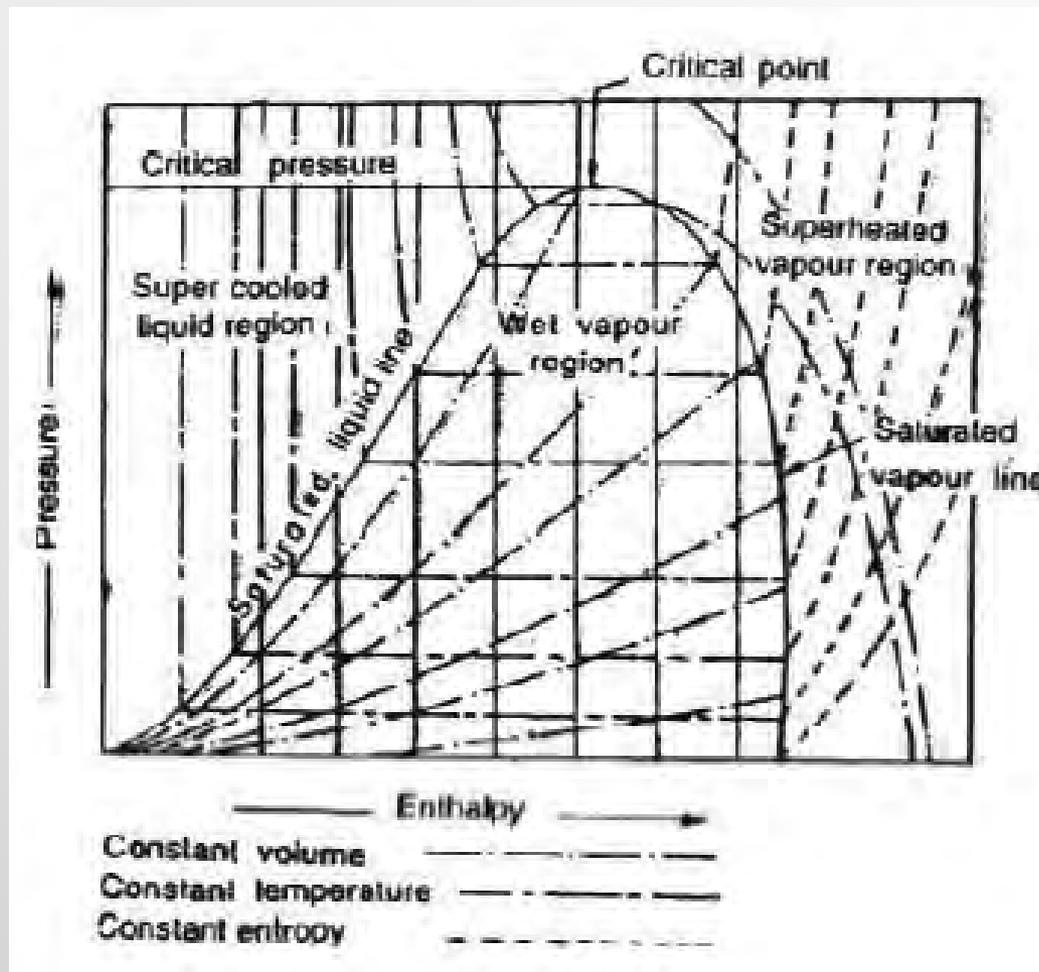
# MECHANISM OF A SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

4. *Expansion valve.* It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator at the low pressure and temperature.

5. *Evaporator.* An evaporator consists of coils of pipe in which the liquid-vapour refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which is to be cooled.

**Note :** In any compression refrigeration system, there are two different pressure conditions. One is called the high pressure side and other is known as low pressure side. The high pressure side includes the discharge line (i.e. piping from the evaporator to the suction valve A).

# PRESSURE-ENTHALPY (P-H) CHART



## PRESSURE-ENTHALPY (P-H) CHART

The most convenient chart for studying the behavior of a refrigerant is the p-h chart, in which the vertical ordinates represent pressure and horizontal ordinates represent enthalpy (i.e. total heat). A typical chart is shown in Fig., in which a few important lines of the complete chart are drawn. The saturated liquid line and the saturated vapour line merge into one another at the critical point. A saturated liquid is one which has a temperature equal to the saturation temperature corresponding to its pressure. The space to the left of the saturated liquid line will, therefore, be sub-cooled liquid region. The space between the liquid and the vapour lines is called wet vapour region and to the right of the saturated vapour line is a superheated vapour region.

In the following pages, we shall draw the p-h chart along with the T-s diagram of the cycle.

## TYPES OF VAPOUR COMPRESSION CYCLES

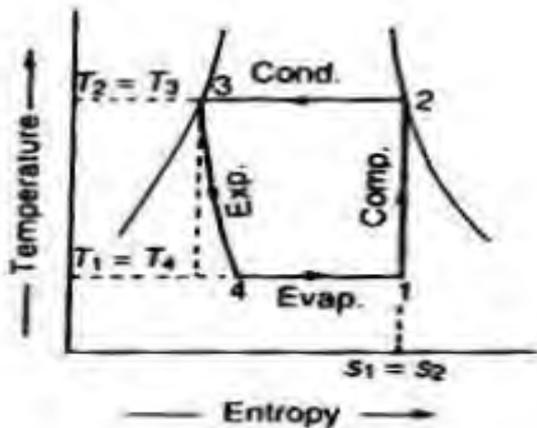
We have already discussed that vapour compression cycle essentially consists of compression, condensation, throttling and evaporation. Many scientists have focused their attention to increase the coefficient of performance of the cycle. Through there are many cycles, yet the following are important from the subject point of view :

1. Cycle with dry saturated vapour after compression,
2. Cycle with wet vapour after compression,
3. Cycle with superheated vapour after compression,
4. Cycle with superheated vapour before compression, and
5. Cycle with undercooling or sub cooling of refrigerant.

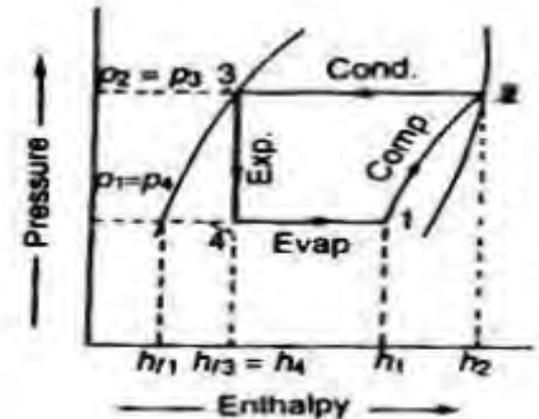
Now we shall discuss all the above mentioned cycles, one by one, in the following pages.

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

A vapour compression cycle with dry saturated vapour after compression is shown on T-s and p-h diagrams. At point 1, let  $T_1$ ,  $p_1$  and  $s_1$  be the temperature, pressure and entropy of the vapour refrigerant respectively. The four process of the cycle are as follows :



(a) T-s diagram.



(b) p-h diagram.

Theoretical vapour compression cycle with dry saturated vapour after compression

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

1. *Compression process.* The vapour refrigerant at low pressure  $p_1$  and temperature  $T_1$  is compressed isentropically to dry saturated vapour as shown by the vertical line 1-2 on T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rises from  $p_1$  to  $p_2$  and  $T_1$  to  $t_2$  respectively.

The work done during isentropic compression per kg of refrigerant is given by

$$w = h_2 - h_1$$

where  $h_1$  = Enthalpy of vapour refrigerant at temperature  $T_1$ , i.e. at suction of the compressor, and  $h_2$  = Enthalpy of the vapour refrigerant at temperature  $T_2$ , i.e. at discharge of the compressor.

2. *Condensing process.* The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure  $p_2$  and temperature  $T_2$ , as shown by the

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

horizontal line 2-3 on T-s and p-h diagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

3. *Expansion process.* the liquid refrigerant at pressure  $p_3 = p_2$  and temperature  $T_3 = T_2$  is expanded by \*throttling process through the expansion valve to a low pressure  $p_4 = p_1$  and temperature  $T_4 = T_1$ , as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram. We have already discussed that some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

**Notes :** (a) In case an expansion cylinder is used in place of throttle or expansion valve to expand the liquid refrigerant, then the refrigerant will expand isentropically as shown by dotted vertical line on T-s diagram in Fig. (a). The isentropic expansion reduces the external work being expanded in running the compressor and increases the refrigerating effect. Thus, the net result of using the expansion cylinder is to increase the coefficient of performance.

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

Since the expansion cylinder system of expanding the liquid refrigerant is quite complicated and involves greater initial cost, therefore its use is not justified for small gain in cooling capacity. Moreover, the flow rate of the refrigerant can be controlled with throttle valve which is not possible in case of expansion cylinder which has a fixed cylinder volume.

**(b)** In modern domestic refrigerators, a capillary (small bore tube) is used in place of an expansion valve.

4. *Vaporizing process.* The liquid-vapour mixture of the refrigerant at pressure  $p_4 = p_1$  and temperature  $T_4 = T_1$  is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which is to be cooled. This heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written as RE. The process of vaporization continues up to point 1 which is the starting point and thus the cycle is completed.

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by

$$R_E = h_1 - h_4 = h_1 - h_{f3} \quad \dots (h_{f3} = h_4)$$

where  $h_{f3}$  = Sensible heat at temperature  $T_3$ , i.e. enthalpy of liquid refrigerant leaving the condenser.

It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant.

∴ Coefficient of performance,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

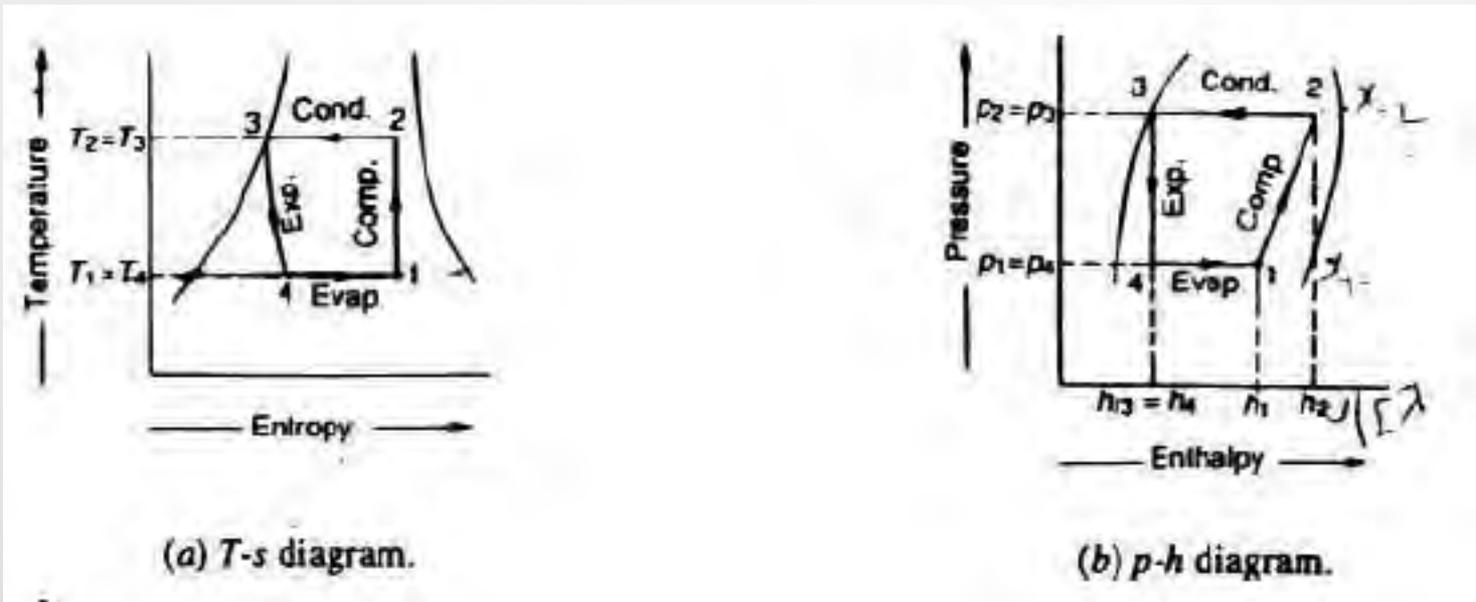
# THEORETICAL VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

**Example 1.** In an ammonia vapour compression system, the pressure in the evaporator is 2 bar. Ammonia at exit is 0.85 dry and at entry its dryness fraction is 0.19. During compression, the work done per kg of ammonia is 150 kJ. Calculate the C.O.P. and the volume of vapour entering the compressor per minute, if the rate of ammonia circulation is 4.5 kg/min. The latent heat and specific volume at 2 bar are 1325 kJ/kg and  $0.58 \text{ m}^3/\text{kg}$  respectively.

**Example 2.** The temperature limits of an ammonia refrigerating system are  $25^\circ \text{C}$  and  $-10^\circ \text{C}$ . If the gas is dry at the end of compression, calculate the coefficient of performance of the cycle assuming no undercooling of the liquid ammonia. Use the following table for properties of ammonia:

<i>Temperature</i> $(^\circ \text{C})$	<i>Liquid heat</i> $(\text{kJ/kg})$	<i>Latent heat</i> $(\text{kJ/kg})$	<i>Liquid entropy</i> $(\text{kJ/kg K})$
25	298.9	1166.94	1.1242
-10	135.37	1297.68	0.5443

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH WET VAPOUR AFTER



## Compression

A vapour compression cycle with wet vapour after compression is shown on  $T-s$  and  $p-h$  diagrams in Fig. (a) and (b) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at points 1 and 2.

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH WET VAPOUR AFTER

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

## SUPERHEAT

In theory, the refrigerant leaves the evaporator as a vapor at point '1', however, in real applications, additional heat, called "superheat" is added to prevent liquid condensation in the lines that can damage the compressor (shown as point 1a). Superheat is the heat added to the vapor beyond what is required to vaporize all of the liquid. Since refrigeration and air-conditioning compressors are designed to compress vapor refrigerant, some superheating is necessary to ensure that no liquid refrigerant can return to the compressor. The amount of superheat is determined by the amount of liquid refrigerant admitted to the evaporator. This, in turn, is controlled by the expansion valve (TXV) and that's why it is also known as metering device. A temperature range of 4° to 12°F of superheat is considered desirable to prevent liquid carry-over into the compressor (flooding back).

# THEORETICAL VAPOUR COMPRESSION CYCLE WITH WET VAPOUR AFTER

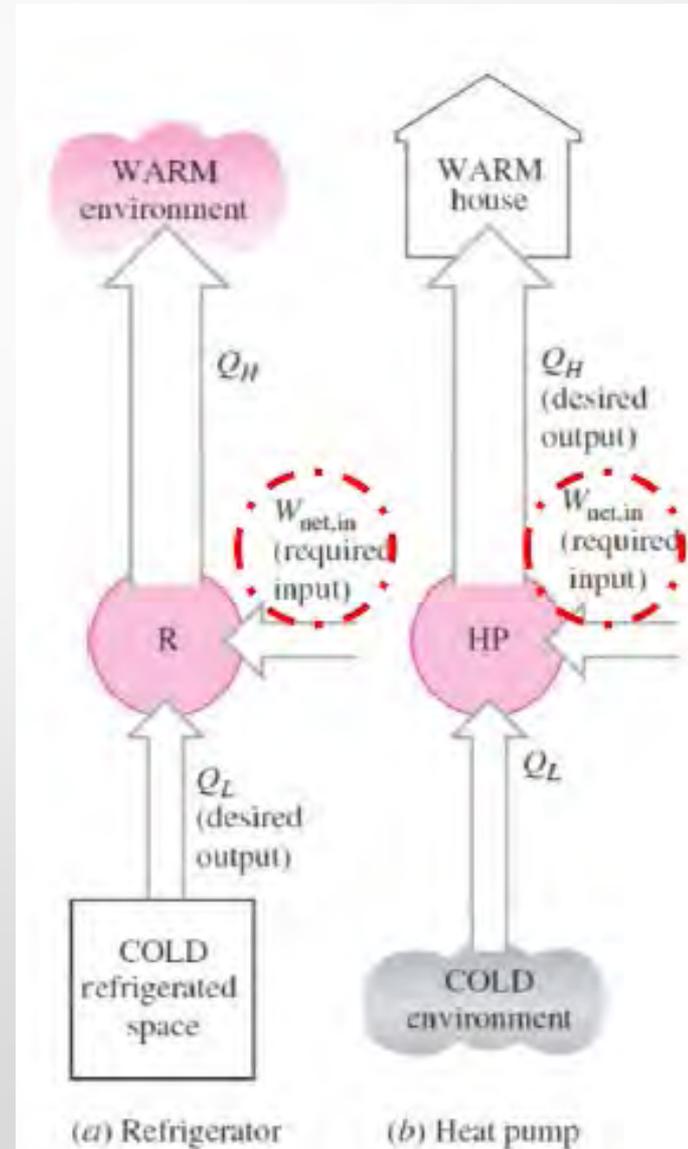
## **SUB-COOLING**

*Sub-cooling is the process of cooling condensed gas below its saturated pressure-temperature. Sub cooling assures that no gas is left at the end of the condensing phase, thus assuring maximum capacity at the expansion valve. Sub-cooling is best accomplished in a separate sub-cooler or a special sub-cooling section of a condenser because tube surface must be submerged in liquid refrigerant for sub-cooling to occur. Sub-cooling can have a dramatic effect in the capacity of a refrigeration system by increasing the capacity of the refrigerant to absorb heat during the evaporation phase for the same compressor kW input.*

# INTRODUCTION

## BASICS OF COMPRESSION PROCESS

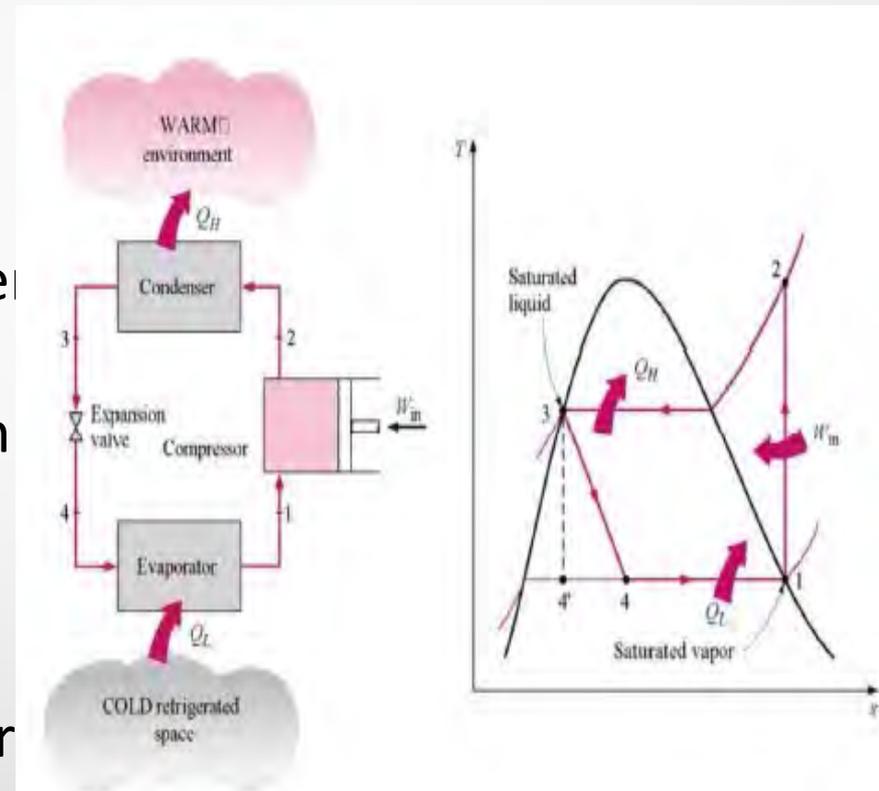
- The compressor is the heart of the system.
- The compressor does just what it's name is.
- It compresses the low pressure refrigerant vapor from the evaporator and compresses it into a high pressure vapor.
- External work done on the system
- Electric power (AC/DC)
- Solar → Electricity → Compressor power



# IDEAL VAPOR COMPRESSION CYCLE

## IDEAL CYCLE CONSISTS OF

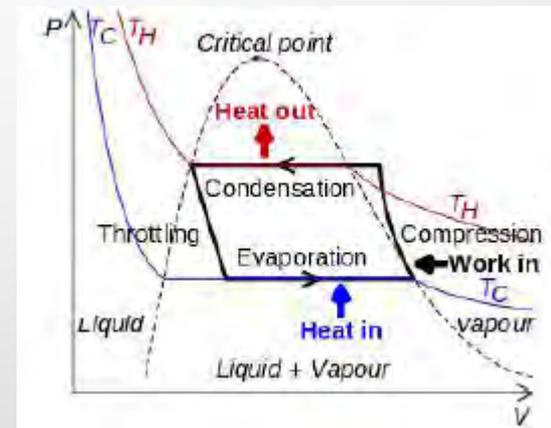
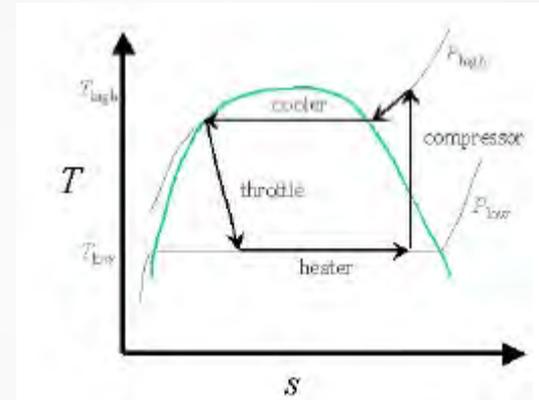
- 1-2: Isentropic compression
- 2-3: Constant pressure heat rejection in the condenser
- 3-4: Throttling in an expansion valve
- 4-1: Constant pressure heat addition in the evaporator





# COMPRESSION PROCESS

- During compression of vapor, work is done on it
- Its enthalpy increases and so does its pressure
- Inlet to the compressor should be in vapor state
- Ideal compression (isentropic) is not possible in real operation.







# Classification of compressors

## Open types

- have a separate housing for the compressor and the motor.
- They rely on lubricant in the system to splash on pump components and seals.
- If not operated frequently, the system can leak its operating gases.
- Open compressors can be driven by non-electric power sources such as combustion engines.



## Classification of compressors

- **Hermetic types** seal the compressor and motor together in the same housing.

*These compressors are leak-free and can sit for long periods unused, but cannot be maintained or repaired.*

- **Semi-hermetic types** also contain the motor and compressor in one housing, but instead of a one-piece housing they incorporate gasketed/bolted covers.

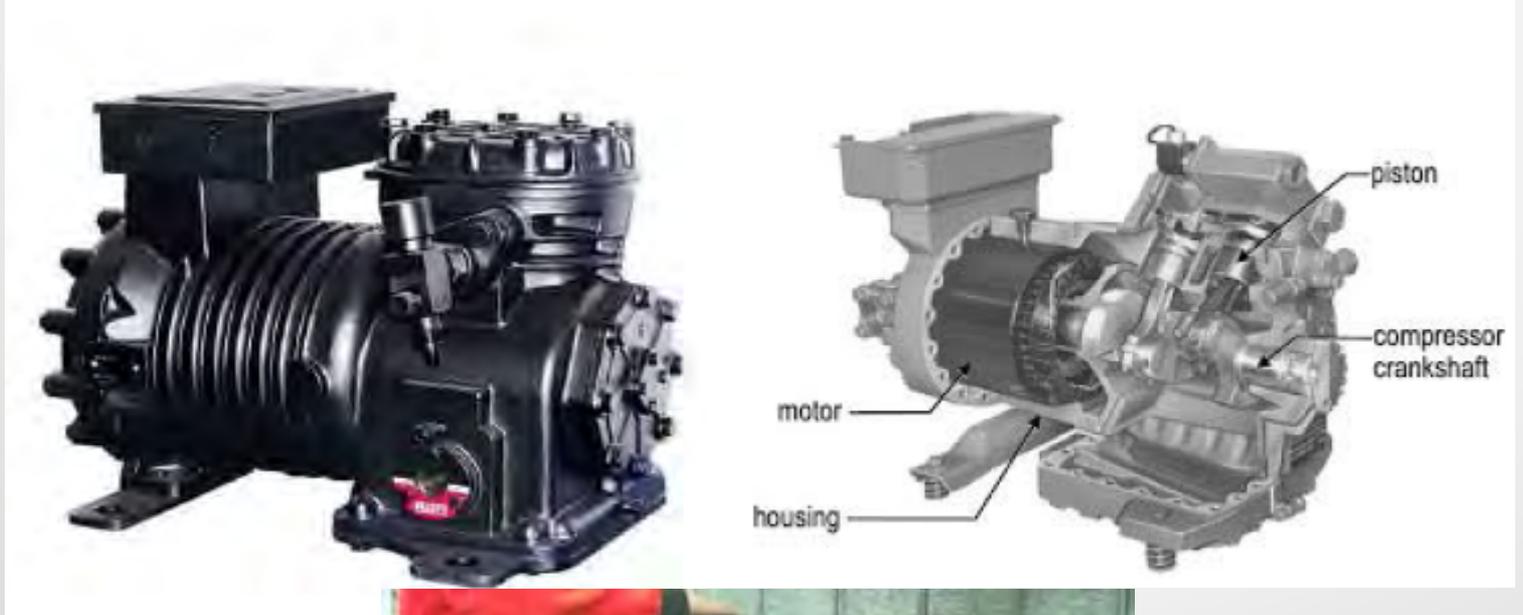
*These can be removed for maintenance and repair of the compressor or motor.*

# OPEN TYPE COMPRESSORS





# SEMI-HERMETIC COMPRESSOR UNITS





# HERMETICALLY SEALED COMPRESSOR UNITS





# TYPES OF COMPRESSORS

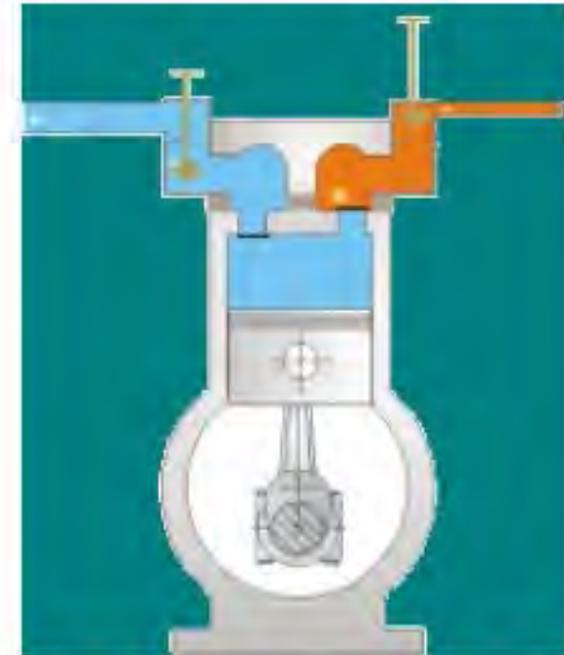
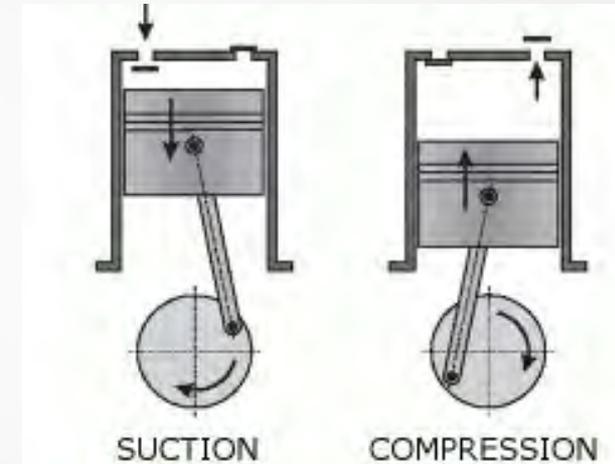
There are basically 5 types of air conditioner compressors that are commonly used in the HVAC industry:

- Reciprocating
- Scroll
- Screw
- Rotary
- Centrifugal



# Reciprocating/Piston Compressors

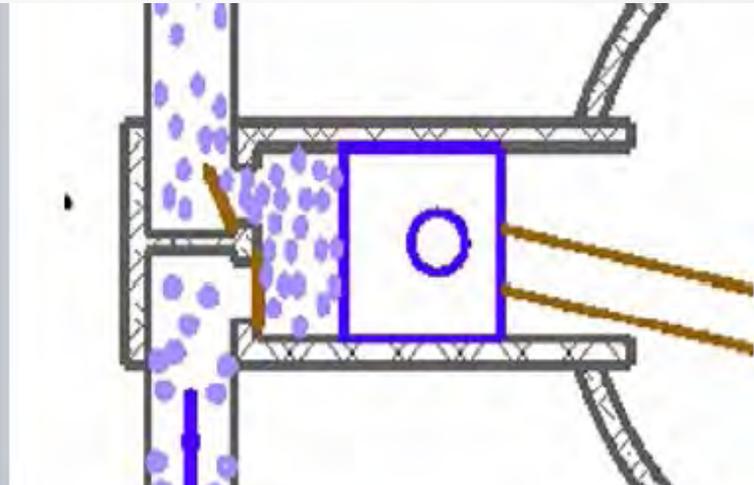
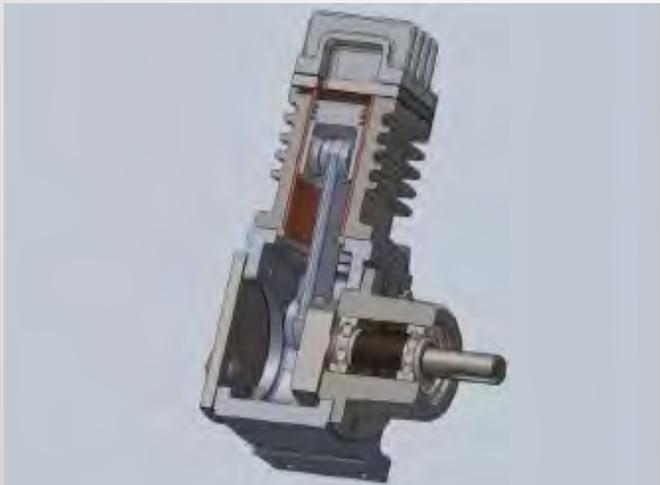
- Piston and cylinder arrangement to provide compressive force - like IC engines.
- Reciprocating motion of the piston due to external power compresses the refrigerant inside the cylinder.
- Low initial cost and a simple, easy to install design.
- Large power output range - can reach extremely high pressures. However, maintenance costs are high, potential vibrational issues.
- Not typically designed to run continuously at full capacity.





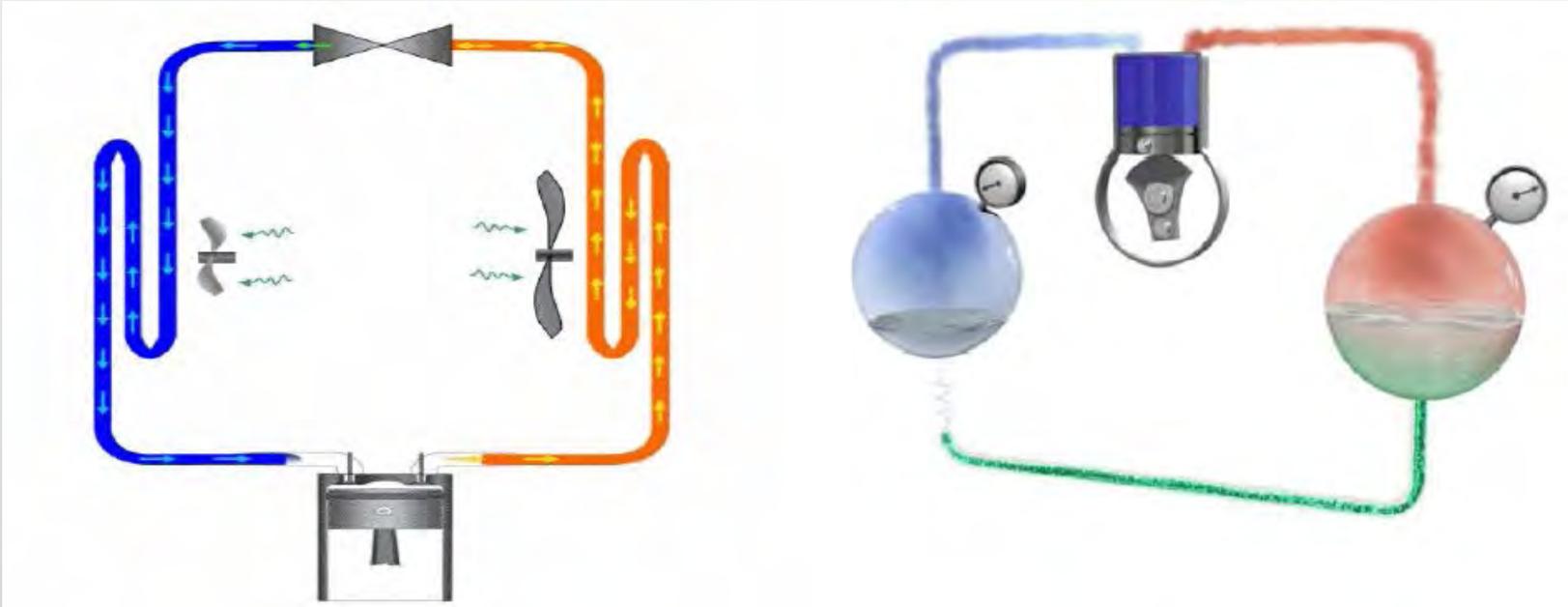
# Reciprocating Compressor

- Suction and discharge valves
- Valve position is controlled by the pressure difference across it
- Can be open, semi-hermetic or fully hermetic





# Reciprocating compressor action in VCR





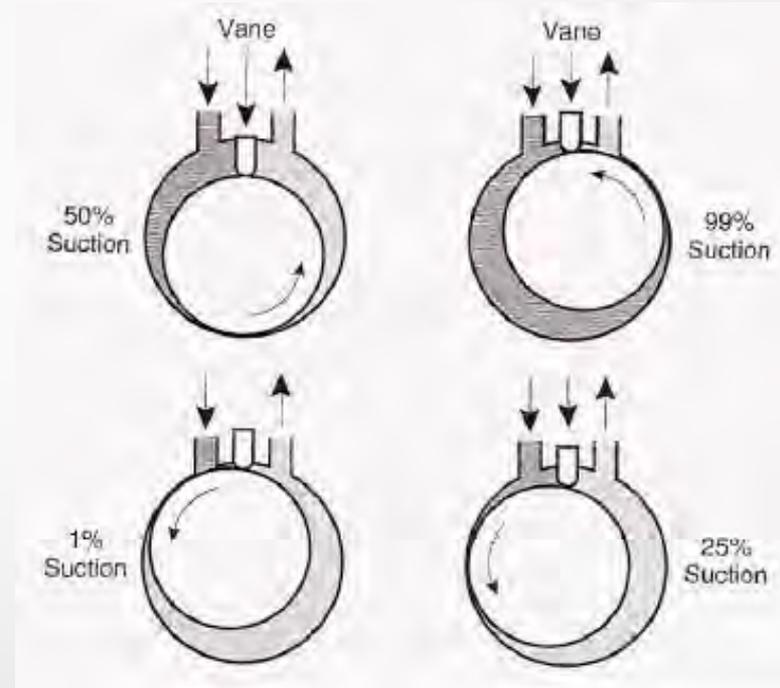
# Rotary Compressors

- Uses circular motion for compression (two rotating elements)
- Rotating blade (vane) type

*Refrigerant is trapped by rotating vanes*

*Refrigerant compresses as volume decreases*

- Stationary blade (vane) type  
*Equipped with only one blade or vane*

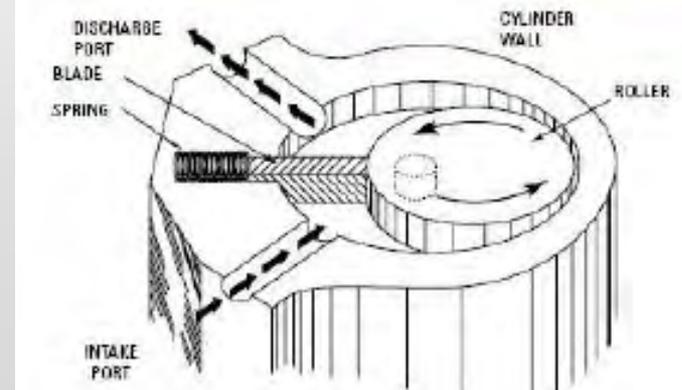
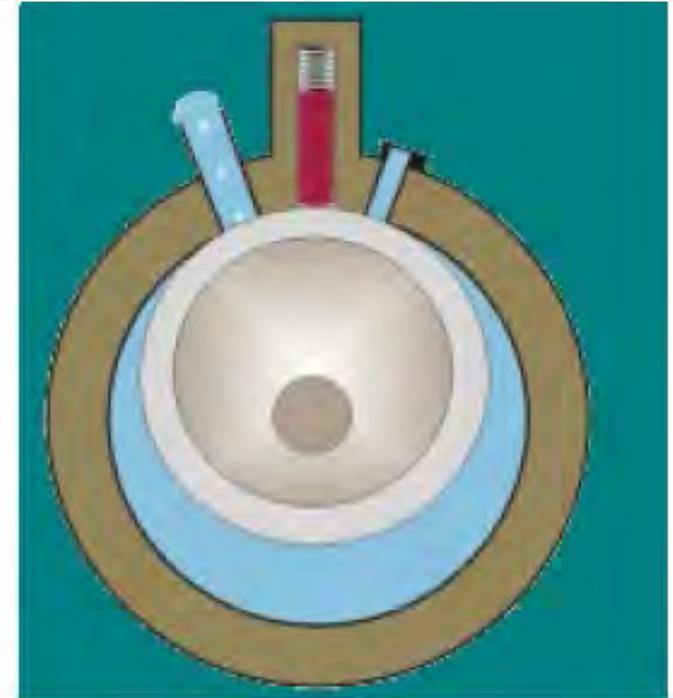




# Rotary Compressors

In general, rotary compressors are:

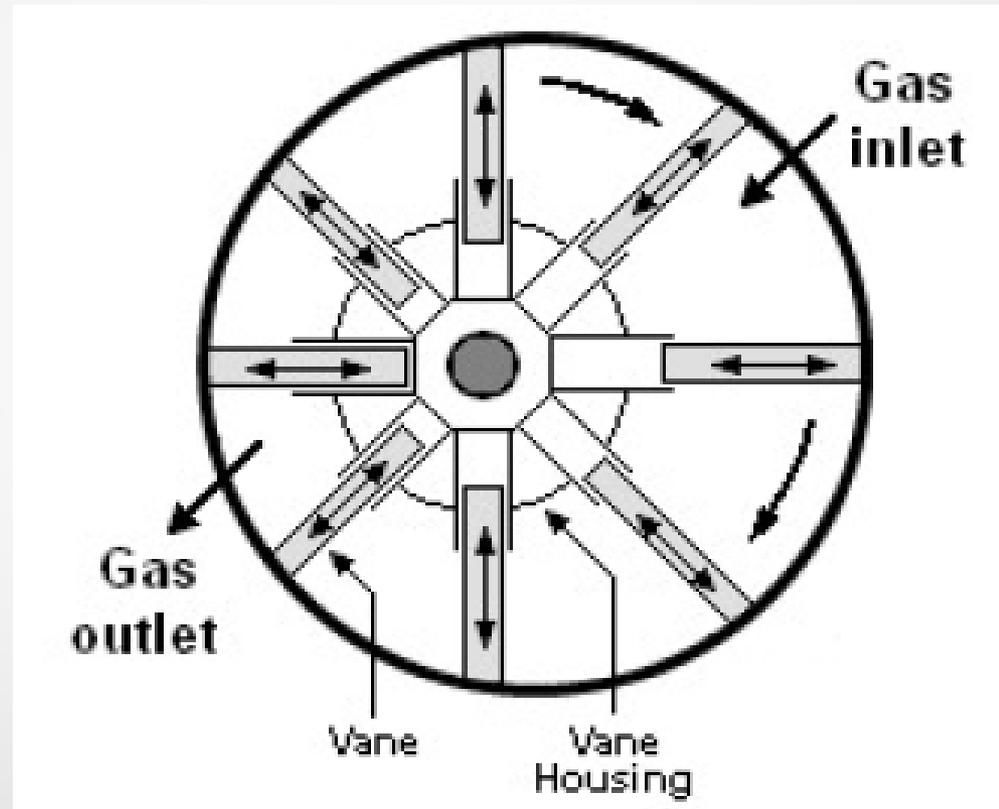
- Fewer moving parts → quite efficient
- Low rotational speeds, low initial and maintenance costs
- Limited to smaller volumes of the gas
- Produce less pressure than other types of compressors.
- Actions of taking in refrigerant and compressing refrigerant occurs simultaneously





# Rotary Compressors

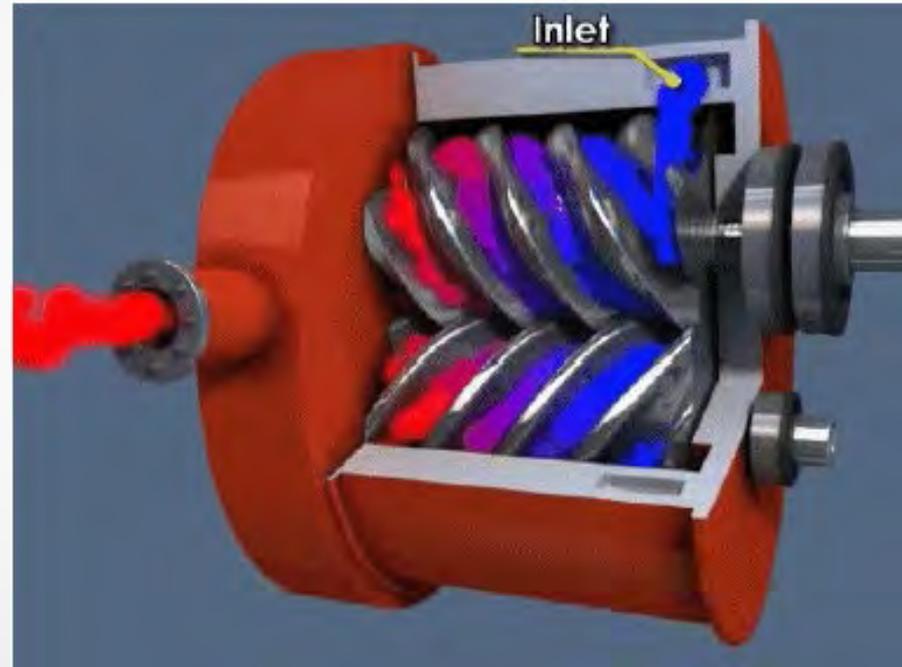
As the off-center drive shaft rotates, so does the vane housing and the vane slides in and out to maintain contact with the cylindrical compressor wall. Gas enters at the largest opening and the compressed gas discharges from the smallest opening.





## Helical rotary (screw type compressors)

- Screw compressors use a pair of helical rotors or screws which mesh together to compress the refrigerant between them.
- The volume of the refrigerant decreases as it flows through the compressor
- A continuous, flowing output is produced
- Compressor capacity is controlled by a slide valve





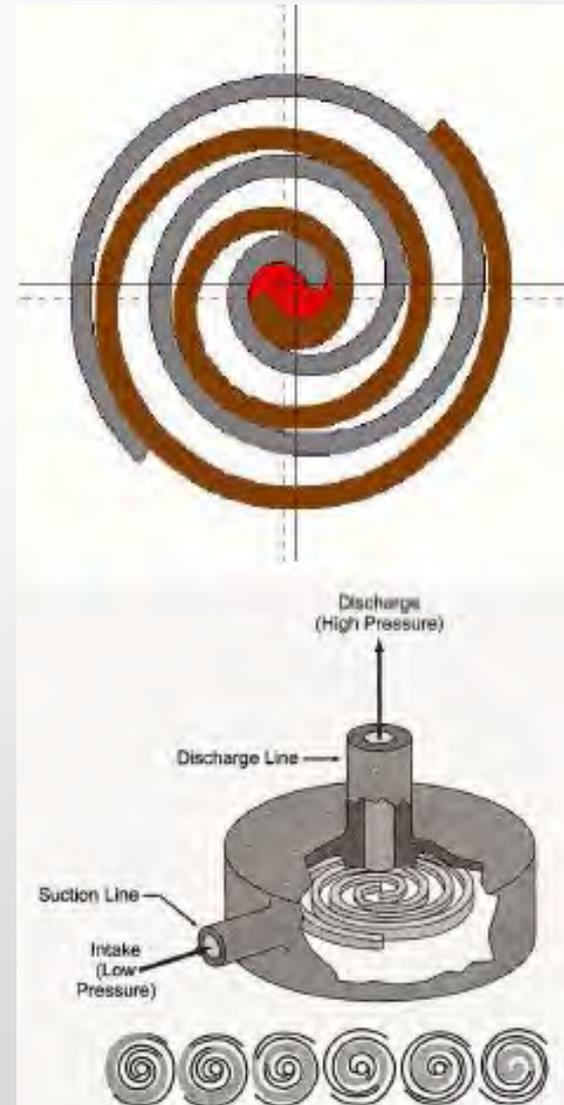
## Screw compressors

- They can produce high pressure for a small quantity of gas and consume less power than reciprocating compressors.
- They have low to medium initial and maintenance costs and few moving parts.
- However, they have difficulty in dirty environments, high rotational speeds, and shorter life expectancies than other designs.



# Scroll type Compressors

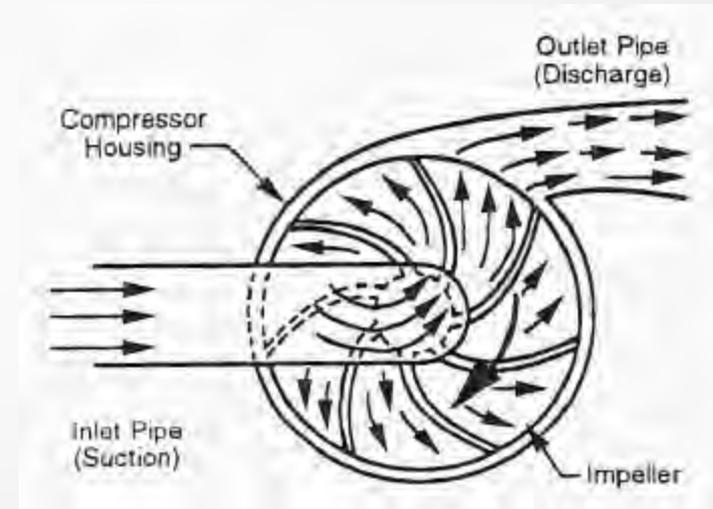
- Utilizes two identically machined scrolls – one scroll is stationary, the other orbits
- The nesting of the scrolls traps vapor
- Gas is introduced from the outer edge refrigerant is discharged from the center
- Scroll compressors are quiet, smooth-operating units with few moving parts
- Highest efficiency ratio of all compressor types.
- However, as fully hermetic designs, they cannot be easily repaired. They also typically cannot rotate in both directions.





# Centrifugal type compressors

- Rely on centrifugal force - utilizes an impeller
- No pistons, valves or cylinders
- Unlike other designs, centrifugal compressors do not operate on the positive displacement principle, but have fixed volume chambers.
- Well suited to compressing large volumes of refrigerant to relatively low pressures.
- Typically used for very large applications
- Capacity is controlled by inlet vanes





# Centrifugal Compressors



- The compressive force generated by an impeller wheel is small, so systems that use centrifugal compressors usually employ two or more stages (impellers wheels) in series to generate high compressive forces.
- Centrifugal compressors are desirable for their simple design, few moving parts, and energy efficiency when operating multiple stages.



# Compressor Specifications



- There are a number of specifications to consider when selecting compressors.
- These include
  - ❖ *Compressor capacity*
  - ❖ *Condensing temperature*
  - ❖ *Evaporating temperature*
  - ❖ *Refrigerant flow rate*
  - ❖ *Power consumption*



## Compressor specifications: Capacity



- Capacity (kW/hr – BTU/hr) measures the ability of a refrigerant compressor to remove heat from the refrigerant gas.
- Nominal capacity ratings are based on a standard set of conditions which include condensing temperature (CT), evaporative temperature (ET), refrigerant, and motor rpm.
- Typically, refrigeration compressors and air conditioning compressors can run at many different values for these parameters, with corresponding changes in their cooling capacity.
- Once in use, compressors can be tweaked and adjusted to the desired capacity and operating conditions.



## Compressor specifications: Other parameters



- Condensing temperature is the range of condensing temperatures over which the compressor is rated to operate.
- Evaporating temperature is the range of evaporative temperatures over which the compressor is rated to operate.
- Flow rate is the rate (by mass) at which the fluid is passed through the compressor, measured in kilograms per hour (kg/hr).
- Power (W) is the input power required to run the compressor motor at a specific operating point.



## Compressor specifications: Other parameters



- Refrigeration compressors and air conditioning compressors also carry power-source specifications defined by voltage/frequency/phase.
- Common choices include 12 VDC and 24 VDC, as well as 115/60/1, 230/50/1, 200-230/60/1, 208-230/60/3, 380-440/50/3



# Compressor Features



- RAC compressors incorporate a number of special features:
- ❖ Low noise - compressor operation generates less noise for applications where a quiet environment is desired.
- ❖ Light weight - compressor is compactly built or constructed with low-density materials for cooling systems which require low weight components.
- ❖ Variable speed - compressor has speed adjustment for running at various operating flow rates and conditions.



## Compressor Features



- ❖ Thermal shut off - compressor features controls which turn the compressor off at high temperatures to prevent it from overheating. They also can provide restart once the compressor has cooled down below a certain temperature.
- ❖ Sealing - describes how the compressor and motor drive are situated in relation to the gas or vapor being compressed.



# REFRIGERANT EVAPORATORS AND CONDENSORS

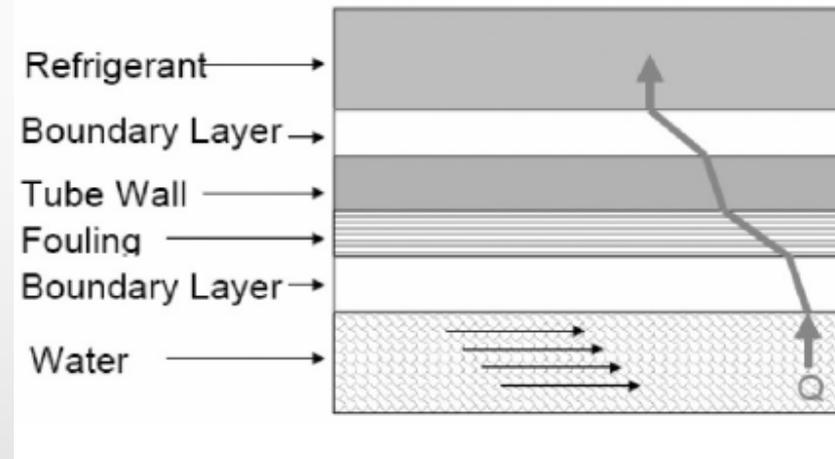
## Purpose of an evaporator:

- The purpose of the evaporator is to receive low-pressure, low temperature fluid from the expansion valve and to bring it in close thermal contact with the load.
- The refrigerant takes up its latent heat from the load and leaves the evaporator as a dry gas.
- The function of the evaporator will be to cool gas, liquid or other product loads.
- In most cases air or a liquid is first cooled, and this is then used to cool the load. e.g., in a cold-room air is cooled and this air cools the stored produce and carries away heat leaking through the structure; in a water chiller, water is circulated to cool the load, etc.



# Thermal path/ Resistances

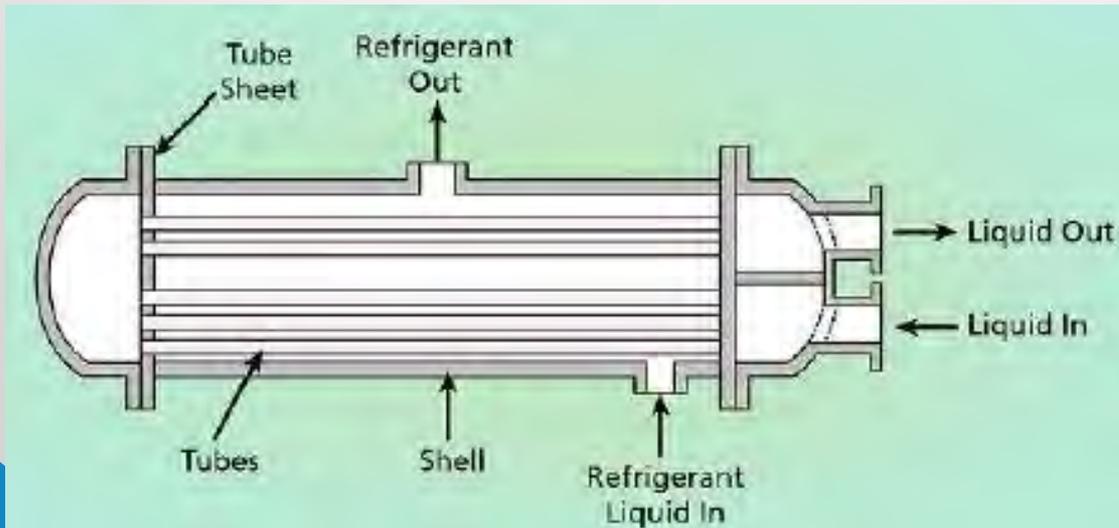
- Heat has to pass from the hot water to cold refrigerant.
- The thermal resistances include:
  - ❖ Water side heat transfer coefficient
  - ❖ Water side fouling
  - ❖ Conduction in tube wall
  - ❖ Refrigerant side fouling (usually low/negligible)
  - ❖ Refrigerant side heat transfer coefficient





# Classification of evaporators

- Evaporators are classified according to their refrigerant flow pattern, their function, geometrical features.
- Evaporators which cool water
- Evaporators which cool air





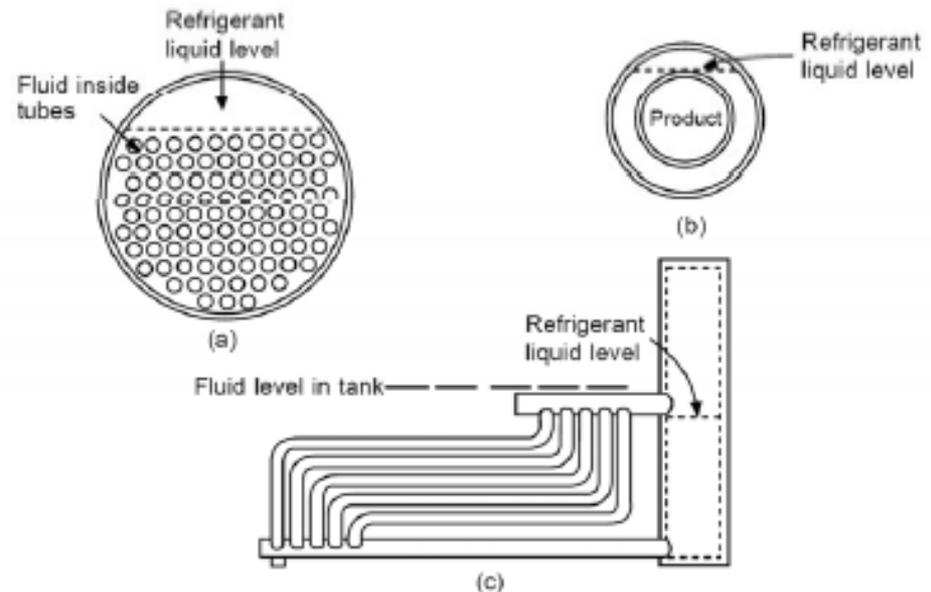
# EVAPORATORS WHICH COOL LIQUID

## (TYPES AND CHARACTERISTIC FEATURES)

### FLOODED TYPE EVAPORATORS

Flooded evaporators.

- (a) Shell-and-tube
- (b) Jacketed
- (c) Raceway

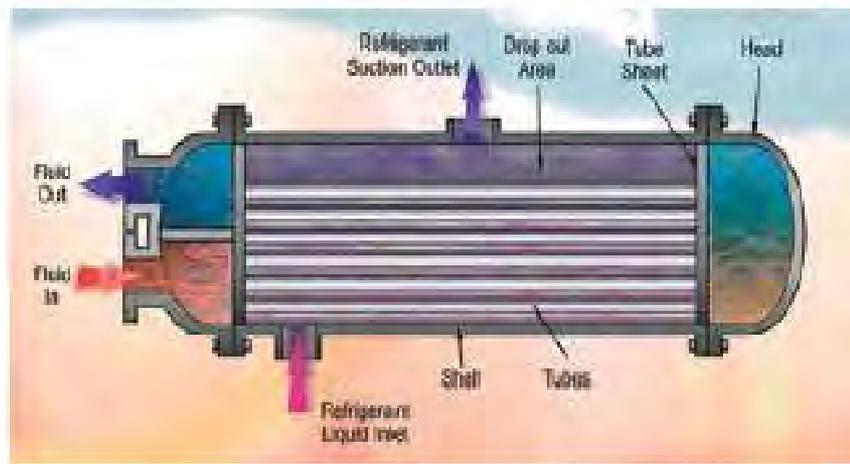
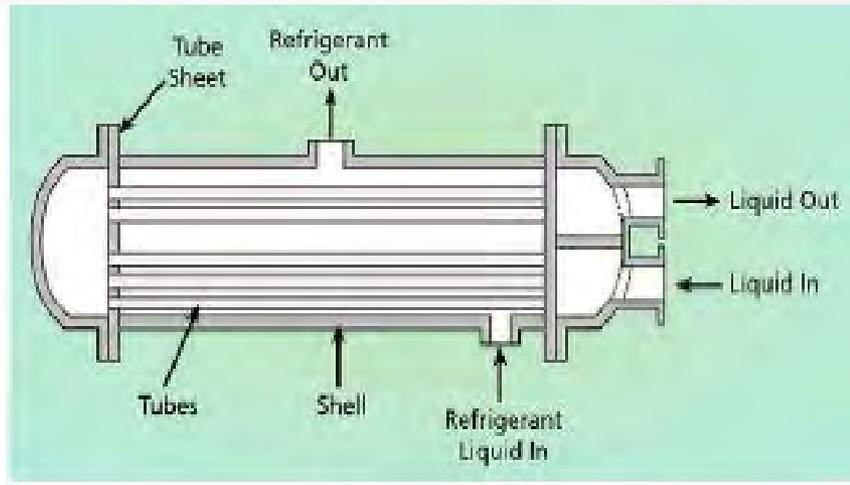




# FLOODED SHELL AND TUBE EVAPORATOR

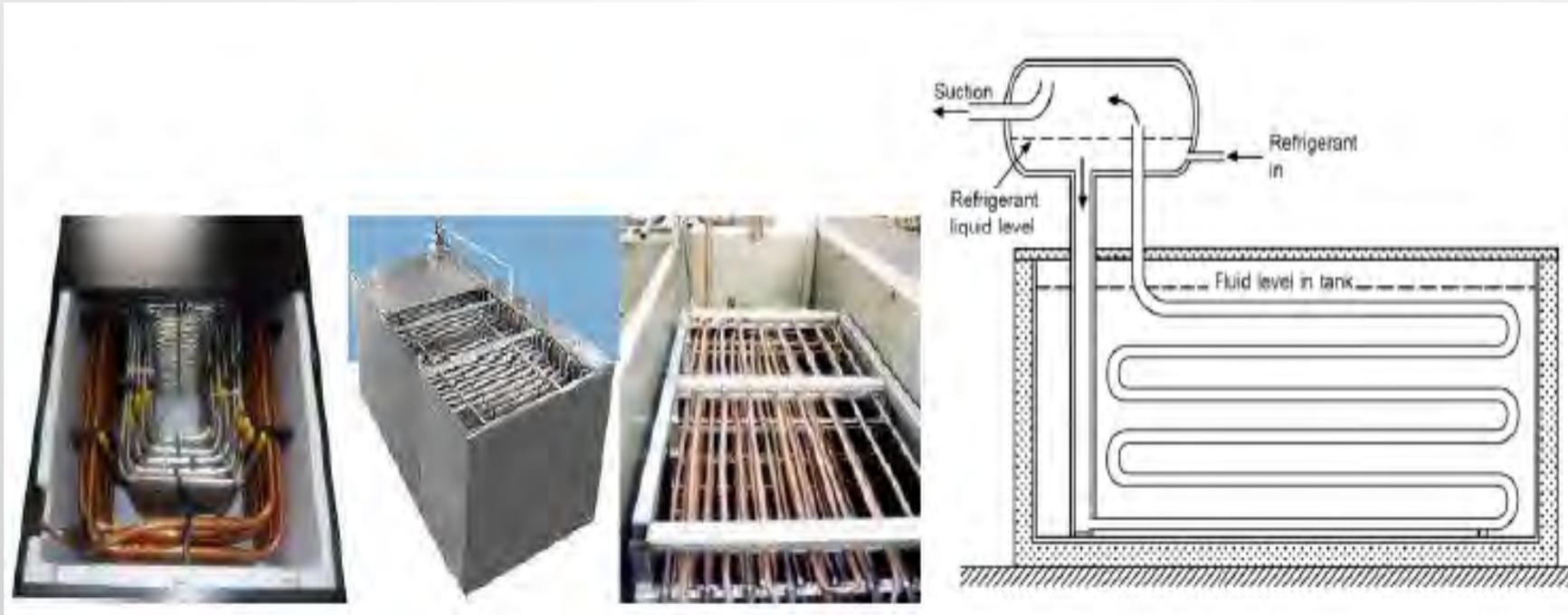


Alfa Laval DH  
Shell-and-tube evaporator



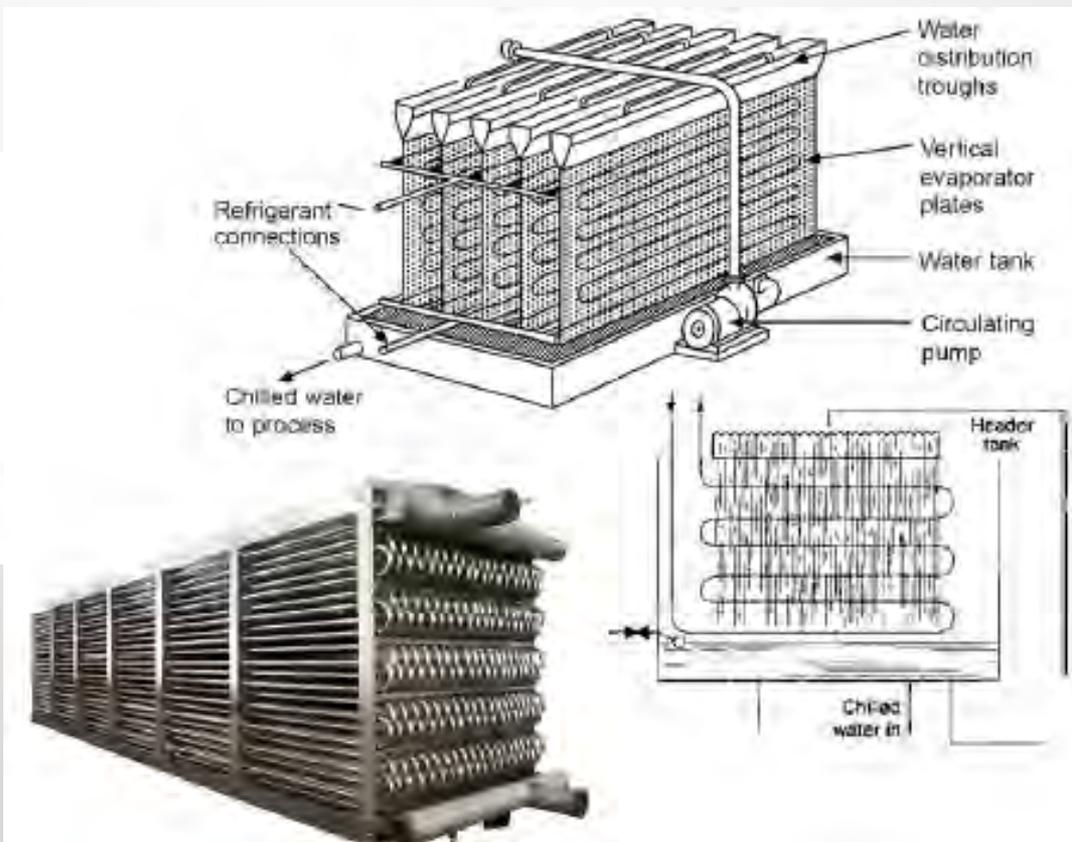
# FLOODED TANK EVAPORATOR

A flooded tank evaporator showing the refrigerant tubes and the tank filled with secondary fluid



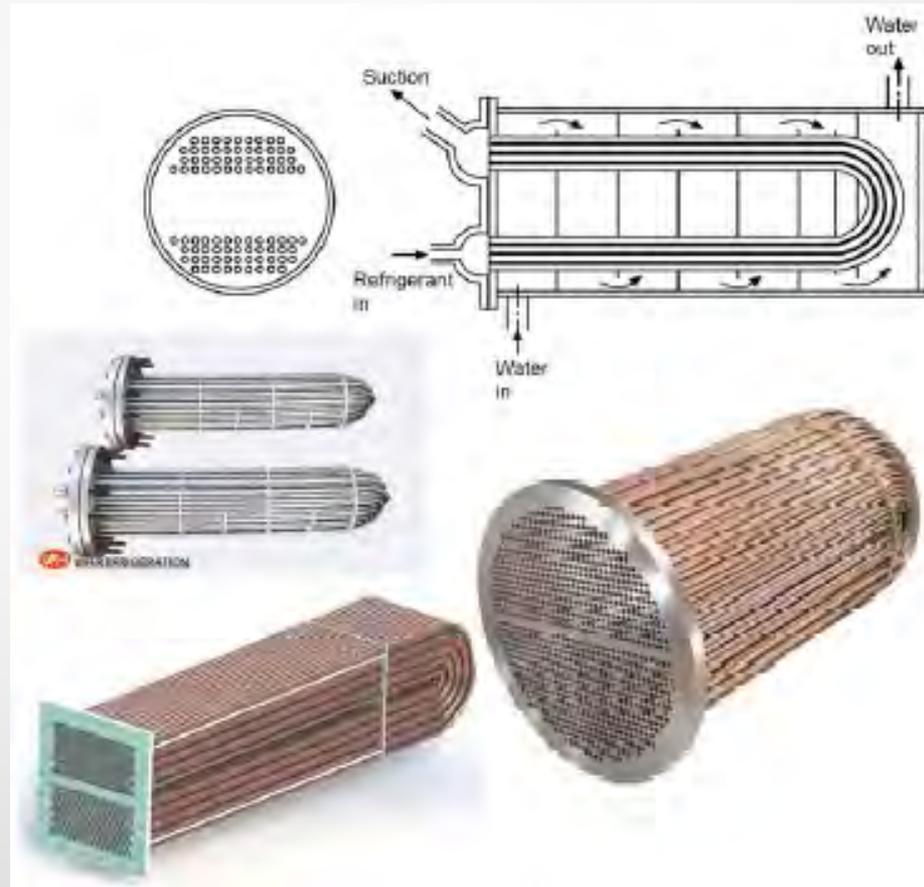
# BAUDELOT COOLER

- Baudelot Coolers are used in industrial applications for chilling a liquid to near its freezing point.
- Normally these coolers are used in systems where water is the liquid being chilled.



# SHELL AND COIL EVAPORATORS

- Commonly, the multi-pass Hx reverses the flow in the tubes by use of one or more sets of "U" bends in the tubes, which allow the fluid to flow back and forth across the length of the Hx.
- A second method to achieve multiple passes is to insert baffles on the shell side. These direct the shell side fluid back and forth across the tubes to achieve the multi-pass effect.

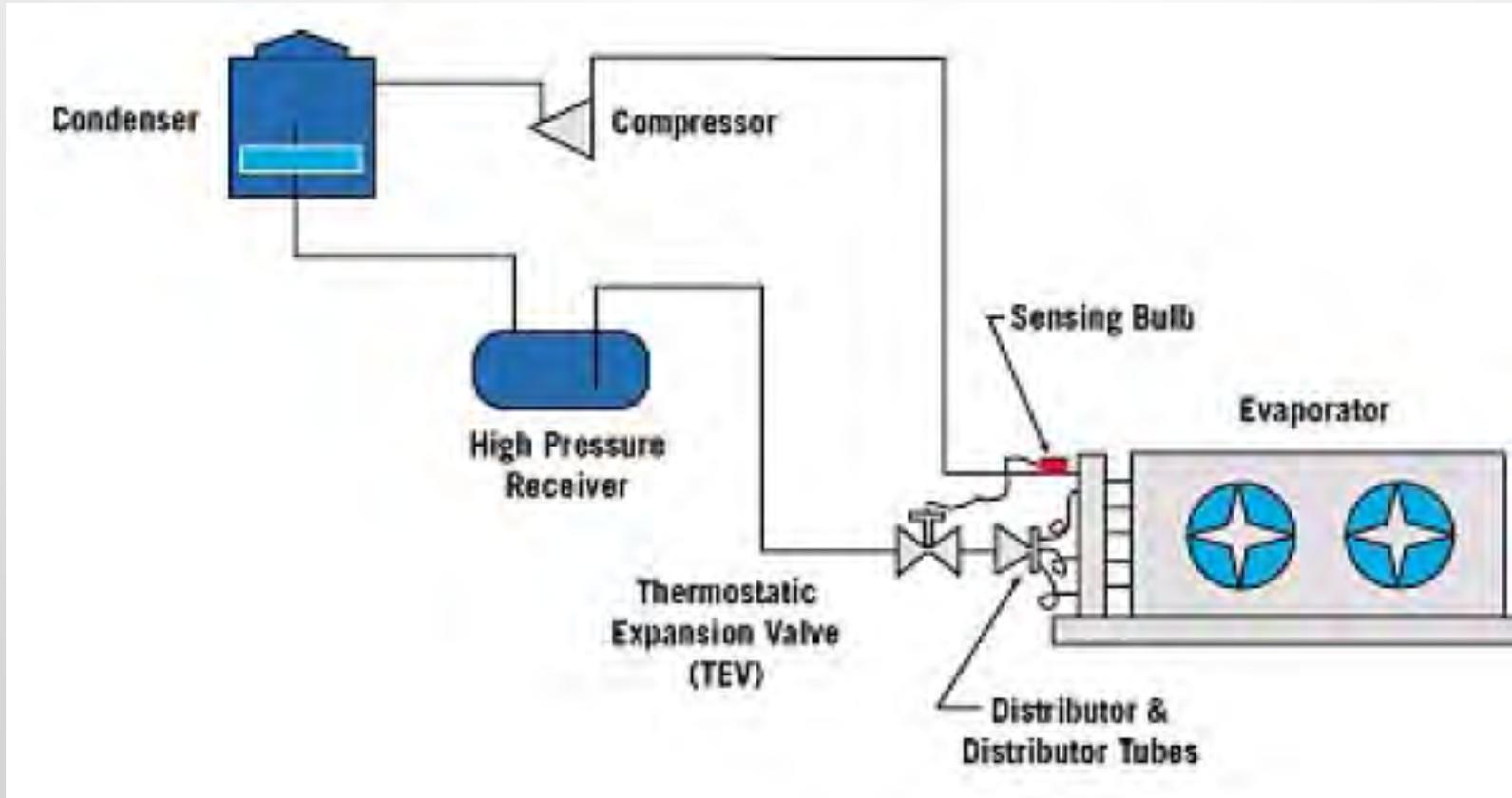




# EVAPORATORS WHICH COOL AIR

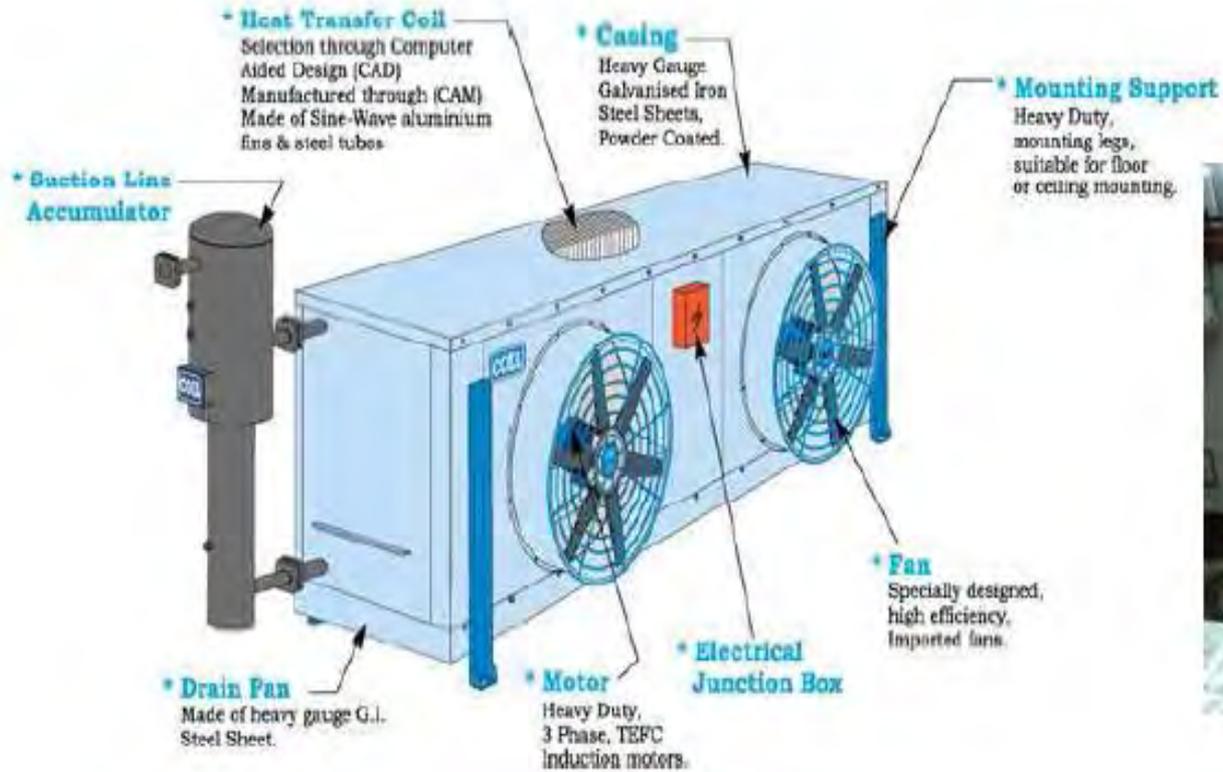
(Basics of evaporation process)

Direct air-cooled evaporators





# FLOOR MOUNTED UNITS



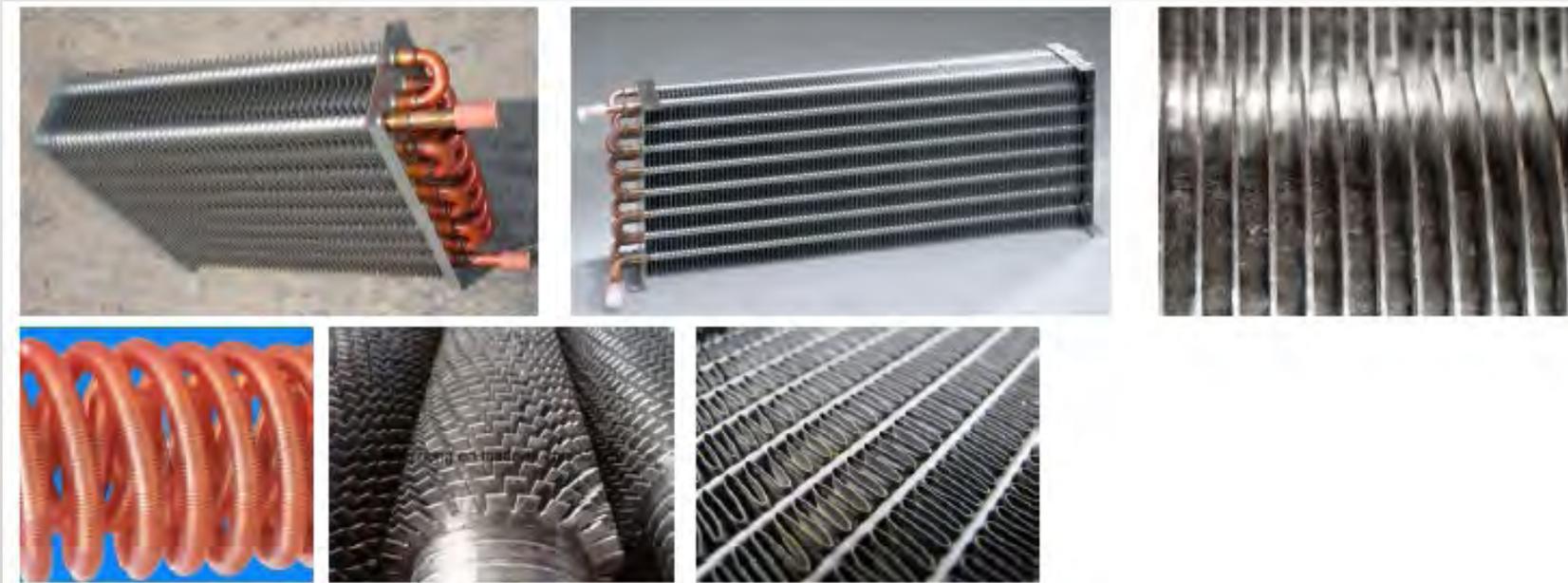


# TUBE GEOMETRY

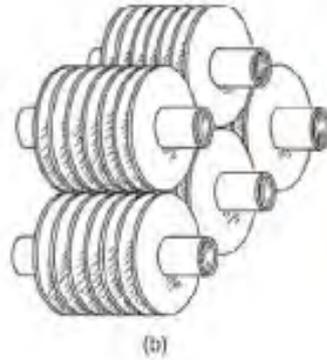
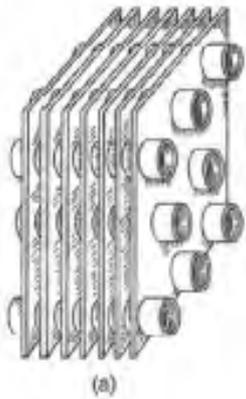
(Finned, unfinned /bare, embossed, internal and external finning)

## Finned tube evaporators

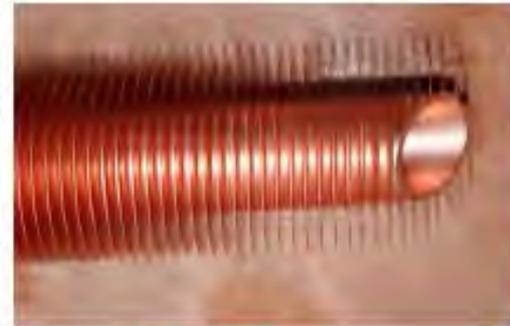
Finning improves heat transfer capacity and hence performance



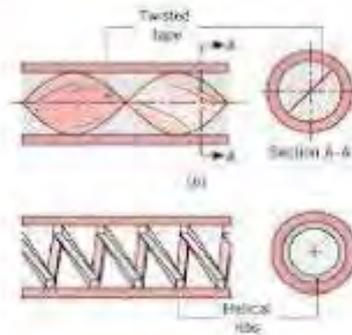
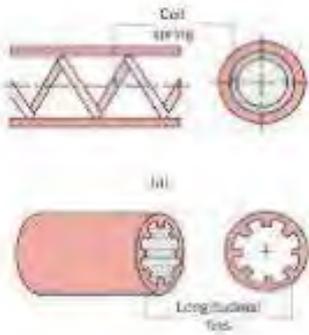
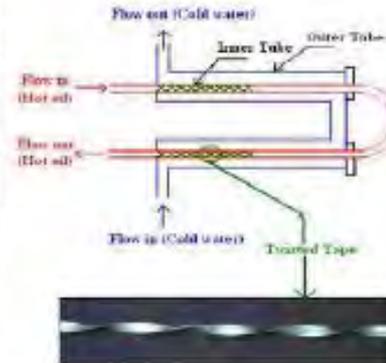
# SOME FINNING METHODS



(c)

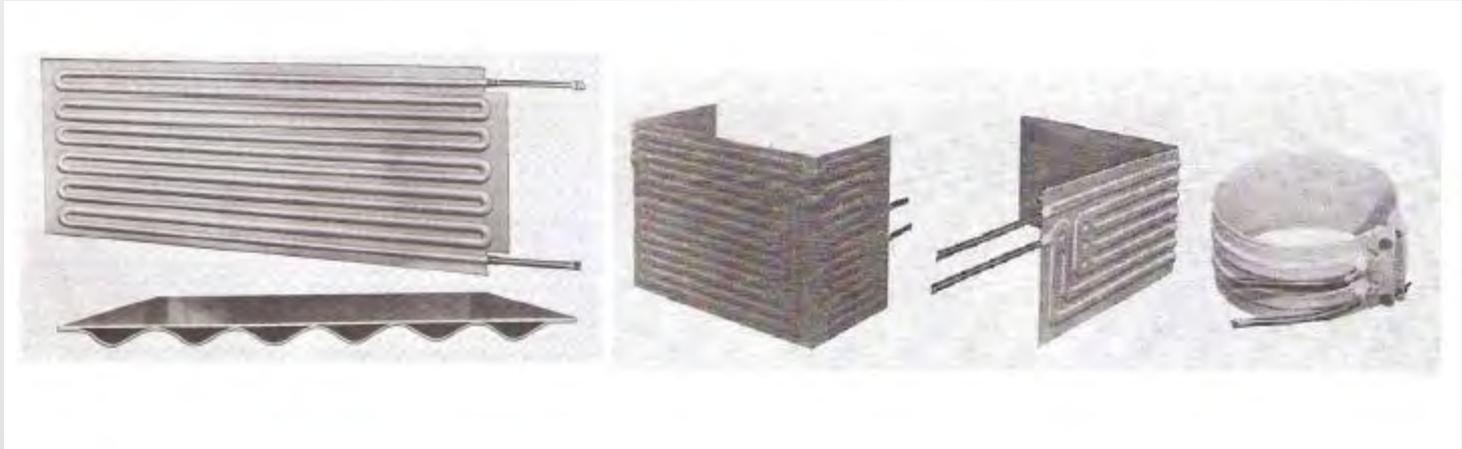


# OTHER METHODS



# BARE TUBE EVAPORATORS

These are simple and cheap



# PLATE SURFACE EVAPORATOR





## DEFROSTING

Air cooling evaporators working below  $0^{\circ}\text{C}$  will accumulate frost which must be removed periodically, since it will obstruct heat transfer.





## DEFROSTING

- Evaporators of suitable and robust construction can be defrosted by brushing, scraping or chipping.
- However, these methods are labour intensive and may lead to damage of the plant.
- Where the surrounding air is always at + 4°C or higher, it will be sufficient to stop the refrigerant for a period and allow the frost to melt off (as in the auto-defrost domestic refrigerator).
- This method can be used for cold-rooms, packaged airconditioners etc., where the service period can be interrupted.



# DEFROSTING METHODS

For lower temperatures, heat must be applied to melt the frost within a reasonable time and ensure that it drains away. Methods used are:

- **Electric resistance heaters:** Elements are within the coil or directly under it.
- **Hot gas:** A branch pipe from the compressor discharge feeds superheated gas to the coil. The compressor must still be working on another evaporator to make hot gas available. Heat storage capsules can be built into the circuit to provide a limited reserve of heat for a small installation.

**Reverse cycle:** The direction of flow of the refrigerant is reverse to make the evaporator act as a condenser. Heat storage or another evaporator are needed as a heat source. In each of these cases, arrangements must be made to remove cold refrigerant from the coil while defrosting is in progress.



# REFRIGERANT CONDENSERS

(Need, Classification, Design)

## Classification of condensers

- Like evaporators, condensers are also classified into several types:
  - ❖ Air cooled condensers
  - ❖ Water cooled condensers
  - ❖ Evaporative condensers



# AIR COOLED CONDENSERS

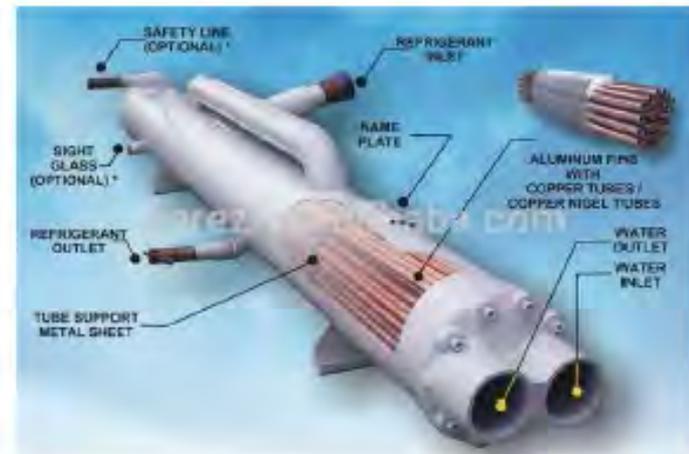
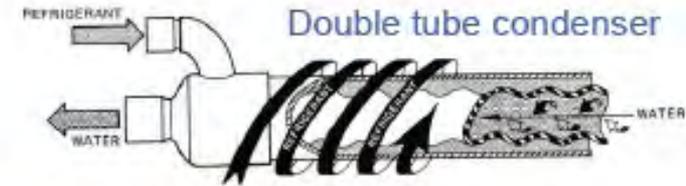
- Heat is removed by air either by natural convection or by forced convection.
- Refrigerant usually flows in metallic tubes (Cu/Al/SS etc.).
- Air velocities are usually below 5 m/s
- Typically not for large systems (> 5 TR, for example): high pressure drop, noise, increase in power consumption
- HTC is usually low and hence nearly always finned



# WATER COOLED CONDENSERS

Water cooled condensers are of three types

- Shell and Tube Condensers
- Shell and Coil Condensers
- Double tube condensers





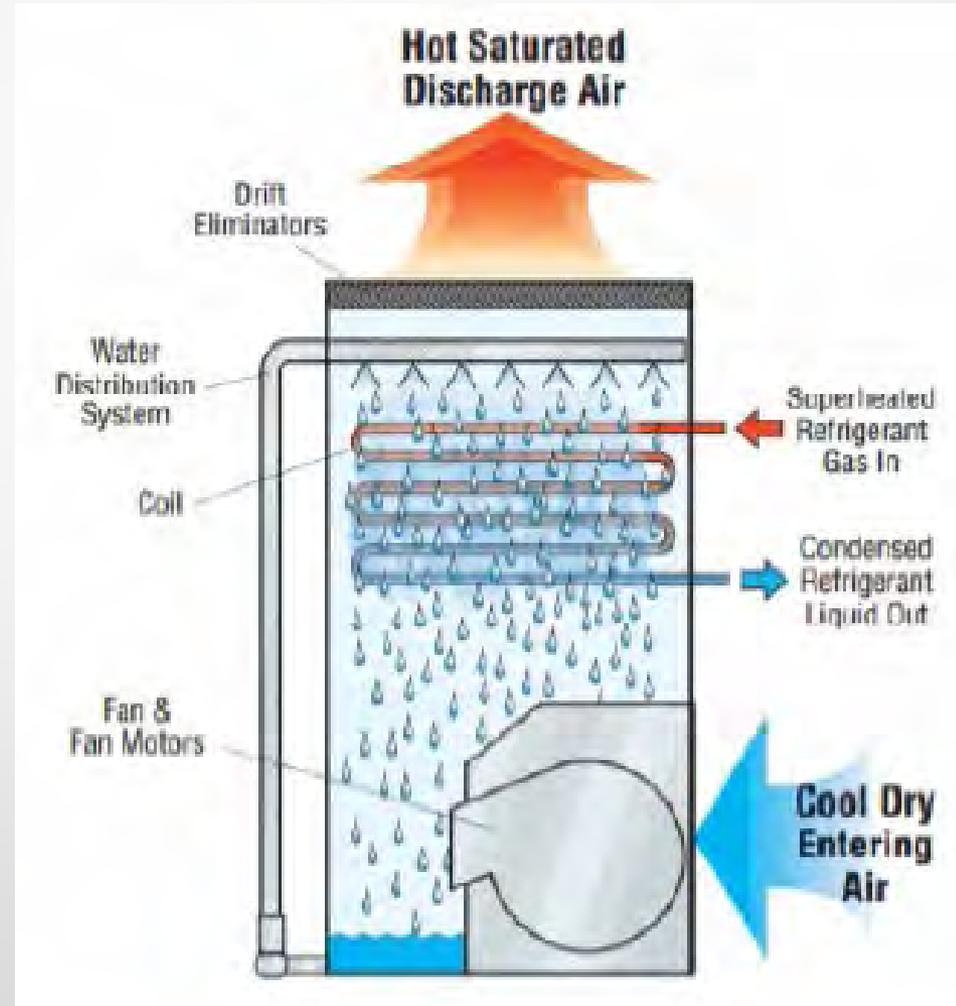
# WATER COOLED CONDENSERS

- The shell can also function as a refrigerant receiver/collector.
- Usually, some sub-cooling also takes place inside the shell.
- Water cooled condensers are used in conjunction with cooling towers.
- Water side heat transfer coefficient can be quite high.
- Cooling towers can be of two types:
  - ❖ Natural draft
  - ❖ Forced draft



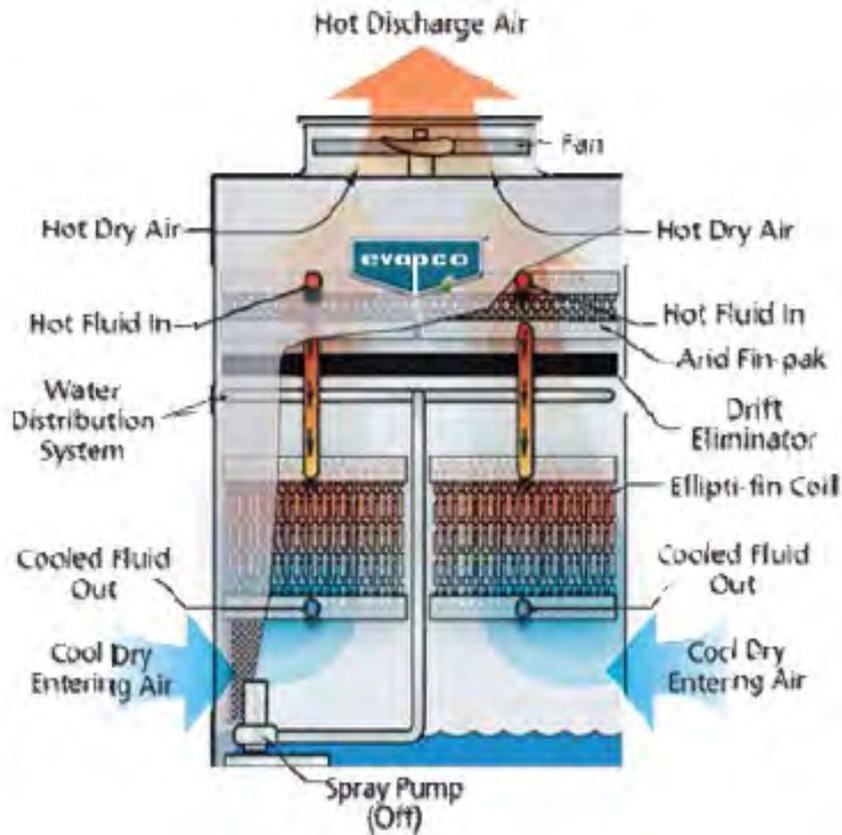
# EVAPORATIVE CONDENSERS

- Here, the refrigerant first loses its heat to water.
- The water is cooled by self evaporation, by forced air flow.
- Thus, an evaporative condenser combines the functions of condenser + cooling tower.





# EVAPORATIVE CONDENSERS



# INTRODUCTION TO REFRIGERANTS

## IN THIS LECTURE...

- Importance of selection of suitable refrigerant.
- Classification of refrigerants into primary and secondary.
- Important differences between primary and secondary refrigerants.
- Refrigerant selection criteria based on thermodynamic, thermophysical, environmental and economic properties.
- Numbering system used for designating refrigerants.
- Comparison between different refrigerants.

# INTRODUCTION

- **Refrigerant** : is the primary working fluid used for absorbing and transmitting heat in a refrigeration system.
- In principle, any fluid can be used as a refrigerant.
- Refrigerants absorb heat at low temperature and low pressure and release heat at a higher temperature and pressure
- Typically, refrigerants undergo **phase-changes** during heat absorption (evaporation) and heat releasing (condensation)
- Air used in an air-cycle refrigeration system can also be considered as a refrigerant (no phase-change).
- We will focus on those fluids that can be used as refrigerants in VCR systems only.

# INTRODUCTION

- The **thermodynamic efficiency** of a refrigeration system depends mainly on its operating temperatures.
- However, important practical issues such as the system design, size, initial and operating costs, safety, reliability, and serviceability etc. depend very much on the type of refrigerant selected for a given application.
- Due to several **environmental issues** such as ozone layer depletion, global warming and their relation to the various refrigerants used, the selection of suitable refrigerant has become one of the most important issues in recent times.
- Replacement of an existing refrigerant by a completely new refrigerant, for whatever reason, is an expensive proposition as it may call for several changes in the design and manufacturing of refrigeration systems.
- Hence, it is very important to understand the issues related to the selection and use of refrigerants.



# PRIMARY AND SECONDARY REFRIGERANTS

- *Primary refrigerants* are those fluids, which are used directly as working fluids, for example in VCR and absorption based systems.
- These fluids provide the necessary refrigeration by undergoing a phase-change process in the evaporator, absorbing latent heat.
- As the name implies, *secondary refrigerants* are those liquids, which are used for transporting thermal energy from one location to other.
- Secondary refrigerants are also known referred to as brines or antifreezes.



# PRIMARY AND SECONDARY REFRIGERANTS

- If the operating temperatures are above  $0^{\circ}\text{C}$ , then pure water can also be used as secondary refrigerant, for example in large air conditioning systems.
- Antifreezes or brines are used when refrigeration is required at sub-zero temperatures.
- Unlike primary refrigerants, secondary refrigerants do not undergo phase change as they transport energy from one location to other.



# PRIMARY AND SECONDARY REFRIGERANTS/ COOLING MEDIUM

- *Secondary refrigerants* are also called as Cooling medium.
- *Cooling Medium*: is a working fluid cooled by the refrigerant to transport the cooling effect between a central plant and remote cooling units and terminals
- *Chilled water, brine, and glycol* are used as cooling media in many refrigeration systems.
- It reduces the extensive circulation of the primary refrigerant.



# PRIMARY AND SECONDARY REFRIGERANTS

- An important property of a secondary refrigerant is its *freezing point*. Generally, the freezing point of a brine will be lower than the freezing point of its constituents (i.e. mixture components).
- The temperature at which freezing of a brine takes place depends on its concentration. The concentration at which a lowest temperature can be reached without solidification is called as *eutectic point*.
- The commonly used secondary refrigerants are the solutions of water and ethylene glycol, propylene glycol or calcium chloride. These solutions are known under the general name of *brines*.



# HISTORY OF REFRIGERATION

- 1830s - Jacob Perkins - Vapor Compression (ether)
- 1851 - John Gorrie - Patent for Air Cycle
- 1859 - R-717 / R-718 (Ammonia / Water)
- 1866 - CO<sub>2</sub> - Marine Applications
- 1873 - R-717 (Ammonia) Commercial Refrigeration
- 1875 - R-764 (Sulfur dioxide) 1920s -R-600a (Isobutane) & R-290 (Propane)
- 1922 - Willis Carrier - R-1130 (Dielene)
- 1926 - R-30 (Methylene Chloride)

## Main challenges

- Low Toxicity
- Non-flammable
- Good Stability
- Atmospheric Boiling Point between -40°C and 0°C



## COMMON REFRIGERANTS IN 1920-30

- Ammonia (R-717)  $\text{NH}_3$
- Carbon Dioxide  $\text{CO}_2$
- Sulfur Dioxide  $\text{SO}_2$
- Hydrocarbons  $\text{C}_n\text{H}_m$
- Methyl Chloride  $\text{CH}_3\text{Cl}$
- Water  $\text{H}_2\text{O}$

Use of Synthetic Refrigerants:  
(Stability, non-toxicity and efficiency)

- 1930: R11, R12
- 1936: R22
- 1961: R507

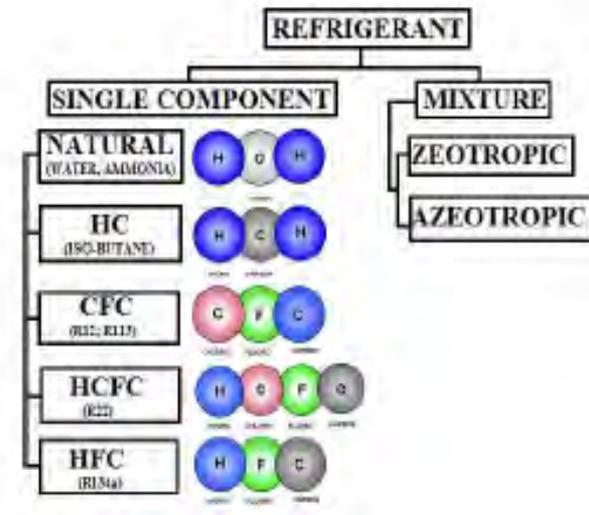
# CLASSES OF REFRIGERANTS

## Organic: Some examples

- Hydro-carbons HC (natural starting molecules from which most refrigerants are synthesized)
  - ❖ Methane (R-50) and Ethane (R-170)
- Chloro-Flouro Carbons (CFC)
- Hydro Chloro-Flouro Carbons (HCFC)
- Hydro-Flouro Carbons (HFC)

## In-organic: Some examples

- Water
- Ammonia
- CO<sub>2</sub>





# HYDROCARBON REFRIGERANTS

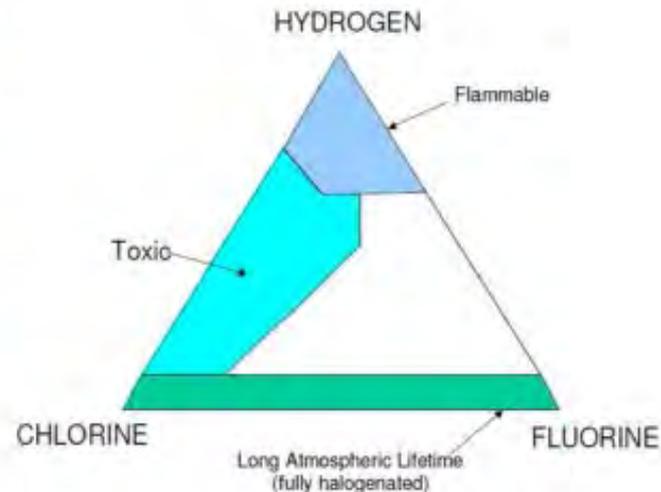
- Many hydrocarbon gases have successfully been used as refrigerants in industrial, commercial and domestic applications.
- Examples:
  - ❖ R170, Ethane,  $C_2H_6$
  - ❖ R290 , Propane  $C_3H_8$
  - ❖ R600, Butane,  $C_4H_{10}$
  - ❖ R600a, Isobutene,  $C_4H_{10}$
  - ❖ Blends of the above gases

# HALOCARBONS

- Halocarbon refrigerants are all synthetically produced and were developed as the FREON family of refrigerants.
- Examples :
  - ❖ CFC's : R11, R12, R113, R114, R115
  - ❖ HCFC's : R22, R123
  - ❖ HFC's : R134a, R404a, R407c, R410a

## PROPERTY TRENDS/ H-CL-F

- CFCs, HCFCs, HFCs & HFOs
- Limited Combinations
  - Adding Chlorine Or Bromine Increases ODP
  - Adding Fluorine Increases GWP
  - Adding Hydrogen Increases Flammability And Lowers Atmospheric Lifetime



# BLENDING OF REFRIGERANTS

- Two or more refrigerants can be suitably blended to achieve the required/desired properties
  - ❖ Flammability
  - ❖ Volumetric Capacity
  - ❖ Limit discharge superheating for lower discharge Temp
- Two Basic Types
  - ❖ Zeotropes
  - ❖ Azeotropes

# AZEOTROPIC REFRIGERANTS

- A stable mixture of two or several refrigerants whose vapor and liquid phases retain identical compositions over a wide range of temperatures.
- Examples:
  - ❖ R-500: 73.8% R12 and 26.2% R152
  - ❖ R-502: 8.8% R22 and 51.2% R115
  - ❖ R-503: 40.1% R23 and 59.9% R13



# ZEOTROPIC REFRIGERANTS

- A zeotropic mixture is one whose composition in liquid phase differs to that in vapor phase. Zeotropic refrigerants therefore do not boil at constant temperatures unlike a zeotropic refrigerants.
  
- Examples:
  - ❖ R404a : R125/143a/134a (44%,52%,4%)
  
  - ❖ R407c : R32/125/134a (23%, 25%, 52%)
  
  - ❖ R410a : R32/125 (50%, 50%)
  
  - ❖ R413a : R600a/218/134a (3%, 9%, 88%)



# PROPERTIES OF REFRIGERANTS

## *ESSENTIAL PROPERTIES OF REFRIGERANTS*

- **Latent heat of vaporization** is one of the most important properties.
- **Chemical stability** under conditions of use is another equally important characteristics.
- Safety codes may require a **non-flammable** refrigerant of low toxicity for most applications.
- **Cost, availability, efficiency, and compatibility** with compressor lubricants and materials with which the equipment is constructed.



# PHYSICAL PROPERTIES

- The refrigerants are arranged in **increasing order of atmospheric boiling point**, from air at  $-194.3^{\circ}\text{C}$  to water at  $100^{\circ}\text{C}$
- **Normal boiling point** is most important because it is a direct indicator of the temperature level at which a refrigerant can be used.



# REFRIGERANT SELECTION CRITERIA

- Selection of refrigerant for a particular application is based on the following requirements:
  - ❖ Thermodynamic and thermo-physical properties.
  - ❖ Environmental and safety properties, like:
    - ✓ Ozone depletion potential (ODP), global warming potential (GWP), and combustibility
    - ✓ Economics of production/maintenance/etc.



# THERMODYNAMIC AND THERMO-PHYSICAL PROPERTIES

- **Suction pressure:** At a given evaporator temperature, the saturation pressure should be above atmospheric for prevention of air or moisture and non-condensable gases ingress into the system and ease of leak detection. Higher suction pressure is better as it leads to smaller compressor displacement
- **Discharge pressure:** At a given condenser temperature, the discharge pressure should be as small as possible to allow lightweight construction of compressor, condenser, pipe, etc.



# THERMODYNAMIC AND THERMO-PHYSICAL PROPERTIES

- *Pressure ratio or Compression ratio*: Should be as small as possible for high volumetric efficiency and low power consumption
- *Latent heat of vaporization*: Should be as large as possible so that the required mass flow rate per unit cooling capacity will be small

$$\ln(P_{\text{sat}}) = -\frac{h_{fg}}{RT} + \frac{s_{fg}}{R}$$

$$\frac{P_c}{P_e} = \exp\left[\frac{h_{fg}}{R}\left(\frac{1}{T_e} - \frac{1}{T_c}\right)\right]$$

- From the above equation, it can be seen that for given condenser and evaporator temperatures, as the latent heat of vaporization increases, the pressure ratio also increases.
- Hence a **trade-off is required** between the latent heat of vaporization and pressure ratio.



# THERMODYNAMIC AND THERMO-PHYSICAL PROPERTIES

- **Isentropic index of compression:** Should be as small as possible so that the temperature rise during compression will be small.
- **Liquid specific heat:** Should be small so that degree of sub cooling will be large leading to smaller amount of flash at evaporator inlet.
- **Vapor specific heat:** Should be large so that the degree of superheating will be small.
- **Thermal conductivity:** Thermal conductivity in both liquid as well as vapor phase should be high for higher heat transfer coefficients.
- **Viscosity:** Viscosity should be small in both liquid and vapor phases for smaller frictional pressure drops.



# ENVIRONMENTAL AND SAFETY PROPERTIES

- In contemporary times, **environment friendliness** of the refrigerant is a major factor in deciding its usage.
- **Ozone Depletion Potential (ODP)**: According to the Montreal protocol, the ODP of refrigerants should be zero, i.e., they should be non-ozone depleting substances.
- Refrigerants having non-zero ODP have either already been phased-out (e.g. R11, R12) or will be in near-future (e.g. R22).
- Since ODP depends mainly on the presence of chlorine or bromine in the molecules, refrigerants having either chlorine (i.e., CFCs and HCFCs) or bromine cannot be used under the new regulations.

# ENVIRONMENTAL AND SAFETY PROPERTIES

- **Global Warming Potential (GWP):** Refrigerants should have as low a GWP value as possible to minimize global warming.
- Refrigerants with zero ODP but a high value of GWP (e.g. R134a) are likely to be regulated in future.
- **Total Equivalent Warming Index (TEWI):** The factor TEWI considers both direct (due to release into atmosphere) and indirect (through energy consumption) contributions of refrigerants to global warming.
- Naturally, refrigerants with as a low a value of TEWI are preferable from global warming point of view.

# ENVIRONMENTAL AND SAFETY PROPERTIES

- **Toxicity:** Ideally, refrigerants used in a refrigeration system should be non-toxic. However, all fluids other than air can be called as toxic as they will cause suffocation when their concentration is large enough.
- Hence, toxicity is a relative term, which becomes meaningful only when the degree of concentration and time of exposure required to produce harmful effects are specified.
- Some fluids are toxic even in small concentrations. Some fluids are mildly toxic, i.e., they are dangerous only when the concentration is large and duration of exposure is long.

# ENVIRONMENTAL AND SAFETY PROPERTIES

- Some refrigerants such as CFCs and HCFCs are non-toxic when mixed with air in normal condition. However, when they come in contact with an open flame or an electrical heating element, they decompose forming highly toxic elements (e.g. phosgene-COCl<sub>2</sub>).
- In general the degree of hazard depends on:
  - ❖ Amount of refrigerant used vs total space
  - ❖ Type of occupancy
  - ❖ Presence of open flames
  - ❖ Odor of refrigerant, and
  - ❖ Maintenance condition



# ENVIRONMENTAL AND SAFETY PROPERTIES

- **Flammability:** The refrigerants should preferably be nonflammable and non-explosive. For flammable refrigerants special precautions should be taken to avoid accidents. Based on the above criteria, ASHRAE has divided refrigerants into **six safety groups** (A1 to A3 and B1 to B3).
- Refrigerants belonging to Group A1 (e.g. R11, R12, R22, R134a, R744, R718) are least hazardous, while refrigerants belonging to Group B3 (e.g. R1140) are most hazardous.
- Finally, **economic properties:** The refrigerant used should preferably be inexpensive and easily available.



# COMPARISON/ ISSUES

Refrigerant Type	Example(s)	Remarks
HCs	HC-290 (Propane), CH <sub>4</sub> (Methane)	Not typically used in commercial A/C products, flammable.
CFCs	CFC-11, 12, 113, 114, 115	Contains ozone depleting chlorine, most harmful, phased out in '95, High GWP.
HCFCs	HCFC-22, HCFC-123	Contains ozone depleting chlorine, Short term replacements, phased out in '10 from new equipment, High GWP.
HFCs	HFC-134a, HFC-407C, HFC-410A,	Contains ZERO ozone depleting chlorine, ZERO ODP. Long term replacements, High GWP.



# NOMENCLATURE OF REFRIGERANTS

## DESIGNATION OF REFRIGERANTS/ NOMENCLATURE

- Since a large number of refrigerants have been developed over the years for a wide variety of applications, a numbering system has been adopted to designate various refrigerants.
- From the number one can get some useful information about the type of refrigerant, its chemical composition, molecular weight etc.
- All the refrigerants are designated by R followed by a unique number.



# ASHRAE STANDARDS (NUMBER 34)

- The number assigned to each refrigerant is related to its chemical composition and the system has been formalized as ASHRAE Standard 34. Broadly, the numbering system is as follows:
  - ❖ 000 Series Methane Based
  - ❖ 100 Series Ethane Based
  - ❖ 200 Series Propane Based
  - ❖ 300 Series Cyclic Organic Compounds
  - ❖ 400 Series Zeotropes
  - ❖ 500 Series Azeotropes
  - ❖ 600 Series Organic Compounds
  - ❖ 700 Series Inorganic Compounds
  - ❖ 1000 Series Unsaturated Organic Compounds



# DESIGNATION OF REFRIGERANTS/ NOMENCLATURE

## 1. Fully saturated, halogenated compounds:

➤ These refrigerants are derivatives of alkanes ( $C_nH_{2n+2}$ ) such as methane ( $CH_4$ ), ethane ( $C_2H_6$ ) and propane ( $C_3H_8$ ).

➤ These refrigerants are designated by R-XYZ, where:

❖ X+1 indicates the number of Carbon (C) atoms

❖ Y-1 indicates number of Hydrogen (H) atoms, and

❖ Z indicates number of Fluorine (F) atoms

❖ The balance indicates the number of Chlorine atoms.

➤ Only 2 digits indicates that the value of X is zero.



## NOMENCLATURE: EXAMPLE

### Example: R 22

- $X = 0 \Rightarrow$  No. of Carbon atoms =  $0 + 1 = 1 \Rightarrow$  derivative of methane ( $\text{CH}_4$ )
- $Y = 2 \Rightarrow$  No. of Hydrogen atoms =  $2 - 1 = 1$
- $Z = 2 \Rightarrow$  No. of Fluorine atoms = 2
- The balance =  $4 - \text{no. of (H+F) atoms} = 4 - 1 - 2 = 1 \Rightarrow$  No. of Chlorine atoms = 1

$\therefore$  The chemical formula of R 22 =  $\text{CHClF}_2$

☉ Similarly it can be shown that the chemical formula of:

- R12 =  $\text{CCl}_2\text{F}_2$
- R134a =  $\text{C}_2\text{H}_2\text{F}_4$  (derivative of ethane)

☉ (letter **a** stands for isomer, e.g. molecules having same chemical composition but different atomic arrangement, e.g. R134 and R134a)

# ISOMERIC REFRIGERANTS

## The Two Isomers of Tetrafluoroethane

1, 1, 2, 2-Tetrafluoroethane

R-134



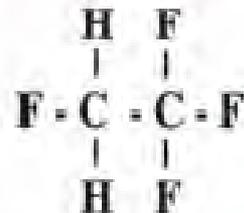
$\text{CHF}_2\text{-CHF}_2$

Attached mass delta

$$39 - 39 = 0$$

1, 1, 1, 2-Tetrafluoroethane

R-134a



$\text{CH}_2\text{F-CF}_3$

Attached mass delta

$$57 - 21 = 36$$

### Chemical name... Tetrafluoroethane

ANSI/ ASHRAE	Chemical Structural Name	Structural Molecular Formula
R-134	1, 1, 2, 2-Tetrafluoroethane	$\text{CHF}_2\text{-CHF}_2$
R-134a	1, 1, 1, 2-Tetrafluoroethane	$\text{CH}_2\text{F-CF}_3$

Molecular

Weight

C... 12.0

H... 1.0

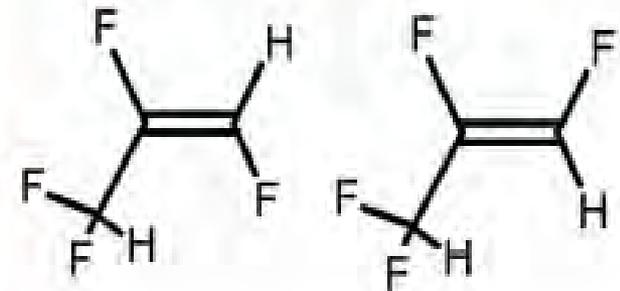
F... 19.0

F... 19.0

2,3,3,3-Tetrafluoropropene, HFO-1234yf, is a hydro-fluoro-olefin (HFO) with the formula  $\text{CH}_2=\text{CFCF}_3$ .

This molecule can have several isomers

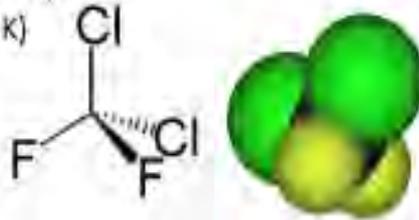
**NEW CLASS OF REFRIGERANT**



# EXAMPLES

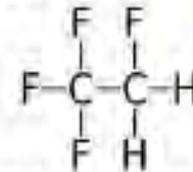
## R12

- **R12 (1bar = -30°C)**
- Molecular formula  $\text{CCl}_2\text{F}_2$
- Molar mass 120.91 g/mol
- Appearance – colorless gas with ether-like odor
- Density 1.486 g/cm<sup>3</sup> (-29.8 °C)
- Melting point -157.7 °C (115.5 K)
- Boiling point -29.8 °C (243.3 K)
  
- ODP = 0.82
- GWP = 8100



## R134a

- **R134a (1bar = -26°C)**
- Molecular formula  $\text{C}_2\text{H}_2\text{F}_4$
- Molar mass 102.03 g/mol
- Appearance Colorless gas
- Density 4.25 kg/m<sup>3</sup>, gas
- Melting point -103.3°C (169.85 K)
- Boiling point -26.3°C (246.85 K)
  
- ODP = 0
- GWP = 1430
  
- Replacement for R11 & R12 (not a drop-in)



# NOMENCLATURE

## 2. Inorganic refrigerants:

These are designated by number 7 followed by the molecular weight of the refrigerant (rounded-off).

- Ex.: Ammonia: Molecular weight is 17, ∴ the designation is R-717
- Carbon dioxide: Molecular weight is 44, ∴ the designation is R-744
- Water: Molecular weight is 18, ∴ the designation is R-718
- ❖ Sulphur dioxide
- ❖ Air

# COMPLETE NOMENCLATURE EXPLAINED

- The first digit on the right is the number of fluorine (F) atoms.
- The second digit from the right is one more than the number of hydrogen (H) atoms.
- The third digit from the right is one less than the number of carbon (C) atoms. When this digit is zero it is omitted from the number.
- The fourth digit from the right is equal to the number of unsaturated carbon-carbon bonds in the compound. When this digit is zero, it is omitted from the number.

# COMPLETE NOMENCLATURE EXPLAINED

- The number of chlorine (Cl) atoms is found by subtracting the sum of the fluorine (F), bromine (Br - if present) and hydrogen (H) atoms from the total number that can be attached to carbon. With one carbon, the total number of attached atoms is four. With two carbons, the total number is six and with three carbons, the total number is eight.
- In some cases, part or all of the chlorine atoms are replaced with bromine and the letter B is used in the number. The number following the letter B shows the number of bromine atoms present.

# COMPLETE NOMENCLATURE EXPLAINED

- Blends are designated by their respective refrigerant numbers and weight proportions. rest of text okay, add this. To differentiate among blends having same components with different proportions, an uppercase letter shall be added to designation. For example, R-402A and R-402B.
- Zeotropic blends that have been commercialized shall be assigned an identifying number in the 400 series. This number designates which components are in the mixture but not the amount of each. The amount of each component is designated as described in 7 above. For example the 60/ 40 weight % mixture of R-12 and R-114 would be R-400 (60/40).

# COMPLETE NOMENCLATURE EXPLAINED

- Azeotropic compounds that have been commercialized shall be assigned an identifying number in the 500 series. It is not necessary to cite the percentages parenthetically once a 500 series is assigned.
- The 600 series has been assigned to miscellaneous organic compounds. Within the organic 600 series, the assignments are serial.
- The 700 series has been assigned to inorganic compounds with relative molecular masses less than 100. Within the 700 series, the relative molecular mass of the compounds is added to 700 to arrive at the identifying refrigerant numbers. When two or more inorganic refrigerants have the same relative molecular masses, uppercase letters (i.e., A, B, C, etc.) are added to distinguish among them.
- The 1000 series has been assigned to unsaturated organic compounds.



## THE OBJECTIVES OF THIS LESSON ARE TO:

- Introduce vapour absorption refrigeration systems
- Explain the basic principle of a vapour absorption refrigeration system
- Compare vapour compression refrigeration systems with continuous vapour absorption refrigeration systems
- Obtain expression for maximum COP of ideal absorption refrigeration system
- Discuss properties of ideal and real refrigerant-absorbent mixtures
- Describe a single stage vapour absorption refrigeration system with solution heat exchanger
- Discuss the desirable properties of refrigerant-absorbent pairs for vapour absorption refrigeration systems and list the commonly used working fluids



## INTRODUCTION

Vapour Absorption Refrigeration Systems (VARs) belong to the class of vapour cycles similar to vapour compression refrigeration systems. However, unlike vapour compression refrigeration systems, the required input to absorption systems is in the form of heat. Hence these systems are also called as heat operated or thermal energy driven systems. Since conventional absorption systems use liquids for absorption of refrigerant, these are also sometimes called as wet absorption systems. Similar to vapour compression refrigeration systems, vapour absorption refrigeration systems have also been commercialized and are widely used in various refrigeration and air conditioning applications. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available. Since conventional absorption systems use natural refrigerants such as water or ammonia they are environment friendly.



## BASIC PRINCIPLE

When a solute such as lithium bromide salt is dissolved in a solvent such as water, the boiling point of the solvent (water) is elevated. On the other hand, if the temperature of the solution (solvent + solute) is held constant, then the effect of dissolving the solute is to reduce the vapour pressure of the solvent below that of the saturation pressure of pure solvent at that temperature. If the solute itself has some vapour pressure (i.e., volatile solute) then the total pressure exerted over the solution is the sum total of the partial pressures of solute and solvent. If the solute is non-volatile (e.g. lithium bromide salt) or if the boiling point difference between the solution and solvent is large ( $\geq 300^{\circ}\text{C}$ ), then the total pressure exerted over the solution will be almost equal to the vapour pressure of the solvent only. In the simplest absorption refrigeration system, refrigeration is obtained by connecting two vessels, with one vessel containing pure solvent and the other containing a solution. Since the pressure is almost equal in both the vessels at equilibrium, the temperature of the solution will be higher than that of the pure solvent. This means that if the solution is at ambient temperature, then the pure solvent will be at a temperature lower than the ambient. Hence refrigeration effect is produced at the vessel containing pure solvent due to this temperature difference. The solvent evaporates due to heat transfer from the surroundings, flows to the vessel containing solution and is absorbed by the solution. This process is continued as long as the composition and temperature of the solution are maintained and liquid solvent is available in the container.



## BASIC PRINCIPLE

For example, Fig shows an arrangement, which consists of two vessels A and B connected to each other through a connecting pipe and a valve. Vessel A is filled with pure water, while vessel B is filled with a solution containing on mass basis 50 percent of water and 50 percent lithium bromide (LiBr salt). Initially the valve connecting these two vessels is closed, and both vessels are at thermal equilibrium with the surroundings, which is at 30°C. At 30°C, the saturation pressure of water is 4.24 kPa, and the equilibrium vapour pressure of water-lithium bromide solution (50 : 50 by mass) at 30°C is 1.22 kPa. Thus at initial equilibrium condition, the pressure in vessel A is 4.24 kPa, while it is 1.22 kPa in vessel B. Now the valve between vessels A and B is opened. Initially due to pressure difference water vapour will flow from vessel A to vessel B, and this vapour will be absorbed by the solution in vessel B. Since absorption in this case is exothermic, heat will be released in vessel B. Now suppose by some means the concentration and temperature of vessel B are maintained constant at 50 % and 30°C, respectively. Then at equilibrium, the pressure in the entire system (vessels A and B) will be 1.22 kPa (equilibrium pressure of 50 % LiBr solution at 30°C).



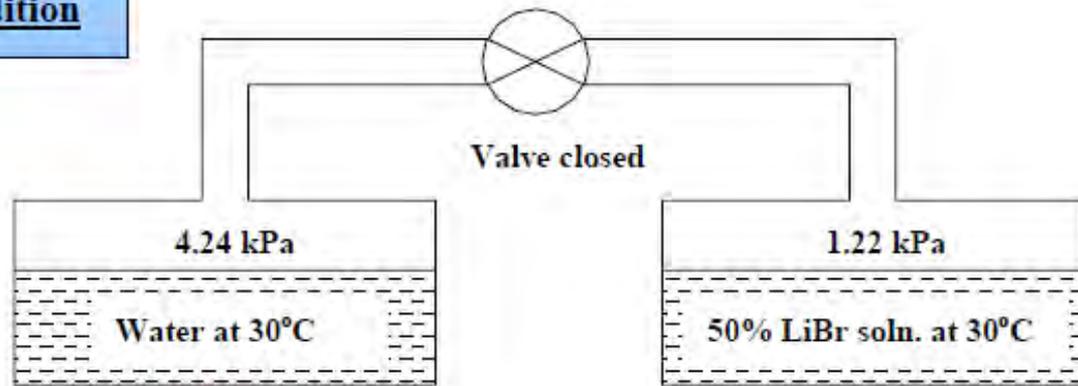
## BASIC PRINCIPLE

The temperature of water in vessel A will be the saturation temperature corresponding to 1.22 kPa, which is equal to about  $10^{\circ}\text{C}$ , as shown in the figure. Since the water temperature in A is lower than the surroundings, a refrigeration effect (Q) can be produced by transferring heat from the surroundings to water at  $10^{\circ}\text{C}$ . Due to this heat transfer, water vaporizes in A, flows to B and is absorbed by the solution in B. The exothermic heat of absorption ( $Q_a$ ) is rejected to the surroundings.



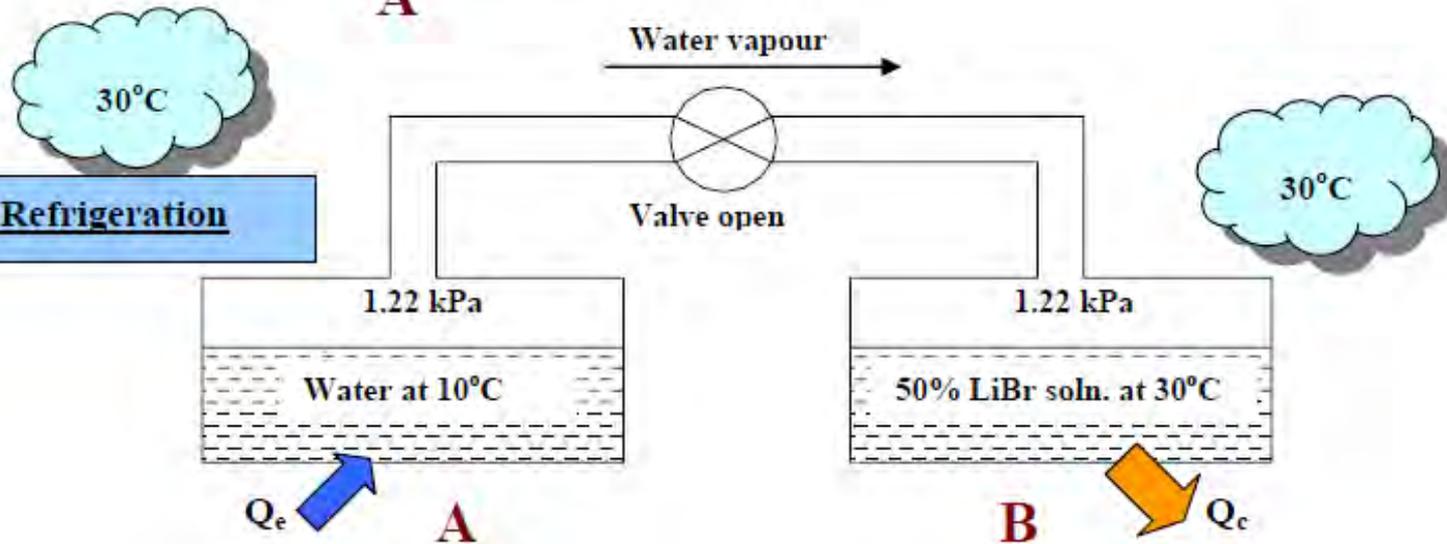
# BASIC PRINCIPLE

## a) Initial condition



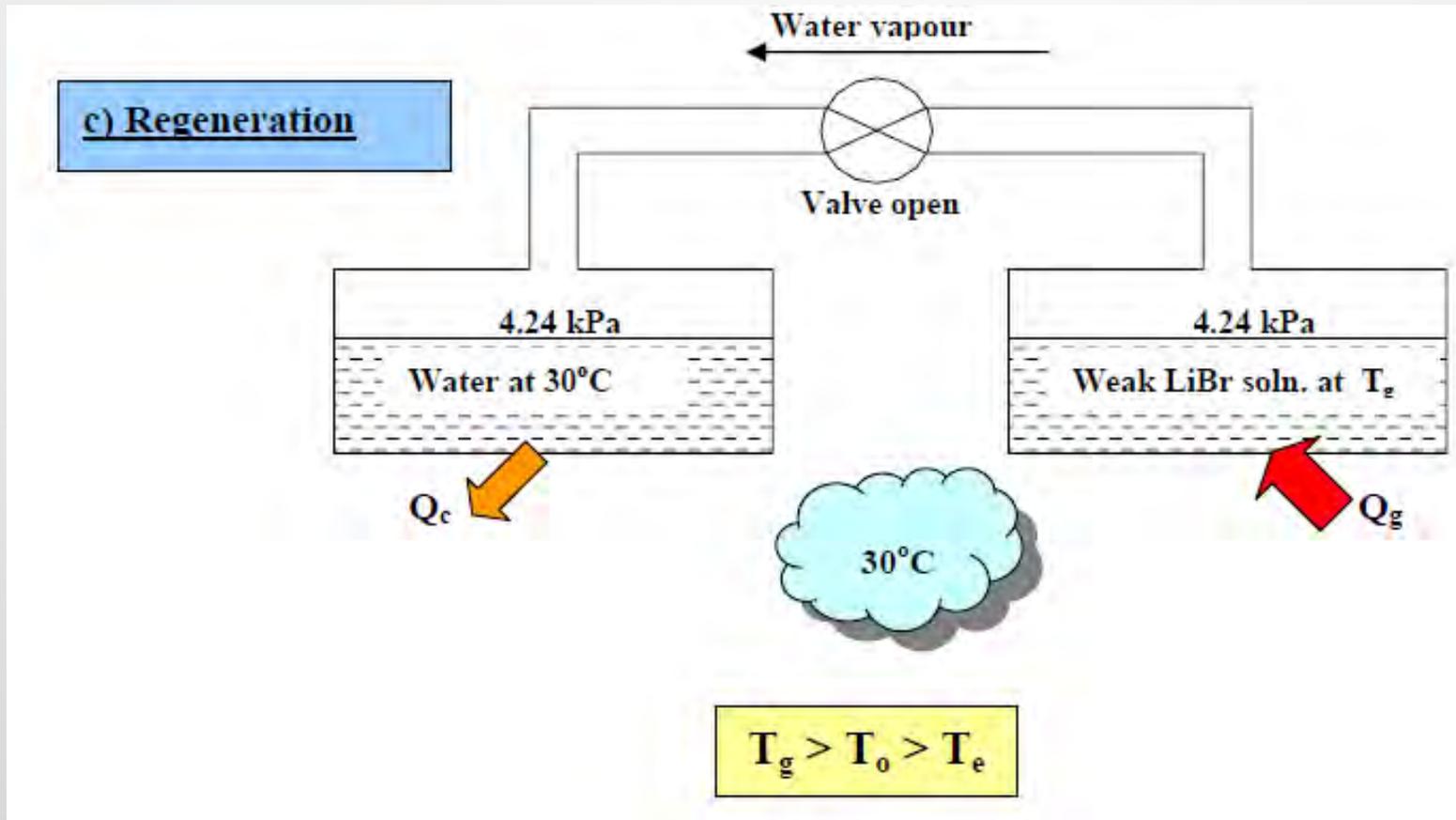
A

## b) Refrigeration

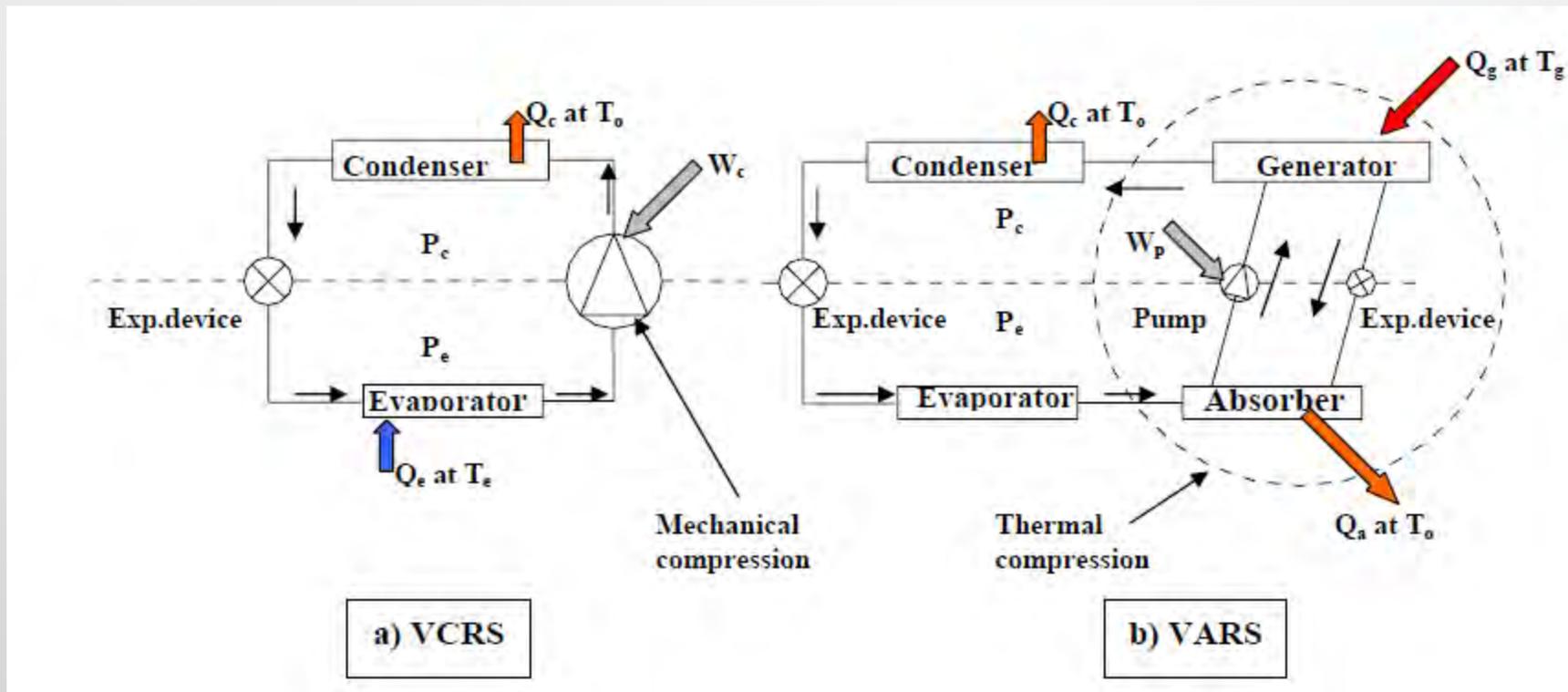




# BASIC PRINCIPLE



# COMPARISON BETWEEN VAPOUR COMPRESSION REFRIGERATION AND VAPOUR ABSORPTION SYSTEM



- a) Vapour compression refrigeration system (VCRS)
- b) Vapour Absorption Refrigeration System (VARS)

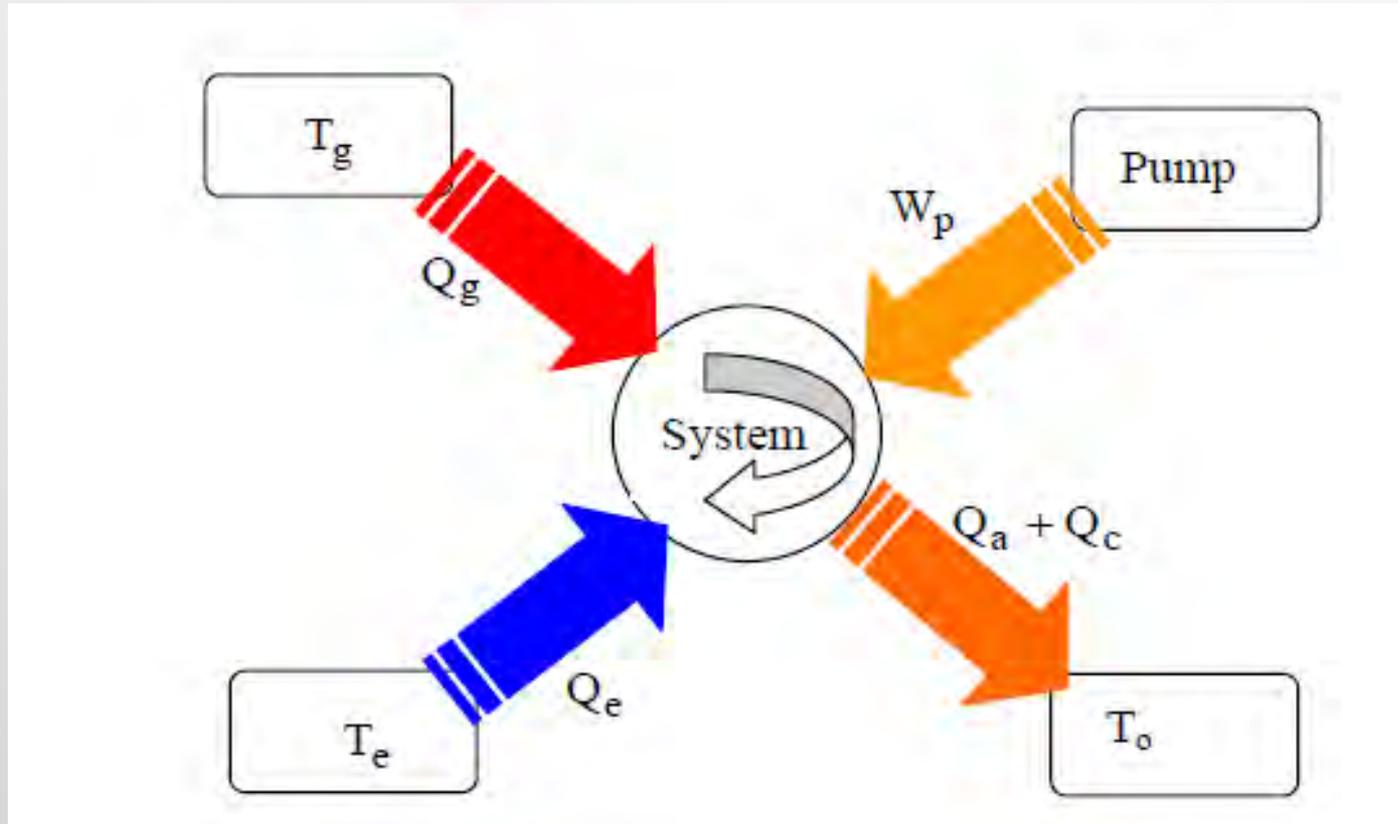
# MAXIMUM COP OF IDEAL ABSORPTION REFRIGERATION SYSTEM

In case of a single stage compression refrigeration system operating between constant evaporator and condenser temperatures, the maximum possible COP is given by Carnot COP:

$$\text{COP}_{\text{Carnot}} = \frac{T_e}{T_c - T_e}$$

If we assume that heat rejection at the absorber and condenser takes place at same external heat sink temperature  $T_o$ , then a vapour absorption refrigeration system operates between three temperature levels,  $T_g$ ,  $T_o$  and  $T_e$ . The maximum possible COP of a refrigeration system operating between three temperature levels can be obtained by applying first and second laws of thermodynamics to the system.

*Various energy transfers in a vapour absorption refrigeration system*



## *Various energy transfers in a vapour absorption refrigeration system*

From first law of thermodynamics,

$$Q_e + Q_g - Q_{c+a} + W_p = 0$$

where  $Q_e$  is the heat transferred to the absorption system at evaporator temperature  $T_e$ ,  $Q_g$  is the heat transferred to the generator of the absorption system at temperature  $T_g$ ,  $Q_{a+c}$  is the heat transferred from the absorber and condenser of the absorption system at temperature  $T_o$  and  $W_p$  is the work input to the solution pump.

From second law of thermodynamics,

$$\Delta S_{\text{total}} = \Delta S_{\text{sys}} + \Delta S_{\text{surr}} \geq 0$$

where  $\Delta S_{\text{total}}$  is the total entropy change which is equal to the sum of entropy change of the system  $\Delta S_{\text{sys}}$  and entropy change of the surroundings  $\Delta S_{\text{surr}}$ . Since the refrigeration system operates in a closed cycle, the entropy change of the working fluid of the system undergoing the cycle is zero, i.e.,  $\Delta S_{\text{sys}} = 0$ . The entropy change of the surroundings is given by:

$$\Delta S_{\text{surr}} = -\frac{Q_e}{T_e} - \frac{Q_g}{T_g} + \frac{Q_{a+c}}{T_o} \geq 0$$

Substituting the expression for first law of thermodynamics in the above equation

## *Various energy transfers in a vapour absorption refrigeration system*

$$Q_g \left( \frac{T_g - T_o}{T_g} \right) \geq Q_e \left( \frac{T_o - T_e}{T_e} \right) - W_p$$

Neglecting solution pump work,  $W_p$ ; the COP of VARS is given by:

$$\text{COP}_{\text{VARS}} = \frac{Q_e}{Q_g} \leq \left( \frac{T_e}{T_o - T_e} \right) \left( \frac{T_g - T_o}{T_g} \right)$$

## *Various energy transfers in a vapour absorption refrigeration system*

An ideal vapour absorption refrigeration system is totally reversible (i.e., both internally and externally reversible). For a completely reversible system the total entropy change (system+surroundings) is zero according to second law, hence for an ideal VARS  $\Delta S_{\text{total,rev}} = 0 \Rightarrow \Delta S_{\text{surr,rev}} = 0$ . Hence:

$$\Delta S_{\text{surr,rev}} = -\frac{Q_e}{T_e} - \frac{Q_g}{T_g} + \frac{Q_{a+c}}{T_o} = 0$$

Hence combining first and second laws and neglecting pump work, the maximum possible COP of an ideal VARS system is given by:

$$\text{COP}_{\text{ideal VARS}} = \frac{Q_e}{Q_g} = \left( \frac{T_e}{T_o - T_e} \right) \left( \frac{T_g - T_o}{T_g} \right)$$

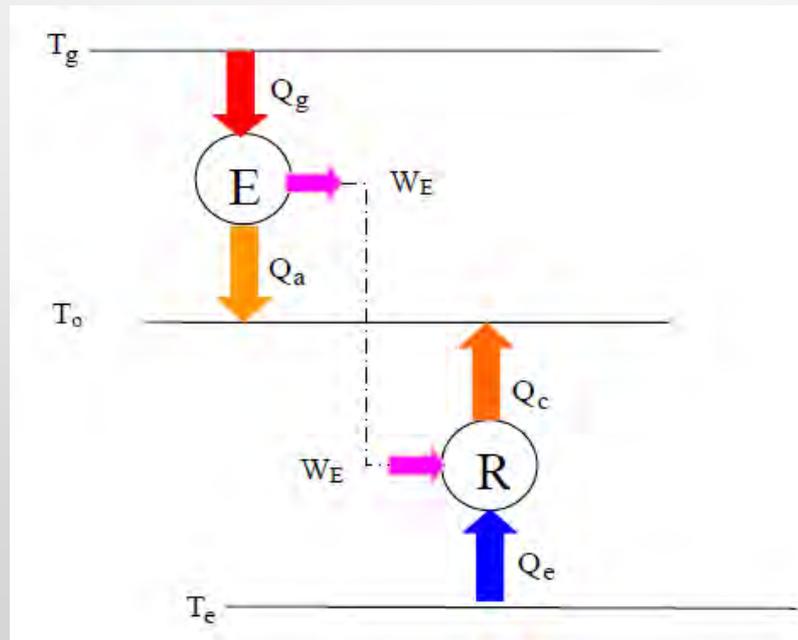
Thus the ideal COP is only a function of operating temperatures similar to Carnot system. It can be seen from the above expression that the ideal COP of VARS system is equal to the product of efficiency of a Carnot heat engine operating between  $T_g$  and  $T_o$  and COP of a Carnot refrigeration system operating between  $T_o$  and  $T_e$ , i.e.,

$$\text{COP}_{\text{ideal VARS}} = \frac{Q_e}{Q_g} = \left( \frac{T_e}{T_o - T_e} \right) \left( \frac{T_g - T_o}{T_g} \right) = \text{COP}_{\text{Carnot}} \cdot \eta_{\text{Carnot}}$$



## VAPOUR ABSORPTION SYSTEM AS A COMBINATION OF HEAT ENGINE AND REFRIGERATOR

an ideal vapour absorption refrigeration system can be considered to be a combined system consisting of a Carnot heat engine and a Carnot refrigerator as shown in Fig. Thus the COP of an ideal VARS increases as generator temperature ( $T_g$ ) and evaporator temperature ( $T_e$ ) increase and heat rejection temperature ( $T_o$ ) decreases. However, the COP of actual VARS will be much less than that of an ideal VARS due to various internal and external irreversibilities present in actual systems.





# PROPERTIES OF REFRIGERANT-ABSORBENT MIXTURES

The solution used in absorption refrigeration systems may be considered as a homogeneous binary mixture of refrigerant and absorbent. Depending upon the boiling point difference between refrigerant and absorbent and the operating temperatures, one may encounter a pure refrigerant vapour or a mixture of refrigerant and absorbent vapour in generator of the absorption system. Unlike pure substances, the thermodynamic state of a binary mixture (in liquid or vapour phase) cannot be fixed by pressure and temperature alone. According to Gibbs' phase rule, one more parameter in addition to temperature and pressure is required to completely fix the thermodynamic state. Generally, the composition of the mixture is taken as the third independent parameter. The composition of a mixture can be expressed either in mass fraction or in mole fraction. The mass fraction of components 1 and 2 in a binary mixture are given by:



# PROPERTIES OF REFRIGERANT-ABSORBENT MIXTURES

$$\xi_1 = \frac{m_1}{m_1 + m_2}; \quad \xi_2 = \frac{m_2}{m_1 + m_2}$$

where  $m_1$  and  $m_2$  are the mass of components 1 and 2, respectively

The mole fraction of components 1 and 2 in a binary mixture are given by:

$$x_1 = \frac{n_1}{n_1 + n_2}; \quad x_2 = \frac{n_2}{n_1 + n_2}$$

where  $n_1$  and  $n_2$  are the number of moles of components 1 and 2, respectively

An important property of a mixture is its miscibility. A mixture is said to be completely miscible if a homogeneous mixture can be formed through any arbitrary range of concentration values. Miscibility of mixtures is influenced by the temperature at which they are mixed. Some mixtures are miscible under certain conditions and immiscible at other conditions. The refrigerant-absorbent mixtures used in absorption refrigeration systems must be completely miscible under all conditions both in liquid and vapour phases.



# IDEAL, HOMOGENEOUS BINARY MIXTURES

A binary mixture of components 1 and 2 is called as an ideal mixture, when it satisfies the following conditions.

Condition 1: The volume of the mixture is equal to the sum of the volumes of its constituents, i.e., upon mixing there is neither contraction nor expansion. Thus the specific volume of the mixture,  $v$  is given by:

$$v = \xi_1 \cdot v_1 + \xi_2 \cdot v_2$$

where  $\xi_1$  and  $\xi_2$  are the mass fractions of components 1 and 2. For a binary mixture,  $\xi_1$  and  $\xi_2$  are related by:

$$\xi_1 + \xi_2 = 1 \Rightarrow \xi_2 = 1 - \xi_1$$



# IDEAL, HOMOGENEOUS BINARY MIXTURES

Condition 2: Neither heat is generated nor absorbed upon mixing, i.e., the heat of solution is zero. Then the specific enthalpy of the mixture,  $h$  is given by:

$$h = \xi_1 \cdot h_1 + \xi_2 \cdot h_2 = \xi_1 \cdot h_1 + (1 - \xi_1)h_2$$

Condition 3: The mixture obeys Raoult's law in liquid phase, i.e., the vapour pressure exerted by components 1 and 2 ( $P_{v,1}$  and  $P$ ) at a temperature  $T$  are given by:

$$P_{v,1} = x_1 \cdot P_{1,sat}$$

$$P_{v,2} = x_2 \cdot P_{2,sat}$$

where  $x_1$  and  $x_2$  are the mole fractions of components 1 and 2 in solution, and  $P_{1,sat}$  and  $P_{2,sat}$  are the saturation pressures of pure components 1 and 2 at temperature  $T$ . The mole fractions  $x$  and  $x$  are related by:

$$x_1 + x_2 = 1 \Rightarrow x_2 = 1 - x_1$$



# IDEAL, HOMOGENEOUS BINARY MIXTURES

Condition 4: The mixture obeys Dalton's law in vapour phase; i.e., the vapour pressure exerted by components 1 and 2 ( $P_{v,1}$  and  $P_{v,2}$ ) in vapour phase at a temperature T are given by:

$$P_{v,1} = y_1 \cdot P_{\text{total}}$$

$$P_{v,2} = y_2 \cdot P_{\text{total}}$$

where  $y_1$  and  $y_2$  are the vapour phase mole fractions of components 1 and 2 and  $P_{\text{total}}$  is the total pressure exerted at temperature T. The vapour phase mole fractions  $y_1$  and  $y_2$  are related by:

$$y_1 + y_2 = 1 \Rightarrow y_2 = 1 - y_1$$

and the total pressure  $P_{\text{total}}$  is given by:

$$P_{\text{total}} = P_{v,1} + P_{v,2}$$

If one of the components, say component 2 is non-volatile compared to component 1 (e.g. component 1 is water and component 2 is lithium bromide salt), then  $y_1 \approx 1$  and  $y_2 \approx 0$ ,  $P_{v,2} \approx 0$ , then from Raoult's and Dalton's laws:

$$P_{\text{total}} \approx P_{v,1} = X_1 \cdot P_{1,\text{sat}}$$



# REAL MIXTURES

Real mixtures deviate from ideal mixtures since:

1. A real solution either contracts or expands upon mixing, i.e.,

$$V \neq \xi_1 \cdot V_1 + \xi_2 \cdot V_2$$

2. Either heat is evolved (exothermic) or heat is absorbed upon mixing;

$$h = \xi_1 \cdot h_1 + (1 - \xi_1)h_2 + \Delta h_{\text{mix}}$$

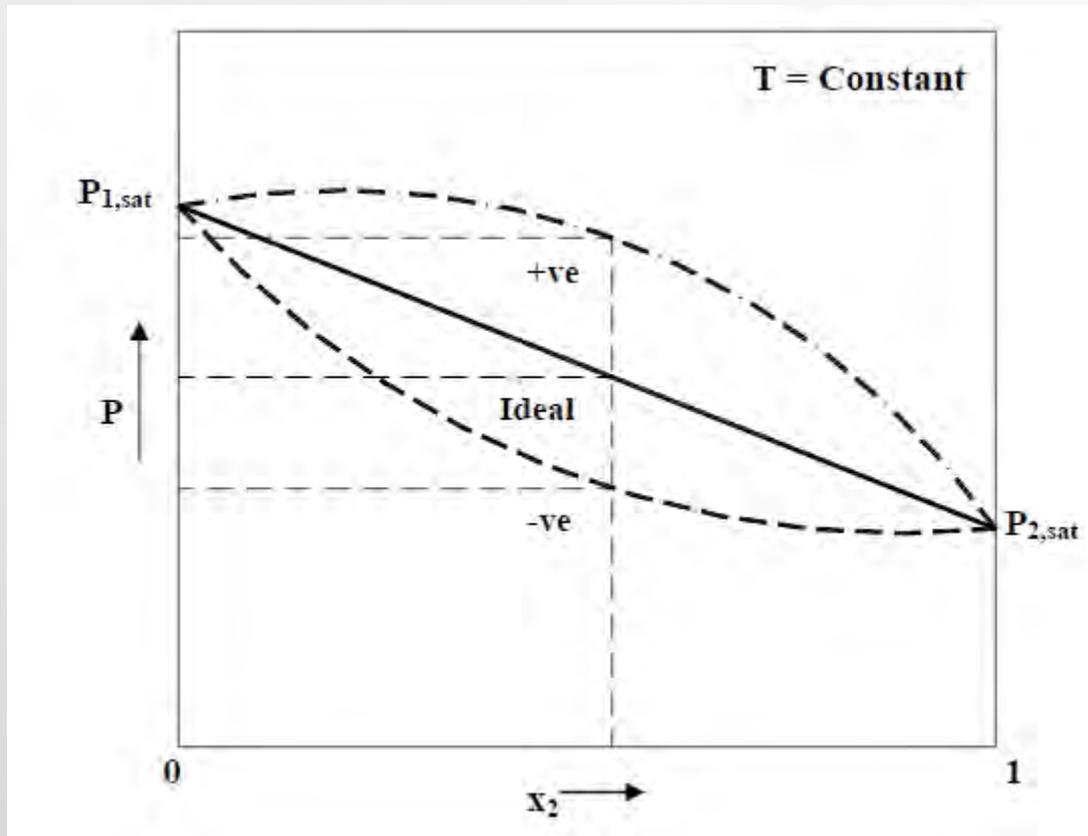
where  $\Delta h_{\text{mix}}$  is the heat of mixing, which is taken as negative when heat is evolved and positive when heat is absorbed.



## REAL MIXTURES

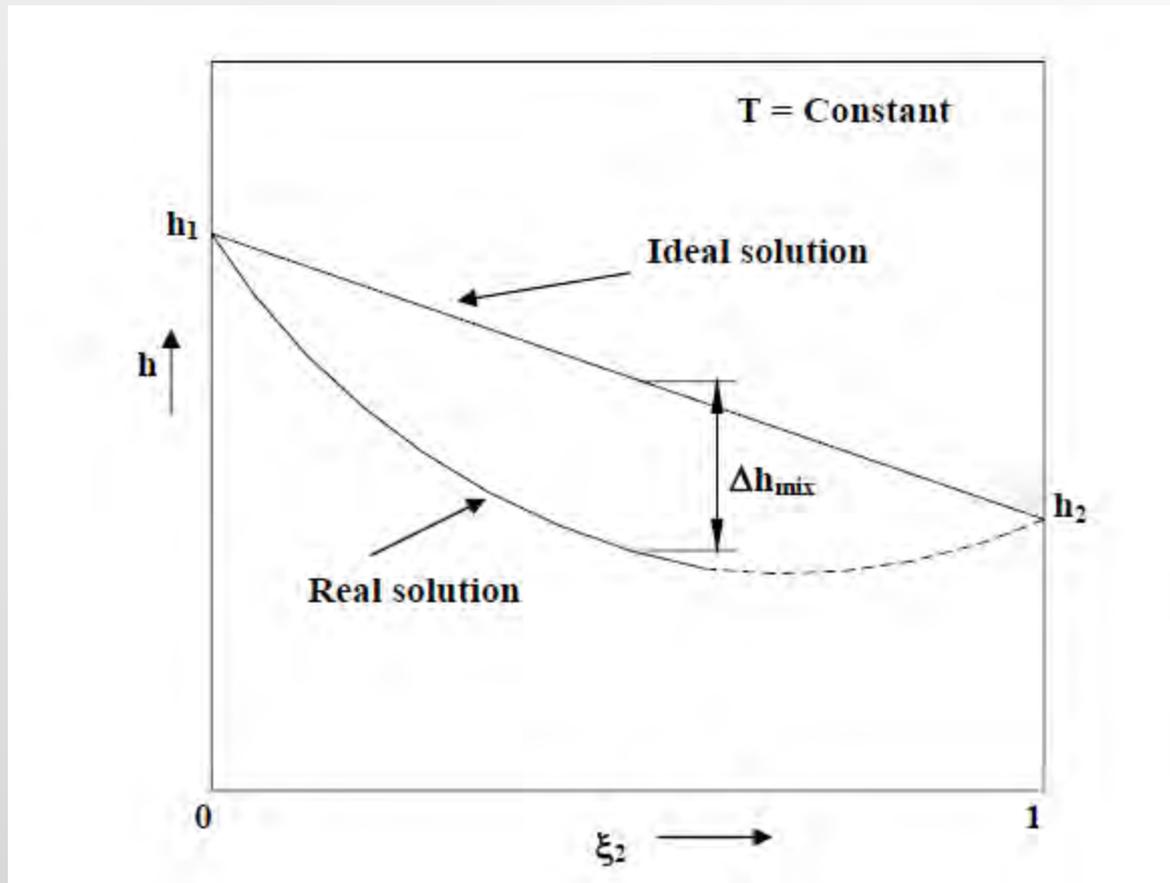
The above two differences between ideal and real mixtures can be attributed to the deviation of real mixtures from Raoult's law. Real mixtures approach ideal mixtures as the mole fraction of the component contributing to vapour pressure approaches unity, i.e., for very dilute solutions. Figure shows the equilibrium pressure variation with liquid phase mole fraction ( $x$ ) of ideal and real binary mixtures with positive (+ve) and negative deviations (-ve) from Raoult's law at a constant temperature. It can be seen that when the deviation from Raoult's law is positive (+ve), the equilibrium vapour pressure will be higher than that predicted by Raoult's law, consequently at a given pressure and composition, the equilibrium temperature of solution will be lower than that predicted by Raoult's law. The converse is true for solutions with -ve deviation from Raoult's law, i.e., the equilibrium temperature at a given pressure and composition will be higher than that predicted by Raoult's law for solution with negative deviation. This behavior can also be shown on specific enthalpy-composition diagram as shown in Fig. for a solution with negative deviation from Raoult's law. Refrigerant-absorbent mixtures used in vapour absorption refrigeration systems exhibit a negative deviation from Raoult's law, i.e., the process of absorption is exothermic with a negative heat of mixing.

# REAL MIXTURES



*Pressure-concentration behavior of ideal and real mixtures at a constant temperature*

# REAL MIXTURES



*Enthalpy-concentration behavior of an ideal mixture and a real mixture with negative deviation from Raoult's law*



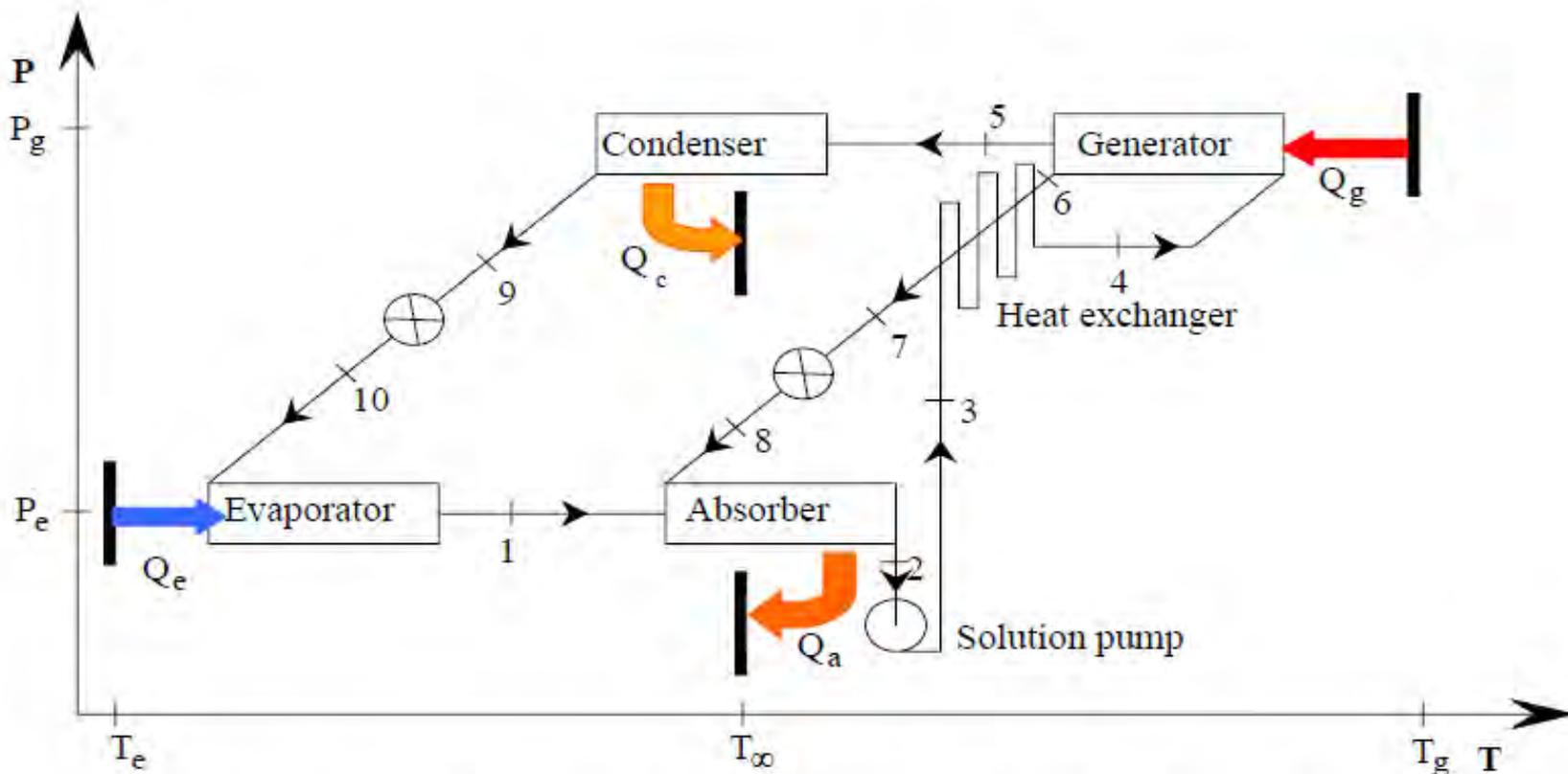
# BASIC VAPOUR ABSORPTION REFRIGERATION SYSTEM

Figure shows a basic vapour absorption refrigeration system with a solution heat exchanger on a pressure vs temperature diagram. As shown in the figure, low temperature and low pressure refrigerant vapour from evaporator at state 1 enters the absorber and is absorbed by solution weak in refrigerant (state 8). The heat of absorption ( $Q_a$ ) is rejected to an external heat sink at  $T_\infty$ . The solution, rich in refrigerant (state 2) is pumped to the generator pressure ( $P_g$ ) by the solution pump (state 3). The pressurized solution gets heated up sensibly as it flows through the solution heat exchanger by extracting heat from hot solution coming from generator (state 4). Heat is supplied to this solution from an external heat source in the generator ( $Q_g$  at  $T_g$ ), as a result refrigerant vapour is generated (absorbent may also boil to give off vapour in case of ammonia-water systems) at state 5. This high-pressure refrigerant vapour condenses in the condenser by rejecting heat of condensation to the external heat sink ( $Q_c$  at  $T_\infty$ ) and leaves the condenser as a high pressure liquid (state 9). This high pressure refrigerant liquid is throttled in the expansion device to evaporator pressure  $P_e$  (state 10) from where it enters the evaporator, extracts heat from low temperature heat source ( $Q_e$  at  $T_e$ ) and leaves the evaporator as vapour at state 1, completing a cycle. The hot solution that is weak in refrigerant (state 6) leaves the generator at high temperature and is cooled sensibly by rejecting heat to the solution going to the generator in the solution heat exchanger (state 7). Then it is throttled to the evaporator pressure in the throttle valve (state 8), from where it enters the absorber to complete the cycle. It can be seen that though not an essential component, the solution heat exchanger is used in practical systems to improve the COP by reducing the heat input in the generator.



# BASIC VAPOUR ABSORPTION REFRIGERATION SYSTEM

A solution heat exchanger as shown in Figure is a counter flow heat exchanger in which the hot solution coming from the generator comes in thermal contact with the cold solution going to the generator. As a result of this heat exchange, less heat input is required in the generator and less heat is rejected in the absorber, thus improving the system performance significantly.



*Basic vapour absorption refrigeration system with a solution heat exchanger on a pressure vs temperature diagram*



# BASIC VAPOUR ABSORPTION REFRIGERATION SYSTEM

The thermodynamic performance of the above system can be evaluated by applying mass and energy balance to each component assuming a steady flow process. In simple theoretical analyses, internal irreversibilities such as pressure drops between the components are generally neglected. To find the performance from the mass and energy balance equations one needs to know inputs such as the type of refrigerant-absorbent mixtures used in the system, operating temperatures, composition of solution at the entry and exit of absorber, effectiveness of solution heat exchanger etc. A simple steady flow analysis of the system will be presented in later sections.



# REFRIGERANT-ABSORBENT COMBINATIONS FOR VARS

The desirable properties of refrigerant-absorbent mixtures for VARS are:

- The refrigerant should exhibit high solubility with solution in the absorber. This is to say that it should exhibit negative deviation from Raoult's law at absorber.
- There should be large difference in the boiling points of refrigerant and absorbent (greater than  $200^{\circ}\text{C}$ ), so that only refrigerant is boiled-off in the generator. This ensures that only pure refrigerant circulates through refrigerant circuit (condenser-expansion valve-evaporator) leading to isothermal heat transfer in evaporator and condenser.
- It should exhibit small heat of mixing so that a high COP can be achieved. However, this requirement contradicts the first requirement. Hence, in practice a trade-off is required between solubility and heat of mixing.
- The refrigerant-absorbent mixture should have high thermal conductivity and low viscosity for high performance.
- It should not undergo crystallization or solidification inside the system.
- The mixture should be safe, chemically stable, non-corrosive, inexpensive and should be available easily.



# REFRIGERANT-ABSORBENT COMBINATIONS FOR VARS

The most commonly used refrigerant-absorbent pairs in commercial systems are:

- Water-Lithium Bromide ( $\text{H}_2\text{O-LiBr}$ ) system for above  $0^\circ\text{C}$  applications such as air conditioning. Here water is the refrigerant and lithium bromide is the absorbent.
- Ammonia-Water ( $\text{NH}_3\text{-H}_2\text{O}$ ) system for refrigeration applications with ammonia as refrigerant and water as absorbent.

Of late efforts are being made to develop other refrigerant-absorbent systems using both natural and synthetic refrigerants to overcome some of the limitations of ( $\text{H}_2\text{O-LiBr}$ ) and ( $\text{NH}_3\text{-H}_2\text{O}$ ) systems.

Currently, large water-lithium bromide ( $\text{H}_2\text{O-LiBr}$ ) systems are extensively used in air conditioning applications, where as large ammonia-water ( $\text{NH}_3\text{-H}_2\text{O}$ ) systems are used in refrigeration applications, while small ammonia-water systems with a third inert gas are used in a pumpless form in small domestic refrigerators (triple fluid vapour absorption systems).



# QUESTIONS

1. Compared to compression systems, absorption systems offer the benefits of:

- a) Higher COPs
- b) Lower refrigeration temperatures
- c) Possibility of using low-grade energy sources
- d) All of the above

2. Absorption of the refrigerant by the absorbent in a vapour absorption refrigeration system is accompanied by:

- a) Absorption of heat
- b) Release of heat
- c) No thermal effects
- d) Reduction in volume



# QUESTIONS

3. An absorption system consisting of only two closed vessels:

- a) Can provide continuous refrigeration
- b) Provides refrigeration intermittently
- c) Can work on solar energy alone
- d) Has no practical application

4. The conventional, continuously operating single stage vapour absorption refrigeration system:

- a) Requires only thermal energy as input
- b) Uses a thermal compressor in place of a mechanical compressor
- c) Does not require a condenser
- d) Consists of two expansion valves



## QUESTIONS

5. For an ideal refrigerant-absorbent mixture:

- a) There is neither expansion nor contraction upon mixing
- b) The mixing process is exothermic
- c) The mixing process is endothermic
- d) Obeys Raoult's law in liquid phase and Dalton's law in vapour phase

6. For a refrigerant-absorbent mixture with a negative deviation from Raoult's law:

- a) The mixing process is exothermic
- b) The mixing process is endothermic
- c) The actual equilibrium temperature will be less than that predicted by Raoult's law
- d) The actual equilibrium temperature will be less more that predicted by Raoult's law

# QUESTIONS

7. Refrigerant-absorbent pairs used in vapour absorption refrigeration systems should:

- a) Exhibit negative deviation from Raoult's law at absorber
- b) Exhibit positive deviation from Raoult's law at absorber
- c) Have large heat of mixing
- d) Have large boiling point difference between refrigerant and absorbent

8. Which of the following statements are true:

- a) Water-lithium bromide systems are used for refrigeration applications above  $0^{\circ}\text{C}$  only
- b) Ammonia-water systems can be used for refrigeration applications below  $0^{\circ}\text{C}$  only
- c) Small ammonia-water systems are used in domestic refrigerators
- d) Small water-lithium bromide systems are used in room air conditioners

# QUESTIONS

9. The operating temperatures of a single stage vapour absorption refrigeration system are: generator: **90°C**; condenser and absorber: **40°C**; evaporator: **0°C**. The system has a refrigeration capacity of **100 kW** and the heat input to the system is **160 kW**. The solution pump work is negligible.

- Find the COP of the system and the total heat rejection rate from the system.
- An inventor claims that by improving the design of all the components of the system he could reduce the heat input to the system to **80 kW** while keeping the refrigeration capacity and operating temperatures same as before. Examine the validity of the claim.



# STEAM JET REFRIGERATION SYSTEM

If water is sprayed into a chamber where a low pressure is maintained, a part of the water will evaporate. The enthalpy of evaporation will cool the remaining water to its saturation temperature at the pressure in the chamber. Obviously lower temperature will require lower pressure. Water freezes at  $0^{\circ}\text{C}$  hence temperature lower than  $4^{\circ}\text{C}$  cannot be obtained with water. In this system, high velocity steam is used to entrain the evaporating water vapour. High-pressure motive steam passes through either convergent or convergent-divergent nozzle where it acquires either sonic or supersonic velocity and low pressure of the order of  $0.009\text{ kPa}$  corresponding to an evaporator temperature of  $4^{\circ}\text{C}$ .

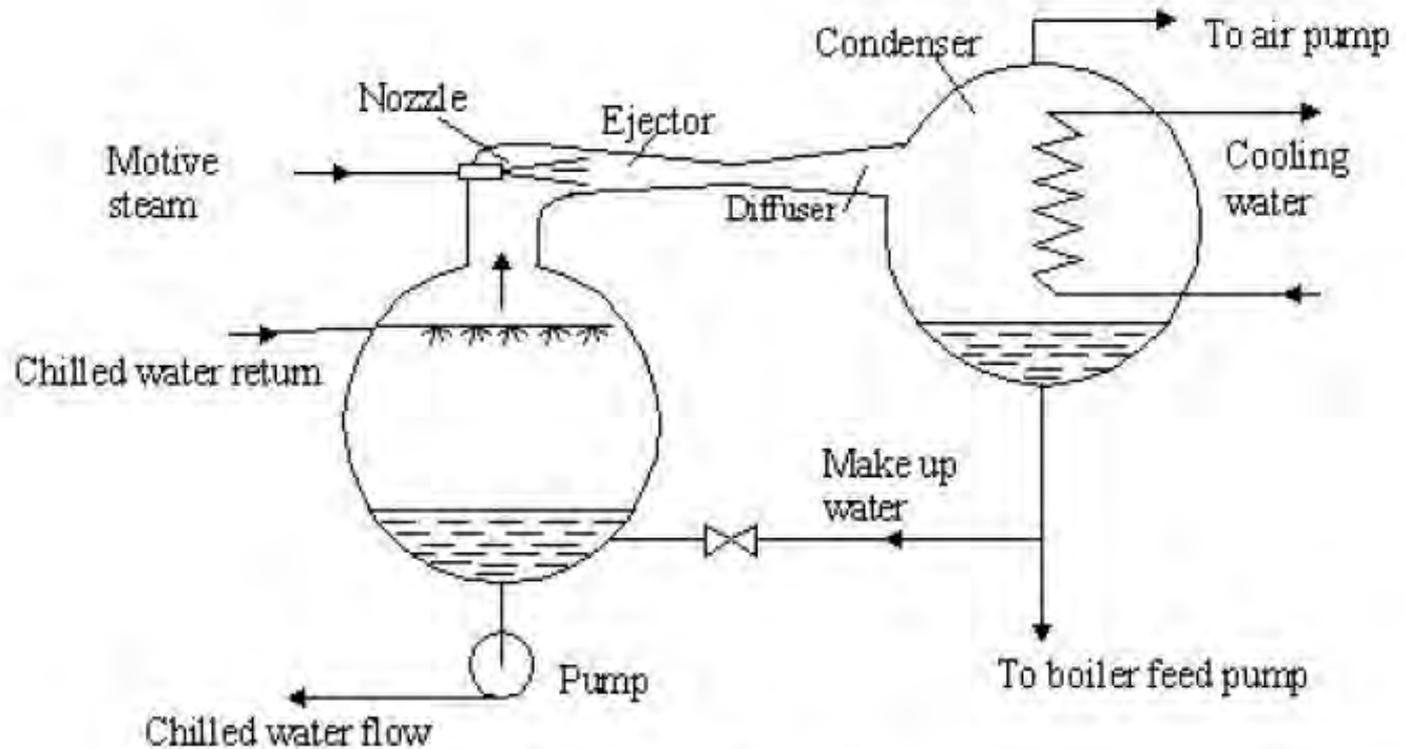


# STEAM JET REFRIGERATION SYSTEM

The high momentum of motive steam entrains or carries along with it the water vapour evaporating from the flash chamber. Because of its high velocity it moves the vapours against the pressure gradient up to the condenser where the pressure is 5.6-7.4 kPa corresponding to condenser temperature of 35-45°C. The motive vapour and the evaporated vapour both are condensed and recycled. This system is known as steam jet refrigeration system. Figure shows a schematic of the system. It can be seen that this system requires a good vacuum to be maintained. Sometimes, booster ejector is used for this purpose. This system is driven by low-grade energy that is process steam in chemical plants or a boiler.



# STEAM JET REFRIGERATION SYSTEM



**Steam jet refrigeration system**

*Schematic of a steam jet refrigeration system*



# STEAM JET REFRIGERATION SYSTEM

In 1838, the Frenchman Pelletan was granted a patent for the compression of steam by means of a jet of motive steam. Around 1900, the Englishman Charles Parsons studied the possibility of reduction of pressure by an entrainment effect from a steam jet. However, the credit for constructing the steam jet refrigeration system goes to the French engineer, Maurice Leblanc who developed the system in 1907-08. In this system, ejectors were used to produce a high velocity steam jet ( $\approx 1200$  m/s). Based on Leblanc's design the first commercial system was made by Westinghouse in 1909 in Paris. Even though the efficiency of the steam jet refrigeration system was low, it was still attractive as water is harmless and the system can run using exhaust steam from a steam engine.



# STEAM JET REFRIGERATION SYSTEM

Between 1928-1930, there was much interest in this type of systems in USA. In USA they were mainly used for air conditioning of factories, cinema theatres, ships and even railway wagons. Several companies such as Westinghouse, Ingersoll Rand and Carrier started commercial production of these systems from 1930. However, gradually these systems were replaced by more efficient vapour absorption systems using LiBr-water. Still, some east European countries such as Czechoslovakia and Russia manufactured these systems as late as 1960s. The ejector principle can also be used to provide refrigeration using fluids other than water, i.e., refrigerants such as CFC-11, CFC-21, CFC-22, CFC-113, CFC-114 etc. The credit for first developing these closed vapour jet refrigeration systems goes to the Russian engineer, I.S. Badylkes around 1955. Using refrigerants other than water, it is possible to achieve temperatures as low as  $-100^{\circ}\text{C}$  with a single stage of compression. The advantages cited for this type of systems are simplicity and robustness, while difficult design and economics are its chief disadvantages.



# THERMOELECTRIC REFRIGERATION SYSTEMS

when current flows through a conductor of a thermocouple that has an initial temperature gradient in it, then heat transfer rate per unit length is proportional to the product of current and the temperature. As the current flow through thermoelectric material it gets heated due to its electrical resistance. This is called the Joulean effect, further, conduction heat transfer from the hot junction to the cold junction transfers heat. Both these heat transfer rates have to be compensated by the Peltier Effect for some useful cooling to be produced. For a long time, thermoelectric cooling based on the Peltier effect remained a laboratory curiosity as the temperature difference that could be obtained using pure metals was too small to be of any practical use. Insulating materials give poor thermoelectric performance because of their small electrical conductivity while metals are not good because of their large thermal conductivity. However, with the discovery of semiconductor materials in 1949-50, the available temperature drop could be increased considerably, giving rise to commercialization of thermoelectric refrigeration systems.

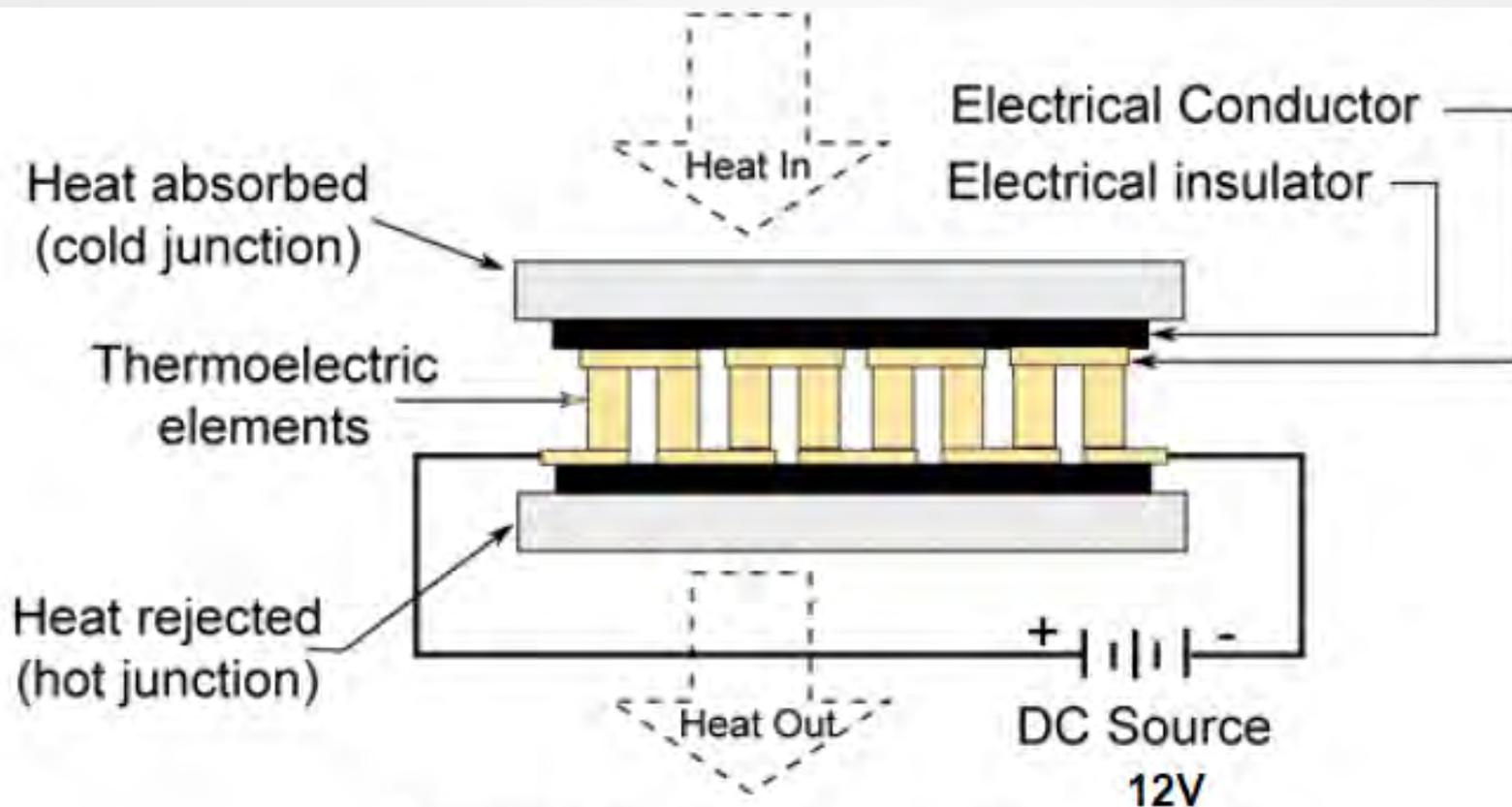


# THERMOELECTRIC REFRIGERATION SYSTEMS

Figure shows the schematic of the thermoelectric refrigeration system based on semiconductor materials. The Russian scientist, A. F. Ioffe is one of the pioneers in the area of thermoelectric refrigeration systems using semiconductors. Several domestic refrigerators based on thermoelectric effect were made in USSR as early as 1949. However, since 1960s these systems are used mainly used for storing medicines, vaccines etc. and in electronic cooling. Development also took place in many other countries. In USA domestic refrigerators, air conditioners, water coolers, air conditioned diving suits etc.



# THERMOELECTRIC REFRIGERATION SYSTEMS



*Schematic of a thermoelectric refrigeration system*

# THERMOELECTRIC REFRIGERATION SYSTEMS

System capacities were typically small due to poor efficiency. However some large refrigeration capacity systems such as a 3000 kcal/h air conditioner and a 6 tonne capacity cold storage were also developed. By using multistaging temperatures as low as  $-145^{\circ}\text{C}$  were obtained. These systems due to their limited performance (limited by the materials) are now used only in certain niche applications such as electronic cooling, mobile coolers etc. Efforts have also been made to club thermoelectric systems with photovoltaic cells with a view to develop solar thermoelectric refrigerators.

# QUESTIONS

**Q1.** In an air cycle refrigeration system, low temperatures are produced due to:

- a) Evaporation of liquid air
- b) Throttling of air
- c) Expansion of air in turbine
- d) None of the above

**Q2.** Air cycle refrigeration systems are most commonly used in:

- a) Domestic refrigerators
- b) Aircraft air conditioning systems
- c) Cold storages
- d) Car air conditioning systems

**Q3.** The required input to the steam jet refrigeration systems is in the form of:

- a) Mechanical energy
- b) Thermal energy
- c) High pressure, motive steam
- d) Both mechanical and thermal energy

# QUESTIONS

**Q4.** A nozzle is used in steam jet refrigeration systems to:

- a) To convert the high pressure motive steam into high velocity steam
- b) To reduce energy consumption
- c) To improve safety aspects
- d) All of the above

**Q5.** The materials used in thermoelectric refrigeration systems should have:

- a) High electrical and thermal conductivity
- b) High electrical conductivity and low thermal conductivity
- c) Low electrical conductivity and high thermal conductivity
- d) Low electrical and thermal conductivity

**Q6.** A thermoelectric refrigeration systems requires:

- a) A high voltage AC (alternating current) input
- b) A low voltage AC input
- c) A high voltage DC (direct current) input
- d) A low voltage DC input

# QUESTIONS

**Q1.** Explain why ice making using nocturnal cooling is difficult on nights when the sky is cloudy?

**Q2.** When you add sufficient amount of glucose to a glass of water, the water becomes cold. Is it an example of refrigeration, if it is, can this method be used for devising a refrigeration system?

**Q3.** To what do you attribute the rapid growth of refrigeration technology over the last century?



# VORTEX TUBE REFRIGERATION SYSTEMS

In 1931, the French engineer Georges Ranque (1898-1973) discovered an interesting phenomenon, which is called “Ranque effect” or “vortex effect”. The tangential injection of air into a cylindrical tube induces to quote his words “ a giratory expansion with simultaneous production of an escape of hot air and an escape of cold air”. Ranque was granted a French patent in 1928 and a US patent in 1934 for this effect.



# VORTEX TUBE REFRIGERATION SYSTEMS

However, the discovery was neglected until after the second world war, when in 1945, Rudolph Hilsch, a German physicist, studied this effect and published a widely read scientific paper on this device. Thus, the vortex tube has also been known as the "*Ranque- Hilsch Tube*". Though the efficiency of this system is quite low, it is very interesting due to its mechanical simplicity and instant cooling. It is convenient where there is a supply of compressed air.



# VORTEX TUBE REFRIGERATION SYSTEMS

The present day vortex tube uses compressed air as a power source, it has no moving parts, and produces hot air from one end and cold air from the other. The volume and temperature of these two airstreams are adjustable with a valve built into the hot air exhaust. Temperatures as low as  $-46^{\circ}\text{C}$  and as high as  $127^{\circ}\text{C}$  are possible. Compressed air is supplied to the vortex tube and passes through nozzles that are tangential to an internal counter bore.



# VORTEX TUBE REFRIGERATION SYSTEMS

These nozzles set the air in a vortex motion. This spinning stream of air turns  $90^\circ$  and passes down the hot tube in the form of a spinning shell, similar to a tornado. A valve at one end of the tube allows some of the warmed air to escape. What does not escape, heads back down the tube as a second vortex inside the low-pressure area of the larger vortex. This inner vortex loses heat and exhausts through the other end as cold air. Currently vortex tube is used for spot cooling of machine parts, in electronic cooling and also in cooling jackets for miners, firemen etc.



## QUESTIONS

**Q1.** In an air cycle refrigeration system, low temperatures are produced due to:

- a) Evaporation of liquid air
- b) Throttling of air
- c) Expansion of air in turbine
- d) None of the above

**Q2.** Air cycle refrigeration systems are most commonly used in:

- a) Domestic refrigerators
- b) Aircraft air conditioning systems
- c) Cold storages
- d) Car air conditioning systems

**Q3.** The required input to the steam jet refrigeration systems is in the form of:

- a) Mechanical energy
- b) Thermal energy
- c) High pressure, motive steam
- d) Both mechanical and thermal energy



## QUESTIONS

**Q4.** A nozzle is used in steam jet refrigeration systems to:

- a) To convert the high pressure motive steam into high velocity steam
- b) To reduce energy consumption
- c) To improve safety aspects
- d) All of the above

**Q5.** The materials used in thermoelectric refrigeration systems should have:

- a) High electrical and thermal conductivity
- b) High electrical conductivity and low thermal conductivity
- c) Low electrical conductivity and high thermal conductivity
- c) Low electrical and thermal conductivity

**Q6.** A thermoelectric refrigeration systems requires:

- a) A high voltage AC (alternating current) input
- b) A low voltage AC input
- c) A high voltage DC (direct current) input
- d) A low voltage DC input



## QUESTIONS

- Q7.** Explain why ice making using nocturnal cooling is difficult on nights when the sky is cloudy?
- Q8.** When you add sufficient amount of glucose to a glass of water, the water becomes cold. Is it an example of refrigeration, if it is, can this method be used for devising a refrigeration system?
- Q9.** To what do you attribute the rapid growth of refrigeration technology over the last century?



## QUESTIONS (Theory)

1. Explain mechanism of a simple vapour compression refrigeration system with neat sketch and p-h and T-s diagram. Mention the advantages and disadvantages of vapour compression system over air refrigeration system.
2. Explain types of vapour compression refrigeration cycles with expression of C.O.P and with help of p-h and T-s diagram:
  - a. Theoretical vapour compression cycle with dry saturated vapour after compression
  - b. Theoretical vapour compression cycle with wet vapour after compression
  - c. Theoretical vapour compression cycle with superheated vapour after compression
  - d. Theoretical vapour compression cycle with superheated vapour before compression
  - e. Theoretical vapour compression cycle with subcooling of refrigerant
3. Discuss various factors affecting the performance of vapour simple compression refrigeration cycle.

## QUESTIONS (Theory)

4. How actual vapour compression refrigeration cycle differs from theoretical vapour compression refrigeration cycle? Explain with T-s diagram.
5. Explain following simple saturation cycle with p-h diagram:
  - a. Simple saturation cycle with flash chamber
  - b. Simple saturation cycle with accumulator OR pre-cooler
  - c. Simple saturation cycle with subcooling of liquid refrigerant by vapour refrigerant
  - d. Simple saturation cycle with subcooling of liquid refrigerant by liquid refrigerant.
6. Mention the limitations of simple vapour compression refrigeration cycle. Give Advantages of compound vapour compression (or Multi-stage) refrigeration system with intercooler.

# QUESTIONS (Theory)

7. Explain following types of compound vapour compression with intercooler with schematic and p-h diagram:
  - a. Two stage compression with liquid flash intercooler
  - b. Two stage compression with flash chamber
  - c. Two stage compression with flash chamber as liquid subcooler
  - d. Two stage compression with water intercooler and liquid subcooler
  - e. Two stage compression with water intercooler, liquid subcooler and liquid flash chamber
  - f. Two stage compression with water intercooler, liquid subcooler and flash intercooler
8. Explain Cascade refrigeration system with neat sketch.

## QUESTIONS (Theory)

9. Explain following types of Multiple evaporators and compression system with schematic and p-h diagram:
  - a. Multiple evaporators at same temperature with single compressor and expansion valve
  - b. Multiple evaporators at different temperatures with single compressor, individual expansion valves and back pressure valves
  - c. Multiple evaporators at different temperatures with single compressor, multiple expansion valves and back pressure valves
  - d. Multiple evaporators at different temperatures with individual compressors and individual expansion valves
  - e. Multiple evaporators at different temperatures with individual compressors and multiple expansion valves
  - f. Multiple evaporators at different temperatures with compound compression, individual expansion valves