1

Introduction

Course Contents

1.1 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. To do this, heat must be removed from the body and it must be transferred to another body whose temperature is below that of the refrigerated body or space need to be cooled.
- 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.
- The subject of refrigeration and air conditioning has evolved out of human need for food and comfort, and its history dates back to centuries. The history of refrigeration is very interesting since every aspect of it, the availability of refrigerants, the prime movers and the developments in compressors and the methods of refrigeration all are a part of it. The French scientist Roger Thyvenot has written an excellent book on the history of refrigeration throughout the world. Here we present only a brief history of the subject with special mention of the pioneers in the field and some important events.

  There are mainly two methods of producing cooling: 1. Natural refrigeration and 2. Artificial refrigeration

1.2 Methods of producing cooling or refrigeration

The cooling may be achieved naturally or by artificially.

1.2.1 Natural Refrigeration

- Following are the methods of natural refrigeration:
  1. Art of Ice making by Nocturnal Cooling
  2. Evaporative Cooling
  3. Cooling by Salt Solutions
- In olden days refrigeration was achieved by natural means such as the use of ice or evaporative cooling. In earlier times, ice was 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.
- In 1806, Frederic Tudor, (who was later called as the "ice king") began the trade in ice by cutting it from the Hudson River and ponds of Massachusetts and exporting it to various

countries including India. In India Tudor's ice was cheaper than the locally manufactured ice by nocturnal cooling. The ice trade in North America was a flourishing business. Ice was transported to southern states of America in train compartments insulated by 0.3m of cork insulation. Trading in ice was also popular in several other countries such as Great Britain, Russia, Canada, Norway and France. In these countries ice was either transported from colder regions or was harvested in winter and stored in icehouses for use in summer. The ice trade reached its peak in 1872 when America alone exported 225000 tonnes of ice to various countries as far as China and Australia. However, with the advent of artificial refrigeration the ice trade gradually declined.

1. 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°C and by early morning hours the water in the trays freezes to ice. This method of ice production was very popular in India.

2. 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.

3. 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 (NaCl) can yield temperatures up to -20°C and Calcium Chloride ($CaCl_2$) up to - 50°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.

1.2.2 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.

1. Vapour Compression Refrigeration Systems:

– The basis of modern refrigeration is the ability of liquids to absorb enormous quantities of heat as they boil and evaporate. Professor William Cullen of the University of Edinburgh demonstrated this in 1755 by placing some water in thermal contact with ether under a receiver of a vacuum pump. The evaporation rate of ether increased due to the vacuum pump and water could be frozen. This process involves two thermodynamic concepts, the vapour pressure and the latent heat. A liquid is in thermal equilibrium with its own vapor at a pressure called the saturation pressure, which depends on the temperature alone. If the pressure is increased for example in a pressure cooker, the water boils at higher temperature. The second concept is that the evaporation of liquid requires latent heat during evaporation. If latent heat is extracted from the liquid, the liquid gets cooled. The temperature of ether will remain constant as long as the vacuum pump maintains a pressure equal to saturation pressure at the desired temperature. This requires the removal of all the vapors formed due to vaporization. If a lower temperature is desired, then a lower saturation pressure will have to be maintained by the vacuum pump. The component of the modern day refrigeration system where cooling is produced by this method is called evaporator.

– 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 SO_2 in 1780 while van Marum and Van Troostwijk liquefied 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.1.1. 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”.

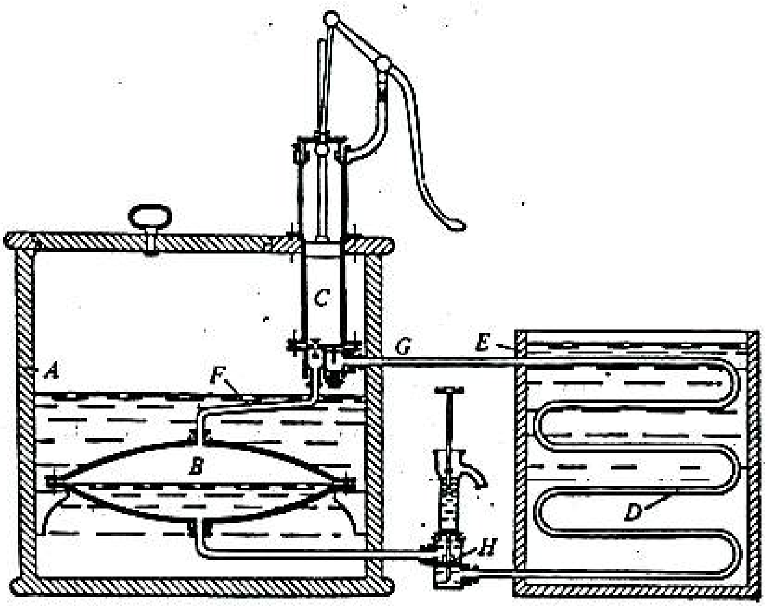


Fig. 1.1. Apparatus described by Jacob Perkins in his patent specification of 1834. The refrigerant (ether or other volatile fluid) boils in evaporator B taking heat from surrounding water in container A. The pump C draws vapour away and compresses it to higher pressure at which it can condense to liquids 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.

– John Hague made Perkins's design into working model with some modifications. This Perkins machine is shown in Fig.1.2. 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, NH_3 and CO_2.

Fig.1.2. Perkins machine built by John Hague

- 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, refrigeration system using dimethyl ether which has a normal boiling point of −23.6°C.
- Carl von Linde in Munich introduced double acting ammonia compressor. It required pressures of more than 10 atmospheres in the condenser. Since the normal boiling point of ammonia is -33.3°C, vacuum was not required on the low pressure side. Since then ammonia is used widely in large refrigeration plants.
- David Boyle, in fact made the first NH_3 system in 1871 in San Francisco. John Enright had also developed a similar system in 1876 in Buffalo N.Y. Franz Windhausen developed carbon dioxide (CO_2) based vapor compression system in Germany in 1886. The carbon dioxide compressor requires a pressure of about 80 atmospheres and therefore a very heavy construction. Linde in 1882 and T.S.C. Lowe in 1887 tried similar systems in USA. The CO_2 system is a very safe system and was used in ship refrigeration until 1960s. Raoul Pictet used SO_2 (NBP -10°C) as refrigerant. Its lowest pressure was high enough to prevent the leakage of air into the system.
- Palmer used C_2H_5Cl in 1890 in a rotary compressor. He mixed it with C_2H_5Br to reduce its flammability. Edmund Copeland and Harry Edwards used iso-butane in 1920 in small refrigerators. It disappeared by 1930 when it was replaced by CH_3Cl. Dichloroethylene (Dielene or Dieline) was used by Carrier in centrifugal compressors in 1922-26.

2. 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. Hence, efforts, starting from 1887 have been made to develop domestic refrigerators using mechanical systems. The initial domestic mechanical refrigerators were costly, not completely automatic and were not very reliable. However, the development of mechanical household refrigerators on a large scale was made possible by the development of small compressors, automatic refrigerant controls, better shaft seals, developments in electrical power systems and induction motors.
- General Electric Company introduced the first domestic refrigerator in 1911, followed by Frigidaire in 1915. Kelvinator launched the domestic mechanical refrigerator in 1918 in USA. In 1925, USA had about 25 million domestic refrigerators of which only 75000 were mechanical. However, the manufacture of domestic refrigerators grew very rapidly, and

by 1949 about 7 million domestic refrigerators were produced annually. With the production volumes increasing the price fell sharply (the price was 600 dollars in 1920 and 155 dollars in 1940). The initial domestic refrigerators used mainly sulphur dioxide as refrigerant. Some units used methyl chloride and methylene chloride. These refrigerants were replaced by Freon-12 in 1930s. In the beginning these refrigerators were equipped with open type compressors driven by belt drive. General Electric Company introduced the first refrigerator with a hermetic compressor in 1926. Soon the open type compressors were completely replaced by the hermetic compressors. First refrigerators used water-cooled condensers, which were soon replaced by air cooled-condensers. Though the development of mechanical domestic refrigerators was very rapid in USA, it was still rarely used in other countries.

- In 1930 only rich families used domestic refrigerators in Europe. The domestic refrigerator based on absorption principle as proposed by Platen and Munters, was first made by Electrolux Company in 1931 in Sweden. In Japan the first mechanical domestic refrigerator was made in 1924. The first dual temperature (freezer-refrigerator) domestic refrigerator was introduced in 1939. The use of mechanical domestic refrigerators grew rapidly all over the world after the Second World War. Today, a domestic refrigerator has become an essential kitchen appliance not only in highly developed countries but also in countries such as India. Except very few almost all the present day domestic refrigerators are mechanical refrigerators that use a hermetic compressor and an air cooled condenser. The modern refrigerators use either HFC-134a (hydro-fluoro-carbon) or iso-butane as refrigerant.

3. 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. Industrial air conditioning is largely responsible for the growth

of modern electronic, pharmaceutical, chemical industries etc. Most of the present day air conditioning systems use either a vapour compression refrigeration system or a vapour absorption refrigeration system. The capacities vary from few kilowatts to megawatts.

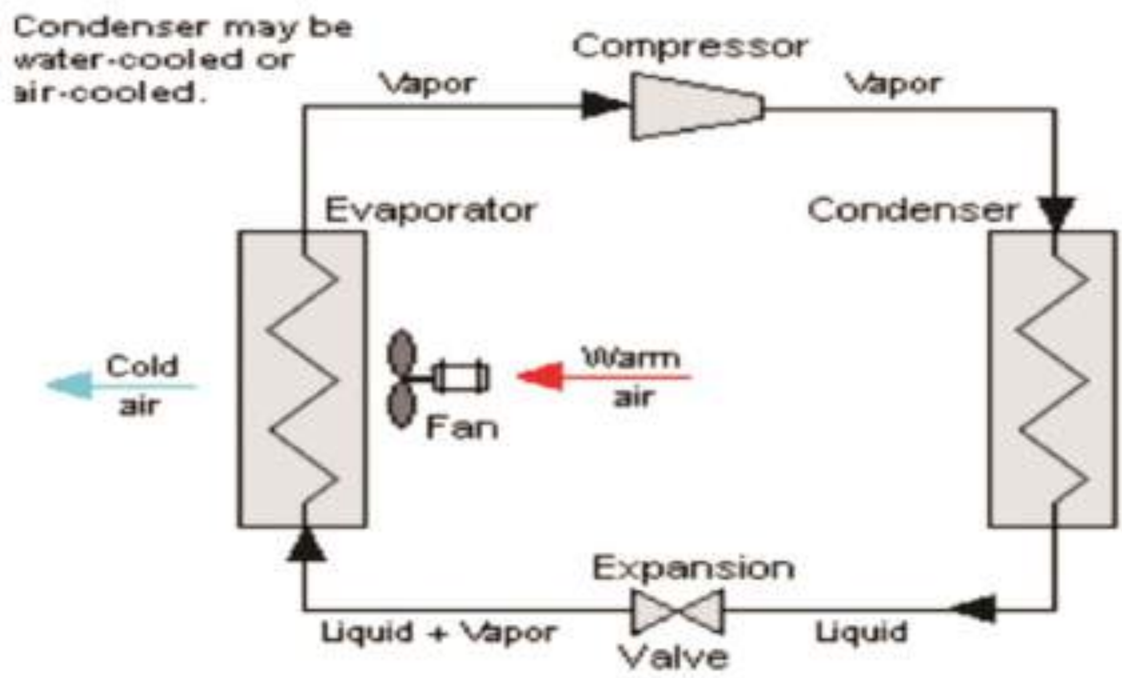


Fig. 1.3 Schematic of vapour compression refrigeration system

- Figure 1.3 shows the basic components of a vapour compression refrigeration system. As shown in the figure the basic system consists of an evaporator, compressor, condenser and an expansion valve. The refrigeration effect is obtained in the cold region as heat is extracted by the vaporization of refrigerant in the evaporator. The refrigerant vapour from the evaporator is compressed in the compressor to a high pressure at which its saturation temperature is greater than the ambient or any other heat sink. Hence when the high pressure, high temperature refrigerant flows through the condenser, condensation of the vapour into liquid takes place by heat rejection to the heat sink. To complete the cycle, the high pressure liquid is made to flow through an expansion valve. In the expansion valve the pressure and temperature of the refrigerant decrease. This low pressure and low temperature refrigerant vapour evaporates in the evaporator taking heat from the cold region. It should be observed that the system operates on a closed cycle. The system requires input in the form of mechanical work. It extracts heat from a cold space and rejects heat to a high temperature heat sink. Fig.1.3. Schematic of a basic vapour compression refrigeration system.
- A refrigeration system can also be used as a heat pump, in which the useful output is the high temperature heat rejected at the condenser. Alternatively, a refrigeration system can be used for providing cooling in summer and heating in winter. Such systems have been built and are available now.

4. Vapour Absorption Refrigeration Systems:

- John Leslie in 1810 kept H_2SO_4 and water in two separate jars connected together. H_2SO_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. H_2SO_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 H_2SO_4.
- Ferdinand Carre invented aqua-ammonia absorption system in 1860. Water is a strong absorbent of NH_3. If 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 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. 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.

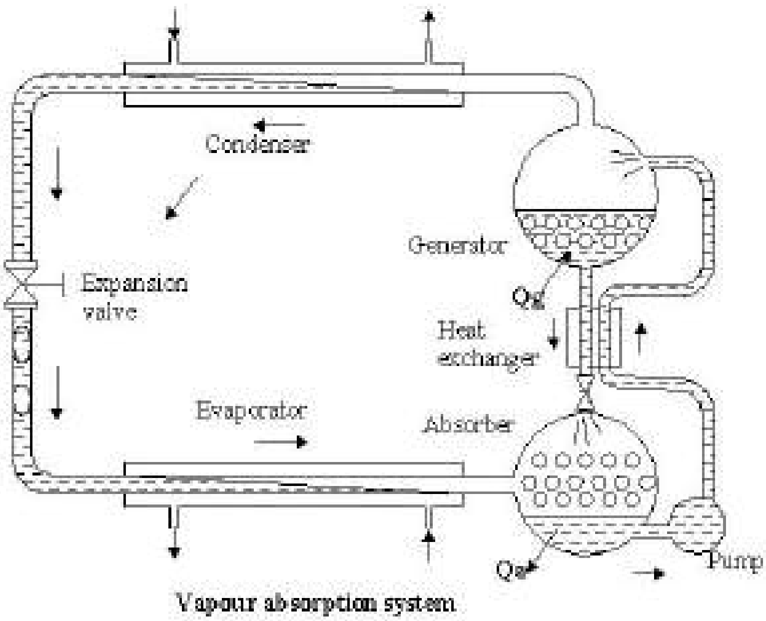


Fig. 1.4 Vapour Absorption Refrigeration Systems

- Figure 1.4 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 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 1.5 shows the schematic of the triple fluid vapour absorption refrigeration system.

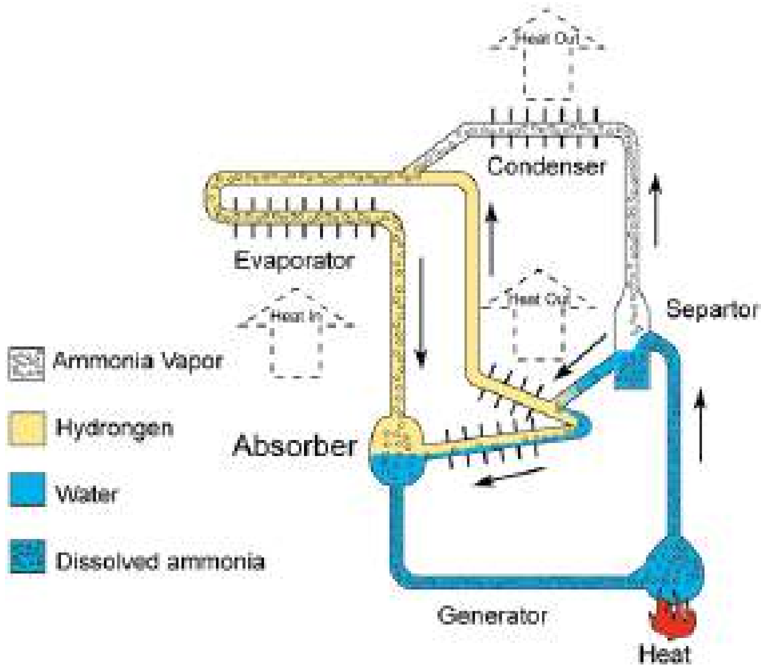


Fig. 1.5 Schematic of triple fluid vapour absorption refrigeration system

- Another vapour absorption system is based on Lithium Bromide (Li-Br) water. This system is used for chilled water air-conditioning system. This is a descendent of Windhausen's machine with LiBr replacing H_2SO_4. In this system LiBr is the absorbent and water is the refrigerant. This system works at vacuum pressures. The condenser and the generator are housed in one cylindrical vessel and the evaporator and the absorber are housed in second vessel. This also runs on low-grade energy requiring a boiler or process steam.

5. Solar energy based refrigeration system

- Attempts have been made to run vapour absorbtion system by solar energy with concentrating and flat plate solar collectors. Several small solar absorption refrigeration systems have been made around 1950s in several countries. Professor G.O.G. of America is one of the pioneers in the area of solar refrigeration using flat plate collectors. A solar refrigeration system that could produce 250 kg of ice per day was installed in Tashkent, USSR in 1953. This system used a parabolic mirror of 10 m^2 area for concentrating the solar radiation. F. Trombe installed an absorption machine with a cylindro-parabolic mirror of 20 m^2 at Montlouis, France, which produced 100 kg of ice per day.
- Serious concentration to solar refrigeration system was given since 1965 due to scarcity of fossil fuel based energy sources. LiBr-water based systems have been developed for air conditioning purposes. The first solar air conditioning system was installed in an experimental solar house in University of Queensland, Australia in 1966. After this several systems based on solar energy were built in many parts of the world including India. In 1976, there were about 500 solar absorption systems in USA alone. Almost all these were based on LiBr-water as these systems do not require very high heating temperatures. These systems were mainly used for space air conditioning.

- Intermediate absorption system based on solar energy have also been built and operated successfully. In these systems, the cooling effect is obtained during the nighttime, while the system gets “charged” during the day using solar energy. Though the efficiency of these systems is rather poor requiring solar collector area, they may find applications in remote and rural areas where space is not a constraint. In addition, these systems are environment friendly as they use eco-friendly refrigerants and run on clean and renewable solar energy.
- Solar absorption system with ammoniacates, sodium thiocyanate, activated charcoal, zeolite as adsorbents and ammonia, alcohols or fluorocarbons as refrigerants have also been in use since 1950s. These systems also do not require a compressor. The refrigerant vapour is driven by the adsorption potential of the adsorbent stored in an adsorbent bed. This bed is connected to an evaporator/condenser, which consists of the pure refrigerant. In intermittent adsorption systems, during the night the refrigerant evaporates and is adsorbed in activated charcoal or zeolite providing cooling effect.
- During daytime the adsorbent bed absorbs solar radiation and drives off the refrigerant stored in the bed. This refrigerant vapour condenses in the condenser and stored in a reservoir for nighttime use. Thus this simple system consists of an adsorbent bed and a heat exchanger, which acts as a condenser during the nighttime and, as an evaporator during the night. Pairs of such reactors can be used for obtaining a continuous cooling.

6. 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^{th} 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 -7^0C, 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 40^0C. Paul Gifford in 1875 perfected the open type of machine.
- 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.

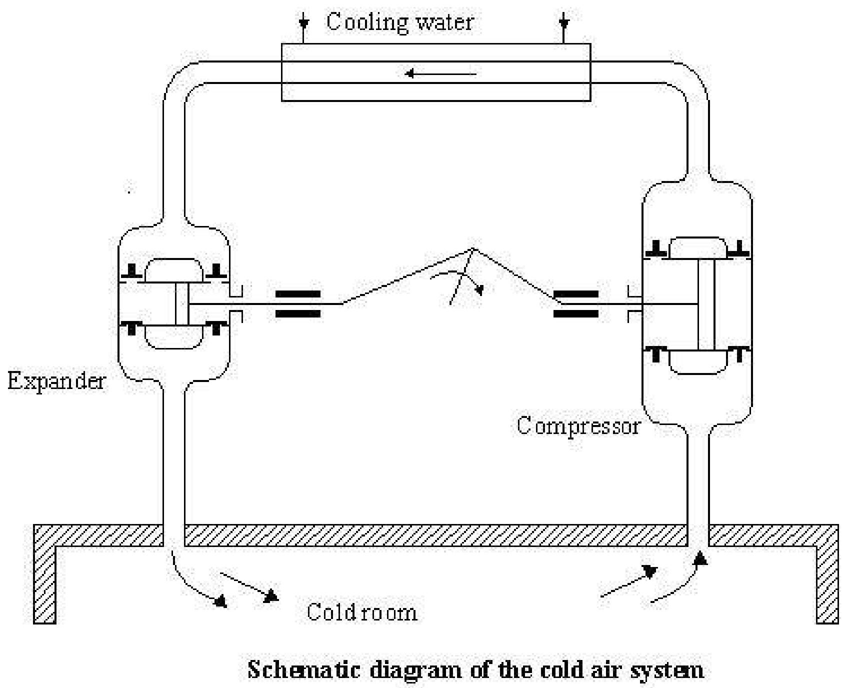


Fig.1.6. Schematic of a basic, open type air cycle refrigeration system

- Figure 1.6 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.

7. 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^oC hence temperature lower than 4^0C 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 both sonic or supersonic velocity and low pressure of the order of 0.009 kPa corresponding to an evaporator temperature of 4^0C. 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 $35\text{-}45^0C$. The motive vapour and the evaporated vapour both are condensed and recycled. This system is known as steam jet refrigeration system.

– Figure 1.7 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.

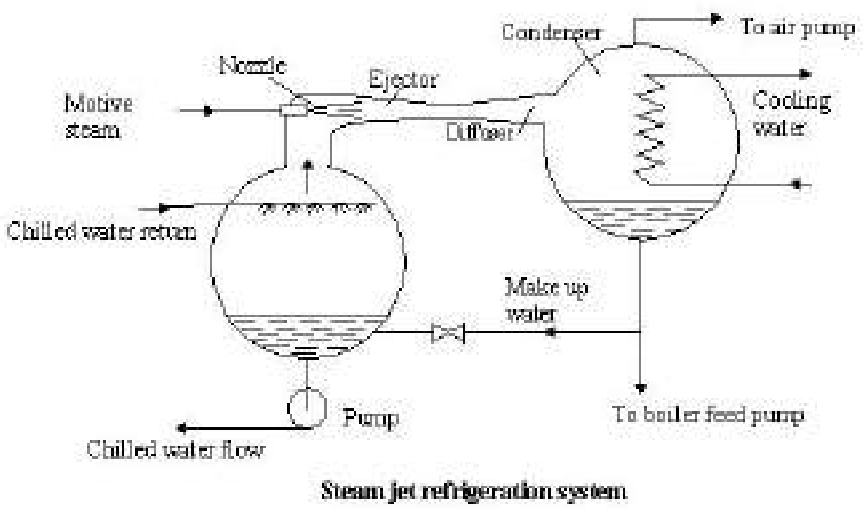


Fig.1.7. Schematic of a steam jet refrigeration system

– 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 (≈ 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. From 1910 onwards, stem jet refrigeration systems were used mainly in breweries, chemical factories, warships etc. In 1926, the French engineer Follain improved the machine by introducing multiple stages of vaporization and condensation of the suction steam.

– 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 –100oC 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.

8. 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. 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.

– Figure 1.8 shows the schematic of the thermoelectric refrigeration system based on semiconductor materials. The Russian scientist, A. F. Loffe 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 using these effects.

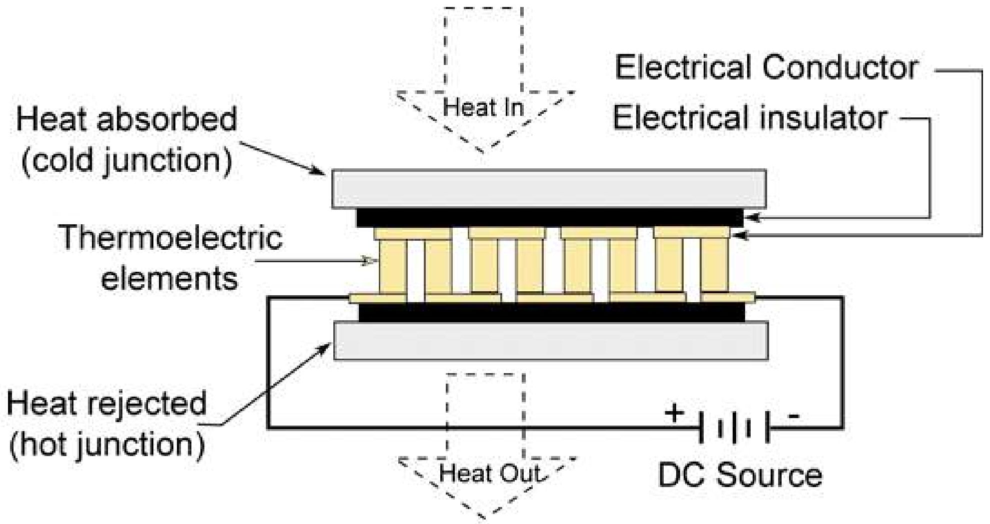
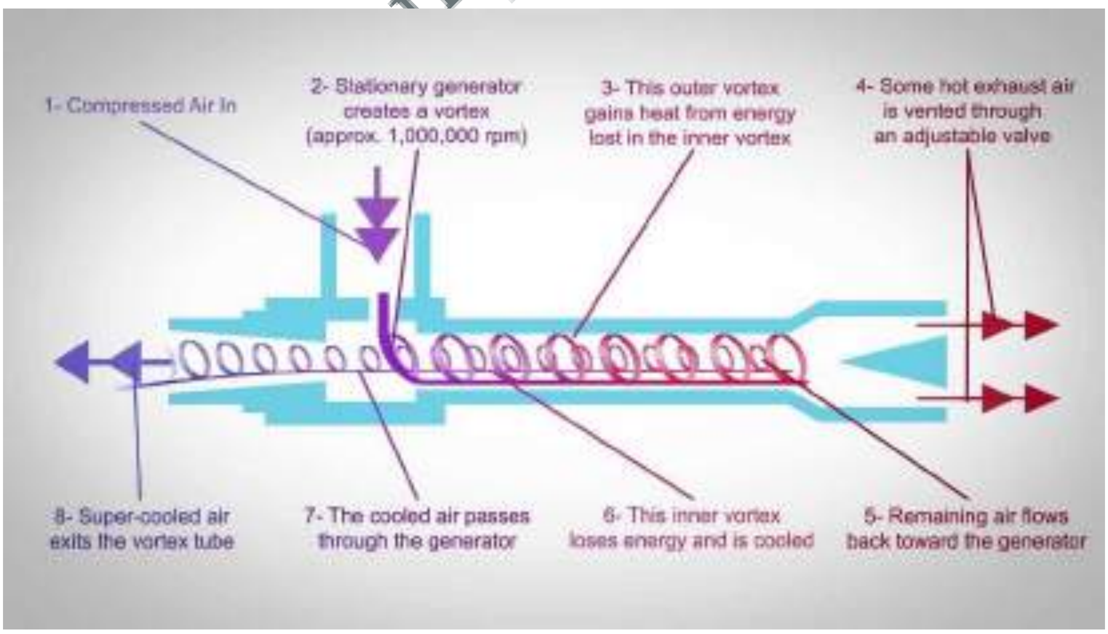


Fig. 1.8. Schematic of a thermoelectric refrigeration system

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°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.

9. **Vortex tube 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. 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. 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°C and as high as 127°C are possible. Compressed air is supplied to the vortex tube and passes through nozzles that are tangential to an internal counter bore. These nozzles set the air in a vortex motion. This spinning stream of air turns 90° 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.

1.3 Units of Refrigeration and Coefficient of performance

– The unit of refrigeration is expressed in terms of 'tonnes of refrigeration' (TR). "A tonne of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tonne (900 kg) of ice at 0°C in 24 hours." Since the latent heat of ice is 335 kJ/kg, therefore one tonne of refrigeration,

$$1TR = 900 \times 335\, kJ\ in\ 24\ hours$$

$$= \frac{900 \times 335}{24 \times 60} = 209.375\, kJ/\min \approx 210\, kJ/\min\ or\ 3.5\, kJ/s$$

Coefficient of Performance of a Refrigerator

– The coefficient of performance is the ratio of heat extracted in the refrigerator to the work done. Mathematically,

$$C.O.P = \frac{Q}{W}$$

Where, Q = Amount of heat extracted in the refrigerator (or the amount of refrigeration produced, or the capacity of a refrigerator)

W = Amount of work done

1.4 Applications of Refrigeration and air conditioning

– Refrigeration deals with cooling of bodies or fluids to temperatures lower than those of surroundings. This involves absorption of heat at a lower temperature and rejection to higher temperature of the surroundings. In olden days, the main purpose of refrigeration was to produce ice, which was used for cooling beverages, food preservation and refrigerated transport etc. Now-a-days refrigeration and air conditioning find so many applications that they have become very essential for mankind, and without refrigeration and air conditioning the basic fabric of the society will be adversely affected.

- Refrigeration and air conditioning are generally treated in a single subject due to the fact that one of the most important applications of refrigeration is in cooling and dehumidification as required for summer air conditioning. Of course, refrigeration is required for many applications other than air conditioning, and air conditioning also involves processes other than cooling and dehumidification. The temperature range of interest in refrigeration extends down to about –100°C. At lower temperatures cryogenic systems are more economical. Now-a-days refrigeration has become an essential part of food chain- from post-harvest heat removal to processing, distribution and storage. Refrigeration has become essential for many chemical and processing industries to improve the standard, quality, precision and efficiency of many manufacturing processes. Ever-new applications of refrigeration arise all the time. Some special applications require small capacities but are technically intriguing and challenging.
- Air-conditioning has made the living conditions more comfortable, hygienic and healthy in offices, work places and homes. As mentioned in Lesson 1, air-conditioning involves control of temperature, humidity, cleanliness of air and its distribution to meet the comfort requirements of human beings and/or some industrial requirements. Air-conditioning involves cooling and dehumidification in summer months; this is essentially done by refrigeration. It also involves heating and humidification in cold climates, which is conventionally done by a boiler unless a heat pump is used.
- The major applications of refrigeration can be grouped into following four major equally important areas.
  1. Food processing, preservation and distribution
  2. Chemical and process industries
  3. Special Applications
  4. Comfort air-conditioning

1.4.1 Food processing, preservation and distribution

1. Storage of Raw Fruits and Vegetables:

- It is well-known that some bacteria are responsible for degradation of food, and enzymatic processing cause ripening of the fruits and vegetables. The growth of bacteria and the rate of enzymatic processes are reduced at low temperature. This helps in reducing the spoilage and improving the shelf life of the food. Table 1.1 shows useful storage life of some plant and animal tissues at various temperatures. It can be seen that the storage temperature affects the useful storage life significantly. In general the storage life of most of the food products depends upon water activity, which essentially depends upon the presence of water in liquid form in the food product and its temperature. Hence, it is possible to preserve various food products for much longer periods under frozen conditions.

Table 1.1 Effect of storage temperature on useful storage life of food products

Food Product	Average useful storage life (days)		
	0°C	22°C	38°C
Meat	6-10	1	< 1
Fish	2-7	1	< 1
Poultry	5-18	1	< 1
Dry meats and fish	> 1000	> 350 & < 1000	> 100 & < 350
Fruits	2 - 180	1 - 20	1 - 7
Dry fruits	> 1000	> 350 & < 1000	> 100 & < 350
Leafy vegetables	3 - 20	1 - 7	1 - 3
Root crops	90 - 300	7 - 50	2 - 20
Dry seeds	> 1000	> 350 & < 1000	> 100 & < 350

– In case of fruits and vegetables, the use of refrigeration starts right after harvesting to remove the post-harvest heat, transport in refrigerated transport to the cold storage or the processing plant. A part of it may be stored in cold storage to maintain its sensory qualities and a part may be distributed to retail shops, where again refrigeration is used for short time storage. Depending upon the size, the required capacity of refrigeration plants for cold storages can be very high. Ammonia is one of the common refrigerants used in cold storages. Figure 1.10 shows the photograph of ammonia based refrigerant plant for a cold storage. Household refrigerator is the user end of cold chain for short time storage.

Fig.1.10 Ammonia based refrigeration plant for a large cold storage

– The cold chain has proved to be very effective in reducing spoilage of food and in food preservation. It is estimated that in India, the post-harvest loss due to inadequate cold storage facilities is high as 30 percent of the total output. The quality of remaining 70 percent is also affected by inadequate cold chain facilities. This shows the importance of proper refrigeration facilities in view of the growing food needs of the ever-growing population. Refrigeration helps in retaining the sensory, nutritional and eating qualities of the food. The excess crop of fruits and vegetables can be stored for use during peak demands and off-season; and transported to remote locations by refrigerated transport.

– In India, storage of potatoes and apples in large scale and some other fruits and vegetables in small scale and frozen storage of peas, beans, cabbage, carrots etc. has improved the standard of living. In general, the shelf life of most of the fruits and vegetables increases by storage at temperatures between 0 to 10°C.

– Table 1.2 shows the typical storage conditions for some fruits and vegetables as recommended by ASHRAE. Nuts, dried fruits and pulses that are prone to bacterial deterioration can also be stored for long periods by this method. The above mentioned fruits, vegetables etc, can be stored in raw state. Some highly perishable items require initial processing before storage. The fast and busy modern day life demands ready-to-eat frozen or refrigerated food packages to eliminate the preparation and cooking time. These items are becoming very popular and these require refrigeration plants.

2. Fish:

– Icing of fish according to ASHRAE Handbook on Applications started way back in 1938. In India, iced fish is still transported by rail and road, and retail stores store it for short periods by this method. Freezing of fish aboard the ship right after catch results in better quality than freezing it after the ship docks. In some ships, it is frozen along with seawater since it takes months before the ships return to dock. Long-term preservation of fish requires cleaning, processing and freezing.

Table 1.2. Recommended storage conditions for fruits and vegetables

	Storage Temperature °C	Relative Humidity %	Maximum recommended storage time	Storage time in cold storages for vegetables in tropical countries
Apples	0 - 4	90 - 95	2 - 6 months	
Beetroot	0	95 - 99	4 - 6 months	
Cabbage	0	95 - 99	5 - 6 months	2 months
Carrots	0	98 - 100	5 - 9 months	2 months
Cauliflower	0	95	3 - 4 weeks	1 week
Cucumber	10 - 13	90 - 95	10 - 14 days	
Eggplant	8 - 12	90 - 95	7 days	
Lettuce	0	95 - 100	2 - 3 weeks	
Melons	7 - 10	90 - 95	2 weeks	
Mushrooms	0 - 4	95	2 - 5	1 day
Onions	0	65 - 70	6 - 8 months	
Oranges	0 - 4	85 - 90	3 - 4 months	
Peas, Green	0	95 - 98	1 - 2 weeks	
Pears	0	90 - 95	2 - 5 months	
Potatoes	4 - 16	90 - 95	2 - 8 months	
Pumpkin	10 - 13	70 - 75	6 - 8 months	
Spinach	0	95	1 - 2 weeks	1 week
Tomatoes	13 – 21	85 - 90	1 - 2 weeks	1 week

3. Meat and poultry:

– These items also require refrigeration right after slaughter during processing, packaging. Short-term storage is done at 0°C. Long-term storage requires freezing and storage at -25°C.

4. Dairy Products:

– The important dairy products are milk, butter, buttermilk and ice cream. To maintain good quality, the milk is cooled in bulk milk coolers immediately after being taken from cow. Bulk milk cooler is a large refrigerated tank that cools it between 10 to 15^oC. Then it is transported to dairy farms, where it is pasteurized. Pasteurization involves heating it to 73^oC and holding it at this temperature for 20 seconds. Thereafter, it is cooled to 3 to 4^oC. The dairies have to have a very large cooling capacity, since a large quantity of milk has to be immediately cooled after arrival. During the lean period, the refrigeration plants of dairies are used to produce ice that is used during peak periods to provide cooling by melting. This reduces the required peak capacity of the refrigeration plant.
– Ice cream manufacture requires pasteurization, thorough mixing, emulsification and stabilization and subsequently cooling to 4 to 5^oC. Then it is cooled to temperature of about – 5^oC in a freezer where it stiffens but still remains in liquid state. It is packaged and hardened at –30 to –25^oC until it becomes solid; and then it is stored at same temperature.
– Buttermilk, curd and cottage cheese are stored at 4 to 10^oC for increase of shelf life. Use of refrigeration during manufacture of these items also increases their shelf life. There are many varieties of cheese available these days. Adding cheese starter like lactic acid and several substances to the milk makes all of these. The whey is separated and solid part is cured for a long time at about 10^oC to make good quality cheese.

5. Beverages**:**

– Production of beer, wine and concentrated fruit juices require refrigeration. The taste of many drinks can be improved by serving them cold or by adding ice to them. This has been one of the favourite past time of aristocracy in all the countries. Natural or man-made ice for this purpose has been made available since a very long time. Fruit juice concentrates have been very popular because of low cost, good taste and nutritional qualities. Juices can be preserved for a longer period of time than the fruits. Also, fruit juice concentrates when frozen can be more easily shipped and transported by road. Orange and other citrus juices, apple juice, grape juice and pineapple juice are very popular. To preserve the taste and flavor of juice, the water is driven out of it by boiling it at low temperature under reduced pressure. The concentrate is frozen and transported at –20^oC.
– Brewing and wine making requires fermentation reaction at controlled temperature, for example lager-type of beer requires 8 to12^oC while wine requires 27-30^oC. Fermentation is an exothermic process; hence heat has to be rejected at controlled temperature.

6. Candy:

– Use of chocolate in candy or its coating with chocolate requires setting at 5-10^oC otherwise it becomes sticky. Further, it is recommended that it be stored at low temperature for best taste.

7. Processing and distribution of frozen food:

– Many vegetables, meat, fish and poultry are frozen to sustain the taste, which nearly duplicates that of the fresh product. Freezing retains the sensory qualities of colour, texture and taste apart from nutritional qualities. The refrigeration systems for frozen food applications are very liberally designed, since the food items are frozen in shortest period of time. The sharp freezing with temperature often below –30^oC, is done so that the ice crystals formed during freezing do not get sufficient time to grow and remain small and do not pierce the cell boundaries and damage them. Ready-to-eat frozen foods, packed dinners and bakery items are also frozen by this method and stored at temperatures of –25 to -20^oC for distribution to retail stores during peak demands or off-season demands.

– Vegetables in this list are beans, corn, peas, carrots, cauliflower and many others. Most of these are blanched before freezing. There are various processes of freezing. Blast freezers give a blast of high velocity air at –30^oC on the food container. In contact freezing, the food is placed between metal plates and metal surfaces that are cooled to −30^oC or lower. Immersion freezing involves immersion of food in low temperature brine. Individual quick freezing (IQF) is done by chilled air at very high velocities like 5-10 m/s that keeps the small vegetable particles or shrimp pieces floating in air without clumping, so that maximum area is available for heat transfer to individual particles. The frozen particles can be easily packaged and transported. The refrigeration capacities in all the freezers are very large since freezing of large quantities is done in a very short time. Liquid nitrogen and carbon dioxide are also used for freezing.
– Of late supermarket refrigeration is gaining popularity all over the world. At present this constitutes the largest sector of refrigeration in developed countries. In a typical supermarket a large variety of products are stored and displayed for sale. Since a wide variety of products are stored, the required storage conditions vary widely. Refrigeration at temperatures greater than 0^oC and less than 0^oC is required, as both frozen and fresh food products are normally stored in the same supermarket. Figure 1.11 shows the photograph of a section of a typical supermarket. Refrigeration systems used for supermarkets have to be highly reliable due to the considerable value of the highly perishable products. To ensure proper refrigeration of all the stored products, a large of refrigerant tubing is used, leading to large refrigerant inventory.

Fig.1.11 Section of a supermarket with refrigerated display cases

1.4.2 Applications of refrigeration in chemical and process industries

– The industries like petroleum refineries, petrochemical plants and paper pulp industries etc. require very large cooling capacities. The requirement of each industry-process wise and equipment-wise is different hence refrigeration system has to be customized and optimized for individual application. The main applications of refrigeration in chemical and process industries involve the following categories.

1. Separation of gases:

– In petrochemical plant, temperatures as low as –150°C with refrigeration capacities as high as 10,000 Tons of Refrigeration (TR) are used for separation of gases by fractional distillation. Some gases condense readily at lower temperatures from the mixtures of hydrocarbon. Propane is used as refrigerant in many of these plants.

2. Condensation of gases:

– Some gases that are produced synthetically are condensed to liquid state by cooling, so that these can be easily stored and transported in liquid state. For example, in synthetic ammonia plant, ammonia is condensed at –10 to 10°C before filling in the cylinders, storage and shipment. This low temperature requires refrigeration.

3. Dehumidification of Air:

– Low humidity air is required in many pharmaceutical industries. It is also required for air liquefaction plants. This is also required to prevent static electricity and prevents short circuits in places where high voltages are used. The air is cooled below its dew point temperature, so that some water vapour condenses out and the air gets dehumidified.

4. Solidification of Solute:

– One of the processes of separation of a substance or pollutant or impurity from liquid mixture is by its solidification at low temperature. Lubricating oil is dewaxed in petroleum industry by cooling it below –25°C. Wax solidifies at about –25°C.

5. Storage as liquid at low pressure:

– Liquid occupies less space than gases. Most of the refrigerants are stored at high pressure. This pressure is usually their saturation pressure at atmospheric temperature. For some gases, saturation pressure at room temperature is very high hence these are stored at relatively low pressure and low temperature. For example natural gas is stored at 0.7 bar gauge pressure and –130°C. Heat gain by the cylinder walls leads to boiling of some gas, which is compressed, cooled and expanded back to 0.7 bar gauge.

6. Removal of Heat of Reaction:

– In many chemical reactions, efficiency is better if the reaction occurs below room temperature. This requires refrigeration. If these reactions are exothermic in nature, then more refrigeration capacities are required. Production of viscose rayon, cellular acetate and synthetic rubber are some of the examples. Fermentation is also one of the examples of this.

7. Cooling for preservation:

– Many compounds decompose at room temperature or these evaporate at a very fast rate. Certain drugs, explosives and natural rubber can be stored for long periods at lower temperatures.

8. Recovery of Solvents:

– In many chemical processes solvents are used, which usually evaporate after reaction. These can be recovered by condensation at low temperature by refrigeration system. Some of the examples are acetone in film manufacture and carbon tetrachloride in textile production.

1.4.3 Special applications of refrigeration

In this category we consider applications other than chemical uses. These are in manufacturing processes, applications in medicine, construction units etc.

1. Cold Treatment of Metals:

– The dimensions of precision parts and gauge blocks can be stabilized by soaking the product at temperature around – 90°C. The hardness and wear resistance of carburized steel can be increased by this process. Keeping the cutting tool at –100°C for 15 minutes can also increase the life of cutting tool. In deep drawing process the ductility of metal increases at low temperature. Mercury patterns frozen by refrigeration can be used for precision casting.

2. Medical:

– Blood plasma and antibiotics are manufactured by freeze-drying process where water is made to sublime at low pressure and low temperature. This does not affect the tissues of blood. Centrifuges refrigerated at –10°C, are used in the manufacture of drugs. Localized refrigeration by liquid nitrogen can be used as anesthesia also.

3. Ice Skating Rinks:

– Due to the advent of artificial refrigeration, sports like ice hockey and skating do not have to depend upon freezing weather. These can be played in indoor stadium where water is frozen into ice on the floor. Refrigerant or brine carrying pipes are embedded below the floor, which cools and freezes the water to ice over the floor.

4. Construction:

– Setting of concrete is an exothermic process. If the heat of setting is not removed the concrete will expand and produce cracks in the structure. Concrete may be cooled by cooling sand, gravel and water before mixing them or by passing chilled water through the pipes embedded in the concrete. Another application is to freeze the wet soil by refrigeration to facilitate its excavation.

5. Desalination of Water:

– In some countries fresh water is scarce and seawater is desalinated to obtain fresh water. Solar energy is used in some cases for desalination. An alternative is to freeze the seawater. The ice thus formed will be relatively free of salt. The ice can be separated and thawed to obtain fresh water.

6. Ice Manufacture:

– This was the classical application of refrigeration. Ice was manufactured in plants by dipping water containers in chilled brine and it used to take about 36 hours to freeze all the water in cans into ice. The ice thus formed was stored in ice warehouses. Now that small freezers and icemakers are available. Hotels and restaurants make their own ice, in a hygienic manner. Household refrigerators also have the facility to make ice in small quantities. The use of ice warehouses is dwindling because of this reason. Coastal areas still have ice plants where it is used for transport of iced fish.

- Refrigeration systems are also required in remote and rural areas for a wide variety of applications such as storage of milk, vegetables, fruits, foodgrains etc., and also for storage of vaccines etc. in health centers. One typical problem with many of the rural and remote areas is the continuous availability of electricity. Since space is not constraint, and most of these areas in tropical countries are blessed with alternate energy sources such as solar energy, biomass etc., it is preferable to use these clean and renewable energy sources in these areas.
- Thermal energy driven absorption systems have been used in some instances. Vapour compression systems that run on photovoltaic (PV) cells have also been developed for small applications. Figure 1.12 shows the schematic of solar PV cell driven vapour compression refrigeration system for vaccine storage.

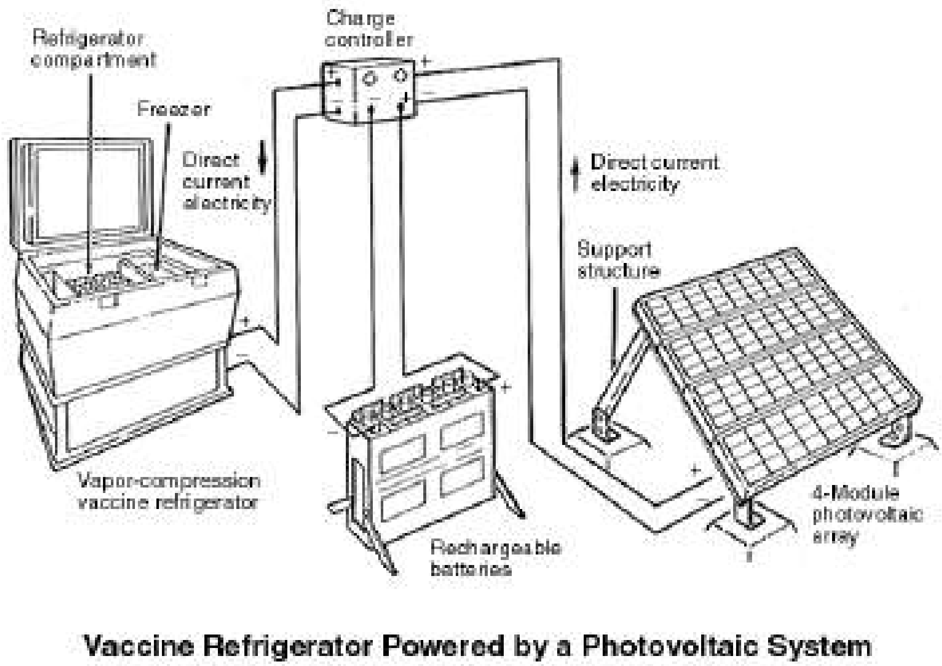


Fig. 1.12 Solar energy driven refrigeration system for vaccine storage

1.4.4 Application of air conditioning:

- Air-conditioning is required for improving processes and materials apart from comfort air-conditioning required for comfort of persons. The life and efficiency of electronic devices increases at lower temperatures. Computer and microprocessor-based equipment also require air-conditioning for their efficient operation. Modern electronic equipment with Very Large Scale Integrated (VLSI) chips dissipates relatively large quantities of energy in a small volume. As a result, unless suitable cooling is provided, the chip temperature can become extremely high. As the computing power of computers increases, more and more cooling will be required in a small volume. Some supercomputers required liquid nitrogen for cooling.

– Air-conditioning applications can be divided into two categories, namely, industrial and comfort air-conditioning.

1. **Industrial Air-conditioning:**

The main purpose of industrial air conditioning systems is to provide conducive conditions so that the required processes can be carried out and required products can be produced. Of course, the industrial air conditioning systems must also provide at least a partial measure of comfort to the people working in the industries. The applications are very diverse, involving cooling of laboratories down to -40^{o}C for engine testing to cooling of farm animals. The following are the applications to name a few.

1. Laboratories: This may involve precision measurement to performance testing of materials, equipment and processes at controlled temperature and relative humidity. Laboratories carrying out research in electronics and biotechnology areas require very clean atmosphere. Many laboratories using high voltage like in LASERS require very low humidity to avoid the sparking.

2. Printing: Some colour printing presses have one press for each colour. The paper passes from one press to another press. The ink of one colour must get dried before it reaches the second press, so that the colours do not smudge. And the paper should not shrink, so that the picture does not get distorted. This requires control over temperature as well humidity. Improper humidity may cause static electricity, curling and buckling of paper.

3. Manufacture of Precision Parts: If the metal parts are maintained at uniform temperature during manufacturing process, these will neither expand nor shrink, maintaining close tolerances. A lower relative humidity will prevent rust formation also. A speck of dust in a switch or relay can cause total or partial malfunction in spacecraft. The manufacture of VLSI chips, microprocessors, computers, aircraft parts, Micro-Electro Mechanical Systems (MEMS), nanomaterial fabrication and many areas of modern progress require a very clean atmosphere and proper control over humidity. Any impurity in the atmosphere will spoil the VLSI chips. The concept of Clean rooms has been introduced for such industries. In fact, all precision industries that use microprocessors require these clean rooms.

4. Textile Industry: The yarn in the textile industry is spun and it moves over spools at very high speeds in modern machines. It is very sensitive to humidity. The generation of static electricity should be avoided. Its flexibility and strength should not change. If it breaks during the process, the plant will have to be stopped and yarn repaired before restarting the plant.

5. Pharmaceutical Industries: In these industries to obtain sterile atmosphere, the airborne bacteria and dust must be removed in the air-conditioning system by filters. These industries require clean rooms. If capsules are made or used in the plant, then air has to be dry otherwise the gelatin of capsules will become sticky.

6. Photographic Material: The raw material used for filmmaking has to be maintained at low temperature, since it deteriorates at high temperature and humidity. The film also

has to be stored at low temperature. The room where film is developed requires 100% replacement by fresh air of the air polluted by chemicals.

7. Farm Animals: The yield of Jersey cows decreases drastically during summer months. Low temperature results in more efficient digestion of food and increase in weight of cow and the milk yield. Animal barns have to be ventilated in any case since their number density is usually very large. In many countries evaporative cooling is used for creating comfort conditions in animal houses.

8. Computer Rooms: These require control of temperature, humidity and cleanliness. The temperature of around 25^{o}C and relative humidity of 50% is maintained in these rooms. The dust spoils the CD drives and printers etc.; hence the rooms have to be kept clean also by using micro filters in the air-conditioning system.

9. Power Plants: Most of the modern power plants are microprocessor controlled. In the earlier designs, the control rooms were very large and were provided with natural ventilation. These days the control rooms are very compact, hence these require air-conditioning for persons and the microprocessors.

10. Vehicular Air-conditioning: Bus, tram, truck, car, recreational vehicle, crane cabin, aircraft and ships all require air-conditioning. In bus, tram, aircraft and ship, the occupancy density is very high and the metabolic heat and water vapour generated by persons has to be rejected. The cooling load in these is very high and rapidly changes that provides a challenge for their design.

2. **Comfort Air-Conditioning:**

– Energy of food is converted into chemical energy for functioning of brain, lungs, heart and other organs and this energy is ultimately rejected to the surroundings. Also the internal organs require a temperature close to 35^{o}C for their efficient operation, and regulatory mechanisms of human body maintain this temperature by rejecting appropriate amount of heat. Human beings do not feel comfortable if some extra effort is required by the body to reject this energy. The air temperature, humidity and velocity at which human body does not have to take any extra action, is called comfort condition or neutral condition.

– The residences, offices, shopping centers, stores, large buildings, theatres, auditorium etc. all have slightly different requirements and require different design. The required cooling capacities also vary widely depending upon the application. The factory assembled room air conditioners are very widely used for small residences, offices etc. these units are available as window type or split type. The capacities of these systems vary from a fraction of a ton (TR) to about 2 TR. These systems use a vapour compression refrigeration system with a sealed compressor and forced convection type evaporators and condensers.

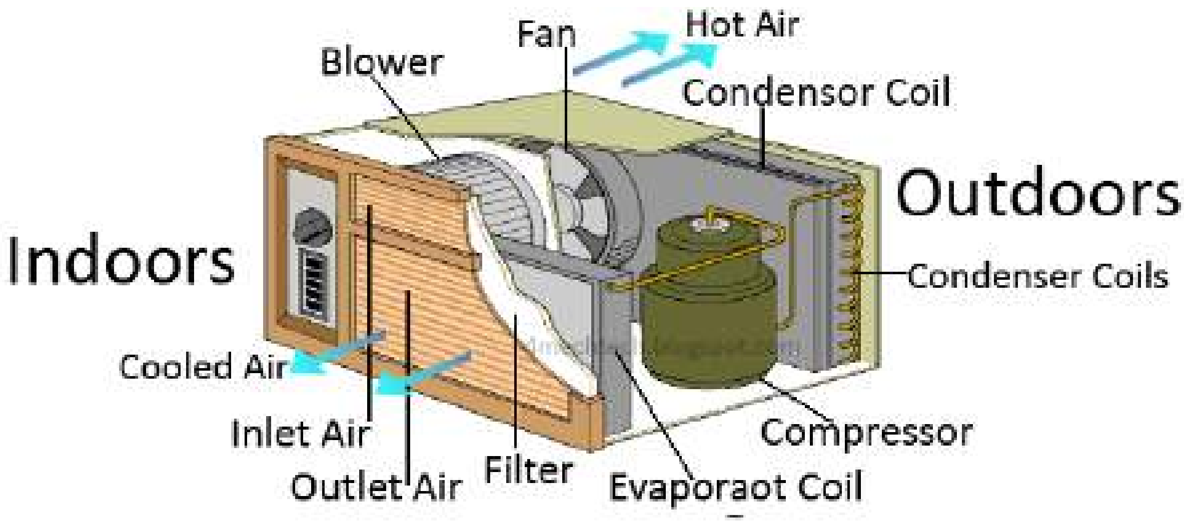


Fig. 1.13 Window air conditioner

– Figure 1.13 shows the schematic of a widow type room air conditioner. In this type all the components are housed in a single outer casing. In a split type air conditioner, the compressor and condenser with fan (commonly known as condensing unit) are housed in a separate casing and are kept away from the indoor unit consisting of the evaporator, blower, filter etc. The outdoor and indoor units are connected by refrigerant piping. For medium sized buildings factory assembled package units are off by various devices and materials. Special quality construction and filters are used for this purpose.

– Restaurants, theatres and other places of amusement require air-conditioning for the comfort of patrons. All places where, a large number of people assemble should have sufficient supply of fresh air to dilute CO_2 and body odours emitted by persons. In addition, people dissipate large quantities of heat that has to be removed by air-conditioning for the comfort of persons. These places have wide variation in air-conditioning load throughout the day. These have large number of persons, which add a lot of water vapour by respiration and perspiration. The food cooked and consumed also adds water vapour. This vapour has to be removed by air-conditioning plant. Hence, these buildings have large latent heat loads. Infiltration of warm outdoor is also large since the large numbers of persons enters and leave the building leading to entry of outdoor air with every door opening. Ventilation requirement is also very large.

– Air-conditioning in stores and supermarkets attracts more customers, induces longer period of stay and thereby increases the sales. Supermarkets have frozen food section, refrigerated food section, dairy and brewage section, all of them requiring different temperatures. The refrigeration system has to cater to different temperatures, apart from air-conditioning. These places also have a wide variation in daily loads depending upon busy and lean hours, and holidays.

– Large commercial buildings are a world of their own; they have their own shopping center, recreation center, gymnasium swimming pool etc. Offices have very high density of persons during office hours and no occupancy during off time. These buildings require integrated concept with optimum utilization of resources and services.

- These have security aspects, fire protection, emergency services, optimum utilization of energy all built-in. Modern buildings of this type are called intelligent buildings where air-conditioning requires large amount of energy and hence is the major focus.
- Since persons have to spend a major part of their time within the building, without much exposure to outdoors, the concept of Indoor Air Quality (IAQ) has become very important. There are a large number of pollutants that are emitted by the materials used in the construction of buildings and brought into the buildings. IAQ addresses to these issues and gives recommendation for their reduction to safe limits. Sick building syndrome is very common in poorly designed air conditioned buildings due to inadequate ventilation and use of improper materials. The sick building syndrome is characterized by the feeling of nausea, headache, eye and throat irritation and the general feeling of being uncomfortable with the indoor environment. In developed countries this is leading to litigation also.
- In the earlier systems little attention was paid to energy conservation, since fuels were abundant and inexpensive. The energy crisis in early seventies, lead to a review of basic principles and increased interest in energy optimization. The concept of low initial cost with no regard to operating cost has become obsolete now. Approaches, concepts and thermodynamic cycles, which were considered impractical at one time, are receiving serious considerations now.
- Earlier, the index of performance used to be first law efficiency, now in addition to that; the second law efficiency is considered so that the available energy utilized and wasted can be clearly seen. Concepts of hybrid cycles, heat recovery systems, alternate refrigerants and mixtures of refrigerants are being proposed to optimize energy use. Large-scale applications of air-conditioning in vast office and industrial complexes and increased awareness of comfort and indoor air quality have lead to challenges in system design and simulations. Developments in electronics, controls and computers have made refrigeration and air-conditioning a high-technology industry.

2

Refrigerants

Course Contents

2.1 Introduction

- Refrigerants are used as working substances in refrigeration systems. It is heat carrying medium which absorbs heat at low temperature and low pressure from space or products being cooled and rejects heat at high temperature and high pressure to the atmosphere.
- A very large number of substances are available, which can be used as refrigerants. But in fact, there is always an unique refrigerant available which is most suited for given application and given system.
- The judicious choice of a refrigerant can be made be made based upon their thermodynamic, physical, chemical, practical, ecological and economic considerations. The natural ice and a mixture of ice and salt were the first refrigerants. In 1834, ether, ammonia, sulphur dioxide, methyl chloride and carbon dioxide came into use as refrigerants in compression cycle refrigeration machines. Most of the early refrigerant materials have been discarded for safety reasons or for lack of chemical or thermal stability. In the present days, many new refrigerants including halo-carbon compounds are used for air-conditioning and refrigeration applications.

2.2 Classification of Refrigerants

- Based upon the working principle, the refrigerants may be classified in two groups.
  1. Primary refrigerants 2. Secondary refrigerants

1. Primary refrigerants

- These are the substances that produce refrigerating effect by absorbing latent heat of evaporation at low temperatures and pressures. They directly take part in the refrigeration system and go through the cyclic processes of condensation and evaporation. e.g water, ammonia, HC, CFCs etc.

2. Secondary refrigerants

- These are the substances that are first cooled by primary refrigerants in refrigeration system and transported to produce refrigerating effect at a location away from the refrigeration plant. They do not go through the cyclic processes of condensation and evaporation in refrigeration system.
- Secondary refrigerants are also known under the name brines or antifreezes. Of course, if the operating temperatures are above $0^{o}C$, 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. 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. The temperature at which freezing of a brine takes place its 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. e.g chilled water, brines and glycol etc.

The primary refrigerants are further classified into the following four groups:

1. Halo-carbon or organic refrigerants,
2. Azeotrope refrigerants
3. Inorganic refrigerants
4. Hydro-carbon refrigerants.

2.4.1 Halo-carbon Refrigerants

The American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) identifies 42 halo-carbon compounds as refrigerants. The various halo-carbon refrigerants are now discussed in detail, as below:

1. R-11, Trichloro-monofluoro-methane (CCl_3F)

– It is stable, non-flammable and non-toxic.
– It has a low side pressure of 0.202 bar at -15°C and high side pressure of 1.2606 bar at 30°C. The latent heat at -15°C is 195 kJ/kg. The NBP (normal boiling point) at atmospheric pressure is 23.77°C.
– Due to its low operating pressures, this refrigerant is exclusively used in large centrifugal compressor systems of 200 TR and above.
– The leaks may be detected by using a soap solution, a halide torch or by using an electronic detector.
– R-11 is the safest solvent used as a flushing agent for cleaning the internal parts of a refrigerator compressor when overhauling systems. It is useful after a system had a motor burn out or after it has a great deal of moisture in the system. By flushing moisture from the system with R-11, evacuation time is shortened.
– The cylinder colour code for R-11 is orange.

2. R-12, Dichloro-difluoro-methane (CCl_2F_2).

– It is a colourless, almost odourless, non-toxic, non-corrosive, non-irritating and non-flammable.
– R-12 has a pressure of 0.82 bar at -15°C and a pressure of 6.4 bar at 30°C. The latent heat of R-12 at -15°C is 159 kJ/kg. Its NBP is -29°C.
– It is used in industrial and commercial applications such as refrigerators, freezers, water coolers, room and window air-conditioning units etc.
– Its principal use is found in reciprocating and rotary compressors, but its use in centrifugal compressors for large commercial air-conditioning is increasing.
– The leak may be detected by soap solution, halide torch or an electronic leak detector. Water is only slightly soluble in R-12. At -18°C, it will hold six parts per million by mass. The solution formed is very slightly corrosive to any of the common metals used in refrigerator construction. The addition of mineral oil to the refrigerant has no effect upon the corrosive action.
– R-12 is more critical as to its moisture content when compared to R-22 and R-502. It is soluble in oil down to -68°C. The oil will begin to separate at this temperature and due to its lightness than the refrigerant, it will collect on the surface of the liquid refrigerant.

The refrigerant is available in a variety of cylinder sizes and the cylinder colour code is white.

3. R-13, Monochloro-trifluoro-methane ($CClF_3$)

– The R-13 has a NBP of -81.4°C at atmospheric pressure and a critical temperature of + 28.8°C.
– This refrigerant is used for the low temperature side of cascade systems. It is suitable with reciprocating compressors.

4. R-14, Carbontetrafluoride (CF_4)

– The R-14 has a NBP of -128°C at atmospheric pressure and critical temperature of -45.5°C. It serves as an ultra-low temperature refrigerant for use in cascade systems.

5. R-21, Dichloro-monoflouro-methane ($CHCl_2F$)

– The R-21 has a NBP of +9°C. It has found its principal use in centrifugal compressor systems for relatively high temperature refrigeration requirements.

6. R-22, Monochloro-diflouro-methane ($CHClF_2$)

– It is stable and is non-toxic, non-corrosive, non-irritating and non-flammable.
– The NBP is -41°C. It has a latent heat of 216.5 kJ/kg at -15°C. The normal head pressure at 30°C is 10.88 bar.
– It is not necessary to use R-22 at below atmospheric pressures in order to obtain the low temperatures. The evaporator pressure of this refrigerant at -15°C is 1.92 bar.
– It is used for refrigeration installations that need a low evaporating temperature, as in fast freezing units which maintain a temperature of -29°C to -40°C. It is also used in air-conditioning units and in household refrigerators. It is used with reciprocating and centrifugal compressors.
– Since water mixes better with R-22 than R- 12 by a ratio of 3 to 1, therefore driers (dessicants) should be used to remove most of the moisture to keep water to a minimum. This refrigerant has good solubility in oil down to -9°C. However, the oil remains fluid enough to flow down the suction line at temperatures as low as -40°C. The oil will begin to separate at this point. Since oil is lighter, therefore it will collect on the surface of the liquid refrigerant.
– The leaks may be detected with a soap solution, a halide torch or with an electronic leak detector. The cylinder colour code for R-22 is green.

7. R-30, Methylene chloride (CH_2Cl_2)

– The R-30 is a clear, water-white liquid with a sweet, non-irritating odour similar to that of chloroform, non-flammable, non-explosive and non-toxic.
– Due to its high NBP of 39.8°C, this refrigerant may be stored in closed cans instead of in compressed gas cylinders.
– The high and low sides of refrigeration system using R-30 operate under a vacuum. Since the volume of vapour at suction conditions is very high, therefore the use of R-30 is restricted to rotary or centrifugal compressors.
– It is extensively used for air conditioning of theatres, auditoriums, and office buildings. Now-a-days, the refrigerant R-11 is used in place of R-30.

- In order to detect leaks in a system using R-30, the pressure must be increased above atmosphere. A halide torch is used for detecting leaks.

8. R-40, Methyl-chloride (CH_3Cl)

- The R-40 is a colourless liquid with a faint, sweet, and non-irritating odour, flammable and explosive when mixed with air in concentrations from 8.1 to 17.2%, non-corrosive in its pure state but it becomes corrosive in the presence of moisture. Thus aluminum, zinc and magnesium alloys should never be used with this refrigerant as they will corrode considerably and pollute the lubricating oil.
- Its NBP is -23.7°C and the usual condenser pressure is 5 to 6.8 bar. The latent heat of vaporisation at -15°C is 423.5 kJ/kg.
- It is a solvent for many materials used in ordinary refrigeration compressors, therefore rubber and gaskets containing rubber should never be used. However, synthetic rubber is not affected by R-40. Thus metallic or asbestos-fibre gaskets containing insoluble binders should be used. The mineral oils are soluble in this refrigerant to a small extent.
- This refrigerant has been used in domestic units with both reciprocating and rotary compressors and in commercial units with reciprocating compressors up to approximately 10 TR capacity. The leaks with R-40 may be detected by soap solution or electronic leak detector.

8. R-100, Ethyl chloride (C_2H_5Cl)

- The R-100 is a colourless, toxic and flammable liquid with low operating pressures. It has NBP of 13.1°C. Due to its low operating pressure, it is not used in refrigerating equipment.

9. R-113, Trichloro-trifluoro-ethane (CCl_2FCClF_2)

- The R-113 has NBP of 47.6°C. It is used in commercial and industrial air-conditioning with centrifugal compressor systems.
- Since this refrigerant has the advantage of remaining liquid at room temperatures and pressures, therefore it can be carried in sealed tins rather than cylinders.

10. R-114, Dichloro-tetrafluoro-ethane ($CClF_2CClF_2$)

- The R-114 has NBP of 3.6°C. At -15°C, it evaporates at a pressure of 0.54 bar and at +30°C it condenses at a pressure of 1.5 bar. Its latent heat of vaporisation at -15°C is 143 kJ/kg.
- It is non-toxic, non-explosive and non corrosive even in the presence of water. It is used in fractional power household refrigerating systems and drinking water coolers employing rotary-vane type compressors.

11. R-123, Dichloro- trifluoro methane (CF_3CHCl_2)

- The R-123 is a potential substitute to R-11. It has about 4.3°C higher boiling point than R-11. It is, therefore, a lower pressure replacement for R-11, thus having larger specific volume of suction vapour. Hence, its use results in 10 to 15% reduction in capacity, if used in existing R-11 centrifugal compressors.

12. R-134a, Tetrafluoro-ethane (CF_3CH_2F)

– The R-134a is considered to be the most preferred substitute for refrigerant R-12. Its boiling point is - 26.15° C which is quite close to the boiling point of R-12 which is -29°C at atmospheric pressure.
– Since the refrigerant R-134a has no chlorine atom, therefore this refrigerant has zero ozone depleting potential (ODP) and has 74% less global warming potential (GWP) as compared to R-12.
– It has lower suction pressure and large suction vapour volume. It is not soluble in mineral oil. Hence, for use in domestic refrigerators (with hermetic units), suitable synthetic oil is used.
– Prevent moisture from getting into the refrigeration system is essential. Since the molecules of R-134a are smaller than R-12, therefore a very sensitive leak detector is used to detect leaks.

13. R-152a, Difluoro-ethane (CH_3CHF_2)

– The R-152a has similar characteristics as R-134a except that R-152a has slight vacuum in the evaporator at -25°C and the discharge temperature is higher because of its high value of the ratio of specific heats.

2.4.2 Azeotrope Refrigerants

– The term `azetrope' refers to a stable mixture of refrigerants whose vapour and liquid phases retain identical compositions over a wide range of temperatures. However, these mixtures, usually, have properties that differ from either of their components. Some of refrigerant are discussed in detail:

1. R-500

– The R-500 is an azeotropic mixture of 73.8% R-12 (CCl_2F_2) and 26.2% of R-152 (CH_3CHF_2). It is non-flammable, low in toxicity and non-corrosive. It is used in both industrial and commercial applications but only in systems with reciprocating compressors. It has a fairly constant vapour pressure temperature curve which is different from the vaporizing curves for either R-152a or R-12.
– This refrigerant offers about 20% greater refrigerating capacity than R-12 for the same size of motor when used for the same purpose. The evaporator pressure of this refrigerant is 1.37 bar at -15°C and its condensing pressure is 7.78 bar at 30°C. It has a boiling point of - 33°C at atmospheric pressure. Its latent heat at -15°C is 192 kJ/kg.
– It can be used whenever a higher capacity than that obtained with R-12 is needed. The solubility of water in R-500 is highly critical. It has fairly high solubility with oil.
– The leakage may be detected by using soap solution, a halide torch, an electronic leak detector or a coloured tracing agent.
– It is necessary to keep moisture out of the system by careful dehydration and by using driers. The cylinder colour code for this refrigerant is yellow.

2. R-502

– The R-502 is an azeotropic mixture of 48.8% R-22 ($CHClF_2$) and 51.2% of R-115 ($CClF_2CF_3$). It is a non-flammable, non-corrosive, practically non-toxic liquid.

- It is suitable where temperatures from -18°C to -51°C are needed. It is often used in frozen food lockers, frozen food processing plants, frozen food display cases and in storage units for frozen foods and ice-cream.
- It is only used with reciprocating compressors. The boiling point of this refrigerant at atmospheric pressure is -46°C. Its evaporating pressure at -15°C is 2.48 bar and the condensing pressure at 30°C is 12.06 bar. Its latent heat at -29°C is 168.6 kJ/kg.
- The R-502 combines many of the good properties of R-12 and R-22. It gives a machine capacity equal to that of R-22 with just about the condensing temperature of a system using R-12.
- Since this refrigerant has a relatively low condensing pressure and temperature, therefore it increases the life of compressor valves and other parts. Better lubrication is possible because of the increased viscosity of the oil at low condensing temperature. It is possible to eliminate liquid injection to cool the compressor because of the low condensing pressure.
- The leaks may be detected by soap solution, halide torch or electronic leak detector. It will hold 1.5 times more moisture at -18°C than R-12. It has fair solubility in oil above 82°C. Below this temperature, the oil tries to separate and tends to collect on the surface of the liquid refrigerant. However, oil is carried back to the compressor at temperatures down to -40°C. The cylinder colour code for this refrigerant is orchid.

3. R-503

- The R-503 is an azeotropic mixture of 40.1% R-23 (CHF_3) and 59.9% of R-13 ($CClF_3$). This is a non-flammable, non-corrosive, practically non-toxic liquid.
- Its boiling temperature at atmospheric pressure is -88°C which is lower than either R-23 or R-13. Its evaporating pressure at -15°C is 17.15 bar. Its critical temperature is 20°C and its critical pressure is 41.15 bar.
- This is a low temperature refrigerant and good for use in the low state of cascade systems which require temperatures in the range of -73°C to -87°C. The latent heat of vaporisation at atmospheric pressure is 173 kJ/kg.
- The leaks in R-503 systems may be detected with the use of soap solution, a halide torch or an electronic leak detector.
- This refrigerant will hold more moisture than some other low temperature refrigerants. The oil does not circulate well at low temperatures. The cascade and other low temperature equipments are normally fitted with oil separators and other devices for returning the oil to the compressor. The cylinder colour code for R-503 is aquamarine.

4. R-504

- The R-504 is an azeotropic mixture of 48.2% R-32 (CH_2F_2) and 51.8 % R-115 ($CClF_2CF_3$). It is non-flammable, non-corrosive and non-toxic. The boiling temperature at atmospheric pressure is -57°C. Its evaporating pressure at -15°C is 5.88 bar and its critical pressure is 48 bar.

- As with all low temperature refrigerants, some difficulty may be experienced with the oil circulation. With the addition of 2 to 5% R-170 (ethane), the oil will be taken into the solution with the refrigerant and will circulate through the system with it.
- The leaks in R-504 systems may be easily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of -40°C to -62°C is desired. The cylinder colour code for R-504 is tan.

2.4.3 Inorganic Refrigerants

- The inorganic refrigerants were exclusively used before the introduction of halo-carbon refrigerants. These refrigerants are still in use due to their inherent thermodynamic and physical properties. The various inorganic refrigerants are given below in detail.

1. R-717 (Ammonia)

- It is one of the oldest and most widely used of all the refrigerants. It is a colourless gas. Its boiling point at atmospheric pressure is -33.3°C and its melting point from the solid is -78°C. The low boiling point makes it possible to have refrigeration at temperatures considerably below 0°C without using pressures below atmospheric in the evaporator.
- Its latent heat of vaporisation at -15°C is 1315 kJ/kg. Thus, large refrigerating effects are possible with relatively small sized machinery. The condenser pressure at 30°C is 10.78 bar. The condensers for R-717 are usually of water cooled type.
- It is a poisonous gas if inhaled in large quantities. In lesser quantities, it is irritating to the eyes, nose and throat. This refrigerant is somewhat flammable and, when mixed with air in the ratio of 16% to 25% of gas by volume, will form an explosive mixture.
- The leaks of this refrigerant may be quickly and easily detected by the use of burning sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphite. This refrigerant attacks copper and bronze in the presence of a little moisture but does not corrode iron or steel.
- Since the refrigerant R-717 is lighter than oil, therefore, its separation does not create any problem. The excess oil in the evaporator may be removed by opening a valve in the bottom of the evaporator. This refrigerant is used in large compression machines using reciprocating compressors and in many absorption type systems where toxicity is secondary.
- The use of this refrigerant is extensively found in cold storage, warehouse plants, ice cream manufacture, ice manufacture, beer manufacture, food freezing plants etc.

2. R-729 (Air)

- The dry air is used as a gaseous refrigerant in some compression systems, particularly in aircraft air-conditioning.

3. R-744 (Carbon dioxide)

- It is non-toxic, non- irritating and non-flammable. The boiling point of this refrigerant is so extremely low (-73.6°C) that at -15°C, a pressure of well over 20.7 bar is required to prevent its evaporation. At a condenser temperature of +30°C, a pressure of approximately 70 bar is required to liquify the gas. Its critical temperature is 31°C and triple point is -56.6°C.

- Due to its high operating pressure, the compressor of a carbon dioxide refrigerator unit is very small even for a comparatively large refrigerating capacity.
- However, because of its low efficiency, it is seldom used in household units, but is used in some industrial applications as dry ice and aboard ships.

4. R-764 (Sulphur dioxide)

- This refrigerant is produced by the combustion of sulphur in air. The boiling point of sulphur dioxide is -10°C at atmospheric pressure. The condensing pressure varies between 4.1 bar and 6.2 bar under normal operating conditions. The latent heat of sulphur dioxide at -15°C is 396 kJ/kg.
- It is a very stable refrigerant with a high critical temperature and it is non-flammable and non-explosive. It has a very unpleasant and irritating odour. This refrigerant is not injurious to food and is used commercially as a ripener and preservative of foods. It is, however, extremely injurious to flowers, plants and shrubbery.
- The sulphur dioxide in its pure state is not corrosive, but when there is moisture present, the mixture forms sulphurous acid which is corrosive to steel. Thus it is very important that the moisture in the refrigerating system be held to a minimum.
- The sulphur dioxide does not mix readily with oil. Therefore, oil lighter than that used with other refrigerants may be used in the compressors. The refrigerant in the evaporator with oil floating on the top has a tendency to have a higher boiling point than that corresponding to its pressure.
- The leaks in the system with sulphur dioxide may be easily detected by means of soap solution or ammonia swab. A dense white smoke forms when sulphur dioxide and ammonia fumes come in contact.

5. R-118 (Water)

- The principal refrigeration use of water is as ice. The high freezing temperature of water limits its use in vapour compression systems. It is used as the absorbent in absorption systems and in steam jet refrigeration system.

2.3 Nomenclature or Designation System for Refrigerants

- It is very difficult to remember and express large number of chemical formulas and their derivatives also. Thus Thomas Migley, Jr. and Charles Kettering carried the pioneer work of synthesizing series of CFCs and introduced simple convenient notation method to designate refrigerant which was also adopted by ASHRAE.
- The refrigerants are internationally designated as `R' followed by certain numbers such as R-11, R-12, R-114 etc.
- A refrigerant followed by a two-digit number indicates that a refrigerant is derived from methane base while three-digit number represents ethane base. The numbers assigned to each refrigerant have a special meaning.

– The general chemical formula for the refrigerant, either for methane or ethane base, is given as $C_mH_nCl_pF_q$, in which $n + p + q = 2m + 2$

where m = Number of carbon atoms,

n = Number of hydrogen atoms,

p = Number of chlorine atoms, and

q = Number of fluorine atoms.

As discussed above, the number of the refrigerant is given by R (m-1) (n + 1) (q).

Let us consider the following refrigerants to find its chemical formula and the number.

1. Dichloro-difluoro-methane (CCl_2F_2)

We see that in this refrigerant

Number of chlorine atoms, $p = 2$

Number of fluorine atoms, $q = 2$

Number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$0 + 2 + 2 = 2m + 2$ or $m = 1$

i.e. Number of carbon atoms = 1

Thus the chemical formula for dichloro-difluoro-methane becomes CCl_2F_2 and the number of refrigerant becomes R (1-1) (0+1) (2) or R-012 i.e. R-12.

2. Dichloro-tetrafluoro-ethane ($C_2Cl_2F_4$)

We see that in this refrigerant

Number of chlorine atoms, $p = 2$

Number of fluorine atoms, $q = 4$

Number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$0 + 2 + 4 = 2m + 2$ or $m = 2$

i.e. Number of carbon atoms = 2

Thus the chemical formula for dichloro-tetrafluoro-ethane becomes $C_2Cl_2F_4$ and the number of refrigerant becomes R (2-1) (0+1) (4) or R-114.

The inorganic refrigerants are designated by adding 700 to the molecular mass of the compound. For example, the molecular mass of ammonia is 17, therefore it is designated by R - (700 + 17) or R-717.

2.4 Desirable Properties of Refrigerants

– A refrigerant is said to be ideal if it has all of the following properties:

1. Low boiling and freezing point,
2. High critical pressure and temperature,
3. High latent heat of vaporisation,
4. Low specific heat of liquid and high specific heat of vapour,
5. Low specific volume of vapour,
6. High thermal conductivity,
7. Non-corrosive to metal,
8. Non-flammable and non-explosive,

9. Non-toxic,
10. Low cost,
11. Easily and regularly available,
12. Easy to liquify at moderate pressure and temperature,
13. Easy of locating leaks by odour or suitable indicator,
14. Mixes well with oil,
15. High coefficient of performance, and
16. Ozone friendly.

Selection of refrigerant for a particular application is based on the following requirements:

i. Thermodynamic and thermo-physical properties
ii. Environmental and safety properties, and
iii. Economics

2.4.1 Thermodynamic and thermo-physical properties

– Following are the thermodynamic and thermo physical requirements:

1. Suction pressure: At a given evaporator temperature, the saturation pressure should be above atmospheric for prevention of air or moisture ingress into the system and ease of leak detection. Higher suction pressure is better as it leads to smaller compressor displacement
2. Discharge pressure: At a given condenser temperature, the discharge pressure should be as small as possible to allow light-weight construction of compressor, condenser etc.
3. Pressure ratio: Should be as small as possible for high volumetric efficiency and low power consumption
4. Latent heat of vaporization: Should be as large as possible so that the required mass flow rate per unit cooling capacity will be small

The above requirements are somewhat contradictory, as the operating pressures, temperatures and latent heat of vaporization are related by Clausius-Clapeyron Equation:

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

In the above equation, P_{sat} is the saturation pressure (in atm.) at a temperature T (in Kelvin), h_{fg} and s_{fg} are enthalpy and entropy of vaporization and R is the gas constant. Since the change in entropy of vaporization is relatively small, from the above equation it can be shown that:

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

In the above equation, P_c and P_e are the condenser and evaporator pressures, T_c and T_e are condenser and evaporator temperatures. 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.

5. Isentropic index of compression: Should be as small as possible so that the temperature rise during compression will be small.

6. Liquid specific heat: Should be small so that degree of subcooling will be large leading to smaller amount of flash gas at evaporator inlet.
7. Vapour specific heat: Should be large so that the degree of superheating will be small
8. Thermal conductivity: Thermal conductivity in both liquid as well as vapour phase should be high for higher heat transfer coefficients.
9. Viscosity: Viscosity should be small in both liquid and vapour phases for smaller frictional pressure drops.

- The thermodynamic properties are interrelated and mainly depend on normal boiling point, critical temperature, molecular weight and structure.
- The normal boiling point indicates the useful temperature levels as it is directly related to the operating pressures. A high critical temperature yields higher COP due to smaller compressor superheat and smaller flash gas losses. On the other hand since the vapour pressure will be low when critical temperature is high, the volumetric capacity will be lower for refrigerants with high critical temperatures. This once again shows a need for trade-off between high COP and high volumetric capacity. It is observed that for most of the refrigerants the ratio of normal boiling point to critical temperature is in the range of 0.6 to 0.7. Thus the normal boiling point is a good indicator of the critical temperature of the refrigerant.
- The important properties such as latent heat of vaporization and specific heat depend on the molecular weight and structure of the molecule. Trouton's rule shows that the latent heat of vaporization will be high for refrigerants having lower molecular weight. The specific heat of refrigerant is related to the structure of the molecule. If specific heat of refrigerant vapour is low then the shape of the vapour dome will be such that the compression process starting with a saturated point terminates in the superheated zone (i.e, compression process will be dry). However, a small value of vapour specific heat indicates higher degree of superheat. Since vapour and liquid specific heats are also related, a large value of vapour specific heat results in a higher value of liquid specific heat, leading to higher flash gas losses. Studies show that in general the optimum value of molar vapour specific heat lies in the range of 40 to 100 kJ/kmol.K.
- The freezing point of the refrigerant should be lower than the lowest operating temperature of the cycle to prevent blockage of refrigerant pipelines.

2.4.2 Environmental and safety properties

- The environmental and safety properties are very important. In fact, at present the environment friendliness of the refrigerant is a major factor in deciding the usefulness of a particular refrigerant.
- The important environmental and safety properties are:

  1. 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. R 11, R 12) or will be phased-out 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

2. Global Warming Potential (GWP): Refrigerants should have as low a GWP value as possible to minimize the problem of global warming. Refrigerants with zero ODP but a high value of GWP (e.g. R134a) are likely to be regulated in future.

3. 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.

4. 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. Thus 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. 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-$COCl_2$). 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

Thus from toxicity point-of-view, the usefulness of a particular refrigerant depends on the specific application.

5. Flammability: The refrigerants should preferably be non-flammable 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.

5. Chemical stability: The refrigerants should be chemically stable as long as they are inside the refrigeration system.

6. Compatibility with common materials of construction (both metals and non-metals)

7. Miscibility with lubricating oils: Oil separators have to be used if the refrigerant is not miscible with lubricating oil (e.g. ammonia). Refrigerants that are completely miscible with oils are easier to handle (e.g. R12). However, for refrigerants with limited solubility

(e.g. R 22) special precautions should be taken while designing the system to ensure oil return to the compressor

8. Dilelectric strength: This is an important property for systems using hermetic compressors. For these systems the refrigerants should have as high a dielectric strength as possible.

9. Ease of leak detection: In the event of leakage of refrigerant from the system, it should be easy to detect the leaks.

10. The refrigerant used should preferably be inexpensive and easily available

2.5 Comparison between different refrigerants

– Synthetic refrigerants that were commonly used for refrigeration, cold storage and air conditioning applications are: R 11 (CFC 11), R 12 (CFC 12), R 22 (HCFC 22), R 502 (CFC 12+HCFC 22) etc. However, these refrigerants have to be phased out due to their Ozone Depletion Potential (ODP). The synthetic replacements for the older refrigerants are: R-134a (HFC-134a) and blends of HFCs. Generally, synthetic refrigerants are non-toxic and non-flammable. However, compared to the natural refrigerants the synthetic refrigerants offer lower performance and they also have higher Global Warming Potential (GWP). As a result, the synthetic refrigerants face an uncertain future. The most commonly used natural refrigerant is ammonia. This is also one of the oldest known refrigerants. Ammonia has good thermodynamic, thermophysical and environmental properties. However, it is toxic and is not compatible with some of the common materials of construction such as copper, which somewhat restricts its application. Other natural refrigerants that are being suggested are hydrocarbons (HCs) and carbon di-oxide (R-744). Though these refrigerants have some specific problems owing to their eco-friendliness, they are being studied widely and are likely to play a prominent role in future.

– Prior to the environmental issues of ozone layer depletion and global warming, the most widely used refrigerants were: R 11, R 12, R 22, R 502 and ammonia. Of these, R 11 was primarily used with centrifugal compressors in air conditioning applications. R 12 was used primarily in small capacity refrigeration and cold storage applications, while the other refrigerants were used in large systems such as large air conditioning plants or cold storages. Among the refrigerants used, except ammonia, all the other refrigerants are synthetic refrigerants and are non-toxic and non-flammable. Though ammonia is toxic, it has been very widely used due to its excellent thermodynamic and thermophysical properties. The scenario changed completely after the discovery of ozone layer depletion in 1974. The depletion of stratospheric ozone layer was attributed to chlorine and bromine containing chemicals such as Halons, CFCs, HCFCs etc. Since ozone layer depletion could lead to catastrophe on a global level, it has been agreed by the global community to phase out the ozone depleting substances (ODS). As a result except ammonia, all the other refrigerants used in cold storages had to be phased-out and a search for suitable replacements began in earnest. At the same time, it was also observed that in addition to ozone layer depletion, most of the conventional synthetic refrigerants also cause significant global warming. In view of the environmental problems caused by the synthetic refrigerants, opinions differed on replacements for conventional refrigerants. The alternate refrigerants can be classified into two broad groups: i) Non-ODS, synthetic refrigerants based on Hydro-Fluoro-Carbons (HFCs) and their blends ii) Natural refrigerants including ammonia, carbon dioxide, hydrocarbons and their blends.

– It should be noted that the use of natural refrigerants such as carbon dioxide, hydrocarbons is not a new phenomena, but is a revival of the once-used-and-discarded technologies in a much better form. Since the natural refrigerants are essentially making a comeback, one advantage of using them is that they are familiar in terms of their strengths and weaknesses. Another important advantage is that they are completely environment friendly, unlike the HFC based refrigerants, which do have considerable global warming potential. The alternate synthetic refrigerants are normally non-toxic and non-flammable. It is also possible to use blends of various HFCs to obtain new refrigerant mixtures with required properties to suit specific applications. However, most of these blends are non-azeotropic in nature, as a result there could be significant temperature glides during evaporation and condensation, and it is also important take precautions to prevent leakage, as this will change the composition of the mixture. Table 2.1 shows a list of refrigerants being replaced and their replacements.

Table 2.1 Application of refrigerants

Refrigerant	Application	Substitute suggested Retrofit (R) /New (N)
R 11(CFC) NBP = 23.7^{o}C H_{fg} at NBP=182.5 kJ/kg T_{cr} =197.98$_o$C C_p/C_v = 1.13 ODP = 1.0 GWP = 3500	Large air conditioning systems Industrial heat pumps As foam blowing agent	R 123 (R,N)
		R 141b (N)
		R 245fa (N)
		n-pentane (R,N)
R 12 (CFC) NBP = -29.8^{o}C h_{fg} at NBP=165.8 kJ/kg T_{cr} =112.04^{o}C C_p/C_v = 1.126 ODP = 1.0 GWP = 7300	Domestic refrigerators Small air conditioners Water coolers Small cold storages	R 22 (R,N)
		R 134a (R,N)
		R 227ea (N)
		R 401A,R 401B (R,N)
		R 411A,R 411B (R,N)
		R 717 (N)
R 22 (HCFC) NBP = -40.8^{o}C h_{fg} at NBP=233.2 kJ/kg T_{cr} =96.02^{o}C C_p/C_v = 1.166	Air conditioning systems Cold storages	R 410A, R 410B (N)
		R 417A (R,N)
		R 407C (R,N)
		R 507,R 507A (R,N)
		R 404A (R,N)

ODP = 0.05 GWP = 1500		R 717 (N)
R 134a (HFC) NBP = -26.15^oC h_{fg} at NBP=222.5 kJ/kg T_{cr} =101.06^oC C_p/C_v = 1.102 ODP = 0.0 GWP = 1200	Used as replacement for R 12 in domestic refrigerators, water coolers, automobile A/Cs etc	**No replacement required** Immiscible in mineral oils Highly hygroscopic
R 717 (NH_3) NBP = -33.35^oC h_{fg} at NBP=1368.9 kJ/kg T_{cr} =133.0^oC C_p/C_v = 1.31 ODP = 0.0 GWP = 0.0	Cold storages Ice plants Food processing Frozen food cabinets	**No replacement required** Toxic and flammable Incompatible with copper Highly efficient Inexpensive and available
R 744 (CO_2) NBP = -78.4^oC h_{fg} at 40^oC=321.3 kJ/kg T_{cr} =31.1^oC C_p/C_v = 1.3 ODP = 0.0 GWP = 1.0	Cold storages Air conditioning systems Simultaneous cooling and heating (Transcritical cycle)	**No replacement required** Very low critical temperature Eco-friendly Inexpensive and available
R718 (H_2O) NBP = 100.^oC h_{fg} at NBP=2257.9 kJ/kg T_{cr} =374.15^oC C_p/C_v = 1.33 ODP = 0.0 GWP = 1.0	Absorption systems Steam jet systems	**No replacement required** High NBP High freezing point Large specific volume Eco-friendly Inexpensive and available
R600a (iso-butane) NBP = -11.73^oC h_{fg} at NBP=367.7 kJ/kg T_{cr} =135.0^oC C_p/C_v = 1.086 ODP = 0.0 GWP = 3.0	Replacement for R 12 Domestic refrigerators Water coolers	**No replacement required** Flammable Eco-friendly

3

Air Refrigeration

Course Contents

3.1 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.
- The advent of high-speed passenger aircraft, jet aircraft and missiles has introduced the need for compact and simple refrigeration systems, capable of high capacity, with minimum reduction of pay load. When the power requirements, needed to transport the additional weight of the refrigerating system are taken into account, the air cycle systems usually prove to be the most efficient. The cooling demands per unit volume of space are heavy. An ordinary passenger aircraft requires a cooling system capable of 8 TR capacity and a super constellation requires a cooling system of more than 8 TR capacity. A jet lighter travelling at 950 km/s needs cooling system of 10 to 20 TR capacity.

3.2 Merits and Demerits of Air Refrigeration system

- Following are the merits and demerits of air refrigeration system.

Merits

1. Air is easily available and there is no cost of refrigerant.
2. The air is nontoxic and non-inflammable.
3. The leakage of air in small amount is tolerable.
4. Since the main compressor is employed for compressed air source, therefore there is no problem of space for extra compressor.
5. The air is light in weight per tonne of refrigeration.
6. The chilled air is directly used for cooling, thus by eliminating the cost of separate evaporator.
7. Since the pressure in the whole system is quite low, therefore piping, ducting quite simple to design, fabricate and maintain.

Demerits

1. It has low coefficient of performance.
2. The rate of air circulation is relatively large.

3.3 Reversed Carnot cycle and its limitations

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

  i. The working fluid is a fixed mass of air that behaves as an ideal gas.

  ii. 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

  iii. All the processes within the cycle are reversible, i.e., the cycle is internally reversible.

  iv. The specific heat of air remains constant throughout the cycle. 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.

– Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure 3.1(a) 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

  Process 1-2 Isentropic compression process: The air is compressed isentropically as shown by the curve 1-2 on p-v and T-s diagrams. During this process, the pressure of air increases from p_1 to p_2 and specific volume decrease from v_1 to v_2 and temperature increase from T_1 to T_2. During isentropic compression, no heat is absorbed or rejected by the air.

  Process 2-3 Isothermal compression process: The air is now compressed isothermally (i.e. at constant temperature, $T_2=T_3$) as shown by the curve 2-3 on p-v and T-s diagrams. During this process, the pressure of air increases from p_2 to p_3, and specific volume decreases from v_2 to v_3. Heat rejected by the air during isothermal compression per kg of air,

$$q_R = q_{2-3} = \text{Area 2-3-3'-2'} = T_3\left(s_2 - s_3\right) = T_2\left(s_2 - s_3\right)$$

  Process 3-4 Isentropic expansion process: The air is now expanded isentropically as shown by the curve 3-4 on p-v and T-s diagrams. The pressure of air decreases from p_3 to p_4, specific volume increases from v_3 to v_4 and the temperature decreases from T_3 to T_4. During isentropic expansion, no heat is absorbed or rejected by the air.

  Process 4-1 Isothermal expansion process: The air is now expanded isothermally (i.e. at constant temperature, $T_4=T_1$) as shown by the curve 4-1 on p-v and T-s diagrams. The pressure of air decreases from p_4 to p_1 and specific volume increases from v_4 to v_1. The heat absorbed by the air (or heat extracted from the cold body) during isothermal expansion per kg of air,

$$q_A = q_{4-1} == \text{Area 4-1-2'-3'} = T_4\left(s_2 - s_3\right) = T_1\left(s_2 - s_3\right)$$

  Work done during the cycle per kg of air,

$$w = q_R - q_A = q_{2-3} - q_{4-1}$$

  Coefficient of performance of the refrigeration system working on reversed Carnot cycle,

$$C.O.P = \frac{Heat\ absorbed}{Work\ done} = \frac{q_A}{w} = \frac{q_A}{q_R - q_A} = \frac{T_1\left(s_2 - s_3\right)}{\left(T_2 - T_1\right)\left(s_2 - s_3\right)}$$

$$C.O.P = \frac{T_1}{T_2 - T_1}$$

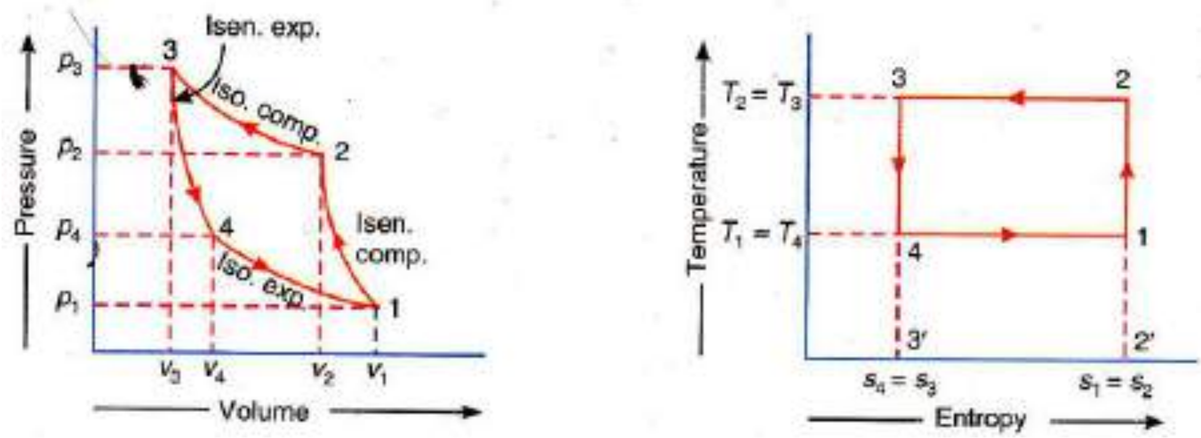


Fig. 3.1 Reverse Carnot refrigeration system in P-v and T-s coordinates

Limitations of reversed Carnot cycle

– 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.

Temperature Limitations of reversed Carnot cycle:

– C.O.P of the reversed Carnot cycle is $C.O.P = \frac{T_1}{T_2 - T_1}$ may be improved by,

1. Decreasing the higher temperature (i.e. temperature of hot body, T_2), or
2. Increasing the lower temperature (i.e. temperature of cold body, T_1).

– It may be noted that temperatures T_1 and T_2 cannot be varied at will, due to functional limitations. The higher temperature T_2 is the temperature of cooling water or air available for rejection of heat and the lower temperature T_1 is the temperature to be maintained in the refrigerator. The heat transfer will take place in the right direction only when the higher temperature is more than the temperature of cooling water or air to which heat is to be rejected, while the lower temperature must be less than the temperature of substance to be cooled.

– Thus, if the temperature of cooling water or air T_2 available for heat rejection is low, the C.O.P. of the Carnot refrigerator will be high. Since T_2 in winter is less than T_2 in summer, therefore, C.O.P. in winter will be higher than C.O.P. in summer. In other words, the Carnot refrigerators work more efficiently in winter than in summer. Similarly, if the lower temperature fixed by the refrigeration application is high, the C.O.P. of the Carnot refrigerator will be high. Thus a Carnot refrigerator used for making ice at 0^0C will have less C.O.P that a Carnot refrigerator used for air conditioned plant in summer at 20^0C when the atmospheric temperature is 40^0C. Thus Carnot C.O.P of a domestic refrigerator is less than the Carnot C.O.P of a domestic air conditioner.

3.4 Necessity of 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:
  1. Large internal heat generation due to occupants, equipment etc.
  2. Heat generation due to skin friction caused by the fast moving aircraft
  3. At high altitudes, the outside pressure will be sub-atmospheric.
  4. Solar radiation
- 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.
- 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.
- 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:
  1. Air is cheap, safe, non-toxic and non-flammable. Leakage of air is not a problem
  2. Cold air can directly be used for cooling thus eliminating the low temperature heat exchanger (open systems) leading to lower weight
  3. 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 equivalent vapours compression system.
  4. Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

3.5 Methods of air craft cooling system

The various methods of Air Refrigeration system for aircraft are as follow.

1. Simple air cooling system
2. Simple air evaporative cooling system
3. Boot strap cooling system
4. Boot strap air evaporative cooling system
5. Reduced ambient air cooling system
6. Regenerative air cooling system

3.5.1 Simple Air Cooling System

A simple air cooling system for aircrafts is shown in Fig. 3.1. The main components of this system are the main compressor driven by a gas turbine, a heat exchanger, a cooling turbine and a cooling air fan. The air required for refrigeration system is bled off from the main compressor. This high pressure and high temperature air is cooled initially in the heat exchanger where ram air is used for cooling. It is further cooled in the cooling turbine by the process of expansion. The work of this turbine is used to drive the cooling fan which draws cooling air through the heat exchanger. This system is good for ground surface cooling and for low flight speeds. The T-s diagram for a simple air cooling system is shown in Fig. 3.1 (b)

The various processes are discussed below:

1. Ramming process. Let the pressure and temperature of ambient air is p_1 and T_1 respectively. The ambient air is rammed isentropically from pressure p_2 and temperature T_2. This ideal ramming action is shown by the vertical line 1-2 in in actual practice, because of internal friction due to irreversibility's, the temperature of the rammed air is more than T_2. Thus the actual ramming process is shown by the curve 1-2' which is adiabatic but not isentropic due to friction. The pressure and temperature of the rammed air is now p_2' and T_2' respectively. During the ideal or actual ramming process, the total energy or enthalpy remains constant i.e. $h_2 = h_2'$, and $T_2 = T_2'$.

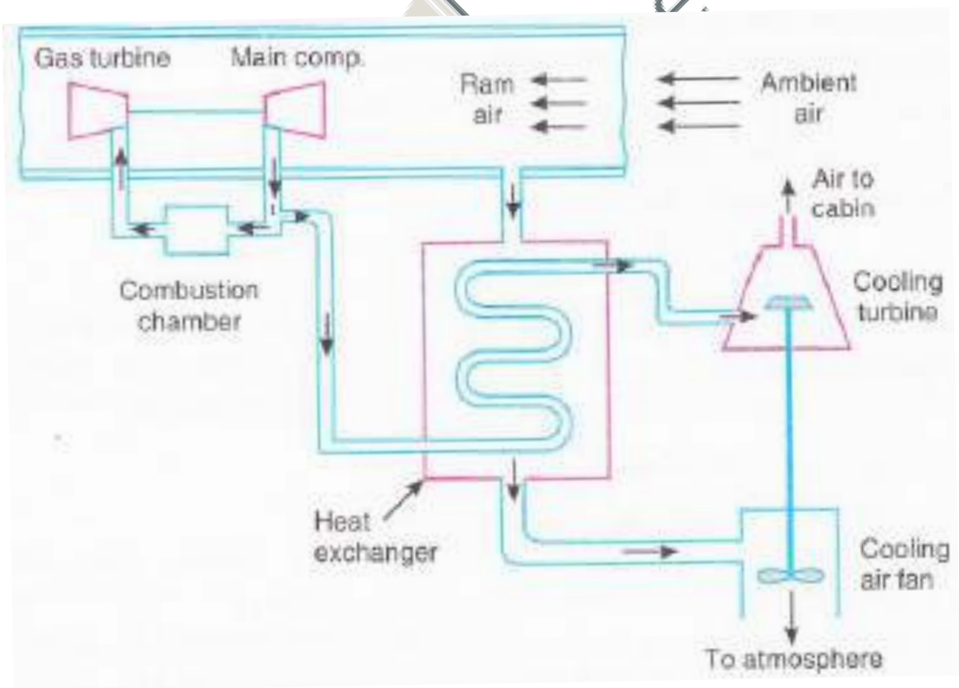


Fig. 3.1 (a) Simple Air Cooling System

Due to the irreversible compression in the ram, the air reaches point 2' instead of point 2 at the same stagnation temperature but at a reduced stagnation pressure p_2'. The pressure p_2' is obtained by Ram efficiency is given by,

$$\eta_R = \frac{\text{Actaul rise in pressure}}{\text{Isentropic rise in pressure}} = \frac{p_2' - p_1}{p_2 - p_1}$$

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

$$W_C = m_a c_p \left(T_3' - T_2'\right)$$

where m_a = Mass of air bled from the main compressor for refrigeration purpose

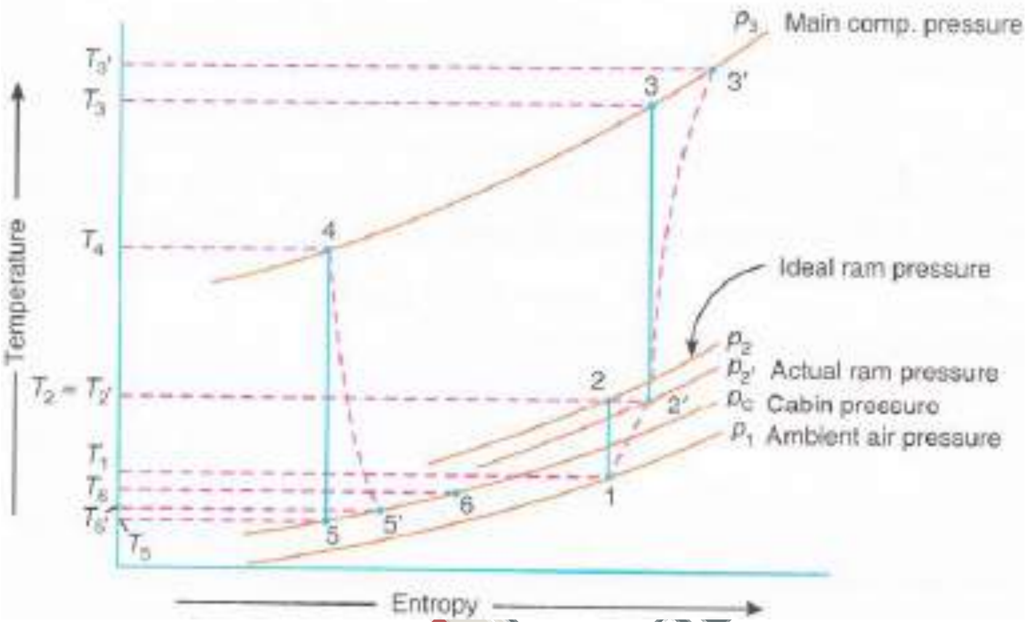


Fig.3.1 (b) T-s diagram for simple air cycle cooling system.

3. Cooling process. The compressed air is cooled by the ram air in the heat exchanger. This process is shown by 3'-4 in Fig. 3.2. In actual practice there is a pressure drop in the heat exchanger. The temperature of air decease from T_3' to T_4 . The heat rejected in the heat exchanger during the cooling process is given by,

$$Q_R = m_a c_p \left(T_3' - T_4\right)$$

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

$$W_T = m_a c_p \left(T_4 - T_5'\right)$$

The work of this cooling turbine is utilized to drive cooling fan which draws cooling air from the heat exchanger.

5. Refrigeration process. The air from cooling turbine is sent to the cabin and cockpit where it gets heated by the heat of equipment and occupancy. This process is shown by the curve 5'-6. The refrigerating effect produced is given by,

$$R.E = m_a c_p \left(T_6 - T_5'\right)$$

where T_6 = Inside temperature of cabin

T_5' = Exit temperature of cooling turbine

The C.O.P of the air cycle,

$$C.O.P = \frac{R.E}{W} = \frac{m_a c_p \left(T_6 - T_5'\right)}{m_a c_p \left(T_3' - T_2'\right)} = \frac{T_6 - T_5'}{T_3' - T_2'}$$

If Q tonnes of refrigeration of cooling load in the cabin, then the air required for the refrigeration is,

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

Power required for the refrigeration system,

$$P = \frac{m_a c_p \left(T_3' - T_2'\right)}{60} \text{ kW}$$

3.5.2 Simple Air Evaporative Cooling system

– A simple air evaporative cooling system is shown in Fig. 3.2 (a). It is similar to the simple cooling system except that the addition of an evaporator between the heat exchanger and cooling turbine. The evaporator provides an additional cooling effect through evaporation of a refrigerant such as water. At high altitudes, the evaporative cooling may be obtained by using alcohol or ammonia. The water, alcohol and ammonia have different refrigerating effects at different altitudes. At 20 000 metres height, water boils at 40°C, alcohol at 9°C and ammonia at - 70°C.

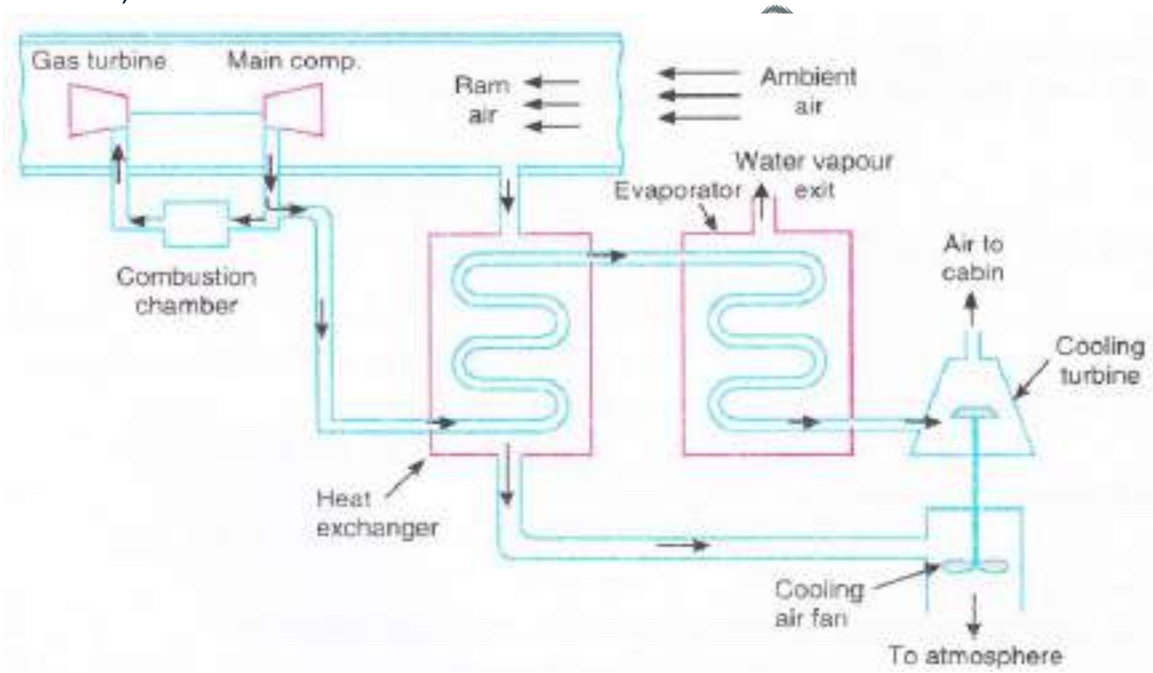


Fig. 3.2 (a) Simple air evaporative cooling system

The T-s diagram for a simple air cycle evaporative cooling system is shown in Fig. 3.2 (b). The various processes in simple air cooling system, except that the cooling process evaporator as shown by 4-4'.

It Q tonnes of refrigeration is the cooling load in the cabin, refrigeration then purpose,

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

Power required for the refrigeration system,

$$P = \frac{m_a c_p \left(T_3' - T_2'\right)}{60} \text{ kW}$$

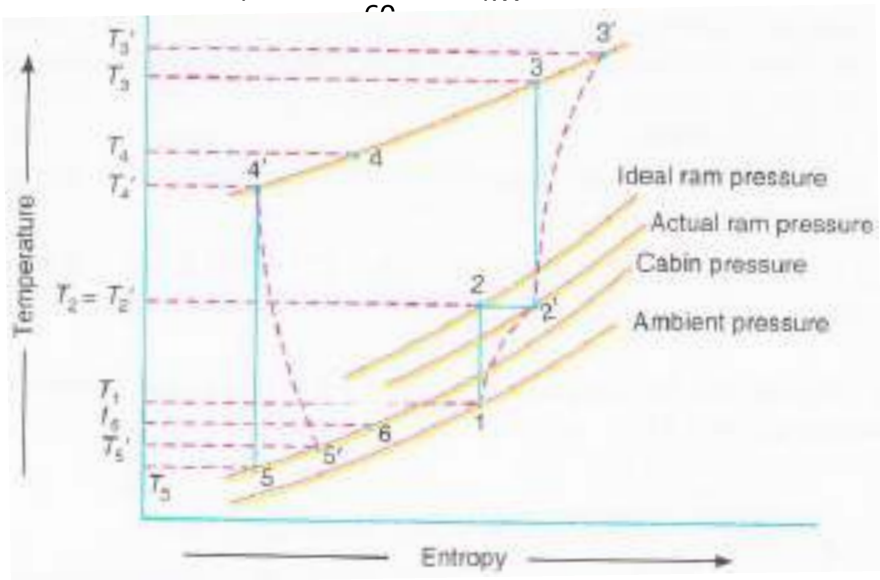


Fig. 3.2 (b) T-s diagram of simple air evaporative cooling system

C.O.P. of the refrigerating system,

$$C.O.P = \frac{210\ Q}{m_a c_p \left(T_3' - T_2'\right)} = \frac{210\ Q}{P \times 60}$$

The initial mass of evaporant required to be carried for the given flight time is given by,

$$m_e = \frac{Q_e t}{h_{fg}}$$

where Q_e = heat to be removed in evaporator kJ/min

t = Flight time in minutes

h_{fg} = Latent heat of evaporation

3.5.3 Boot-strap Air Cooling System

- A boot-strap air cooling system is shown in Fig. 3.3 (a). This cooling system has two heat exchangers instead of one and a cooling turbine drives a secondary compressor instead of cooling fan. The air bled from the main compressor is first cooled by the ram air in the first heat exchanger. This cooled air, after compression in the secondary compressor, is led to the second heat exchanger where it is again cooled by the ram air before passing to the cooling turbine. This type of cooling system is mostly used in transport type aircraft.
- T-s diagram for a boot-strap air cycle cooling system is shown in Fig. 3.3 (b). The various processes are as follows:
  1. The process 1- 2 represents the isentropic ramming of ambient air from pressure p_1, and temperature T_1 to pressure p_2 and temperature T_2. The process 1- 2' represents the actual ramming process because of internal friction due to irreversibilities.

2. The process 2'- 3 represents the isentropic compression of air in the main compressor and the process 2'- 3' represents the actual compression of air because of internal friction due to irreversibilities.

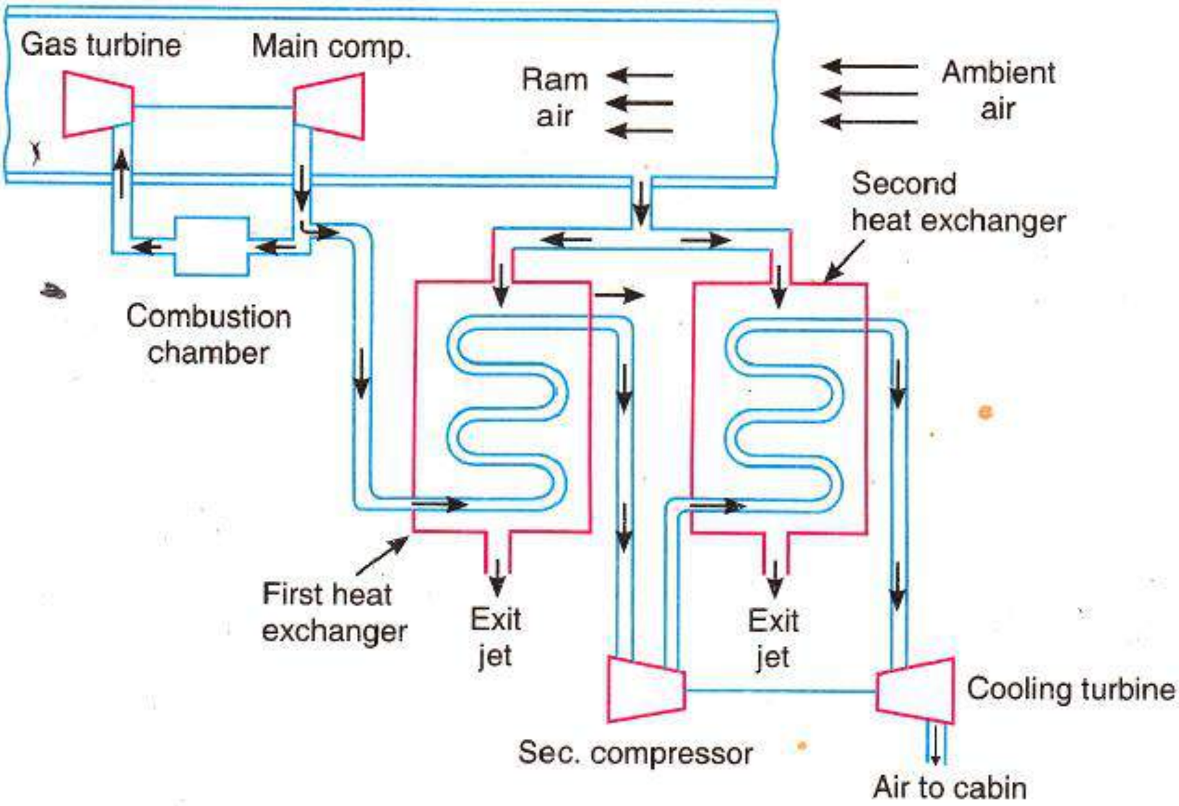


Fig.3.3 (a) Boot-strap Air Cooling System

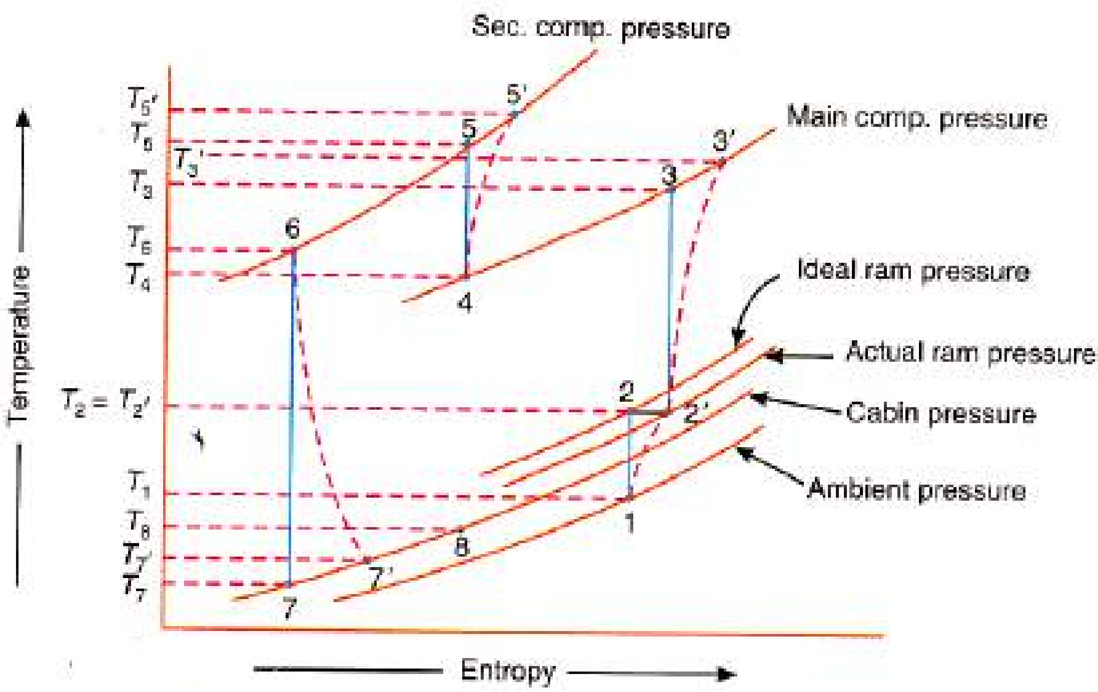


Fig. 3.3 (a) T-s diagram for Boot-strap Air Cooling System

3. The process 3'-4 represents the cooling by ram air in the first heat exchanger. The pressure drop in the heat exchanger is neglected.
4. The process 4 - 5 represents the isentropic compression of cooled air, from first heat exchanger, in the secondary compressor. The process 4 - 5' represents the actual compression process because of internal friction due to irreversibilities.
5. The process 5'- 6 represents the cooling by ram air in the second heat exchanger. The pressure drop in the heat exchanger in neglected.
6. The process 6 - 7 represents the isentropic expansion of cooled air in the cooling turbine upto the cabin pressure. The process 6 - 7'represents actual expansion of the cooled air in the cooling turbine.
7. The process 7'- 8 represents the heating of air upto the cabin temperature T_8.

If Q tonnes of refrigeration of cooling load in the cabin, then the air required for the refrigeration is,

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

Power required for the refrigeration system,

$$P = \frac{m_a c_p\left(T_3' - T_2'\right)}{60} \text{ kW}$$

C.O.P. of the refrigerating system,

$$C.O.P = \frac{210\ Q}{m_a c_p\left(T_3' - T_2'\right)} = \frac{210\ Q}{P \times 60}$$

3.5.4 Boot-strap Air Evaporative Cooling System

A boot-strap air cycle evaporative cooling system is shown in Fig. 3.4 (a). It is similar to the boot-strap air cycle cooling system except that the addition of an evaporator between the second heat exchanger and the cooling turbine.

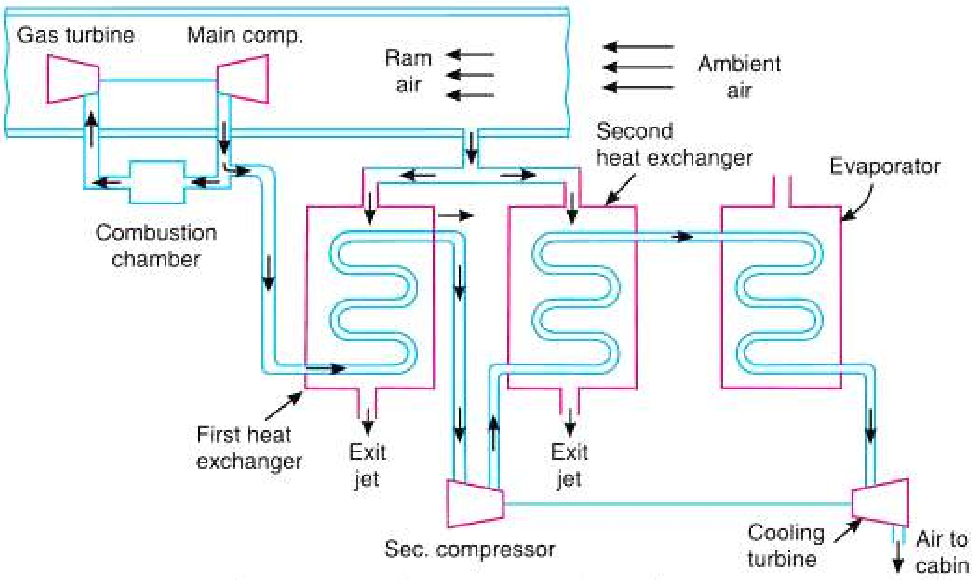


Fig.3.4 (a) Boot-strap Air Evaporative Cooling System

The T-s diagram for a boot-strap air evaporative cooling system is shown in Fig 3.4 (b). The various processes of this cycle are same as simple boot strap air cooling system except the process 5’-6 which represents cooling in the evaporator using any suitable evaporant.

If Q tonnes of refrigeration of cooling load in the cabin, then the air required for the refrigeration is,

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

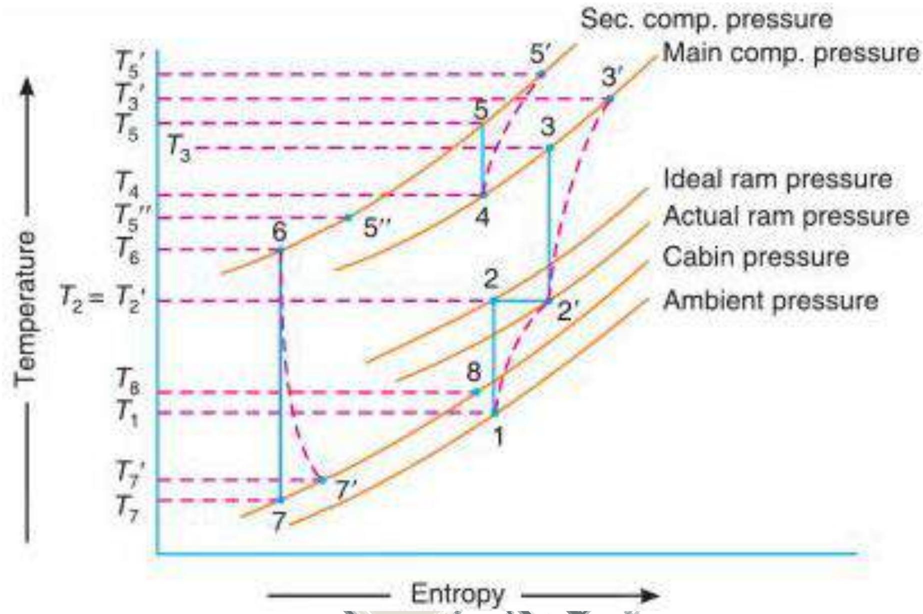


Fig. 3.4 (b) T-s diagram of Boot-strap Air Evaporative Cooling System

Power required for the refrigeration system,

$$P = \frac{m_a c_p\left(T_3' - T_2'\right)}{60} \text{ kW}$$

C.O.P. of the refrigerating system,

$$C.O.P = \frac{210\ Q}{m_a c_p\left(T_3' - T_2'\right)} = \frac{210\ Q}{P \times 60}$$

3.5.5 Regenerative Air Cooling System

– The regenerative air cooling system is shown in Fig. 3.5 (a). It is a modification of a simple air cooling system with the addition of a regenerative heat exchanger. The high pressure and high temperature air from the main compressor is first cooled by the ram air in the heat exchanger. This air is further cooled in the regenerative heat exchanger with a portion of the air bled after expansion in the cooling turbine. This type of cooling system is used for supersonic aircrafts and rockets. The diagram for the regenerative air cooling system is shown in Fig. 3.5 (b). The various processes are as follows:

1. The process 1- 2 represents the isentropic ramming of air and process 1- 2' represents the actual ramming process because of internal friction due to irreversibilities.

2. The process 2'- 3 represents the isentropic compression of air in the main compressor and the process 2'- 3' represents the actual compression of air because of internal friction due to irreversibilities.

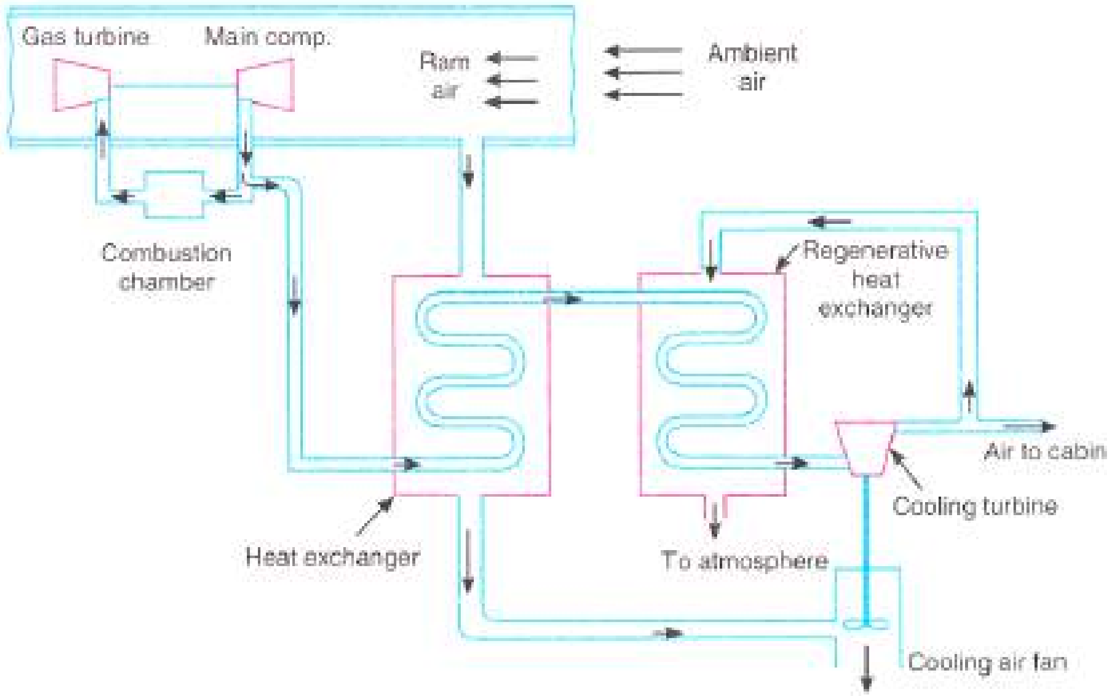


Fig. 3.5 (a) Regenerative air cooling system

3. The process 3'-4 represents the cooling by ram air in the heat exchanger.
4. The process 4 - 5 represents cooling of air in the regenerative heat exchanger.
5. The process 5 -6 represents the isentropic expansion of cooled air in the cooling turbine upto the cabin pressure and process 5 - 6' represents actual expansion of the cooled air in the cooling turbine.
6. The process 6'- 7 represents the heating of air upto the cabin temperature T_7.

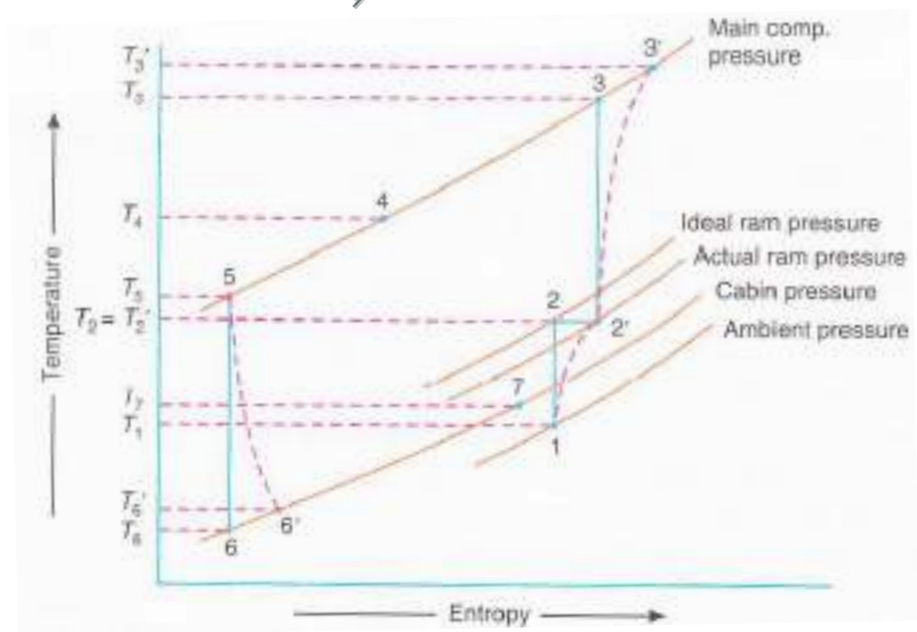


Fig. 3.5 (b) T-s diagram for regenerative air cooling system

If Q tonnes of refrigeration of cooling load in the cabin, then the air required for the refrigeration is,

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

Let $m_1 =$ Total mass of air bled from the main compressor

$m_2 =$ Mass of cold air bled from the cooling turbine for regenerative heat exchanger

For the energy balanced of regenerative heat exchanger, we have

$$m_2 c_p\left(T_8 - T_6'\right) = m_1 c_p\left(T_4 - T_5\right)$$

$$m_2 = \frac{m_1\left(T_4 - T_5\right)}{\left(T_8 - T_6'\right)}$$

where $T_8 =$ Temperature of air leaving to atmosphere from the for regenerative heat exchanger

Power required for the refrigeration system,

$$P = \frac{m_1 c_p\left(T_3' - T_2'\right)}{60} \text{ kW}$$

C.O.P. of the refrigerating system,

$$C.O.P = \frac{210\ Q}{m_a c_p\left(T_3' - T_2'\right)} = \frac{210\ Q}{P \times 60}$$

3.5.6 Reduced Ambient Air Cooling System

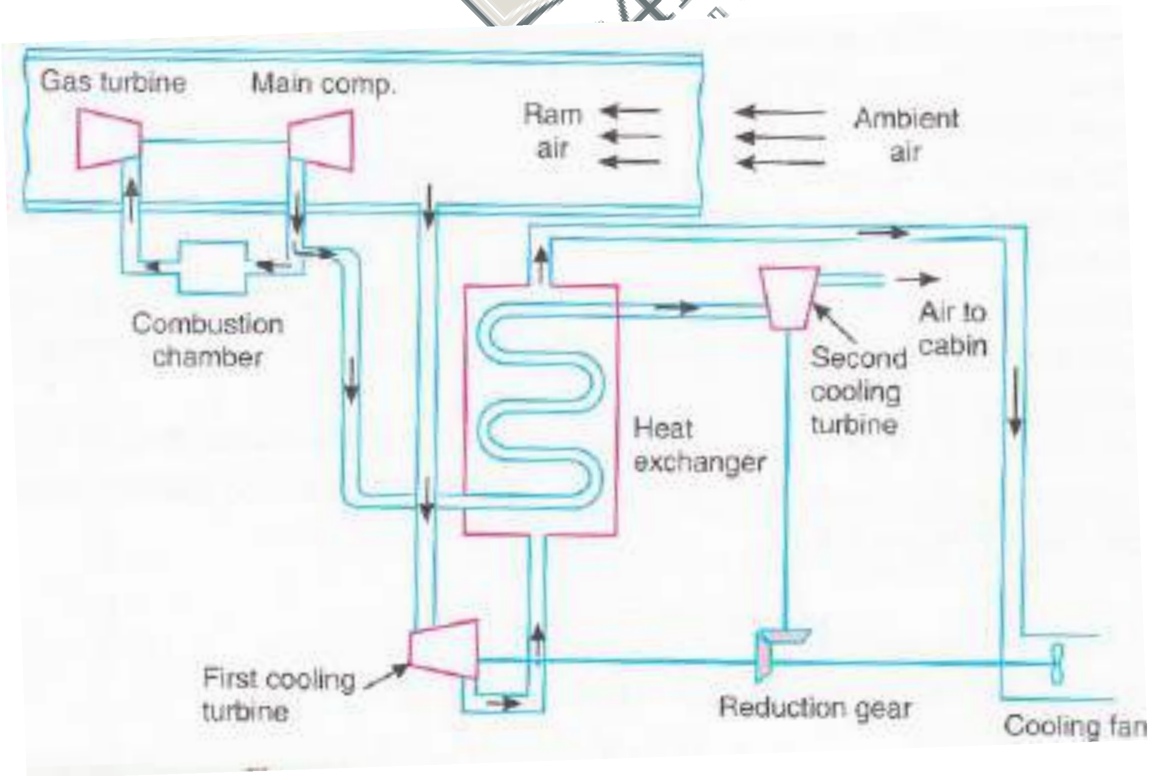


Fig. 3.6 (a) T-s diagram for reduced ambient air cooling system

– The diagram for the regenerative air cooling system is shown in Fig. 3.6 (a) and T-s diagram in Fig. 3.6 (b). The various processes are as follows:

1. The process 1-2 represents isentropic ramming of air and process 1- 2' represents actual ramming of air because of internal friction due to irreversibility's.
2. The process 2'-3 represents isentropic compression of air in the main compressor and the process 2'-3' represents actual compression of air because of internal friction due to irreversibility's.
3. The process 3'-4 represents cooling of compressed air by ram air in the heat exchanger.
4. The process 4-5 represents cooling of air in the regenerative heat exchanger.
5. The process 5-6 represents isentropic expansion of air in the cooling turbine up to the cabin pressure and the process 5-6' represents actual expansion of air in the cooling turbine.
6. The process 6'-7 represents heating of air up to the cabin temperature T_7.

If a Q tonne of refrigeration is the cooling load in the cabin, then the quantity of air required for the refrigeration purpose will be

$$m_a = \frac{210Q}{c_p\left(T_6 - T_5'\right)} \text{ kg/min}$$

Power required for the refrigeration system,

$$P = \frac{m_1 c_p\left(T_3' - T_2'\right)}{60} \text{ kW}$$

C.O.P. of the refrigerating system,

$$C.O.P = \frac{210\ Q}{m_a c_p\left(T_3' - T_2'\right)} = \frac{210\ Q}{P \times 60}$$

4A

Vapour Compression Refrigeration System

Course Contents

4.1 Introduction

– The 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 condition. The refrigerants usually used for this purpose are NH_3, CO_2 and SO_2. The refrigerant does not leave the system, but is 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.

4.2 Advantages and Disadvantages of Vapour Compression Refrigeration system over Air Refrigeration system

Following are the advantages and disadvantages of the 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.

4.3 Mechanism of a Simple vapour Compression

– Fig. 4.1 shows the schematic diagram 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 high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged into 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.

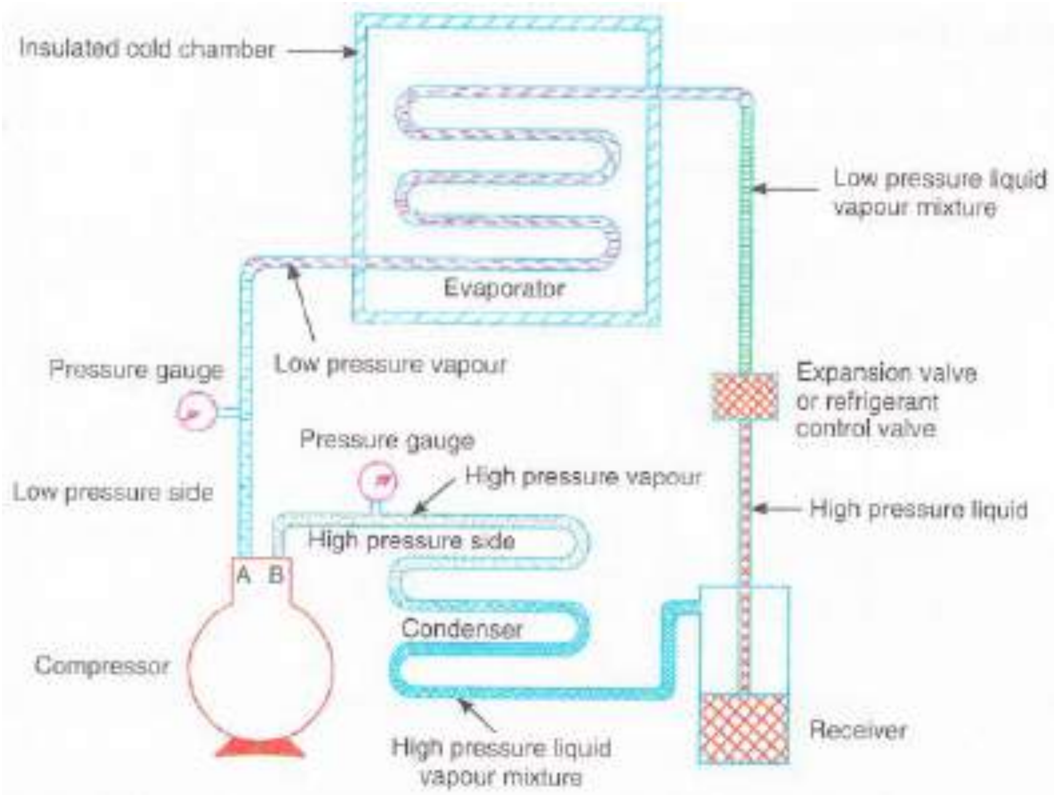


Fig. 4.1. Simple vapour compression refrigeration system

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 vaporisation from the medium (air, water or brine) which is to be cooled.

4.4 Pressure-Enthalpy (p-h) Chart

– The most convenient chart for studying the behaviour 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. 4.2, 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 at vapour region and to the right of the saturated vapour line is a superheated vapour region.

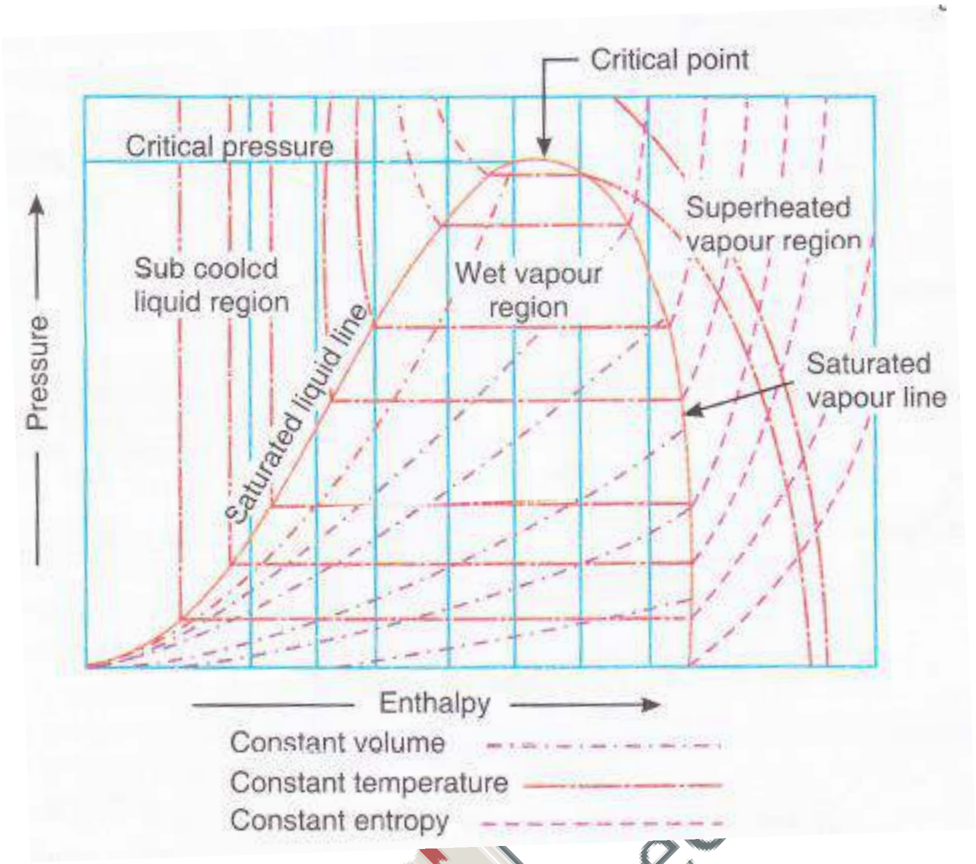


Fig. 4.2 Pressure -enthalpy (p-h) chart

4.5 Types of vapour compression cycles with p-h and T-s diagram

Following are the 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 subcooling of refrigerant.

4.5.1 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 in Fig. 4.3. At point 1 let T_1, p_1 and s_1, be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycle are as follows:

1. Compression process: The vapour refrigerant at at low pressure p_1, and temperature T_1 is compressed isentropically to dry saturated vapour as shown by the vertical line 1-2 and 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, at suction of the compressor, and

h_2 = Enthalpy of vapour refrigerant at temperature T_2, at discharge of the compressor

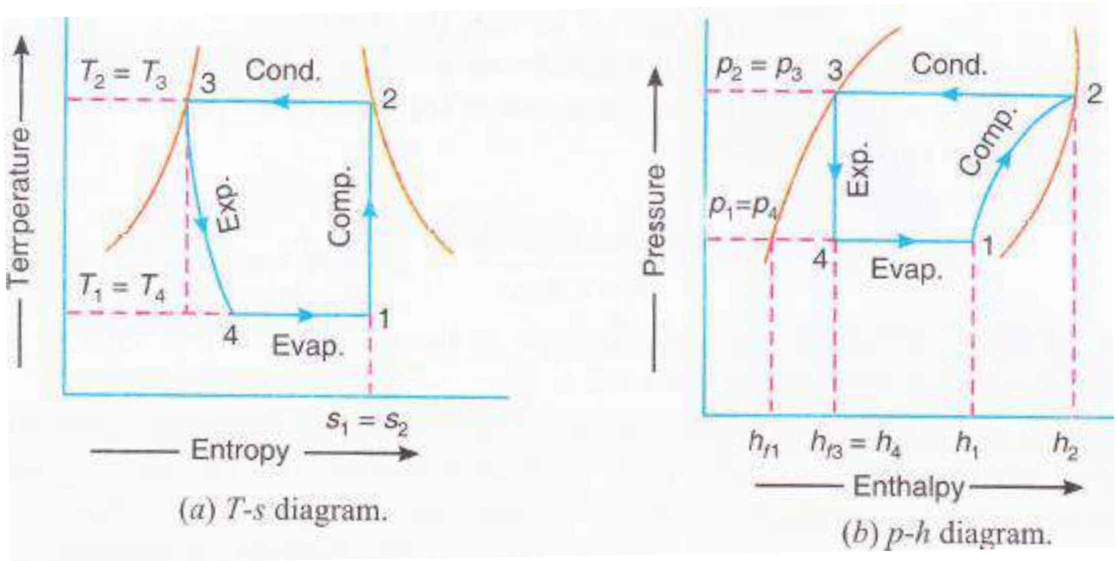


Fig. 4.3 Theoretical vapour compression cycle with dry saturated vapour after compression

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 temperature at constant pressure as shown by the 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 p-h diagram. During throttling some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator and no heat is absorbed by the liquid refrigerant.

4. Vaporisation: 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 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine) which is to be cooled. This heat which is absorbed by the refrigerant is called refrigerating effect. The process of vaporisation continues up to point 1 which is the starting point and thus the cycle is completed.

Now the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by,

$$R.E = h_1 - h_4 = h_1 - h_{f3}$$

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

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

4.5.2 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. 4.4. 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.

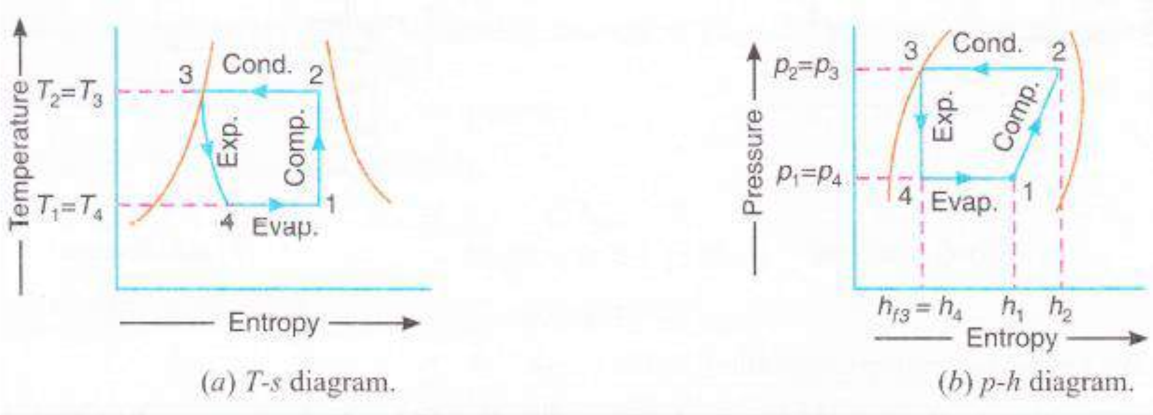


Fig. 4.4 Theoretical vapour compression cycle with wet vapour after compression

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

4.5.3 Theoretical Vapour Compression Cycle with Superheated Vapour after Compression

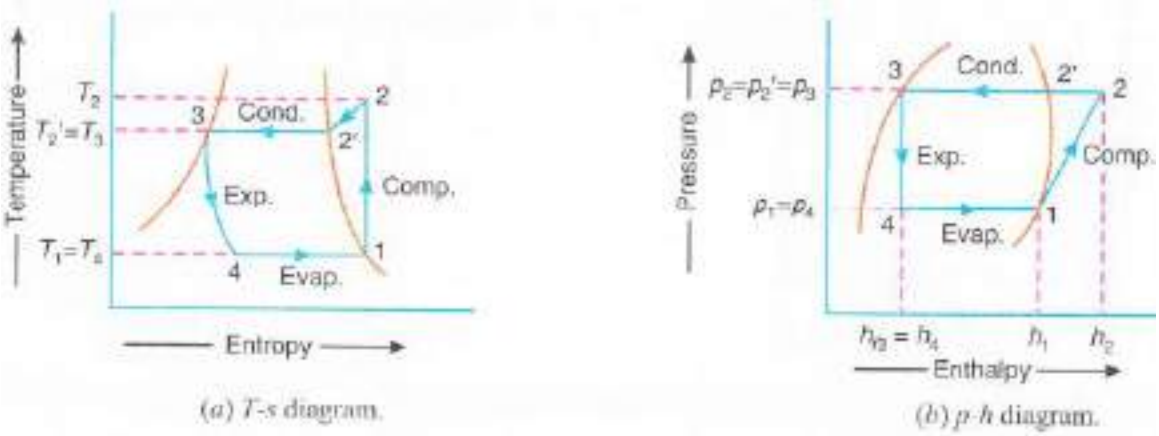


Fig. 4.5 Theoretical vapour compression cycle with superheated vapour after compression

– A vapour compression cycle with superheated vapour after compression is shown on T-s and p-h diagrams in Fig. 4.5. In this cycle, the enthalpy at point 2 is found out with the help of degree of superheat. The degree of superheat may be found out by equating the entropies at points 1 and 2.

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

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

- As shown in Fig. 4.5 the superheating increases the refrigerating effect and the amount of work done in the compressor. Since the increase in refrigerating effect is less as compared to the increase in work done, therefore, the net effect of superheating is to have low coefficient of performance.
- In this cycle, the cooling of superheated vapour will take place in two stages. Firstly, it will be condensed to dry saturated stage at constant pressure (shown by graph 2-2') and secondly, it will be condensed at constant temperature (shown by graph 2'-3). The remaining cycle is same as discussed in the last article.

4.5.3 Theoretical Vapour Compression Cycle with Superheated Vapour before Compression

- A vapour compression cycle with superheated vapour before compression is shown on T-s and p-h diagrams in Fig. 4.6. In this cycle, the evaporation starts at point 4 and continues upto point 1', when it is dry saturated. The vapour is now superheated before entering the compressor upto the point 1.

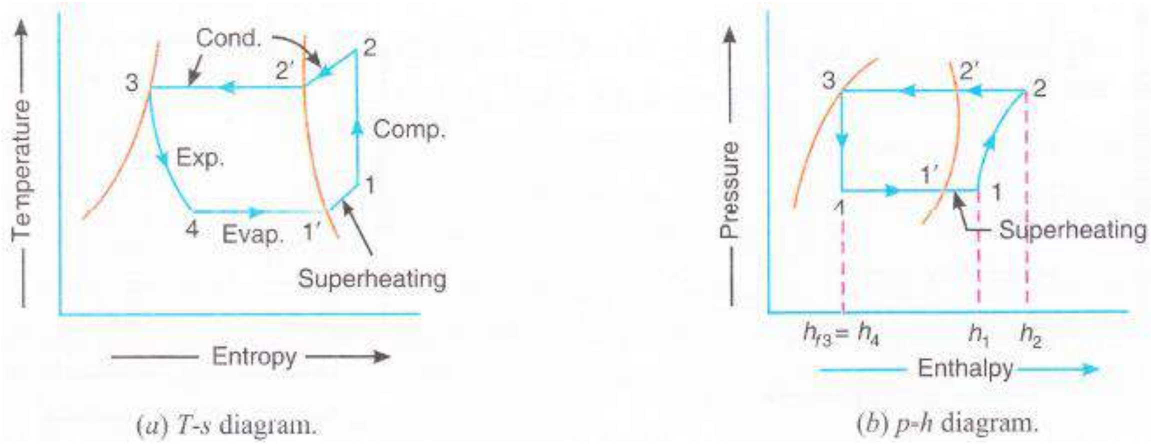


Fig. 4.6 Theoretical vapour compression cycle with superheated vapour before compression

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

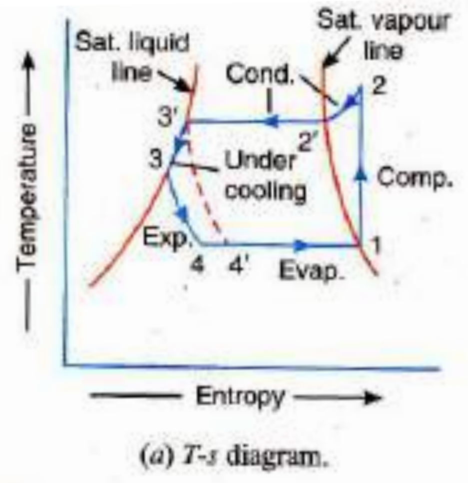
$$\text{C.O.P} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

- In this cycle, the heat is absorbed (or extracted) in two stages. Firstly from point 4 to point 1'and secondly from point 1' to point 1. The remaining cycle is same as discussed in the previous article.

4.5.4 Theoretical Vapour Compression Cycle with Undercooling or Subcooling of Refrigerant

- Sometimes, the refrigerant, after condensation process 2'-3', is cooled below the saturation temperature (T_3') before expansion by throttling. Such a process is called undercooling of the refrigerant and is generally done along the liquid line as shown in

Fig. 4.7. The ultimate effect of the undercooling is to increase the value of coefficient of performance under the same set of conditions.



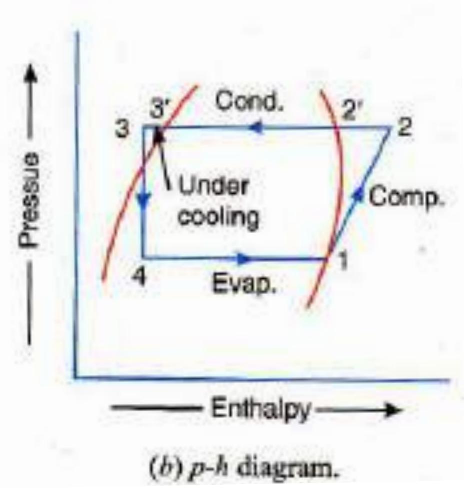


Fig. 4.7 Theoretical vapour compression cycle with subcooling of refrigerant

- The process of undercooling is generality brought about by circulating more quantity of cooling water through the condenser or by using water colder than the main circulating water. Sometimes, this process is also brought about by employing a heat exchanger. In actual practice, the refrigerant is superheated after compression and undercooled before throttling, as shown in Fig. 4.7. The refrigerating effect is increased by adopting both the superheating and undercooling process as compared to a cycle without them, which is shown by dotted line.
- Now the coefficient of performance may be found out as usual from the relation,

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

4.6 Actual Vapour Compression Cycle

- The actual vapour compression cycle differs from the theoretical vapour compression cycle runny ways, some of which are unavoidable and cause losses.

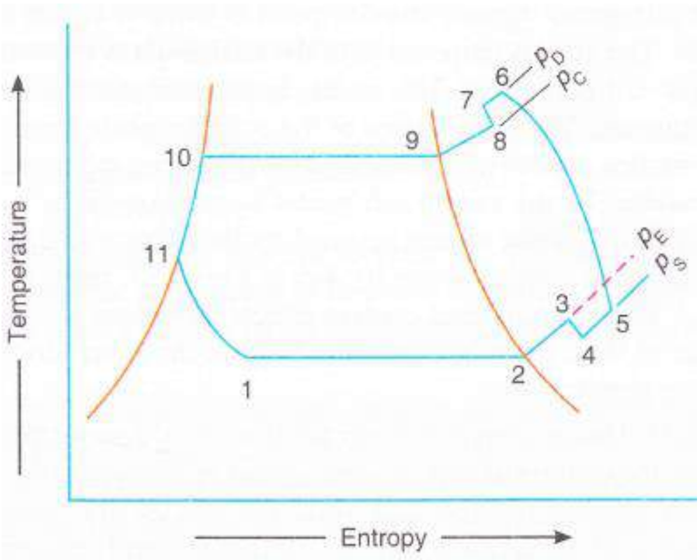


Fig .4.8 Actual Vapour Compression Cycle

- The main deviations between the theoretical cycle and actual cycle are as follows:
  1. The vapour refrigerant leaving the evaporator is in superheated state.
  2. The compression of refrigerant is neither isentropic nor polytropic.
  3. The liquid refrigerant before entering the expansion valve is sub-cooled in the condenser.
  4. The pressure drops in the evaporator and condenser.
- The actual vapour compression cycle on T-s diagram is shown in Fig. 4.8. The various processes are discussed below:

(a) Process 1-2-3. This process shows the flow of refrigerant in the evaporator. The point 1 represents the entry of refrigerant into the evaporator and the point 3 represents the exit of refrigerant from evaporator in a superheated state. The point 3 also represents the entry of refrigerant into the compressor in a superheated condition.

The superheating of vapour refrigerant from point 2 to point 3 may be due to:

1. Automatic control of expansion valve so that the refrigerant leaves the evaporator as the superheated vapour.
2. Picking up of larger amount of heat from the evaporator through pipes located within cooled space.
3. Picking up of heat from the suction and the compressor suction valve. i.e the pipe connecting the evaporator delivery and the compressor suction valve.

- In the first and second case of superheating the vapour refrigerant, the refrigerating effect as well as the compressor work is increased. The coefficient of performance, as compared to saturation cycle at the same suction pressure may be greater, less or unchanged.
- The superheating also causes increase in the required displacement of compressor and load on the compressor and condenser. This is indicated by 2-3 on T-s diagram as shown in Fig. 3.8.

(b) Process 3-4-5-6-7-8. This process represents the flow of refrigerant through the compressor. When the refrigerant enters the compressor through the suction valve at point 3, the pressure falls to point 4 due to frictional resistance to flow. Thus the actual suction pressure is lower than the evaporator pressure (p_E). During suction and prior to compression, temperature of the cold refrigerant vapour rises to point 5 when it comes in contact with the compressor cylinder walls. The actual compression of the refrigerant is shown by 5-6 in which is neither isentropic nor polytropic. This is due to the heat transfer between the cylinder walls and the vapour refrigerant. The temperature of the cylinder walls is some-what in between the temperatures of cold suction vapour refrigerant and hot discharge vapour refrigerant. It may assume that the heat absorbed by the vapour refrigerant from the cylinder walls during the part of the compression stroke is equal to heat rejected by the vapour refrigerant to the cylinder walls. Like the heating effect at suction given by 4-5 in Figure 4.8 , there is a cooling effect discharge as given by 6-7. These heating and cooling effects take place at constant pressure at constant pressure to the frictional resistance of flow, there is a pressure drop i.e. the actual discharge pressure (p_D) is more than the condenser pressure (p_C).

(c) Process 8-9-10-11. This process represents the flow of refrigerant through the condenser. The process 8-9 represents the cooling of superheated vapour refrigerant to the dry saturated state. The process 9-10 shows the removal of latent heat which changes the dry saturated refrigerant in to liquid refrigerant. The process 10-11 represents the sub-cooling of liquid refrigerant in the condenser before passing through the expansion valve. This is desirable as it increases the refrigerating effect per kg of the refrigerant flow. It also reduces the volume of the refrigerant partially evaporated from the liquid refrigerant while passing through the expansion valve. The increase in refrigerating effect can be obtained by large quantities of circulating cooling water which should be at a temperature much lower than the condensing temperatures.

(d) Process 11-1. This process represents the expansion of subcooled liquid refrigerant by throttling from the condenser pressure to the evaporator pressure.

4.7 Factors affecting the performance of simple VCR system

1. Suction pressure

– In actual practice, the suction pressure (or evaporator pressure) decreases due to the frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle 1'-2'-3'-4' when the suction pressure decreases from p_s to p_s', as shown on p-h diagram in Fig. 4.9 (a).

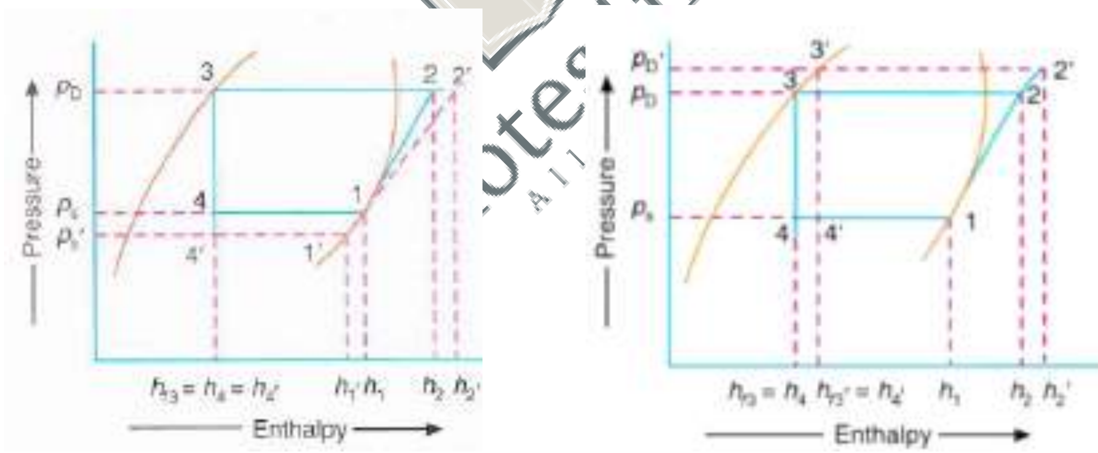


Fig. 4.9 (a) Effect of suction pressure *(b) Effect of discharge pressure*

– It may he noted that the decrease in suction pressure

1. decreases the refrigerating effect from $(h_1 - h_4)$ to $(h_1' - h_4')$ and
2. increases the work required for compression from $(h_2 - h_1)$ to $(h_2' - h_1')$

– Since the C.O.P of the system is the ratio of refrigerating effect to the work done, therefore with the decrease in suction pressure, the net effect is to decrease the C.O.P of the refrigerating system, the refrigerating capacity of the system decreases and the refrigeration cost increases for the same amount of refrigerant flow.

2. Discharge Pressure

- In actual practice, the discharge pressure (or condenser pressure) increases due to the frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle 1’-2’-3’-4' when the discharge pressure increases from pD to pD’, as shown on p-h diagram in Fig. 4.9 (b).
- It may he noted that the increase in discharge pressure
  1. decreases the refrigerating effect from $(h_1 - h_4)$ to $(h_1 - h_4')$ and
  2. increases the work required for compression from $(h_2 - h_1)$ to $(h_2' - h_1)$
- From above diagram with the increase in discharge pressure, the net effect is to decrease the C.O.P of the refrigerating system, the refrigerating capacity of the system decreases and the refrigeration cost increases for the same amount of refrigerant flow.

4.8 Improvements in Simple Saturation Cycle

The simple saturation cycle may be improved by the following methods:

1. By introducing the flash chamber between the expansion valve and the evaporator.
2. By using the accumulator or pre-cooler.
3. By subcooling the liquid refrigerant by the vapour refrigerant.
4. By subcooling the liquid refrigerant leaving the condenser by liquid refrigerant from the expansion valve.

4.8.1 Simple Saturation Cycle with Flash Chamber

- We have already discussed that when the high pressure liquid refrigerant from the condenser passes through the expansion valve, some of it evaporates. This partial evaporation of the liquid refrigerant is known as flash. It may be noted that the vapour formed during expansion is of no use to the evaporator to producing refrigerating effect as compared to the liquid refrigerant which carries the heat in the form of latent heat. This form the vapour, which is incapable of producing any refrigerating effect, can be bypassed around the evaporator and supplied directly to the suction of the compressor. This is done by introducing a flash chamber between the expansion valve and the evaporator as shown in Fig. 4.10. The flash chamber is an insulated container and it separates the liquid and vapour due to centrifugal effect. Thus the mass of the refrigerant passing through the evaporator reduces.
- Let us consider that a certain amount of refrigerant is circulating through the condenser. This refrigerant after passing through the expansion valve, is supplied to the flash chamber which separate the liquid and vapour refrigerant_ The liquid refrigerant from the flash chamber is supplied to the evaporator and the vapour refrigerant flows directly from the flash chamber to the suction of the compressor. The p-h diagram of the cycle is shown in Fig. 4.11.

  Let, m_1 = mass of liquid refrigerant supplied to the evaporator

  m_2 = mass of refrigerant (liquid and vapour) circulating through the condenser or leaving the expansion valve.

Mass of vapour refrigerant flowing directly from the flash chamber to the suction of the compressor is $m_2 - m_1$.

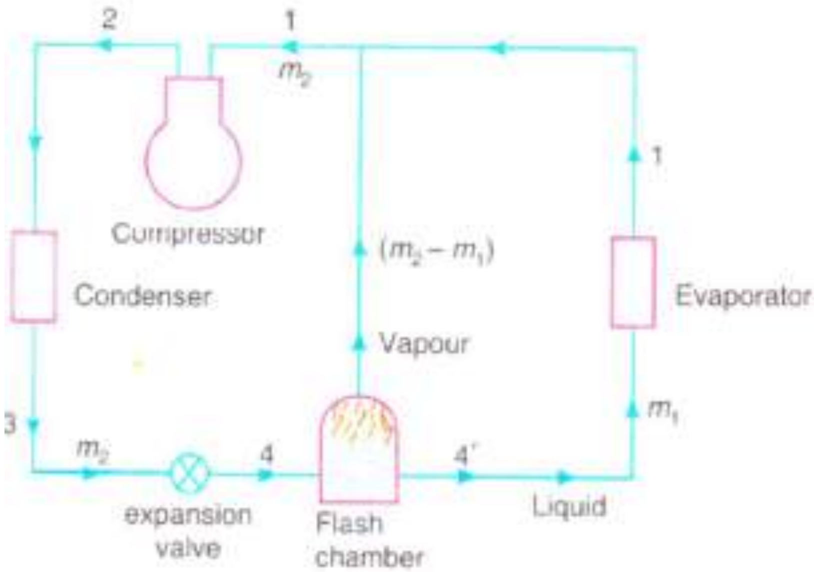


Fig. 4.10 Simple saturation cycle with flash chamber

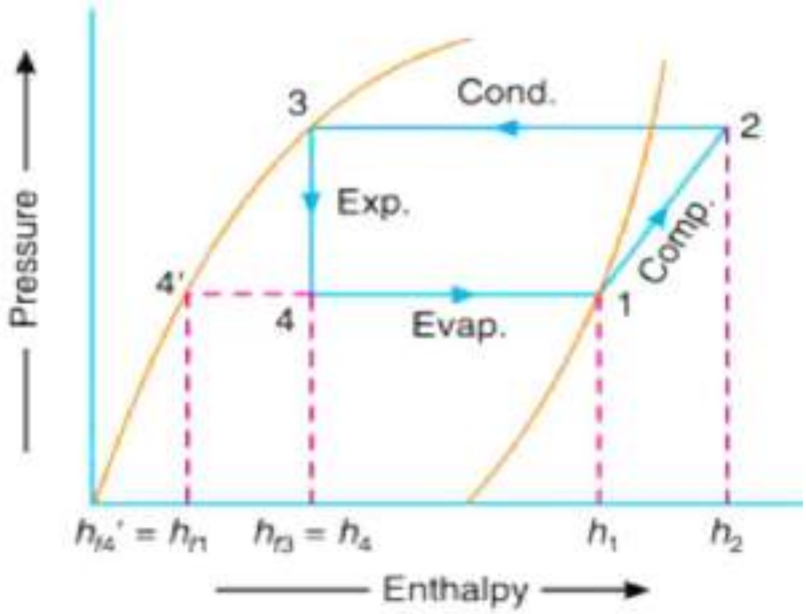


Fig. 4.11 p-h diagram of simple saturation cycle with flash chamber

Heat taken by flash chamber = heat given out by flash chamber

Now considering the thermal equilibrium of the flash chamber,

$$m_2 h_4 = m_1 h_{f4'} + (m_2 - m_1) h_1$$

$$m_2 (h_1 - h_4) = m_1 (h_1 - h_{f4'})$$

$$m_1 = m_2 \left[\frac{h_1 - h_4}{h_1 - h_{f4'}} \right]$$

Now refrigerating effect is given by,

$$R.E = m_1 (h_1 - h_{f4'})$$

$$= m_2 \left[\frac{h_1 - h_{f3}}{h_1 - h_{f4'}} \right] \left(h_1 - h_{f4'} \right) = m_2 \left(h_1 - h_{f3} \right)$$

and work done in compressor,

$$W = m_2 \left(h_2 - h_1 \right)$$

$$C.O.P = \frac{R.E}{W} = \frac{m_2 \left(h_1 - h_{f3} \right)}{m_2 \left(h_2 - h_1 \right)} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

And power required to drive the compressor,

$$\mathrm{P} = \frac{m_2 \left(h_2 - h_1 \right)}{60} kW$$

From above, we see that the refrigerating effect, coefficient of performance and the power required are same as that of a simple saturation cycle when the flash chamber is not used. Thus the use of flash chamber has no effect on the thermodynamic cycle. The only effect resulting from the use of flash chamber is the reduction in the mass of refrigerant flowing through the evaporator and hence the reduction in size of evaporator.

4.8.1 Simple Saturation Cycle with Accumulator or Pre-cooler

– Sometimes, the liquid refrigerant passing through the evaporator is not completely evaporated. If the compressor is supplied with liquid along with vapour refrigerant, then the compressor has to do an additional work of evaporating and raising the temperature of liquid refrigerant. It will also upset the normal working of the compressor which is meant only for compressing the pure vapour refrigerant.
– In order to avoid this difficulty, an insulated vessel, known as accumulator or pre-cooler. used in the system, as shown in Fig. 4.12. The accumulator receives the discharge (a mixture liquid and vapour refrigerant) from the expansion valve and supplies the liquid refrigerant only to the evaporator, as in the case of flash chamber.

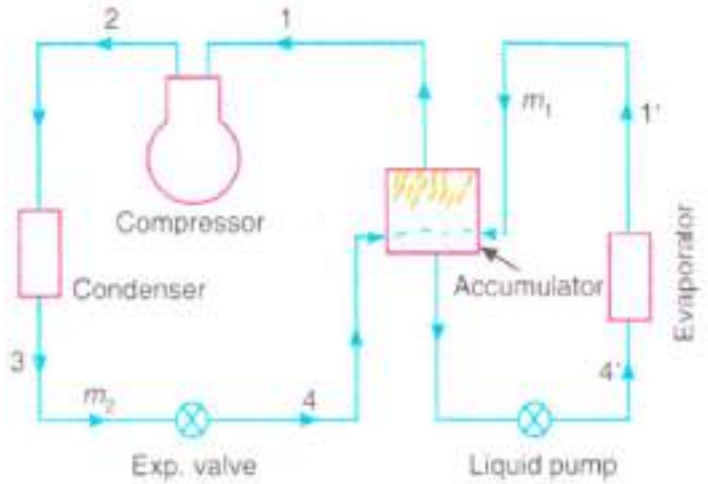


Fig. 4.12 Simple saturation cycle with accumulator or pre-cooler

– The discharge from the evaporator is sent again to the accumulator which helps to keep off the liquid from entering the compressor. Thus the accumulator supplies dry and saturated vapour to the compressor. A liquid pump is provided in the system in order to maintain circulation of the refrigerant in the evaporator.

Let m_1 = Mass of liquid refrigerant circulating through evaporator, and
m_2 = mass of refrigerant flowing in the condenser

– When all the liquid refrigerant does not evaporate in the evaporator, it is represented by point 1' on p-h diagram as shown in Fig. 4.13. Let the mass of refrigerant that leaves the evaporator at point 1' is same.

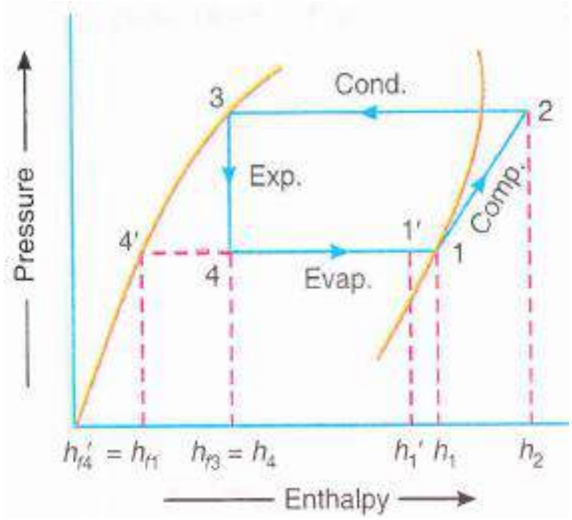


Fig. 4.13 p-h diagram of simple saturation cycle with accumulator

– Consider the thermal equilibrium of the accumulator. Since the accumulator is an insulated vessel, therefore there is no heat exchange between the accumulator and the atmosphere. Mathematically,

Heat taken in by the accumulator = Heat given out by the accumulator

$$m_2h_4 + m_1h_{1'} = m_2h_1 + m_1h_{f4'}$$

$$m_1\left(h_{1'} - h_{f4'}\right) = m_2\left(h_1 - h_4\right)$$

$$m_1 = m_2\left[\frac{h_1 - h_4}{h_{1'} - h_{f4'}}\right] = m_2\left[\frac{h_1 - h_{f3}}{h_{1'} - h_{f4'}}\right]$$

Now refrigerating effect is given by,

$$R.E = m_1\left(h_1 - h_{f4'}\right)$$

$$= m_2\left[\frac{h_1 - h_{f3}}{h_1 - h_{f4'}}\right]\left(h_1 - h_{f4'}\right) = m_2\left(h_1 - h_{f3}\right)$$

Work done in compressor,

$$W = m_2\left(h_2 - h_1\right)$$

$$C.O.P = \frac{R.E}{W} = \frac{m_2\left(h_1 - h_{f3}\right)}{m_2\left(h_2 - h_1\right)} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Power required to drive the compressor,

$$\mathrm{P} = \frac{m_2\left(h_2 - h_1\right)}{60} kW$$

From above, we see that when the accumulator is used in the system, the refrigerating effect, coefficient of performance, and power required is same as the simple saturation cycle. The accumulator is used only to protect the liquid refrigerant to flow into the compressor and thus dry compression is always ensured.

4.8.2 Simple Saturation Cycle with Sub-Cooling of Liquid Refrigerant by Vapour Refrigerant

We know that the liquid refrigerant leaving the condenser is at a higher temperature than the vapour refrigerant leaving the evaporator. The liquid refrigerant leaving the condenser can be subcooled by passing it through a heat exchanger is supplied with saturated vapour from the evaporator as shown in Fig. 4.14. In the heat exchanger, the liquid refrigerant gives heat to the refrigerant. The p-h diagram of the cycle is shown in Fig. 4.16.

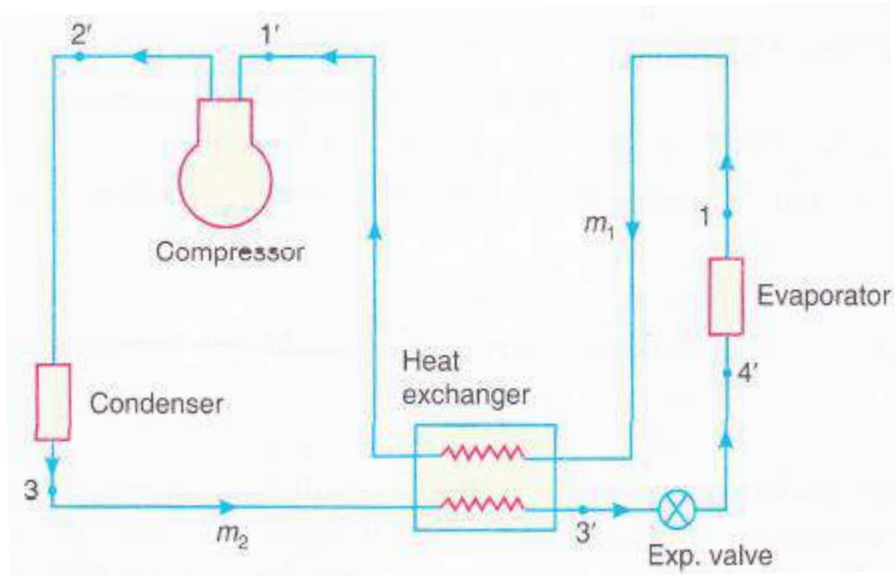


Fig. 4.14 Simple saturation cycle with sub-cooling of liquid refrigerant by vapour refrigerant

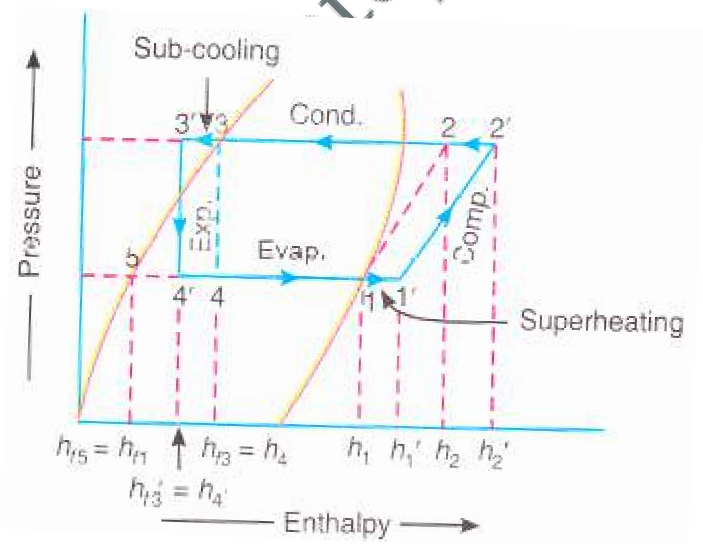


Fig. 4.15 p-h diagram of simple saturation cycle with subcooling of liquid refrigerant by vapour refrigerant

m_1 = Mass of the vapour refrigerant

m_2 = Mass of the liquid refrigerant

T_3 = Temperature of liquid refrigerant entering the heat exchanger,

T_3' = Temperature of liquid refrigerant leaving exchanger,

T_1 = Temperature of vapour refrigerant entering exchanger,
T_1' = Temperature of vapour refrigerant leaving exchanger,
c_{pv} = Specific heat of vapour refrigerant, and
c_{pl} = Specific heat of liquid refrigerant
Considering the thermal equilibrium of heat exchanger
Heat lost by liquid refrigerant = Heat gained by vapour refrigerant

$$m_2 \times c_{pl}\left(T_3 - T_{3'}\right) = m_1 \times c_{pv}\left(T_{1'} - T_1\right)$$

Since $m_2 = m_1$ and from energy balance,

$$\left(h_{f3} - h_{f3'}\right) = \left(h_{1'} - h_1\right)$$

In the ideal case, the liquid and vapour refrigerant leave the heat exchanger at the same temperature, say T_m

$$T_{1'} = T_{3'} = T_m$$

Knowing the condition of points 1 and 3, the condition of points 1' and 3' can be obtained by trial and error method on p-h chart the refrigerating effect from $\left(h_1 - h_{f3}\right)$ to $\left(h_1 - h_{f3'}\right)$ per kg of refrigerant as compared with simple saturation cycle.

If Q tonnes of refrigeration is the load on the evaporator, then the mass of refrigerant (m_R) required to be circulated through the evaporator for sub-cooled cycle is given by,

$$m_R = \frac{210Q}{h_1 - h_{f3'}} kg/\min$$

Power required to drive the compressor,

$$P_1 = \frac{m_R\left(h_{2'} - h_{1'}\right)}{60} kW = \frac{210Q}{60}\left[\frac{h_{2'} - h_{1'}}{h_1 - h_{f3'}}\right] kW$$

Power required to drive the compressor without heat exchanger (for simple saturation cycle)

$$P_2 = \frac{m_R\left(h_2 - h_1\right)}{60} kW = \frac{210Q}{60}\left[\frac{h_2 - h_1}{h_1 - h_{f3}}\right] kW$$

Excess power required to drive the compressor as compared to simple saturation cycle

$$P_{excess} = P_1 - P_2$$

$$= \frac{210Q}{60}\left[\frac{h_{2'} - h_{1'}}{h_1 - h_{f3'}} - \frac{h_2 - h_1}{h_1 - h_{f3}}\right] kW$$

From above, we see that sub-cooling the liquid refrigerant by vapour refrigerant , the C.O.P. of the cycle is reduced.

4.8.3 Simple Saturation Cycle with Sub-cooling of Liquid Refrigerant by Liquid Refrigerant

– We know that the liquid refrigerant leaving the condenser is at a higher temperature than the liquid refrigerant leaving the expansion valve. The liquid refrigerant leaving the condenser can be sub-cooled by passing it through a heat exchanger which is supplied with liquid refrigerant from the expansion valve, as shown in Fig. 4.16. In the heat exchanger, the liquid refrigerant from the condenser gives heat to the liquid refrigerant from the expansion valve. The p-h diagram of the cycle is shown in Fig. 4.17.

m_1 = Mass of liquid refrigerant circulating through evaporator, and

m_2 = mass of refrigerant flowing in the condenser

m_3 = Mass of liquid refrigerant supplied to the heat exchanger, from the expansion valve

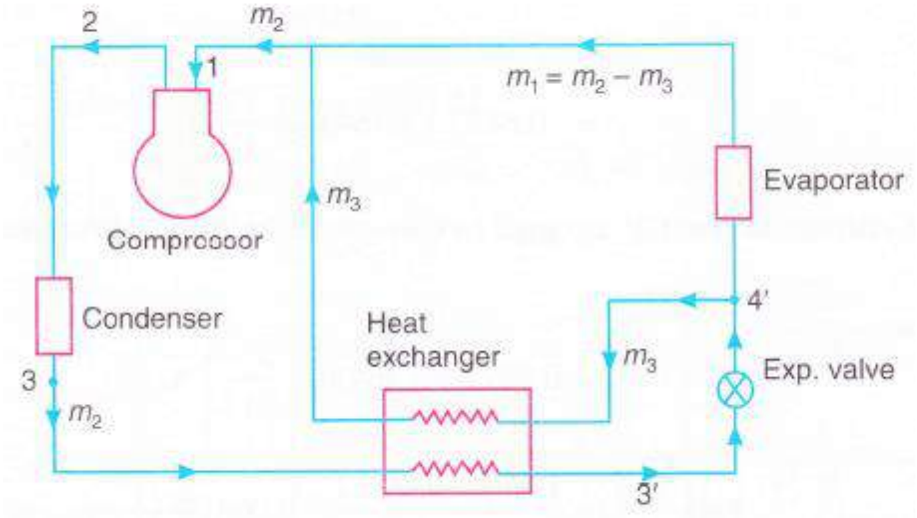


Fig. 4.16 Simple saturation cycle with sub-cooling of liquid refrigerant by liquid refrigerant

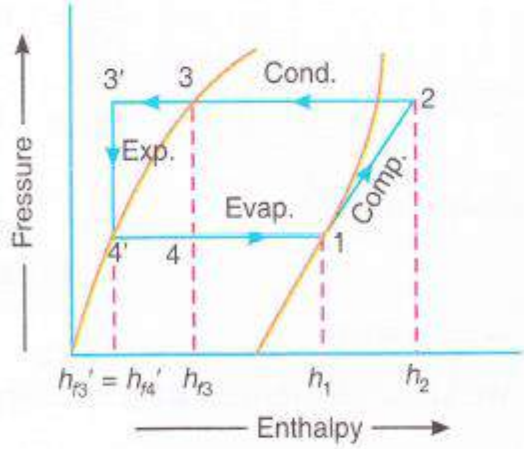


Fig. 4.17 p-h diagram of simple saturation cycle with sub-cooling of liquid refrigerant by liquid refrigerant

m_1 = Mass of liquid refrigerant circulating through evaporator, and

m_2 = mass of refrigerant flowing in the condenser

m_3 = Mass of liquid refrigerant supplied to the heat exchanger, from the expansion valve

Considering the thermal equilibrium of the heat exchanger,

Heat lost by liquid refrigerant from condenser = Heat gained by liquid refrigerant from expansion valve

$$m_2\left(h_{f3} - h_{f3'}\right) = m_3\left(h_1 - h_{f4'}\right)$$

$$m_3 = m_2\left[\frac{h_{f3} - h_{f3'}}{h_1 - h_{f4'}}\right] = m_2\left[\frac{h_{f3} - h_{f3'}}{h_1 - h_{f3'}}\right]$$

Now refrigerating effect is given by,

$$R.E = m_1\left(h_1 - h_{f4'}\right) = \left(m_2 - m_3\right)\left(h_1 - h_{f4'}\right)$$

$$\left[m_2 - m_2\left(\frac{h_{f3} - h_{f3'}}{h_1 - h_{f4'}}\right)\right]\left(h_1 - h_{f4'}\right)$$

$$m_2\left(h_1 - h_{f4'}\right) - m_2\left(h_{f3} - h_{f3'}\right)$$

$$m_2 h_1 - m_2 h_{f4'} - m_2 h_{f3} = m_2\left(h_1 - h_{f3}\right)$$

$$= m_2\left[\frac{h_1 - h_{f3}}{h_1 - h_{f4'}}\right]\left(h_1 - h_{f4'}\right) = m_2\left(h_1 - h_{f3}\right)$$

and work done in compressor,

$$W = m_2\left(h_2 - h_1\right)$$

$$C.O.P = \frac{R.E}{W} = \frac{m_2\left(h_1 - h_{f3}\right)}{m_2\left(h_2 - h_1\right)} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

If Q tonnes of refrigeration is the load on the evaporator, them the mass of refrigerant required to be circulated through the evaporator is given by,

$$m_1 = \frac{210Q}{h_1 - h_{f4'}} kg/\min$$

$$m_2 - m_3 = \frac{210Q}{h_1 - h_{f4'}}$$

Substitute the value of m_3 then we obtain,

$$m_2 - m_2\left[\frac{h_{f3} - h_{f3'}}{h_1 - h_{f4'}}\right] = \frac{210Q}{h_1 - h_{f4'}}$$

$$m_2\left[1 - \frac{h_{f3} - h_{f3'}}{h_1 - h_{f4'}}\right] = \frac{210Q}{h_1 - h_{f4'}}$$

$$m_2\left[\frac{h_1 - h_{f4'} - h_{f3} + h_{f3'}}{h_1 - h_{f4'}}\right] = \frac{210Q}{h_1 - h_{f4'}}$$

$$m_2 = \frac{210Q}{h_1 - h_{f3}}$$

Power required to drive the compressor,

$$P = \frac{m_2\left(h_2 - h_1\right)}{60} kW$$

Substitute the value of m_2 in the above equation,

$$P = \frac{210Q}{60}\left[\frac{h_2 - h_1}{h_1 - h_{f3}}\right] kW$$

4B

Compound Compression Refrigeration System

Course Contents

4.1 Introduction

- In the previous chapter, we have discussed the simple vapour compression refrigeration system in which the low pressure vapour refrigerant from the evaporator is compressed in a single stage (or a single compressor) and then delivered to a condenser at a high pressure. But sometimes, the vapour refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigerating systems In such cases either we should compress the vapour refrigerant by employing a single stage compressor with a very high pressure ratio between the condenser and evaporator or compress it in two or more compressors placed in series. The compression carried out in two or more compressors is called compound or multistage compression.
- In vapour compression refrigeration systems, the major operating cost is the energy input to the system in the form of mechanical work. Thus any method of increasing C.O.P is advantageous so long as it does not involve too heavy an increase in other operating expenses, as well as initial plant cost and consequent maintenance.
- Since the coefficient of performance of refrigeration system is the ratio of refrigerating effect to the compression work, therefore the coefficient of performance can be increased either by increasing the refrigerating effect or by increasing the compression work. In vapour compression refrigeration system compression work is greatly reduced if the refrigerant is compressed very close to the saturated vapour line. This can be achieved by compressing the refrigerant in more stages with intermediate Intercooling. But it is economical only where the pressure ratio is considerable as would be the case when very low evaporator temperatures are desired or when high condenser temperature may be required. The compound compression is generally economical in large plants.
- The refrigerating effect can be increased by maintaining the condition of the refrigerant in more liquid state at the entrance to the evaporator. This can be achieved by expanding the refrigerant very close to the saturated liquid line. It may be noted that by subcooling the refrigerant by removing the flashed vapour, as they are during multistage expansion, the expansion can be brought close to the liquid line.

4.2 Advantages of Compound Vapour Compression Refrigeration system with Intercooler

Following are the advantages and disadvantages of the system over single stage compression:

1. The work done per kg of refrigerant is reduced in compound compression with intercooler as compared to single stage compression for the same delivery pressure.
2. It improves the volumetric efficiency for the given pressure ratio.
3. The sizes of the two cylinders (i.e. high pressure and low pressure) may be adjusted to suit the volume and pressure of the refrigerant.
4. It reduces the leakage loss considerably.
5. It gives more uniform torque, and hence a smaller size flywheel is needed.

6. It provides effective lubrication because of lower temperature range.
7. It reduces the cost of compressor.

4.3 Types of Compound Vapour Compression with Intercooler

– In compound compression vapour refrigeration systems, the superheated vapour refrigerant leaving the first stage of compression is cooled by suitable method before being fed to the second stage of compression and so on. Such type of cooling the refrigerant is called intercooling. Following are the various types of compound compression with intercoolers:
  1. Two stage compression with liquid intercooler.
  2. Two stage compression with water intercooler.
  3. Two stage compression with water intercooler, liquid subcooler and liquid flash chamber.
  4. Two stage compression with water intercooler, liquid subcooler and flash intercooler.

4.3.1 Two Stage Compression with Liquid Intercooler

The arrangement of a two stage compression with liquid intercooler and the corresponding p-h diagram is shown is shown in Fig. 4.1 (a) and (b).

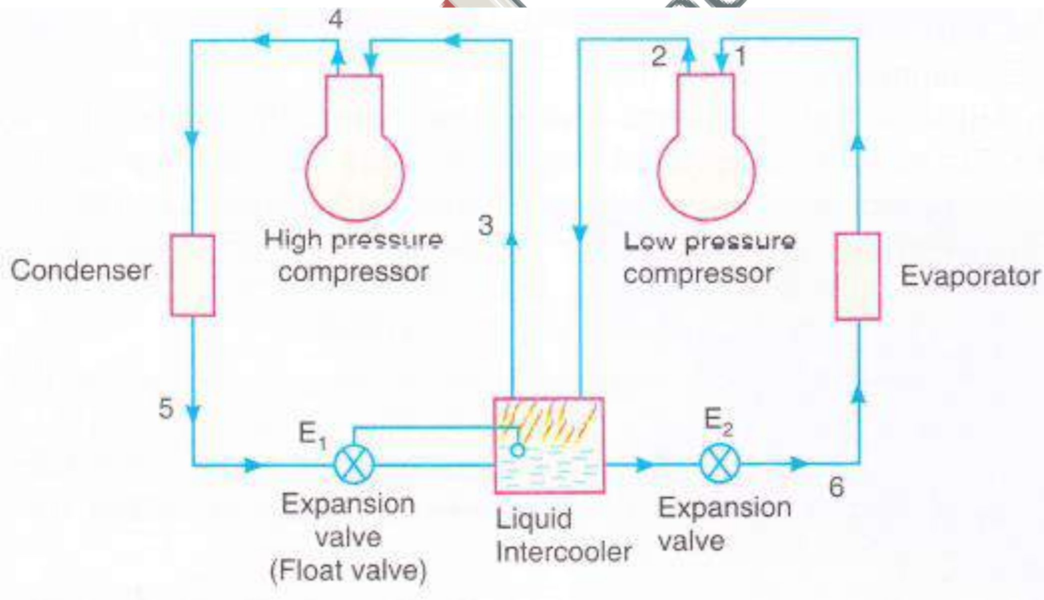


Fig. 4.1 (a) Two stage compression with liquid intercooler

The various points on the p-h diagram are plotted as discussed below:

1. First of all, draw a horizontal pressure line representing the evaporator pressure p_E (or suction pressure of low pressure compressor) which intersects the saturated vapour line at point 1. At this point, the saturated vapour is supplied to the low pressure compressor. Let, at point l, the enthalpy of the saturated vapour h_1, and entropy s_{v1}.

2. The saturated vapour refrigerant admitted at point 1 is compressed isentropically in the low pressure compressor and delivers the refrigerant in a superheated state. The pressure rises from p_E to p_2. The curve 1-2 represents the isentropic compression in the low pressure

compressor. In order to obtain point 2, draw a line from point 1, with entropy equal to s_{v1} along the constant entropy line intersecting the intermediate pressure line p_2 at point 2. Let enthalpy at this point is h_2.

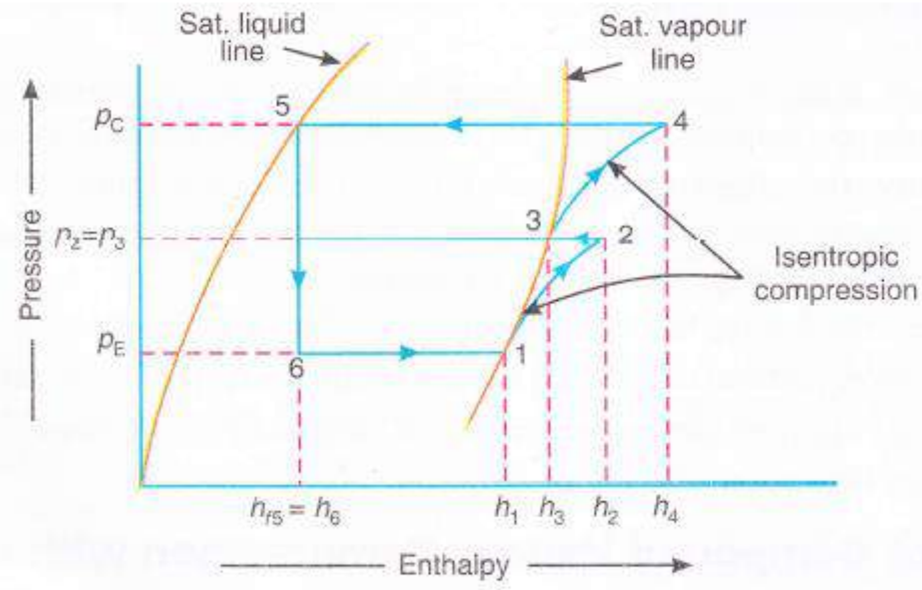


Fig. 4.1(b) p-h diagram of two stage compression with liquid intercooler

3. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is cooled (or desuperheated) at constant pressure $p_2 = p_3$ in a liquid intercooler by the liquid refrigerant from the condenser. The refrigerant leaving the liquid intercooler is in saturated vapour state. The line 2-3 represents the cooling or desuperheating process.

Let the enthalpy and entropy at point 3 is h_3 and s_{v3} respectively.

4. The dry saturated vapour refrigerant is now supplied to high pressure compressor where it is compressed isentropically from intermediate or intercooler pressure p_2 to condenser pressure p_C. The curve 3-4 represents the isentropic compression in the high pressure compressor. The point 4 on the p-h diagram is obtained by drawing a line of entropy equal to s_{v3} along the constant entropy line as shown in Fig. 4.1 (b). Let the enthalpy of superheated vapour refrigerant at point 4 is h_4.

5. The superheated vapour refrigerant leaving the high pressure compressor at point 4 is now passed through the condenser at constant pressure p_c as shown by a horizontal line 4- 5. The condensing process 4-5 changes the state of refrigerant from superheated vapour to saturated liquid.

6. The high pressure saturated liquid refrigerant from the condenser is passed to the intercooler where some of liquid refrigerant evaporates in desuperheating the superheated vapour refrigerant from the low pressure compressor. In order to make up for the liquid evaporated, i.e. to maintain a constant liquid level, an expansion valve E_1 which acts as a float valve, is provided.

7. The liquid refrigerant from the intercooler is first expanded in an expansion valve E_2 and then evaporated in the evaporator to saturated vapour condition.

Let m_1 = Mass of refrigerant passing through the evaporator (or low pressure compressor) in kg/min, and

m_2 = Mass of refrigerant passing through the condenser (or high pressure compressor) in kg/min

The high pressure compressor in a given system will compress the mass of refrigerant from pressure compressor (m_1) and the mass of liquid evaporated in the liquid intercooler during cooling or desuperheating of superheated vapour refrigerant from low pressure compressor. If m_3 is the mass of liquid evaporated in the intercooler, then

$$m_3 = m_2 - m_1$$

The value of m_2, may be obtained by considering the thermal equilibrium for the liquid cooler as shown in Fig. 4.1 (c).

Heat taken by the liquid intercooler = Heat given by the liquid intercooler

$$m_2 h_{f5} + m_1 h_2 = m_1 h_6 + m_1 h_3$$

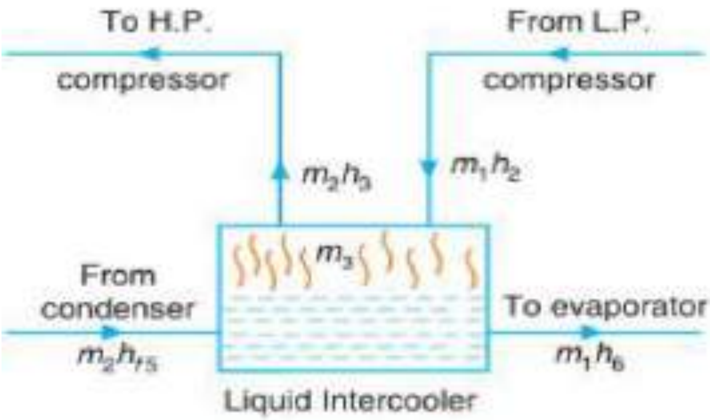


Fig. 4.1 (c) Thermal equilibrium for the liquid cooler

$$m_2 = \frac{m_1(h_2 - h_6)}{h_3 - h_{f5}} = \frac{m_1(h_2 - h_{f5})}{h_3 - h_{f5}}$$

The mass of liquid refrigerant evaporated in the intercooler,

$$m_3 = m_2 - m_1 = \frac{m_1(h_2 - h_6)}{h_3 - h_{f5}} - m_1 = \frac{m_1(h_2 - h_3)}{h_3 - h_{f5}}$$

Refrigerating effect is given by,

$$R.E = m_1(h_1 - h_{f5}) = 210Q \text{ kJ/min}$$

Total work done in both the compressors,

$$W = m_1(h_2 - h_1) + m_2(h_4 - h_3)$$

Power required to drive the system,

$$P = \frac{m_1(h_2 - h_1) + m_2(h_4 - h_3)}{60} kW$$

The C.O.P of the system is,

$$C.O.P = \frac{R.E}{W} = \frac{m_1(h_1 - h_{f5})}{m_1(h_2 - h_1) + m_2(h_4 - h_3)} = \frac{210Q}{P \times 60}$$

4.3.2 Two Stage Compression with Water Intercooler and Liquid Sub-cooler

The arrangement of a two-stage compression with water intercooler and the corresponding p-h diagram shown in Fig. 4.2 (a) and (b).

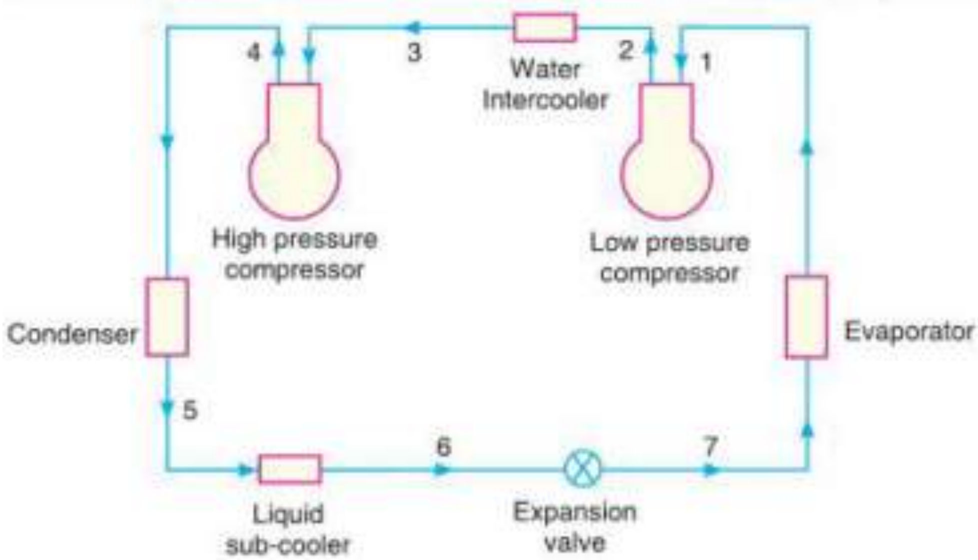


Fig. 4.2 (a) Two stage compression with water intercooler and liquid sub-cooler

The various processes in this system are as follows:

1. The saturated vapour refrigerant at the evaporator pressure p_E is admitted to low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropically from the evaporator pressure p_E to the water intercooler pressure $p_{2,}$ as shown by the curve 1-2.

2. The refrigerant leaving the low pressure compressor at point 2 is in superheated state. This superheated vapour refrigerant is now passed through the water intercooler at constant pressure, in order to reduce the degree of superheat. The line 2-3 represents the water intercooling or desuperheating process.

The refrigerant leaving the water intercooler at point 3 (which is still in the superheated state) is compressed isentropically in the high pressure compressor to the condenser pressure p_c. The curve 3-4 shows the isentropic compression in high pressure compressor.

4. The discharge from the high pressure compressor is now passed through the condenser which changes the state of refrigerant from superheated vapour to saturated liquid as shown by process 4-5.

5. The temperature of the saturated liquid refrigerant is further reduced by passing it through a liquid sub-cooler as shown by process 5-6.

6. The liquid refrigerant from the sub-cooler is now expanded in an expansion valve (process 6-7) before being sent to the evaporator for evaporation (process 7-1).

It may he noted that water intercooling reduces the work to be done in high pressure compressor. It also reduces the specific volume of the refrigerant which requires a compressor of less capacity for stroke volume). The complete desuperbeating of the vapour refrigerant is not possible in case of water intercooling. It is due to the fact that temperature of the cooling water used in water intercooler is not available sufficiently low so as to desuperheat the vapour completely.

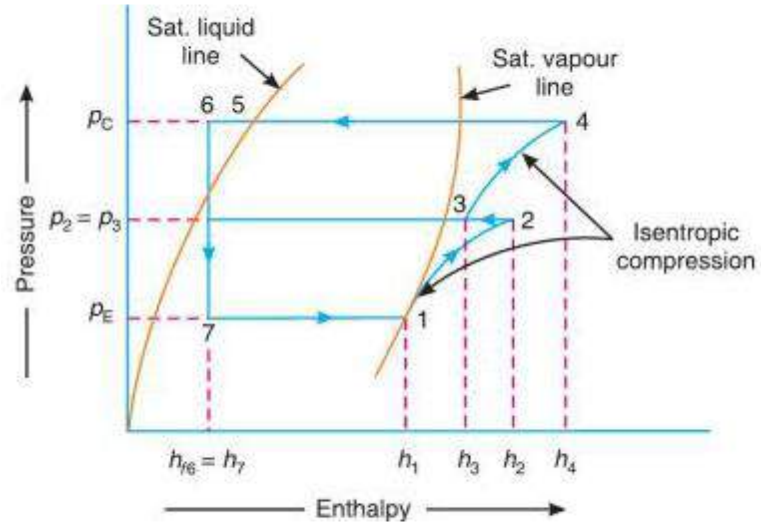


Fig. 4.2 (b) p-h diagram of two stage compression with water intercooler and liquid sub-cooler

Let, Q = load on the evaporator in tonnes of refrigeration.

Mass of refrigerant passing through the evaporator,

$$m = \frac{210Q}{h_1 - h_7} = \frac{210Q}{h_1 - h_{f6}} kg/\min$$

Mass of refrigerant passing through the compressor is same, therefore, total work in bothe the compressors,

$$W = m\left[\left(h_2 - h_1\right) + \left(h_4 - h_3\right)\right]$$

Power required to drive the system,

$$P = \frac{m\left[\left(h_2 - h_1\right) + \left(h_4 - h_3\right)\right]}{60} kW$$

Refrigerating effect is given by,

$$R.E = m\left(h_1 - h_{f6}\right) = 210Q \text{ kJ/min}$$

The C.O.P of the system is,

$$C.O.P = \frac{R.E}{W} = \frac{m\left(h_1 - h_{f6}\right)}{m\left[\left(h_2 - h_1\right) + \left(h_4 - h_3\right)\right]} = \frac{210Q}{P \times 60}$$

4.3.3 Two Stage Compression with water Intercooler, Liquid Sub -cooler and Liquid Flash Chamber

The arrangement of a two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber and corresponding p-h diagram is shown Fig. 4.3 (a) and (b). The various processes, in this system, are as follows:

1. The saturated vapour refrigerant at the evaporator pressure p_E is admitted to pressure compressor at point 1. In this compressor, the refrigerant is compressed to isentropically from evaporator pressure p_E to water intercooler (or flash chamber pressure p_F as shown by the curve 1-2).

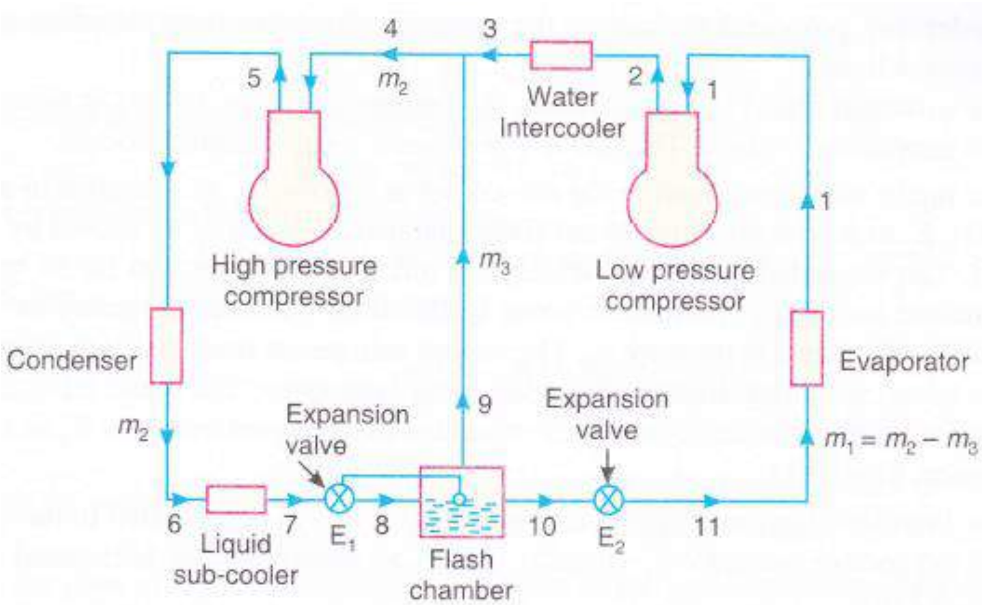


Fig. 4.3 (a) Two stage compression with water intercooler, liquid sub -cooler and liquid flash chamber

2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure p_F in order to reduce the degree of superheat (i.e. from temperature t_2 to t_3). The line 2-3 represents the water intercooling or de-superheating process.

3. The superheated vapour refrigerant leaving the water intercooler at point 3 is mixed with the vapour refrigerant supplied by the flash chamber at point 9. The condition of refrigerant after mixing is shown by point 4 which is in superheated state. Let the temperature at this point is t_4.

4. The superheated vapour refrigerant admitted at point 4 to the high pressure compressor is compressed isentropically from the intercooler or flash chamber pressure p_F to condenser pressure p_C as shown by curve 4-5. The temperature rises from t_4 to t_5.

5. The superheated vapour leaving the high pressure compressor at pressure p_c is passed through a condenser at constant pressure as shown by a horizontal line 5-6. The condensing process 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.

6. The saturated liquid refrigerant from the condenser is now cooled in liquid sub-cooler to a temperature say t_7. The line 6-7 represents a sub-cooling process.

7. The liquid refrigerant leaving the sub-cooler at pressure p_C equal to the flash chamber pressure p_F, as shown by vertical line 7-8. The expanded refrigerant which is a mixture of vapour and liquid refrigerants is admitted to a flash chamber at point 8. The flash chamber separates the vapour and liquid refrigerants at pressure p_F. The vapour refrigerant from the flash chamber at point 9 is mixed with the refrigerant from the water intercooler. The liquid refrigerant from the flash chamber at point 10 is further expanded in an expansion valve E_2 shown by the vertical line 10-11.

8. The liquid refrigerant leaving the expansion valve E_2 is evaporated in the evaporator at the evaporated pressure p_E as shown by the horizontal 11-1.

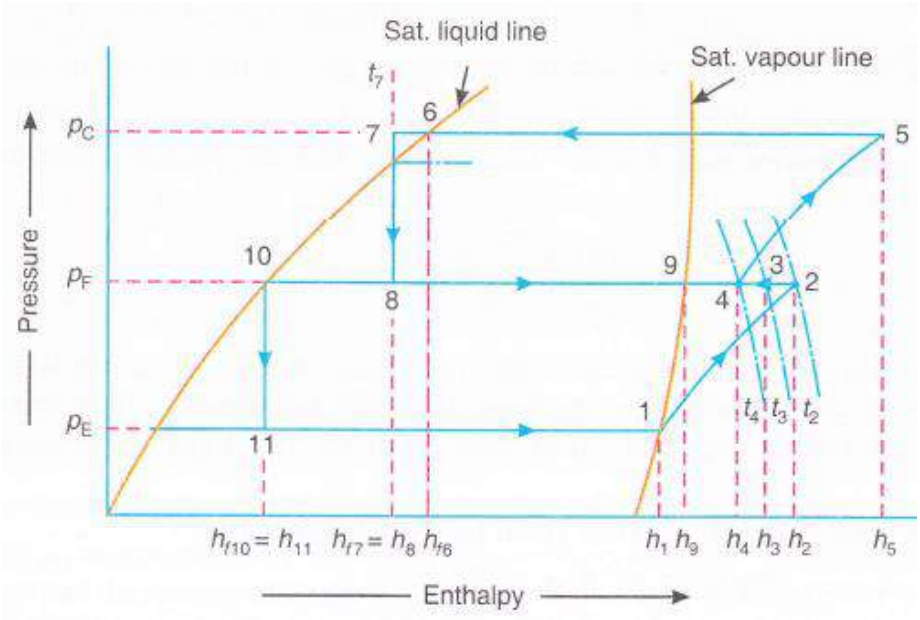


Fig. 4.3 (b) p-h diagram

Let m_2 = Mass of refrigerant passing through the condenser or high pressure compressor and

m_3 = Mass of vapour refrigerant formed in the flash chamber,

Mass of refrigerant passing through the evaporator,

$$m_1 = m_2 - m_3$$

Let, Q = load on the evaporator in tonnes of refrigeration, then the mass of refrigerant passing through the evaporator,

$$m_1 = \frac{210Q}{h_1 - h_{11}} = \frac{210Q}{h_1 - h_{f10}} kg/\min$$

Now consider the thermal equilibrium of the flash chamber,

Heat taken by the flash chamber = Heat given by the flash chamber

$$m_2 h_8 = m_3 h_9 + m_1 h_{f10} = m_3 h_9 + (m_2 - m_3) h_{f10}$$

$$m_2 \left(h_8 - h_{f10}\right) = m_3 \left(h_9 - h_{f10}\right)$$

$$m_3 = m_2 \left(\frac{h_8 - h_{f10}}{h_9 - h_{f10}}\right) = m_2 \left(\frac{h_{f7} - h_{f10}}{h_9 - h_{f10}}\right)$$

The enthalpy of the mixed refrigerant at point 4 may be calculated by,

$$m_2 h_4 = m_3 h_9 + m_1 h_3 = m_3 h_9 + (m_2 - m_3) h_3$$

Refrigerating effect is given by,

$$R.E = m_1 \left(h_1 - h_{11}\right) = 210Q \text{ kJ/min}$$

Total work done in low and high pressure compressors,

$$W = W_L + W_H = m_1 \left(h_2 - h_1\right) + m_2 \left(h_5 - h_4\right)$$

Power required to drive the system,

$$P = \frac{m_1 \left(h_2 - h_1\right) + m_2 \left(h_5 - h_4\right)}{60} kW$$

The C.O.P of the system is,

$$C.O.P=\frac{R.E}{W}=\frac{m_1\left(h_1-h_{11}\right)}{m_1\left(h_2-h_1\right)+m_2\left(h_5-h_4\right)}=\frac{210Q}{P\times 60}$$

4.3.4 Two Stage Compression with Water Intercooler, Liquid Sub-Cooler and Flash Intercooler

- A two stage compression with water intercooler, liquid sub-cooler and flash intercooler and corresponding p-h diagram is shown in Fig. 4.4 (a) and (b).
- When the vapour refrigerant from the low pressure compressor is passed through the water intercooler, its temperature does not reduce to the saturated vapour line or even very near to it, before admitting it to the high pressure compressor [Refer point 4 of Fig. 4.4 (b). In fact, with water cooling there may be no saving of work in compression. But the improvement in performance and the reduction in compression work may be achieved by using flash chamber as an intercooler as well as flash separator.
- The various processes, in this system, are as follows:

1. The saturated vapour refrigerant at the evaporator pressure p_E is admitted to the low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropically from evaporator pressure p_E to the flash intercooler pressure p_F, as shown by the curve 1-2.

2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure p_F, in order to reduce the degree of superheat (i.e. from temperature t_2 to t_3). The line 2-3 represents the water intercooling or desuperheating process.

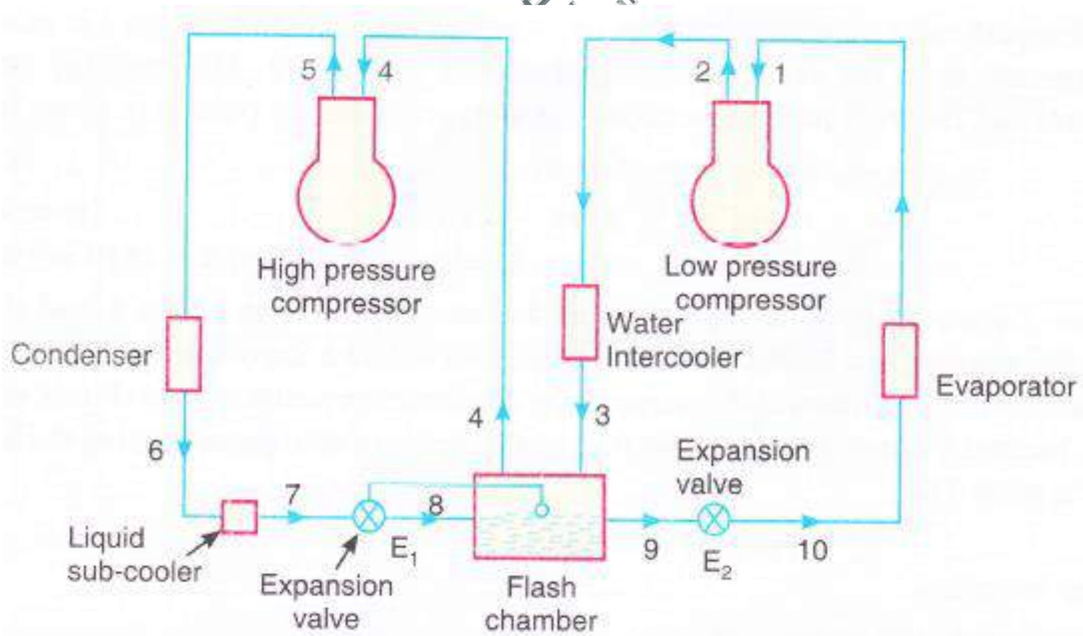


Fig. 4.4 (a) Two stage compression with water intercooler, liquid sub-cooler and flash intercooler

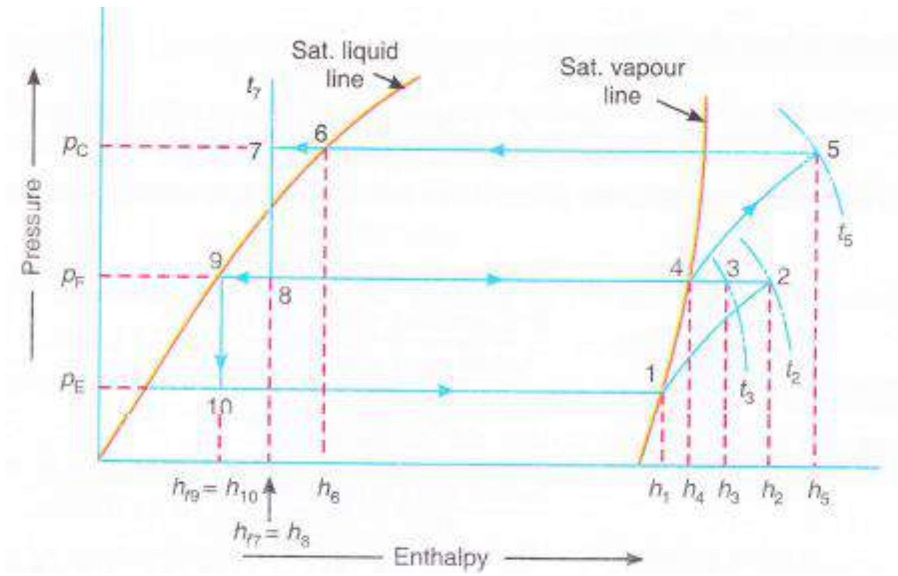


Fig. 4.4 (b) p-h diagram

3. The superheated vapour refrigerant leaving the water intercooler at point 3 is passed through a flash intercooler which cools the superheated vapour refrigerant to saturated vapour refrigerant as shown by the line 3-4. The cooling of superheated vapour refrigerant is done by the evaporation of a part of the liquid refrigerant from the flash intercooler placed at point 8.

4. The saturated vapour refrigerant leaving the flash intercooler enters compressor at point 4 where it is compressed isentropically from flash intercooler pressure p_F to condenser pressure p_c, as shown by the curve 4-5.

5. The superheated vapour refrigerant leaving the high pressure compressor at pressure is passed through a condenser at constant pressure. The condensing process as shown by line 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.

6. The saturated liquid refrigerant leaving the condenser at point 6 is now cooled at constant pressure p_c in the liquid sub-cooler to a temperature t_7 as shown in Fig. 4.4 (b).

The line 6-7 shows the sub-cooling process.

The liquid refrigerant leaving the sub-cooler at point 7 is expanded in an expansion valve E_1 to a pressure equal to the flash intercooler pressure p_F, as shown by the vertical line 7-8. The expanded refrigerant (which is a mixture of vapour and liquid refrigerant) is admitted to flash intercooler at point 8 which also acts as a flash separator.

8. The liquid refrigerant leaving the flash intercooler at point 9 is passed through the second expansion valve E_2 (process 9-10) and then evaporated in the evaporator as shown by the horizontal line 10-11.

Let m_1 = Mass of the refrigerant passing through the evaporator (or low pressure compressor), and

m_2 = Mass of the refrigerant passing through the condenser (or high pressure compressor).

If Q tonne of refrigeration is the load on the evaporator, then the mass of refrigerant passing through the evaporator is given by,

$$m_1 = \frac{210Q}{h_1 - h_{10}} = \frac{210Q}{h_1 - h_{f9}} kg/\min$$

Now consider the thermal equilibrium of the flash chamber,

Heat taken by the flash intercooler = Heat given by the flash intercooler

$$m_2 h_8 + m_1 h_3 = m_2 h_4 + m_1 h_{f9}$$

$$m_1 \left(h_3 - h_{f9}\right) = m_2 \left(h_4 - h_8\right)$$

$$m_2 = m_1 \left(\frac{h_3 - h_{f9}}{h_4 - h_8}\right) = m_2 \left(\frac{h_3 - h_{f9}}{h_4 - h_{f7}}\right) kg/\min$$

Refrigerating effect is given by,

$$R.E = m_1 \left(h_1 - h_{10}\right) = 210Q \text{ kJ/min}$$

Total work done in low and high pressure compressors,

$$W = W_L + W_H = m_1 \left(h_2 - h_1\right) + m_2 \left(h_5 - h_4\right)$$

Power required to drive the system,

$$P = \frac{m_1 \left(h_2 - h_1\right) + m_2 \left(h_5 - h_4\right)}{60} kW$$

The C.O.P of the system is,

$$C.O.P = \frac{R.E}{W} = \frac{m_1 \left(h_1 - h_{11}\right)}{m_1 \left(h_2 - h_1\right) + m_2 \left(h_5 - h_4\right)} = \frac{210Q}{P \times 60}$$

4.4 Cascade Refrigeration system

The cascade refrigeration system consists of two or more vapour compression systems in series which use refrigerants with progressively lower boiling two-stage cascade system using two refrigerants is shown in Fig. 4.5 (a) and its p-h diagrams are shown in Fig. 4.5 (b) respectively.

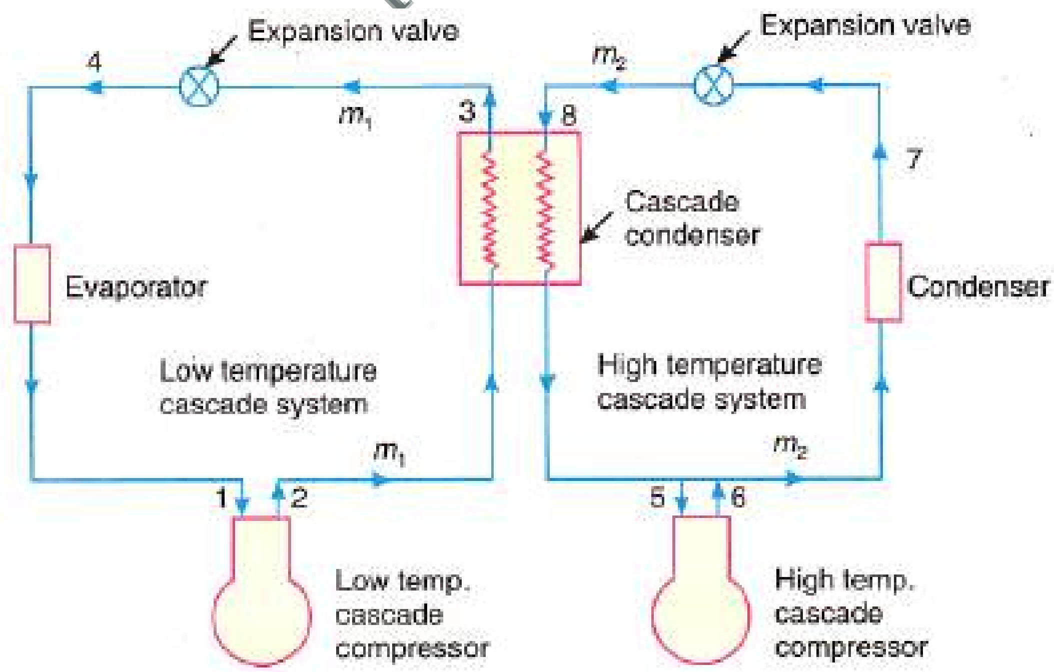
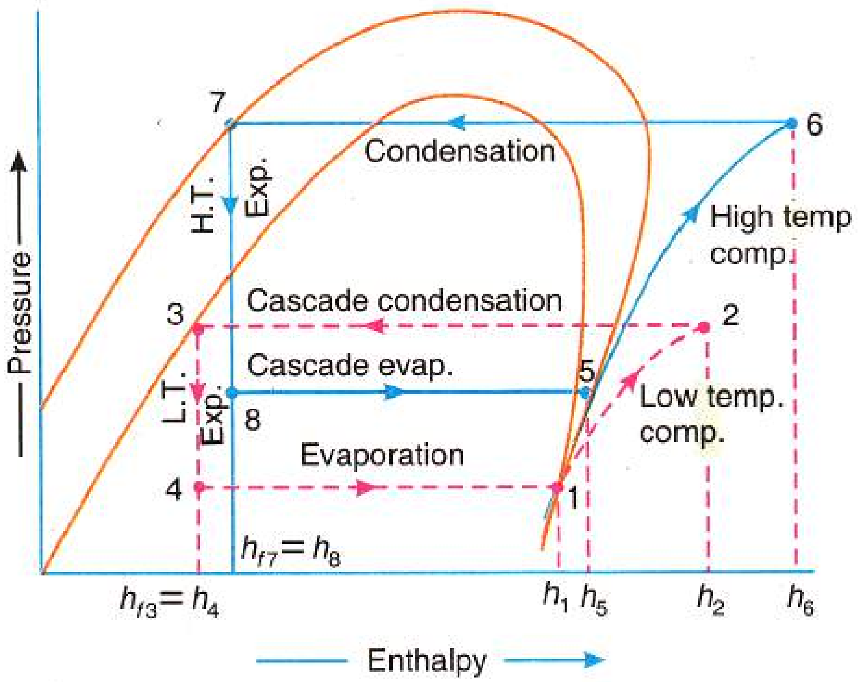


Fig. 4.5 (a) Two stage cascade system



(*b*) *p-h* diagram.

Fig. 4.5 (b) Two stage cascade system

- In this system, a cascade condenser serves as an evaporator for the high temperature cascade system and a condenser for the low temperature cascade system. The only useful refrigerating effect is produced in the evaporator of the low temperature cascade system. The principal advantage of the cascade system is that it permits the use of two different refrigerants. The high temperature cascade system uses a refrigerant with high boiling temperature such as R-12 or R-22. The low temperature cascade system uses a refrigerant with low boiling temperature such as R-13. These low boiling temperature refrigerants have extremely high pressure which ensures a smaller compressor displacement in the low temperature cascade system and a higher coefficient of performance.
- The cascade system was first used by Pietet in 1877 for liquefaction of oxygen, employing sulphur dioxide (SO_2) and carbon dioxide (CO_2) as intermediate refrigerants. Another set of refrigerants commonly used for liquefaction of gases in a three-stage cascade system is ammonia (NH_3), ethylene (C_2H_4) and methane (CH_4).

Coefficient of Performance of a Two-Stage Cascade refrigeration system

- The schematic and p-h diagram of a two-stage cascade system is shown in Fig. 4.5. If Q tonnes of refrigeration is the load on the low temperature cascade system, the mass of refrigerant flowing through the low temperature cascade system is given by,

$$m_1 = \frac{210Q}{h_1 - h_4} \text{ kg/min}$$

The mass of refrigerant m_2 required in the high temperature cascade system in order to liquefy the refrigerant of low temperature cascade system in the cascade condenser may be obtained by balancing the heat of both the systems. In other words, the heat absorbed in high temperature cascade system must be equal to the heat rejected in the low temperature cascade system. Mathematically,

$$m_2\left(h_5-h_8\right)=m_1\left(h_2-h_{f3}\right)$$

$$m_2=\frac{m_1\left(h_2-h_{f3}\right)}{h_5-h_8}$$

We know that the total work done by the system,

$$W=m_1\left(h_2-h_1\right)+m_2\left(h_6-h_5\right)$$

and refrigerating effect is,

$$R.E=210Q \text{ kJ/min}$$

Coefficient of performance of the system,

$$C.O.P=\frac{R.E}{W}=\frac{210Q}{m_1\left(h_2-h_1\right)+m_2\left(h_6-h_5\right)}$$

and power required to drive the system

$$P=\frac{m_1\left(h_2-h_1\right)+m_2\left(h_6-h_5\right)}{60} \text{ kW}$$

4C

Multiple Evaporator and Compressor System

Course Contents

4.1 Introduction

– In the single evaporator system entire load is carried by a single evaporator at one temperature. But in many refrigeration installations different temperatures are required to be maintained at various points in the plant such as in hotels, large restaurants, institution, industrial plants and food markets where the food quantities are received in large quantities and stored at different temperatures. For example, the fresh fruits, fresh vegetables, fresh cut meats, frozen product, dairy products, canned goods, bottled goods have all different conditions of temperature and humidity for storage. In such cases, each location is cooled by its own evaporator in order to obtain more satisfactory control of the condition.

4.2 Types of Multiple evaporator and compressor system

– Following types of multiple evaporator and compressor systems are important from the subject point of view:

1. Multiple evaporators at the same temperature with single compressor and expansion valve.
2. Multiple evaporators at different temperature with single compressor, individual expansion valves and back pressure valves.
3. Multiple evaporators at different temperature with single compressor, multiple expansion valves and back pressure valves.
4. Multiple evaporators at different temperature with individual compressors and individual expansion valves
5. Multiple evaporators at different temperatures with individual compressors and multiple expansion valves.
6. Multiple evaporators at different temperatures with compound compression and individual expansion valves.
7. Multiple evaporators, at different temperature with compound compression, individual expansion valves and flash intercoolers.
8. Multiple evaporators at different temperatures with compound compression, multiple expansion valves and flash intercoolers.

All above mentioned types of multiple evaporators and compressor systems are discussed in detail in the following pages

4.2.1 Multiple evaporators at the same temperature with single compressor and expansion valve

– The arrangement of system with three evaporators EP_1, EP_2 and EP_3 at the same temperature with single compressor and expansion valve and corresponding p-h diagram shown in Fig. 4.1 (a) and (b).

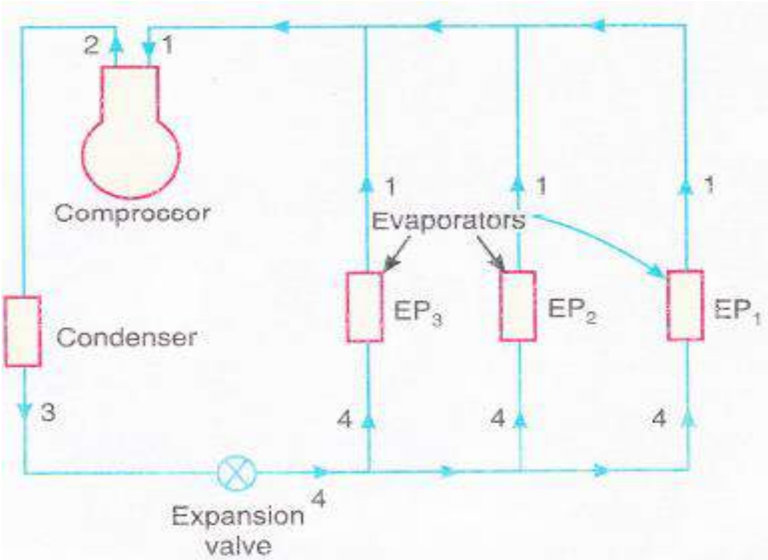


Fig. 4.1 (a) Multiple evaporators at the same temperature with single compressor and expansion valve

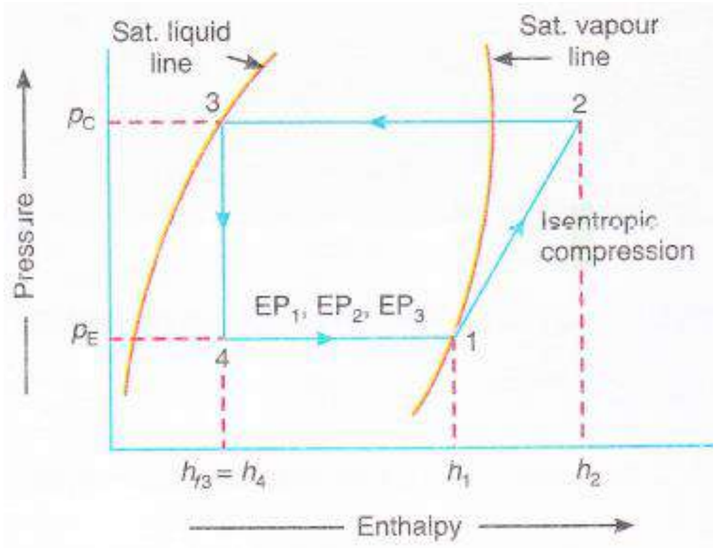


Fig. 4.1 (b) p-h diagram

Let, Q_1, Q_2 and Q_3 = loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1,

$$m_1 = \frac{210Q_1}{h_1 - h_4} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_1,

$$m_2 = \frac{210Q_2}{h_1 - h_4} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the third evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_1 - h_4} \text{ kg/min}$$

Total mass of refrigerant required to be circulated through the evaporators or compressor,

$$m = m_1 + m_2 + m_3$$

Work done in the compressor,

$$W = (m_1 + m_2 + m_3)(h_2 - h_1)$$

Total refrigerating effect is,

$$R.E = (m_1 + m_2 + m_3)(h_1 - h_4)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{(m_1 + m_2 + m_3)(h_1 - h_4)}{(m_1 + m_2 + m_3)(h_2 - h_1)} = \frac{h_1 - h_4}{h_2 - h_1}$$

4.2.2 Multiple Evaporators at Different Temperature with Single Compressor, Individual Expansion Valves and Back Pressure Valves

– The arrangement of system consists of three evaporators EP_1, EP_2 and EP_3 at different temperature with single compressor, individual expansion valves and back pressure valves and corresponding p-h diagram shown in Fig. 4.2 (a) and (b).

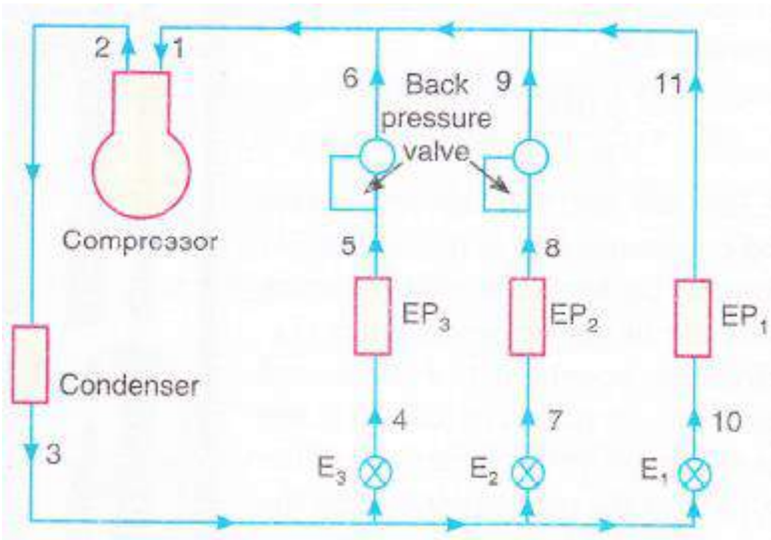


Fig. 4.2 (a) Multiple evaporators at different temperatures with single compressor, individual expansion valves and back pressure valves

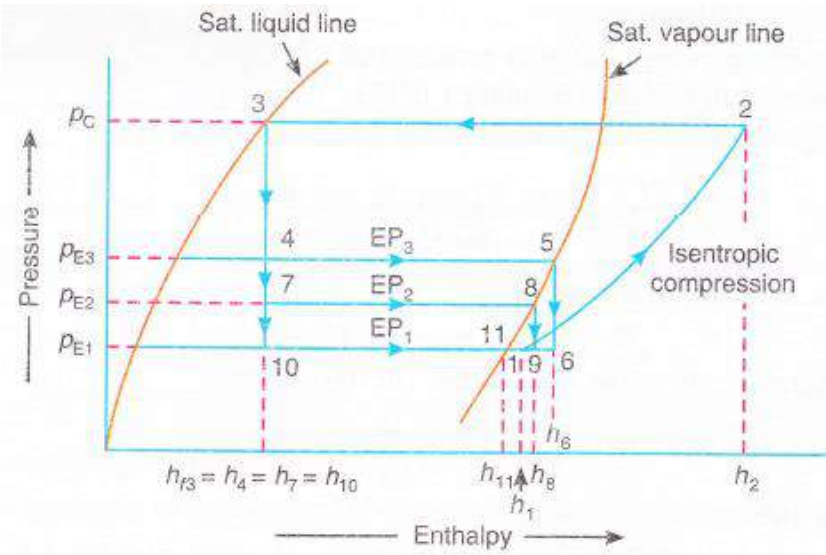


Fig. 4.2 (b) p-h diagram

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1,

$$m_1 = \frac{210Q_1}{h_{11} - h_{10}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_1,

$$m_2 = \frac{210Q_2}{h_8 - h_7} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the third evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_5 - h_4} \text{ kg/min}$$

From Fig. 4.2 (a), we see that the refrigerant corning out of the third evaporator EP_3 at pressure p_{E3} is further expanded through the back pressure valve shown by 5-6, to a pressure of the first evaporator p_{E1}. Similarly, the refrigerant coming out E the second evaporator EP_2 pressure p_{E2}, is further expanded in back pressure valve as shown by 8-9, to a pressure of the first evaporator EP_1. Now the refrigerant leaving the back pressure valves at points 6 and 9 are mixed together with the refrigerant leaving the first evaporator at point 11, at the pressure of the first evaporator (p_{E1}) which is the suction pressure of the compressor.

The condition of the refrigerant after mixing and entering in the compressor is shown by point 1. The enthalpy at this point is given by,

$$h_1 = \frac{m_1 h_{11} + m_2 h_8 + m_3 h_5}{m_1 + m_2 + m_3}$$

Work done in the compressor,

$$W = (m_1 + m_2 + m_3)(h_2 - h_1)$$

Total refrigerating effect is,

$$R.E = m_1(h_{11} - h_{10}) + m_2(h_8 - h_7) + m_3(h_5 - h_4)$$
$$= 210Q_1 + 210Q_2 + 210Q_3$$

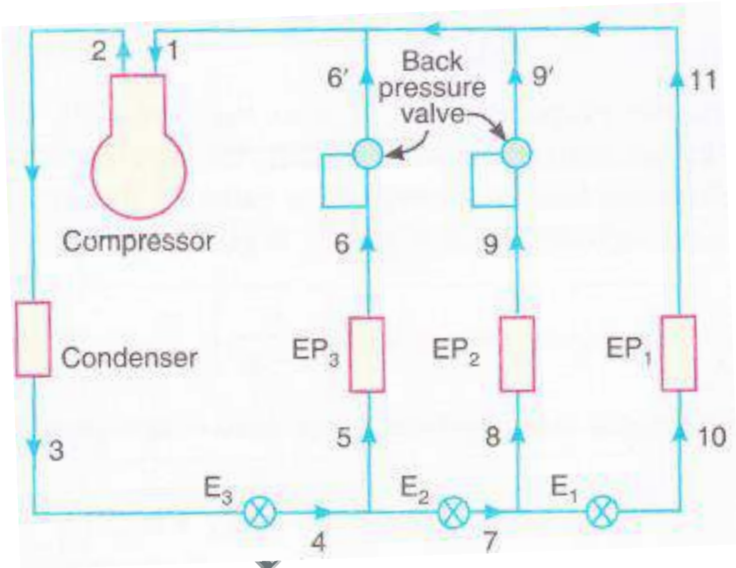
Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210(Q_1 + Q_2 + Q_3)}{(m_1 + m_2 + m_3)(h_2 - h_1)}$$

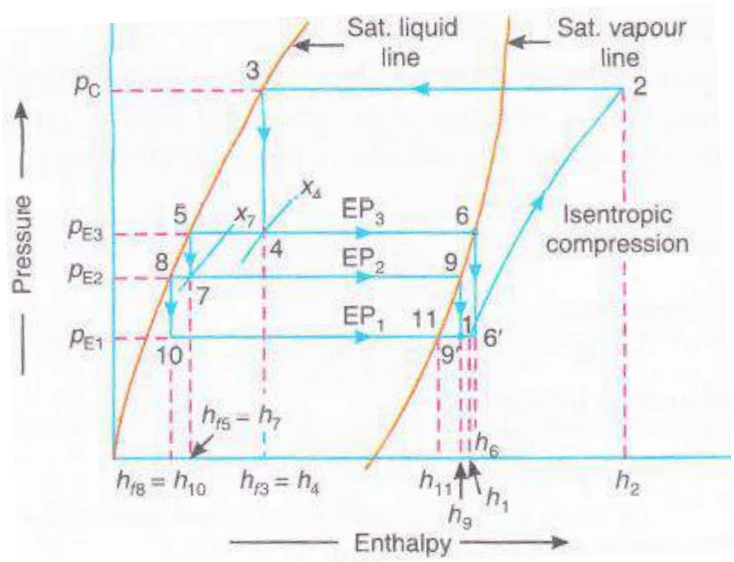
4.2.3 Multiple Evaporators at Different Temperature with Single Compressor, Multiple Expansion Valves and Back Pressure Valves

- The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different tempera lures with single compressor, multiple expansion valves E_1, E_2, and E_3 and back pressure valves and the corresponding p-h diagram is shown in as shown in Fig. 4.3 (a) and (b).
- In this system the refrigerant flows from the condenser through expansion valve E_3 where its pressure reduced from the condenser pressure p_c to the pressure of third

evaporator (i.e. highest temperature evaporator) EP_3 (i.e. p_{E3}). All the vapour formed after leaving the expansion valve E_3 plus enough liquid to take care of the load of evaporator EP_3 passes through this evaporator EP_3. The remaining refrigerant then flows through the expansion valve E_2 where its pressure is reduced from p_{E3} to p_{E2}. Again all the vapour formed after leaving the expansion valve E_2 plus enough liquid to take care of the load of evaporator EP_2 passes through the evaporator EP_2. The remaining liquid now flows through the expansion valve E_1 and supplies it to first evaporator (i.e. lowest temperature evaporator) EP_1. The vapour refrigerants coming out of the second and third evaporators EP_2 and EP_3 are further expanded through the back pressure valves to reduce their pressures to p_{E1} as shown by 9-9' and 6-6' respectively. Now the refrigerants leaving the back pressure valves at points 6 and 9 are mixed together with the refrigerant leaving the first evaporator at point 11, at pressure p_{E1} which is the suction pressure of the compressor.



4.3 (a) Multiple evaporators at different temperature with single compressor, multiple expansion valves and back pressure valves



4.3 (b) p-h diagram

Let Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration

We know that the mass of refrigerant required to be circulated (at point 10) through the first evaporator or the lowest temperature evaporator EP_1,

$$m_{e1} = m_1 = \frac{210Q_1}{h_{11} - h_{10}} \text{ kg/min}$$

The mass of refrigerant required to be circulated (at point 7) through the second evaporator or the intermediate temperature evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_9 - h_7} \text{ kg/min}$$

We have discussed above that the second evaporator EP_2 is also supplied with the vapour formed during expansion of m_{e1} kg/min refrigerant while passing through expansion valve E_2. If x_7 is the dryness fraction of the refrigerant leaving the expansion valve E_2, then the mass of vapour formed by m_{e1} while passing through the expansion valve E_2 is given by

$$m_{2'} = m_{e1}\left(\frac{x_7}{1 - x_7}\right)$$

Total mass of refrigerant though the second evaporator EP_2 (at point 8),

$$m_{e2} = m_2 + m_{2'} = m_2 + m_{e1}\left(\frac{x_7}{1 - x_7}\right)$$

Similarly the mass of refrigerant required to be circulated (at point 4) through the third evaporator or the highest temperature evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_6 - h_4} \text{ kg/min}$$

The evaporator EP_3 is also supplied with the vapour formed by expansion of $\left(m_{e1} + m_{e2}\right)$ kg/min refrigerant while passing through the expansion valve E_3. If x_4 is dryness fraction of refrigerant leaving the expansion valve E_3, then mass of vapour formed by $\left(m_{e1} + m_{e2}\right)$ while passing through expansion valve E_3 is given by,

$$m_{3'} = \left(m_{e1} + m_{e2}\right)\left(\frac{x_4}{1 - x_4}\right)$$

Total mass of refrigerant though the third evaporator EP_3 (at point 5),

$$m_{e3} = m_3 + m_{3'} = m_3 + \left(m_{e1} + m_{e2}\right)\left(\frac{x_4}{1 - x_4}\right)$$

It may be noted that vapour formed during expansion while passing through expansion valves E_2 and E_3 do not take any part in refrigeration.

The vapour refrigerant coming out of the three evaporators are mixed together and then supplied to the compressor. Let the condition of the refrigerant entering into the compressor shown by point 1 on p-h diagram. The enthalpy at point 1 is given by,

$$h_1 = \frac{m_{e1}h_{11} + m_{e2}h_9 + m_{e3}h_6}{m_{e1} + m_{e2} + m_{e3}}$$

Work done in the compressor,

$$W = (m_{e1} + m_{e2} + m_{e3})(h_2 - h_1)$$

Total refrigerating effect is,

$$R.E = m_1(h_{11} - h_{10}) + m_2(h_9 - h_7) + m_3(h_6 - h_4)$$
$$= 210(Q_1 + Q_2 + Q_3)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210(Q_1 + Q_2 + Q_3)}{(m_1 + m_2 + m_3)(h_2 - h_1)}$$

4.2.4 Multiple Evaporators at Different Temperature with Individual Compressors and Individual Expansion Valves

The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different temperatures with individual compressors C_1, C_2 and C_3 and individual expansion valves E_1, E_2, and E_3 and the corresponding p-h diagram is shown in as shown in Fig. 4.4 (a) and (b).

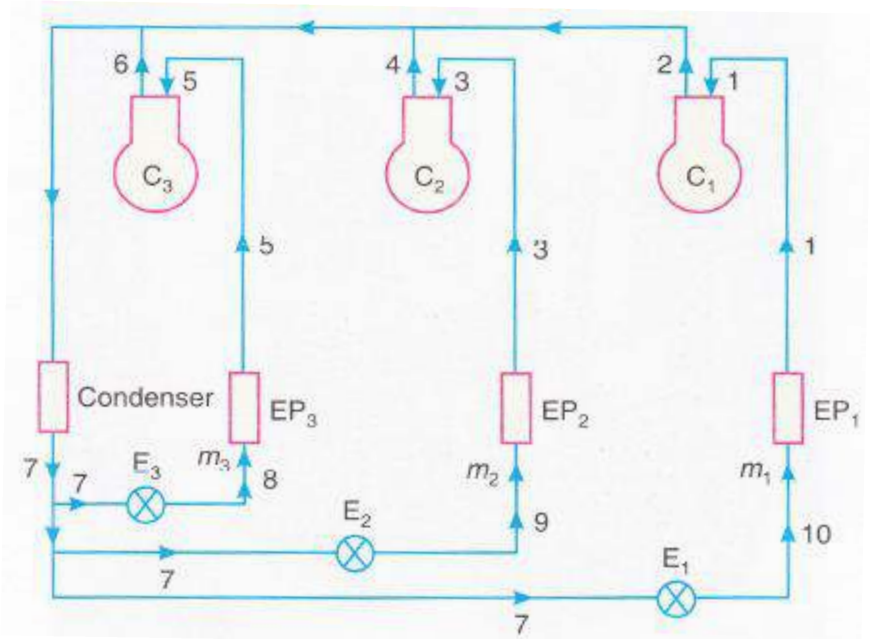


Fig. 4.4 (a) Multiple evaporators at different temperature with individual compressors and individual expansion valves

Work required to drive the compressor C_1,

$$W_1 = m_1(h_2 - h_1)$$

Similarly work required to drive the compressor C_2,

$$W_2 = m_2(h_4 - h_3)$$

Work required to drive the compressor C_1,

$$W_3 = m_3(h_6 - h_5)$$

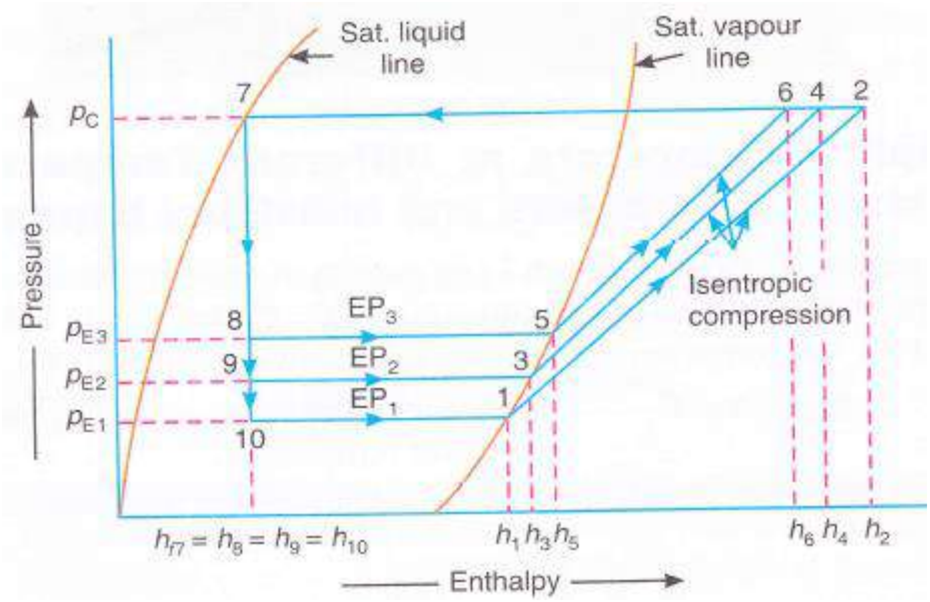


Fig. 4.4 (b) p-h diagram

Total work required in the three compressors

$$W = m_1(h_2 - h_1) + m_2(h_4 - h_3) + m_3(h_6 - h_5)$$

Total refrigerating effect is,

$$R.E = 210(Q_1 + Q_2 + Q_3)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210(Q_1 + Q_2 + Q_3)}{m_1(h_2 - h_1) + m_2(h_4 - h_3) + m_3(h_6 - h_5)}$$

4.2.5 Multiple Evaporators at Different Temperature with Individual Compressors and Multiple Expansion Valves

The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different temperatures with individual compressors C_1, C_2 and C_3 and multiple expansion valves at E_1, E_2, and E_3 and the corresponding p-h diagram is shown in as shown in Fig. 4.5 (a) and (b).

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1,

$$m_1 = \frac{210Q_1}{h_1 - h_{10}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_3 - h_9} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the third evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_5 - h_8} \text{ kg/min}$$

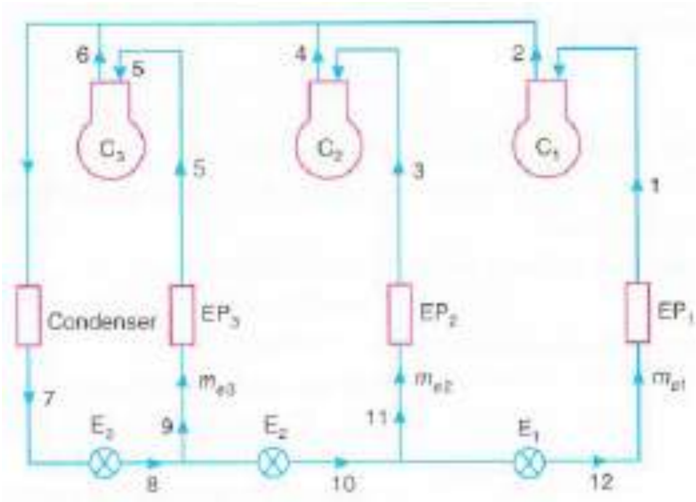


Fig. 4.5 (a) Multiple Evaporators at Different Temperature with Individual Compressors and Multiple Expansion Valves

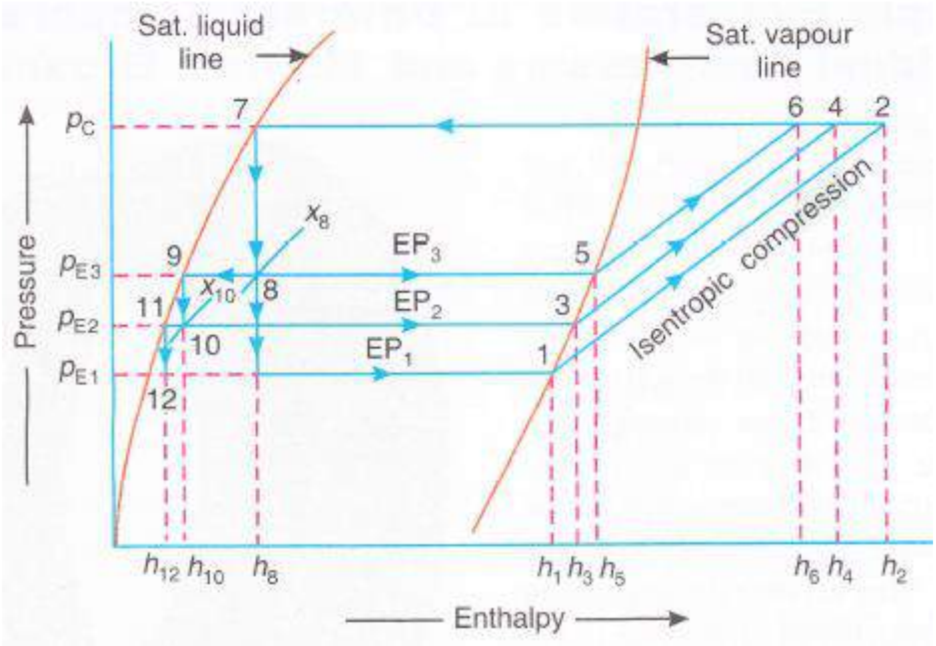


Fig. 4.5 (b) p-h diagram

In this system all the refrigerant from the condenser flows through the expansion valve E_1 where its pressure is reduced from the condenser pressure p_C to the pressure of the third evaporator. All the vapour formed after leaving the expansion valve E_3 plus enough liquid to take care of the load of evaporator EP_3 passes through this evaporator EP_3. The remaining refrigerant then flows through the expansion valve E_2 where its pressure is reduced from p_{E3} to p_{E2}. Again all the vapour formed after leaving the expansion valve E_2 plus enough liquid to take care of the load of evaporator EP_2 passes through this evaporator EP_2. The remaining refrigerant then flows through the expansion valve E_1 which supplies it to the first evaporator (lowest temperature evaporator) EP_1.

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated (at point 12) through the first evaporator EP_1,

$$m_{e1} = m_1 = \frac{210Q_1}{h_1 - h_{12}} \text{ kg/min}$$

Mass of refrigerant required to be circulated (at point 10) through the second evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_3 - h_{10}} \text{ kg/min}$$

We have discussed above that the second evaporator EP_2 is also supplied with the vapour formed during expansion of m_{e1} kg/min refrigerant while passing through expansion valve E_2. If x_{10} is the dryness fraction of the refrigerant leaving the expansion valve E_2, then the mass of vapour formed by m_{e1} while passing through the expansion valve E_2 is given by,

$$m_{2'} = m_{e1}\left(\frac{x_{10}}{1 - x_{10}}\right)$$

Total mass of refrigerant though the second evaporator EP_2 (at point 11),

$$m_{e2} = m_2 + m_{2'} = m_2 + m_{e1}\left(\frac{x_{10}}{1 - x_{10}}\right)$$

Similarly the mass of refrigerant required to be circulated (at point 8) through the third evaporator or the highest temperature evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_5 - h_8} \text{ kg/min}$$

The evaporator EP_3 is also supplied with the vapour formed by expansion of $\left(m_{e1} + m_{e2}\right)$ kg/min refrigerant while passing through the expansion valve E_3. If x_8 is dryness fraction of refrigerant leaving the expansion valve E_3, then mass of vapour formed by $\left(m_{e1} + m_{e2}\right)$ while passing through expansion valve E_3 is given by,

$$m_{3'} = \left(m_{e1} + m_{e2}\right)\left(\frac{x_8}{1 - x_8}\right)$$

Total mass of refrigerant though the third evaporator EP_3 (at point 9),

$$m_{e3} = m_3 + m_{3'} = m_3 + \left(m_{e1} + m_{e2}\right)\left(\frac{x_8}{1 - x_8}\right)$$

Total mass of refrigerant flowing though the condenser,

$$m = m_{e1} + m_{e2} + m_{e3}$$

Work required to drive the compressor C_1,

$$W_1 = m_1\left(h_2 - h_1\right)$$

Similarly work required to drive the compressor C_2,

$$W_2 = m_2\left(h_4 - h_3\right)$$

Work required to drive the compressor C_1,

$$W_3 = m_3\left(h_6 - h_5\right)$$

Total work required in the three compressors

$$W = m_1\left(h_2 - h_1\right) + m_2\left(h_4 - h_3\right) + m_3\left(h_6 - h_5\right)$$

Total refrigerating effect is,

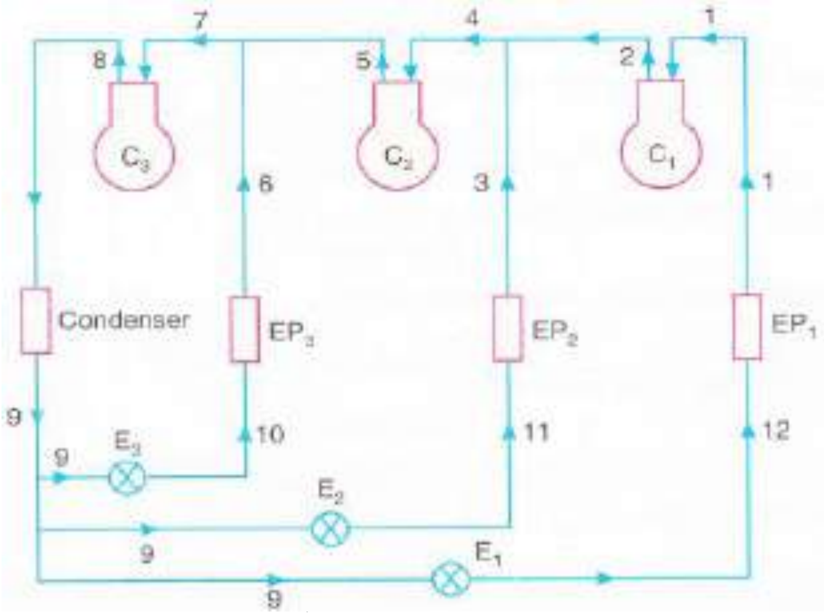
$$R.E = 210\left(Q_1 + Q_2 + Q_3\right)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210\left(Q_1 + Q_2 + Q_3\right)}{m_1\left(h_2 - h_1\right) + m_2\left(h_4 - h_3\right) + m_3\left(h_6 - h_5\right)}$$

4.2.6 Multiple Evaporators at Different Temperature with Compound Compression and Individual Expansion Valves

The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different temperatures with compound compression and individual expansion valves E_1, E_2, and E_3 and the corresponding p-h diagram is shown in as shown in Fig. 4.6 (a) and (b).



4.6 (a) Multiple evaporators at different temperature with compound compression and individual expansion valves

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1,

$$m_1 = \frac{210Q_1}{h_1 - h_{12}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_2 - h_{11}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the third evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_5 - h_8} \text{ kg/min}$$

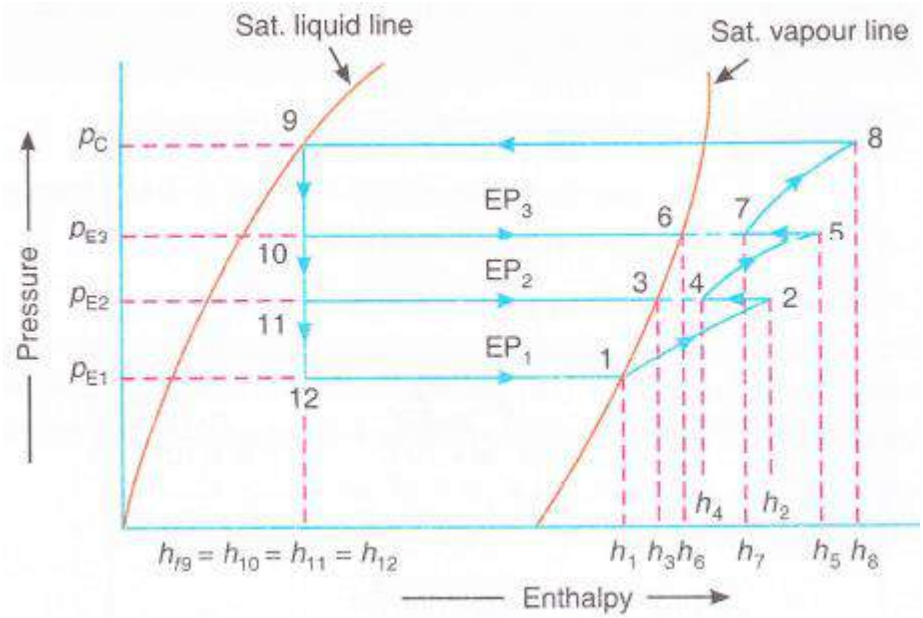


Fig. 4.6 (b) p-h diagram

From Fig. 4.6 (a), we see that the mass of refrigerant (m_1) coming out from the first compressor C_1 is mixed with the mass of refrigerant (m_2) coming out from the second evaporator EP_2 before entering into the second compressor C_2. The condition of the mixed refrigerant entering into the second compressor C_2 is shown by point 4 on the p-h diagram. The enthalpy at point 4 is given by

$$(m_1 + m_2)h_4 = m_1 h_2 + m_2 h_3$$

$$h_4 = \frac{m_1 h_2 + m_2 h_3}{m_1 + m_2}$$

Similarly the refrigerant coming out from the second compressor C_2 (*i.e.* $m_1 + m_2$) is mixed with the refrigerant coming out from the third evaporator EP_3 before entering into the third compressor C_3. The condition of the mixed refrigerant entering into the third compressor C_3 is shown by point 7 on the p-h diagram. The enthalpy at point 7 is given by

$$(m_1 + m_2 + m_3)h_7 = (m_1 + m_2)h_5 + m_3 h_6$$

$$h_7 = \frac{(m_1 + m_2)h_5 + m_3 h_6}{m_1 + m_2 + m_3}$$

Work required to drive the compressor C_1,

$$W_1 = m_1(h_2 - h_1)$$

Similarly work required to drive the compressor C_2,

$$W_2 = (m_1 + m_2)(h_5 - h_4)$$

Work required to drive the compressor C_3,

$$W_3 = (m_1 + m_2 + m_3)(h_8 - h_7)$$

Total work required in the three compressors

$$W = W_1 + W_2 + W_3$$

Total refrigerating effect is,

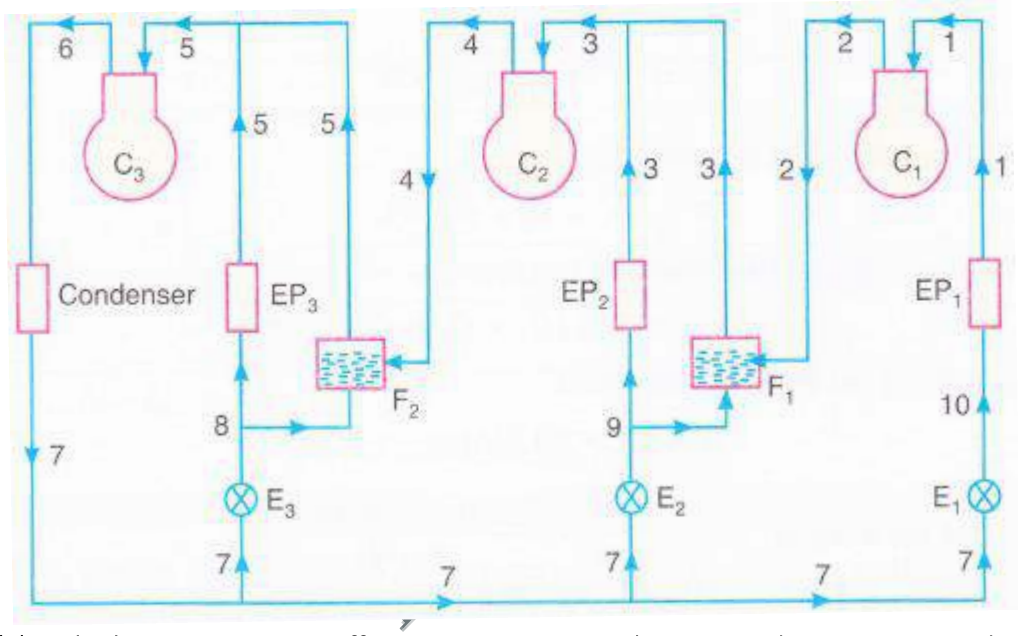
$$R.E = 210\left(Q_1 + Q_2 + Q_3\right)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210\left(Q_1 + Q_2 + Q_3\right)}{W}$$

4.2.7 Multiple Evaporators at Different Temperature with Compound Compression, Individual Expansion Valves and Flash Intercoolers

The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different temperatures with compound compression (three stage) and individual expansion valves E_1, E_2, and E_3. The flash intercooler F_1 and F_2 are provided in the system for cooling the superheated vapour leaving the compressors C_1 and C_2 respectively and the corresponding p-h diagram is shown in as shown in Fig. 4.7 (a) and (b).



4.7 (a) Multiple Evaporators at Different Temperature with Compound Compression, Individual Expansion Valves and Flash Intercoolers

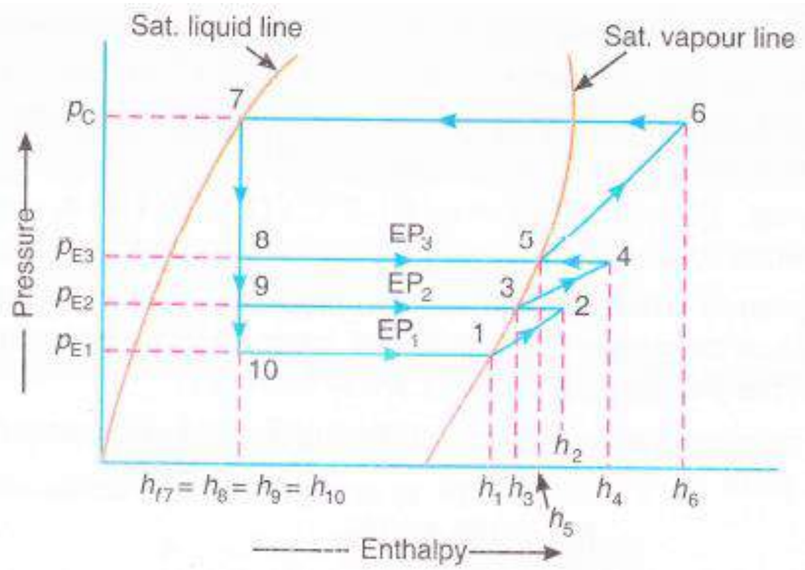


Fig. 4.7 (b) p-h diagram

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1 or through the first compressor C_1

$$m_{e1} = m_1 = \frac{210Q_1}{h_1 - h_{10}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_3 - h_9} \text{ kg/min}$$

Mass of refrigerant required to be by-passed (at point 9) to the flash intercooler F_1 for desuperheating the superheated vapour refrigerant m_{c1} coming from first compressor C_1
To the dry saturated condition as at point 3 is given by

$$m_{2'} = \frac{m_{c1}(h_2 - h_3)}{h_3 - h_9}$$

Total mass of refrigerant passing through the second compressor C_2,

$$m_{c2} = m_{c1} + m_2 + m_{2'}$$

Mass of refrigerant required to be circulated through the third evaporator EP_3,

$$m_3 = \frac{210Q_3}{h_5 - h_8} \text{ kg/min}$$

Mass of refrigerant required to be by-passed (at point 8) to the flash intercooler F_2 for desuperheating the superheated vapour refrigerant m_{c2} coming from second compressor C_2 to the dry saturated condition as at point 5 is given by,

$$m_{3'} = \frac{m_{c2}(h_4 - h_5)}{h_5 - h_8}$$

Total mass of refrigerant passing through the third compressor C_3,

$$m_{c3} = m_{c2} + m_3 + m_{3'}$$

Work required to drive the compressor C_1,

$$W_1 = m_{c1}(h_2 - h_1)$$

Similarly work required to drive the compressor C_2,

$$W_2 = m_{c2}(h_4 - h_3)$$

Work required to drive the compressor C_3,

$$W_3 = m_{c3}(h_6 - h_5)$$

Total work required in the three compressors

$$W = W_1 + W_2 + W_3$$

Total refrigerating effect is,

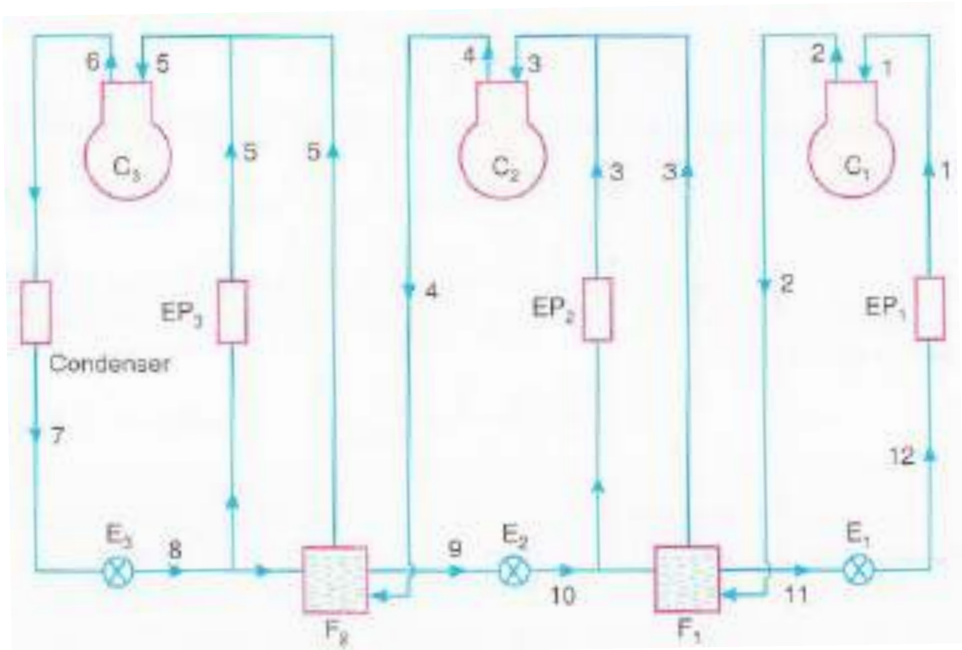
$$R.E = 210(Q_1 + Q_2 + Q_3)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210\left(Q_1 + Q_2 + Q_3\right)}{W}$$

4.2.8 Multiple Evaporators at Different Temperature with Compound Compression, Multiple Expansion Valves and Flash Intercoolers

The arrangement consists of three evaporators EP_1, EP_2 and EP_3 operating at different temperatures with compound compression (three stage) and multiple expansion valves E_1, E_2, and E_3. The flash intercooler F_1 and F_2 are provided in the system for cooling the superheated vapour leaving the compressors C_1 and C_2 respectively and the corresponding p-h diagram is shown in as shown in Fig. 4.8 (a) and (b).



4.8 (a) Multiple Evaporators at Different Temperature with Compound Compression, Multiple Expansion Valves and Flash Intercoolers

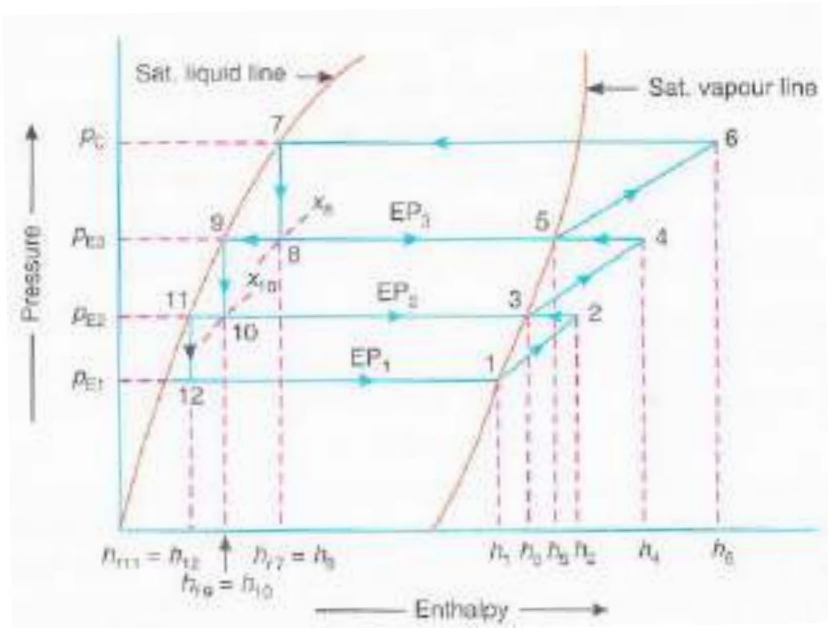


Fig.4.8 (b) p-h diagram

Let, Q_1, Q_2 and Q_3 = Loads on the evaporators EP_1, EP_2 and EP_3 respectively in tonnes of refrigeration.

Mass of refrigerant required to be circulated through the first evaporator EP_1 or passing through the first compressor C_1

$$m_{c1} = m_1 = \frac{210Q_1}{h_1 - h_{12}} \text{ kg/min}$$

Mass of refrigerant required to be circulated through the second evaporator EP_2,

$$m_2 = \frac{210Q_2}{h_3 - h_{10}} \text{ kg/min}$$

Mass of refrigerant required (at point 10) in the flash intercooler F_1 for desuperheating the superheated vapour refrigerant m_{c1} coming from first compressor C_1 to the dry saturated condition as at point 3 is given by,

$$m_{2'} = \frac{m_{c1}\left(h_2 - h_3\right)}{h_3 - h_{10}}$$

The second evaporator EP_2 is supplied with the vapour formed during expansion of m_{c1} kg/min refrigerant while passing through expansion valve E_2. If x_{10} is the dryness fraction of the refrigerant leaving the expansion valve E_2, then the mass of vapour formed by m_{e1} while passing through the expansion valve E_2 is given by,

$$m_{2''} = m_{c1}\left(\frac{x_{10}}{1 - x_{10}}\right)$$

Total mass of refrigerant passing through the second compressor C_2,

$$m_{c2} = m_{c1} + m_2 + m_{2'} + m_{2''}$$

Similarly the mass of refrigerant required to be circulated through the third evaporator EP_3

$$m_3 = \frac{210Q_3}{h_5 - h_8} \text{ kg/min}$$

Mass of refrigerant required (at point 8) in the flash intercooler F_2 for desuperheating the superheated vapour refrigerant m_{c2} coming from second compressor C_2 to the dry saturated condition as at point 5 is given by,

$$m_{3'} = \frac{m_{c2}\left(h_4 - h_5\right)}{h_5 - h_8}$$

The evaporator EP_3 is also supplied with the vapour formed by expansion of m_{c2} kg/min refrigerant while passing through the expansion valve E_3. If x_8 is dryness fraction of refrigerant leaving the expansion valve E_3, then mass of vapour formed by m_{c2} while passing through expansion valve E_3 is given by,

$$m_{3''} = m_{c2}\left(\frac{x_8}{1 - x_8}\right)$$

Total mass of refrigerant though the third compressor C_3

$$m_{c3} = m_{c2} + m_3 + m_{3'} + m_{3''}$$

Work required to drive the compressor C_1,

$$W_1 = m_{c1}\left(h_2 - h_1\right)$$

Similarly work required to drive the compressor C_2,

$$W_2 = m_{c2}\left(h_4 - h_3\right)$$

Work required to drive the compressor C_3,

$$W_3 = m_{c3}\left(h_6 - h_5\right)$$

Total work required in the three compressors

$$W = m_{c1}\left(h_2 - h_1\right) + m_{c2}\left(h_4 - h_3\right) + m_{c3}\left(h_6 - h_5\right)$$

Total refrigerating effect is,

$$R.E = 210\left(Q_1 + Q_2 + Q_3\right)$$

Coefficient of performance of the system,

$$C.O.P = \frac{R.E}{W} = \frac{210\left(Q_1 + Q_2 + Q_3\right)}{m_{c1}\left(h_2 - h_1\right) + m_{c2}\left(h_4 - h_3\right) + m_{c3}\left(h_6 - h_5\right)}$$

5

Vapour Absorption Refrigeration System

Course Contents

5.1 Introduction

- The vapour absorption refrigeration system is one of the oldest methods of producing refrigerating effect. The principle of vapour absorption was first discovered by Michael Faraday in 1824 while performing a set of experiments to liquify certain gases. The first vapour absorption refrigeration machine was developed by a French scientist, Ferdinand Cane, in 1860. This system may be used in both the domestic and large industrial refrigerating plants. The refrigerant, commonly used in a vapour absorption system, is ammonia.
- The vapour absorption system uses heat energy, instead of mechanical energy as in vapour compression systems, in order to change the conditions of the refrigerant required for the operation of the refrigeration cycle. We have discussed in the previous chapters that the function of a compressor, in a vapour compression system, is to withdraw the vapour refrigerant from the evaporator. It then raises its temperature and pressure higher than the cooling agent in the condenser so that the higher pressure vapour can reject heat in the condenser. The liquid refrigerant leaving the condenser is now ready to expand to the evaporator conditions again.
- In the vapour absorption system, the compressor is replaced by an absorber, a pump, a generator and a pressure reducing valve. These components in vapour absorption system perform the same function as that of a compressor in vapour compression system. In this system, the vapour refrigerant from the evaporator is drawn into an absorber where it is absorbed by the weak solution of the refrigerant forming a strong solution. This strong solution is pumped to the generator where it is heated by some external source. During the heating process, the vapour refrigerant is driven off by the solution and enters into the condenser where it is liquified. The liquid refrigerant then flows into the evaporator and thus the cycle is completed.

5.2 Simple Vapour Absorption System

- The simple vapour absorption system, as shown in Fig. 5.1, consists of an absorber, a pump, a generator and a pressure reducing valve to replace the compressor of vapour compression system. The other components of the system are condenser, receiver, expansion valve and evaporator as in the vapour compression system.
- In this system, the low pressure ammonia vapour leaving the evaporator enters the absorber where it is absorbed by the cold water in the absorber. The water has the ability to absorb very large quantities of ammonia vapour and the solution, thus formed, is known as aqua-ammonia. The absorption of ammonia vapour in water lowers the pressure in the absorber which in turn draws more ammonia vapour from the evaporator and thus raises the temperature of solution. Some form of cooling arrangement (usually water cooling) is employed in the absorber to remove the heat of solution evolved there. This is necessary in order to increase the absorption capacity of water because at higher temperature water absorbs less ammonia vapour. The strong solution thus formed in the absorber is pumped to the generator by the liquid pump.

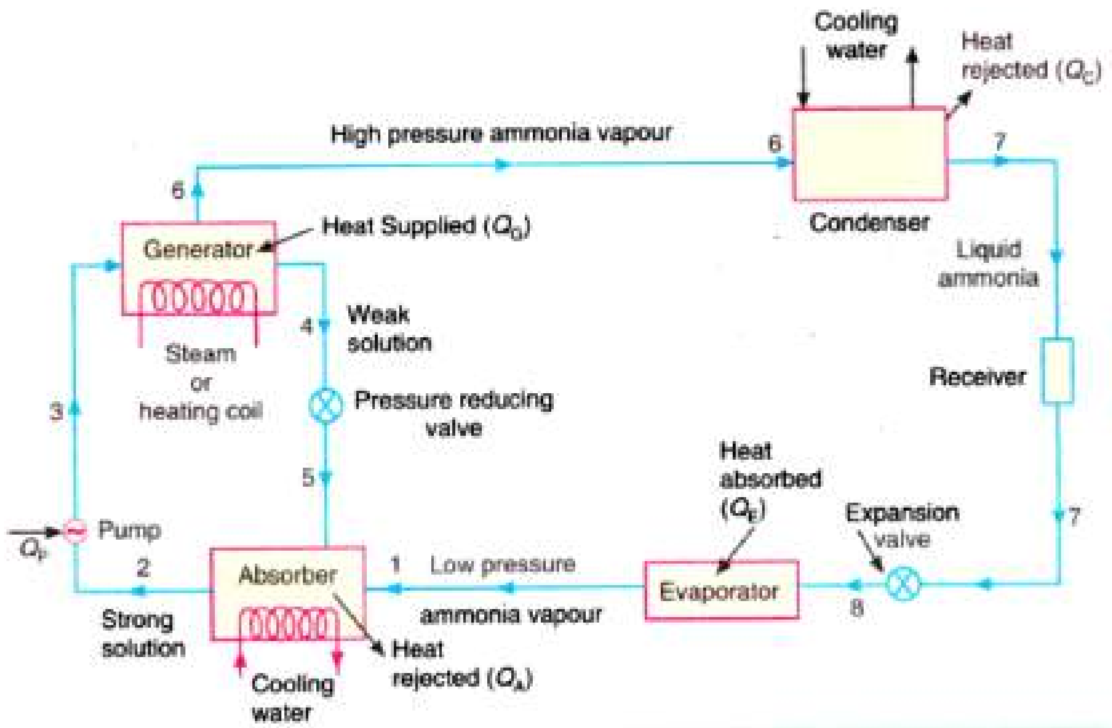


Fig. 5.1 Simple vapour absorption system

– The pump increases the pressure of the solution up to 10 bar. The strong solution of ammonia in the generator is heated by some external source such as gas or steam. During the heating process, the ammonia vapour is driven off the solution at high pressure leaving behind the hot weak ammonia solution in the generator. This weak ammonia solution flows back to the absorber at low pressure after passing through the pressure reducing valve. The high pressure ammonia vapour from the generator is condensed in the condenser to high pressure liquid ammonia. This liquid ammonia is passed to the expansion valve through the receiver and then to the evaporator. This completes the simple vapour absorption cycle.

5.3 Practical Vapour Absorption System

– The simple absorption system as discussed in the previous article is not very economical. In order to make the system more practical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in Fig. 5.2. These accessories help to improve the performance and working of the plant, as discussed below:

1. Analyser. When ammonia is vaporised in the generator, some water is also vaporised and will flow into the condenser along with the ammonia vapour in the simple system. If these unwanted water particles are not removed before entering into the condenser, they will enter into the expansion valve where they freeze and choke the pipeline. In order to remove these unwanted particles flowing to the condenser, an analyser is used. The analyser may be built as an integral part of the generator or made as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier are introduced at the top of the analyser and flow downward over the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator.

The vapour is cooled and most of the water vapour condenses, so that mainly ammonia vapour (approximately 99%) leaves the top of the analyser. Since the aqua is heated by the vapour, less external heat is required in the generator.

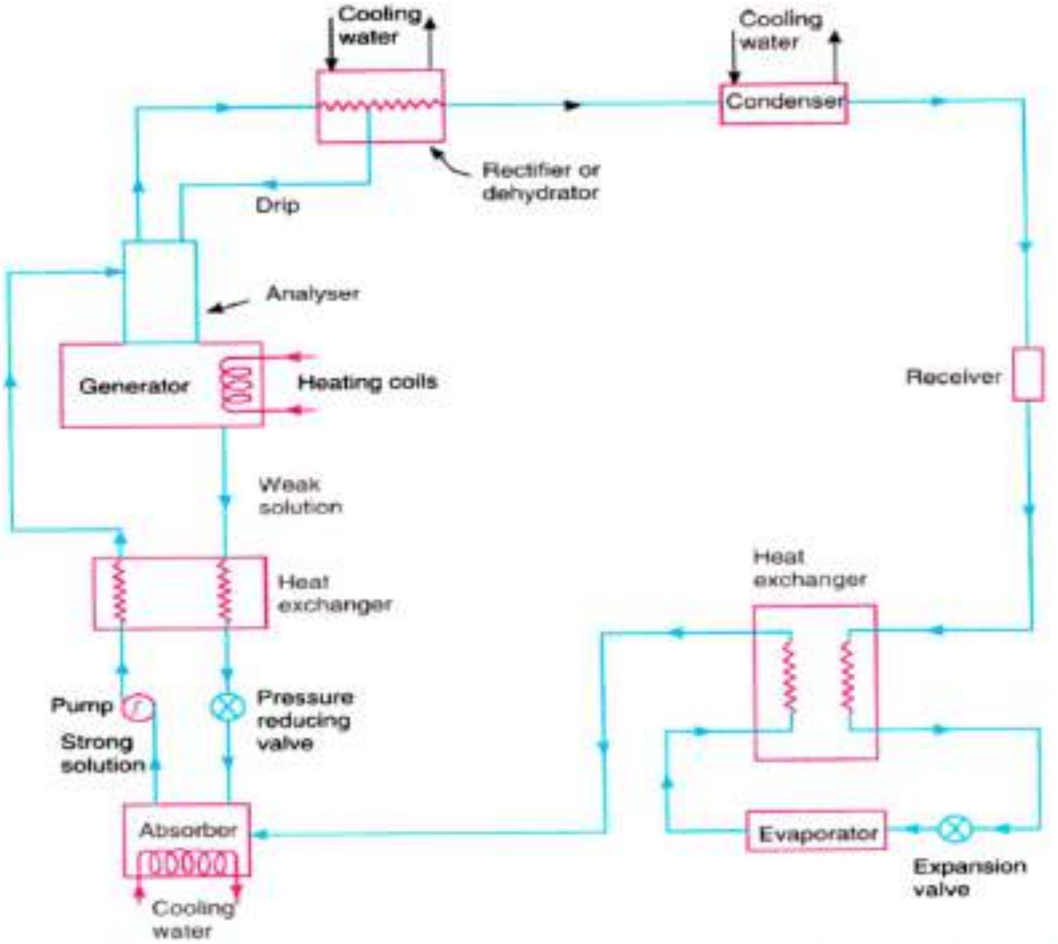


Fig. 5.2 Practical vapour absorption system

2. Rectifier. In case the water vapour are not completely removed in the analyser, a closed type vapour cooler called rectifier (also known as dehydrator) is used. It is generally water cooled and may be of the double pipe, shell and coil or shell and tube type. Its function is to cool further the ammonia vapour leaving the analyser so that the remaining water vapour are condensed. Thus, only dry or anhydrous ammonia vapour flow to the condenser. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

Note: A strong ammonia solution contains as much ammonia as possible whereas a weak ammonia solution contains considerably less ammonia.

3. Heat exchangers. The heat exchanger provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to the generator and the amount of cooling required for the absorber. Thus the economy of the plant increase.

The heat exchanger provided between the condenser and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub-cooled by the low temperature ammonia vapour from the evaporator as shown in Fig. 6.2. This sub-cooled liquid is now passed to the expansion valve and then to the evaporator.

In this system, the net refrigerating effect is the heat absorbed by the refrigerant in the evaporator. The total energy supplied to the system is the sum of work done by the pump and the heat supplied in the generator. Therefore, the coefficient of performance of the system is given by,

$$\text{C.O.P} = \frac{\text{Heat absorbed in evaporator}}{\text{Work done by pump + Heat supplied in generator}}$$

5.4 Thermodynamic Requirements of Refrigerant- Absorbent Mixture

– The two main thermodynamic requirements of the refrigerant-absorbent mixture are as follows:

  1. Solubility requirement. The refrigerant should have more than Raoult's law solubility in the absorbent so that a strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerant vapour.

  2. Boiling points requirement. There should be a large difference in the normal boiling points of the two substances, at least 200°C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus, almost absorbent-free refrigerant is boiled off from the generator and the absorbent alone returns to the absorber.

  In addition, the refrigerant-absorbent mixture should possess the following desirable characteristics:

  a. It should have low viscosity to minimise pump work.
  b. It should have low freezing point.
  c. It should have good chemical and thermal stability.
  d. The irreversible chemical reactions of all kinds such as decomposition, polymerization, corrosion etc. are to be avoided.

Property of Ideal Refrigerant-Absorbent Combination

The ideal refrigerant-absorbent combination should possess the following qualities:

1. The refrigerant should have high affinity for the absorber at low temperature and less affinity at high temperature.
2. The combination should have high degree of negative deviation from Raoult's law.
3. The mixture should have low specific heat and low viscosity.
4. The mixture (solution) should be non-corrosive.
5. The mixture should have a small heat.
6. The mixture should have low freezing point.
7. There should be a large difference in normal boiling points of the refrigerants and the absorbent.

5.5 Advantages of Vapour Absorption Refrigeration System over Vapour Compression Refrigeration System

– Following are the advantages of vapour absorption system over vapour compression system:
  1. In the vapour absorption system, the only moving part of the entire system is a pump which has a small motor. Thus, the operation of this system is essentially quiet and is subjected to little wear. The vapour compression system of the same capacity has more wear, tear and noise due to moving parts of the compressor.
  2. The vapour absorption system uses heat energy to change the condition of the refrigerant from the evaporator. The vapour compression system uses mechanical energy to change the condition of the refrigerant from the evaporator.
  3. The vapour absorption systems are usually designed to use steam, either at high pressure or low pressure. The exhaust steam from furnaces and solar energy may also be used. Thus this system can be used where the electric power is difficult to obtain or is very expensive.
  4. The vapour absorption systems can operate at reduced evaporator pressure and temperature by increasing the steam pressure to the generator, with little decrease in capacity. But the capacity of vapour compression system drops rapidly with lowered evaporator pressure.
  5. The load variations do not affect the performance of a vapour absorption system. The load variations are met by controlling the quantity of aqua circulated and the quantity of steam supplied to the generator. The performance of a vapour compression system at partial loads is poor.
  6. In the vapour absorption system, the liquid refrigerant leaving the evaporator has no bad effect on the system except that of reducing the refrigerating effect. In the vapour compression system, it is essential to superheat the vapour refrigerant leaving the evaporator so that no liquid may enter the compressor.
  7. The vapour absorption systems can be built in capacities well above 1000 tonnes of refrigeration each, which is the largest size for single compressor units.
  8. The space requirements and automatic control requirements favour the absorption system more and more as the desired evaporator temperature drops.

5.6 Domestic Refrigerator (NH_2-H_2) Refrigerator

– This type of refrigerator is also called three-fluid absorption system. The main purpose of this system is to eliminate the pump so that in the absence of moving parts, the machine becomes noiseless. The three fluids used in this system are ammonia, hydrogen and water. The ammonia is used as a refrigerant because it possesses most of the desirable properties. It is toxic, but due to absence of moving parts, there is very little chance for the leakage and the total amount of refrigerant used is small. The hydrogen, being the lightest gas, is used to increase the rate of evaporation of the liquid ammonia passing through the evaporator. The hydrogen is also non-corrosive and insoluble in water. This is used in the low-pressure side of the system. The water is used as a solvent because it has the ability to absorb ammonia readily. The principle of operation of a domestic Electrolux type refrigerator, as shown in Fig. 6.3 is discussed below:

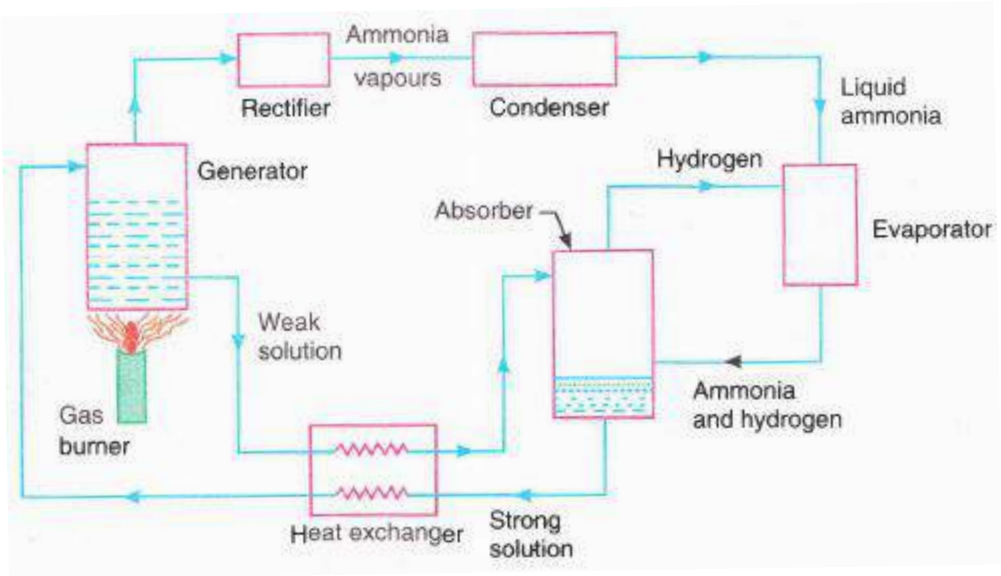


Fig. 5.3 Domestic Electrolux type refrigerator

- The strong ammonia solution from the absorber through heat exchanger is heated in the generator by applying heat from an external source, usually a gas burner. During this heating process, ammonia vapour are removed from the solution and passed to the condenser. A rectifier or a water separator fitted before the condenser removes water vapour carried with the ammonia vapour, so that dry ammonia vapour are supplied to the condenser. This water vapour, if not removed, will enter into the evaporator causing freezing and choking of the machine. The hot weak solution left behind in the generator flows to the absorber through the heat exchanger. This hot weak solution while passing through the exchanger is cooled. The heat removed by the weak solution is utilised in raising the temperature of strong solution passing through the heat exchanger. In this way, the absorption is accelerated and the improvement in the performance of a plant is achieved.
- The ammonia vapour in the condenser are condensed by using external cooling source. The liquid refrigerant leaving the condenser flows under gravity to the evaporator where it meets the hydrogen gas. The hydrogen gas which is being fed to the evaporator permits the liquid ammonia to evaporate at a low pressure and temperature according to Dalton's principle. During the process of evaporation, the ammonia absorbs latent heat from the refrigerated space and thus produces cooling effect. The mixture of ammonia vapour and hydrogen is passed to the absorber where ammonia is absorbed in water while the hydrogen rises to the top and flows back to the evaporator. This completes the cycle. The coefficient of performance of this refrigerator is given by:

$$\text{C.O.P} = \frac{\text{Heat absorbed in the evaporator}}{\text{Heat supplied in evaporator}}$$

5.7 Lithium Bromide Absorption Refrigeration System

- The lithium bromide absorption refrigeration system uses a solution of lithium bromide in water. In this system, the water is being used as a refrigerant whereas lithium bromide, which is a highly hydroscopic salt, as an absorbent. The lithium bromide solution has a strong affinity for water vapour because of its very low vapour pressure. Since lithium bromide solution is corrosive, therefore, inhibitors should be added in order to protect the metal parts of the system against corrosion. Lithium chromate is often used as a corrosion inhibitor. This system is very popular for air-conditioning in which low refrigeration temperatures (not below 0°C) are required.

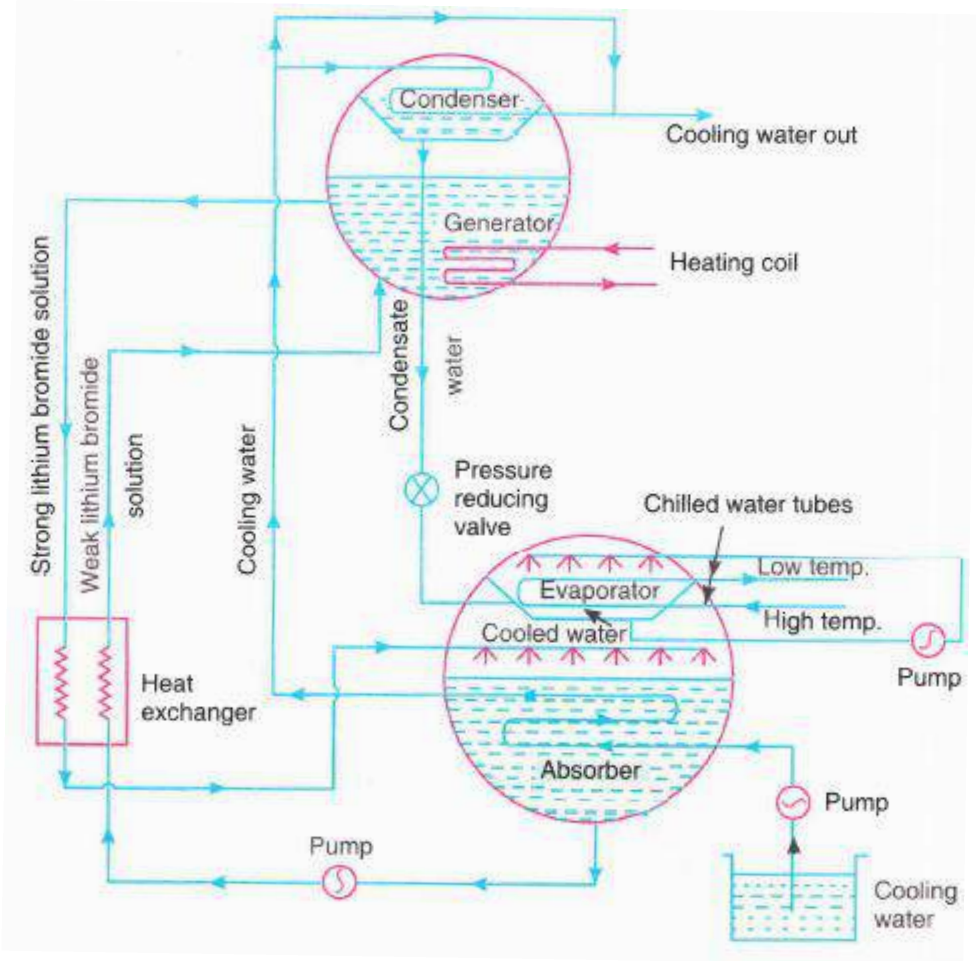


Fig. 5.4 Lithium Bromide absorption refrigeration system

- Fig. 5.4 shows a lithium bromide vapour absorption system. In this system, the absorber and the evaporator are placed in one shell which operates at the same low pressure of the system. The generator and condenser are placed in another shell which operates at the same high pressure of the system. The principle of operation of this system is discussed below:
- The water for air-conditioning coils or process requirements is chilled as it is pumped through the chilled water tubes in the evaporator by giving up heat to the refrigerant water sprayed over the tubes. Since the pressure inside the evaporator is maintained very low, therefore, the refrigerant water evaporates. The water vapour thus formed

will be absorbed by the strong lithium bromide solution which is sprayed in the absorber. In absorbing the water vapour, the lithium bromide solution helps in maintaining very low pressure (high vacuum) needed in the evaporator, and the solution

– The water for air-conditioning coils or process requirements is chilled as it is pumped through the chilled water tubes in the evaporator by giving up heat to the refrigerant water sprayed over the tubes. Since the pressure inside the evaporator is maintained very low, therefore, the refrigerant water evaporates. The water vapour thus formed will be absorbed by the strong lithium bromide solution which is sprayed in the absorber. In absorbing the water vapour, the lithium bromide solution helps in maintaining very low pressure (high vacuum) needed in the evaporator, and the solution becomes weak. This weak solution is pumped by a pump to the generator where it is heated up by using steam or hot water in the heating coils. A portion of water is evaporated by the heat and the solution now becomes more strong. This strong solution is passed through the heat exchanger and then sprayed in the absorber as discussed above. The weak solution of lithium bromide from the absorber to the generator is also passed through the heat exchanger. This weak solution gets heat from the strong solution in the heat exchanger, thus reducing the quantity of steam required to heat the weak solution in the generator.
– The refrigerant water vapour formed in the generator due to heating of solution are passed to the condenser where they are cooled and condensed by the cooling water flowing through the condenser water tubes. The cooling water for condensing is pumped from the cooling water pond or tower. This cooling water first enters the absorber where it takes away the heat of condensation and dilution. The condensate from the condenser is supplied to the evaporator to compensate the water vapour formed in the evaporator. The pressure reducing valve reduces the pressure of condensate from the condenser pressure to the evaporator pressure. The cooled water from the evaporator is pumped and sprayed in the evaporator in order to cool the water for air-conditioning flowing through the chilled tubes. This completes the cycle.

6

Refrigeration System Components

Course Contents

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6.2 Working of Condenser

6.3 Factors affecting the condenser capacity

6.4 Classification of Condensers

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- 6.4.3 Evaporative condensers

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6.7 Factors affecting the evaporator capacity

6.8 Classification of Evaporators

6.9

6.1 Introduction

- The condenser is an important device used in the high pressure side of a refrigeration system. Its function is to remove heat of the hot vapour refrigerant discharged from the compressor. The hot vapour refrigerant consists of the heat absorbed by the evaporator and the heat of compression added by the mechanical energy of the compressor motor. The heat from the hot vapour refrigerant in a condenser is removed first by transferring it to the walls of the condenser tubes and then from the tubes to the condensing or cooling medium. The cooling medium may be air or water or a combination of the two.
- The selection of a condenser depends upon the capacity of the refrigerating system, the type refrigerant used and the type of cooling medium available.

6.2 Working of a Condenser

- The working of a condenser may be best understood by considering a simple refrigerating system and the corresponding p-h diagram showing three stages of a refrigerant cooling is shown in Fig. 6.1 (a) and (b).

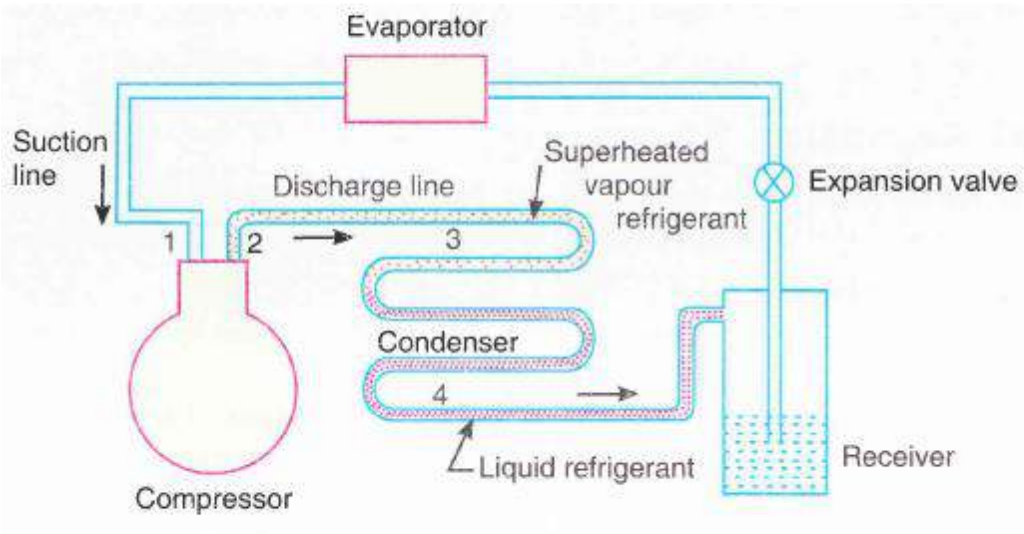


Fig. 6.1 (a) Schematic diagram of simple refrigeration system

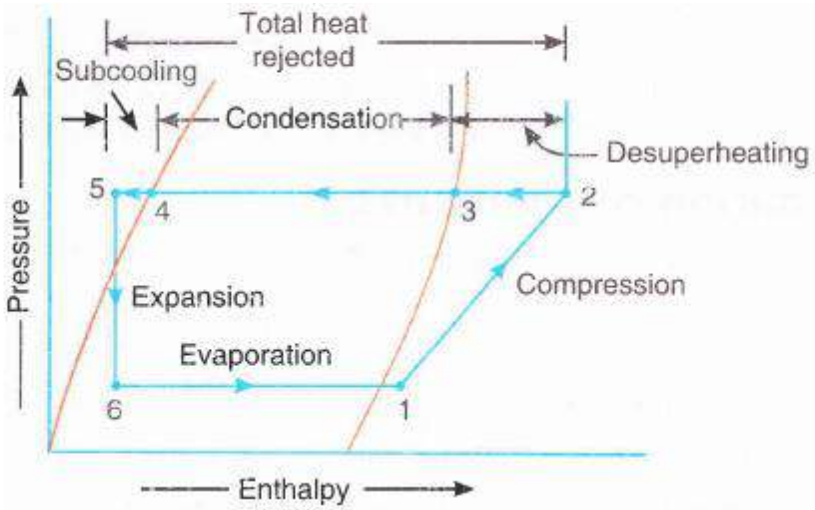


Fig. 6.1 (b) p-h diagram

The compressor draws in the superheated vapour refrigerant that contains the heat it absorbed in the evaporator. The compressor adds more heat (i.e. heat of compression) to the superheated vapour. This highly superheated vapour from the compressor is pumped to the condenser through the discharge line. The condenser cools the refrigerant in the following three stages:

1. First of all, the superheated vapour is cooled to saturation temperature (called desuperheating) corresponding to the pressure of the refrigerant. This is shown by the line 2-3 in 6.1 (b). The desuperheating occurs in the discharge line and in the first few coils of the condenser.
2. Now the saturated vapour refrigerant gives up its latent heat and is condensed to a saturated liquid refrigerant. This process, called condensation, is shown by the line 3-4.
3. The temperature of the liquid refrigerant is reduced below its saturation temperature (i.e. sub-cooled) in order to increase the refrigeration effect. This process is shown by the line 4-5.

6.3 Factors Affecting the Condenser Capacity

– The condenser capacity is the ability of the condenser to transfer heat from the hot refrigerant to the condensing medium. The heat transfer capacity of a condenser depends upon following factors:

1. Material. Since the different materials have different abilities of heat transfer, therefore the size of a condenser of a given capacity can be varied by selecting the right material. It may noted that higher the ability of a material to transfer heat, the smaller will be the size of condenser

2. Amount of contact. The condenser capacity may be varied by controlling the amount contact between the condenser surface and the condensing medium. This can be done by varying the surface area of the condenser and the rate of flow of the condensing medium over condenser surface. The amount of liquid refrigerant level in the condenser also affects the amount of contact between the vapour refrigerant and the condensing medium. The portion of condenser used for liquid sub-cooling cannot condense any vapour refrigerant.

3. Temperature difference. The heat transfer capacity of a condenser greatly depends up the temperature difference between the condensing medium and the vapour refrigerant. As temperature difference increases, the heat transfer rate increases and therefore the condenser capacity increases. Generally, this temperature difference cannot be controlled. But when temperature difference becomes so great that it becomes a problem, devices are available that v change the amount of condensing surface and the air flow rate to control condenser capacity.

6.4 Classification of Condensers

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

1. Air-cooled condensers,
2. Water-cooled condensers, and

3. Evaporative condensers. These condensers are discussed, in detail, in the following pages.

6.4.1 Air-Cooled Condensers

- An air-cooled condenser is one in which the removal of heat is done by air. It consists of or copper tubing through which the refrigerant flows. The size of tube usually ranges from to 18 mm outside diameter, depending upon the size of condenser. Generally copper tubes used because of its excellent heat transfer ability. The condensers with steel tubes are used in ammonia refrigerating systems. The tubes are usually provided with plate type fins to increase the surface area for heat transfer as shown in Fig. 6.2. The fins are usually made from aluminum because of its light weight. The fins spacing is quite wide to reduce dust clogging.

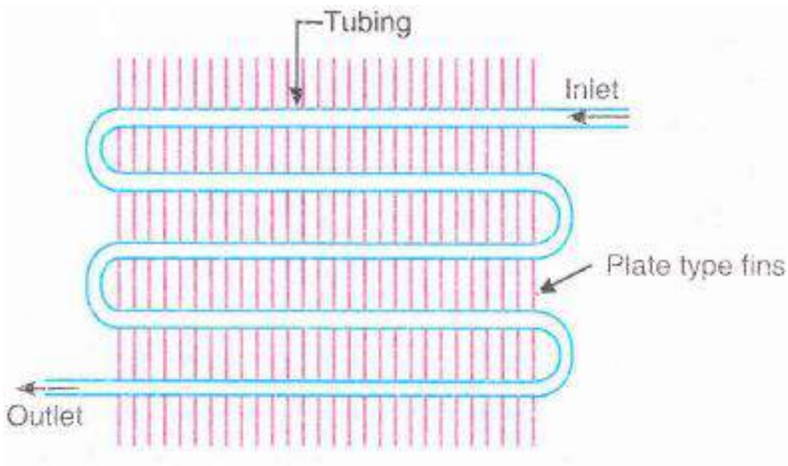


Fig. 6.2 Air-cooled condensers

- The condenser with single row of tubing provides the most efficient heat transfer. This is because the air temperature rises as it passes through each row of tubing. The temperature difference between the air and the vapour refrigerant decreases in each row of tubing and therefore each row becomes less effective.
- The air-cooled condensers may have two or more rows of tubing, but the condensers with up to six rows of tubing are common. Some condensers have seven or eight rows. However more than eight row of tubing are usually not efficient. This is because air temperature will be too close to the condenser temperature to absorb any more heat after passing through eight rows of tubing.

➢ **Types of Air-Cooled Condensers**

Following are the two types of air-cooled condensers:

1. Natural convection air-cooled condensers.

- In natural convection air-cooled condenser the heat transfer from the condenser coils to the air is by natural convection. As the air comes in contact with the warm condenser tubes, it absorbs heat from the refrigerant and thus temperature of air increases. The warm air, being lighter, rises up and the cold air from below rises to take away the heat from the condenser. This cycle continues in natural convection air-cook condensers. Since the rate of heat transfer in natural convection condenser is slower, therefore they require a larger surface area as compared to forced convection condensers. The natural convection air-cooled condensers are used only in small-capacity applications such as domestic refrigerators freezers, water coolers and room air-conditioners.

2. Forced convection air-cooled condensers.

– In forced convection air-cooled condensers the fan (either propeller or centrifugal) is used to force the air over the condenser coils to increase its heat transfer capacity. The forced convection condensers may be divided into the following two groups:
  (a) The base mounted air-cooled condensers and (b) Remote air cooled condensers
– The base mounted air-cooled condensers, using propeller fans are mounted on the same base of compressor, motor, receiver and other controls. The entire arrangement is called a condensing unit. In small units, the compressor is belt-driven from the motor and the fan required to force the air through the condenser is mounted on the shaft of the motor. The use of this type of compressor for indoor units is limited up to 3 kW capacity motor only. These condensing units are used on packaged refrigeration systems of 10 tonnes or less.
– Remote air cooled condensers, the remote air-cooled condensers are used on systems above 10 tonnes and are available up to 125 tonnes. The systems above 125 tonnes usually have two or more condensers. These condensers may be horizontal or vertical. They can be located either outside or inside the building.
– The remote condensers located outside the building can be mounted on a foundation on the ground, on the roof or on the side of a building away from the walls. These condensers usually use propeller fans because they have low resistance to air flow and free air discharge. They require 18 to 36 m3/min of air per tonne of capacity. The propeller fans can move this volume of air as long as the resistance to air flow is low. To prevent any resistance to air flow, the fan intake on vertical outdoor condensers usually faces the prevailing winds. If this is not possible, the air discharge side is usually covered with a shield to deflect opposing winds.
– The remote condensers located inside the building usually require duct work to carry air to and from the unit. The duct work restricts air flow to and from the condenser and causes high air pressure drop. Therefore, inside condensers usually use centrifugal fans which can move the necessary volume of air against the resistance to air flow.

6.4.2 Water-Cooled Condensers

– A water-cooled condenser is one in which water is used as the condensing medium. They are always preferred where an adequate supply of clear inexpensive water and means of water disposal are available. These condensers are commonly used in commercial and industrial refrigerating units. The water-cooled condensers may use either of the following two water systems:

1. Waste water system or 2. Recalculated water system

1. Waste water system

– In waste water system the water after circulating in the condenser is discharged to a sewer, as shown in Fig. 6.3. This system is used on small units and in locations where large cities of fresh inexpensive water and a sewer system large enough to handle the waste water me available. The most common source of fresh water supply is the city main.

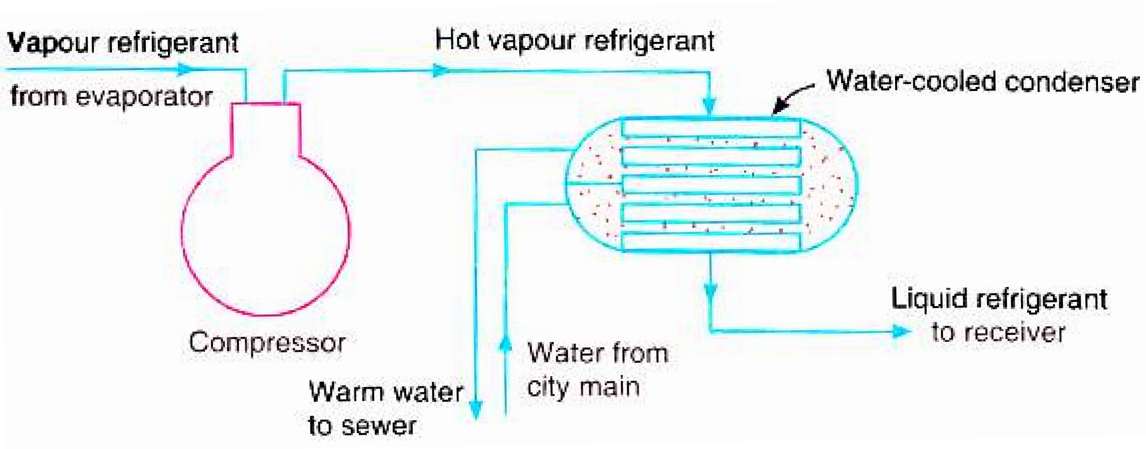


Fig. 6.3 Water-cooled condenser using waste water system

2. Recirculated water system

– As shown in Fig. 6.4, the same water circulating in the denser is cooled and used again and again. Thus this system requires some type of watering device. The cooling water towers and spray ponds are the most common cooling devices used in a recirculated water system. The warm water from the condenser is led to the cooling tower where it is cooled by self-evaporation into a stream of air. The water pumps are used to circulate the water through the system and then to the cooling tower which is usually located on the roof. Once a recirculated water system is filled with water, the only additional water required is make-up water. The make-up water simply replaces the water that evaporates from the cooling tower or spray pond.

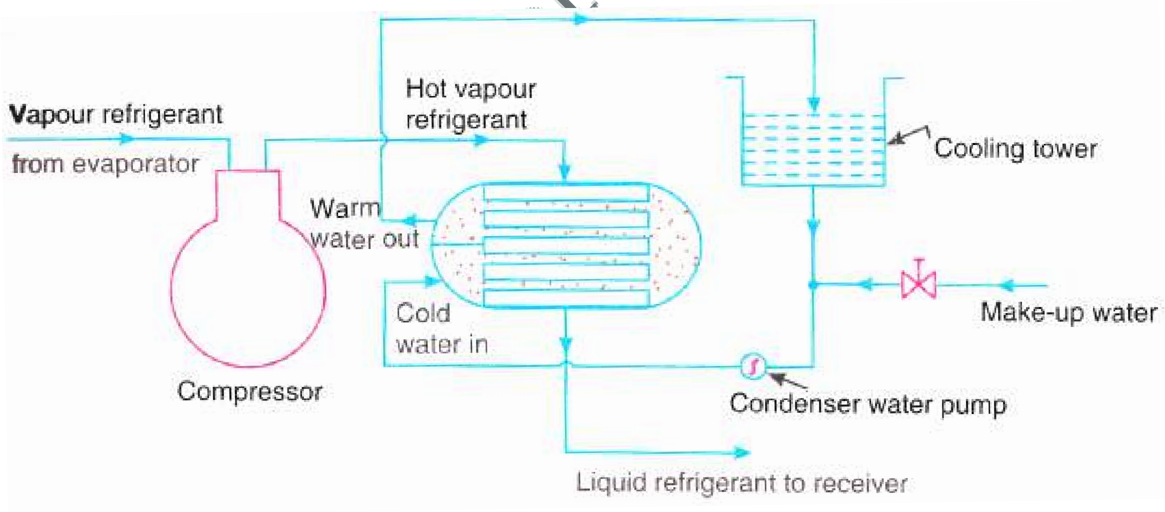


Fig. 6.4 Water-cooled condenser recirculated water system

➢ **Types of Water-Cooled Condensers**

The water-cooled condensers are classified, according to their construction, into the following three groups:

1. Tube-in-tube or double-tube condensers

– The tube-in-tube or double-tube condenser, as shown in Fig. 6.5, consists of a water tube inside a large refrigerant tube.

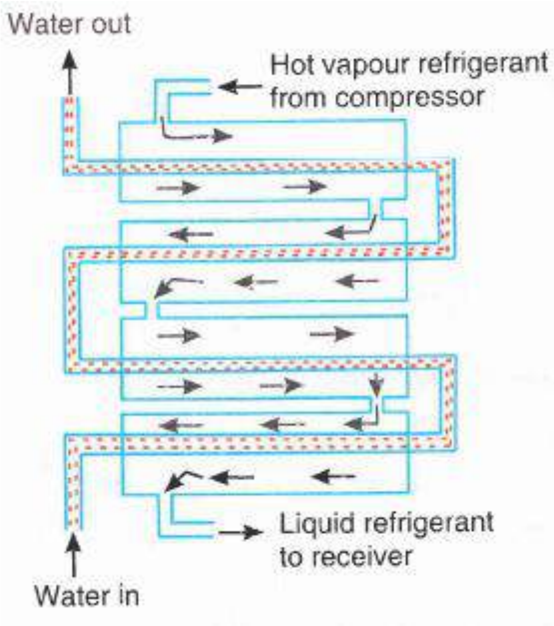


Fig. 6.5 Tube-in-tube or double-tube condenser

– In this type of condenser, the hot vapour refrigerant enters at the top of the condenser. The water absorbs the from the refrigerant and the condensed liquid refrigerant flows at the bottom. Since the refrigerant tubes are exposed to ambient air, therefore some of the heat is also absorbed by ambient air by natural convection.

– The cold water in the inner tubes may flow in either direction. When the water enters a: bottom and flows in the direction opposite to the refrigerant, it is said to be a counter-flow system. On the other hand, when the water enters at the top and flows in the same direction ac refrigerant, it is said to be a parallel-flow system.

– The counter-flow system, as shown in Fig. 6.5 is preferred in all types of water-coo' condensers because it gives high rate of heat transfer. Since the coldest water is used for final cooling of the liquid refrigerant and the warmest water absorbs heat from the hottest vapour refrigerant, therefore the temperature difference between the water and refrigerant remains fair constant throughout the condenser. In case of parallel-flow system, as the water and refrigerant flow in the same direction, therefore the temperature difference between them increases. Thus ability of water to absorb heat decreases at it passes through the condenser.

2. Shell and coil condensers

– A shell and coil condenser, as shown in Fig. 6.6, consists of one or more water coils enclosed in a welded steel shell. Both the finned and bare coil types are available.

– The shell and coil condenser may be either vertical (as shown in the figure) or horizontal. In this type of condenser, the hot vapour refrigerant enters at the top of the shell and surrounds the water coils. As the vapour condenses, it drops to the bottom of the shell which often serves as a receiver. Most vertical type shell and coil condensers use counter flow water system as it is more efficient then parallel flow water system. In the shell and coil condensers, coiled tubing is free to expand and with temperature changes

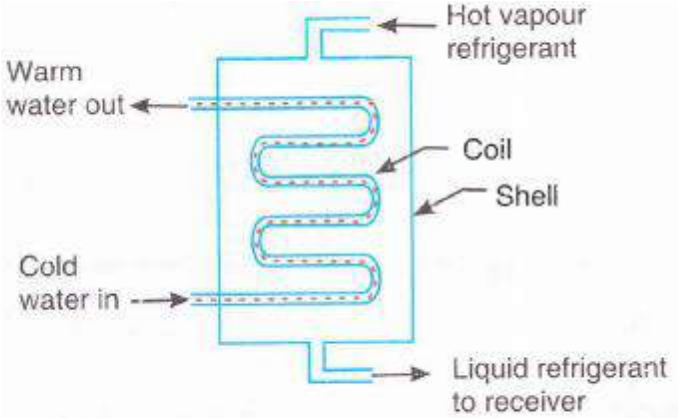


Fig. 6.6 Shell and coil condenser

Because of its spring action and can withstand any strain caused temperature changes. Since the water coils are enclosed in a welded steel shell, therefore the cleaning of these coils is not possible. The coils are cleaned with chemicals. The shell and coil condensers are used for units up to 50 tonnes capacity.

3. Shell and tube condensers

– The shell and tube condenser, as shown in Fig. 6.7 consists cylindrical steel shell containing a number of straight water tubes. The tubes are expanded into the tube sheet holes to form a vapour-tight fit. The tube sheets are welded to the shell at both the ends. The removable water boxes are bolted to the tube sheet at each end to facilitate cleaning of the condenser. The intermediate supports are provided in the shell to avoid sagging of the tubes.

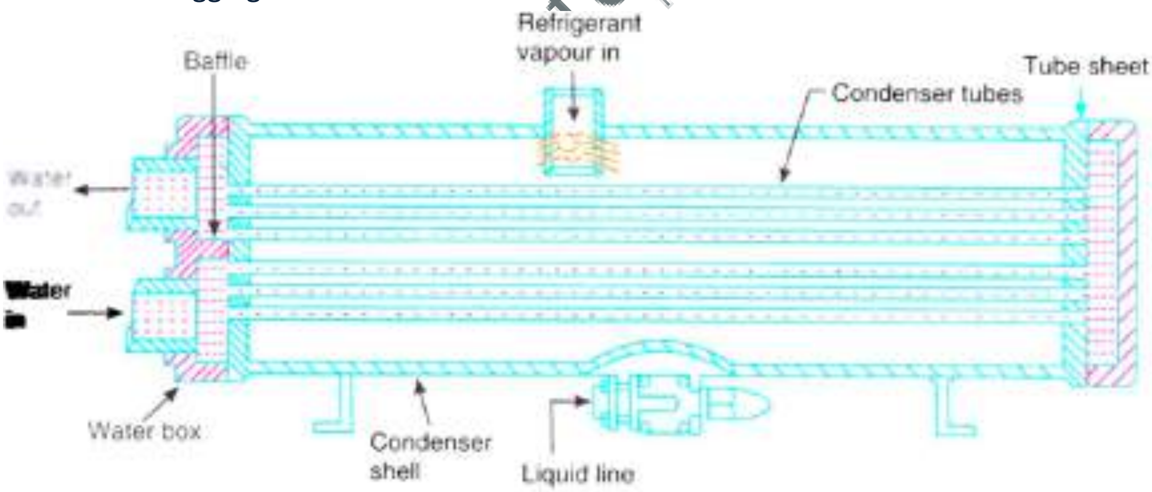


Fig. 6.7 Shell and tube condenser

– The condenser tubes are made either from steel or copper, with or without fins. The steel tubes without fins are usually used in ammonia refrigerating systems because ammonia corrodes copper tubing.

– In this type of condenser, the hot vapour refrigerant enters at the top of the shell and condenses as it comes in contact with water tubes. The condensed liquid refrigerant drops to the bottom of the shell which often serves as a receiver. However, if the maximum storage capacity for the liquid refrigerant is less than the total charge of the system, then a receiver of adequate capacity is to be added in case the pump down facility is to be provided as in ice plants, cold storages etc. In some condensers, extra rows of water tubes are provided at the lower end of the condenser for sub-cooling of the liquid refrigerant below the condensing temperature.

6.4.3 Evaporative condenser

- In evaporative condensers, both air and water are used to extract heat from the condensing refrigerant. Figure 6.8 shows the schematic of an evaporative condenser. Evaporative condensers combine the features of a cooling tower and water-cooled condenser in a single unit. In these condensers the water is sprayed from top part on a bank of tubes carrying the refrigerant and air is induced upwards. There is a thin water film around the condenser tubes from which evaporative cooling takes place. The heat transfer coefficient for evaporative cooling is very large. Hence, the refrigeration system can be operated at low condensing temperatures (about 11 to 13 K above the wet bulb temperature of air). The water spray countercurrent to the airflow acts as cooling tower. The role of air is primarily to increase the rate of evaporation of water. The required air flow rates are in the range of 350 to 500 m^3/h per TR of refrigeration capacity.

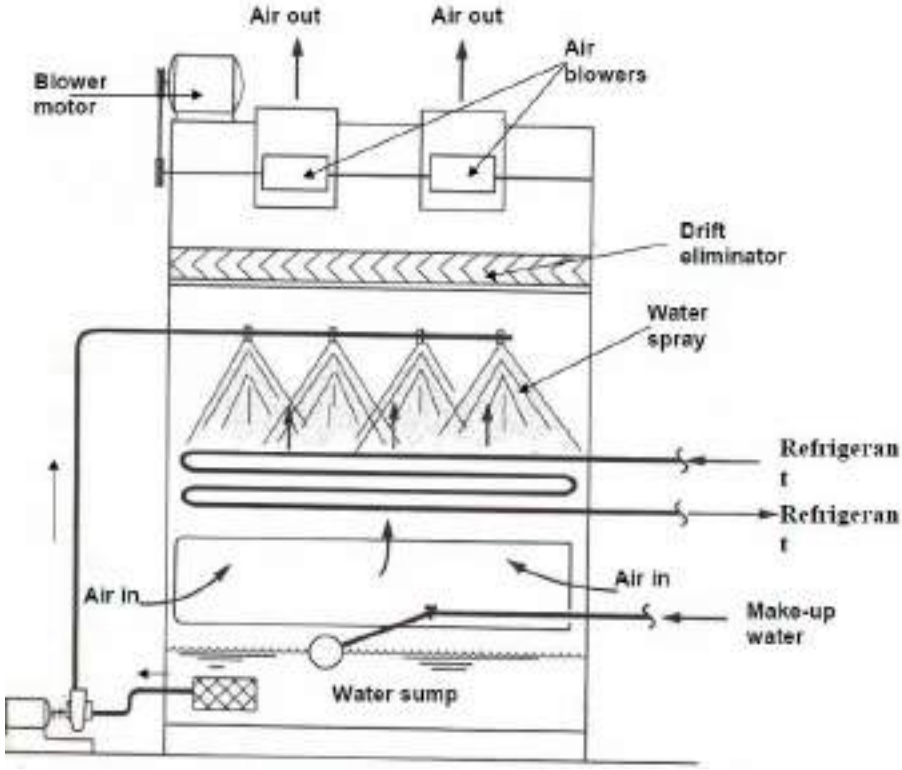


Fig. 6.8 Evaporative condenser

- Evaporative condensers are used in medium to large capacity systems. These are normally cheaper compared to water cooled condensers, which require a separate cooling tower. Evaporative condensers are used in places where water is scarce. Since water is used in a closed loop, only a small part of the water evaporates. Make-up water is supplied to take care of the evaporative loss. The water consumption is typically very low, about 5 percent of an equivalent water cooled condenser with a cooling tower. However, since condenser has to be kept outside, this type of condenser requires a longer length of refrigerant tubing, which calls for larger refrigerant inventory and higher pressure drops. Since the condenser is kept outside, to prevent the water from freezing, when outside temperatures are very low, a heater is placed in the water tank. When outside temperatures are very low it is possible to switch-off the water pump and run only the blowers, so that the condenser acts as an air cooled condenser.

– Another simple form of condenser used normally in older type cold storages is called as atmospheric condenser. The principle of the atmospheric condenser is similar to evaporative condenser, with a difference that the air flow over the condenser takes place by natural means as no fans or blowers are used. A spray system sprays water over condenser tubes. Heat transfer outside the tubes takes by both sensible cooling and evaporation, as a result the external heat transfer coefficient is relatively large. The condenser pipes are normally large, and they can be either horizontal or vertical. Though these condensers are effective and economical they are being replaced with other types of condensers due to the problems such as algae formation on condenser tubes, uncertainty due to external air circulation etc.

6.5 Comparison between air cooled and water cooled condenser

The Salient features of air cooled and water cooled condensers are shown below in Table 6.1. The advantages and disadvantages of each type are discussed below.

Sr No.	Air cooled condenser	Water cooled condenser
1.	Very simple construction, thus less initial and maintained cost.	Complicated construction, thus high initial and maintained cost.
2.	No handling problem.	Difficult to handle.
3.	Not required piping arrangement for carrying air.	Pipes are required to take water to and from the condenser.
4.	No problem in disposing of used air.	Problem of in disposing of used water unless a recirculation system is provided.
5.	No corrosion, less fouling effect	Corrosion occurs inside the tubes, high fouling effect
6.	Low heat transfer capacity due to low thermal conductivity of air	High heat transfer capacity due to high thermal conductivity of water
7.	Used for less capacity plants	Used for large capacity plants
8.	Power required to drive fan is excessive, thus objectionable noise.	No fan noise
9.	Not uniform distribution of air on condenser surface	Even distribution of water on condenser surface
10.	High flexibility	Low flexibility

6.6 Working of evaporator

- The evaporator is an important device used in the low pressure side of a refrigeration system. The liquid refrigerant from the expansion valve enters into the evaporator where it boils and changes into vapour. The function of an evaporator is to absorb heat from the surrounding location or medium which is to be cooled by means of a refrigerant. The temperature of the boiling refrigerant in the evaporator must always be less than that of the surrounding medium so that the heat flows to the refrigerant. The evaporator becomes cold and remains cold due to the following two reasons:
    1. The temperature of the evaporator coil is low due to the low temperature of the refrigerant inside the coil.
    2. The low temperature of the refrigerant remains unchanged because any heat it absorbs is converted to latent heat as boiling proceeds
- The liquid refrigerant at low pressure enters the evaporator at point 6, as shown in Fig. 6.1 (b). As the liquid refrigerant passes through the evaporator coil, it continually absorbs heat through coil walls, from the medium being cooled. During this, the refrigerant continues to boil and evaporate and the entire liquid refrigerant has evaporated and only vapours refrigerant remains in the evaporator coil. The liquid refrigerant's ability to convert absorbed heat to latent heat is now used up.
- Since the vapour refrigerant at point is still colder than the medium being cooled therefore the vapour refrigerant continues to absorb heat. This heat absorption causes an increase in the sensible heat (or temperature) of the vapour refrigerant. The vapour temperature continues to rise until the vapour leaves the evaporator to the suction line at point 1. At this point, the temperature of the vapour is above the saturation temperature and the vapour refrigerant superheated.

6.7 Factors Affecting the Heat Transfer Capacity of an Evaporator

- Though there are many factors upon which the heat transfer capacity of an evaporator depends are given below.

1. Material. In order to have rapid heat transfer in an evaporator, the material used for the construction of an evaporator coil should be a good conductor of heat. The material which is not affected by the refrigerant must also be selected. Since metals are best conductors of heat, they are always used for evaporators. Iron and steel can be used with all common Refrigerants. Brass and copper are used with all refrigerants except ammonia. Aluminium should not be used with freon.

2. Temperature difference. The temperature difference between the refrigerant within the evaporator and the product to be cooled plays an important role in the heat transfer capacity of an evaporator. The following table shows the suggested temperature difference for some of the products to be cooled. It may be noted that a too low temperature difference (below 8°C) may cause slime on certain products such as meat or poultry. On the other hand, too high a temperature difference causes excessive dehydration.

3. Velocity of refrigerant. The velocity of refrigerant also affects the heat transfer capacity of an evaporator. If the velocity of refrigerant flowing through the evaporator increases, the overall heat transfer coefficient also increases. But this increased velocity will cause greater pressure loss in the evaporator. Thus the only recommended velocities for different refrigerants which give high heat transfer rates and allowable pressure loss should be used.

4. Thickness of the evaporator coil wall. The thickness of the evaporator coil wall affects the heat transfer capacity of the evaporator. In general, the thicker the wall, the slower the rate of heat transfer. Since the refrigerant in the evaporator coils is under pressure, therefore the evaporator walls must be thick enough to withstand the effects of that pressure. It may be note that the thickness has only a slight effect on total heat transfer capacity because the evaporators usually made from highly conductive materials.

5. Contact surface area. An important factor affecting the evaporator capacity is the contact surface available between the walls of evaporator coil and the medium being cooled. The amount of contact surface, in turn, depends basically on the physical size and shape of the evaporator coil.

6.8 Classification of Evaporator

– There are several ways of classifying the evaporators as below:

1. According to the type of construction
   - (a) Bare tube coil evaporator
   - (b) Finned tube evaporator
   - (c) Plate evaporator
   - (d) Shell and tube evaporator
   - (e) Shell and coil evaporator
   - (f) Tube in tube evaporator
2. According to the manner in which liquid refrigerant is fed
   - (a) Flooded evaporator
   - (b) Dry expansion evaporator
3. According to mode of heat transfer
   - (a) Natural convection evaporator
   - (b) Forced convection evaporator
4. According to operating conditions
   - (a) Frosting evaporator
   - (b) Non frosting evaporator
   - (c) Defrosting evaporator

6.4.4 Bare tube evaporator

– The simplest type of evaporator is the bare tube coil evaporator, as shown in Fig. 6.9. The bare tube coil evaporators are also known as prime surface evaporators. This type of evaporator offers relatively little surface area as compared to other types of coils. The amount of surface area may be increased by extending the length of tube. But there are disadvantages of excessive tube length. The effective length of the tube is limited by the capacity of expansion valve. If the tube is too long for the valve's capacity, the, liquid refrigerant will tend to completely vaporize early in the progress through the tube, thus leading to excessive superheating at the outlet. The long tubes will also cause considerably greater pressure drop between the inlet and outlet of evaporator. This results in a reduced suction line pressure.

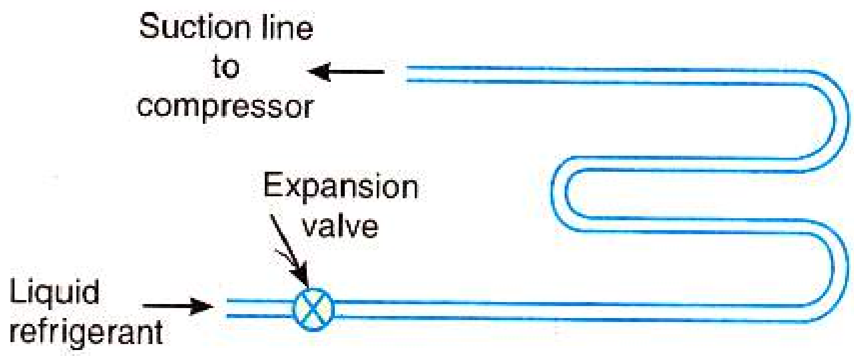


Fig. 6.9 Bare tube coil evaporative

– The diameter of the tube in relation to tube length may also be critical. If the tube diameter is too large, the refrigerant velocity will be too low and the volume of refrigerant will be too great in relation to the surface area of the tube to allow complete vaporization. This, in turn may allow liquid refrigerant to enter the suction line with possible damage to the compressor. On the other hand, if the diameter is too small, the pressure drop due to friction may be too high and will reduce the system efficiency.
– The bare tube coil evaporators may be used for any type of refrigeration requirement. Its use is however, limited to applications where the box temperatures are under 0°C and in liquid cooling, because the accumulation of ice or frost on these evaporators has less effect on the heat transfer than on those equipped with fins. The bare tube coil evaporators are also extensively used household refrigerators because they are easier to keep clean.

6.4.5 Finned evaporator

– The finned evaporator, as shown in Fig. 6.10 consists of bare tubes or coils over which the plates or fins are fastened.
– The metal fins are constructed of thin sheets of metal having good thermal conductivity. The shape, size or spacing of the fins can be adapted to provide best rate of heat transfer for a given application. Since the fins greatly increase the contact surfaces for heat transfer, therefore finned evaporators are also called extended surface evaporators.

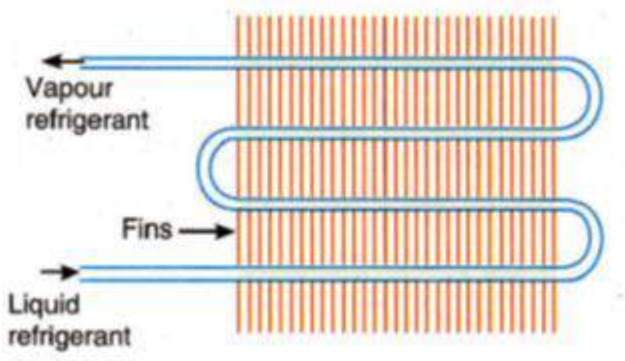


Fig. 6.10 Finned evaporator

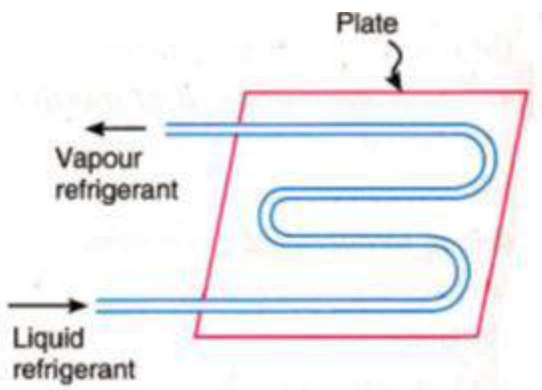


Fig. 6.11 Plate evaporator

- The finned evaporators are primarily designed for air conditioning applications where the refrigerator temperature is above 0^oC. Because of the rapid heat transfer of the finned evaporator, it will defrost itself on the off cycle when the temperature of the coil is near 0^oC. A finned coil should never be never allowed to frost because the accumulation of frost between the fins reduces the capacity. The air conditioning coils, which operate at suction temperatures which are high enough so that frosting never occurs, have fin spacing as small as 3 mm. The finned coils which frost on the on cycle and defrost on the off cycle have wider fin spacing.

6.4.6 Plate evaporator

- A common type of plate evaporator is shown in Fig. 6.11 is consist of number of horizontal tubes enclosed in a cylindrical shell. The inlet and outlet headers with are either welded on one side of a plate or between the two plates which are welded together at the edges. The plate evaporators are generally used in household refrigerators, home freezers beverage coolers, ice cream cabinets, locker plants etc.

6.4.7 Shell and Tube Evaporators

- The shell and tube evaporator, as shown in Fig. 6.12 is consists of a number of horizontal tubes enclosed in a cylindrical shell. The inlet and outlet headers with perforated metal tube sheets are connected at each end of the tubes.

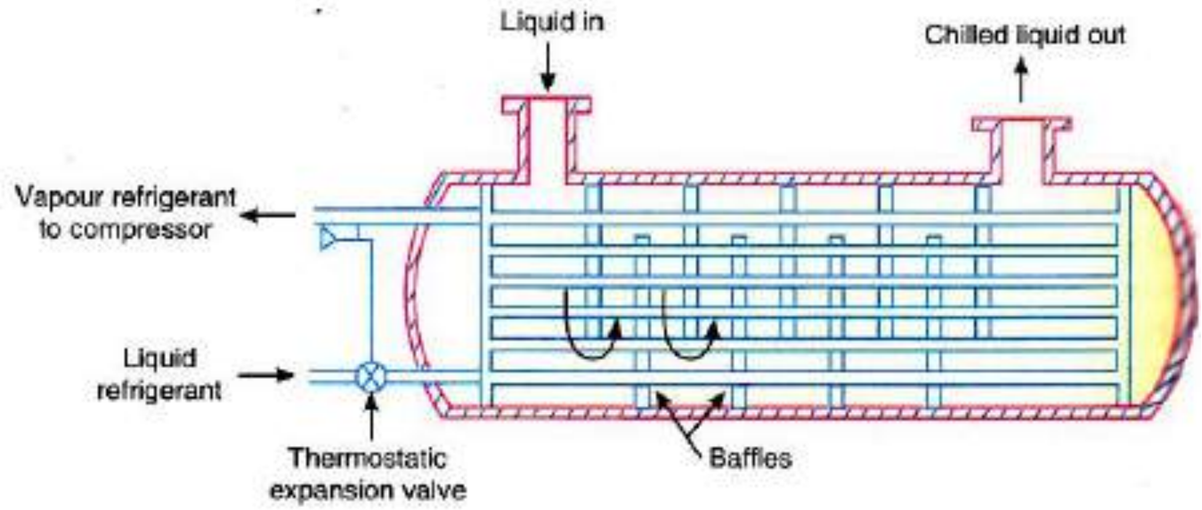
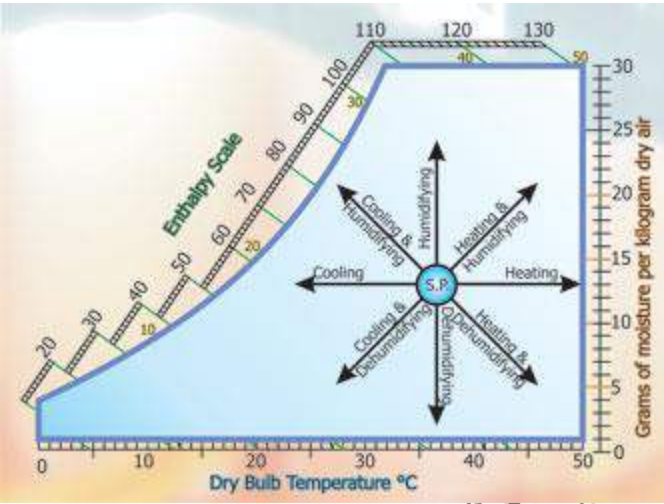


Fig. 6.12 Shell and Tube Evaporators

7

PSYCHROMETRY & PSYCHROMETRIC TERMS



Course Contents

7.1Air-Conditioning

Air-conditioning is a process that simultaneously conditions air; distributes it combined with the outdoor air to the conditioned space; and at the same time controls and maintains the required space's temperature, humidity, air movement, air cleanliness, sound level, and pressure differential within predetermined limits for the health and comfort of the occupants, for product processing, or both.

7.2 Air-Conditioning Systems

An air-conditioning or HVAC&R system consists of components and equipment arranged in sequential order to heat or cool, humidify or dehumidify, clean and purify, attenuate objectionable equipment noise, transport the conditioned outdoor air and recirculate air to the conditioned space, and control and maintain an indoor or enclosed environment at optimum energy use.

The types of buildings which the air-conditioning system serves can be classified as:

- Institutional buildings, such as hospitals and nursing homes
- Commercial buildings, such as offices, stores, and shopping centers
- Residential buildings, including single-family and multifamily low-rise buildings of three or fewer stories above grade
- Manufacturing buildings, which manufacture and store products

7.3 Psychrometrics

Psychrometrics is the study of the thermodynamic properties of moist air. It is used extensively to illustrate and analyze the characteristics of various air conditioning processes and cycles.

7.4 Psychrometric Properties & Psychrometric Relations

7.4.1 Moist Air

Above the surface of the earth is a layer of air called the **atmosphere or atmospheric air**. The lower atmosphere, or homosphere, is composed of moist air, that is, a mixture of dry air and water vapor.

Psychrometrics is the science of studying the thermodynamic properties of moist air. It is widely used to illustrate and analyze the change in properties and the thermal characteristics of the air-conditioning process and cycles.

The composition of dry air varies slightly at different geographic locations and from time to time. The approximate composition of dry air by volume is nitrogen, 79.08%; oxygen, 20.95%; argon, 0.93%; carbon dioxide, 0.03%; other gases (e.g., neon, sulfur dioxide), 0.01%. The amount of water vapor contained in the moist air within the temperature range 0 to 100 °F changes from 0.05 to 3% by mass. The variation of water vapor has a critical influence on the characteristics of moist air.

The equation of state for an ideal gas that describes the relationship between its thermodynamic properties is

$$pv = RT$$

Or

$$pV = mRT$$

Where, p = pressure of the gas

v = specific volume of the gas

R = gas constant

T = absolute temperature of the gas

V = volume of the gas

m = mass of the gas

The most exact calculation of the thermodynamic properties of moist air is based on the formulations recommended by Hyland and Wexler (1983) of the U.S. National Bureau of Standards. The psychrometric charts and tables developed by ASHRAE are calculated and plotted from these formulations. According to Nelson et al. (1986), at a temperature between 0 and 100 °F, enthalpy and specific volume calculations using ideal gas Equations show a maximum deviation of 0.5% from the results of Hyland and Wexler's exact formulations. Therefore, ideal gas equations are used in the development and calculation of psychrometric formulations in this handbook.

Although air contaminants may seriously affect human health, they have little effect on the thermodynamic properties of moist air. For thermal analysis, moist air may be treated as a binary mixture of dry air and water vapor.

Applying ***Dalton's law*** to moist air:

$$p_b = p_a + p_w$$

Dalton's law is summarized from the experimental results and is more accurate at low gas pressure. Dalton's law can also be extended, as the Gibbs-Dalton law, to describe the relationship of internal energy, enthalpy, and entropy of the gaseous constituents in a mixture.

7.4.2 Specific Humidity, Humidity Ratio (*w*)

The humidity ratio of moist air w is the ratio of the mass of water vapor m_v to the mass of dry air m_a contained in the mixture of the moist air, in kg/kg of dry air. The humidity ratio can be calculated as

$$w = \frac{m_v}{m_a}$$

Since dry air and water vapor can occupy the same volume at the same temperature, we can apply the ideal gas equation and Dalton's law for dry air and water vapor. Equation of w can be rewritten as

$p_a v_a = m_a R_a T_d$ And $p_v v_v = m_v R_v T_d$

Substituting, R_a = 0.287 kJ/kg K & R_v = 0.461 kJ/kg K

So,
$$w = \frac{m_v}{m_a} = \frac{R_a p_v}{R_v p_a} = \frac{0.287 p_v}{0.461 p_a} = 0.622 \frac{p_v}{(p_b - p_v)}$$

Where, P_v = vapour pressure of air

P_b = Barometric Pressure

7.4.3 Relative Humidity (φ)

The relative humidity is the ratio of actual mass of water vapour in a given volume of moist air to the mass of water vapour in the same volume of saturated air at the same temperature and pressure. It is denoted by φ.

Mathematically,

$$\varphi = \frac{m_v}{m_s}$$

$p_v v_v = m_v R_v T_v$ *and*

$p_s v_s = m_s R_s T_s$

According to definitions,

$v_v = v_s$

$T_v = T_s$

$R_v = R_s$ = 0.461 kJ/KgK

So,

$$\varphi = \frac{m_v}{m_s} = \frac{p_v}{p_s}$$

Thus, the relative humidity is also defined as the ratio of actual partial pressure of water vapour in a given volume of moist air to the partial pressure of water vapour in the same volume of saturated air at the same temperature.

7.4.4 Degree of saturation or Percentage Humidity (μ)

The degree of saturation is the ratio of the humidity ratio W to the humidity ratio of a saturated mixture W_s at the same temperature and pressure, i.e.,

$$\mu = \frac{W}{W_s}$$

7.4.5 Pressure of water vapour (p_v)

According to carrier's equation, the partial pressure of water vapour,

$$p_v = p_w - \frac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44t_w}$$

Where,

p_w = Saturation pressure @ WBT

p_b = barometric pressure

t_d = Dry bulb temp.(DBT)

t_w = Wet bulb temp.(WBT)

7.4.6 Dew-point temperature (*DPT*)

If unsaturated moist air is cooled at constant pressure, then the temperature at which the moisture in the air begins to condense is known as ***dew-point temperature*** *(DPT)* of air.

7.4.7 Specific volume or Vapour density or absolute humidity

The specific volume is defined as the number of cubic meters of moist air per kilogram of dry air. From perfect gas equation since the volumes occupied by the individual substances are the same, the specific volume is also equal to the number of cubic meters of dry air per kilogram of dry air, i.e.,

$$p_a v_a = m_a R_a T_d \text{ and } p_v v_v = m_v R_v T_d$$

$$v_a/m_a = \rho_a \text{ and } v_v/m_v = \rho_v$$

so,

$$\rho_a = \frac{p_a}{R_a T_d}$$

$$\rho_v = \frac{Wp_a}{R_a T_d} = \frac{W(p_b - p_v)}{R_a T_d}$$

7.4.8 Enthalpy of moist air

The enthalpy of moist air is the sum of the enthalpy of the dry air and the enthalpy of the water vapour. Enthalpy values are always based on some reference value. For moist air, the enthalpy of dry air is given a zero value at 0 ^{0}C, and for water vapour the enthalpy of saturated water is taken as zero at 0 ^{0}C.

The enthalpy of moist air is given by:

$$h=h_a + Wh_g = c_p t_d + W(h_{fg} + c_{pw} t_d)$$

$$h = 1.022\ t_d + W(h_{fgdp}+2.3t_{dp})$$

7.5 Ideal Adiabatic Saturation Process *or*

Thermodynamic Dry Bulb Temperature and Wet Bulb Temperature

If moist air at an initial temperature T_1, humidity ratio w_1, enthalpy h_1, and pressure p flows over a water surface of infinite length in a well-insulated chamber, as shown in Fig. 7.1, liquid water will evaporate into water vapor and will disperse in the air. The humidity ratio of the moist air will gradually increase until the air can absorb no more moisture.

As there is no heat transfer between this insulated chamber and the surroundings, the latent heat required for the evaporation of water will come from the sensible heat released by the moist air.

This process results in a drop in temperature of the moist air. At the end of this evaporation process, the moist air is always saturated. Such a process is called an ***ideal adiabatic saturation process***, where an adiabatic process is defined as a process without heat transfer to or from the process.

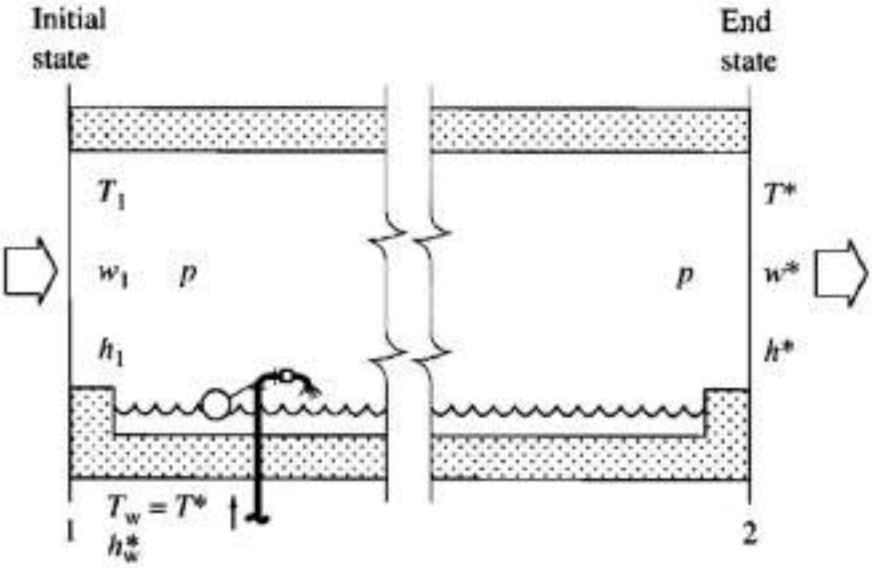


Fig.7.1 *Ideal adiabatic saturation process*

Thermodynamic Wet-Bulb Temperature

For any state of moist air, there exists a thermodynamic wet-bulb temperature *T** that exactly equals the saturated temperature of the moist air at the end of the ideal adiabatic saturation process at constant pressure. Applying a steady flow energy equation, we have

$$h_1 + (w_s^* - w_1)h_w^* = h_s^*$$

Where,

h_1, h_s^* = enthalpy of moist air at initial state and enthalpy of saturated air at end of ideal adiabatic saturation process, Btu/ lb (kJ /kg)

w_1, w_s^* = humidity ratio of moist air at initial state and humidity ratio of saturated air at end of ideal adiabatic saturation process, lb / lb (kg /kg of dry air)

h_w^* = enthalpy of water as it is added to chamber at a temperature T^*, Btu/ lb (kJ /kg)

The thermodynamic wet-bulb temperature T^*, °F (°C), is a unique property of a given moist air sample that depends only on the initial properties of the moist air—w_1, h_1 and p. It is also a fictitious property that only hypothetically exists at the end of an ideal adiabatic saturation process.

Psychrometer

A psychrometer is an instrument that permits one to determine the relative humidity of a moist air sample by measuring its dry-bulb and wet-bulb temperatures. **Fig.7.2** shows a psychrometer, which consists of two thermometers. The sensing bulb of one of the thermometers is always kept dry. The temperature reading of the dry bulb is called the ***dry-bulb temperature.*** The sensing bulb of the other thermometer is wrapped with a piece of cotton wick, one end of which dips into a cup of distilled water. The surface of this bulb is always wet; therefore, the temperature that this bulb measures is called the ***wet-bulb temperature***. The dry bulb is separated from the wet bulb by a radiation- shielding plate. Both dry and wet bulbs are cylindrical.

Wet-Bulb Temperature

A sling-type psychrometer, as shown in ***Fig.7.2***, is an instrument that determines the temperature, relative humidity, and thus the state of the moist air by measuring its dry bulb and wet bulb temperatures. It consists of two mercury-in-glass thermometers. The sensing bulb of one of them is dry and is called the ***dry bulb temperature***. Another sensing bulb is wrapped with a piece of cotton wick, one end of which dips into a water tube. This wetted sensing bulb is called the wet bulb and the temperature measured by it is called the ***wet bulb temperature*** T_w and the difference between the dry-bulb and wet-bulb temperatures is called the ***wet-bulb depression***.

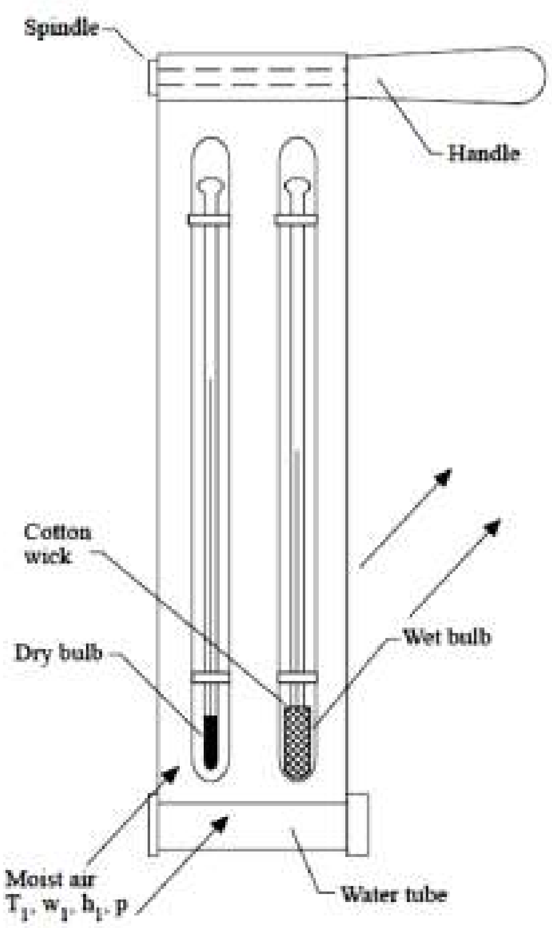


Fig. 7.2 *A sling psychrometer.*

7.6 Psychrometric Chart

A Psychrometric chart graphically represents the thermodynamic properties of moist air. Standard psychrometric charts are bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or humidity ratio (ordinate). The Left Hand Side of the psychrometric chart is bounded by the saturation line. **Fig. 7.3** shows the schematic of a psychrometric chart. Psychrometric charts are readily available for standard barometric pressure of 101.325 kPa at sea level and for normal temperatures (0-50^{0}C). ASHRAE has also developed psychrometric charts for other temperatures and barometric pressures (for low temperatures: -40 to 10^{0}C, high temperatures 10 to 120 ^{0}C and very high temperatures 100 to 120^{0}C)

Measurement of psychrometric properties

Based on Gibbs' phase rule, the thermodynamic state of moist air is uniquely fixed if the barometric pressure and two other independent properties are known. This means that at a given barometric pressure, the state of moist air can be determined by measuring any two independent properties. One of them could be the dry-bulb temperature (DBT), as the measurement of this temperature is fairly simple and accurate. The accurate measurement of other independent parameters such as humidity ratio is very difficult in practice. Since measurement of temperatures is easier, it would be convenient if the other independent parameter is also a temperature. Of course, this could be the dew-point temperature (DPT), but it is observed that accurate measurement of dew-point temperature is difficult. In this context, a new independent temperature parameter called the *wet-bulb temperature* (WBT) is defined. Compared to DPT, it is easier to measure the wet-bulb temperature of moist air. Thus knowing the dry-bulb and wet-bulb temperatures from measurements, it is possible to find the other properties of moist air.

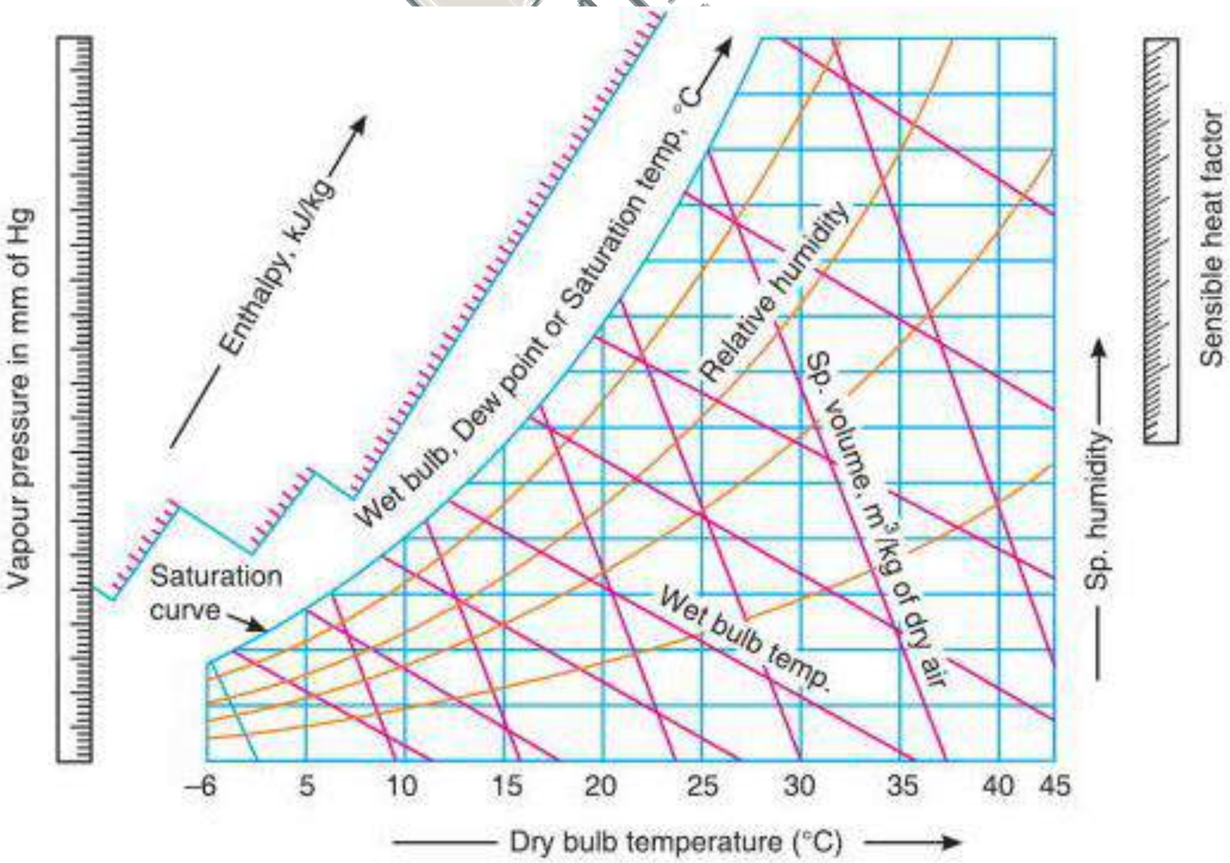


Fig. 7.3 *Psychrometric chart*

7.7 Psychrometric Processes

In the design and analysis of air conditioning plants, the fundamental requirement is to identify the various processes being performed on air. Once identified, the processes can be analyzed by applying the laws of conservation of mass and energy. All these processes can be plotted easily on a psychrometric chart. This is very useful for quick visualization and also for identifying the changes taking place in important properties such as temperature, humidity ratio, enthalpy etc. The important processes that air undergoes in a typical air conditioning plant are discussed below.

Important Psychrometric processes:

a) Sensible cooling:

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant, the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. **Fig.7.4** shows the sensible cooling process O-A on a psychrometric chart.

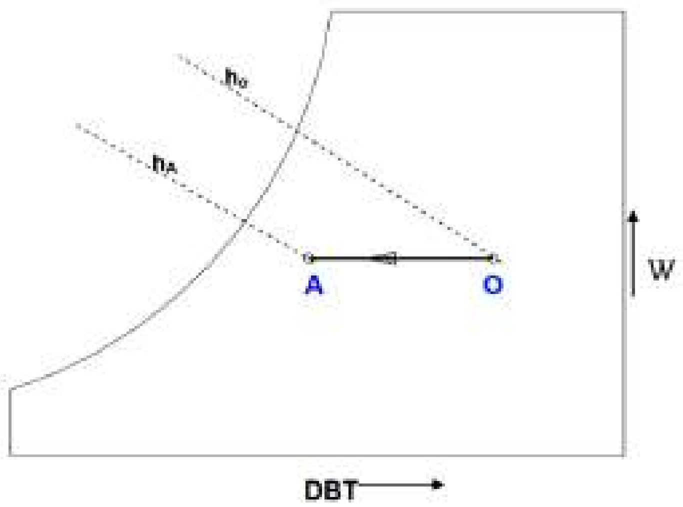


Fig.7.4 *Sensible cooling process O-A on psychrometric chart*

b) Sensible heating (Process O-B):

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil.

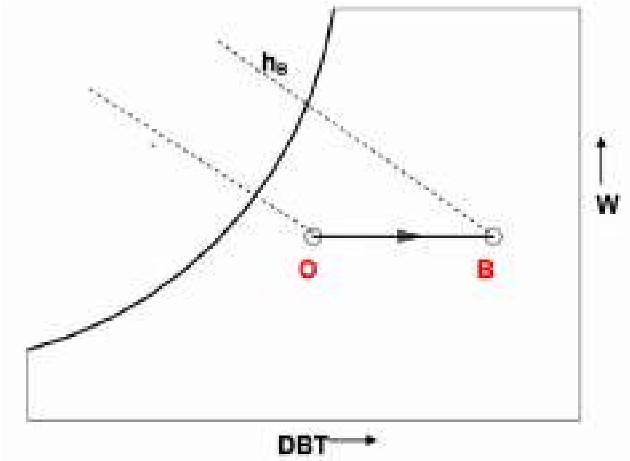


***Fig.7.5** Sensible heating process O-B on psychrometric chart*

c) Cooling and dehumidification (Process O-C):

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in **Fig.7.6**, some of the water vapor in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. This is the process air undergoes in a typical air conditioning system. Although the actual process path will vary depending upon the type of cold surface, the surface temperature, and flow conditions, for simplicity the process line is assumed to be a straight line.

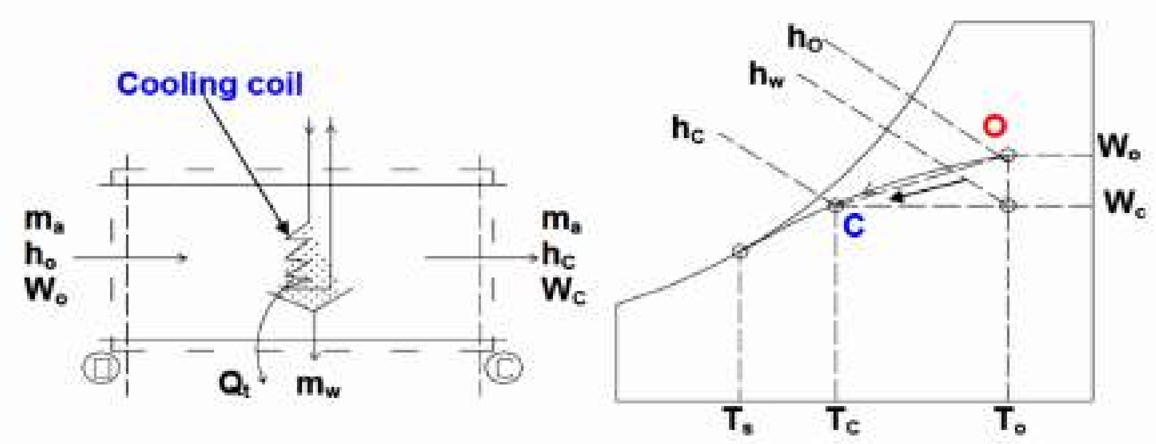


***Fig.7.6** Cooling and dehumidification process (O-C)*

The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations. It can be observed that the cooling and de-humidification process involves both latent and sensible heat transfer processes.

d) Heating and Humidification (Process O-D):

During winter it is essential to heat and humidify the room air for comfort. As shown in **Fig.7.7**, this is normally done by first sensibly heating the air and then adding water vapour to the air stream through steam nozzles as shown in the **Fig.7.7**

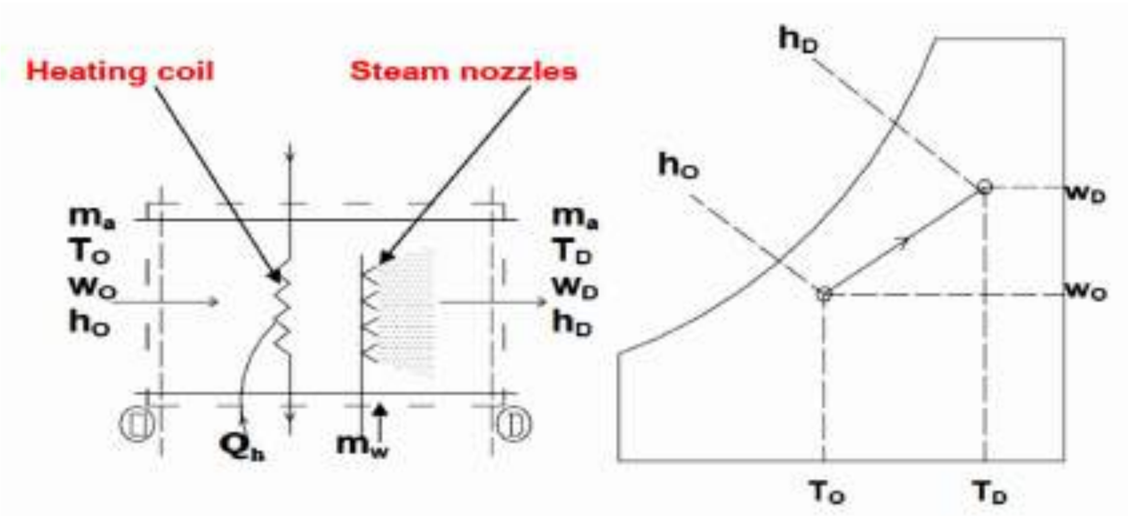


Fig.7.7*: Heating and humidification process*

Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

e) Cooling & humidification (Process O-E):

As the name implies, during this process, the air temperature drops and its humidity increases. This process is shown in **Fig.7.8**. As shown in the figure, this can be achieved by spraying cool water in the air stream.

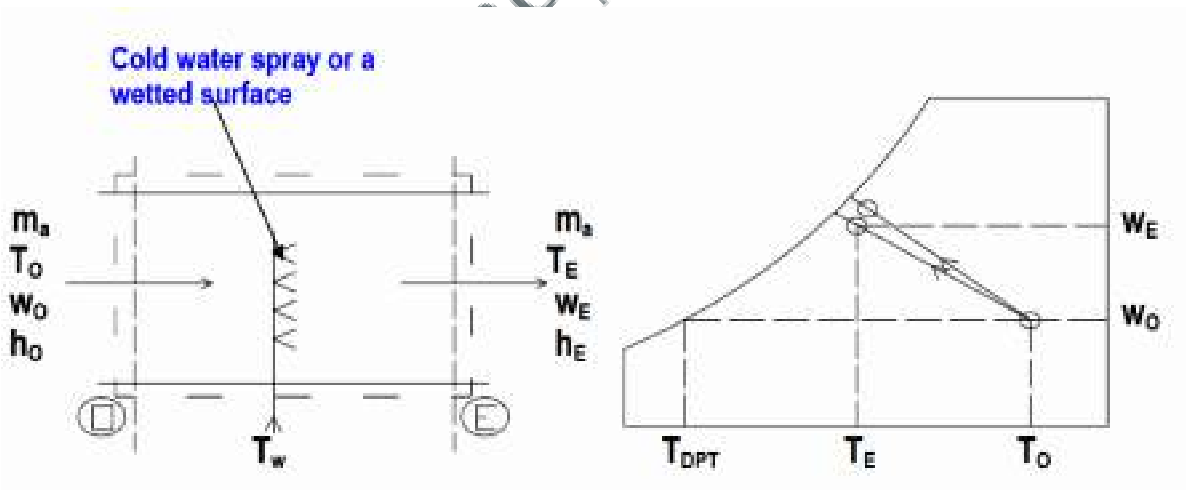


Fig.7.8 *Cooling and humidification process*

The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew-point temperature to avoid condensation ($T_{DPT} < T_w < T_O$).

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature. If the temperature of the water sprayed is equal to the wet bulb temperature of air, then the net transfer rate will be zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water. Under a special case when the spray water is entirely recirculated and is neither heated nor cooled, the system is perfectly insulated and the make-up water is supplied at WBT, then at steady-state, the air undergoes an adiabatic saturation process, during which its WBT remains constant. This is the process of adiabatic. The process of cooling and humidification is encountered in a wide variety of devices such as evaporative coolers, cooling towers etc.

f) Heating and de-humidification (Process O-F):

This process can be achieved by using a hygroscopic material, which absorbs or adsorbs the water vapor from the moisture. If this process is thermally isolated, then the enthalpy of air remains constant, as a result the temperature of air increases as its moisture content decreases as shown in **Fig.7.9.** This hygroscopic material can be a solid or a liquid. In general, the absorption of water by the hygroscopic material is an exothermic reaction, as a result heat is released during this process, which is transferred to air and the enthalpy of air increases.

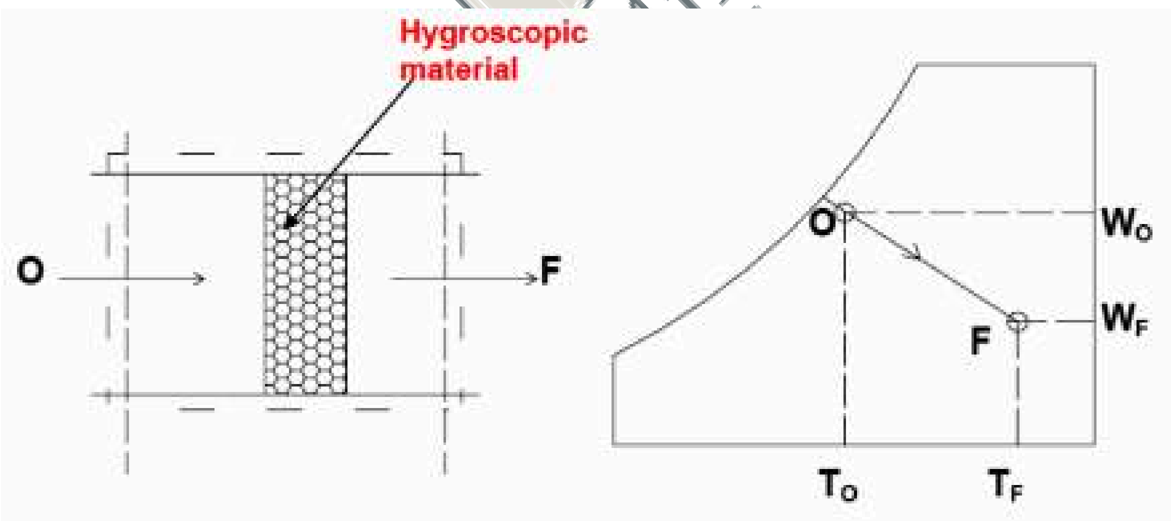


***Fig.7.9**. Chemical de-humidification process*

g) Mixing of air streams:

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture.

i) Without condensation:

Fig.7.10 shows an adiabatic mixing of two moist air streams during which no condensation of moisture takes place. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance.

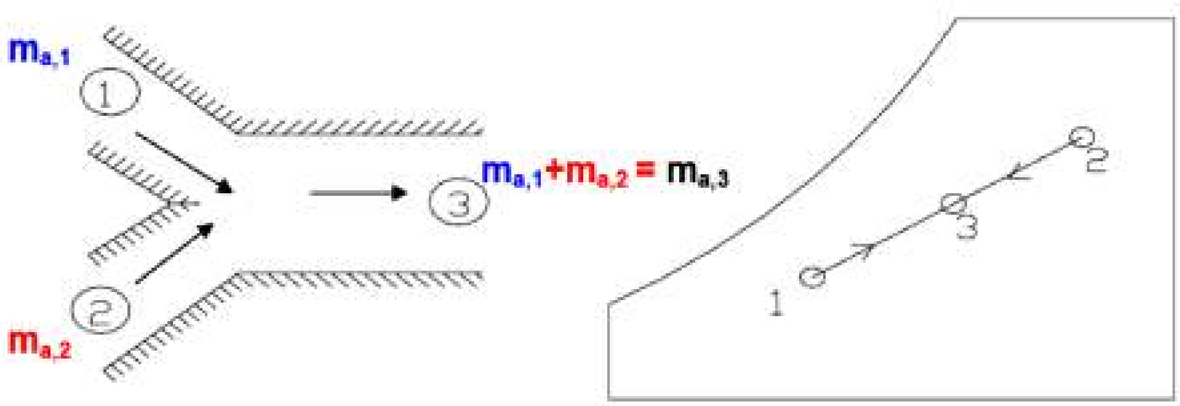


***Fig.7.10**. Mixing of two air streams without condensation*

From the mass balance of dry air and water vapor:

$$m_{a1}w_1 + m_{a2}w_2 = m_{a3}w_3$$

From energy balance:

$$m_{a1}h_1 + m_{a2}h_2 = m_{a3}h_3$$

From the above equations, it can be observed that the final enthalpy and humidity ratio of mixture are weighted averages of inlet enthalpies and humidity ratios. A generally valid approximation is that the final temperature of the mixture is the weighted average of the inlet temperatures. With this approximation, the point on the psychrometric chart representing the mixture lies on a straight line connecting the two inlet states. Hence, the ratio of distances on the line, i.e., **(1-3)/(2-3)** is equal to the ratio of flow rates $\mathbf{m_{a2}/m_{a1}}$. The resulting error (due to the assumption that the humid specific heats being constant) is usually less than 1 percent.

ii) Mixing with condensation:

As shown in **Fig.7.11**, when very cold and dry air mixes with warm air at high relative humidity, the resulting mixture condition may lie in the two-phase region, as a result there will be condensation of water vapor and some amount of water will leave the system as liquid water. Due to this, the humidity ratio of the resulting mixture (point 3) will be less than that at point 4. Corresponding to this will be an increase in temperature of air due to the release of latent heat of condensation. This process rarely occurs in an air conditioning system, but this is the phenomenon which results in the formation of fog or frost (if the mixture temperature is below 0 ^{0}C). This happens in winter when the cold air near the earth mixes with the humid and warm air, which develops towards the evening or after rains.

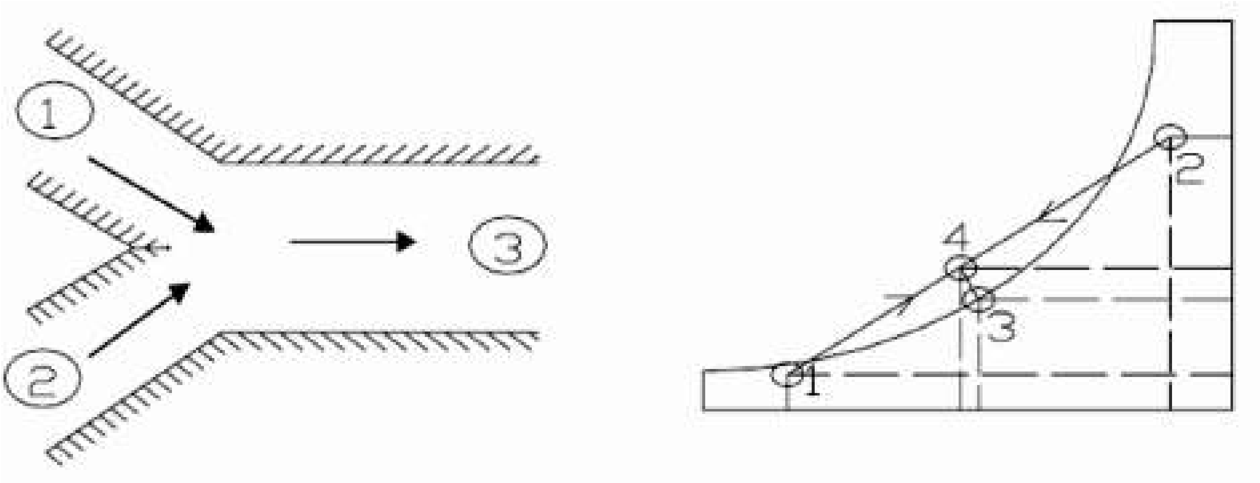


Fig.7.11. *Mixing of two air streams with condensation*

7.8 Air Washers

An air washer is a device for conditioning air. As shown in **Fig.7.12**, in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes.

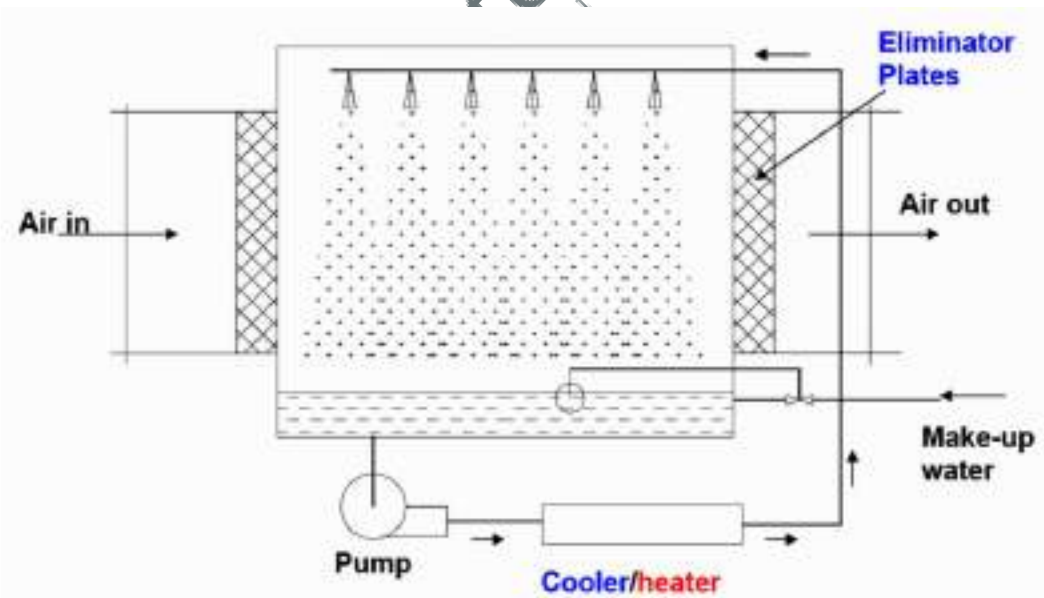


Fig. 7.12 *Air Washer*

In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer. As a consequence of the 2nd law, the heat transfer between air and water droplets will be in the direction of decreasing temperature gradient.

Similarly, the mass transfer will be in the direction of decreasing vapor pressure gradient. For example,

a) Cooling and dehumidification: $t_w < t_{DPT}$. Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled. Here both latent and sensible heat transfers are from air to water. This is shown by Process O-A in *Fig.1.13*.

b) Adiabatic saturation: $t_w = t_{WBT}$. Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or heating of water is required. That is this is a case of pure water recirculation. This is shown by Process O-B in *Fig.1.13*. Thus the process that takes place in a perfectly insulated evaporative cooler.

c) Cooling and humidification: $t_{DPT} < tw < t_{WBT}$. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from air to water, hence, water has to be cooled externally. This is shown by Process O-C in **Fig.7.13.**

d) Cooling and humidification: $t_{WBT} < t_w < t_{DBT}$. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is shown by Process O-D in **Fig.7.13**. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser.

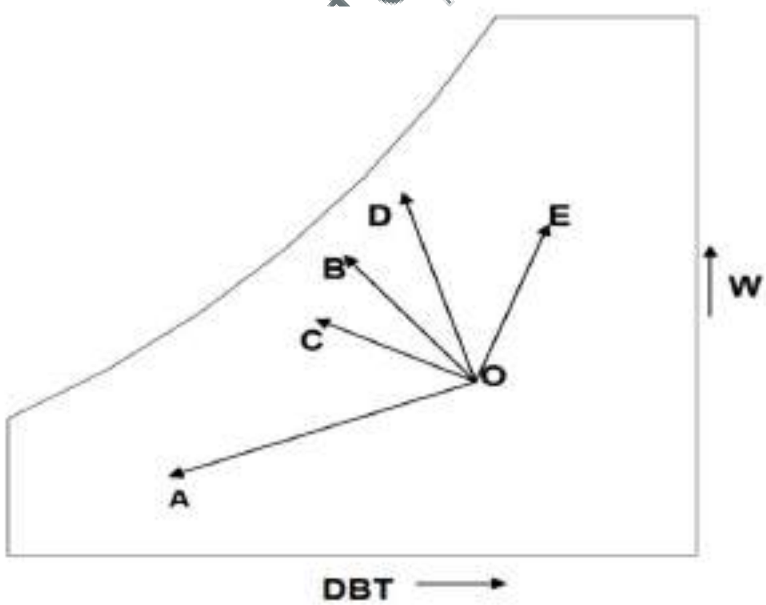


Fig. 7.13 *Various psychrometric processes that can take place in an air washer*

e) Heating and humidification: $t_w > t_{DBT}$. Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally. This is shown by Process O-E in

Fig.7.13. Thus, it can be seen that an air washer works as a year-round air conditioning system. Though air washer is a and extremely useful simple device, it is not commonly used for comfort air conditioning applications due to concerns about health resulting from bacterial or fungal growth on the wetted surfaces. However, it can be used in industrial applications.

➢ Examples

1) Moist air at 30°C, 1.01325 bars has a relative humidity of 80%. Determine without using the psychrometric chart

1) Partial pressures of water vapour and air
2) Specific humidity
3) Specific Volume and
4) Dew point temperature

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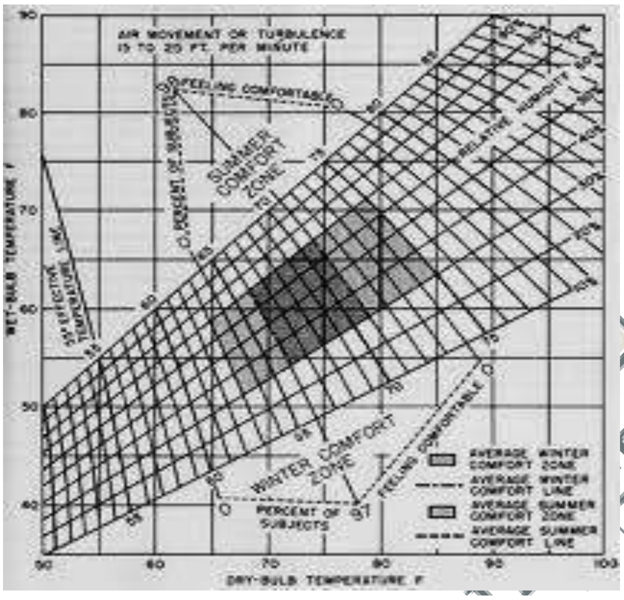
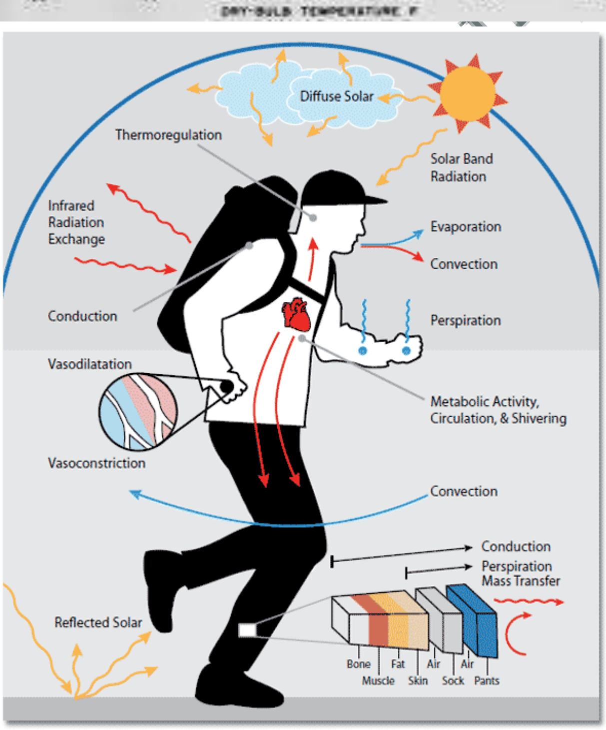
8

HUMAN COMFORT





Course Contents

8.1 Introduction

Thermal and atmospheric conditions in an enclosed space are usually controlled in order to ensure (1) the health and comfort of the occupants or (2) the proper functioning of sensitive electronic equipment, such as computers, or certain manufacturing processes that have a limited range of temperature and humidity tolerance. The former is referred to as ***comfort conditioning***, and the latter is called ***process air conditioning***. The conditions required for optimum operation of machinery may or may not coincide with those conducive to human comfort.

On the other hand, this improvement in comfort has come about at the expense of greater equipment installation, maintenance, and energy costs. A substantial portion of the energy consumed in buildings is related to the maintenance of comfortable environmental conditions. In fact, approximately 20 percent of the *total* U.S. energy consumption is directed toward this task.

But this doesn't have to continue to be the case. With an understanding of the factors that determine comfort in relation to climate conditions, designers may select design strategies that provide human comfort more economically. Thus, prior to investigating the energy-consuming mechanical systems in buildings, we will begin by discussing the concepts of human comfort.

8.2 Comfort Conditions

Besides being aesthetically pleasing, the human environment must provide light, air, and thermal comfort. In addition, proper acoustics and hygiene are important. Air requirements and thermal comfort are covered in this chapter

Comfort is best defined as the absence of discomfort. People feel uncomfortable when they are too hot or too cold, or when the air is odorous and stale. Positive comfort conditions are those that do not distract by causing unpleasant sensations of temperature, drafts, humidity, or other aspects of the environment. Ideally, in a properly conditioned space, people should not be aware of equipment noise, heat, or air motion.

The feeling of comfort or, more accurately, discomfort is based on a network of sense organs: the eyes, ears, nose, tactile sensors, heat sensors, and brain. ***Thermal comfort*** is that state of mind that is satisfied with the thermal environment; it is thus the condition of minimal stimulation of the skin's heat sensors and of the heat-sensing portion of the brain.

The environmental conditions conducive to thermal comfort are not absolute, but rather vary with the individual's metabolism, the nature of the activity engaged in, and the body's ability to adjust to a wider or narrower range of ambient.

For comfort and efficiency, the human body requires a fairly narrow range of environmental conditions compared with the full scope of those found in nature. The factors that affect humans pleasantly or adversely include:

1. Temperature of the surrounding air
2. Radiant temperatures of the surrounding surfaces
3. Humidity of the air
4. Air motion
5. Odors
6. Dust
7. Aesthetics
8. Acoustics
9. Lighting

Of these, the first four relate to thermal interactions between people and their immediate environment. In order to illustrate how thermal interactions affect human comfort, the explanation below describes the body temperature control mechanisms and how environmental conditions affect them.

Heat vs. Temperature

The sense of touch tells whether objects are hot or cold, but it can be misleading in telling just how hot or cold they are. The sense of touch is influenced more by the rapidity with which objects conduct heat to or from the body than by the actual temperature of the objects. Thus, steel feels colder than wood at the same temperature because heat is conducted away from the fingers more quickly by steel than by wood. As another example, consider the act of removing a pan of biscuits from an oven. Our early childhood training would tell us to avoid touching the hot pan, but at the same time, we would have no trouble picking up the biscuits themselves. The pan and biscuits are at the same temperature, but the metal is a better conductor of heat and may burn us. As this example illustrates, the sensors on our skin are poor gauges of temperature, but rather are designed to sense the degree of heat flow.

Heat

By definition, ***heat*** is a form of energy that flows from a point at one temperature to another point at a lower temperature. There are two forms of heat of concern in planning for comfort: (1) sensible heat and (2) latent heat. The first is the one we usually have in mind when we speak of heat.

Sensible Heat

Sensible heat is an expression of the degree of molecular excitation of a given mass. Such excitation can be caused by a variety of sources, such as exposure to radiation, friction between two objects, chemical reaction, or contact with a hotter object. When the temperature of a substance changes, it is the heat content of the object that is changing. Every material has a property called its ***specific heat***, which identifies how much its temperature changes due to a given input of sensible heat.

The three means of transferring sensible heat are radiation, convection, and conduction. All bodies emit ***thermal radiation***. The net exchange of radiant heat between two bodies is a function of the difference in temperature between the two bodies. When radiation encounters a mass, one of three things happens: (1) the radiation continues its journey unaffected (in which case it is said to be transmitted), (2) it is deflected from its course (in which case it is said to be reflected), or (3) its journey comes to an end (and it is said to be absorbed). Usually, the response of radiation to a material is some combination of transmission, reflection, and absorption. The radiation characteristics of a material are determined by its temperature, emissivity (emitting characteristics), absorptivity, reflectivity, and transmissivity. Conduction is the process whereby molecular excitation spreads through a substance or from one substance to another by direct contact. Convection occurs in fluids and is the process of carrying heat stored in a particle of the fluid to another location where the heat can conduct away.

Latent Heat

Heat that changes the state of matter from solid to liquid or liquid to gas is called *latent heat.* The ***latent heat of fusion*** is that which is needed to melt a solid object into a liquid. A property of the material, it is expressed per unit mass (per pound or per kilogram). The *latent heat of vaporization* is the heat required to change a liquid to a gas. When a gas liquefies (condenses) or when a liquid solidifies, it releases its latent heat.

Enthalpy

Enthalpy is the sum of the sensible and latent heat of a substance. For example, the air in our ambient environment is actually a mixture of air and water vapor. If the total heat content or enthalpy of air is known, and the enthalpy of the desired comfort condition is also known, the difference between them is the enthalpy or heat that must be added (by heating and humidification) or removed (by cooling and dehumidification).

Temperature

Temperature is a measure of the degree of heat intensity. Then temperature of a body is an expression of its molecular excitation. The temperature difference between two points indicates a potential for heat to move from the warmer to the colder point. The English system of units uses the Fahrenheit degree scale, while in SI units the Celsius degree scale is used. Note that temperature is a measure of heat *intensity,* whereas a Btu or joule is a measure of the *amount* of heat energy.

Body Temperature Control

Human beings are essentially constant-temperature animals with a normal internal body temperature of about 98.6°F (37.0°C). Heat is produced in the body as a result of metabolic activity, so its production can be controlled, to some extent, by controlling metabolism. Given a set metabolic rate, however, the body must reject heat at the proper rate in order to maintain thermal equilibrium. If the internal temperature rises or falls beyond its normal range, mental and physical operation is curtailed, and if the temperature deviation is extreme, serious physiological disorders or even death can result. Sometimes the body's own immunological system initiates a body temperature rise in order to kill infections or viruses. The importance of maintaining a fairly precise internal temperature is illustrated in **Fig. 8.1**, which shows the consequences of deep-body temperature deviations. When body temperature falls, respiratory activity particularly in muscle tissue automatically increases and generates more heat. Shivering is the extreme manifestation of this form of body temperature control. An extremely sensitive portion of the brain called the ***hypothalmus*** constantly registers the temperature of the blood and seems to be stimulated by minute changes in blood temperature originating anywhere in the body (this could result from drinking a hot beverage or a change in skin surface temperature). The skin also has sensors that signal to the brain the level of heat gain or loss at the skin.

It is the hypothalmus that appears to trigger the heat control mechanisms to either increase or decrease heat loss. This is accomplished by controlling the flow of blood to the skin by constricting or dilating the blood vessels within it. Since blood has high thermal conductivity, this is a very effective means of rapid thermal control of the body. By controlling peripheral blood flow, the body is able to (1) increase skin temperature to speed up elimination of body heat, (2) support sweating, or (3) reduce heat loss in the cold. When body temperature rises above normal, the blood vessels in the skin dilate, bringing more heat-carrying blood to the surface. This results in a higher skin temperature and, consequently, increased heat loss. At the same time, sweat glands are stimulated, opening the pores of the skin to the passage of body fluids which evaporate on the surface of the skin and thereby cool the body. This evaporating perspiration is responsible for a great deal of heat loss. A minor amount of heat is also lost continuously by evaporation of water from the lungs and respiratory tracts.

When body heat loss is high, people experience a feeling of lassitude and mental dullness brought about by the fact that an increased amount of the blood pumped by the heart goes directly from the heart to the skin and back to the heart, bypassing the brain and other organs.

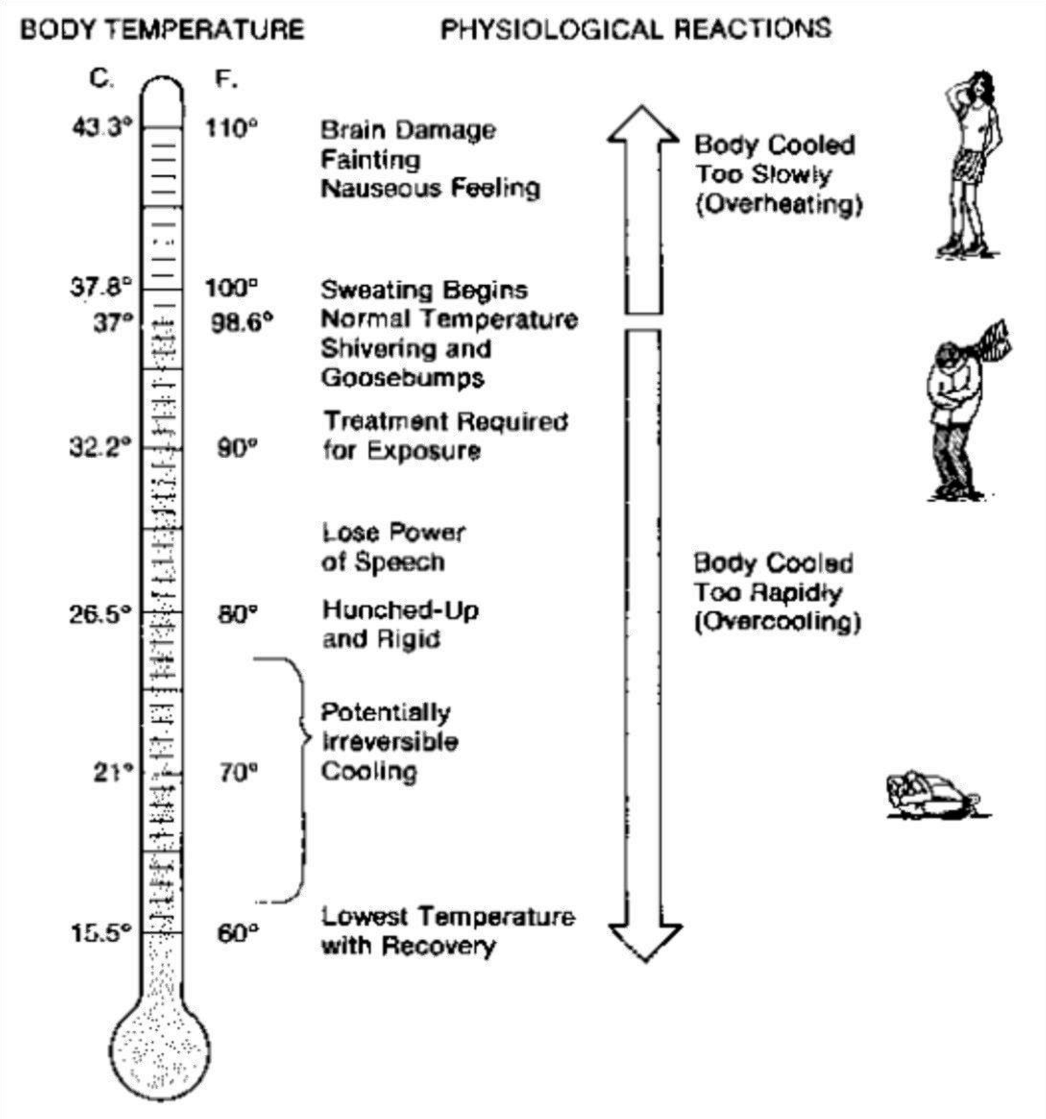


Fig.8.1 *Physiological reactions to body temperature*

A hot environment also increases strain on the heart, since it has to beat more rapidly to pump more blood to the periphery of the body. When the body loses more heat to a cold environment than it produces, it decreases heat loss by constricting the outer blood vessels, thereby reducing blood flow to the outer surface of the skin. This converts the skin surface to a layer of insulation between the interior of the body and the environment. It has about the same effect as putting on a light sweater. If the body is still losing too much heat, the control device increases heat production by calling for involuntary muscular activity or shivering. When heat loss is too great, the body tends to hunch up and undergo muscular tension, resulting in a strained posture and physical exhaustion if the condition persists for any length of time. Within limits, the body can acclimate itself to thermal environmental change. Such limits are not large, especially when the change is abrupt, such as when passing from indoors to outdoors. The slower seasonal changes are accommodated more easily. Changes in clothing assist the acclimatization. Whenever the body cannot adjust itself to the thermal environment, heat stroke or freezing to death is inevitable.

The physiological interpretation of comfort is the achievement of thermal equilibrium at our normal body temperature with the minimum amount of bodily regulation. We feel uncomfortable when our body has to work too hard to maintain thermal equilibrium. Under conditions of comfort, heat production equals heat loss without any action necessary by the heat control mechanisms. When the comfort condition exists, the mind is alert and the body operates at maximum efficiency.

It has been found that maximum productivity occurs under this condition and that industrial accidents increase at higher and lower temperatures. Postural awkwardness due to a cold feeling results in just as many accidents as does mental dullness caused by a too warm environment.

8.3 Heat Balance

Like all mammals, humans "burn" food for energy and must discard the excess heat. This is accomplished by evaporation coupled with the three modes of sensible heat transfer: conduction, convection, and radiation. For a person to remain healthy, the heat must not be lost too fast or too slowly, and a very narrow range of body temperature must be maintained. The body is in a state of thermal equilibrium with its environment when it loses heat at exactly the same rate as it gains heat. Mathematically, the relationship between the body's heat production and all its other heat gains and losses is:

$$\text{Heat production} = \text{heat loss}$$

OR

$$M = E \pm R \pm C \pm S$$

Where:

M = metabolic rate

E = rate of heat loss by evaporation, respiration, and elimination

R = radiation rate

C = conduction and convection rate

S = body heat storage rate

Equation is illustrated in **Fig. 8.2**.

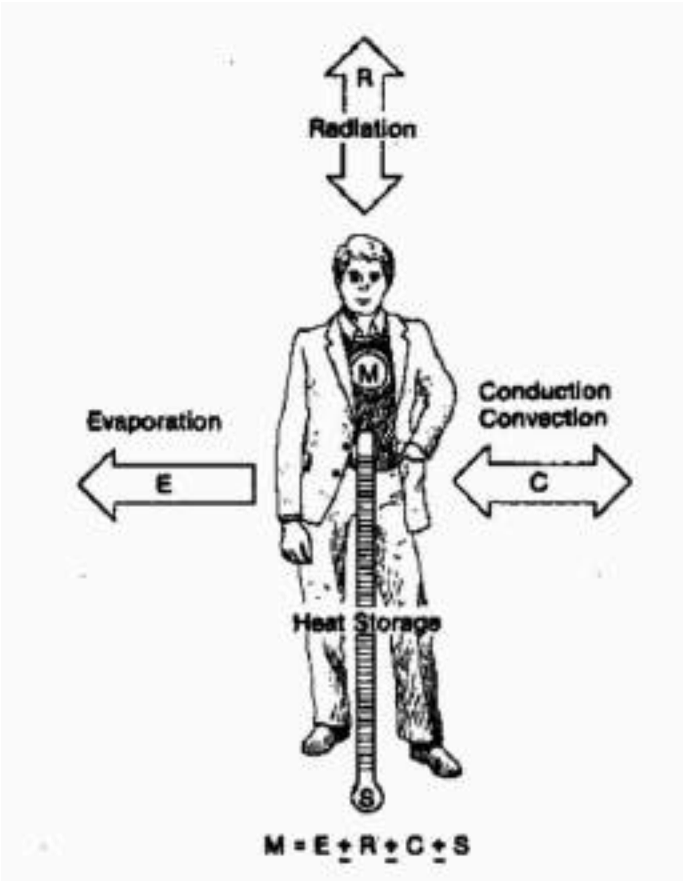


Fig. 8.2. *Heat balance of the human body interacting with its environment.*

The body always produces heat, so the metabolic rate (M) is always positive, varying with the degree of exertion. If environmental conditions are such that the combined heat loss from radiation, conduction, convection, and evaporation is less than the body's rate of heat production, the excess heat must be stored in body tissue. But body heat storage (S) is always small because the body has a limited thermal storage capacity. Therefore, as its interior becomes warmer, the body reacts to correct the situation by increasing blood flow to the skin surface and increasing perspiration.

As a result, body heat loss is increased, thereby maintaining the desired body temperature and the balance expressed by Equation. The converse condition where heat loss is greater than body heat production causes a reversal of the above process and, if necessary, shivering. This increased activity raises the metabolic rate.

Table 8.1 indicates the environmental and human factors that influence each of the major terms in Equation. Metabolism is discussed at greater length later in this chapter, while the other major factors—evaporation, radiation, conduction, and convection—are discussed below.

Table 8.1 Factors Influencing the Heat Balance Equation

Factor	Environment	Human
Metabolism (M)	Little effect Activity	Weight Surface area Age Sex
Evaporation (E)	Wet-bulb temperature Dry-bulb temperature Velocity	Ability to produce sweat Surface area Clothing
Radiation (R)	Temperature difference between bodies Emissivity of surfaces	Surface area Clothing
Convection (C)	Dry-bulb temperature Velocity	Clothing Mean body surface temperature Surface area

8.3.1 Evaporation

The body can either gain or lose heat by radiation (R) and conductive-convective heat transfer (C), depending on the temperature of the surrounding objects and ambient air. By contrast, evaporation (E) is exclusively a cooling mechanism. Evaporative losses usually play an insignificant role in the body's heat balance at cool temperatures. They become the predominant factor, however, when ambient temperatures are so high that radiant or convective heat losses cannot occur. At comfortable temperatures, there is a steady flow of sensible heat from the skin to the surrounding air. The amount of this sensible heat depends upon the temperature difference between the skin and air. Although the deep body temperature remains relatively constant, the skin temperature may vary from 40° to 105°F (4° to 41°C) according to the surrounding temperature, humidity, and air velocity.

During the heating season, the average surface temperature of an adult indoors wearing comfortable clothing is approximately 80°F (27°C). At lower surrounding temperatures, the skin temperature is correspondingly lower. When the surrounding environment is about 70°F (21°C), most people lose sensible heat at a rate that makes them feel comfortable. If the ambient temperature rises to the skin temperature, the sensible heat loss drops to zero. If the ambient temperature continues to rise, the body gains heat from the environment, and the only way it can lose heat is by increasing evaporation.

Evaporative heat losses also increase at high activity levels, when the metabolic heat production rises. A person engaged in strenuous physical work may sweat as much as a quart of fluid in an hour. The rate of evaporation and evaporative heat loss is determined by the evaporation potential of the air. It is dependent to a minor degree on the relative humidity of the surrounding air and, to a much greater extent, on the velocity of air motion. Moisture, which is evaporated from the skin surface, is carried away by the passing air stream. Sufficient heat must be added to the perspiration to vaporize it, and this heat is drawn from the body. This heat loss equals the latent heat of vaporization of all the moisture evaporated. It is thus commonly known as the latent heat component of the total heat rejected by the body.

While the skin sweats only at moderate to high temperatures, evaporative losses of water from the respiratory passages and lungs occur continuously. The breath "seen" when exhaled in frosty weather is evidence that the air leaving the lungs has high moisture content. We generally exhale air that is saturated (100 percent RH), and even at rest, the body requires about 100 Btuh (30 W) of heat to evaporate this moisture from the lungs into the inhaled air. Since it takes a considerable amount of heat to convert water into vapor, the evaporative heat loss from our lungs and skin plays an important role in disposing of body heat.

8.3.2 Radiation

Radiation is the net exchange of radiant energy between two bodies across an open space. The human body gains or loses radiant heat, for example, when exposed to an open fire, the sun, or a window on a cold winter day.

Each body the earth, the sun, a human body, a wall, a window, or a piece of furniture interacts with every other body in a direct line of sight with it. Radiation affects two bodies only when they are in sight of each other. This means that the energy cannot go around corners or be affected by air motion. For example, when we are uncomfortably hot in the direct light of the sun, we can cut off the radiant energy coming directly from the sun by stepping into the shade of a tree. Since air is a poor absorber of radiant heat, nearly all radiant exchanges are with solid surfaces to which we are exposed.

Radiant heat may travel toward or away from a human body, depending on whether the radiating temperatures of surrounding surfaces are higher or lower than the body's temperature. In a cold room, the warmer body or its clothing transmits radiant heat to all cooler surfaces such as walls, glass, and other construction within view. If there is a cold window in sight, it will typically have the largest impact in terms of draining heat away and making the body feel colder. By closing the drapes, a person can block the radiant transfer in the same way that a person can cut off the radiant energy from the sun by stepping into the shade of a tree.

The rate of radiant transfer depends on the temperature differential, the thermal absorptivity of the surfaces, and the distance between the surfaces. The body gains or loses heat by radiation according to the difference between the body surface (bare skin and clothing) temperature, and the MRT of the surrounding surfaces.

The MRT is a weighted average of the temperatures of all the surfaces in direct line of sight of the body (see **Fig. 8.3**) although the MRT tends to stabilize near the room air temperature; it is also affected by large glass areas, degree of insulation, hot lights, and so on. The inside surface temperature of an insulated wall will be much closer to the room air temperature than will that of an uninsulated wall.

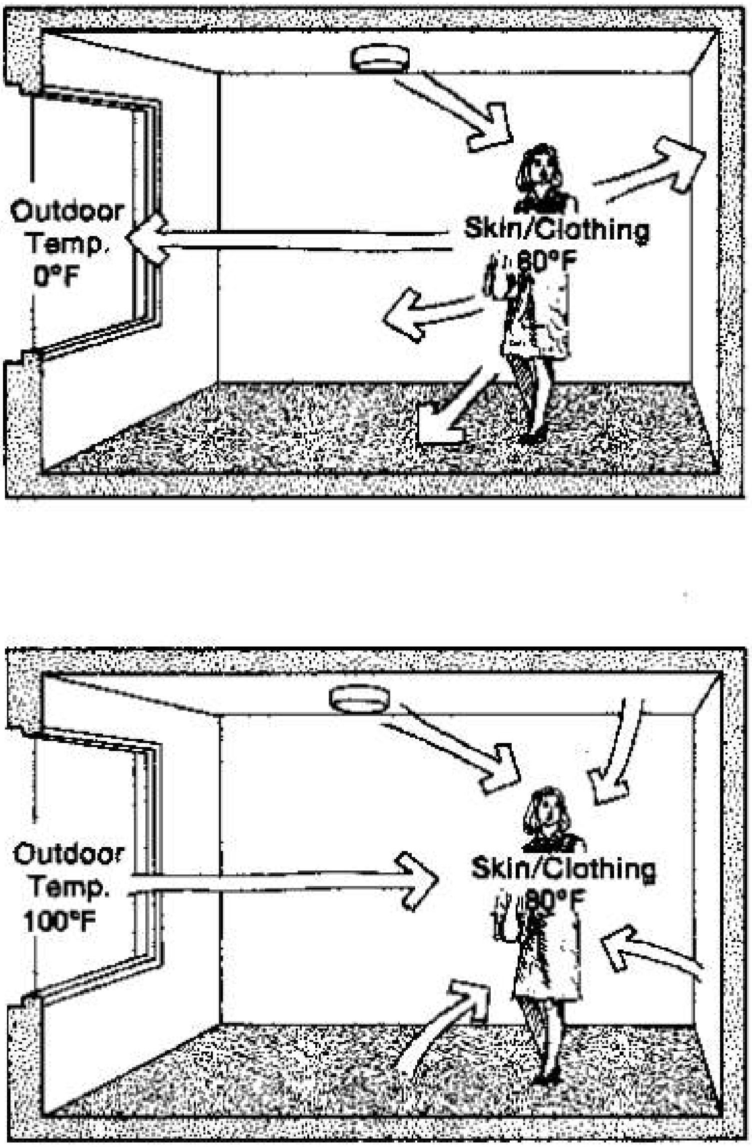


Fig. 8.3. *Radiant heat transfer with surrounding surfaces.*

If the MRT is below the body temperature, the radiant heat term, R, in Equation is a positive number, and the body is losing radiant heat. If the MRT is above the body temperature, R is negative, and the body is gaining radiant heat. This could be a benefit if the room air temperature is cool, causing excess body heat loss, while it would be detrimental if the ambient conditions are hot and humid, and the body is already having trouble rejecting heat.

It should be kept in mind that the body loses radiant heat according to its surface temperature. For a comfortable, normally dressed adult, the weighted average temperature of the bare skin and clothed surfaces is about 80°F (27°C). In still air at a temperature near skin temperature, radiant exchange is the principal form of heat exchange between the body and its environment.

8.3.3 Convection-Conduction

Air passing over the skin surface is instrumental not only to the evaporation of moisture, but also to the transference of sensible heat to or from the body. The faster the rate of air movement, the larger the temperature difference between the body and surrounding air, and the larger the body surface area, the greater the rate of heat transfer.

When the air temperature is lower than the skin (and clothing) temperature, the convective heat term in equation 1.2 is "plus," and the body loses heat to the air. If the air is warmer than the skin temperature, the convective heat term is "minus," and the body gains heat from the air. Convection becomes increasingly effective at dissipating heat as air temperature decreases and air movement increases.

The conduction heat loss or gain occurs through contact of the body with physical objects such as the floor and chairs. If two chairs one with a metal seat and the other with a fabric seat have been in a 70°F (21°C) room for a period of time, they will both have a temperature of 70°F (21°C), but the metal one will feel colder than the one with the woven seat. There are two reasons for this. First, metal is a good conductor, and it is the rate at which heat is conducted away not the temperature that we feel. Also, the metal chair has a smoother surface, which makes a good contact and thus facilitates better conduction. Clothing also plays an important role in conductive heat transfer, insulating us from the warm or cold surface, just as a pot holder protects us from a hot pot.

8.3.4 Combined Effects

The physiological basis of comfort was previously stated as the achievement of thermal equilibrium with a minimal amount of body regulation of M, E, R, and C. **Fig. 2.4** shows the relation between all these factors for lightly clothed and unclothed subjects at rest. Note that convective and radiant heat loss is greater for the lightly clothed subjects. Also, the heat loss by convection and radiation decreases with increasing air temperature, while evaporative heat loss increases with increasing air temperature. Heat loss by evaporation is relatively constant below certain air temperatures approximately 75°F (24°C) for the heavily clothed subject and 85°F (29°C) for the lightly clothed subject. The metabolic rate at a given activity level is stable when the temperature ranges from about 70° to 90°F (21° to 32°C).

To illustrate the various modes of heat loss operating in conjunction, consider a person outdoors in 100°F (38°C) air temperature. Referring to Equation, the convective heat loss is "minus" because the body is gaining heat from the air. The MRT is much higher than the body surface temperature the sidewalk, street, building walls, sunny sky, and everything else in the range of view of the body is warmer than the body surface temperature. Thus, the radiant heat term is also "minus" because the body is gaining radiant heat.

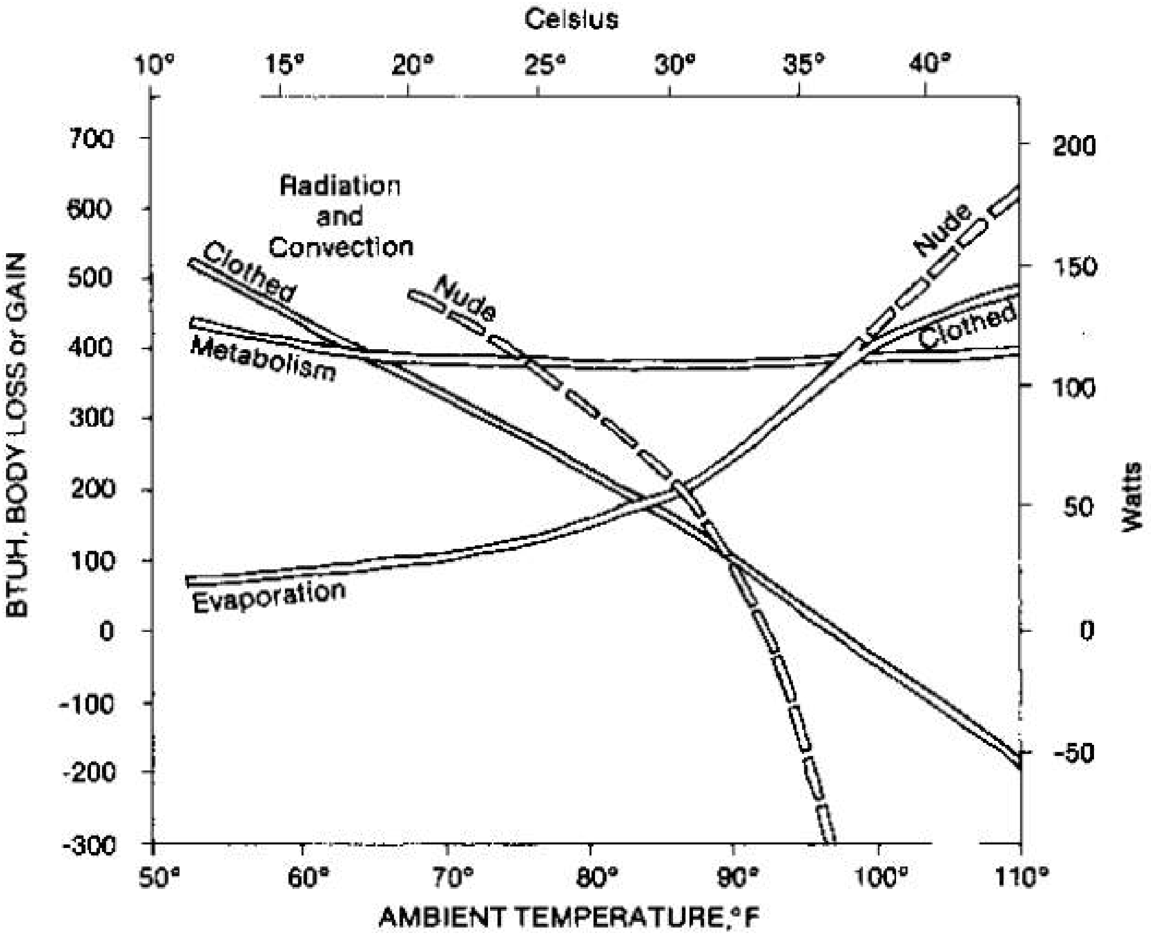


Fig. 8.4. *Relationship between metabolism, evaporation, radiation, convection, and temperature.*

But as the person walks down the sidewalk, the metabolism produces about 700 Btuh (200 W), and all that heat must be lost in addition to that gained by convection and radiation in order to maintain the heat balance. The total the body must lose may be over 1,000 Btuh (300 W), all by evaporation. The sweat glands automatically open, and the resultant moisture emitted onto the body surface then evaporates. The heat drawn from the body to evaporate the moisture keeps the skin cool as long as the surrounding air will carry away the water vapor so that more can be evaporated. This in turn keeps the deep-body temperature close to 98.6°F (37.0°C). As the dry-bulb temperature of the surrounding air rises from the comfortable 70s to the 80s and 90s, less sensible heat (convective and radiative) is lost by the body, while the latent heat (evaporation) loss increases. Thus, if a body at rest produces 400 Btuh (117 W), it may lose 290 Btuh (85 W) of sensible heat and 110 Btuh (32 W) of latent heat at 70°F (21°C). At higher temperatures, the sensible component drops to nearly zero, and the latent heat must increase to almost the full 400 Btuh (117 W) in order to lose the same amount of heat.

When people work under conditions of high temperature and extremely high humidity, both the sensible heat loss and the evaporation of moisture from their skins are reduced. Under these conditions, the rate of evaporation must be increased by blowing air rapidly over the body.

8.4 Environmental Factors

Satisfaction with the thermal environment is a complex, subjective response to many interacting variables. Our perception of comfort is influenced by these variables, which include the characteristics of the physical environment, amount of clothing, activity level, and the demographic character of the subject (age, sex, health, etc.). Researchers have identified the seven major determinants of thermal comfort response:

1. Air (dry-bulb) temperature
2. Humidity
3. Mean radiant temperature
4. Air movement
5. Clothing
6. Activity level
7. Rate of change of any of the above

As any one of these variables changes, the others need to be adjusted to maintain the thermal equilibrium between heat gain and heat loss in order for a person to continue to feel comfortable. The important environmental parameters are temperature, humidity, radiation, and air movement, while the important personal parameters are clothing and activity. The personal parameters have already been covered, so the following discussion concentrates on the environmental parameters.

8.4.1 Dry-Bulb Temperature

Dry-bulb temperature affects the rate of convective and evaporative body heat loss. It is perhaps the most important determinant of comfort, since a narrow range of comfortable temperatures can be established almost independently of the other variables. There is actually a fairly wide range of temperatures that can provide comfort when combined with the proper combination of relative humidity, MRT, and air flow. But as any one of these conditions varies, the dry-bulb temperature must be adjusted in order to maintain comfort conditions.

Temperature drifts or ramps are gradual temperature changes over time. *Drifts* refer to passive temperature changes, while ***ramps*** are actively controlled temperature changes. People may feel comfortable with temperatures that rise or fall like a ramp over the course of time, even though they would be uncomfortable if some of the temperatures were held constant. Ideal comfort standards call for a change of no more than 1°F/hr (0.6°C/hr) during occupancy, provided that the temperature excursion doesn't extend far beyond the specified comfort conditions and for very long.

8.4.2 Humidity

Humidity is the amount of water vapor in a given space. The density of water vapor per unit volume of air is called ***absolute humidity***. It is expressed in units of kg (of water) per cubic meter (of dry air). The ***humidity ratio*** or ***specific humidity*** is the weight of water vapor per unit weight of dry air; it is given in either grains per kg or kg per kg of dry air.

The amount of moisture that air can hold is a function of the temperature. The warmer the air, the more moisture it can hold. The amount of water present in the air relative to the maximum amount it can hold at a given temperature without causing condensation (water present ÷ maximum water holding capability) is known as the ***degree of saturation***.

This ratio multiplied by 100 is the ***percentage humidity***. This percentage is a measure of the dryness of air. Low percentages indicate relative dryness, and high percentages indicate high moisture.

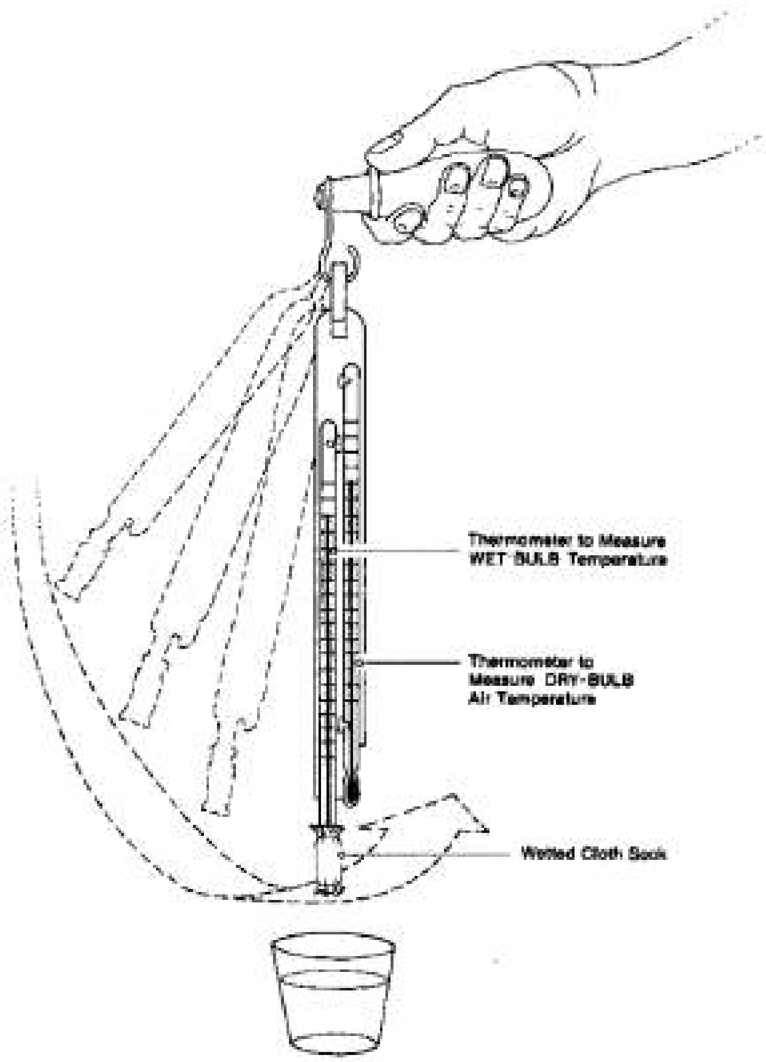


Fig.8.5 *Sling psychrometer.*

Humidity can be expressed as dew point, RH, wet-bulb temperature, or vapor pressure. None of these, however, by itself defines the amount of moisture present without knowledge of one of the others or the coincident dry-bulb temperature. In general, any of these five parameters can be found by means of tables or a psychrometric chart if any two of them are already known. A common and simple instrument for determining humidity is the sling psychrometer shown in **Fig.8.5**. It consists of two mercury-filled glass thermometers mounted side by side on a frame fitted with a handle by which the device can be whirled through the air. One of the thermometers has a cloth sock that is wetted. As moisture from the wet sock evaporates into the moving air, the wet-bulb temperature drops. The drier the air surrounding the sling psychrometer, the more moisture that can evaporate from the sock. This evaporation lowers the wet-bulb temperature accordingly. The greater the difference between the wet-bulb and dry-bulb temperatures (called the ***wet-bulb depression***), the lower the RH. A table is normally provided with the device for correlating dry- and wet-bulb temperatures with RH.

8.4.3 Mean Radiant Temperature

As an illustration of the importance of radiant temperature, experiments have been conducted in rooms in which the surface temperatures were controlled. Subjects were surprised to find out that they were warm at air temperatures of 50°F (10°C) when the room surfaces were sufficiently heated and that they were cool at air temperatures of 120°F (49°C) when room surfaces were cooled.

The MRT affects the rate of radiant heat loss from the body. Since the surrounding surface temperatures may vary widely, the MRT is a weighted average of all radiating surface temperatures within line of sight. Two-dimensionally, it can be calculated as follows:

$$MRT = \frac{\Sigma T\theta}{360} = \frac{T_1\theta_1 + T_2\theta_2 + \ldots\ldots\ldots + T_n\theta_n}{360}$$

Where,

T = surface temperature

θ = surface exposure angle (relative to occupant) in degrees

8.4.4 Air Movement

Air motion significantly affects body heat transfer by convection and evaporation. Air movement results from free (natural) and forced convection as well as from the occupants' bodily movements. The faster the motion, the greater the rate of heat flow by both convection and evaporation.

Table 8.2 Subjective respond to air motion

Air Velocity		Occupant Reaction
fpm	m/s	
0 to 10	0 to 0.05	Complaints about stagnation
10 to 50	0.05 to 0.25	Generally favorable (air outlet devices normally designed for 50 fpm in the occupied zone)
50 to 100	0.25 to 0.51	Awareness of air motion, but may be comfortable, depending on moving air temperature and room conditions

100 to 200	0.51 to 1.02	Constant awareness of air motion, but can be acceptable (e.g., in some factories) if air supply is intermittent and if moving air temperature and room conditions are acceptable
200 (about 2 mph)	1.02 and above	Complaints about blowing of papers and hair, and other and above annoyances

When ambient temperatures are within acceptable limits, there is no minimum air movement that must be provided for thermal comfort. The natural convection of air over the surface of the body allows for the continuous dissipation of body heat. When ambient temperatures rise, however, natural air flow velocity is no longer sufficient and must be artificially increased, such as by the use of fans. Typical human responses to air motion are shown in **Table 8.2**.

In general, insufficient air motion promotes stuffiness and air stratification. Stratification causes air temperatures to vary from floor to ceiling. When air motion is too rapid, unpleasant drafts are felt by the room occupants. The exact limits to acceptable air motion in the occupied zone are a function of the overall room conditions of temperature, humidity, and MRT, along with the temperature and humidity conditions of the moving air stream.

8.5 Thermal Comfort Standards

Thermal Indices

Thermal sensations can be described as feelings of being hot, warm, neutral, cool, cold, and a range of classifications in between. There have been numerous attempts to find a single index—integrating some or all of the environmental factors—that could be used to determine thermal comfort conditions (for a given metabolic rate and amount of clothing). The following are the most common of these indices still in use.

8.5.1 Dry- and Wet-Bulb Temperatures

The simplest practical index of cold and warmth is the reading obtained with an ordinary dry-bulb thermometer. This long-established gauge is fairly effective in judging comfort for average humidity (40 to 60 percent RH), especially in cold conditions. In the heat, when humidity greatly affects the efficiency of body temperature regulation by sweating, the significance of the dry-bulb temperature is limited. The wet-bulb temperature represents an improvement over the simple dry-bulb temperature by taking humidity into account.

8.5.2 Operative Temperature

Operative temperature is the uniform temperature of an imaginary enclosure in which the occupant would exchange the same heat by radiation and convection as in the actual environment. An alternative definition of operative temperature is an average of MRT and dry-bulb temperatures weighted by the respective radiation and convection heat transfer coefficients.

Humid operative temperature is the uniform temperature of an imaginary environment at 100 percent RH with which the occupant would exchange the same heat by radiation, convection, conductance through clothing, and evaporation as in the actual environment.

8.5.3 New Effective Temperature

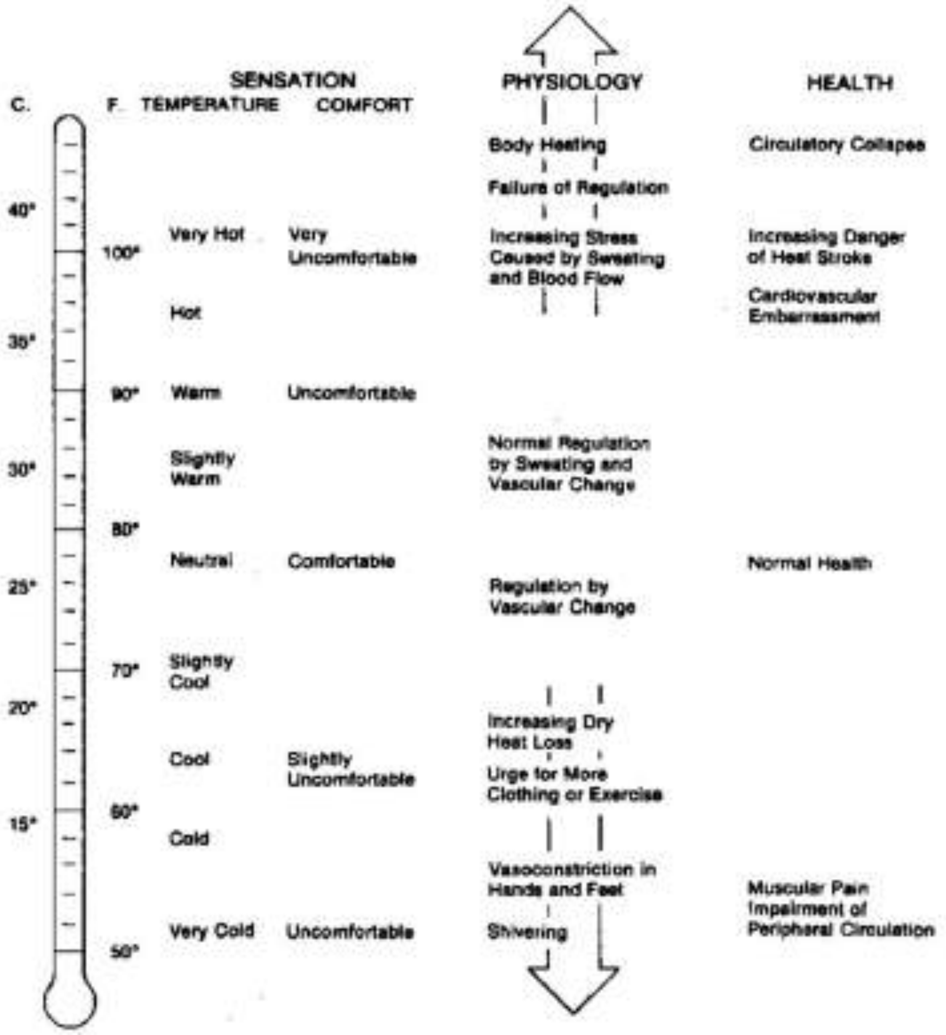


Fig. 8.6 *The ET* scale correlated to physiological reactions, comfort, and health.*

Effective temperature is not an actual temperature in the sense that it can be measured by a thermometer. It is an experimentally determined index of the various combinations of dry-bulb temperature, humidity, radiant conditions, and air movement that induce the same thermal sensation. Those combinations that induce the same feeling of warmth or cold are called thermo-equivalent conditions.

The ***new effective temperature*** (*ET**) of a given space is defined as the dry-bulb temperature of a thermo-equivalent environment at 50 percent RH and a specific uniform radiation condition. Thus, any space has an ET* of 70°F (21°C) when it induces a sensation of warmth like that experienced in still air at 70°F (21°C), 50 percent RH, and the proper radiant conditions. ET* is, in general, a reliable indicator of discomfort or dissatisfaction with the thermal environment. If ET* could be envisioned as a thermometer scale, it would appear as in **Fig.8.6**.

8.6 The Comfort Chart

The comfort chart, shown in **Fig.8.7**, correlates the perception of comfort with the various environmental factors known to influence it. The dry-bulb temperature is indicated along the bottom. The right side of the chart contains a dew point scale, and the left side a wet-bulb temperature scale indicating guide marks for imaginary lines sloping diagonally down from left to right. The lines curving upward from left to right represent RHs. ET* lines are also drawn. These are the sloping dashed lines that cross the RH lines and are labeled in increments of 5°F. At any point along any one of these lines, an individual will experience the same thermal sensation and will have the same amount of skin wetness due to regulatory sweating. Clo levels at which 94 percent of occupants will find acceptable comfort are also indicated.

Notice that the comfort chart in **Fig.8.7** is derived from what is called the psychrometric chart two ***comfort envelopes*** or zones are defined by the shaded regions on the comfort chart—one for winter and one for summer. The thermal conditions within these envelopes are estimated to be acceptable to 80 percent of the occupants when wearing the clothing ensemble indicated. To satisfy 90 percent of the people, the limits of the acceptable comfort zone are sharply reduced to one-third of the above ranges.

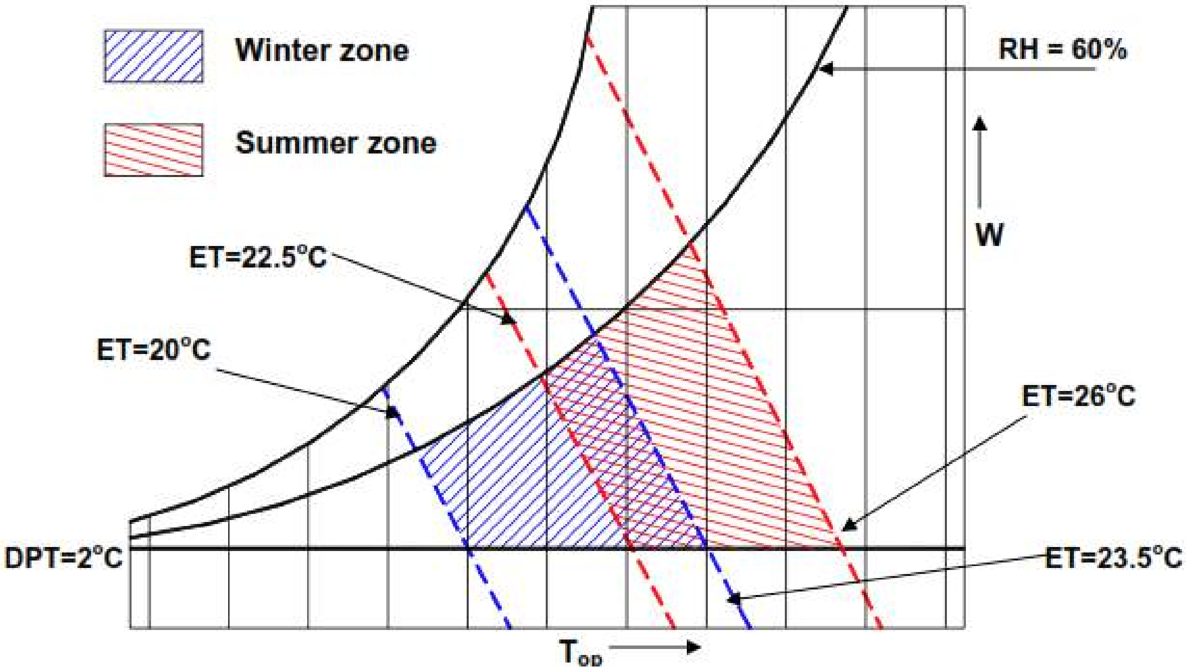


Fig. 8.7 *The comfort chart*

The zones overlap in the 73° to 75°F (23° to 24°C) range. In this region, people in summer dress tend to be slightly cool, while those in winter clothing feel a slightly warm sensation.

Fig.8.7 generally applies when altitudes range from sea level to 7,000 feet (2,134 m), MRT is nearly equal to dry bulb air temperature, and air velocity is less than 40 fpm (0.2 m/s). Under these conditions, thermal comfort can be defined in terms of two variables: dry-bulb air temperature and humidity.

Mean radiant temperature is actually just as important as air temperature in affecting comfort. When air movement in an indoor environment is low, the operative temperature is approximately the average of air temperature and MRT. When the MRT in the occupied zone significantly differs from the air temperature, the operative temperature should be substituted for the dry-bulb temperature scale along the bottom of **Fig.8.7**.

The comfort chart is primarily useful for occupants with a minimum of 1 hour of exposure, 0.6 clo (standard shirtsleeve indoor office clothing), and a 1-met (seated or sedentary) activity level. It is secondarily useful at higher temperatures to identify when there is a risk of sedentary heat stress. Although the ET* scale is based on 1-hour exposure, data show no significant changes in response with longer exposures unless the limits of heat stress ET* greater than 90°F (32°C) are approached. As the ET* lines show, humidity between about 20 and 55 percent RH has only a small effect on thermal comfort. Its effect on discomfort increases with both temperature and the degree of regulatory sweating. Evaporative heat loss near the comfort range is only about 25 percent of the total heat loss. As the temperature increases, this percentage grows until the ambient temperature equals skin (and clothes) temperature, at which point evaporation accounts for 100 percent of the heat loss.

The upper and lower humidity limits on the comfort envelope of **Fig.8.7** are based on considerations of respiratory health, mold growth, and other moisture-related phenomena in addition to comfort. Humidification in winter must be limited at times to prevent condensation on cold building surfaces such as windows.

The environmental parameters of temperature, radiation, humidity, and air movement necessary for thermal comfort depend upon the occupant’s clothing and activity level.

The comfort chart was developed from ASHRAE research, which has usually been limited to lightly clothed occupants (0.5 to 0.6 clo) engaged in sedentary activities. The reasoning behind this approach is that 90 percent of people’s indoor work and leisure time is spent at or near the sedentary activity level. In line with this rationale, the comfort envelope defined in **Fig.8.7** strictly applies only to sedentary and slightly active, normally clothed persons at low air velocities, when the MRT is equal to air temperature. For other conditions, the comfort zone must be adjusted accordingly.

For example, comfort can be maintained at temperatures as low as 68°F (20°C) for an individual wearing a clothing ensemble measuring less than 1.34 clo if he or she gets up and moves around for at least 10 minutes out of every half hour. On the other end of the scale, comfort conditions may be extended upward to 82°F (28°C) with a fan-induced air velocity of 160 fpm (0.8 m/s).

Within the comfort envelope of *Fig.8.7*, there is no minimum air movement necessary for thermal comfort. However, the maximum allowable air motion is lower in the winter than in the summer. In the wintertime, the average air movement within the occupied zone should not exceed 30 fpm (0.15 m/s). If the temperature is less than the optimum or neutral sensation temperature, slight increases in air velocity or irregularity of air movement can cause uncomfortable localized drafts. While possibly of little consequence in an active factory, this can create significant problems in professional buildings, religious buildings, and other places where people are seated and wearing light indoor clothing.

In the summer, the average air movement in the occupied zone can go as high as 50 fpm (0.25 m/s) under standard temperature and humidity conditions. Above 79°F (26°C), comfort can be maintained by increasing the average air motion 30 fpm for each °F (0.275 m/s for each °C) of increased temperature up to a maximum of 160 fpm (0.8 m/s). At that point, loose paper, hair, and other light objects start blowing around.

As average steady-state activity increases above the sedentary or slightly active (1.2-met) levels, sweating increases. To maintain comfort, the air motion must be increased, or the operative temperature must be decreased.

The development of reliable thermal comfort indices has been very important for the effective control of the thermal environment using no more energy and equipment than is necessary. The five variables affecting comfort for a given activity or room function are dry-bulb temperature, humidity, MRT, air movement, and clothing. Sometimes when one of these conditions is out of the comfort range, adjusting one or more of the other conditions will restore comfort with the addition of little or no additional energy.

The most commonly recommended design conditions for comfort are:

ET* = 75°F (24°C)

Dry-bulb air temperature = MRT

Relative humidity = 40% (20 to 60% range)

Air velocity less than 40 fpm (0.2 m/s)

8.7 Design Considerations

The comfort chart is useful for determining design conditions to be met by a building envelope and its ***heating, ventilation***, and ***air conditioning*** (HVAC) equipment. But considerable judgment must still be exercised if this chart is to be employed as a guide. For example, noticeably uneven radiation from hot and cold surfaces, temperature stratification in the air, a wide disparity between air temperature and MRT, a chilly draft, contact with a warm or cool floor, and other factors can cause local discomfort and reduce the thermal acceptability level of the space. Although a person may feel thermally neutral in general preferring neither a warmer nor a cooler environment thermal discomfort may exist if one part of the body is warm and another is cold.

This relates to both the building envelope design and the system for HVAC. A higher air temperature may be necessary to compensate for extensive cold glass areas. Or if large radiant heating panels are contemplated, a lower air temperature might be allowable, since radiant gain to the body will permit greater convective and evaporative losses. Direct sunlight from large windows or skylights *requires* lower room air temperature in order to compensate for the high radiant gain. In order to avoid having to lower the air temperature in the summer, the windows or skylights could be shaded.

When radiant surfaces are too cool for comfort, the air temperature must be increased from 0.3° to 1° for every 1° reduction in MRT, depending on room conditions. Since the comfort chart is the result of observations of healthy, clothed, sedentary subjects, spaces to be conditioned for very active, ill, or nude persons may require considerably different conditions for comfort than those indicated.

The condition, clothing, and activity level of the occupants, as well as the humidity, MRT, and air motion conditions in the space, must also be taken into consideration.

People engaged in physical work need a lower effective temperature for comfort than do sedentary ones. The greater the activity and the more clothing worn, the lower the effective temperature must be for comfort. Although the latent heat liberated by people engaged in any physical activity raises sharply, the liberated sensible heat changes very little. For example, an average man seated at rest in a room at 80°F DB gives off 180 Btuh of sensible heat and 150 Btuh of latent heat, or a total of 330 Btuh (27°C, 53 W, 44 W, and 97 W, respectively). Now, if he engages in light bench work, he will liberate 220 Btuh of sensible heat and 530 Btuh of latent heat (64 W and 155 W). This represents a sensible heat increase of about 20 percent but a latent increase of about 250 percent.

Other distinctions between winter and summer that must be taken into consideration in selecting design conditions are:

- The MRT of perimeter and top-floor spaces are lower in winter than in summer due to the influence of outdoor air conditions on the building shell.
- Conventions of clothing insulation are different in winter and summer.
- Expectations (psychological and physiological acclimatization) of indoor thermal conditions vary according to season.

8.8 Air Quality and Quantity

Besides the thermal conditions of an environment, comfort and health also depend on the composition of the air itself. For example, people feel uncomfortable when the air is odorous or stale. The quality of air in a space can even seriously affect its ability to support life. Under heavy occupancy of a space, the concentration of carbon dioxide can rise to deleterious levels. In addition, excessive accumulations of some air contaminants become hazardous to both plants and animals.

Air Contaminants

Air normally contains both oxygen and small amounts of carbon dioxide (0.03 percent), along with varying amounts of particulate materials referred to as *permanent atmospheric impurities.* These materials arise from such natural processes as wind erosion, evaporation of sea spray, and volcanic eruption. The concentrations of these materials in the air vary considerably but are usually below the concentrations caused by human activity.

Air composition can change drastically. In sewers, sewage treatment plants, tunnels, and mines, the oxygen content of air may become so low that it cannot support human life. Concentrations of people in confined spaces, such as theaters, require the removal of carbon dioxide given off by respiration and replacement with oxygen.

At atmospheric pressure, oxygen concentrations of less than 12 percent or carbon dioxide concentrations greater than 5 percent are dangerous even for short periods. Smaller deviations from normal concentrations can be hazardous under prolonged exposures.

Artificial contaminants are numerous, originating from a variety of human activities. Contaminants in the indoor environment, of which tobacco is a prime example, are of particular concern to building designers.

Air contaminants can be particulate or gaseous, organic or inorganic, visible or invisible, toxic or harmless. Loose classifications are (1) dust, fumes, and smoke, which are chiefly *solid* particulates (although smoke often contains liquid particles); (2) mist and fog, which are *liquid* particulates; and (3) vapors and gases, which are *no particulates.*

Dust consists of solid particles projected into the air by natural forces such as wind, volcanic eruption, or earthquakes, or by human activities. *Fumes* are solid airborne particles usually 100 times smaller than dust particles, commonly formed by condensation of vapors of normally solid materials. Fumes that are permitted to age tend to agglomerate into larger clusters. *Smoke* is made up of solid or liquid particles about the same size as fumes, produced by the incomplete combustion of organic substances such as tobacco, wood, coal, and oil. This class also encompasses *airborne living organisms* which range in size from submicroscopic viruses to the larger pollen grains. Included are bacteria and fungus spores, but not the smallest insects. *Mist* is defined as very small airborne droplets of a liquid that are formed by atomizing, spraying, mixing, violent chemical reactions, evaporation, or escape of a dissolved gas when pressure is released. Sneezing expels or atomizes very small droplets containing microorganisms that become air contaminants. *Fog* is very fine airborne droplets usually formed by condensation of vapor. Fogs are composed of droplets that are smaller than those in mists, but the distinction is insignificant, and both terms are commonly used to indicate the same condition.

Ventilation

The concentration of indoor air contaminants and odors can be maintained below levels known to impair health or cause discomfort, by the controlled introduction of fresh air to exchange with room air. This is known as ventilation. Humans require fresh air for an adequate supply of oxygen, which is necessary for metabolism of food to sustain life. Carbon and hydrogen in foods are oxidized to carbon dioxide and water, which are eliminated by the body as waste products. The rate at which oxygen is consumed and carbon dioxide generated depends on physical activity, and the ratio of carbohydrates, fats, and protein eaten.

It was once thought that the carbon dioxide content of the air from respiration was responsible for the condition of stale air experienced in places of concentrated occupancy.

Actually, the sense of staleness is primarily a result of the buildup of heat, moisture, and unpleasant odors given off by the body. While high carbon dioxide levels are responsible for headaches and loss of judgment, acute discomfort from odors, and health problems from other sources of air contamination contamination usually occurs long before the carbon dioxide concentration raises that high. The generally accepted safe limit is a 0.5 percent concentration for healthy, sedentary occupants eating a normal diet. This corresponds to 2.25 CFM (cubic feet per minute) of outdoor air per person where the outdoor air contains a normal proportion of carbon dioxide.

When ventilation air is brought in from outdoors, it may be by either mechanical (active) or natural (passive) means. In spaces with low-density occupancy and exterior walls, sufficient outside air may be introduced by leakage through doors and windows. Interior zones and heavily populated areas, however, require the introduction of ventilated air by mechanical equipment. Also, if the outside air needs to be conditioned, it should be passed through the conditioning equipment first and then delivered to the space. Whether ventilation air is brought in from outside or is predominantly recirculated, it must still be introduced at a rate sufficient to remove objectionable odors and contaminants from the space. With proper air distribution, the motion of the ventilation air blends with the room air, creating a unified thermal condition. And as the air gently passes by the occupants, it carries away heat, humidity, and odors given off by the body. This should result in a feeling of freshness.

8.9 Summary

The human body is essentially a constant-temperature device. Heat is continuously produced by bodily processes and dissipated in an automatically regulated manner to maintain the body temperature at its correct level despite variations in ambient conditions. In terms of physiology, the experience of comfort is the achievement of thermal equilibrium with the minimum amount of body regulation.

The human body normally rejects heat to the environment using evaporative cooling (sweating) and the heat transfer mechanisms of radiation, convection, and conduction.

The relative roles of these heat transfer mechanisms are determined by the individual's metabolism, clothing, and activity level, as well as by the surrounding environmental conditions of radiation, humidity, air temperature, and air motion. The acceptable value of each of these features is not fixed, but can vary in conjunction with one or more of the others. It is possible for the body to vary its own balance of losses, for example, through increased sweating; or the insulating value of the clothing worn can be varied to a limited degree to compensate for conditions beyond the body's ability to make its own adequate adjustment.

The comfort of a given individual is affected by many variables. Health, age, activity, clothing, gender, food, and acclimatization are all determining factors of the comfort conditions for any particular person. Since these factors will not be identical for all people, room conditions are provided under which a majority of the expected occupants will feel comfortable.

In addition to its thermal climate, the air quality of each indoor environment affects the sense of comfort. Air may contain a variety of possible contaminants that may or may not be harmful to human occupants. Along with possible toxicity, contaminants can impart odors to the space, and the toxicity and odor intensity are often related to the concentration of impurities. To reduce health hazards and eliminate objectionable odors, concentrations of impurities are controlled either by dilution from outside air ventilation or by treatment of the air in an air conditioning system, or by both. Proper fresh air distribution throughout a space is important for mixing the air in order to achieve acceptable overall quality, and for keeping the air steadily moving around the occupants to carry away heat, moisture, and odors generated by them.

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9

Load Analysis

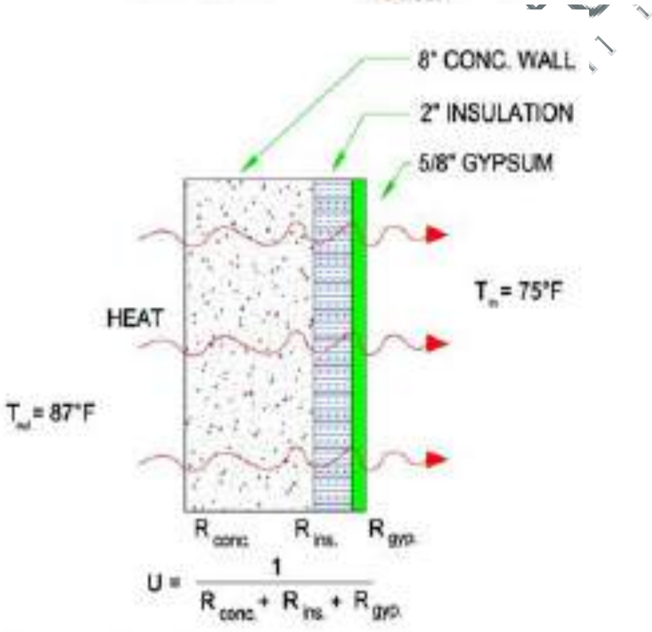


FIGURE 6: SIMPLIFIED HEAT GAIN THROUGH ROOF OR WALL

$$Q = U \times A \times (T_{outdoor} - T_{indoor})$$

Course Contents

9.1 Introduction

The load analysis is one of the most important steps in the design of an air conditioning system.

Historically, the air conditioning load calculation was a technique that started as a modification of heating load estimates prior to the emergence of buildings "designed" for air conditioning in the late 1940s and early 1950s. As the initial cooling load estimate techniques were developed, they simply addressed heat gain in place of heat loss. The winter outdoor design temperature was replaced with a summer outdoor design temperature. It was quickly recognized that the air conditioning system needed not only to have a capacity to remove heat but also to remove moisture, so the load estimates were expanded to perform this calculation.

Since the time of these early calculation methods modeled after the heating load estimates much change has transpired, not only in calculation technology but also in building technology. The increased knowledge gained from improved techniques in calculating air conditioning loads is in the process of leading the industry and designers of systems to a new paradigm in system design, if not building design. One of these changing paradigms will represent a return to regionalism in building designs based upon climatological characteristics. This move will result in significantly improved architectural design and enhanced building economics.

To calculate the cooling load needed to remove the amount of heat energy from a class room by the HVAC equipment to maintain the room at indoor design temperature when worst case outdoor design temperature is being experienced.

9.2 Terminology

Before we go further in cooling load calculations, we need to define and understand some important terminology.

Space: is either a volume or a site without a partition or a partitioned room or group of rooms. The load analysis is one of the most important steps in the design of an air conditioning system.

Zone: is a space or group of spaces within a building with heating and/or cooling requirements sufficiently similar so that comfort conditions can be maintained throughout by a single controlling device.

Cooling Load Temperature Difference (CLTD): an equivalent temperature difference used for calculating the instantaneous external cooling load across a wall or roof.

Sensible Heat Gain: is the energy added to the space by conduction, convection and/or radiation.

Latent Heat Gain: is the energy added to the space when moisture is added to the space by means of vapor emitted by the occupants, generated by a process or through air infiltration from outside or adjacent areas.

Radiant Heat Gain: the rate at which heat absorbed is by the surfaces enclosing the space and the objects within the space.

Space Heat Gain: is the rate at which heat enters into and/or is generated within the conditioned space during a given time interval.

Space Cooling Load: is the rate at which energy must be removed from a space to maintain a constant space air temperature.

Space Heat Extraction Rate: the rate at which heat is removed from the conditioned space and is equal to the space cooling load if the room temperature remains constant.

Dry Bulb Temperature: is the ordinary temperature of the air.

Wet Bulb Temperature: is the temperature of the air with consisting of moisture.

Dew Point Temperature: is the temperature to which air must be cooled in order to reach saturation or at which the condensation of water vapor in a space begins for a given state of humidity and pressure.

Relative humidity: describes how far the air is from saturation. It is a useful term for expressing the amount of water vapor when discussing the amount and rate of evaporation. One way to approach saturation, a relative humidity of 100%, is to cool the air. It is therefore useful to know how much the air needs to be cooled to reach saturation.

Relative humidity: is a term used to describe the amount of water vapor in a mixture of air and water vapor.

Space heat gain: the amount of heat is entering the space.

Space cooling load: the amount of energy must be removed from the space to keep temperature and relative humidity constant.

Space heat extraction: the amount of the energy that is removed by the HVAC.

Cooling load (coil): how much energy is removed by the cooling coil serving various spaces plus any loads external to the spaces such as duct heat gain, duct leakage, fan heat and outdoor makeup air.

9.3 The Method of How the Heat Enters the Space

- Solar radiation through the window or any transparent surfaces
- Heat conduction through walls, roof and windows of the class.
- Heat conduction through interior partitions, ceilings and floors.
- The generated heat by the occupants such as lights, appliances, equipment and

processes.

- The loads that are results of ventilation and infiltration of outdoor air.
- Other miscellaneous heat gains.

9.4 Sensible Heat Loads

It's about heat at which a substance absorbs. During rising the temperature of the substance, the substance doesn't change state. Sensible heat gain is directly added to the conditioned space by conduction, convection, and radiation. Note that the sensible heat gain entering the conditioned space does not equal the sensible cooling load during the same time interval because of the stored heat in the building envelope. Only the convective heat becomes cooling load instantaneously.

9.4.1 The factors involve the sensible heat load:

- Heat transmitted thru floors, ceilings, walls.
- Occupant's body heat.
- Appliance & Light heat.
- Solar Heat gain thru glass.
- Infiltration of outside air.
- Air introduced by Ventilation.

9.5 Latent Heat Loads

Latent heat gain occurs when moisture is added to the space either from internal sources (e.g. vapor emitted by occupants and equipment) or from outdoor air as a result of infiltration or ventilation to maintain proper indoor air quality.

9.5.1 The factors involve the Latent heat load:

- Moisture-laden outside air form Infiltration & Ventilation.
- Occupant Respiration & Activities.
- Moisture from Equipment & Appliances.

9.6 Space Heat Gain and Cooling Load (Heat Storage Effect)

The heat that is collected from the heat sources (conduction, convection, solar radiation, lightning, people, equipment, etc...) doesn't go directly to heating the room. However, only some part of the heat sources that is absorbed air in the conditioned space (class), leading to a quick change in its temperature. Most of the radiation heat especially from sun, lighting, people is first absorbed by the internal surfaces, which include ceiling, floor, internal walls, furniture etc. Due to the large but finite thermal capacity of the roof, floor, walls etc., their temperature increases slowly due to absorption of radiant heat. The radiant portion introduces a time lag and also a decrement factor depending upon the dynamic characteristics of the surfaces. Due to the time lag, the effect of radiation will be felt even when the source of radiation, in this case the sun is removed.

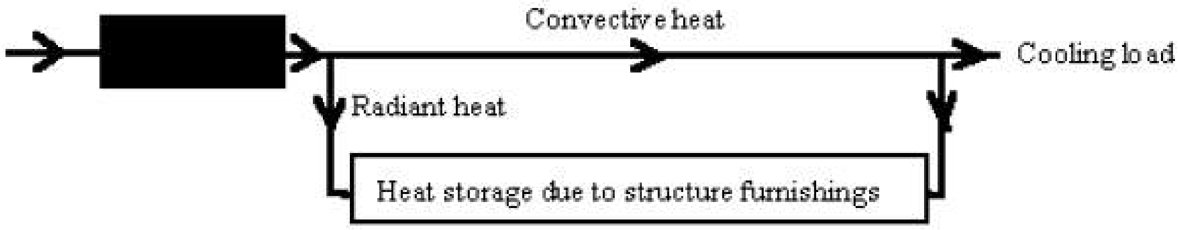


Fig. 9.1 shows differences between Space Heat Gain and Space Cooling Load.

As you can see from **Fig.9.1** that shows difference between immediate heat gain and cooling load is due to heat storage affect.

The relation between heat gain and cooling load and the effect of the mass of the structure (light, medium & heavy) is shown below. From figure (2) it is evident that, there is a delay in the peak heat, especially for heavy construction.

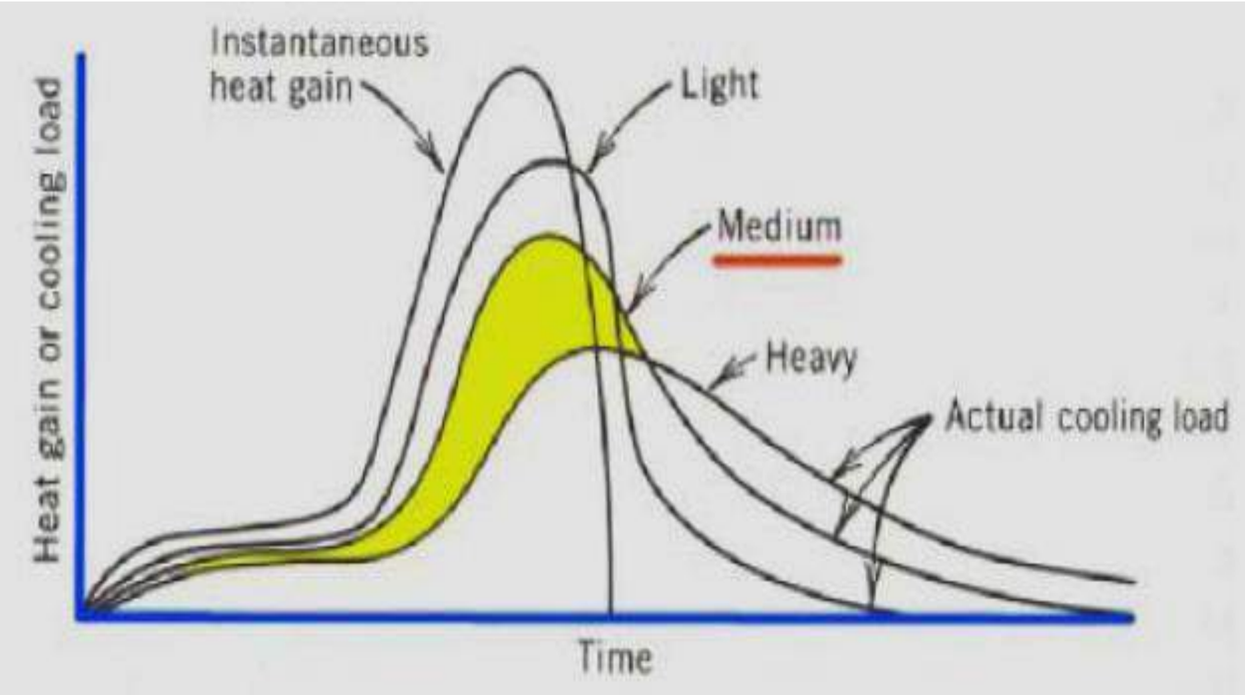


Fig. 9.2 shows actual cooling load and solar heat gain for light, medium and heavy construction.

9.7 Space Cooling and Cooling Load (Coil)

Space cooling is the rate at which heat must be removed from the spaces to maintain air temperature at a constant value. Cooling load, on the other hand, is the rate at which energy is removed at the cooling coil that serves one or more conditioned spaces in any central air conditioning system. It is equal to the instantaneous sum of the space cooling loads for all spaces served by the system plus any additional load imposed on the system external to the conditioned spaces items such as fan energy, fan location, duct heat gain, duct leakage, heat extraction lighting systems and type of return air systems all affect component sizing.

9.8 Components of Cooling Load

The total class room cooling load consists of heat transferred through the class room envelope (walls, roof, floor, windows, doors etc.) and heat generated by occupants, equipment, and lights. The load due to heat transfer through the envelope is called as external load, while all other loads are called as internal loads. The percentage of external versus internal load varies with class room type, site climate, and class room design. The total cooling load on any building consists of both sensible as well as latent load components. The sensible load affects the dry bulb temperature, while the latent load affects the moisture content of the conditioned space.

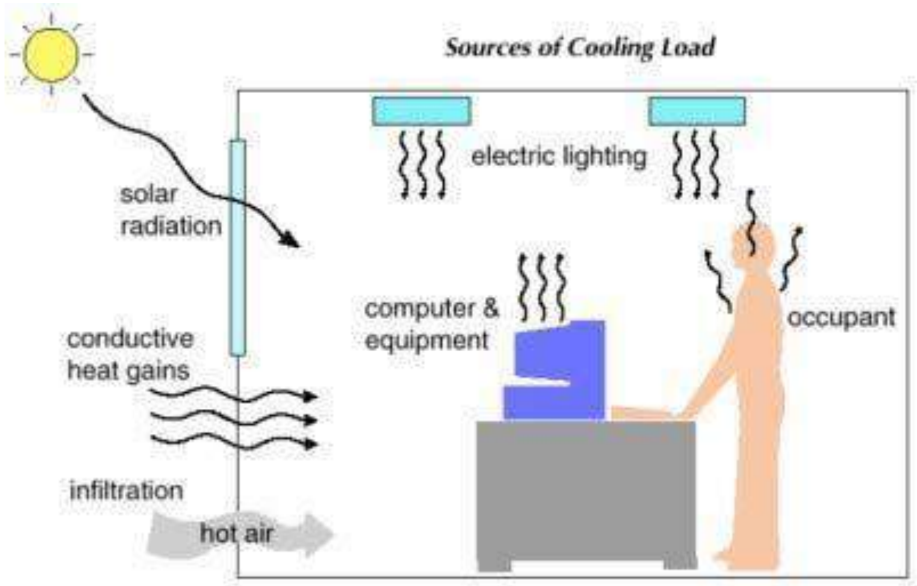


Fig. 9.3 *shows the external loads and internal loads.*

Class room may be classified as externally loaded and internally loaded as you can see from **Fig.9.3**. In externally loaded class room, the cooling load on the class room is mainly due to heat transfer between the surroundings and the internal conditioned space. Since the surrounding conditions are highly variable in any given day, the cooling load of an externally loaded building varies widely. In internally loaded class room, the cooling load is mainly due to internal heat generating sources such as occupants, lights or appliances. In general the heat generation due to internal heat sources may remain fairly constant, and since the heat transfer from the variable surroundings is much less compared to the internal heat sources, the cooling load of an internally loaded class room remains fairly constant. Obviously from energy efficiency and economics points of view, the system design strategy for an externally loaded class room should be different from an internally loaded building. Hence, prior knowledge of whether the building is externally loaded or internally loaded is essential for effective system design.

9.9 CLTD/SCL/CLF Methods of Load Calculation

CLTD is a theoretical temperature difference that accounts for the combined effects of inside and outside air temp difference, daily temp range, solar radiation and heat storage in the construction assembly/building mass. It is affected by orientation, tilt, month, day, hour, latitude, etc. CLTD factors are used for adjustment to conductive heat gains from walls, roof, floor and glass.

CLF accounts for the fact that all the radiant energy that enters the conditioned space at a particular time does not become a part of the cooling load instantly. The CLF values for various surfaces have been calculated as functions of solar time and orientation and are available in the form of tables in ASHRAE Handbooks. CLF factors are used for adjustment to heat gains from internal loads such as lights, occupancy, power appliances.

SCL factors are used for adjustment to transmission heat gains from glass.

9.10 External Cooling Load

9.10.1 Roof

If the roof is exposed directly to the sun, it absorbs maximum heat. If there is other room above the air-conditioned room, then the amount of heat gained by the roof reduces. The heat gained by the partitions of the room depends upon the type of partition. Roof calculation formula is given below:

$$Q = U \times A \times (CLTD)$$

Where,

Q = cooling load.

U = Coefficient of heat transfer roof or wall or glass.

A = area of roof.

CLTD = cooling load temperature difference.

Since the ASHRAE tables provide hourly CLTD values for one typical set of conditions i.e. outdoor maximum temperature of 95°F with mean temperature of 85°C and daily range of 21°F, the equation is further adjusted to apply correction factors for conditions other than the mentioned base case. Thus,

$$Q_{\text{Roof}} = U \times A \times CLTD_{\text{Roof Corrected}}$$

The typical steps to calculate the Roof load are as follows

1. Determine roof construction and overall heat transfer coefficient (U)
2. Select roof no. from ASHRAE Table which is closest to matching actual roof construction.

3. Select CLTD Roof for time of interest, typically on an hourly basis
4. Corrections: CLTD $_{\text{Roof Corrected}}$= [CLTD $_{\text{Roof}}$+ (25.5 – TR) +(TM – 29.4)]

   Where
   - (25.5 – TR) = indoor design temperature correction
   - (TM – 29.4) = outdoor design temperature correction
   - TR = Indoor room temperature
   - TM = Mean outdoor temperature
   - T_{max} = Maximum outdoor temperature
   - TM = T_{max} – (Daily Range) / 2
5. Calculate roof area (A)
6. $Q_{\text{Roof}} = U \times A \times CLTD_{\text{Roof Corrected}}$

9.10.2 Walls

The walls of the room gain heat from the sun by way of conduction. The amount of heat depends on the wall material and its alignment with respect to sun. If the wall of the room is exposed to the west direction, it will gain maximum heat between 2 to 5 pm. The southern wall will gain maximum heat in the mid-day between 12 to 2 pm. The heat gained by the wall facing north direction is the least. The heat gained by the walls in day-time gets stored in them, and it is released into the rooms at the night time thus causing excessive heating of the room. If the walls of the room are insulated the amount of heat gained by them reduces drastically.

The cooling load from walls is treated in a similar way as roof:

$$Q_{\text{Wall}} = U \times A \times CLTD_{\text{Wall Corrected}}$$

Where,

Q_{Wall} = Load through the walls.

U = Thermal Transmittance for walls.

A = area of walls.

CLTD = Cooling Load Temperature Difference for walls.

The typical steps to calculate the Wall load are as follows:

1. Determine wall construction and overall heat transfer coefficient (U)
2. Select wall no. from ASHRAE Table which is closest to matching actual wall construction.
3. Select CLTD Roof for time of interest, typically on an hourly basis
4. Corrections: CLTD $_{\text{Wall Corrected}}$= [CLTD $_{\text{Wall}}$+ (25.5 – TR) +(TM – 29.4)]

   Where,
   - (25.5 – TR) = indoor design temperature correction
   - (TM – 29.4) = outdoor design temperature correction
   - TR = Indoor room temperature

- TM = Mean outdoor temperature
- T_{max} = Maximum outdoor temperature
- TM = T_{max} – (Daily Range) / 2

5. Calculate roof area (A)
6. $Q_{Wall} = U \times A \times CLTD_{Wall\ Corrected}$

9.10.3 Solar load through glass

Solar load through glass has two components: 1) Conductive and 2) Solar Transmission. The absorbed and then conductive portion of the radiation through the windows is treated like the roof & walls where CLTD values for standard glazing are tabulated in ASHARE fundamentals handbook. For solar transmission, the cooling load is calculated by the cooling load SCL factor and shading coefficient (SC).

The cooling load equations for glass are:

$$\text{Conductive } Q_{Glass\ Conductive} = U \times A \times CLTD_{Glass\ Corrected}$$

$$\text{Solar Transmission } Q_{Glass\ Solar} = A \times SC \times SCL$$

Where

$Q_{Conductive}$ = Conductive load through the glass.

Q_{Solar} = Solar transmission load through the glass.

U = Thermal Transmittance for glass.

A = area of glass.

CLTD = Cooling Load Temperature Difference for glass.

SC = Shading coefficient.

SCL = Solar Cooling Load Factor.

Steps for conductive calculations

1. For the glass types used, select from ASHRAE tables the overall heat transfer coefficient (U).
2. Select CLTD Glass for time of interest, typically on an hourly basis.
3. Corrections: CLTD Glass Corrected= [CLTD Glass+ (25.5 – TR) +(TM – 29.4)].
4. Calculate glass area (A).
5. $Q_{Glass} = U \times A \times CLTD_{Glass\ Corrected}$

Steps for solar transmission calculations:

1. Determine shading coefficient (SC)
2. Determine zone type
3. Determine solar cooling load factor (SCL)

4. Calculate glass area (A).
5. $Q_{Glass\ Solar} = A * SC * SCL$

9.10.4 Partition, ceilings and floors

The various internal loads consist of sensible and latent heat transfers due to occupants, products, processes appliances and lighting. The lighting load is only sensible. The conversion of sensible heat gain (from lighting, people, appliances, etc.) to space cooling load is affected by the thermal storage characteristics of that space and is thus subject to appropriate cooling load factors (CLF) to account for the time lag of the cooling load caused by the building mass. The weighting factors equation determines the CLF factors. CLF = Q cooling load / Q internal gains.

Note that the latent heat gains are considered instantaneous.

9.11 Internal Cooling Loads

The various internal loads consist of sensible and latent heat transfers due to occupants, products, processes appliances and lighting. The lighting load is only sensible. The conversion of sensible heat gain (from lighting, people, appliances, etc.) to space cooling load is affected by the thermal storage characteristics of that space and is thus subject to appropriate cooling load factors (CLF) to account for the time lag of the cooling load caused by the building mass. The weighting factors equation determines the CLF factors.

$$CLF = Q_{cooling\ load} / Q_{internal\ gains}$$

Note that the latent heat gains are considered instantaneous.

3.11.1 People

Human beings release both sensible heat and latent heat to the conditioned space when they stay in it as you can see in the Table 1. You might have noticed that when a small room is filled with people, it tends to become warmer. People emit heat primarily through breathing and perspiration, and, to a lesser extent, through radiation. This heat translates into an increased cooling load on your cooling systems. The heat gain by the occupants in the building is separated into sensible and latent heat. The number of people, the type of activity they are performing, and the CLF determines sensible and latent heat. The CLF is determined by the time the occupants come into the building and for how long they stay in the building.

Table 1 Heat gain from occupants at various activities

Activity	Total heat, Btu/h		Sensible heat, Btu/h	Latent heat, Btu/h
	Adult, male	Adjusted		
Seated at rest	400	350	210	140
Seated, very light work, writing	480	420	230	190
Seated, eating	520	580	255	325
Seated, light work, typing,	640	510	255	255
Standing, light work or walking	800	640	315	325
slowly,	880	780	345	435
Light bench work	1040	1040	345	695
Light machine work, walking 3mi/hr	1360	1280	405	875
Moderate dancing	1600	1600	565	1035

The heat gain from the occupancy or people is given be equation:

$$Q_{\text{sensible}} = N\,(QS)\,(CLF)$$

$$Q_{\text{latent}} = N\,(QL)$$

Where

N = number of students in space (class room).

QS, QL = Sensible and Latent heat gain from occupancy is given in Table 1.

CLF = Cooling Load Factor, by hour of occupancy.

9.11.2 Lights

The primary source of heat from lighting comes from light-emitting elements. **Fig.9.4**, indicate and explain more about the lights effects on the heat gain when it's off or on. Calculation of this load component is not straightforward; the rate of heat gain at any given moment can be quite different from the heat equivalent of power supplied instantaneously to those lights. Only part of the energy from lights is in the form of convective heat, which is picked up instantaneously by the air-conditioning apparatus. The remaining portion is in the form of radiation, which affects the conditioned space only after having been absorbed and re-released by walls, floors, furniture, etc. This absorbed energy contributes to space cooling load only after a time lag, with some part of such energy still present and reradiating after the lights have been switched off.

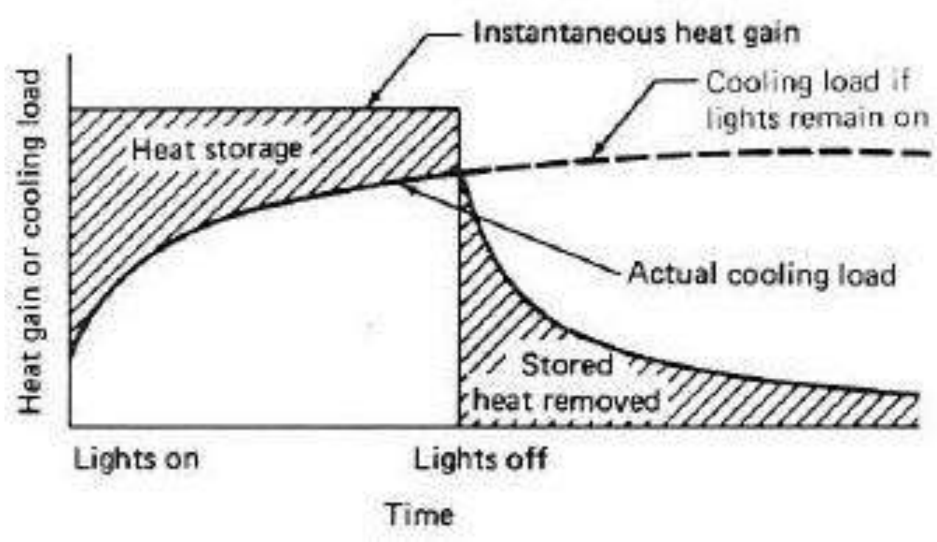


Fig.9.4 *shows actual cooling load from fluorescent lights.*

Generally, the instantaneous rate of heat gain from electric lighting may be calculated from:

$$Q = 3.41 \times W \times F_{UT} \times F_{SA}$$

Cooling load factors are used to convert instantaneous heat gain from lighting to the sensible cooling load; thus the equation is modified to:

$$Q = 3.41 \times W \times F_{UT} \times F_{SA} \times (CLF).$$

Where,

W = Watts input from electrical lighting plan or lighting load data.

F_{UT} = Lighting use factor, as appropriate.

F_{SA} = special ballast allowance factor, as appropriate.

CLF = Cooling Load Factor, by hour of occupancy.

9.11.3 Appliances

In a cooling load estimate, heat gain from all appliances-electrical, gas, or steam-should be taken into account. Because of the variety of appliances, applications, schedules, use, and installations, estimates can be very subjective. Often, the only information available about heat gain from equipment is that on its name-plate.

$$Q_{Sensible} = Q_{in} \text{ x } F_u \text{ x } F_r \text{ x } (CLF)$$

$$Q_{Latent} = Q_{in} \text{ x } F_u$$

Where,

Q_{in} = rated energy input from appliances. Use manufacturer's data. For computers, monitors, printers and miscellaneous office equipment, see 2001 ASHRAE Fundamentals.

F_u = Usage factor. See 1997 ASHRAE Fundamentals.

F_r = Radiation factor. See 1997 ASHRAE Fundamentals.

CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals,

9.11.4 Infiltration Air

$$Q_{sensible} = 1.08 \text{ x CFM x } (T_o - T_i)$$

$$Q_{latent} = 4840 \text{ x CFM x } (W_o - W_i)$$

$$Q_{total} = 4.5 \text{ x CFM x } (h_o - h_i)$$

Where,

CFM = Infiltration air flow rate.

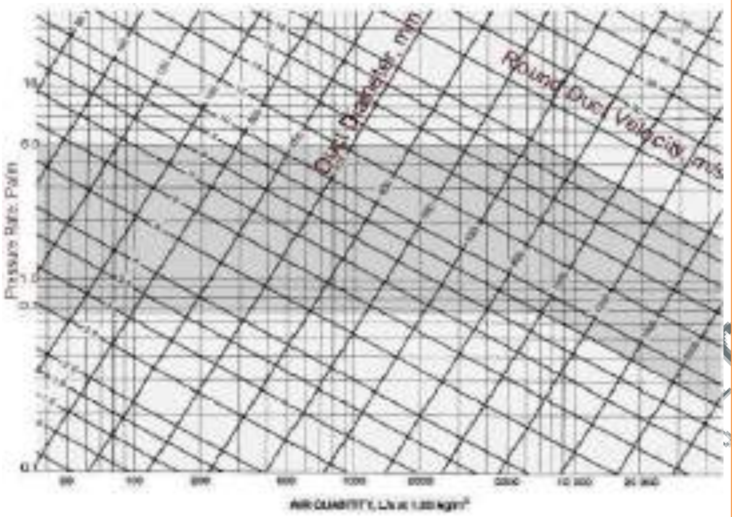
T_o, T_i = Outside/Inside dry bulb temperature.

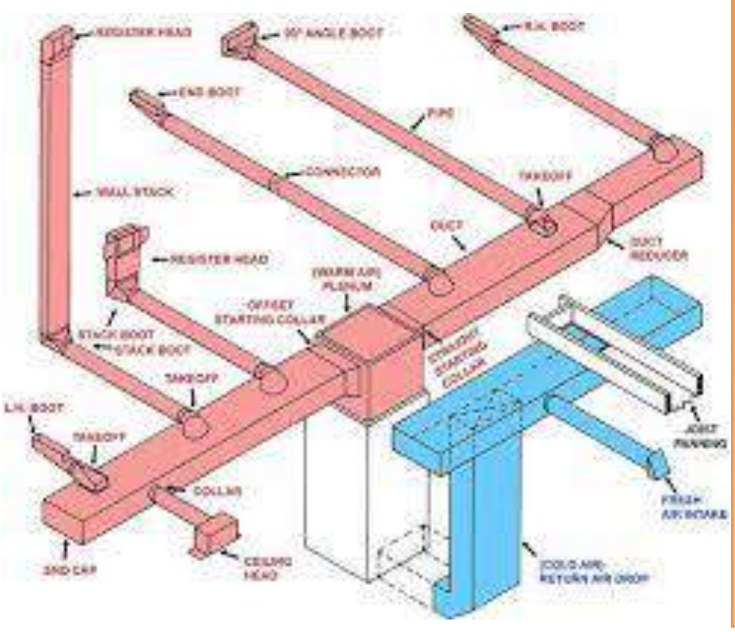
W_o, W_i = Outside/Inside humidity ratio.

h_o, h_i = Outside/Inside air enthalpy.

10

Duct Design





Course Contents

10.1 Introduction

The chief requirements of an air conditioning duct system are:

1. It should convey specified rates of air flow to prescribed locations.
2. It should be economical in combined initial cost, fan operating cost and cost of building space.
3. It should not transmit or generate objectionable noise.

Generally at the time of designing an air conditioning duct system, the required airflow rates are known from load calculations. The location of fans and air outlets are fixed initially. The duct layout is then made taking into account the space available and ease of construction. In principle, required amount of air can be conveyed through the air conditioning ducts by a number of combinations. However, for a given system, only one set results in the optimum design. Hence, it is essential to identify the relevant design parameters and then optimize the design.

10.2 General Rules for Duct Design

1. Air should be conveyed as directly as possible to save space, power and material.
2. Sudden changes in directions should be avoided. When not possible to avoid sudden changes, turning vanes should be used to reduce pressure loss.
3. Diverging sections should be gradual. Angle of divergence $\leq 20^{o}$
4. Aspect ratio should be as close to 1.0 as possible. Normally, it should not exceed 4.
5. Air velocities should be within permissible limits to reduce noise and vibration.
6. Duct material should be as smooth as possible to reduce frictional losses

10.3 Classification of Duct Systems

Ducts are classified based on the load on duct due to air pressure and turbulence. The classification varies from application to application, such as for residences, commercial systems, industrial systems etc. For example, one such classification is given below:

Low pressure systems: Velocity $\leq$ 10 m/s, static pressure $\leq$ 5 cm H_2O (g)

Medium pressure systems: Velocity $\leq$ 10 m/s, static pressure $\leq$ 15 cm H_2O (g)

High pressure systems: Velocity > 10 m/s, static pressure $15<p \leq 25$ cm H_2O (g)

High velocities in the ducts results in:

1. Smaller ducts and hence, lower initial cost and lower space requirement.
2. Higher pressure drop and hence larger fan power consumption.

3. Increased noise and hence a need for noise attenuation.

Recommended air velocities depend mainly on the application and the noise criteria. Typical recommended velocities are:

Residences: 3 m/s to 5 m/s

Theatres: 4 to 6.5 m/s

Restaurants: 7.5 m/s to 10 m/s

If nothing is specified, then a velocity of 5 to 8 m/s is used for main ducts and a velocity of 4 to 6 m/s is used for the branches. The allowable air velocities can be as high as 30 m/s in ships and aircrafts to reduce the space requirement.

4.4 Duct Material

The three most common types of duct material used in home construction are metal, rigid fiberglass duct board, and flex-duct. Both metal and fiberglass duct board are rigid and installed in pieces, while flex-duct comes in long sections. Flex-duct is usually installed in a single, continuous piece between the register and plenum box or plenum box and air handler. Be careful not to tear the soft lining material. The flex-duct must also not be pinched or constricted. Long flex-duct runs can restrict air flow, so they must be installed carefully. Flex-duct takeoffs, while often airtight in appearance, can have substantial leakage and should be sealed with mastic. Always select duct insulation with a shiny, metal foil exterior covering to reduce radiant heat gain and to act as a vapor barrier.

Round and rectangular metal duct must be sealed with mastic and insulated during installation. It is important to seal the seams and joints first, because the insulation does not stop air leaks. Metal ducts often use fiberglass insulation having an attached metal foil vapor barrier. The duct insulation should be at least R-6, and the vapor barrier should be installed to the outside of the insulation - facing away from the duct. The seams in the insulation are usually stapled together around the duct and then taped. Duct insulation in homes at least two-years old provides visible clues about duct leakage – if the insulation is removed, lines of dirt in the fiberglass often show where air leakage has occurred. Sometimes, rectangular metal duct used for plenums and larger trunk duct runs is insulated internally with duct liner, a high density material that should be at least 1-inch thick. Many homeowners have concerns about the long term effects of the duct liner exposed to the air flowing through the system. They prefer to insulate the outside of the ductwork, rather than the inside. Internally insulated metal ducts cannot be cleaned as easily as externally insulated ducts. For acoustical dampening in transfer ducts (from room to room or hall) the internal lining is preferred.

10.5 Commonly Used Duct Design Methods

Fig.10.1 shows the schematic of a typical supply air duct layout. As shown in the figure, supply air from the fan is distributed to five outlets (1 to 5), which are located in five different conditioned zones. The letters A to I denote the portions of the duct to different outlets. Thus A-B is the duct running from the supply air fan to zone 1, A-B-C is the duct running from supply fan to conditioned zone and so on. These are known as duct runs. The run with the highest pressure drop is called as the index run. From load and psychrometric calculations the required supply airflow rates to each conditioned space are known. From the building layout and the location of the supply fan, the length of each duct run is known. The purpose of the duct design is to select suitable dimensions of duct for each run and then to select a fan, which can provide the required supply airflow rate to each conditioned zone.

Due to the several issues involved, the design of an air conditioning duct system in large buildings could be a sophisticated operation requiring the use of Computer Aided Design (CAD) software. However, the following methods are most commonly used for simpler lay-outs such as the one shown in **Fig.10.1**.

1. Velocity method

2. Equal Friction Method

3. Static Regain method FAN

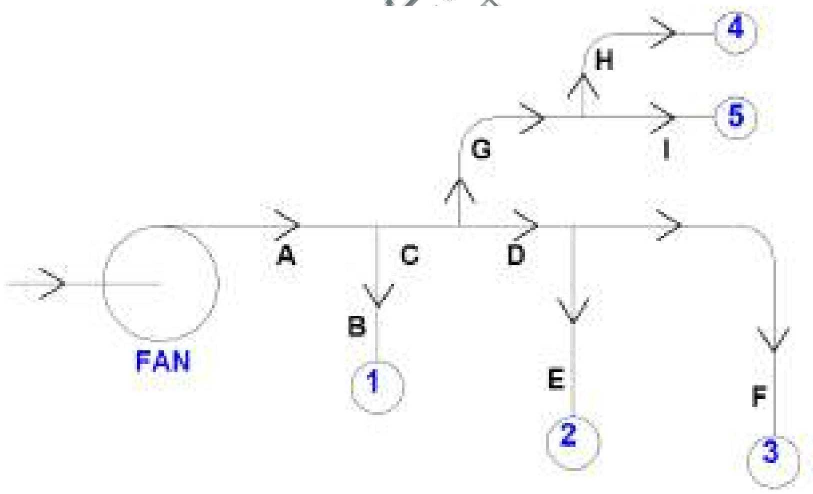


Fig.10.1*: Typical air conditioning duct lay-out*

10.5.1 Velocity method

The various steps involved in this method are:

1. Select suitable velocities in the main and branch ducts.

2. Find the diameters of main and branch ducts from airflow rates and velocities for circular ducts. For rectangular ducts, find the cross-sectional area from flow rate and velocity, and then by fixing the aspect ratio, find the two sides of the rectangular duct.
3. From the velocities and duct dimensions obtained in the previous step, find the frictional pressure drop for main and branch ducts using friction chart or equation.
4. From the duct layout, dimensions and airflow rates, find the dynamic pressure losses for all the bends and fittings.
5. Select a fan that can provide sufficient FTP for the index run.
6. Balancing dampers have to be installed in each run. The damper in the index run is left completely open, while the other dampers are throttled to reduce the flow rate to the required design values.

The velocity method is one of the simplest ways of designing the duct system for both supply and return air. However, the application of this method requires selection of suitable velocities in different duct runs, which requires experience. Wrong selection of velocities can lead to very large ducts, which, occupy large building space and increases the cost, or very small ducts which lead to large pressure drop and hence necessitates the selection of a large fan leading to higher fan cost and running cost. In addition, the method is not very efficient as it requires partial closing of all the dampers except the one in the index run, so that the total pressure drop in each run will be same.

For example, let the duct run A-C-G-H be the index run and the total pressure drop in the index run is 100 Pa. If the pressure drop in the shortest duct run (say A-B) is 10 Pa, then the damper in this run has to be closed to provide an additional pressure drop of 90 Pa, so that the required airflow rate to the conditioned zone 1 can be maintained. Similarly the dampers in the other duct runs also have to be closed partially, so that the total pressure drop with damper partially closed in each run will be equal to the pressure drop in the index run with its damper left open fully.

10.5.2 Equal friction method

In this method the frictional pressure drop per unit length in the main and branch ducts ($\Delta p_f/L$) are kept same, i.e.,

$$\left(\frac{\Delta p_f}{L}\right)_A = \left(\frac{\Delta p_f}{L}\right)_B = \left(\frac{\Delta p_f}{L}\right)_C = \left(\frac{\Delta p_f}{L}\right)_D = \ldots\ldots$$

Then the stepwise procedure for designing the duct system is as follows:

Select a suitable frictional pressure drop per unit length ($\Delta p_f/L$) so that the combined initial and running costs are minimized.

Then the equivalent diameter of the main duct (A) is obtained from the selected value of ($\Delta p_f/L$) and the airflow rate. As shown in Fig.1, airflow rate in the main duct Q_A is equal to the sum total of airflow rates to all the conditioned zones, i.e.,

$$Q_A = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 = \sum_{i=1}^{n} Q_i$$

From the airflow rate and ($\Delta p_f/L$) the equivalent diameter of the main duct (D_{eq},A) can be obtained either from the friction chart or using the frictional pressure drop equation, i.e.,

$$D_{eq,A} = \left(\frac{0.022243 Q_A^{1.852}}{\left(\frac{\Delta p_f}{L}\right)_A} \right)^{\left(\frac{1}{4.973}\right)}$$

Since the frictional pressure drop per unit length is same for all the duct runs, the equivalent diameters of the other duct runs, B to I are obtained from the equation:

$$\left(\frac{Q^{1.852}}{D^{4.973}}\right)_A = \left(\frac{Q^{1.852}}{D^{4.973}}\right)_B = \left(\frac{Q^{1.852}}{D^{4.973}}\right)_C = \ldots\ldots\ldots$$

If the ducts are rectangular, then the two sides of the rectangular duct of each run are obtained from the equivalent diameter of that run and by fixing aspect ratio as explained earlier. Thus the dimensions of the all the duct runs can be obtained. The velocity of air through each duct is obtained from the volumetric flow rate and the cross-sectional area.

Next from the dimensions of the ducts in each run, the total frictional pressure drop of that run is obtained by multiplying the frictional pressure drop per unit length and the length, i.e.,

$$\Delta P_{f,A} = \left(\frac{\Delta p_f}{L}\right)_A .L_A ; \Delta P_{f,B} = \left(\frac{\Delta p_f}{L}\right)_B .L_B$$

Next the dynamic pressure losses in each duct run are obtained based on the type of bends or fittings used in that run.

Next the total pressure drop in each duct run is obtained by summing up the frictional and dynamic losses of that run, i.e.,

$$\Delta P_A = \Delta p_{f,A} + \Delta p_{d,A} ; \Delta P_B = \Delta p_{f,B} + \Delta p_{d,B}$$

Next the fan is selected to suit the index run with the highest pressure loss. Dampers are installed in all the duct runs to balance the total pressure loss.

Equal friction method is simple and is most widely used conventional method. This method usually yields a better design than the velocity method as most of the available pressure drop is dissipated as friction in the duct runs, rather than in the balancing dampers. This method is generally suitable when the ducts are not too long, and it can be used for both supply and return ducts. However, similar to velocity method, the equal friction method also requires partial closure of dampers in all but the index run, which may generate noise. If the ducts are too long then the total pressure drop will be high and due to dampering, ducts near the fan get over-pressurized.

10.5.3. Static Regain Method

This method is commonly used for high velocity systems with long duct runs, especially in large systems. In this method the static pressure is maintained same before each terminal or branch. The procedure followed is as given below:

Velocity in the main duct leaving the fan is selected first.

Velocities in each successive runs are reduced such that the gain in static pressure due to reduction in velocity pressure equals the frictional pressure drop in the next duct section. Thus the static pressure before each terminal or branch is maintained constant. For example, **Fig.10.2** shows a part of the duct run with two sections 1 and 2 before two branch take-offs. The velocity at 1 is greater than that at 2, such that the static pressure is same at 1 and 2. Then using the static regain factor, one can write:

$$\Delta p_{f,2} + \Delta p_{d,2} = R(p_{v,1} - p_{v,2})$$

where $\Delta p_{f,2}$ and $\Delta p_{d,2}$ are the frictional and dynamic losses between 1 and 2, and $p_{v,1}$ and $p_{v,2}$ are the velocity pressures at 1 and 2 respectively.

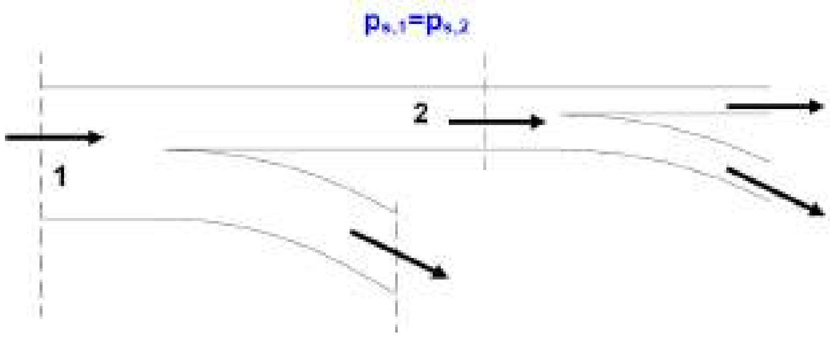


Fig.10.2 *Principle of Static Regain Method*

If section 1 is the outlet of the fan, then its dimensions are known from the flow rate and velocity (initially selected), however, since both the dimensions and velocity at section 2 are not known, a trial-and-error method has to be followed to solve the above equation, which gives required dimensions of the section at 2.

The procedure is followed in the direction of airflow, and the dimensions of the downstream ducts are obtained.

As before, the total pressure drop is obtained from the pressure drop in the longest run and a fan is accordingly selected.

Static Regain method yields a more balanced system and does not call for unnecessary dampering. However, as velocity reduces in the direction of airflow, the duct size may increase in the airflow direction. Also the velocity at the exit of the longer duct runs may become too small for proper air distribution in the conditioned space.

4.6 Performance of Duct Systems

For the duct system with air in turbulent flow, the total pressure loss (ΔP_t) is proportional to the square of flow rate; i.e.,

Total Pressure drop, $\Delta P_t \propto Q^2$

Or Total Pressure drop, $\Delta P_t = C \cdot Q^2$

Where C is the resistance offered by the duct system. Once the duct system is designed and installed, the value of C is supposed to remain constant. However, if the air filters installed in the duct become dirty and/or if the damper position is altered, then the value of C changes. Thus variation of total pressure drop with airflow rate is parabolic in nature as shown in **Fig.10.3**. In this figure, the curve A refers to the performance of the duct at design conditions, while curve B refers to the performance under the conditions of a dirty filter and/or a higher damper closure and curve C refers to the performance when the damper is opened more.

From the duct characteristic curve for constant resistance, one can write,

$$\frac{\Delta p_{t,1}}{\Delta p_{t,2}} = \frac{(Q_1)^2}{(Q_2)^2}$$

Thus knowing the total pressure drop and airflow rate at design condition (say 1), one can obtain the total pressure drop at an off-design condition 2, using the above equation.

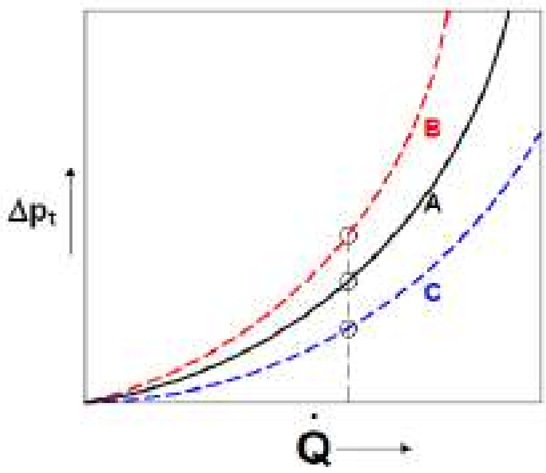


Fig.10.3: *Variation of total pressure drop with flow rate for a given duct system*

10.7 System Balancing and Optimization

In large buildings, after the Air Handling Unit is installed, it has to be balanced for satisfactory performance. System balancing requires as a first step, measurements of actual airflow rates at all supply air outlets and return air inlets. Then the dampers are adjusted so that the actual measured flow rate corresponds to the specified flow rates. System balancing may also require adjusting the fan speed to get required temperature drop across the cooling or heating coils and required airflow rates in the conditioned zone. Balancing a large air conditioning system can be a very expensive and time consuming method and may require very accurate instruments for measuring air flow rates and temperatures. However, system balancing is always recommended to get the full benefit from the total cost incurred on air conditioning system.

Large air conditioning systems require optimization of the duct design so as to minimize the total cost, which includes the initial cost of the system and the lifetime operating cost. At present very sophisticated commercial computer software are available for optimizing the duct design. One such method is called as T-Method. The reader should refer to advanced textbooks or ASHRAE handbooks for details on duct optimization methods.

10.8 Fans

The fan is an essential and one of the most important components of almost all air conditioning systems. Thus a basic understanding of fan performance characteristics is essential in the design of air conditioning systems. The centrifugal fan is most commonly used in air conditioning systems as it can efficiently move large quantities of air over a large range of pressures. The operating principle of a centrifugal fan is similar to that of a centrifugal compressor discussed earlier. The centrifugal fan with forward-curved blades is widely used in low-pressure air conditioning systems. The more efficient backward-curved and airfoil type fans are used in large capacity, high-pressure systems.

10.9 Fan Laws

The fan laws are a group of relations that are used to predict the effect of change of operating parameters of the fan on its performance. The fan laws are valid for fans, which are geometrically and dynamically similar. The fan laws have great practical use, as it is not economically feasible to test fans of all sizes under all possible conditions.

The important operating parameters of a fan of fixed diameter are:

1. Density of air (ρ) which depends on its temperature and pressure.
2. Operating speed of the fan (ω in rps), and
3. Size of the fan.

Here the fan laws related to the density of air and the rotative speed of the fan are considered. The effect of the size of the fan is important at the time of designing the fan. For a given air conditioning system with fixed dimensions, fittings etc. it can be easily shown that:

Air flow rate, $Q \propto \omega$

Static pressure rise, $\Delta p_s \propto \frac{\rho V^2}{2}$

Fan power input, $W \propto Q(\Delta p_s) + Q\left(\frac{\rho V^2}{2}\right)$

From the expression for fan power input, it can be seen that the 1st term on the RHS accounts for power input required for increasing the static pressure of air and the 2nd term on RHS accounts for the power input required to impart kinetic energy to air as it flows through the fan. Using the above relations, the following fan laws can be obtained.

Law 1: Density of air ρ remains constant and the speed ω varies:

$$Q \propto \omega \ ; \ \Delta p_s \propto \omega^2 \text{ and } W \propto \omega^3$$

Law 2: Airflow rate remains constant and the density ρ varies:

$$Q = \text{Constant} \ ; \ \Delta p_s \propto \rho \text{ and } W \propto \rho$$

Law 3: Static pressure rise Δp_s remains constant and density ρ varies:

$$Q \propto \frac{1}{\sqrt{\rho}} \ ; \ \Delta p_s = \text{Constant} \ ; \ \omega \propto \frac{1}{\sqrt{\rho}} \text{ and } W \propto \frac{1}{\sqrt{\rho}}$$

10.10 Interaction between Fan and Duct System

Fig.10.4 shows the variation of FTP of a centrifugal fan (fan performance curve) and variation of total pressure loss of a duct system (duct performance curve) as functions of the airflow rate. As shown in the figure, the point of intersection of the fan performance curve and the duct performance curve yield the balance point for the combined performance of fan and duct system. Point 1 gives a balance point between the fan and duct system when the rotative speed of fan is ω_1.

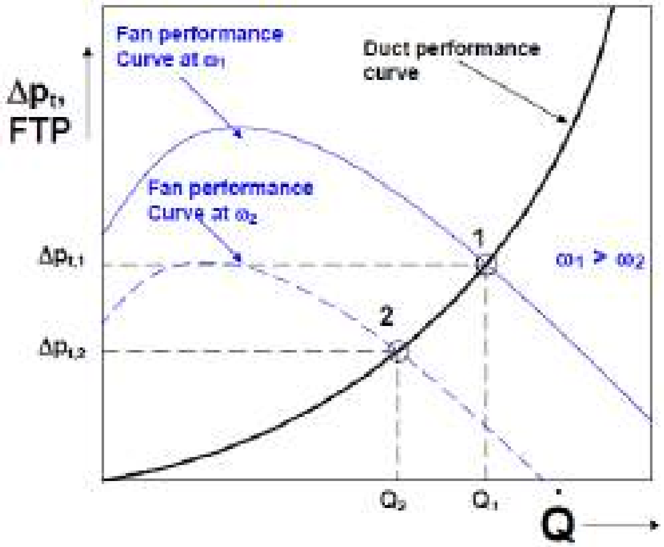
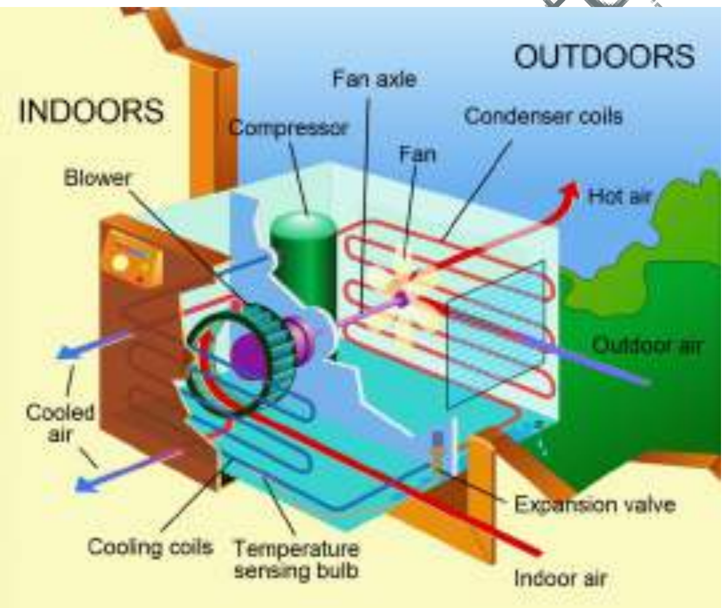


Fig.10.4 : *Fan and Duct Performance Curve and Balance Points*

At this condition the airflow rate is Q and the total pressure loss which is equal to the FTP is $\Delta p_{t,1}$. Now if the flow rate is reduced to Q, then the total pressure loss reduces to $\Delta p_{t,2}$. To match the reduced flow rate and the reduced pressure loss, the speed of the fan has to be reduced to ω_2 or the position of the inlet guide vanes of the centrifugal fan have to be adjusted to reduce the flow rate. This will give rise to a new balance point at 2. Thus the fan and duct system have to be matched when there is a change in the operating conditions.

11

Air Conditioning Systems



Course Contents

11.1 Introduction

In order to maintain required conditions inside the conditioned space, energy has to be either supplied or extracted from the conditioned space. The energy in the form of sensible as well as latent heat has to be supplied to the space in winter and extracted from the conditioned space in case of summer. An air conditioning system consists of an air conditioning plant and a thermal distribution system as shown in **Fig.11.1**. As shown in the figure, the air conditioning (A/C) plant acts either as a heat source (in case of winter systems) or as a heat sink (in case of summer systems). Air, water or refrigerant are used as media for transferring energy from the air conditioning plant to the conditioned space. A thermal distribution system is required to circulate the media between the conditioned space and the A/C plant. Another important function of the thermal distribution system is to introduce the required amount of fresh air into the conditioned space so that the required Indoor Air Quality (IAQ) can be maintained.

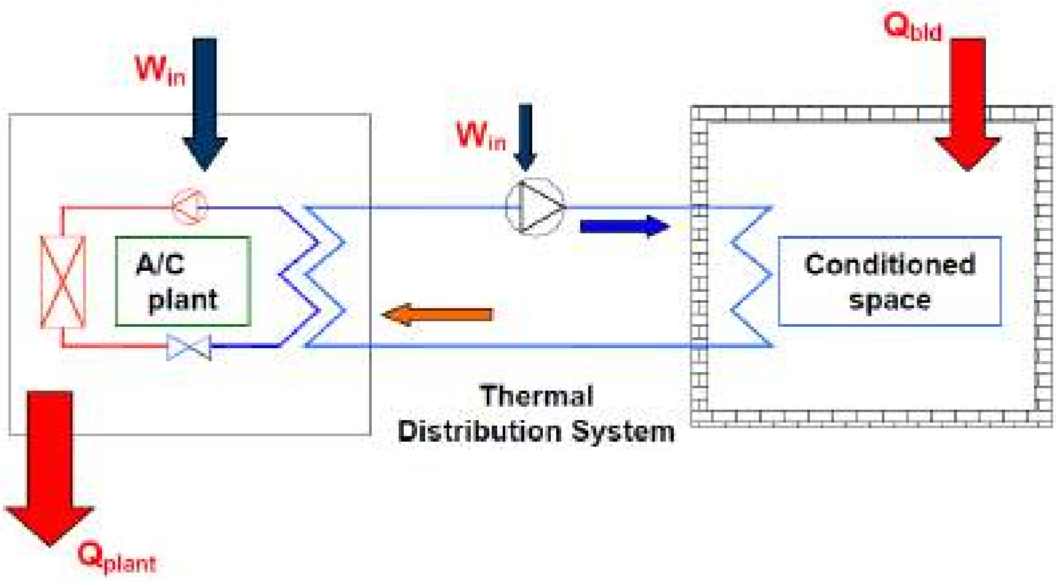


Fig. 11.1 *Schematic of summer air conditioning system with the thermal distribution system*

11.2 Selection Criteria for Air Conditioning Systems

Selection of a suitable air conditioning system depends on:

1. Capacity, performance and spatial requirements
2. Initial and running costs
3. Required system reliability and flexibility
4. Maintainability
5. Architectural constraints

The relative importance of the above factors varies from building owner to owner and may vary from project to project. The typical space requirement for large air conditioning systems may vary from about 4 percent to about 9 percent of the gross building area, depending upon the type of the system. Normally based on the selection criteria, the choice is narrowed down to 2 to 3 systems, out of which one will be selected finally.

11.3 Classification of Air Conditioning Systems

Based on the fluid media used in the thermal distribution system, air conditioning systems can be classified as:

1. All air systems
2. All water systems
3. Air- water systems
4. Unitary refrigerant based systems

11.4 All Air Systems

As the name implies, in an all air system air is used as the media that transports energy from the conditioned space to the A/C plant. In these systems air is processed in the A/C plant and this processed air is then conveyed to the conditioned space through insulated ducts using blowers and fans. This air extracts (or supplies in case of winter) the required amount of sensible and latent heat from the conditioned space. The return air from the conditioned space is conveyed back to the plant, where it again undergoes the required processing thus completing the cycle. No additional processing of air is required in the conditioned space. All air systems can be further classified into:

1. Single duct systems, or
2. Dual duct systems

The single duct systems can provide either cooling or heating using the same duct, but not both heating and cooling simultaneously. These systems can be further classified into:

1. Constant volume, single zone systems
2. Constant volume, multiple zone systems
3. Variable volume systems

The dual duct systems can provide both cooling and heating simultaneously. These systems can be further classified into:

1. Dual duct, constant volume systems
2. Dual duct variable volume systems

11.4.1 Single duct, constant volume, single zone systems

Fig.11.2 shows the classic, single duct, single zone, constant volume systems. As shown in the figure, outdoor air (OD air) for ventilation and recirculated air (RC air) are mixed in the required proportions using the dampers and the mixed air is made to flow through a cooling and dehumidifying coil, a heating coil and a humidifier using a an insulated ducting and a supply fan. As the air flows through these coils the temperature and moisture content of the air are brought to the required values. Then this air is supplied to the conditioned space, where it meets the building cooling or heating requirements. The return air leaves the conditioned space, a part of it is recirculated and the remaining part is vented to the atmosphere. A thermostat senses the temperature of air in the conditioned space and controls the amount of cooling or heating provided in the coils so that the supply air temperature can be controlled as per requirement. A humidistat measures the humidity ratio in the conditioned space and controls the amount of water vapour added in the humidifier and hence the supply air humidity ratio as per requirement.

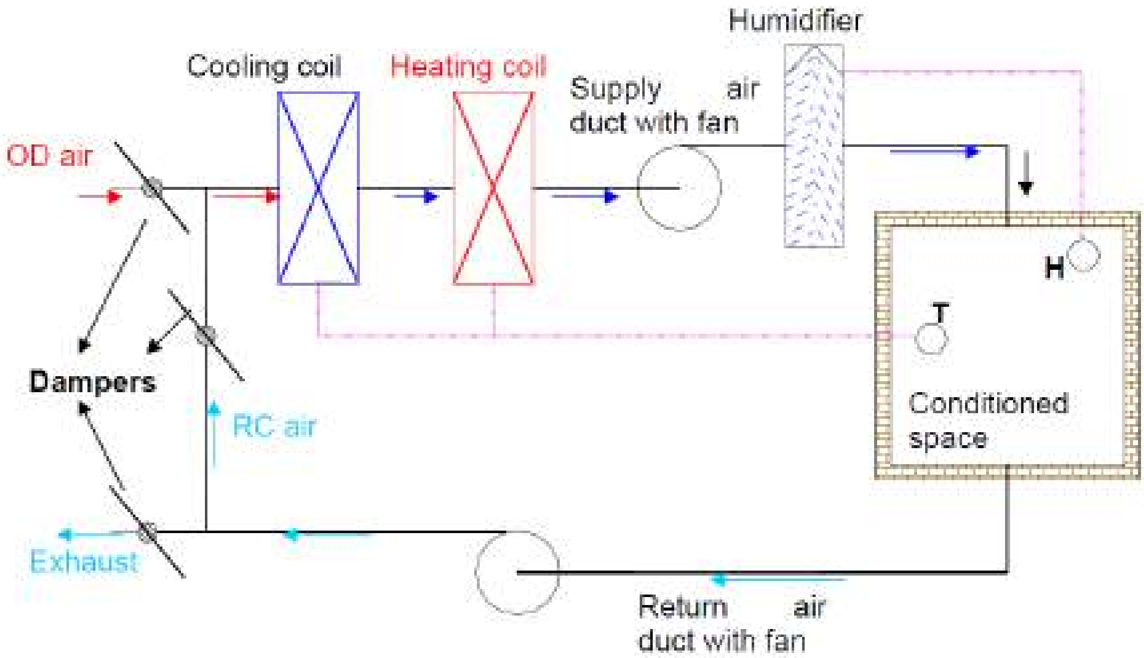


Fig.11.2 *A Constant volume, single zone system*

This system is called as a single duct system as there is only one supply duct, through which either hot air or cold air flows, but not both simultaneously. It is called as a constant volume system as the volumetric flow rate of supply air is always maintained constant. It is a single zone system as the control is based on temperature and humidity ratio measured at a single point. Here a zone refers to a space controlled by one thermostat. However, the single zone may consist of a single room or one floor or whole of a building consisting of several rooms. The cooling/ heating capacity in the single zone, constant volume systems is regulated by regulating the supply air temperature and humidity ratio, while keeping the supply airflow rate constant. A separate sub-system controls the amount of OD air supplied by controlling the damper position.

Since a single zone system is controlled by a single thermostat and humidistat, it is important to locate these sensors in a proper location, so that they are indicative of zone conditions. The supply air conditions are controlled by either coil control or face-and-bypass control.

In coil control, supply air temperature is controlled by varying the flow rate of cold and hot water in the cooling and heating coils, respectively. As the cooling season gradually changes to heating season, the cooling coil valve is gradually closed and heating coil valve is opened. Though coil control is simpler, using this type of control it is not possible to control the zone humidity precisely as the dehumidification rate in the cooling coil decreases with cold water flow rate. Thus at low cold water flow rates, the humidity ratio of the conditioned space is likely to be higher than required.

In face-and-bypass control, the cold and hot water flow rates are maintained constant, but the amount of air flowing over the coils are decreased or increased by opening or closing the by-pass dampers, respectively. By this method it is possible to control the zone humidity more precisely, however, this type of control occupies more space physically and is also expensive compared to coil control.

Applications of single duct, single zone, constant volume systems:

1. A space with uniform loads, such as large open areas with small external loads e.g. theatres, auditoria, and departmental stores.
2. Spaces requiring precision control such as laboratories

The Multiple, single zone systems can be used in large buildings such as factories, office buildings etc.

11.4.2 Single duct, constant volume, multiple zone systems

For very large buildings with several zones of different cooling/heating requirements, it is not economically feasible to provide separate single zone systems for each zone. For such cases, multiple zone systems are suitable. **Fig.11.3** shows a single duct, multiple zone system with terminal reheat coils. In these systems all the air is cooled and dehumidified (for summer) or heated and humidified (for winter) to a given minimum or maximum temperature and humidity ratio. A constant volume of this air is supplied to the reheat coil of each zone. In the reheat coil the supply air temperature is increased further to a required level depending upon the load on that particular zone. This is achieved by a zone thermostat, which controls the amount of reheat, and hence the supply air temperature. The reheat coil may run on either electricity or hot water.

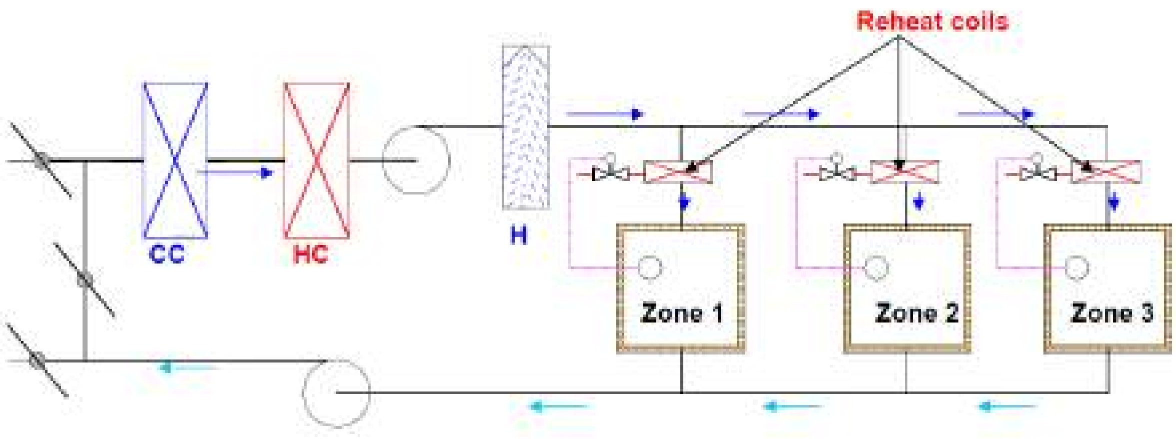


Fig.11.3. Single duct, constant volume system with multiple zones and reheat coils

Advantages of single duct, multiple zone, constant volume systems with reheat coils:

a) Relatively small space requirement
b) Excellent temperature and humidity control over a wide range of zone loads
c) Proper ventilation and air quality in each zone is maintained as the supply air amount is kept constant under all conditions

Disadvantages of single duct, multiple zone, constant volume systems with reheat coils:

a) High energy consumption for cooling, as the air is first cooled to a very low temperature and is then heated in the reheat coils. Thus energy is required first for cooling and then for reheating. The energy consumption can partly be reduced by increasing the supply air temperature, such that at least one reheat coil can be switched-off all the time. The energy consumption can also be reduced by using waste heat (such as heat rejected in the condensers) in the reheat coil.
b) Simultaneous cooling and heating is not possible.

11.4.3 Single duct, variable air volume (VAV) systems

Fig.11.4 shows a single duct, multiple zone, and variable air volume system for summer air conditioning applications. As shown, in these systems air is cooled and dehumidified to a required level in the cooling and dehumidifying coil (CC). A variable volume of this air is supplied to each zone. The amount of air supplied to each zone is controlled by a zone damper, which in turn is controlled by that zone thermostat as shown in the figure. Thus the temperature of supply air to each zone remains constant, whereas its flow rate varies depending upon the load on that particular zone.

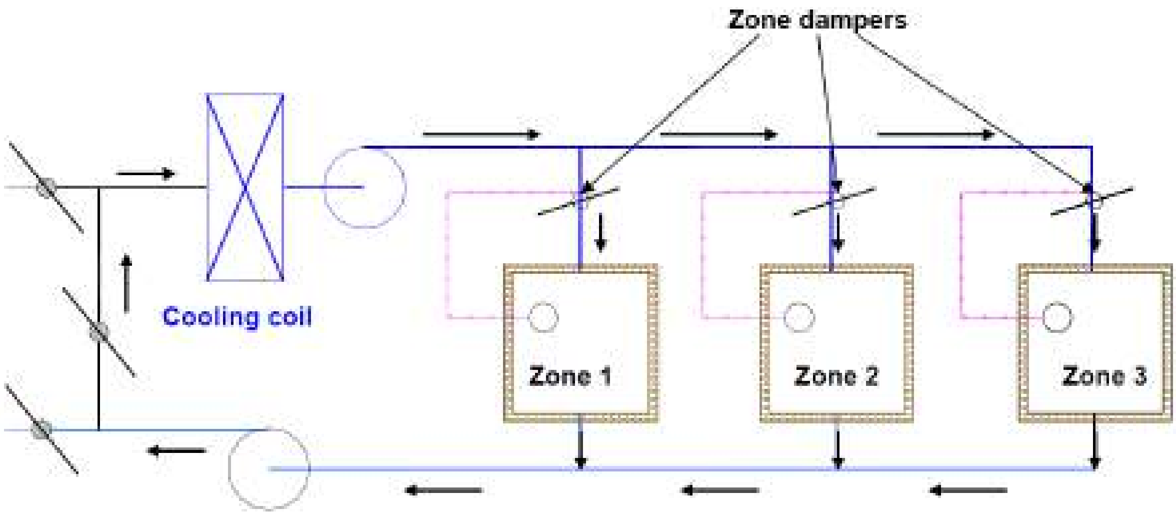


Fig.11.4 *Single duct, multiple zone, variable air flow system*

Compared to constant volume systems, the variable air volume systems offer advantages such as:

a) Lower energy consumption in the cooling system as air is not cooled to very low temperatures and then reheated as in constant volume systems.
b) Lower energy consumption also results due to lower fan power input due to lower flow rate, when the load is low. These systems lead to significantly lower power consumption, especially in perimeter zones where variations in solar load and outside temperature allows for reduced air flow rates.

However, since the flow rate is controlled, there could be problems with ventilation, IAQ and room air distribution when the zone loads are very low. In addition it is difficult to control humidity precisely using VAV systems. Balancing of dampers could be difficult if the airflow rate varies widely. However, by combining VAV systems with terminal reheat it is possible to maintain the air flow rate at a minimum required level to ensure proper ventilation and room air distribution. Many other variations of VAV systems are available to cater to a wide variety of applications.

11.4.4 Dual duct, constant volume systems

Fig.11.5 shows the schematic of a dual duct, constant volume system. As shown in the figure, in a dual duct system the supply air fan splits the flow into two streams. One stream flow through the cooling coil and gets cooled and dehumidified to about 13^{o}C, while the other stream flow the heating coil and is heated to about 35–45^{o}C. The cold and hot streams flow through separate ducts. Before each conditioned space or zone, the cold and hot air streams are mixed in required proportions using a mixing box arrangement, which is controlled by the zone thermostat. The total volume of air supplied to each zone remains constant, however, the supply air temperature varies depending upon load.

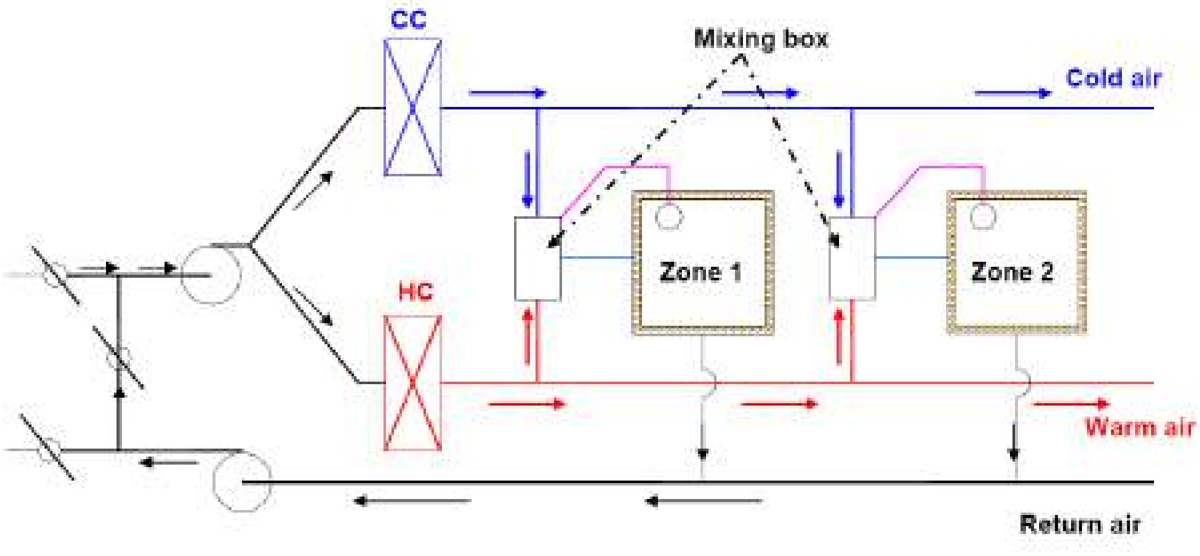


Fig.11.5 *Duel duct, constant volume system*

Advantages of dual duct systems

1. Since total airflow rate to each zone is constant, it is possible to maintain proper IAQ and room air distribution.
2. Cooling in some zones and heating in other zones can be achieved simultaneously
3. System is very responsive to variations in the zone load, thus it is possible to maintain required conditions precisely.

Disadvantages of dual duct systems

1. Occupies more space as both cold air and hot air ducts have to be sized to handle all the air flow rate, if required.
2. Not very energy efficient due to the need for simultaneous cooling and heating of the air streams. However, the energy efficiency can be improved by completely shutting down the cooling coil when the outside temperature is low and mixing supply air from fan with hot air in the mixing box. Similarly, when the outside weather is hot, the heating coil can be completely shut down, and the cold air from the cooling coil can be mixed with supply air from the fan in the mixing box.

11.4.5 Dual duct, variable air volume systems

These systems are similar to dual duct, constant volume systems with the only difference that instead of maintaining constant flow rates to each zone, the mixing boxes reduce the air flow rate as the load on the zone drops.

11.4.6 Outdoor air control in all air systems

As mentioned in a previous lecture, outdoor air is required for ventilation purposes. In all air systems, a sub-system controls the amount of outdoor air by controlling the position of exhaust, re-circulated and outdoor air dampers. From mass balance, since the outdoor airflow rate should normally be equal to the exhaust airflow rate (unless building pressurization or de-pressurization is required), both the exhaust and outdoor air dampers open or close in unison. Again from mass balance, when the outdoor air damper opens the re-circulated air damper closes, and vice versa. The control system maintains a minimum amount of outdoor air (about 10 to 20% of supply air flow rate as required for ventilation) when the outdoor is too cold ($\leq -30^oC$) or too warm ($\geq 24^oC$). For energy conservation, the amount of outdoor air can be increased gradually as the outdoor air temperature increases from -30^oC to about 13^oC. A 100 percent outdoor air can be used when the outdoor air temperature is between 13^oC to about 24^oC. By this method it is possible to reduce the annual energy consumption of the air conditioning system significantly, while maintaining the required conditions in the conditioned space.

11.4.7. Advantages of all air systems

1. All air systems offer the greatest potential for energy conservation by utilizing the outdoor air effectively.
2. By using high-quality controls it is possible to maintain the temperature and relative humidity of the conditioned space within ± **0.15^oC** (DBT) and ± **0.5**%, respectively.
3. Using dual duct systems, it is possible to provide simultaneous cooling and heating. Changeover from summer to winter and vice versa is relatively simple in all air systems.
4. It is possible to provide good room air distribution and ventilation under all conditions of load.
5. Building pressurization can be achieved easily.
6. The complete air conditioning plant including the supply and return air fans can be located away from the conditioned space. Due to this it is possible to use a wide variety of air filters and avoid noise in the conditioned space.

11.4.8. Disadvantages of all air systems

1. They occupy more space and thus reduce the available floor space in the buildings. It could be difficult to provide air conditioning in high-rise buildings with the plant on the ground floor or basement due to space constraints.
2. Retrofitting may not always be possible due to the space requirement.
3. Balancing of air in large and particularly with variable air volume systems could be difficult.

11.4.9. Applications of all air systems

All air systems can be used in both comfort as well as industrial air conditioning applications. They are especially suited to buildings that require individual control of multiple zones, such as office buildings, classrooms, laboratories, hospitals, hotels, ships etc. They are also used extensively in applications that require very close control of the conditions in the conditioned space such as clean rooms, computer rooms, operation theatres, research facilities etc.

11.5. All Water Systems

In all water systems the fluid used in the thermal distribution system is water, i.e., water transports energy between the conditioned space and the air conditioning plant. When cooling is required in the conditioned space then cold water is circulated between the conditioned space and the plant, while hot water is circulated through the distribution system when heating is required. Since only water is transported to the conditioned space, provision must be there for supplying required amount of treated, outdoor air to the conditioned space for ventilation purposes. Depending upon the number of pipes used, the all water systems can be classified into a 2-pipe system or a 4-pipe system.

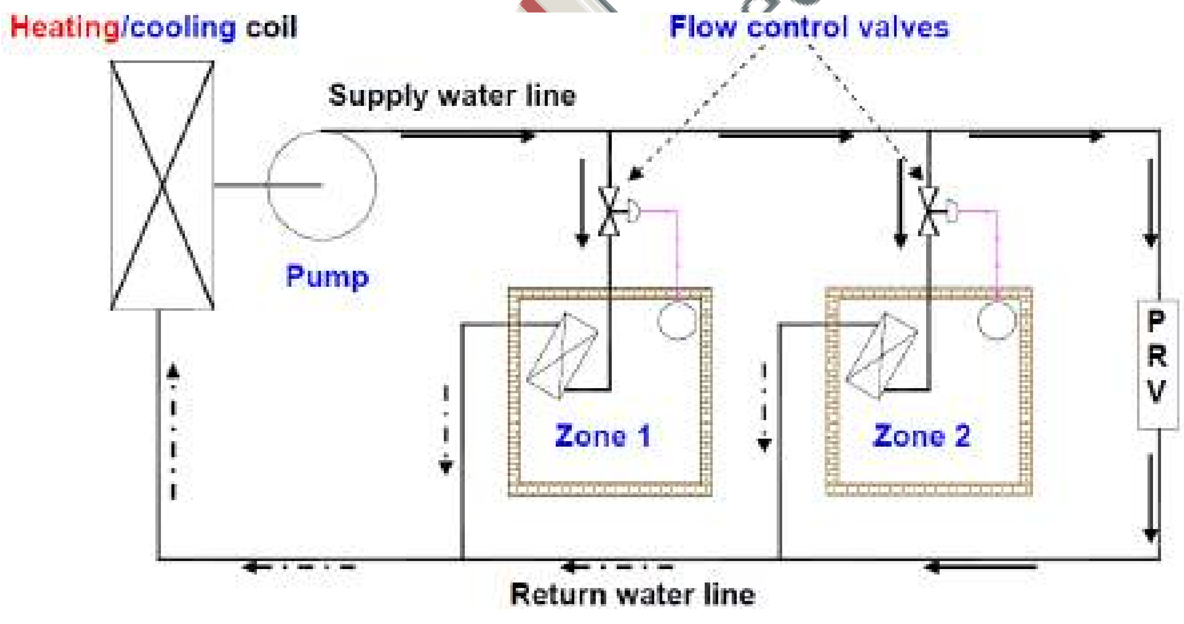


Fig.11.6 *A two-pipe, all water system*

A 2-pipe system is used for either cooling only or heating only application, but cannot be used for simultaneous cooling and heating. Figure 36.6 shows the schematic of a 2-pipe, all water system. As shown in the figure and as the name implies, a 2-pipe system consists of two pipes – one for supply of cold/hot water to the conditioned space and the other for the return water. A cooling or heating coil provides the required cold or hot water. As the supply water flows through the conditioned space, required heat transfer between the water and conditioned space takes place, and the return water flows back to the cooling or heating coil. A flow control valve controls the flow rate of hot or cold water to the conditioned space and thereby meets the required building heating or cooling load. The flow control valve is controlled by the zone thermostat. As already mentioned, a separate arrangement must be made for providing the required amount of ventilation air to the conditioned space. A pressure relief valve (PRV) is installed in the water line for maintaining balanced flow rate.

A 4-pipe system consists of two supply pipelines – one for cold water and one for hot water; and two return water pipelines. The cold and hot water are mixed in a required proportion depending upon the zone load, and the mixed water is supplied to the conditioned space. The return water is split into two streams, one stream flows to the heating coil while the other flows to the cooling coil.

Heat transfer between the cold/hot water and the conditioned space takes place either by convection, conduction or radiation or a combination of these. The cold/hot water may flow through bare pipes located in the conditioned space or one of the following equipment can be used for transferring heat:

1. Fan coil units
2. Convectors
3. Radiators etc.

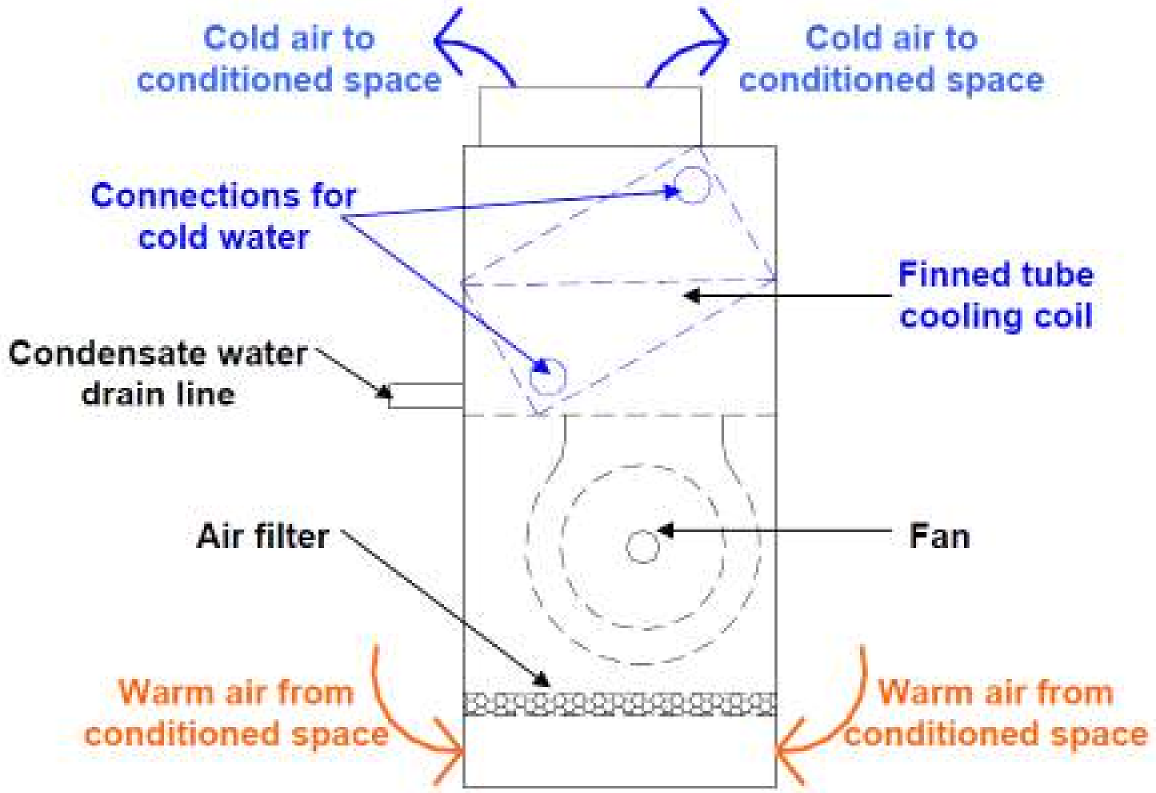


Fig.11.7 *A Basic fan coil unit for cooling purpose*

A fan coil unit is located inside the conditioned space and consists of a heating and/or cooling coil, a fan, air filter, drain tray and controls. **Fig.11.7** shows the schematic of a fan coil unit used for cooling applications. As shown in the figure, the basic components of a fan coil unit are: finned tube cooling coil, fan, air filter, insulated drain tray with provision for draining condensate water and connections for cold water lines. The cold water circulates through the finned tube coil while the blower draws warm air from the conditioned space and blows it over the cooling coil. As the air flows through the cooling coil it is cooled and dehumidified. The cold and dehumidified air is supplied to the conditioned space for providing required conditions inside the conditioned space. The water condensed due to dehumidification of room air has to be drained continuously. A cleanable or replaceable filter is located in the upstream of the fan to prevent dust accumulation on the cooling coil and also to protect the fan and motor from dust. Fan coil units for domestic air conditioning are available in the airflow range of 100 to 600 l/s, with multi-speed, high efficiency fans. In some designs, the fan coil unit also consists of a heating coil, which could be in the form of an electric heater or steam or hot water coil. Electric heater is used with 2-pipe systems, while the hot water/steam coils are used with 4-pipe systems. The fan coil units are either floor mounted, window mounted or ceiling mounted. The capacity of a fan coil unit can be controlled either by controlling the cold water flow rate or by controlling air flow rate or both. The airflow rate can be controlled either by a damper arrangement or by varying the fan speed. The control may be manual or automatic, in which case, a room thermostat controls the capacity. Since in the fan coil unit there is no provision for ventilation, a separate arrangement must be made to take care of ventilation. A fan coil unit with a provision for introducing treated ventilation air to the conditioned space is called as **unit ventilator**.

A **convector** consists of a finned tube coil through which hot or cold fluid flows. Heat transfer between the coil and surrounding air takes place by natural convection only, hence no fans are used for moving air. Convectors are very widely used for heating applications, and very rarely are used for cooling applications.

In a **radiator**, the heat transfer between the coil and the surrounding air is primarily by radiation. Some amount of heat is also transferred by natural convection. Radiators are widely used for heating applications, however, in recent times they are also being used for cooling applications.

11.5.1 Advantages of all water systems

1. The thermal distribution system requires very less space compared to all air systems. Thus there is no penalty in terms of conditioned floor space. Also the plant size will be small due to the absence of large supply air fans.
2. Individual room control is possible, and at the same time the system offers all the benefits of a large central system.
3. Since the temperature of hot water required for space heating is small, it is possible to use solar or waste heat for winter heating.
4. It can be used for new as well existing buildings (retrofitting).

5. Simultaneous cooling and heating is possible with 4-pipe systems.

11.5.2. Disadvantages of all water systems

1. Requires higher maintenance compared to all air systems, particularly in the conditioned space.
2. Draining of condensate water can be messy and may also create health problems if water stagnates in the drain tray. This problem can be eliminated, if dehumidification is provided by a central ventilation system, and the cooling coil is used only for sensible cooling of room air.
3. If ventilation is provided by opening windows or wall apertures, then, it is difficult to ensure positive ventilation under all circumstances, as this depends on wind and stack effects.
4. Control of humidity, particularly during summer is difficult using chilled water control valves.

11.5.3. Applications of all water systems

All water systems using fan coil units are most suitable in buildings requiring individual room control, such as hotels, apartment buildings and office buildings.

11.6. Air-Water Systems

In air-water systems both air and water are used for providing required conditions in the conditioned space. The air and water are cooled or heated in a central plant. The air supplied to the conditioned space from the central plant is called as primary air, while the water supplied from the plant is called as secondary water. The complete system consists of a central plant for cooling or heating of water and air, ducting system with fans for conveying air, water pipelines and pumps for conveying water and a room terminal. The room terminal may be in the form of a fan coil unit, an induction unit or a radiation panel. **Fig.11.8** shows the schematic of a basic air-water system. Even though only one conditioned space is shown in the schematic, in actual systems, the air-water systems can simultaneously serve several conditioned spaces.

Normally a constant volume of primary air is supplied to each zone depending upon the ventilation requirement and the required sensible cooling capacity at maximum building load. For summer air conditioning, the primary air is cooled and dehumidified in the central plant, so that it can offset all the building latent load. Chilled water is supplied to the conditioned space to partly offset the building sensible cooling load only. Since the chilled water coil kept in the conditioned space has to take care of only sensible load, condensation of room air inside the conditioned space is avoided thereby avoiding the problems of condensate drainage and related problems in the conditioned space. As mentioned, the primary takes care of the ventilation requirement of the conditioned space, hence unlike in all water systems, there is no need for separate ventilation systems. In winter, moisture can be added to the primary air in the central plant and hot water is circulated through the coil kept in the conditioned space. The secondary water lines can be of 2-pipe, 3-pipe or 4-pipe type similar to all water systems.

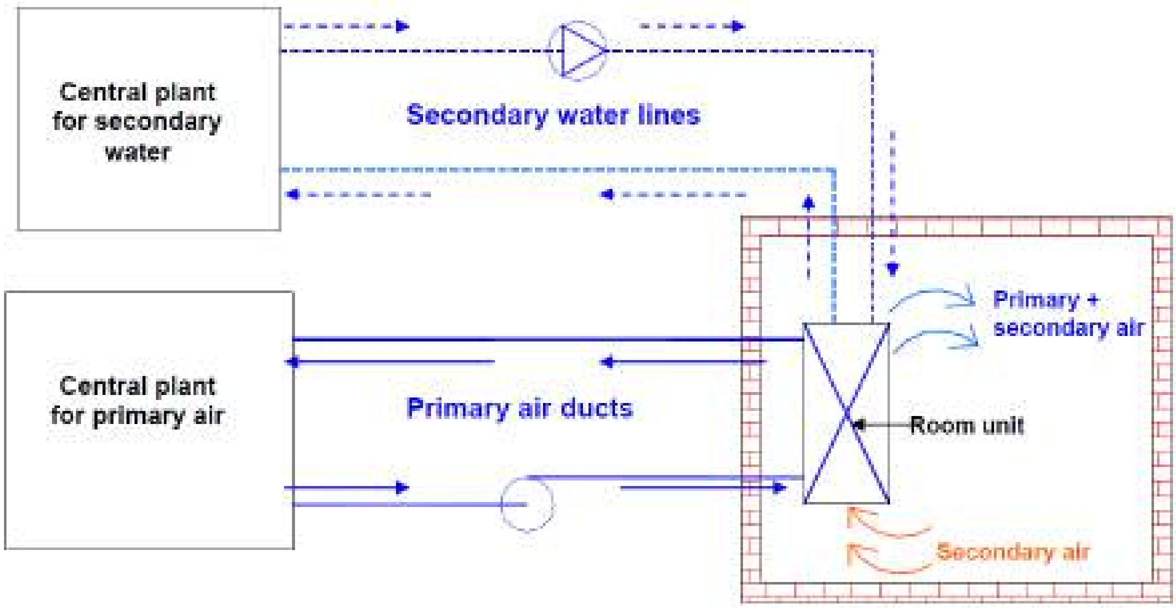


Fig.11.8: *A basic air-water system*

As mentioned the room unit may be in the form of a fan coil unit, an induction unit or in the form of a radiant panel. In an induction unit the cooling/heating coil is an integral part of the primary air system. The primary air supplied at medium to high pressure to the induction unit, induces flow of secondary air from the conditioned space. The secondary air is sensibly cooled or heated as it flows through the cooling/heating coil. The primary and secondary air are mixed and supplied to the conditioned space. The fan coil units are similar to the ones used in all water systems.

11.6.1 Advantages of air-water systems

1. Individual zone control is possible in an economic manner using room thermostats, which control either the secondary water flow rate or the secondary air (in fan coil units) or both.
2. It is possible to provide simultaneous cooling and heating using primary air and secondary water.
3. Space requirement is reduced, as the amount of primary supplied is less than that of an all air systems.
4. Positive ventilation can be ensured under all conditions.
5. Since no latent heat transfer is required in the cooling coil kept in the conditioned space, the coil operates dry and its life thereby increases and problems related to odours or fungal growth in conditioned space is avoided.

6. The conditioned space can sometimes be heated with the help of the heating coil and secondary air, thus avoiding supply of primary air during winter.
7. Service of indoor units is relatively simpler compared to all water systems.

11.6.2 Disadvantages of air-water systems

1. Operation and control are complicated due to the need for handling and controlling both primary air and secondary water.
2. In general these systems are limited to perimeter zones.
3. The secondary water coils in the conditioned space can become dirty if the quality of filters used in the room units is not good.
4. Since a constant amount of primary air is supplied to conditioned space, and room control is only through the control of room cooling/heating coils, shutting down the supply of primary air to unoccupied spaces is not possible.
5. If there is abnormally high latent load on the building, then condensation may take place on the cooling coil of secondary water.
6. Initial cost could be high compared to all air systems.

11.6.3 Applications of air-water systems

These systems are mainly used in exterior buildings with large sensible loads and where close control of humidity in the conditioned space is not required. These systems are thus suitable for office buildings, hospitals, schools, hotels, apartments etc.

11.7 Unitary refrigerant based systems

Unitary refrigerant based systems consist of several separate air conditioning units with individual refrigeration systems. These systems are factory assembled and tested as per standard specifications, and are available in the form of package units of varying capacity and type. Each package consists of refrigeration and/or heating units with fans, filters, controls etc. Depending upon the requirement these are available in the form of window air conditioners, split air conditioners, heat pumps, and ductable systems with air cooled or water cooled condensing units etc. The capacities may range from fraction of TR to about 100 TR for cooling. Depending upon the capacity, unitary refrigerant based systems are available as single units which cater to a single conditioned space, or multiple units for several conditioned spaces. **Fig.11.9** shows the schematic of a typical window type, room air conditioner, which is available in cooling capacities varying from about 0.3 TR to about 3.0 TR. As the name implies, these units are normally mounted either in the window sill or through the wall. As shown in the figure, this type of unit consists of single package which includes the cooling and dehumidification coil, condenser coil, a hermetic compressor, expansion device (capillary tube), condenser fan, evaporator fan, room air filter and controls. A drain tray is provided at the bottom to take care of the condensate water. Both evaporator and condensers are plate fin-and-tube, forced convection type coils. For rooms that do not have external windows or walls, a split type room air conditioner can be used. In these air conditioners, the condensing unit comprising of the condenser, compressor and condenser fan with motor are located outside, while the indoor unit consisting of the evaporator, evaporator fan with motor, expansion valve and air filter is located inside the conditioned room. The indoor and outdoor units are connected by refrigerant piping. In split type air conditioners, the condensed water has to be taken away from the conditioned space using separate drain pipes. In the room air conditioners (both window mounted and split type), the cooling capacity is controlled by switching the compressor on-and-off. Sometimes, in addition to the on-and-off, the fan speed can also be regulated to have a modular control of capacity. It is also possible to switch off the refrigeration system completely and run only the blower for air circulation.

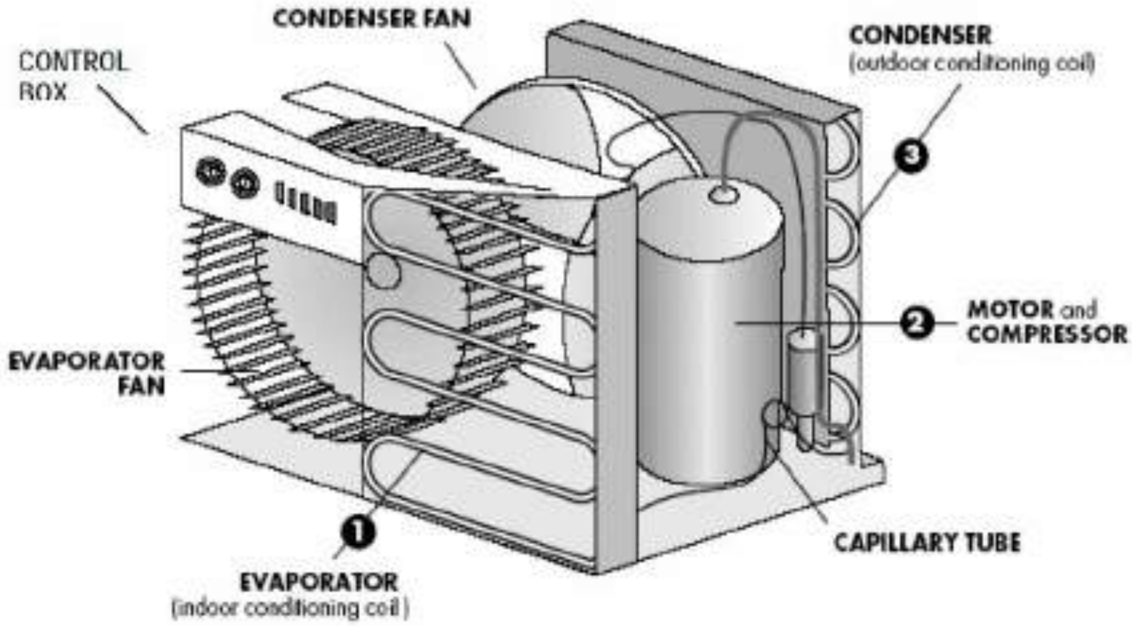


Fig.11.9 *A typical window type room air conditioner*

Fig.11.10 shows a typical package unit with a remote condensing unit. As shown, in a typical package unit, the remote condensing unit consists of the compressor and a condenser, while the indoor unit consists of the plate fin-and-tube type, evaporator, a blower, air filter, drain tray and an arrangement for connecting supply air and return air ducts. These units are available in capacities ranging from about 5 TR to up to about 100 TR. The condenser used in these systems could be either air cooled or water cooled. This type of system can be used for providing air conditioning in a large room or it can cater to several small rooms with suitable supply and return ducts. It is also possible to house the entire refrigeration in a single package with connections for water lines to the water cooled condenser and supply and return air ducts. Larger systems are either constant air volume type or variable air volume type. They may also include heating coils along with the evaporator.

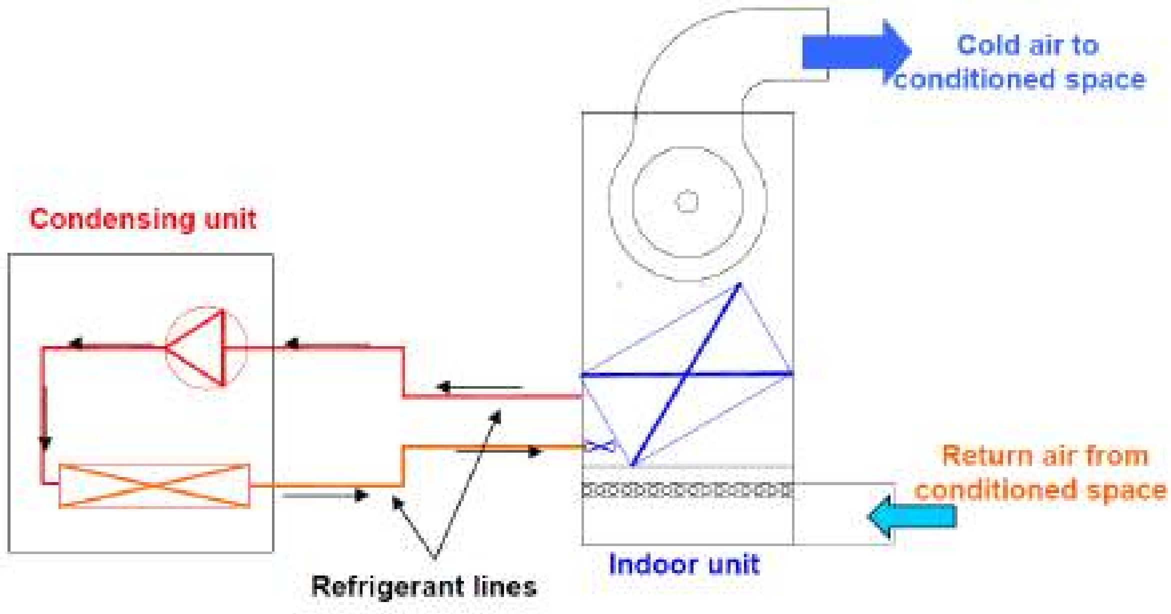


Fig.11.10 *A typical package unit with remote condensing unit*

Most of the unitary systems have a provision for supplying outdoor air for ventilation purposes. The type of control depends generally on the capacity of the unit. The control system could be as simple as a simple thermostat based on-off control as in room air conditioners to sophisticated microprocessor based control with multiple compressors or variable air volume control or a combination of both.

11.7.1 Advantages of unitary refrigerant based systems

1. Individual room control is simple and inexpensive.
2. Each conditioned space has individual air distribution with simple adjustment by the occupants.
3. Performance of the system is guaranteed by the manufacturer.
4. System installation is simple and takes very less time.
5. Operation of the system is simple and there is no need for a trained operator.
6. Initial cost is normally low compared to central systems.
7. Retrofitting is easy as the required floor space is small.

11.7.2 Disadvantages of unitary refrigerant based systems

1. As the components are selected and matched by the manufacturer, the system is less flexible in terms of air flow rate, condenser and evaporator sizes.
2. Power consumption per TR could be higher compared to central systems.
3. Close control of space humidity is generally difficult.
4. Noise level in the conditioned space could be higher.
5. Limited ventilation capabilities.

6. Systems are generally designed to meet the appliance standards, rather than the building standards.
7. May not be appealing aesthetically.
8. The space temperature may experience a swing if on-off control is used as in room air conditioners.
9. Limited options for controlling room air distribution.
10. Equipment life is relatively short.

11.7.3 Applications of unitary refrigerant based systems

Unitary refrigerant based systems are used where stringent control of conditioned space temperature and humidity is not required and where the initial cost should be low with a small lead time. These systems can be used for air conditioning individual rooms to large office buildings, classrooms, hotels, shopping centers, nursing homes etc. These systems are especially suited for existing building with a limitation on available floor space for air conditioning systems.