UNIT- 5 - REFRIGERATION CYCLES & REFRIGERANTS

5.1 INTRODUCTION

For specific applications, efficiencies of both living and non-living beings depend to a great extent on the physical environment. The nature keeps conditions in the physical environment in the dynamic state ranging from one extreme to the other. Temperature, humidity, pressure and air motion are some of the important environment variables that at any location keep changing throughout the year. Adaptation to these many a times unpredictable variations are not possible and thus working efficiently is not feasible either for the living beings or the non-living ones. Thus for any specific purpose, control of the environment is essential. Refrigeration and air-conditioning is the subject which deals with the techniques to control the environments of the living and non-living subjects and thus provide them comforts to enable them to perform better and have longer lives.

5.2 DEFINITIONS

Refrigeration:

Refrigeration is defined as a method of reducing the temperature of a system below that of the surroundings and maintaining it at the lower temperature by continuously extracting the heat from it.

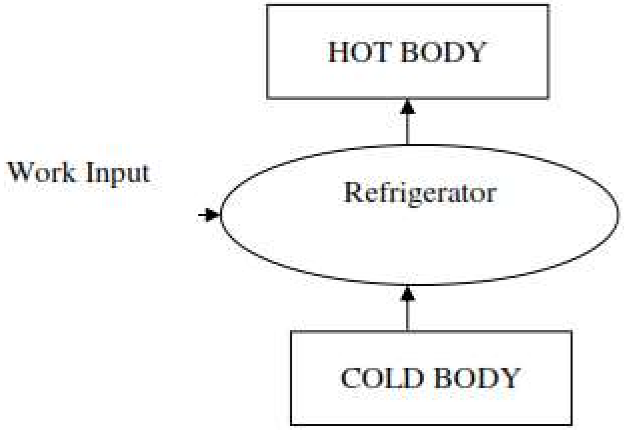


Fig 5.1

The principle of refrigeration is based on second law of thermodynamics. It states that heat does not flow from a low temperature body to a high temperature body without the help of an external work. In refrigeration process, the heat is continuously removed from a system at lower temperature and transfers it to the surroundings at a higher temperature. This operation according to second law of thermodynamics can only be performed by the aid of the external work. Therefore in a refrigeration system, power is to be supplied to remove heat continuously from the refrigerator to keep it cool at a temperature less than the surroundings. The refrigeration cycle is based on reversible Carnot cycle.

Refrigeration effect:

The rate at which the heat is absorbed in a cycle of from the interior space to be cooled is called refrigeration effect. It is defined as the quantity of heat removed to the time taken. It is also called as the capacity of a refrigerator.

Ton of Refrigeration (or) Unit of Refrigeration (TR):

The standard unit of refrigeration is *ton refrigeration* or simply *ton* denoted by TR. It is equivalent to the rate of heat transfer needed to produce 1 ton (2000 lbs) of ice at 32 ^{0}F from water at 32 ^{0}F in one day, i.e., 24 hours. The enthalpy of solidification of water from and at 32 ^{0}F in British thermal unit is 144 Btu/lb. Thus

$$1\ \mathrm{TR} = \frac{2000\ \mathrm{lb} \times 144\ \mathrm{Btu/lb}}{24\ \mathrm{hr}}$$
$$= 12000\ \mathrm{Btu/hr} = 200\ \mathrm{Btu/min}$$

In general, 1 TR means 200 Btu of heat removal per minute. Thus if a refrigeration system is capable of cooling at the rate of 400 Btu/min, it is a 2 ton machine. A machine of 20 ton rating is capable of cooling at a rate of 20 × 200 = 4000 Btu/min. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries, the standard MKS unit of kcal/hr is used. In the MKS it can be seen that

$$1\ \mathrm{TR} = 12000\ \mathrm{Btu/hr} = \frac{12000}{3.968} = 3024.2\ \mathrm{kcal/hr}$$
$$= 50.4\ \mathrm{kcal/min} \approx 50\ \mathrm{kcal/min}$$

If Btu ton unit is expressed into SI system, it is found to be **210 kJ/min or 3.5 kW.**

Co-efficient of Performance (COP):

The Co-efficient of Performance is defined as the ratio of heat absorbed in a system to the work supplied.

The theoretical Coefficient of Performance (Carnot), (COP a standard measure of refrigeration efficiency of an ideal refrigeration system) depends on two key system temperatures: evaporator temperature T_e and condenser temperature T_c

COP is given as: $\mathrm{COP}_{\mathrm{Carnot}} = T_e / (T_c - T_e)$

This expression also indicates that higher COP_{Carnot} is achieved with higher evaporator temperatures and lower condenser temperatures. But COP is only a ratio of temperatures, and does not take into account the type of compressor. Hence the COP normally used in industry is calculated as follows:

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

Where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW.

Ice making capacity:

It is the ability of the refrigeration system to make ice. In other words, it is the capacity of refrigeration system to remove heat from water to make ice.

Relative COP:

It is the ratio of actual COP to the theoretical COP of a refrigerator. Actual COP is measured during a test and theoretical COP is obtained by applying the laws of thermodynamics.

5.3 REFRIGERATOR AND HEAT PUMP

The vapor compression refrigeration cycle is a common method for transferring heat from a low temperature to a high temperature.

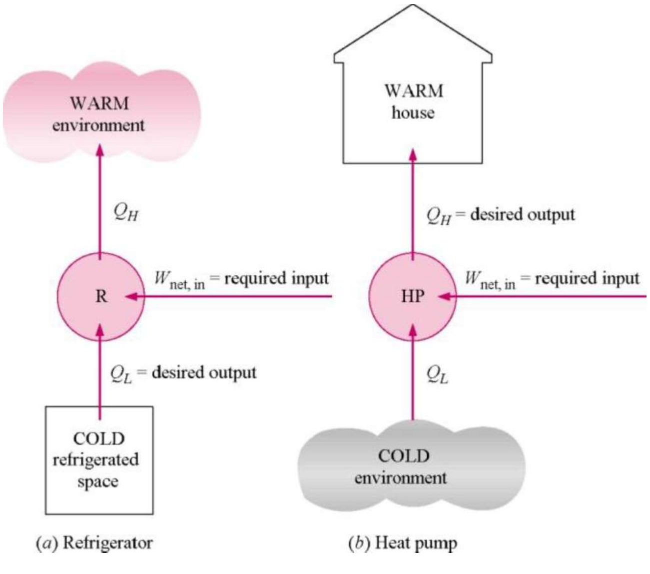


Fig 5.2

The above figure shows the objectives of refrigerators and heat pumps. The purpose of a refrigerator is the removal of heat, called the cooling load, from a low temperature medium. The purpose of a heat pump is the transfer of heat to a high temperature medium, called the heating load. When we are interested in the heat energy removed from a low temperature space, the device is called a refrigerator. When we are interested in the heat energy supplied to the high temperature space, the device is called a heat pump. In general, the term "heat pump" is used to describe the cycle as heat energy is removed from the low temperature space and rejected to the high temperature space.

The performance of refrigerators and heat pumps is expressed in terms of *coefficient of performance* (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

Both COP_R and COP_{HP} can be larger than 1. Under the same operating conditions, the COPs are related by

$$COP_{HP} = COP_R + 1$$

5.4 TYPES OF REFRIGERATION

Refrigeration is classified as based on working substance used

- Air refrigeration system (Bell-Coleman cycle)
- Water refrigeration system
- Ice refrigeration system
- Refrigeration by special fluid (low boiling point fluids – Refrigerants) (Reversed Carnot cycle)
    - Vapour compression refrigeration system (VCR)
    - Vapour absorbtion refrigeration system (VAR)
    - Vapour adsorbtion refrigeration system and etc.,

5.5 Simple Vapour Compression Refrigeration System (VCR)

It consists of the following essential parts:

Compressor

The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged into the condenser through the delivery or discharge valve B.

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.

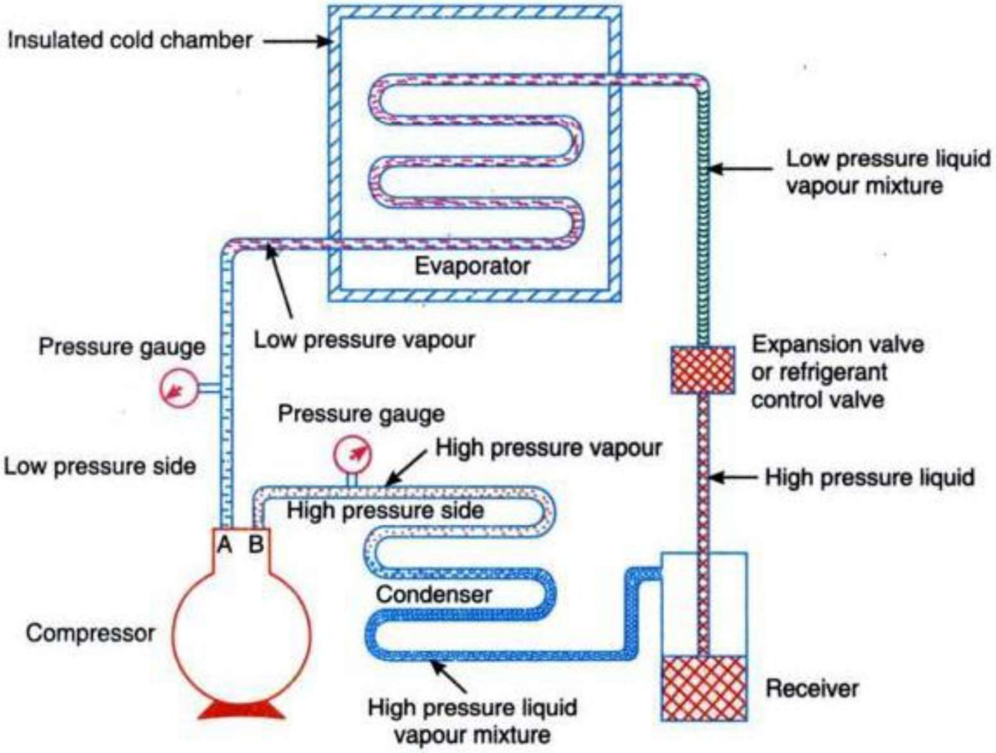


Fig 5.3

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.

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 vaporized in the evaporator at the low pressure and temperature

Evaporator

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

The Simple Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle has four components: evaporator, compressor, condenser, and expansion (or throttle) valve. The most widely used refrigeration cycle is the *vapor-compression refrigeration cycle.* In an ideal or simple vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor and is cooled to the saturated liquid state in the condenser. It is then throttled to the evaporator pressure and vaporizes as it absorbs heat from the refrigerated space.

The ideal vapor compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

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

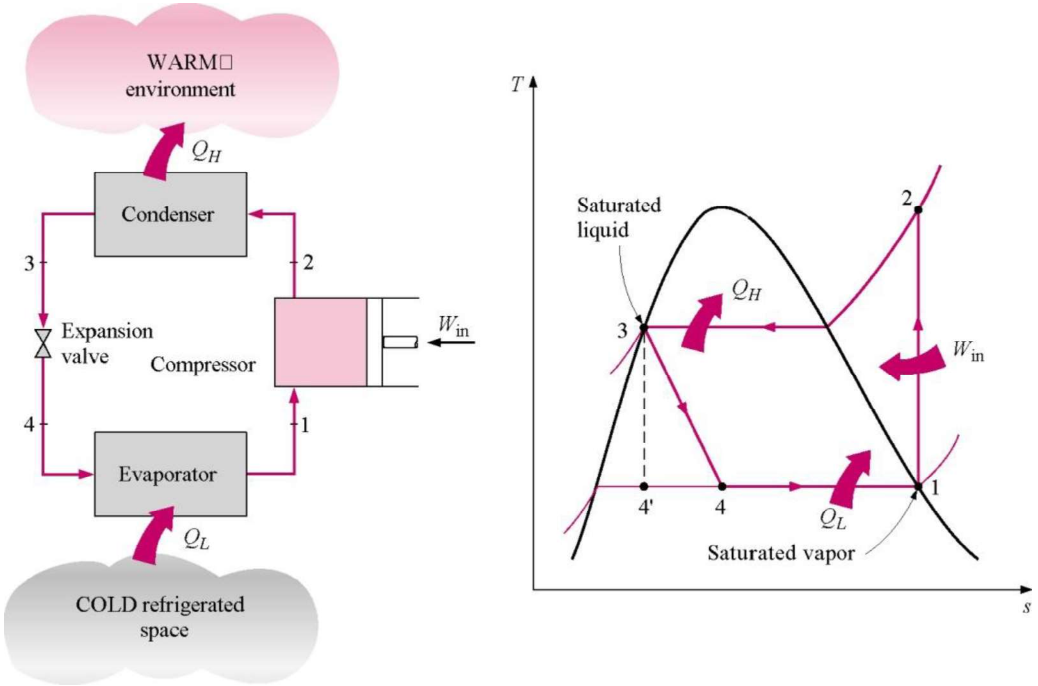


Fig 5.4

The P-h diagram is another convenient diagram often used to illustrate the refrigeration cycle.

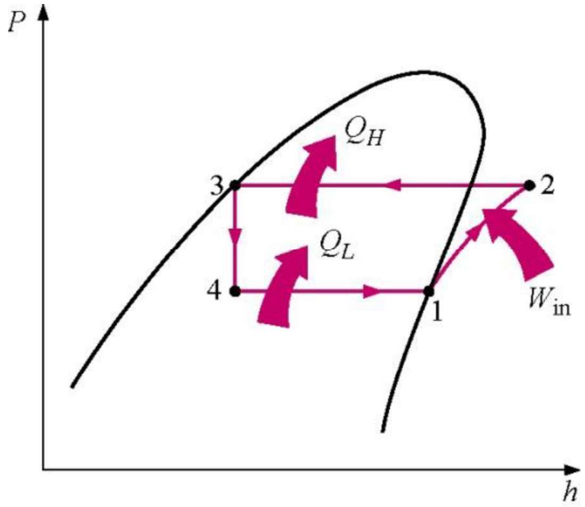


Fig 5.5

The ordinary household refrigerator is a good example of the application of this cycle.

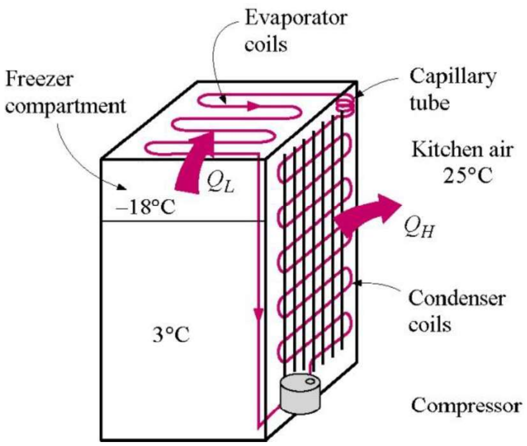


Fig 5.6

Results of First and Second Law Analysis for Steady-Flow

Component	Process	First Law Result
Compressor	s = Const.	$\dot{W}_{in} = \dot{m}(h_2 - h_1)$
Condenser	P = Const.	$\dot{Q}_H = \dot{m}(h_2 - h_3)$
Throttle Valve	Δs > 0 $\dot{W}_{net} = 0$ $\dot{Q}_{net} = 0$	$h_4 = h_3$
Evaporator	P = Const.	$\dot{Q}_L = \dot{m}(h_1 - h_4)$

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

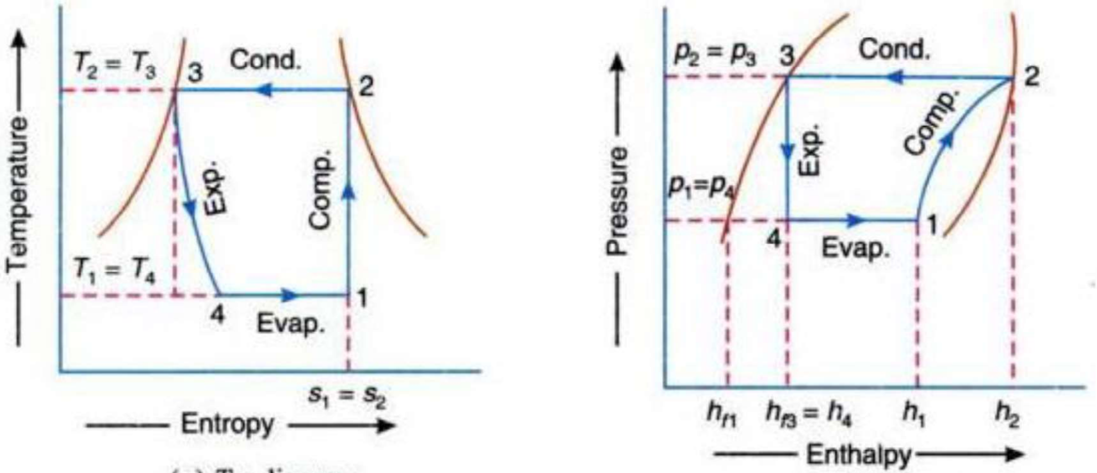
$$COP_{HP} = \frac{\dot{Q}_H}{\dot{W}_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

Methods to enhance the COP of simple vapour compression refrigeration system

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.

5.6 Theoretical Vapour Compression Cycle with Dry Saturated Vapour after Compression

A vapour compression cycle with dry saturated vapour after compression is shown in the following Figures (a) and (b) respectively. At point 1, let T_1, p_1, S_1 be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycle are as follows:



(*a*) *T-s* diagram.

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

Fig 5.7

Compression Process

The vapour refrigerant at low pressure $p1$ and temperatureT1 is compressed isentropic ally to dry saturated vapour as shown by the vertical line 1-2 on the T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rise from $p1$ to $p2$ and $T1$ to $T2$ respectively.

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

$w = h2 – h1$

where $h1$ = Enthalpy of vapour refrigerant at temperature $T1$, i.e. at suction of the compressor, and

$h2$ = Enthalpy of the vapour refrigerant at temperature $T2$. i.e. at discharge of the compressor.

Condensing Process

The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure p2 and temperature T2 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.

Expansion Process

The liquid refrigerant at pressure p3 = p2 and temperature T3 = T2, is expanded by throttling process through the expansion valve to a low pressure p4 = p1 and Temperature T4 = T1 as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

Vaporizing Process

The liquid-vapour mixture of the refrigerant at pressure p4 = p1 and temperature T4 = T1 is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which, is to be cooled, This heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written as R_E. The process of vaporization continues up to point 1 which is the starting point and thus the cycle is completed.

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

$$R_E = h_1 - h_4 = h_1 - hf_3$$

where hf3 = Sensible heat at temperature T3, i.e. enthalpy of liquid refrigerant leaving the condenser.

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

Coefficient of performance, C.O.P. = (Refrigerating effect)/(Work done)

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{h_1 - h_{f3}}{h_2 - h_1}$$

5.7 Theoretical Vapour Compression Cycle with Wet Vapour after Compression

(*a*) *T-s* diagram.

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

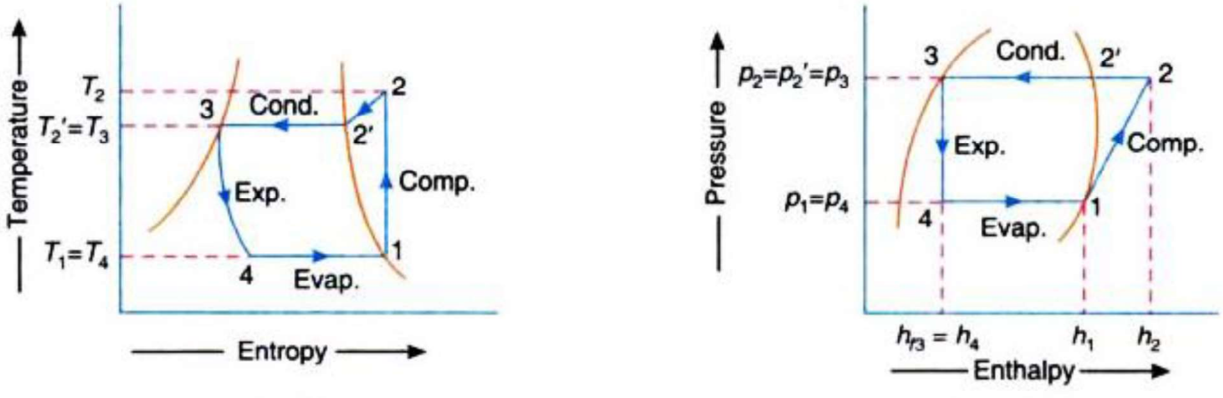
Fig 5.8

From the above figure, the end point of compression is lies in the region of wet (liquid and vapour). The enthalpy and entropy at this point is calculated by following formula

$$h_2 = h_{f2} + x_2 h_{fg2}$$

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2}$$

5.8 Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(*a*) *T-s* diagram.

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

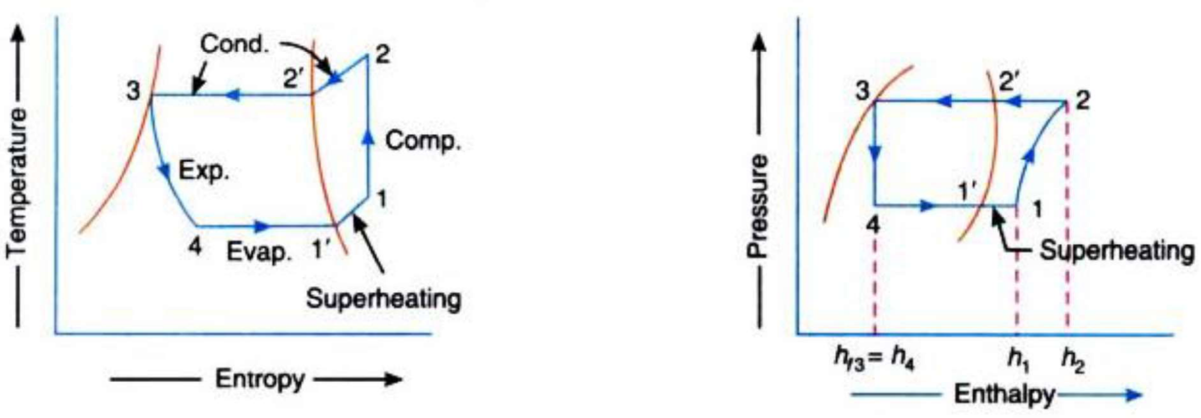
Fig 5.9

From the above figure, the end point of compression is lies in the region of superheated vapour. The enthalpy and entropy at this point is calculated by following formula

$$h_2 = h_{2'} + c_p \times \text{Degree of superheat} = h_{2'} + c_p (T_2 - T_{2'})$$

$$s_2 = s_{2'} + 2.3\, c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

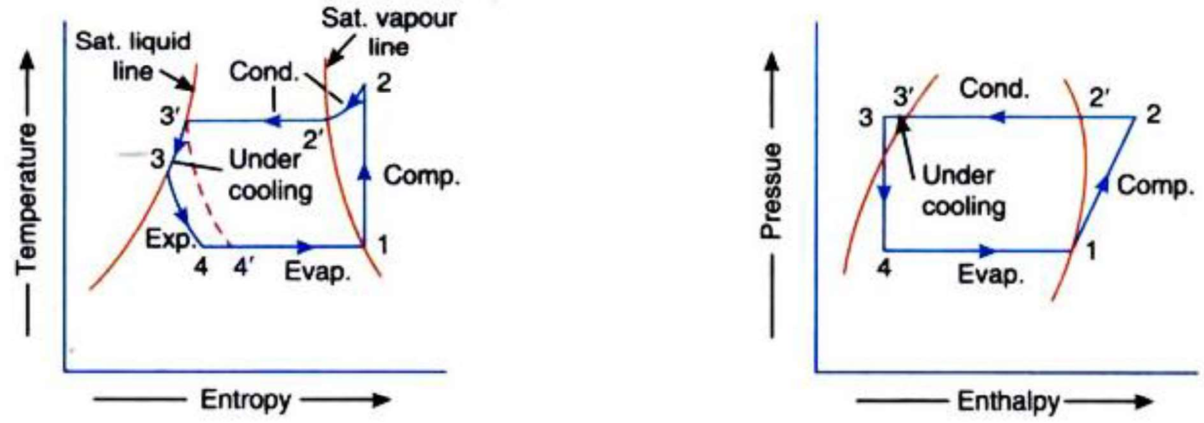
5.9 Theoretical Vapour Compression Cycle with Superheated Vapour before Compression



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Fig 5.10

5.10 Theoretical Vapour Compression Cycle with Sub-Cooling or Under cooling of Refrigerant



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Fig 5.11

$$h_{f3} = h_{f3'} - c_p \times \text{Degree of undercooling}$$

The process 3-3', cooling of the refrigerant temperature below its saturation temperature value is called **sub cooling or under cooling** at the end of condensation process.

5.11 REFRIGERANTS

The working agent in a refrigerating system that absorbs carries or releases heat from the place to be cooled or refrigerated can be termed as a refrigerant. This heat transfer generally takes place through a phase change of the refrigerant. A more complete definition of a refrigerant could be given as follows:

"Refrigerant is the fluid used for heat transfer in a refrigerating system that absorbs heat during evaporation from the region of low temperature and pressure, and releases heat during condensation at a region of higher temperature and pressure."

Primary and secondary refrigerants

Primary refrigerants are those which can be directly used for the purpose of refrigeration. If the refrigerant is allowed to flow freely into the space to be refrigerated and there is no danger of possible harm to human beings, then primary refrigerants are used. The refrigerants used in home refrigerators like Freon-12 are primary refrigerants.

On the other hand, there may be certain situations in which we cannot allow the refrigerant to come in direct contact with the items being refrigerated, and then the refrigerant used is termed as a secondary refrigerant. As for example, we cannot allow a toxic refrigerant to be used for air conditioning in residential buildings. There are some refrigerants which are highly inflammable and so their direct use is forbidden for safety reasons. Again, it may so happen that if direct refrigeration, such as in cooling a big cold storage, is allowed, then the amount of refrigerant required may be so large that its cost becomes prohibitively high. These are some typical situations for which we favor the use of secondary refrigerants. Water and brine solutions are common examples of secondary refrigerants.

5.12 Classification of refrigerants

Refrigerants can be broadly classified based on the following:

Working Principle

Under this heading, we have the primary or common refrigerants and the secondary refrigerants.

The primary refrigerants are those that pass through the processes of compression, cooling or condensation, expansion and evaporation or warming up during cyclic processes. Ammonia, R12, R22, carbon dioxide come under this class of refrigerants. On the other hand, the medium which does not go through the cyclic processes in a refrigeration system and is only used as a medium for heat transfer are referred to as secondary refrigerants. Water, brine solutions of sodium chloride and calcium chloride come under this category.

Safety Considerations

Under this heading, we have the following three sub-divisions.

Safe refrigerants

These are the non-toxic, non-flammable refrigerants such as R11, R12, R13, R14, R21, R22, R113, R114, methyl chloride, carbon dioxide, water etc.

Toxic and moderately flammable

Dichloroethylene methyl format, ethylchloride, sulphur dioxide, ammonia etc. come under this category.

Highly flammable refrigerants

The refrigerants under this category are butane, isobutene, propane, ethane, methane, ethylene etc.

Chemical Compositions

Halocarbon compounds

These are obtained by replacing one or more hydrogen atoms in ethane or methane with alogens.

Azeotropes

These are the mixtures of two or more refrigerants and behave as a compound.

Oxygen and Nitrogen Compounds

Refrigerants having either oxygen or nitrogen molecules in their structure, such as ammonia, are grouped separately and have a separate nomenclature from the halogenated refrigerants.

Cyclic organic Compounds

The compounds coming under this class are R316, R317 and R318.

Inorganic Compounds

These are further divided into two categories: Cryogenic and Non-cryogenic. Cryogenic fluids are those which are applied for achieving temperatures as low as – 160 ^{0}C to – 273 ^{0}C. Above this temperature range, we can use a multi-stage refrigeration system to realise the desired temperature. But below – 160^0 C, this is not possible since the COP of the cycle becomes very low. To attain temperatures below –160 ^{0}C, we use refrigerants such as nitrogen, oxygen, helium, hydrogen etc. and for temperatures close to – 273^{0}C, magnetic cooling is employed. The inorganic compounds which are employed above the cryogenic temperature ranges come under the remaining sub-division of inorganic refrigerants.

Unsaturated Compounds

Compounds such as ethylene, propylene etc., are grouped under this head and grouped under the 1000 series for convenience.

Miscellaneous

This group contains those compounds which cannot be grouped under the other components. They are indicated by the 700 series with the last numbers being their molecular weight. Examples include air, carbon dioxide, sulphur dioxide etc. As we can see from the above sub-divisions, they are not mutually exclusive. A compound may come under more than one sub-division. Hence, the importance of adopting the various naming conventions to designate the different refrigerants cannot be underestimated.

5.13 Designation of refrigerants

The American Society of Refrigerating Engineers (ASRE) has developed certain conventions for use in naming different types of refrigerants. These naming conventions differ according to the type of refrigerant. Each refrigerant type is denoted by a different series. Thus, we have separate series for halogenated refrigerants and other types. The naming conventions are simple and easy to follow. These conventions are now accepted worldwide and help to name the large variety of refrigerants available commercially nowadays.

Halocarbon Compounds

These are represented by a three digit nomenclature. Here, the first digit represents the number of carbon atoms in the compound minus one, the second digit stands for the number of hydrogen atoms plus one while the third digit stands for the number of fluorine atoms. The remaining atoms are chlorine. As an example, let us consider the refrigerant having R22 as its three digit nomenclature.

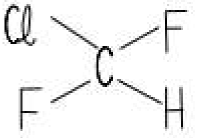
According to the above mentioned convention,

No. of C atoms in R22: $C - 1 = 0 \Rightarrow C = 1$

No. of H atoms in R22: $H + 1 = 2 \Rightarrow H = 1$

No. of F atoms in R22: F = 2

Since there is only one carbon atom in the compound, this compound has originated from the methane series (CH). From the calculation, we can see there is one hydrogen atom and two fluorine atoms. The remaining valence bond of carbon will be balanced by chlorine. Thus, the substance is



Graphical Representation of Monochloro-Difluoro-Methane

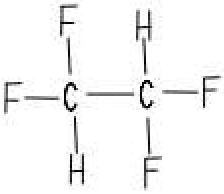
Therefore, chemical formula of R22 is $CHClF_2$ and has the name Monochloro-difluoro-methane. Taking again the example of R134, we can calculate its chemical formula as above which gives us

No. of C atoms: C – 1 = 1 => C = 2

No. of H atoms: H + 1 = 3 => H = 2

No. of F atoms: F = 4

Therefore, no. of Cl atoms: Cl = 0



Graphical Representation of Tetrafluoroethane

The compound is $C_2H_2F_2$ and its name is Tetrafluoroethane. The non-halogenated refrigerants follow a different naming convention which is dependent upon the series of the refrigerant.

5.14 DESIRABLE PROPERTIES OF REFRIGERANTS

The vast number of refrigerants available in the market today allows us to choose a refrigerant depending upon the operating conditions of the refrigeration system. As such, there is no refrigerant that can be advantageously used under all operating conditions and in all types of refrigeration systems. In spite of that, we can state certain desirable properties that a refrigerant should posses. These properties can be divided into favorable thermodynamic, chemical and physical properties:

5.14.1 Thermodynamic Properties

Critical Temperature and Pressure

The critical temperature of the refrigerant should be as high as possible above the condensing temperature in order to have a greater heat transfer at a constant temperature. If this is not taken care of, then we will have excessive power consumption by the refrigeration system. The critical pressure should be moderate and positive. A very high pressure will make the system heavy and bulky whereas in case of very low pressures, there is a possibility of air leaking into the refrigerating system.

Specific Heat

The specific heat of the liquid should be as small as possible. This ensures that the irreversibilities associated with throttling are small and there is greater subcooling of the liquid. On the other hand, the specific heat of vapor should be high to have less superheating of the vapor.

Enthalpy of Vaporization

This should be as large as possible to minimize the area under superheat and the area reduction due to throttling. Also, the higher value of enthalpy of vaporization lowers the required flow rate per ton of refrigeration.

Conductivity

The conductivity of the refrigerant should be as high as possible so that the size of the evaporator and condenser is manageable. From this viewpoint, ammonia has a better conductivity than that of R12 or R22 and is more suitable than the latter. But, ammonia is toxic and this does not allow its use in home refrigeration systems.

Evaporator and Condenser Pressure

Both the evaporator and condenser pressures need to be above atmospheric pressure otherwise there is a possibility of air leaking into the system. Presence of air drastically reduces the capacity of the refrigeration system. Also, due to presence of moisture in air, acids or other corrosive compounds may form and this may affect the tubing of the refrigeration system.

Compression Ratio

The compression ratio needs to be as small as possible otherwise the leakage of refrigerant occurs across the piston. Also, the volumetric efficiency is affected.

Freezing Point

It should be as low as possible or else there will be a possibility of blockage of passages during flow of fluid through evaporator.

Volume of Refrigerant Handled Per Ton of Refrigeration

This should be as small as possible in order to have a small size of the compressor. The type of compressor is decided by this value. For refrigerants like R12, R500, R22 etc., a reciprocating compressor is suitable. For others like R11 and water, a centrifugal compressor is required to handle the large volume.

Coefficient of Performance

The Coefficient of performance or COP has a direct bearing on the running cost of the refrigeration system. Higher the magnitude of COP, lower will be the running cost. Since, the COP of any refrigeration system is limited by the Carnot COP, for large operating pressures a multi-stage refrigeration system should be employed. CO_2 has a very low COP. Hence, it is not suitable for use as a refrigerant.

Density

The density of the refrigerant should be as large as possible. In reciprocating compressors, the pressure rise is accomplished by squeezing the entrapped fluid inside the piston-cylinder assembly. Hence, density decides the size of the cylinder. Again in centrifugal compressors pressure rise is related to the density of the vapor. A high value of density results in high pressure rise.

Compression Temperature

Whenever a refrigerant gets compressed, there is a rise in the temperature of the refrigerant resulting in the heating of the cylinder walls of the compressor. This necessitates external cooling of the cylinder walls to prevent volumetric and material losses. Refrigerants having lowest compression temperatures are thus better than others.

5.14.2 Chemical Properties

Chemical Stability and Inertness

It should be chemically stable for the operating ranges of temperature. Also, it should not react with the materials of the refrigeration system or with which it comes into contact. Further, it should be chemically inert and must not undergo polymerization reactions at either the lower or higher ranges of temperatures.

Action on Rubber or Plastics

Rubber and plastics are used extensively in the refrigeration system. These materials are mostly used in the seals and gaskets of the refrigeration system. They help to prevent the leakage of the refrigerant and ensure the smooth functioning of the compressor. The refrigerant should not react with them or else there might be leakage of refrigerant from the system or loss of functioning of the compressor.

Flammability

The refrigerant should be inert and not catch fire when subjected to high temperatures. From this viewpoint CO_2 is the most suitable as it is not only non-flammable, but also acts as a fire-extinguisher. Ethane, butane, isobutene are highly undesirable as they catch fire quickly.

Effect on Oil

The refrigerant should not react with the lubricating oil else, there is a possibility of loss of lubricating action due to either thickening or thinning of the oil. It should not be soluble in the oil else there will be reduction in the viscosity of the lubricating oil.

Effect on Commodity

If the refrigerant is directly used for chilling, then it should not affect the commodity kept in the conditioned space. Also, in case where direct cooling is not employed, the refrigerant should still not affect the commodity if there is any leakage.

Toxicity

The refrigerant used in air conditioning, food preservation etc. should not be toxic as they will come into contact with human beings.

5.14.3 Physical Properties

Leakage and Detection

Since pressures higher than atmospheric are usually employed in refrigeration systems, there is a possibility of leakage of refrigerants after long period of operation. It is desirable to detect this leak early else the system would operate under reduced capacity or stop functioning altogether. Hence, it is desirable that the refrigerant has a pungent smell so that its leakage can be detected immediately.

Miscibility with Oil

The refrigerant should not be miscible with the oil else the lubricating strength will be reduced.

Viscosity

It should be as small as possible to ensure that the pressure drop in the system is as small as possible. A low viscosity refrigerant will require less energy for its circulation through the refrigeration system.

5.14.4 Safety Criteria

Under safety criteria, we consider the toxicity, flammability, action on perishable food and formation of explosive compound on exposure to air. An ideal refrigerant should be non-toxic, non-flammable, have no effect on food products and should not react with atmospheric air. No refrigerant satisfy these criteria fully. We can therefore, group refrigerants into different sub-groups based on their flammability and toxicity levels.

5.14.5 Economic Criteria

Apart from the thermodynamic, chemical, physical and safety criteria, there is another criterion by which we judge an ideal refrigerant. The economic criterion takes into account the cost of the refrigerant, the availability and supply levels of the refrigerant, cost of storage and handling. We discuss each of these in detail below.

Cost of Refrigerant

The cost of the refrigerant has a big impact on the overall cost of the refrigeration system. Hence, its cost should be as low as possible. From this viewpoint, ammonia and water are ideally suited, but their low thermodynamic and chemical properties restrict their use in all types of refrigeration systems. Particularly, for flooded type evaporator or condenser, the refrigerant amount required is high and their cost needs to be factored in while making the initial investments.

Availability and Supply

The refrigerant should be easily available in the market and in abundant quantity. This ensures that the cost of the refrigerant is not prohibitive. An abundant and free supply of the refrigerant ensures that refrigeration systems will be designed specifically for use with them.

Storage and Handling

The refrigerant should be such that it can be conveniently stored and handled during transportation and charging. It should be stored in as small a pressure vessel as possible. Also, if we have to handle a toxic or flammable refrigerant, then the cost involved will be higher compared to handling and storage cost of non-toxic and non-flammable refrigerant.

5.15 COMMON REFRIGERANTS

The refrigerants which are available commercially in the market are numerous. Some of them which are in common use are mentioned below:

Air

Air (molecular weight 28.97, specific heats C*p* = 1.04 kJ/kgK and Cv = 0.712 kJ/kg-K) is one of the earliest refrigerant to be used in the refrigeration systems. Its advantages are that it is available free of cost, is non-toxic and non-flammable and does not affect the commodity if pure. However, air suffers from a number of drawbacks. Air contains moisture and this reacts with the material of the evaporator and condenser severely affecting their working capacity. Further, there is a possibility that the passages may be blocked by the formation of ice from this moisture. The COP of air is of the order of 0.6 and thus, not suitable for use in refrigeration systems on a commercial scale. It is mainly used for air conditioning in aircrafts where efficiency of operation is of secondary importance.

Ammonia

Ammonia (molecular weight 17) is one of the oldest refrigerants and it was commonly employed in places where toxicity effects were of secondary importance. Its advantages are its low cost, low specific volume, high COP (of the order of 4.0) and high refrigeration effect per unit mass of the refrigerant. Its primary drawback is its toxicity which prevents its use in air conditioning and food preservation systems. Ammonia has a boiling point of -33^{0}C at atmospheric pressure.

Carbon Dioxide

Carbon dioxide (molecular weight 44) is a non-toxic and non-poisonous refrigerant. Also, it is not only non-flammable but and is an excellent extinguishing agent as well. Its other advantages are that it is chemically stable, immiscible with the lubricating oil and does not affect the metal used in the system. It has a low specific volume and this requires volume displacement per ton of refrigeration. However, its critical pressure is too high. Also, its critical temperature is only 31^{0}C which makes it unsuitable for use in countries with a hot climate like India. It is an excellent refrigerant for low temperature refrigeration.

Sulphur Dioxide

Sulphur dioxide (molecular weight 64) is a colourless, suffocating and irritating gas and is twice as heavy as air at atmospheric conditions. It was mostly used as a household refrigerant in the older days, but has since been discarded for better refrigerants. It suffers from a lot of disadvantages. Sulphur dioxide reacts with water forming sulphurous acid, which in presence of oxygen becomes sulphuric acid, a corrosive compound for metals. It is non-flammable but attacks foodstuff on coming in contact with it. It is also partially miscible with the lubricating oil.

Hydrocarbons

This group consists of colourless fluids normally in gaseous state and made up of various combinations of carbon and hydrogen. Most of the refrigerants from this category are suitable for low temperature refrigeration. Isobutane falls in this category and has been suitable for domestic refrigeration. They are non-poisonous, but are flammable and highly explosive when exposed to air. The molecular weight and boiling point of each gas varies according to the number of hydrogen and carbon atoms. The larger the number of hydrogen and carbon atoms, the heavier is the gas and higher is its boiling point.

Halocarbon Refrigerants

The halocarbon refrigerants are formed by replacing one or more of hydrogen atoms of methane or ethane by one or more atoms of the three halogens: fluorine, chlorine or bromine. Some of the refrigerants coming under this category are mentioned below:

Refrigerant R12

The refrigerant R12 is the most widely used refrigerant in the domestic and large commercial establishments. Its chemical formula is CCl_2F_2 and its boiling point is -30^0C at 1 bar. It is a non-flammable, non-explosive, non-irritating, non-toxic and odourless refrigerant. It remains chemically stable up to 550^0C. Also, it does not affect the material of the refrigeration system. It is available in abundance and is quite cheap. However, its use is being discontinued nowadays for its contribution to ozone depletion which will be discussed later.

Refrigerant R13

Its chemical formula is $CClF_3$. It is a non-flammable, non-toxic and stable refrigerant. It is very suitable for achieving low temperatures in a cascade refrigeration system. Its specific volume is high and therefore, it is suitable for centrifugal compressors. However, it also has a negative effect on ozone depletion.

Refrigerant R22

Its chemical formula is $CHClF_2$. It is also a non-toxic, non-flammable, non-corrosive and non-irritating refrigerant. It is the most common refrigerant for use in large refrigeration systems and is preferred to R12.

Refrigerant R114

Its chemical formula is $C_2Cl_2F_4$. Its boiling point corresponding to 1 bar is about 3^0C. It has properties very similar to those of R12 with respect to water and oil combination. It is not suitable for low temperature refrigeration since it has negative evaporator pressure even at around 9^0C. It is non-toxic, non-explosive and non-corrosive even in the presence of water.

All refrigerants properties can be referred from standard refrigeration tables. Some refrigerant properties are listed below

5.16 VAPOUR ABSORPTION SYSTEM

Introduction

In a vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid. In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a 'generator'. The solubility of the refrigerant and absorbent must be suitable and the plant which uses ammonia as the refrigerant and water as absorbent will be described.

Simple Vapour Absorption System

Refer Fig. 5.12 for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution. The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.

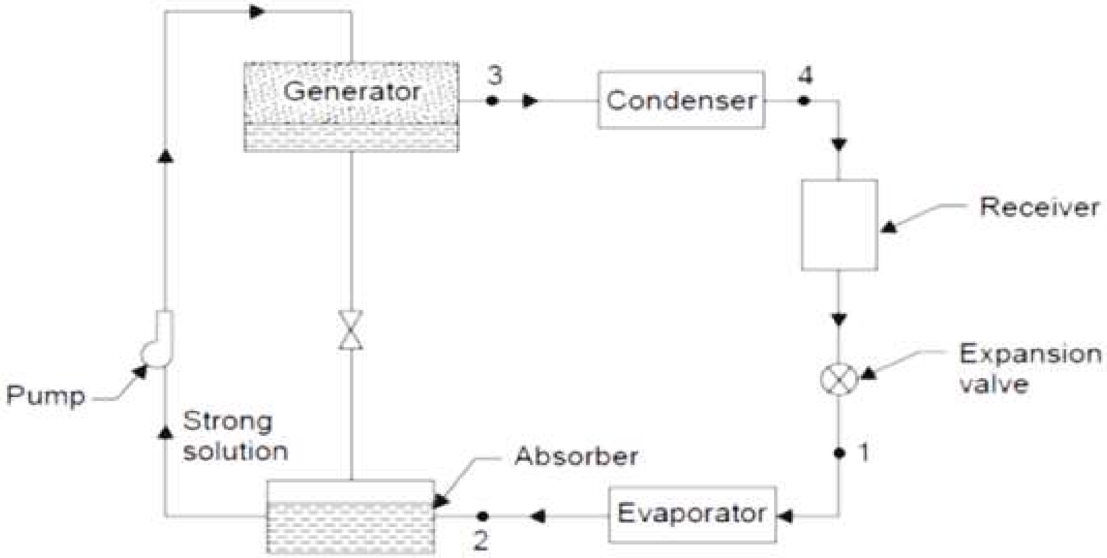
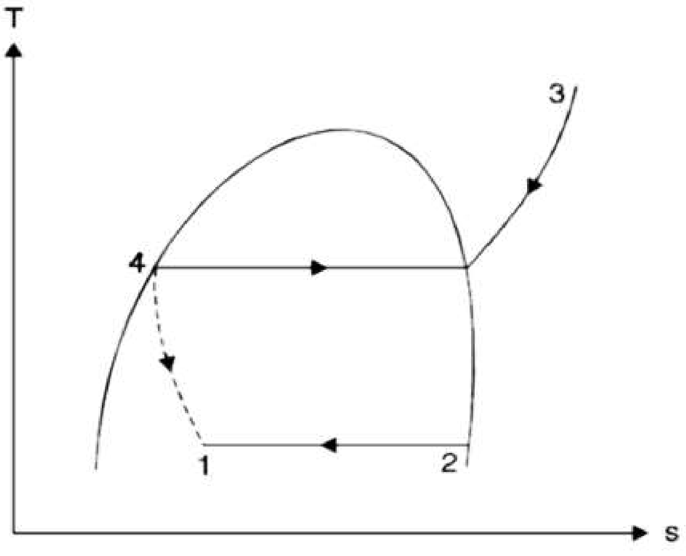


Fig 5.12



Simple vapour absorption system—*T-s* diagram.

Fig 5.13

In this system the work done on compression is less than in vapour compression cycle (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form e.g. steam or gas heating.

Practical Vapour Absorption System

Refer Fig. 5.14. Although a simple vapour absorption system can provide refrigeration yet its operating efficiency is low. The following accessories are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyser. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber.

The strong solution which is pumped from the absorber to the generator must be heated ; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently cost of heating the generator and cost of cooling the absorber are reduced.

2. **Analyser.** An analyser consists of a series of trays mounted above the generator. Its main function is to remove partly some of the unwanted water particles associated with ammonia vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze ; as a result the pipe line may get choked.

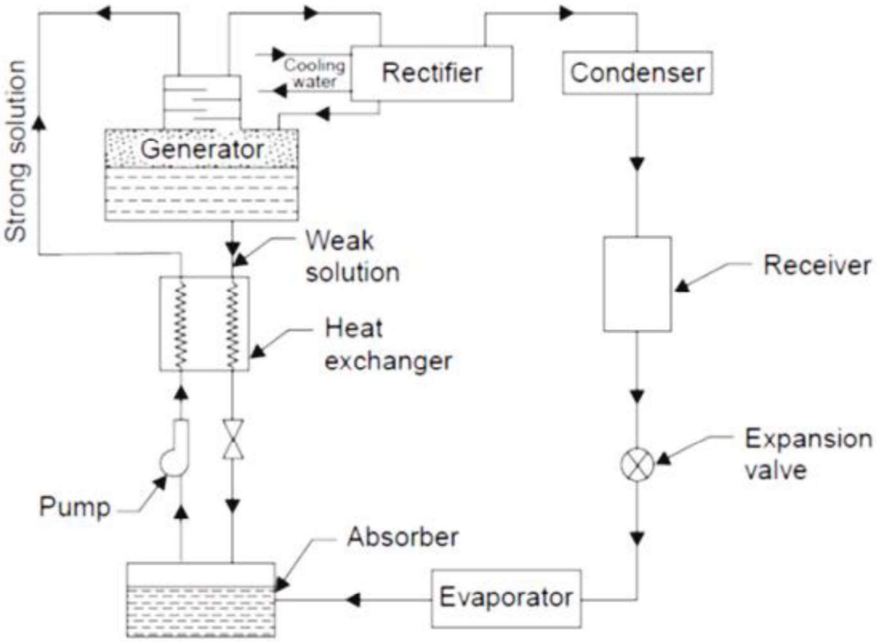


Fig 5.14

3. **Rectifier.** A rectifier is a water-cooled heat exchanger which condenses water vapour and some ammonia and sends back to the generator. Thus final reduction or elimination of the percentage of water vapour takes place in a rectifier. The co-efficient of performance (C.O.P.) of this system is given by :

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

5.17 PSYCHROMETRY AND AIR – CONDITIONING

Psychrometric properties, Use of psychrometric chart, Psychrometric process – Sensible heat exchange process, Latent heat exchange process, Adiabatic mixing, Evaporative cooling, Property calculations of air-vapour mixtures.

Principles of air-conditioning, Types of air conditioning systems – summer, winter, year round air conditioners, Concept of RSHF, GSHF, ESHF, Simple problems.

5.17.1 CONCEPT OF PSYCHROMETRY AND PSYCHROMETRICS

Air comprises of fixed gases principally, nitrogen and oxygen with an admixture of water vapour in varying amounts. In atmospheric air water is always present and its relative weight averages less than 1% of the weight of atmospheric air in temperate climates and less than 3% by weight under the most extreme natural climatic conditions, it is nevertheless one of most important factors in human comfort and has significant effects on many materials. Its effect on human activities is in fact altogether disproportionate to its relative weights. *The art of measuring* the *moisture content of air is termed* "**psychrometry**". *The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effect of atmospheric moisture on material and human comfort may properly be termed* "**psychrometrics**".

DEFINITIONS

Some of the more important definitions are given below :

1. **Dry air.** The international joint committee on Psychrometric Data has adopted the following exact composition of air expressed in mole fractions (Volumetric) Oxygen 0.2095, Nitrogen 0.7809, Argon 0.0093, Carbon dioxide 0.0003. Traces of rare gases are neglected. Molecular weight of air for all air conditioning calculations will be taken as 28.97. Hence the gas constant, *Rair* = 0.287 kJ/kg K Dry air is never found in practice. Air always contains some moisture. Hence the common designation "air" *usually means moist* air. The term '*dry air*' is used to indicate the *water free contents* of air having any degree of moisture.

2. **Saturated air.** Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it. For *a given temperature, a given quantity of air can be saturated with a fixed quantity of moisture.* At higher temperatures, it requires a larger quantity of moisture to saturate it. At saturation, vapour pressure of moisture in air corresponds to the saturation pressure given in steam tables corresponding to the given temperature of air.

3. **Dry-bulb temperature (DBT).** It *is the temperature of air as registered by an ordinary thermometer* (*tdb*).

4. **Wet-bulb temperature (WBT).** *It is the temperature registered by a thermometer when the bulb is covered by a wetted wick and is exposed to a current of rapidly moving air* (*twb*).

5. **Adiabatic saturation temperature.** *It is the temperature at which the water or ice can saturate air by evaporating adiabatically into it*. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) (*twb*).

6. **Wet bulb depression.** It *is the difference between dry-bulb and wet bulb temperatures* (*tdb* – *twb*).

7. **Dew point temperature (DPT).** *It is the temperature to which air must be cooled at constant pressure in order to cause condensation of any of its water vapour*. It is equal to steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (*tdp*).

8. **Dew point depression.** *It is the difference between the dry bulb and dew point temperatures* (*tdb – tdp*).

9. **Specific humidity (Humidity ratio).** *It is the ratio of the mass of water vapour per unit mass of dry air in the mixture of vapour and air*, it is generally expressed as grams of water per kg of dry air. For a given barometric pressure it is a function of dew point temperature alone.

10. **Relative humidity (RH), (φ).** *It is the ratio of the partial pressure of water vapour in the mixture to the saturated partial pressure at the dry bulb temperature, expressed as percentage.*

11. **Sensible heat.** *It is the heat that changes the temperature of a substance when added to or abstracted from it.*

12. **Latent heat.** *It is the heat that does not affect the temperature but changes the state of substance when added to or abstracted from it.*

13. **Enthalpy.** *It is the combination energy which represents the sum of internal and flow energy in a steady flow process*. It is determined from an arbitrary datum point for the air mixture and is expressed as kJ per kg of dry air (*h*).

Note. When air is saturated DBT, WBT, DPT are *equal.*

5.17.2 PSYCHROMETRIC RELATIONS

Pressure

Dalton's law of partial pressure is employed to determine the pressure of a mixture of gases. This law states that the total pressure of a mixture of gases is equal to the sum of partial pressures which the component gases would exert if each existed alone in the mixture volume at the mixture temperature. Precise measurements made during the last few years indicate that this law as well as Boyle's and Charle's laws are only approximately correct. Modern tables of atmospheric air properties are based on the correct versions. For calculating partial pressure of water vapour in the air many equations have been proposed, probably Dr. Carrier's equation is most widely used.

$$p_v = (p_{vs})_{wb} - \frac{[pt - (p_{vs})_{wb}](t_{db} - t_{wb})}{1527.4 - 1.3\, t_{wb}}$$

where p_v = Partial pressure of water vapour,
p_{vs} = Partial pressure of water vapour when air is fully saturated,
p_t = Total pressure of moist air,
t_{db} = Dry bulb temperature (°C), and
t_{wb} = Wet bulb temperature (°C).

Specific humidity W :

$$\text{Specific humidity} = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}}$$

or

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

Also,

$$m_a = \frac{p_a V}{R_a T}$$

and

$$m_v = \frac{p_v \times V}{R_v \times T}$$

where p_a = Partial pressure of dry air,

p_v = Partial pressure of water vapour,

V = Volume of mixture,

R_a = Characteristic gas constant for dry air, and

R_v = Characteristic gas constant for water vapour.

From equations (10.2) and (10.3)

$$W = \frac{p_v \times V}{R_v \times T} \times \frac{R_a T}{p_a V} = \frac{R_a}{R_v} \times \frac{p_v}{p_a}$$

But

$$R_a = \frac{R_0}{M_a} \qquad \left(= \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K in SI units} \right)$$

$$R_v = \frac{R_0}{M_v} \qquad \left(= \frac{8.3143}{18} = 0.462 \text{ kJ/kg K in SI units} \right)$$

where R_0 = Universal gas constant,

M_a = Molecular weight of air, and

M_v = Molecular weight of water vapour.

$$\therefore \qquad W = \frac{0.287}{0.462} \cdot \frac{p_v}{p_a} = 0.622 \; \frac{p_v}{p_t - p_v}$$

$$W = 0.622 \; \frac{p_v}{p_t - p_v}$$

Degree of saturation (μ) :

$$\text{Degree of saturation} = \frac{\text{Mass of water vapour associated with unit mass of dry air}}{\text{Mass of water vapour associated with saturated unit mass of dry saturated air}}$$

i.e.,
$$\mu = \frac{W}{W_s}$$

where, W_s = Specific humidity of air when air is fully saturated

$$\therefore \quad \mu = \frac{0.622\left(\dfrac{p_v}{p_t - p_v}\right)}{0.622\left(\dfrac{p_{vs}}{p_t - p_{vs}}\right)} = \frac{p_v(p_t - p_{vs})}{p_{vs}(p_t - p_v)}$$

$$= \frac{p_v}{p_s}\left[\frac{\left(1 - \dfrac{p_{vs}}{p_t}\right)}{\left(1 - \dfrac{p_v}{p_t}\right)}\right]$$

where p_{vs} = Partial pressure of water vapour when air is fully saturated (p_{vs} can be calculated from steam tables corresponding to the dry bulb temperature of the air).

Relative humidity (RH), φ :

$$\text{Relative humidity, } \phi = \frac{\text{Mass of water vapour in a given volume}}{\text{Mass of water vapour in the same volume if saturated at the same temp.}}$$

$$= \frac{m}{m_{vs}} = \frac{\dfrac{p_v\, T}{R_v\, T}}{\dfrac{p_{vs}\, T}{R_v\, T}} = \frac{p_v}{p_{vs}}$$

Enthalpy of moist air

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in kJ/kg of dry air

$$h = h_{\text{air}} + W \,.\, h_{\text{vapour}}$$
$$= c_p t_{db} + W \,.\, h_{\text{vapour}}$$

where h = Enthalpy of mixture/kg of dry air,

h_{air} = Enthalpy of 1 kg of dry air,

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables,

W = Specific humidity in kg/kg of dry air, and

c_p = Specific heat of dry air normally assumed as 1.005 kJ/kg K.

Also $$h_{\text{vapour}} = h_g + c_{ps} (t_{db} - t_{dp})$$

where h_g = Enthalpy of saturated steam at dew point temperature,

and c_{ps} = 1.88 kJ/kg K.

However, a better approximation is given by the following relationship :

$$h_{\text{vapour}} = 2500 + 1.88 t_{db} \text{ kJ/kg of water vapour}$$

where t_{db} is dry bulb temperature in ºC, and the datum state is liquid water at 0ºC.

$$\therefore \qquad h = 1.005\ t_{db} + W(2500 + 1.88\ t_{db}) \text{ kJ/kg dry air.}$$

5.18 PSYCHROMETRIC CHARTS

The psychrometric charts are prepared to represent graphically all the necessary moist air properties used for air conditioning calculations. The values are based on actual measurements verified for thermodynamic consistency. For psychrometric charts the most convenient co-ordinates are dry bulb temperature of air vapour mixture as the abcissa and moisture content (kg/kg of dry air) or water vapour pressure as the ordinate. Depending upon whether the humidity contents is abcissa or ordinate with temperature co-ordinate, the charts are generally classified as Mollier chart and Carrier chart. Carrier chart having *tdb* as the abcissa and *W* as the ordinate finds a wide application.

The chart is constructed as under :

1. *The dry bulb temperature* (ºC) of unit mass of dry air for different humidity contents or humidity ratios are indicated by *vertical lines drawn parallel to the ordinate.*

2. *The mass of water vapour* in kg (or grams) per kg of dry air is drawn parallel to the abcissa for different values of dry bulb temperature. It is the *major vertical scale of the chart.*

3. *Pressure of water vapour in mm of mercury* is shown in the scale at left and is the absolute pressure of steam.

4. *Dew point temperatures* are temperatures corresponding to the boiling points of water at low pressures of water vapour and are shown in the scale on the *upper curved line.* The dew points for different low pressures are read on *diagonal co-ordinates*

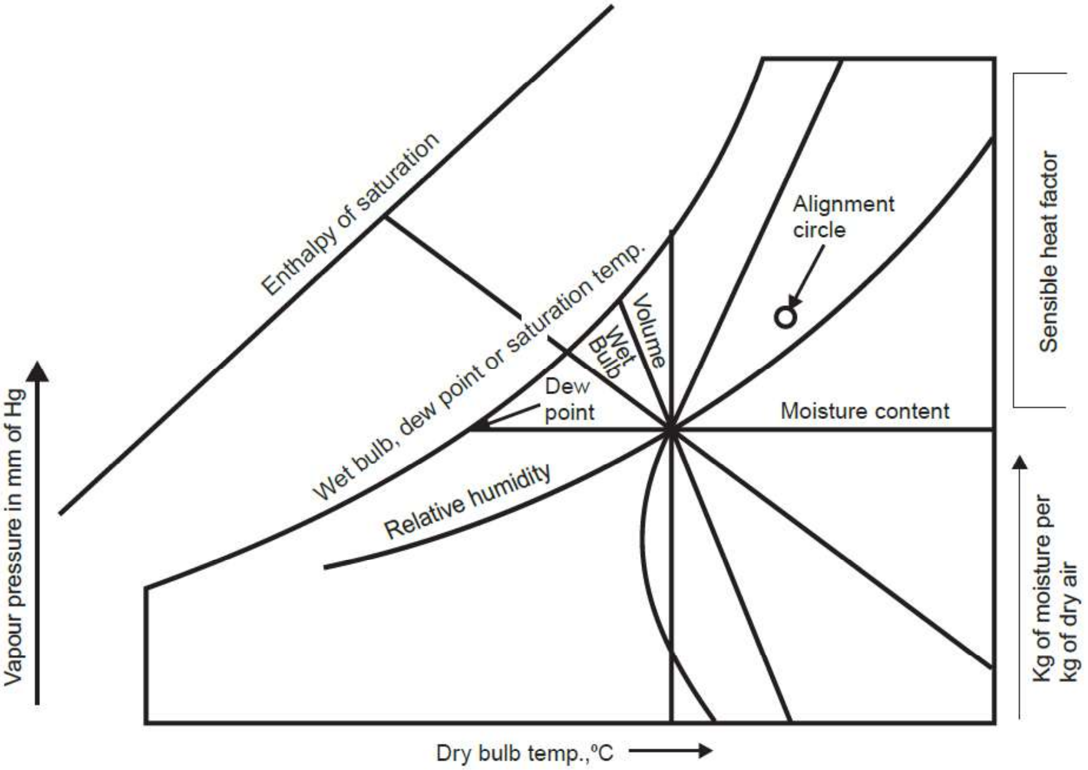


Fig 5.15

5. Constant relative humidity lines in per cent are indicated by marking off vertical distancesbetween the saturation line or the upper curved line and the base of the chart. The relative humidity curve depicts quantity (kg) of moisture actually present in the air as a percentage of the total amount possible at various dry bulb temperatures and masses of vapour.

6. *Enthalpy or total heat* at saturation temperature in kJ/kg of dry air is shown by a diagonal system of co-ordinates. The scale on the diagonal line is separate from the body of the chart and is indicated above the saturation line.

7. *Wet bulb temperatures* are shown on the diagonal co-ordinates coinciding with heat coordinates. The scale of wet bulb temperatures is shown on the saturation curve. The diagonals run downwards to the right at an angle of 30º to the horizontal.

8. The volume of air vapour mixture per kg of dry air (specific volume) is also indicated by a set of diagonal co-ordinates but at an angle of 60º with the horizontal. The other properties of air vapour mixtures can be determined by using formulae (already discussed).

In relation to the psychrometric chart, these terms can quickly indicate many things about the condition of air, for example :

1. If dry bulb and wet bulb temperatures are known, the relative humidity can be read from the chart.

2. If the dry bulb and relative humidity are known, the wet bulb temperature can be determined.

3. If wet bulb temperature and relative humidity are known, the dry bulb temperature can be found. 4. If wet bulb and dry bulb temperatures are known, the dew point can be found.

5. If wet bulb and relative humidity are known, dew point can be read from the chart.

6. If dry-bulb and relative humidity are known, dew point can be found.

7. The quantity (kg) of moisture in air can be determined from any of the following combinations :

(*i*) Dry bulb temperature and relative humidity ;

(*ii*) Dry bulb temperature and dew point ;

(*iii*) Wet bulb temperature and relative humidity ;

(*iv*) Wet bulb temperature and dew point temperature ;

(*v*) Dry bulb temperature and wet bulb temperature ; and

(*vi*) Dew point temperature alone.

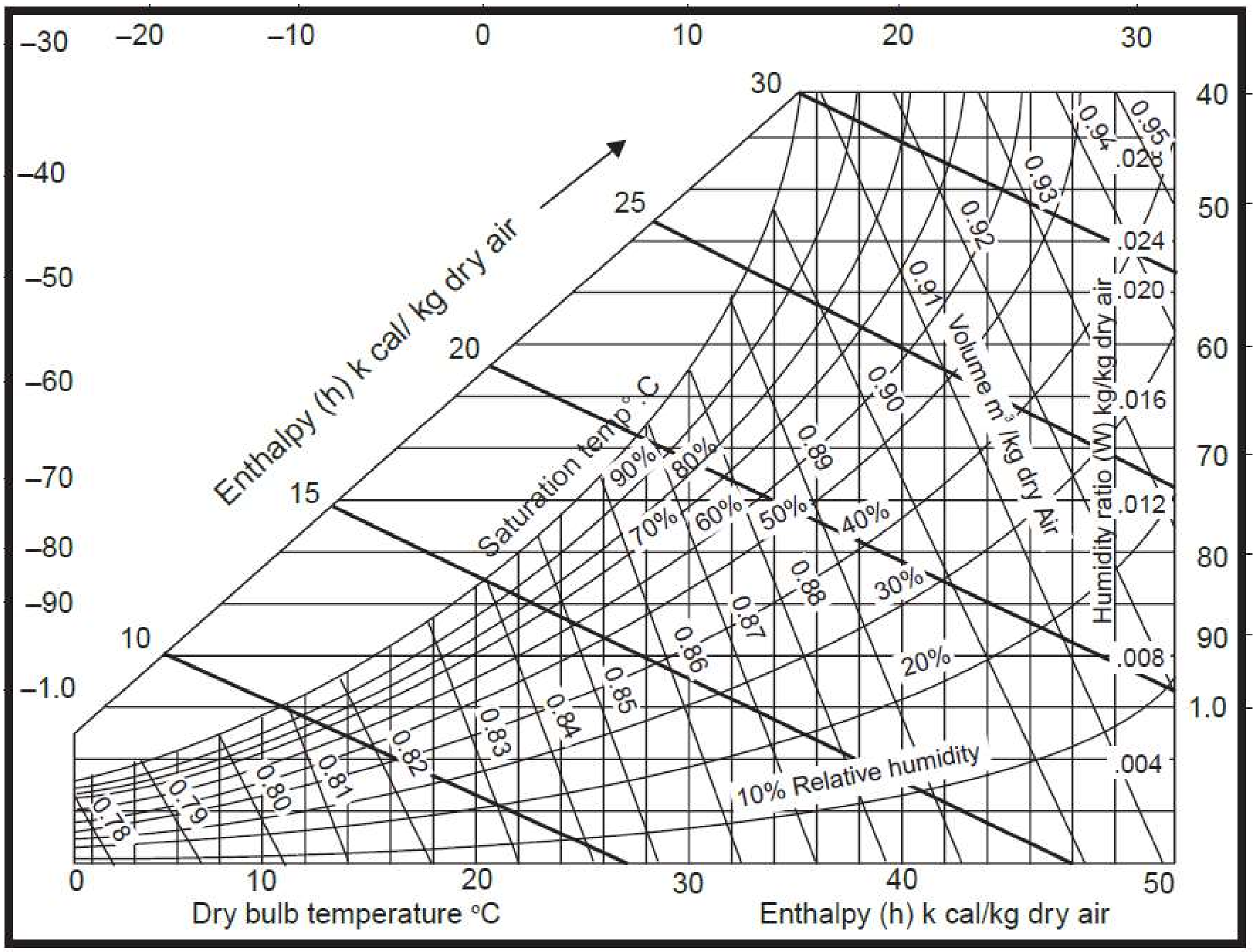


Fig 5.16 Carrier Psychrometric chart

5.19 PSYCHROMETRIC PROCESSES

In order to condition air to the conditions of human comfort or of the optimum control of an industrial process required, certain processes are to be carried out on the outside air available. The processes affecting the *psychrometric properties of air are called* ***psychrometric processes.***

These processes involve mixing of air streams, heating, cooling, humidifying, dehumidifying, adiabatic saturation and mostly the combinations of these.

The important psychrometric processes are enumerated and explained in the following text

1. Mixing of air streams
2. Sensible heating

3. Sensible cooling

4. Cooling and dehumidification

5. Cooling and humidification

6. Heating and dehumidification

7. Heating and humidification.

Mixing of Air Streams

Refer Figs. 5.17 and 5.18 Mixing of several air streams is the process which is very frequently used in air conditioning. This mixing normally takes place without the addition or rejection of either heat or moisture, i.e., adiabatically and at constant total moisture content. Thus we can write the following equations:

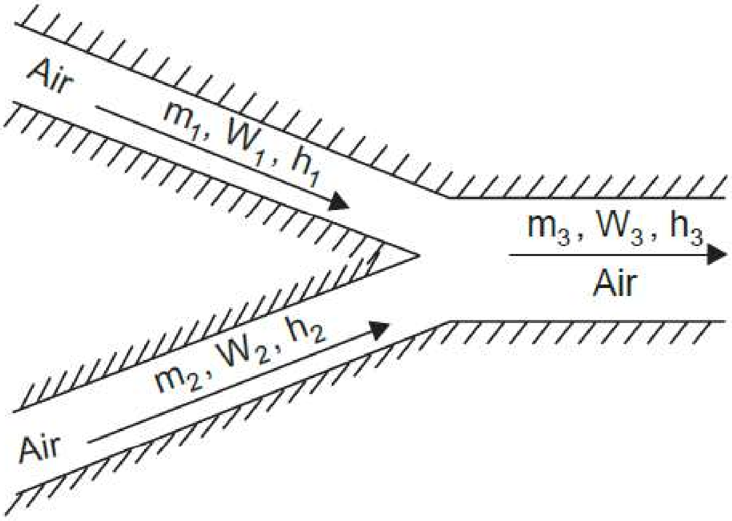


Fig 5.17 Mixing of Air streams

$$m_1 + m_2 = m_3$$
$$m_1 W_1 + m_2 W_2 = m_3 W_3$$
$$m_1 h_1 + m_2 h_2 = m_3 h_3$$

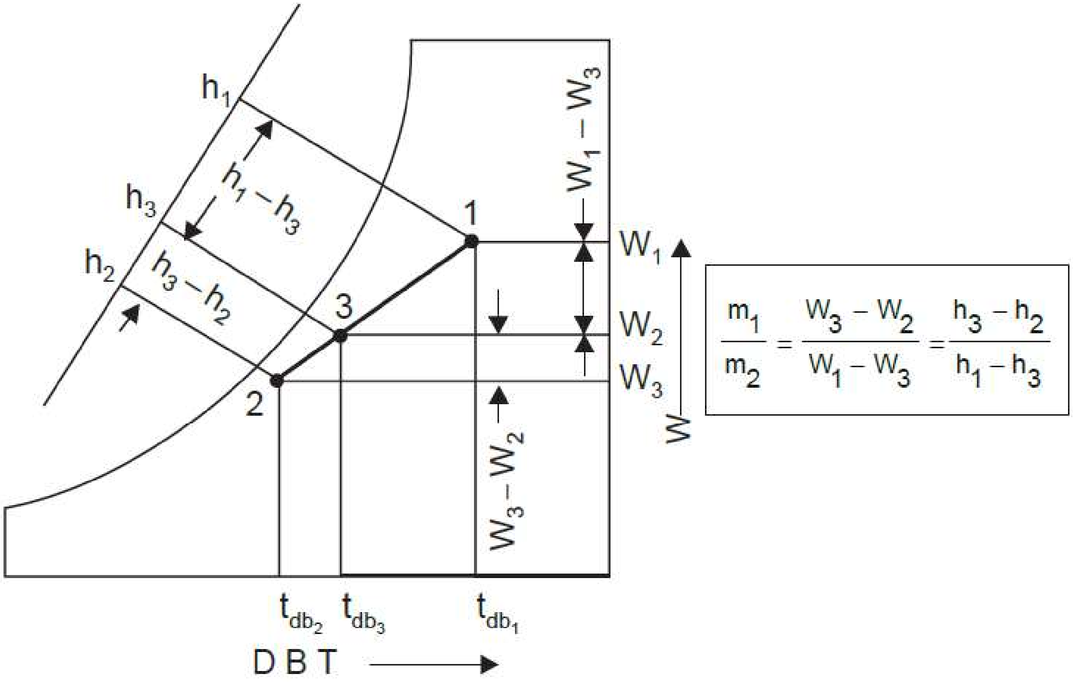


Fig 5.18

Rearranging of last two equations gives the following :

$$m_1(W_1 - W_3) = m_2(W_3 - W_2)$$
$$m_1(h_1 - h_3) = m_2(h_3 - h_2)$$
$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

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. Figure 5.19 shows the sensible cooling process O-A on a psychrometric chart. The heat transfer rate during this process is given by:

$$Q_c = m_a(h_O - h_A) = m_a c_{pm}(T_O - T_A)$$

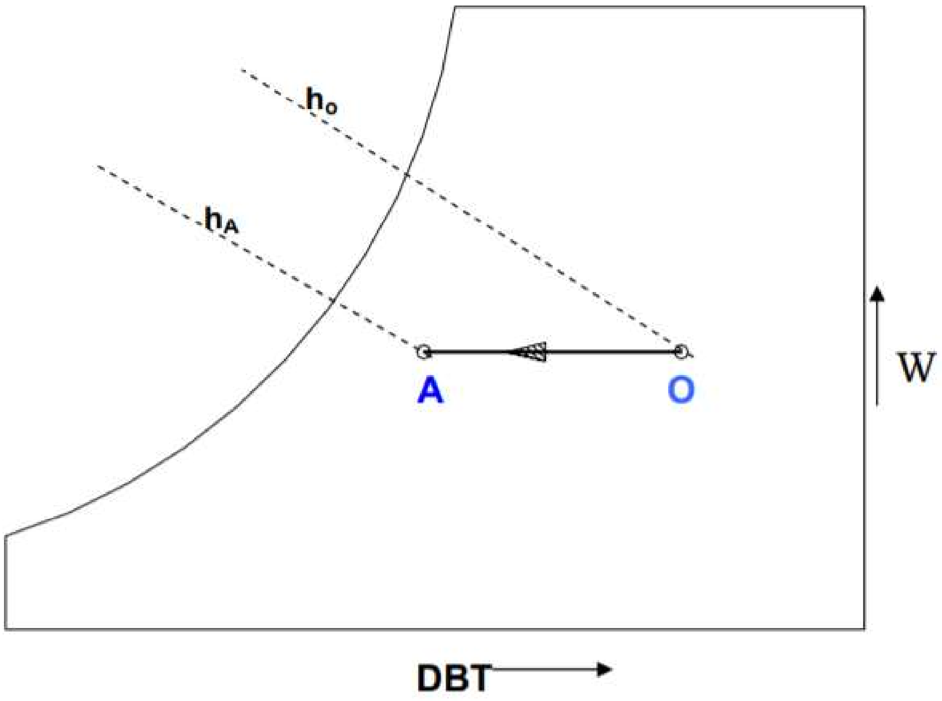


Fig 5.19 Sensible cooling process

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. The heat transfer rate during this process is given by:

$$Q_h = m_a(h_B - h_O) = m_a c_{pm}(T_B - T_O)$$

where c_{pm} is the humid specific heat (≈1.0216 kJ/kg dry air) and ma is the mass flow rate of dry air (kg/s). Figure 5.20 shows the sensible heating process on a psychrometric chart.

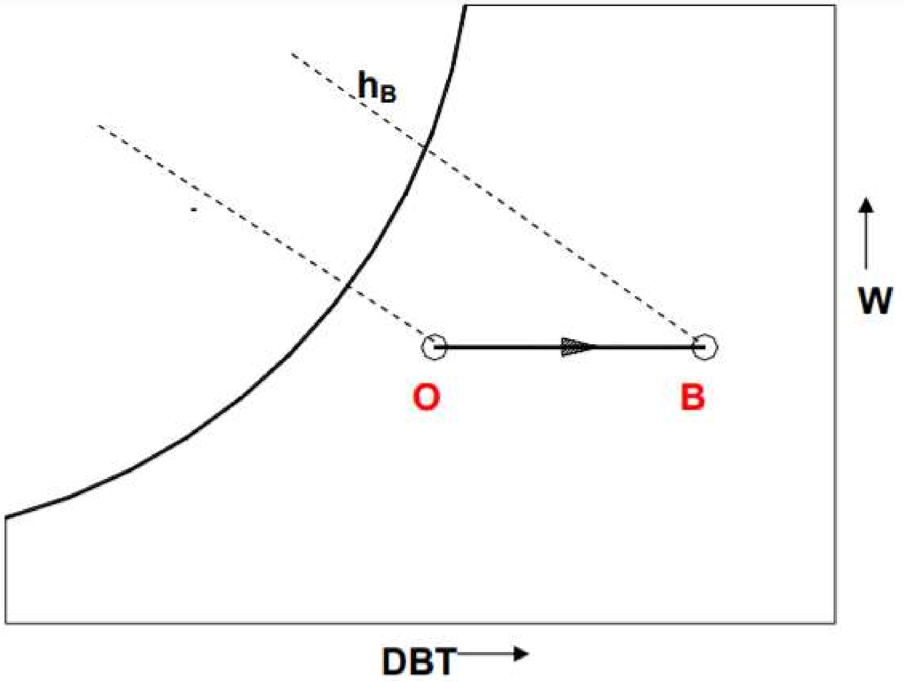


Fig 5.20 Sensible heating process

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.5.21, 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. 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 as given below: By applying mass balance for the water:

$$m_a.w_O = m_a.w_C + m_w$$

By applying energy balance:

$$m_a.h_O = Q_t + m_w.h_w + m_a.h_C$$

from the above two equations, the load on the cooling coil, Qt is given by:

$$Q_t = m_a(h_O - h_C) - m_a(w_O - w_C)h_w$$

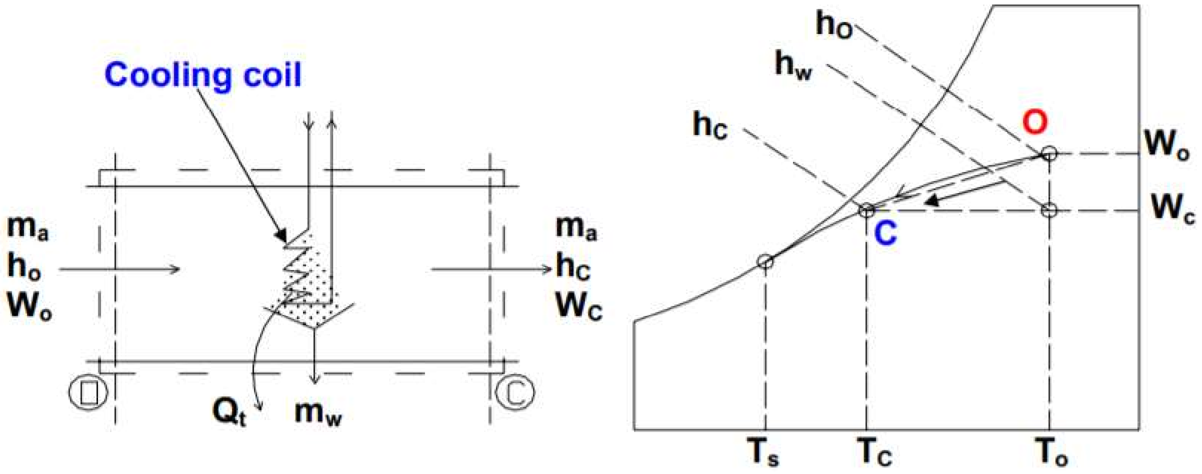


Fig 5.21 Cooling and Dehumidification process

the 2 term on the RHS of the above equation is normally small compared to the other terms, so it can be neglected. Hence,

$$Q_t = m_a(h_O - h_C)$$

It can be observed that the cooling and de-humidification process involves both latent and sensible heat transfer processes, hence, the total, latent and sensible heat transfer rates (Qt, Ql and Qs) can be written as:

$$\begin{aligned} &Q_t = Q_l + Q_s \\ \text{where}\quad &Q_l = m_a(h_O - h_w) = m_a.h_{fg}(w_O - w_C) \\ &Q_s = m_a(h_w - h_C) = m_a.c_{pm}(T_O - T_C) \end{aligned}$$

By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (SHF) is defined. SHF is defined as the ratio of sensible to total heat transfer rate, i.e.,

$$SHF = Q_s / Q_t = Q_s /(Q_s + Q_l)$$

From the above equation, one can deduce that a SHF of 1.0 corresponds to no latent heat transfer and a SHF of 0 corresponds to no sensible heat transfer. A SHF of 0.75 to 0.80 is quite common in air conditioning systems in a normal dry-climate. A lower value of SHF, say 0.6, implies a high latent heat load such as that occurs in a humid climate.

By pass factor

$$\mathbf{BPF} = \frac{T_C - T_S}{T_O - T_S}$$

It can be easily seen that, higher the by-pass factor larger will be the difference between air outlet temperature and the cooling coil temperature. When BPF is 1.0, all the air by-passes the coil and there will not be any cooling or de-humidification. In practice, the by-pass factor can be increased by increasing the number of rows in a cooling coil or by decreasing the air velocity or by reducing the fin pitch.

Alternatively, a contact factor(CF) can be defined which is given by:

CF=1-BPF

Heating and Humidification (Process O-D):

During winter it is essential to heat and humidify the room air for comfort. As shown in Fig.5.22., 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 figure.

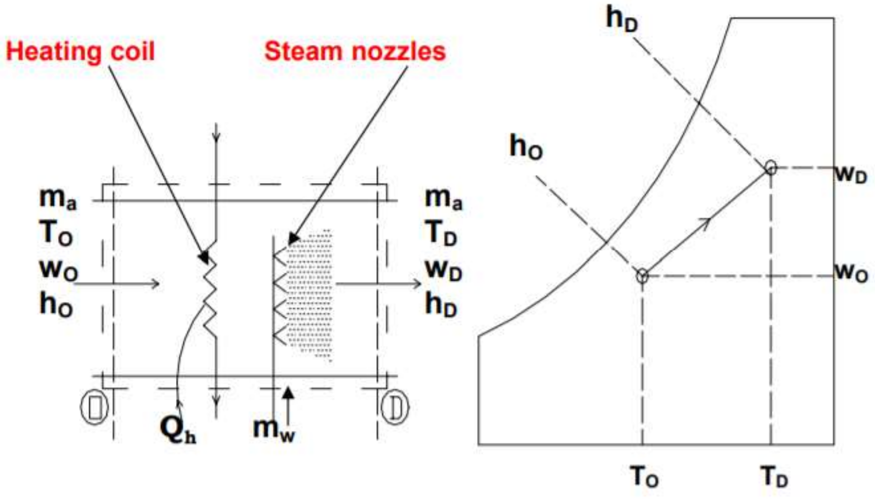


Fig 5.22

Mass balance of water vapor for the control volume yields the rate at which steam has to be added, i.e., mw:

$$m_w = m_a(w_D - w_O)$$

where ma is the mass flow rate of dry air.

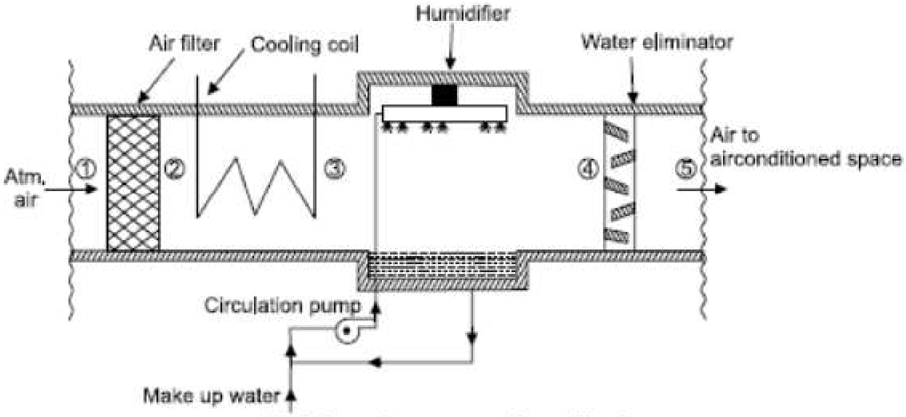
From energy balance:

$$Q_h = m_a(h_D - h_O) - m_w h_w$$

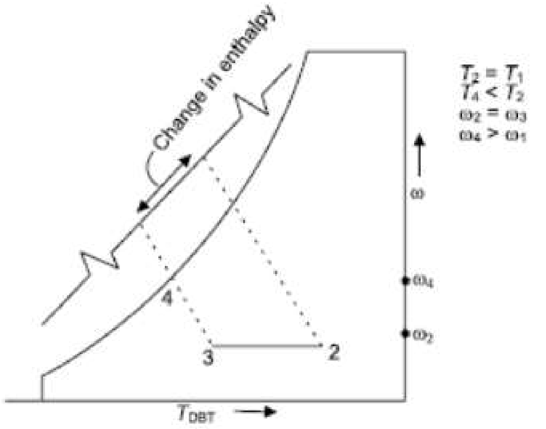
where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam. Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

5.20 AIR CONDITIONING SYSTEMS

Air conditioning systems require basic arrangement for getting refrigeration effect through cooling coil followed by subsequent humidification/dehumidification and heating etc. in order to provide air conditioned space with air at desired temperature and humidity. Air conditioning systems require different arrangements depending upon the atmospheric air condition and comfort condition requirement. Such as summer air conditioning systems and inter air conditioning systems are different. These systems have different arrangement if outdoor conditions are hot and humid, hot and dry etc. Summer air conditioning system for hot and dry outdoor condition is given in Fig. 5.23. Here the comfort conditions may require delivery of air to air-conditioned space at about 25ºC DBT and 60% relative humidity where the outdoor conditions may be up to 40–44º C DBT and 20% relative humidity in Indian conditions. Generic arrangement has air blower which blows air across the air filter between (1) and (2). Air coming out from filter passes over cooling coils and is subsequently sent for humidification between states (3) and (4). Large size water particles carried by air are retained by water eliminator. Air finally coming out at state (5) is sent to air conditioned space. Here psychrometric representation is made considering negligible change in humidity in water eliminator.



(*a*) Schematic summer airconditioning



(*b*) Representation on psychrometric chart

Fig 5.23 Summer air conditioning system

Winter Air Conditioning System:

In winter AC System, the inlet is heated by the heater, and in winter season due to less present in the air, we also need to add the moisture particle to the air, generally, a humidification system is added to maintain the moisture quantity.

Working of Winter Air Conditioning System:

In winter air conditioning, the air is heated and is accompanied by humidification. The outside air flows through a damper and mixes up with the recirculated air which is obtained from the conditioned space. The mixture here passes through a filter to remove dirt, dust, and other impurities.

The air now passes through a preheat coil to prevent possible freezing of water due to which dry bulb temperature increases to a very high value and the relative humidity drops to a low value.

This air is being pumped into the humidifier.

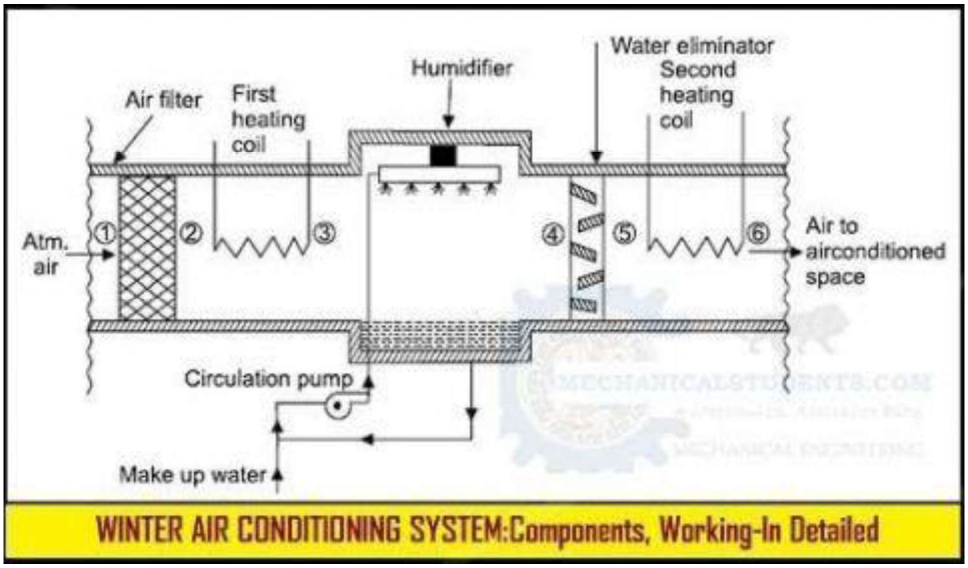


Fig 5.24 Winter air conditioning system

So, humidification of air (addition of moisture) is done and then the air is made to pass through a reheat coil to bring the air to the designed dry bulb temperature. Now the conditioned air is supplied to the conditioned space by mea fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the air known as recirculated air is again conditioned as shown in the figure. So it is again humidified due to which it reaches a point of 80% or 100% RH where the DBT is very low. So in order to get the desired dry bulb temperature, again the process of reheating is done where the desired percentage 40% RH is also obtained. A damper is used in order to control the area and have an intake of the required amount of air.

1. A 5 tonne refrigerator plant uses RR as refrigerant. It enters the compressor at -5°C as saturated vapour. Condension takes place at 32°C and there is no under cooling of refrigerant liquid. Assuming isentropic compression,determine COP of the plant, mass flow of refrigerant, power required to run the compressor in kw. The properties of R-12 are given table.

T(°C)	**P(bar)**	**Enthalpy(kw/kg)**		**Entropy(KJ/kgk)**
		hf	**hg**	**Sg**
32	**7.85**	**130.5**	**264.5**	**1.542**
-5	**2.61**	**-**	**249.3**	**1.557**

Solution:

Beginning of compression in dry and end of compression is superheated. So the P-h and T-S diagrams are

From table, at point 1

T_1=-5°C=268K

hg_1=2493kJ/Kg , Sg_1=1.557KJ/Kgk

At point 2

T_2=32°C=305k, hf_2=130.5KJ/Kg, hg_2=264.5, Sg_2=1.542KJ/KgK

From ph diagram, At point eqn(1) (dry).

At -5°C, i.e at 268k

hg_1=249.3KJ/Kg=hg

h_1=249.3KJ/Kg

At 32°C, i.e at 305K

hg_2=264.5KJ/Kg=h_2'

h_2'=264.5KJ/Kg

Entropy is constant during the compression process so,

$S_1=S_2$

From T- S diagram

At point (1) dry,

S_1=Sg at -5°C

$S_1=Sg_1$=1.557KJ/Kgk

$S_1=S_2$=1.557KJ/Kgk

At point (2) (super heated)

$S_1=S_2' + Cp \ln (T_2/T_2')$

$1.557=S_2'+1.884 \ln(T_2/305)$ --------(1)

S_2'=Sg at 32°C.

$Sg_2=1.542=S_2$

S_2'=1.542KJ/Kg k

$1.884 \ln (T_2/305)=0.015$

T_2=307.44k

For super heated vapour the enthalpy is

$h_2=h_2'+C_p(T_2-T_1')$

h_2= 264.5 + 1.884 (307.44-305)

h_2= 269.1 KJ/Kg

From P-h diagram, we know that,

$h_3= h_4$

$h_3=h_f$ at 32°C

$h_{f\,2}=130.5= h_f$

We Know that,

COP=Refrigeration effect / Work done= $(h_1-h_4) / (h_2-h_1)$

= (2493-130.5)/(269.1-249.3)=6

Refrigeration effect = m× (h1- h_4)

m= (2×210)/ (249.3-130.5)

m = 8.84Kg/min

Work done = Refrigeration effect/ cop

= (2×210)/6 = 175 KJ/min

Power = 2.92kw.

2. A refrigerator works between -7°C and 27°C the vapour is dry at the end of adiabatic compression. Assuming there is no under cooling determine (i) cop (ii) power of the compressor to remove a heat load of 12140KJ/hr.The properties of refrigerant are given in

T(°C)	sensible Heat (h_f)	Latent heat(h_{fg}) KJ/Kgk)	Entropy of liquid (KJ/Kgk)	Entropy of vapour Sg (KJ/Kgk)
-7	-29.3	1297.9	-0.109	4.748
27	1117.23	1172.3	0.427	4.333

table.

Solution:

The vapour is dry at end of compression i.e, beginning of compression is wet and of compression is dry saturated.

At point (1)

T_1=-7°C=266k, h_{fg1}=1297.9 KJ/Kg, S_{f1}=-0.109 KJ/KgK

h_{f1}=-29.3 KJ/Kg, S_{fg1}=4.478 KJ/KgK

At point (2)

T_2=27°C=300k, h_{fg2}=1172.3 KJ/Kg, S_{f2}= 0.427KJ/KgK

h_{f2}=117.23 KJ/Kg, S_{fg2}=4.333KJ/KgK

We point $S_1=S_2$

At point (1) (wet)

$S_1=S_{wet}=S_{f1}+x_1+S_{fg1}$

S_1= -0.109+x_1(S_{fg1}- S_{f1}) ($S_{fg}=S_g-S_f$)

S_1= -0.109+x_1(4.857)

At point (2) (dry)

$S_2=S_{g2}$=4.33KJ/KgK

S_2=4.33KJ/KgK

$S_1=S_2$ So, 4.33= -0.109+x_1(4.857)

Dryness fraction

x_1=0.913

At point (1) (wet)

$h_1= h_{f1}+ x_1\times h_{fg1}$

h_1=-29.3+0.913×1297.3

h_1=1156.3 KJ/Kg

At point (2) (dry)

$h_2= h_{f2}+h_{g2}$

h_2= 117.23+1172.3

h_2= 1289.53 KJ/Kg

From P-h diagram

$h_3=h_4$

$h_3=h_{f2}$

h_3=1172.3 KJ/Kg

h_4=117.23 KJ/Kg

COP= $(h_1-h_4)/(h_2-h_1)$ = (1156.3-117.23) / (1289.53-1156.3) = 7.7

Work done = Heat removed/ COP

= 12140/7.7

Power = 0.43 KJ/hr

3. Air enters the compressor of air craft system at 100kpa, 277k and is compressed to 300kpa with an isentropic efficiency of 72%. After being cooled to 328k and air expands is 100kpa and an η_{Isen}=78% the load is 3 tons and find COP, power, mass flow rate.

Given data:

P_1= 100kpa, T_3=38k

T_1= 277k, P_4=100kpa

P_2= 300kpa, η_T=78%

η_c= 72%

Solution:

process 1-2 Isentropic compression

$T_2= (P_2/P_1)^{\gamma-1/\gamma} \times T_1$

$T_2= (300/100)^{1.4-1/1.4}$

T_2=379.14k

$\eta_c=(T_2-T_1)/(T_2'-T_1)$

0.72= (379.14-277)/ (T_2'-277)

T_2=418.86k

Process 3-4 isentropic compression

$T_3/T_4= (P_3/P_4)^{\gamma-1/\gamma}$

$328/T_4= (300/100)^{\gamma-1/\gamma}$

$T_4=239.64k$

$\eta_t=(T_3-T_4')/(T_3-T_4)$

$0.78= (328-T_4')/ (328-239.64)$

$T_4'= 259.08k$

$COP= (T_1-T_4')/ (T_2'-T_1)$

$COP= (277-259.08)/ (418.86-277) =0.17$

1 tonne= 3.5kw of heat

3tonne= 3×3.5=10.5kw

Energy balance.

Heat energy absorbed by I_{ce}=Heat rejected by air

$= m\times C_p\times (T_1-T_4')$

$10.5= m_a\times 1.005\times(277-259.08)$

Mass of air, $m_a= 0.583$Kg/sec.

Power, $P=m_a\times Cp_a\times (T_2'-T_1)$

$= 0.583\times 1.005\times (418.86-277)$

$= 83.12kW$

4. **An ammonia refrigerator process 20tons of ice per day from and at 0°C.The condensation and evaporation takes at 20°C and -20°C respectively the temperature of the vapour at the end of Isentropic compression is 50°C and there is no under cooling of the liquid.COP=70% of theoretical COP. Determine (i) Rate of NH_3 circulation (ii) size of compressor, N=240rpm,L=D, η_{vol}=80%. Take Laten heat of I_{cc}=335kJ/Kg, C_p= 2.8 kJ/Kg,**

V_{s1}=0.624m^{3}kg. Use the following properties of ammonia.

Sat.Temp(°C)	**Enthalpy(kJ/Kg)**		**Entropy(kJ/Kgk)**		
	h_f	**h_g**	**S_f**	**S_g**	
20	**274.98**	**1461.58**	**1.0434**	**5.0919**	
20	**89.72**	**1419.05**	**0.3682**	**5.6204**	**[Apr 2003]**

Given data: 20 tons of Icc per day at °C

T_3= 20°C

T_1=-20°C

T3=50°C

COP =70% of theoretical cop

N=240rpm

L=D

h_v=80%

Latent heat of Ice=335KJ/Kg

C_p=2.8KJ/Kgk

V_{s1}=0.624m^3/kg

Solution:-

The refrigeration effect=20×3.5=77.55kw

h_1=1419.05KJ/Kg

h_{g2}=1461.58KJ/Kg at 20°C

h_{f3}=274.98 KJ/Kg

h_2=h_{g2}+C_p (T_2-20)

h_2=1461.58+2.8(50-20)

h_2=1545.58 KJ/Kg

COP= $(h_1-h_{f3})/(h_1-h_2)$

= (1419.05-274.98)/(1545.58-1419.05)

=9.04.

5. Explain the working principle of vapor compression refrigeration system with neat sketch.

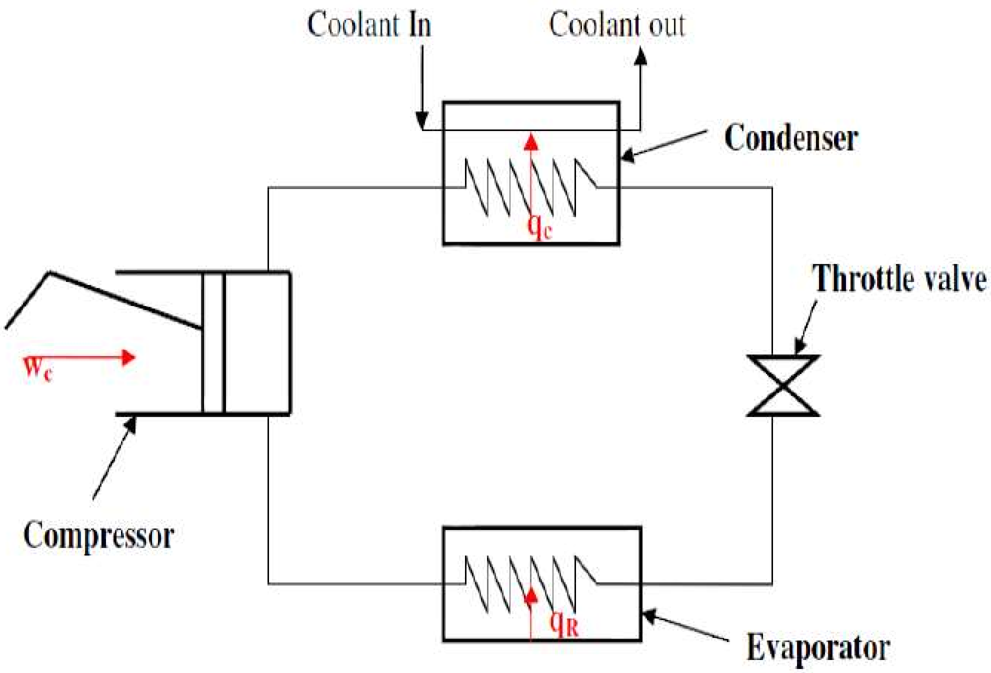
Schematic layout of vapour compression system

Process 1-2: Isentropic compression of the refrigerant from state 1 to state 2. During this process work done is done on the refrigerant by the surroundings. At the end of the process the refrigerant will be in super heated vapour state.

Process 2-3: Constant pressure condensation of the refrigerant in the condenser till it becomes a saturated liquid.

Process 3-4: Throttling expansion of the refrigerant from condenser pressure to the evaporator pressure. Process 4-1: Constant pressure vapourisation of the refrigerant in the evaporator till it becomes a dry saturated vapour. During this process heat is absorbed by the refrigerant from the place to be refrigerated. Applying steady flow steady state energy equation to the evaporator and neglecting the changes in kinetic and potential energies we have Refrigeration effect = QR = m (h1 – h4) Since process 3-4 is a throttling process, h4 = h3. Hence QR = m (h1 – h3) Similarly, by applying steady flow, steady state energy equation to compressor we get Compressor work input = Wc = m (h2 – h1)

Hence COP = Qr / Wc = (h1 – h4) / (h2 – h1)



Advantages of Vapour compression refrigeration system over air refrigeration system: Since the working cycle approaches closer to carnot cycle, the C.O.P is quite high. Operational cost of vapour compression system is just above 1/4th of air refrigeration system. Since the heat

removed consists of the latent heat of vapour, the amount of liquid circulated is less and as a result the size of the evaporator is smaller. Any desired temperature of the evaporator can be achieved just by adjusting the throttle valve.

Disadvantages of Vapour compression refrigeration system over air refrigeration system:

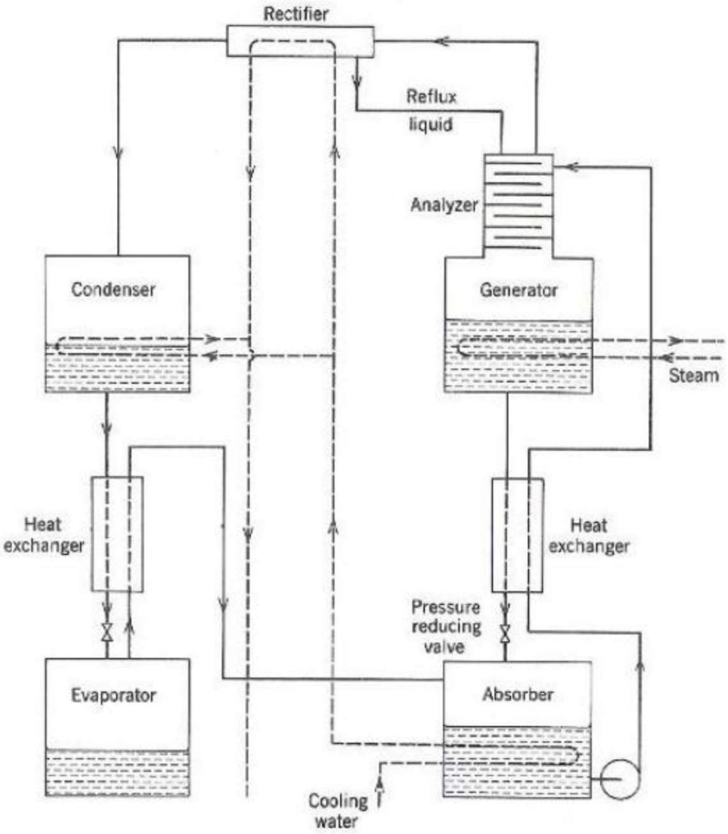
Initial investment is high Prevention of leakage of refrigerant is a major problem

6. Explain and working principles of Ammonia – water vapour absorption refrigeration system with neat sketch. [May 2011]

1) Evaporator: It is in the evaporator where the refrigerant pure ammonia (NH3) in liquid state produces the cooling effect. It absorbs the heat from the substance to be cooled and gets evaporated. From here, the ammonia passes to the absorber in the gaseous state.

2) Absorber: In the absorber the weak solution of ammonia-water is already present. The water, used as the absorbent in the solution, is unsaturated and it has the capacity to absorb more ammonia gas. As the ammonia from evaporator enters the absorber, it is readily absorbed by water and the strong solution of ammonia-water is formed. During the process of absorption heat is liberated which can reduce the ammonia absorption capacity of water; hence the absorber is cooled by the cooling water. Due to absorption of ammonia, strong solution of ammonia-water is formed in the absorber.

3) Pump: The strong solution of ammonia and water is pumped by the pump at high pressure to the generator.



Schematic layout of Ammonia –water vapour absorption system

4) Generator: The strong solution of ammonia refrigerant and water absorbent are heated by the external source of heat such as steam or hot water. It can also be heated by other sources like natural gas, electric heater, waste exhaust heat etc. Due to heating the refrigerant ammonia gets vaporized and it leaves the generator. However, since water has strong affinity for ammonia and its vaporization point is quite low some water particles also get carried away with ammonia refrigerant, so it is important to pass this refrigerant through analyzer.

5) Analyzer: One of the major disadvantages of the ammonia-water vapor absorption refrigeration system is that the water in the solution has quite low vaporizing temperature, hence when ammonia refrigerant gets vaporized in the generator some water also gets vaporized. Thus the ammonia refrigerant leaving the generator carries appreciable amount of water vapor. If this water vapor is allowed to be carried to the evaporator, the capacity of the refrigeration system would reduce. The water vapor from ammonia refrigerant is removed by analyzer and the rectifier. The analyzer is a sort of the distillation column that is located at the top of the generator. The analyzer consists of number of plates positioned horizontally. When the ammonia refrigerant along with the water vapor particles enters the analyzer, the solution is cooled. Since water has higher saturation temperature, water vapor gets condensed into the water particles that drip down into the generator. The ammonia refrigerant in the gaseous state continues to rise up and it moves to the rectifier.

6) Rectifier or the reflex condenser: The rectifier is a sort of the heat exchanger cooled by the water, which is also used for cooling the condenser. Due to cooling the remaining water vapor mixed with the ammonia refrigerant also gets condensed along with some particles of ammonia. This weak solution of water and ammonia drains down to the analyzer and then to the generator.

7) Condenser and expansion valve: The pure ammonia refrigerant in the vapor state and at high pressure then enters the condenser where it is cooled by water. The refrigerant ammonia gets converted into the liquid state and it then passes through the expansion valve where its temperature and pressure falls down suddenly. Ammonia refrigerant finally enters the evaporator, where it produces the cooling effect. This cycle keeps on repeating continuously. Meanwhile, when ammonia gets vaporized in the generator, weak solution of ammonia and water is left in it. This solution is expanded in the expansion valve and passed back to the absorber and its cycle repeats.

7. Comparison between vapour compression and vapour absorption systems.

Compression systems	Absorption systems
Work operated	Heat operated
High COP	Low COP
Performance very sensitive to evaporator temperatures.	Performance not very sensitive to evaporator temperatures.
System COP reduces considerably at part loads.	COP does not reduce significantly with load.
Liquid at the exit of evaporator may damage compressor.	Presence of liquid at evaporator exit is not a serious problem.
Performance of sensitive to evaporator superheat.	Evaporator superheat is not very important.
Many moving parts	Very few moving parts
Regular maintenance required	Very low maintenance required
Higher noise and vibration	Less noise and vibration.

8. List out the Properties of Refrigerants

Toxicity: It obviously desirable that the refrigerant have little effect on people.

Inflammability: Although refrigerants are entirely sealed from the atmosphere, leaks are bound to develop. If the refrigerant is inflammable and the system is located where ignition of the refrigerant may occur, a great hazard is involved.

Boiling Point: An ideal refrigerant must have low boiling temperature at atmospheric pressure.

Freezing Point: An ideal refrigerant must have a very low freezing point because the refrigerant should not freeze at low evaporator temperatures.

Evaporator and condenser pressure: In order to avoid the leakage of the atmosphere air and also to enable the detection of the leakage of the refrigerant, both the Evaporator and condenser pressure should be slightly above the atmosphere pressure.

Chemical Stability: An ideal refrigerant must not decompose under operating conditions.

Latent heat of Evaporation: The Latent heat of Evaporation must be very high so that a minimum amount of refrigerant will accomplish the desired result; in other words, it increases the refrigeration effect.

Specific Volume: The Specific Volume of the refrigerant must be low. The lower specific volume of the refrigerant at the compressor reduces the size of the compressor.

Specific heat of liquid vapour: A good refrigerant must have low specific heat when it is in liquid state and high specific heat when it is vaporized.

Viscosity: The viscosity of the refrigerant t both the liquid and vapour state must be very low as improved the heat transfer and reduces the pumping pressure.

Corrosiveness: A good refrigerant should be non-corrosive to prevent the corrosion of the metallic parts of the refrigerator.

Odour: A good refrigerant must be odour less, otherwise some foodstuff such as meat, butter, etc loses their taste.

Oil solvent properties: A good refrigerant must be not react with the lubricating oil used in the refrigerator for lubricating the parts of the compressor.

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