

THERMAL SYSTEMS

SMEX1044

UNIT III: REFRIGERATION AND AIR CONDITIONING

SIMPLE VAPOUR COMPRESSION SYSTEM

Introduction

Out of all refrigeration systems, the vapour compression system is the most important system from the view point of commercial and domestic utility. It is the most practical form of refrigeration. In this system the working fluid is a vapour. It readily evaporates and condenses or changes alternately between the vapour and liquid phases without leaving the refrigerating plant. During evaporation, it absorbs heat from the cold body. This heat is used as its latent heat for converting it from the liquid to vapour. In condensing or cooling or liquifying, it rejects heat to external body, thus creating a cooling effect in the working fluid. This refrigeration system thus acts as a latent heat pump since it pumps its latent heat from the cold body or brine and rejects it or delivers it to the external hot body or cooling medium. The principle upon which the vapour compression system works apply to all the vapours for which tables of Thermodynamic properties are available.

Simple Vapour Compression Cycle

In a simple vapour compression system fundamental processes are completed in one cycle

These are:

1. Compression
2. Condensation
3. Expansion
4. Vapourisation.

The flow diagram of such a cycle is shown in Fig. 1

The vapour at low temperature and pressure (state '2') enters the "compressor" where it is compressed isentropically and subsequently its temperature and pressure increase considerably (state '3'). This vapour after leaving the compressor enters the "condenser" where it is condensed into high pressure liquid (state '4') and is collected in a "receiver tank". From receiver tank it passes through the "expansion valve", here it is throttled down to a lower pressure and has a low temperature (state '1'). After finding its way through expansion "valve" it finally passes on to "evaporator" where it extracts heat from the surroundings or circulating fluid being refrigerated and vapourises to low pressure vapour (state '2').

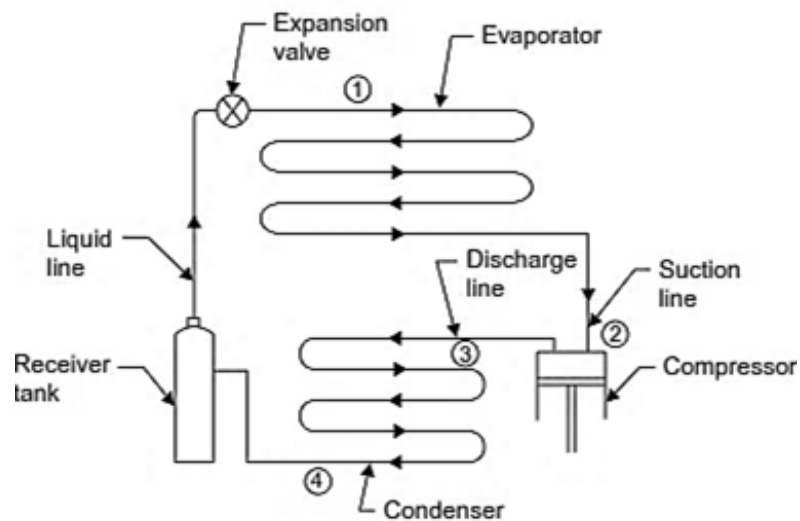


Fig. 1. Vapour compression system.

Merits and demerits of vapour compression system over Air refrigeration system:

Merits:

1. C.O.P. is quite high as the working of the cycle is very near to that of reversed Carnot cycle.
2. When used on ground level the running cost of vapour-compression refrigeration system is only 1/5th of air refrigeration system.
3. For the same refrigerating effect the size of the evaporator is smaller.
4. The required temperature of the evaporator can be achieved simply by adjusting the throttle valve of the same unit.

Demerits:

1. Initial cost is high.
2. The major disadvantages are inflammability, leakage of vapours and toxicity. These have been overcome to a great extent by improvement in design.

Functions of Parts of a Simple Vapour Compression System

Here follows the brief description of various parts of a simple vapour compression system shown in Fig. 1.

- 1. Compressor.** The function of a compressor is to remove the vapour from the evaporator, and to raise its temperature and pressure to a point such that it (vapour) can be condensed with available condensing media.
- 2. Discharge line (or hot gas line).** A hot gas or discharge line delivers the high-pressure, high-temperature vapour from the discharge of the compressor to the condenser.
- 3. Condenser.** The function of a condenser is to provide a heat transfer surface through which heat passes from the hot refrigerant vapour to the condensing medium.
- 4. Receiver tank.** A receiver tank is used to provide storage for a condensed liquid so that a constant supply of liquid is available to the evaporator as required.
- 5. Liquid line.** A liquid line carries the liquid refrigerant from the receiver tank to the refrigerant flow control.
- 6. Expansion valve (refrigerant flow control).** Its function is to meter the proper amount of refrigerant to the evaporator and to reduce the pressure of liquid entering the evaporator so that liquid will vapourize in the evaporator at the desired low temperature and take out sufficient amount of heat.
- 7. Evaporator.** An evaporator provides a heat transfer surface through which heat can pass from the refrigerated space into the vapourizing refrigerant.
- 8. Suction line.** The suction line conveys the low pressure vapour from the evaporator to the suction inlet of the compressor.

Vapour Compression Cycle on Temperature-Entropy (T-s) Diagram

We shall consider the following three cases:

- 1. When the vapour is dry and saturated at the end of compression.** Fig. 2 represents the vapour compression cycle, on T-s diagram the points 1, 2, 3 and 4 correspond to the state points 1, 2, 3 and 4 in Fig. 1.

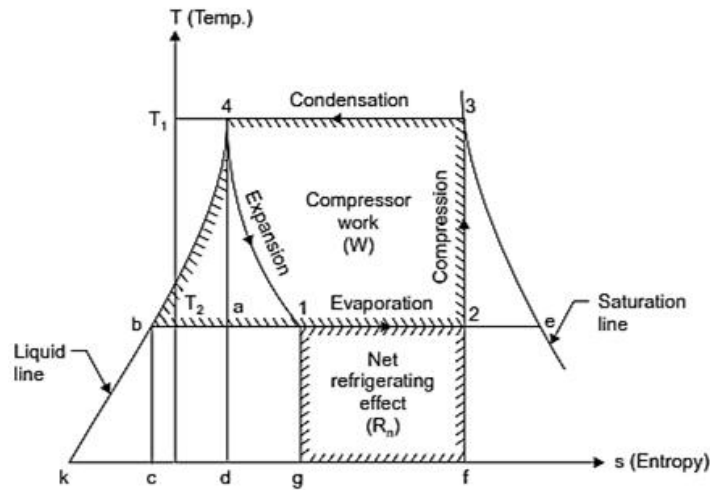


Fig. 2.T-s diagram.

At point '2' the vapour which is at low temperature (T_2) and low pressure enters the compressor's cylinder and is compressed adiabatically to '3' when its temperature increases to the temperature T_1 . It is then condensed in the condenser (line 3-4) where it gives up its latent heat to the condensing medium. It then undergoes throttling expansion while passing through the expansion valve and it's again reduces to T_1 , it is represented by the line 4-1. From the T-s diagram it may be noted that due to this expansion the liquid partially evaporates, as its dryness fraction is represented by the ratio (b_1/b_2). At '1' it enters the evaporator where it is further evaporated at constant pressure and constant temperature to the point '2' and the cycle is completed.

Work done by the compressor = W = Area '2-3-4-b-2'

Heat absorbed = Area 2-1-g-f-2'

$$\text{C.O.P.} = \frac{\text{Heat extracted or refrigerating effect}}{\text{Work done}} = \frac{\text{Area '2-1-g-f-2'}}$$

$$\begin{aligned} \text{C.O.P.} &= \frac{h_2 - h_1}{h_3 - h_2} \\ &= \frac{h_2 - h_4}{h_3 - h_2} \end{aligned}$$

($h_1 = h_4$, since during the throttling expansion 4-1 the total heat content remains unchanged)

2. When the vapour is superheated after compression. If the compression of the vapour is continued after it has become dry, the vapour will be superheated, and its effect on T-s diagram is shown in Fig. 3. The vapour enters the compressor at condition '2' and is compressed to '3' where it is superheated to temperature T_{sup} . Then it enters the condenser. Here firstly superheated vapour cools to temperature T_1 (represented by line 3-3') and then it condenses at constant temperature along the line 3'-4; the remaining of the cycle ; however is the same as before.

Now, Work done = Area 2-3-3'-4-b-2

and Heat extracted/absorbed = Area 2-1-g-f-2

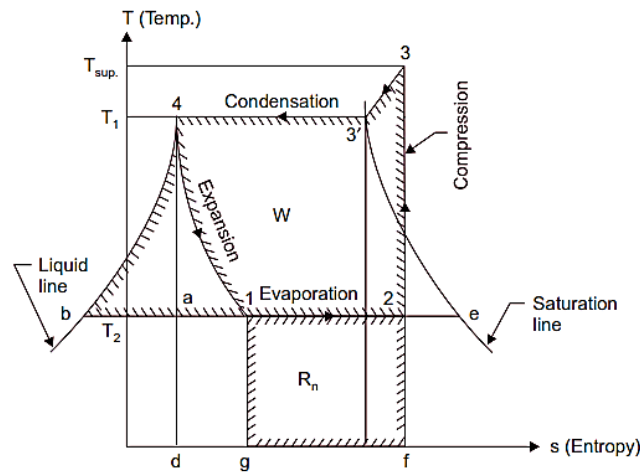


Fig. 3.T-s diagram.

$$\text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-g-f-2'}}{\text{Area '2-3-3'-4-b-2'}} = \frac{h_2 - h_1}{h_3 - h_2}$$

In this case $h_3 = h_3 + c_p(T_{sup} - T_{sat})$ and h_3 = total heat of dry and saturated vapour at the point '3'.

3. When the vapour is wet after compression. Refer Fig. 4

Work done by the compressor = Area 2-3-4-b-2

Heat extracted = Area 2-1-g-f-2

$$\text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-g-f-2'}}{\text{Area '2-3-4-b-2'}} = \frac{h_2 - h_1}{h_3 - h_2}$$

Note. If the vapour is not superheated after compression, the operation is called 'WET COMPRESSION' and if the vapour is superheated at the end of compression, it is known as 'DRY COMPRESSION'. Dry compression, in actual practice is always preferred as it gives higher volumetric efficiency and mechanical efficiency and there are less chances of compressor damage.

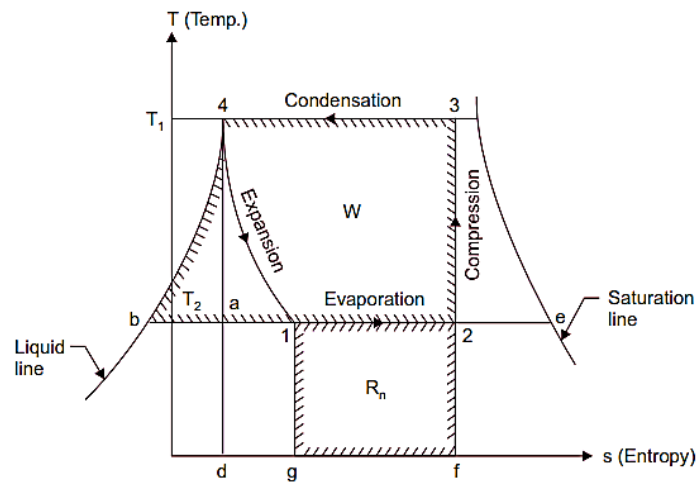


Fig. 4 T-s diagram.

Simple Vapour Compression Cycle on p-h Chart

Fig. 5 shows a simple vapour compression cycle on a p-h chart. The points 1, 2, 3 and 4 correspond to the points marked in Fig. 1

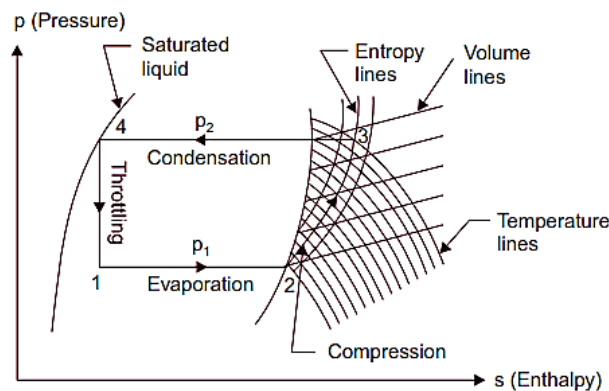


Fig. 5. Simple vapour compression cycle on p-h chart.

The dry saturated vapour (at state 2) is drawn by the compressor from evaporator at lower pressure p_1 and then it (vapour) is compressed isentropically to the upper pressure p_2 . The isentropic compression is shown by the line 2-3. Since the vapour is dry and saturated at

the start of compression it becomes superheated at the end of compression as given by point 3. The process of condensation which takes place at constant pressure is given by the line 3-4. The vapour now reduced to saturated liquid is throttled through the expansion valve and the process is shown by the line 4-1. At the point 1 a mixture of vapour and liquid enters the evaporator where it gets dry saturated as shown by the point 2. The cycle is thus completed.

Heat extracted (or refrigerating effect produced),

$$R_n = h_2 - h_1$$

$$W = h_3 - h_1$$

$$\text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

The values of h_1 , h_2 and h_3 can be directly read from p-h chart.

Factors Affecting the Performance of a Vapour Compression System

The factors which affect the performance of a vapour compression system are given below :

1. **Effect of suction pressure.** The effect of decrease in suction pressure is shown in Fig. 6.

The C.O.P. of the original cycle,

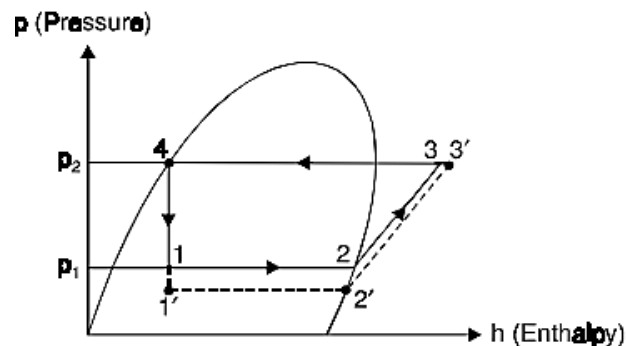


Fig. 6. Effect of decrease in suction pressure.

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

The C.O.P. of the cycle when suction pressure is decreased,

$$\begin{aligned} \text{C.O.P.} &= \frac{h_2' - h_1'}{h_3' - h_2'} \\ &= \frac{(h_2 - h_1) - (h_2 - h_2')}{(h_3 - h_2) + (h_2 - h_2') + (h_3' - h_3)} \\ &\quad (\because h_1 = h_1') \end{aligned}$$

This shows that the refrigerating effect is decreased and work required is increased. The net effect is to reduce the refrigerating capacity of the system (with the same amount of refrigerant flow) and the C.O.P.

2. **Effect of delivery pressure.** Fig. 7 shows the effect of increase in delivery pressure.

C.O.P. of the original cycle,

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

C.O.P. of the cycle when delivery pressure is increased,

$$\text{C.O.P.} = \frac{h_2 - h_1'}{h_3' - h_2} = \frac{(h_2 - h_1) - (h_1' - h_1)}{(h_3 - h_2) + (h_3' - h_3)}$$

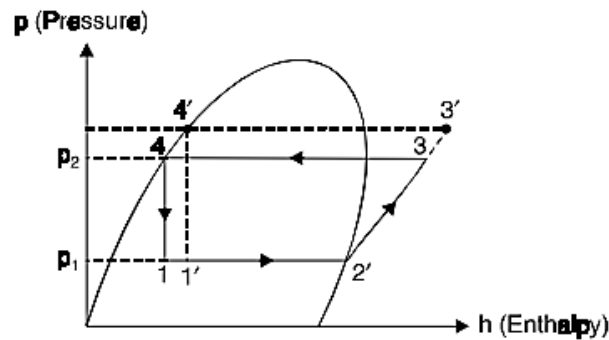


Fig. 7 Effect of increase in delivery pressure

The effect of increasing the delivery/discharge pressure is just similar to the effect of decreasing the suction pressure. The only difference is that the effect of decreasing the suction pressure is more predominant than the effect of increasing the discharge pressure.

The following points may be noted :

- (i) As the discharge temperature required in the summer is more as compared with winter, the same machine will give less refrigerating effect (load capacity decreased) at a higher cost.
- (ii) The increase in discharge pressure is necessary for high condensing temperatures and decrease in suction pressure is necessary to maintain low temperature in the evaporator.

Effect of superheating. As may be seen from the Fig. 8 the effect of superheating is to increase the refrigerating effect but this increase in refrigerating effect is at the cost of increase in amount of work spent to attain the upper pressure limit. Since the increase in work is more as compared to increase in refrigerating effect, therefore overall effect of superheating is to give a low value of C.O.P.

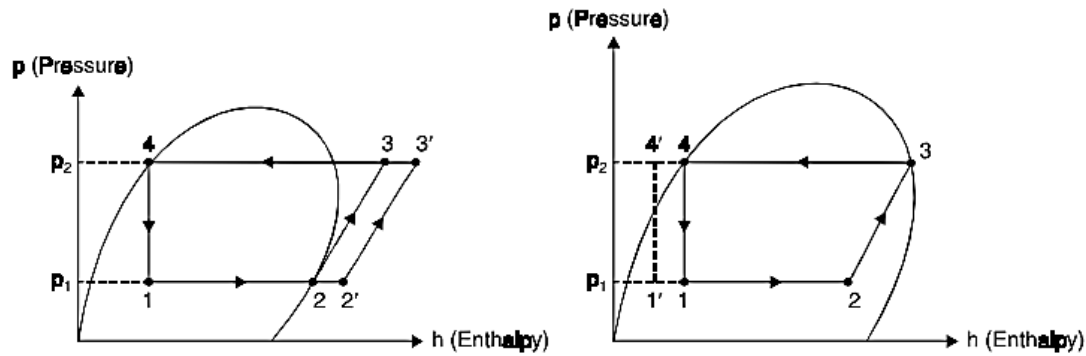


Fig. 8.Effect of superheating.Fig. 9.Effect of sub-cooling of liquid.

4. Effect of sub-cooling of liquid. ‘Sub-cooling’ is the process of cooling the liquid refrigerant below the condensing temperature for a given pressure . In Fig. 14.18 the process of subcooling is shown by 4-4' . As is evident from the figure the effect of subcooling is to increase the refrigerating effect. Thus sub-cooling results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.

The sub-cooling or undercooling may be done by any of the following methods:

- (i)Inserting a special coil between the condenser and the expansion valve.
- (ii)Circulating greater quantity of cooling water through the condenser.
- (iii)Using water cooler than main circulating water.

5. Effect of suction temperature and condenser temperature. The performance of the vapour compression refrigerating cycle varies considerably with both vapourising and condensing temperatures. Of the two, the vapourising temperature has far the greater effect. It is seen that the capacity and performance of the refrigerating system improve as the vapourising temperature increases and the condensing temperature decreases . Thus refrigerating system should always be designed to operate at the highest possible vapourising temperature and lowest possible condensing temperature, of course, keeping in view the requirements of the application.

VAPOUR ABSORPTION SYSTEM

Introduction

In 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. 10 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.

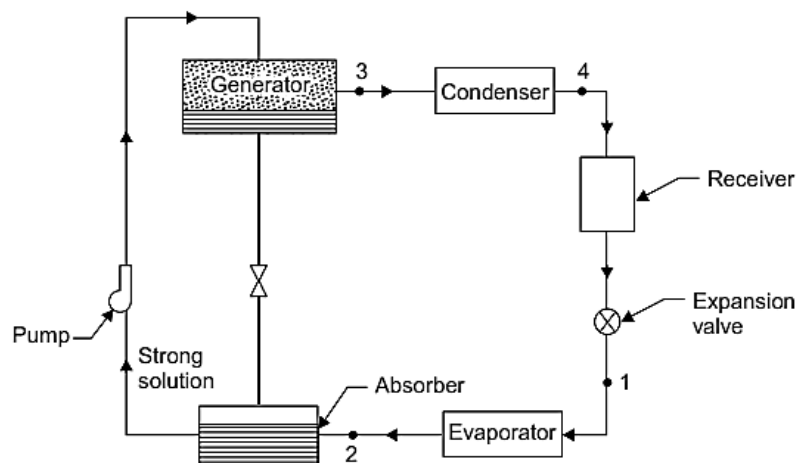


Fig. 10. (a) Simple vapour absorption system

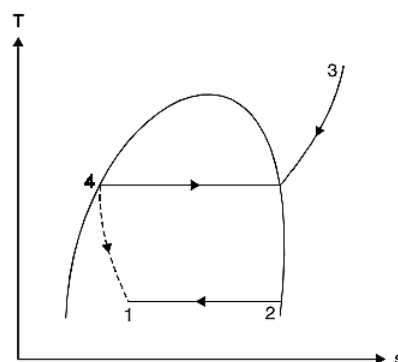


Fig. 10. (b) Simple vapour absorption system—T-s diagram

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. 11. 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.

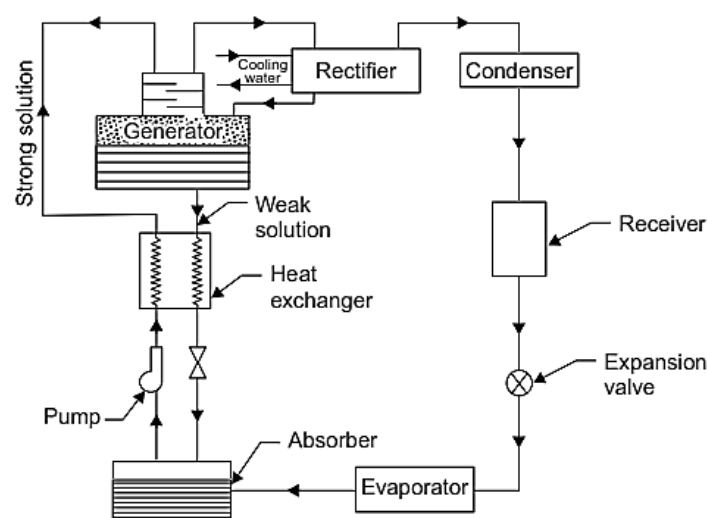


Fig. 11 Practical Vapour Absorption System

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 :

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

COMPARISON BETWEEN VAPOUR COMPRESSION AND VAPOUR ABSORPTION SYSTEMS

S. No.	Particulars	Vapour compression system	Vapour absorption system
1	Type of energy supplied	Mechanical-a high grade energy	Mainly heat-a low grade energy
2	Energy supply	Low	High
3	Wear and tear	More	Less
4	Performance at Part loads.	Poor	System not affected by variationsloads
5	Suitability	Used where high grade mechanical energy is available	Can also be used at remote places as it can work even with a simple kerosene lamp (of course in small capacities)
6	Charging of refrigerant	Simple	Difficult
7	Leakage of refrigerant	More chances	No chance as there is no compressor or any reciprocating component to cause leakage.
8	Damage	Liquid traces in suction line may damage the compressor	Liquid traces of refrigerant present in piping at the exit ofevaporator constitute no danger.

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,

$$R_{air} = \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K}$$

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 (t_{db}).

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 (t_{wb}).

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) ($t_{db} - t_{wb}$).

6. **Wet bulb depression.** It is the difference between dry-bulb and wet bulb temperatures.

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 (t_{dp}).

8. **Dew point depression.** It is the difference between the dry bulb and dew point temperatures ($t_{db} - t_{dp}$).

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.

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 Charles'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.

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

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:

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

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

$$\text{Also, } m_a = \frac{p_a V}{R_a T}$$

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

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.

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

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

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

Where

R_0 = Universal gas constant,

M_a = Molecular weight of air, and

M_v = Molecular weight of water vapour.

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

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

The masses of air and water vapour in terms of specific volumes are given by expression as

$$m_a = \frac{V}{v_a} \quad \text{and} \quad m_v = \frac{V}{v_v}$$

Where

v_a = Specific volume of dry air, and

v_v = Specific volume of water vapour.

$$W = \frac{v_a}{v_v}$$

Degree of saturation (μ):

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

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

W_s = Specific humidity of air when air is fully saturated

$$\begin{aligned}\mu &= \frac{0.622 \left(\frac{p_v}{p_t - p_v} \right)}{0.622 \left(\frac{p_{vs}}{p_t - p_{vs}} \right)} = \frac{p_v(p_t - p_{vs})}{p_{vs}(p_t - p_v)} \\ &= \frac{p_v}{p_s} \left[\frac{\left(1 - \frac{p_{vs}}{p_t} \right)}{\left(1 - \frac{p_v}{p_t} \right)} \right]\end{aligned}$$

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 :

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

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

$$\phi = \frac{p_a W}{0.622} \times \frac{1}{p_{vs}} = 1.6 W \frac{p_a}{p_{vs}}$$

Note 1. Relative humidity as compared to specific humidity plays a vital role in comfort air-conditioning and industrial air-conditioning. Relative humidity signifies the absorption capacity of air. If initial relative humidity of air is less it will absorb more moisture.

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2. W , μ and ϕ cannot be conveniently measured as they require measurement of p_v and p_{vs} . The value of p_v can be obtained from the measurement of the wet bulb temperature and the value of p_{vs} can be calculated from steam tables corresponding to given air temperature.

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.

$$h = h_{\text{air}} + W \cdot h_{\text{vapour}}$$

$$= c_p t_{db} + W \cdot h_{\text{vapour}}$$

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.

$$\therefore h = c_p t_{db} + W[h_g + c_{ps}(t_{db} - t_{dp})]$$

$$= (c_p + c_{ps} W) t_{db} + W(h_g - c_{ps} t_{dp})$$

$$= c_{pm} t_{db} + W(h_g - c_{ps} t_{dp})$$

Where $C_{pm} = (C_p + C_{ps} W)$ is the specific heat of humid air or humid specific heat.

The value of C_{pm} is taken as 1.021 kJ/kg dry air per K. It is the heat capacity of $(1 + W)$ kg of moisture per kg of dry air.

$h_{\text{vapour}} = h_g$ at dry bulb temperature. So,

$$h = c_p t_{db} + W h_g.$$

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 °C, and the datum state is liquid water at 0°C.

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

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 abscissa and moisture content (kg/kg of dry air) or water vapour pressure as the ordinate. Depending upon whether the humidity contents are abscissa or

ordinate with temperature co-ordinate, the charts are generally classified as Mollier chart and Carrier chart. Carrier chart having t_{db} as the abscissa and W as the ordinate finds a wide application. The chart is constructed as under :

1. The dry bulb temperature ($^{\circ}\text{C}$) 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 abscissa 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.

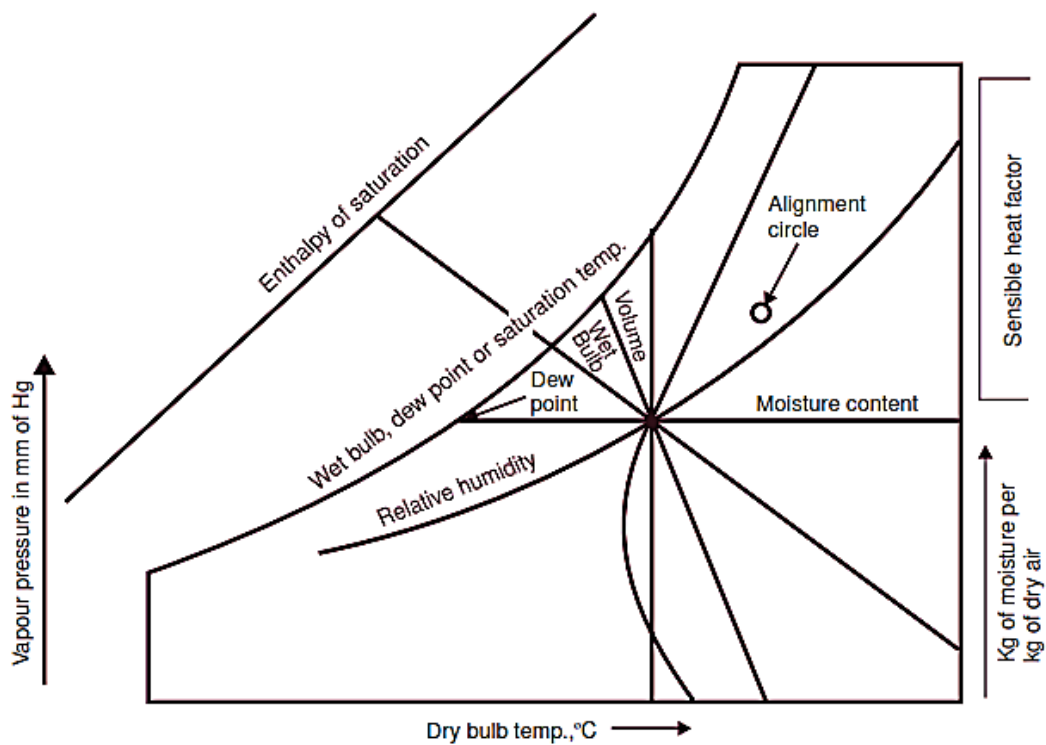


Fig.a. Skeleton psychrometric chart.

5. Constant relative humidity lines in per cent are indicated by marking off vertical distances between 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 30° 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 60° 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.

Figs. a and b show the skeleton psychrometric chart and lines on carrier chart respectively.

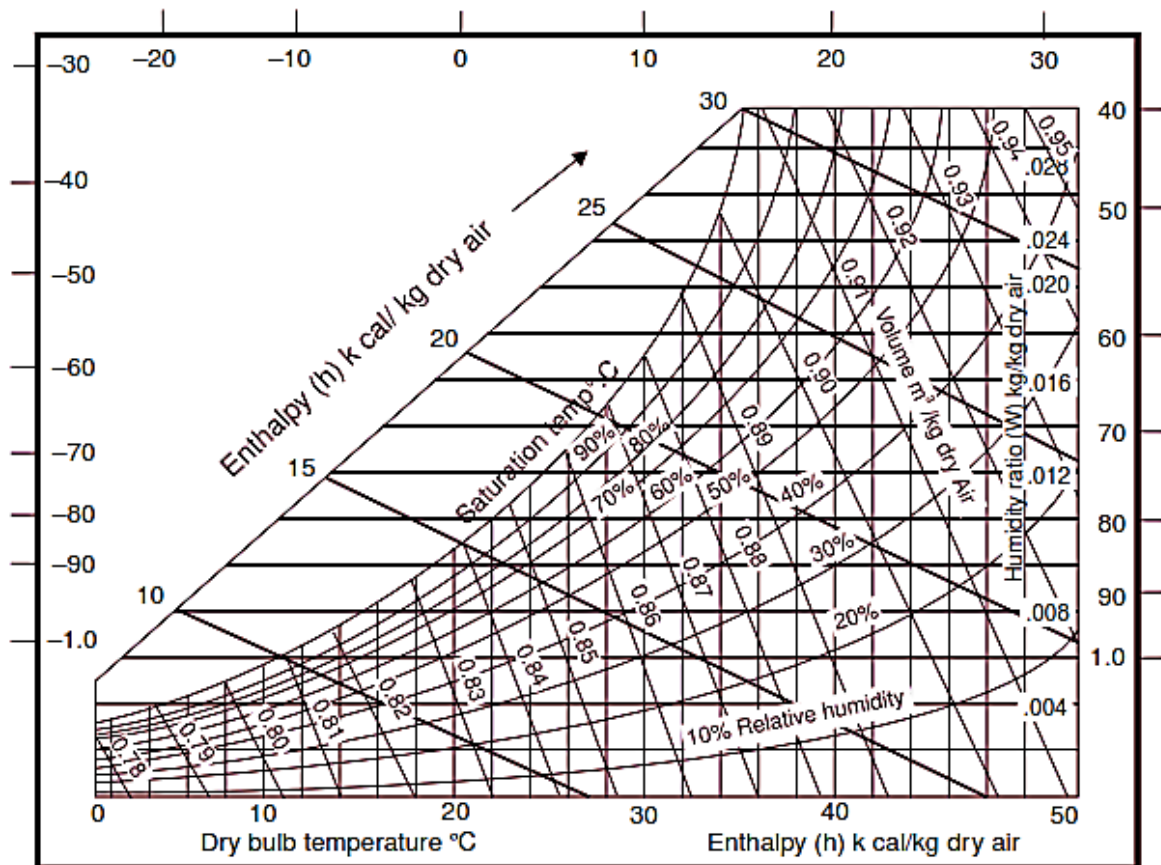


Fig b..Carrier chart.

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. C and D. 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 :

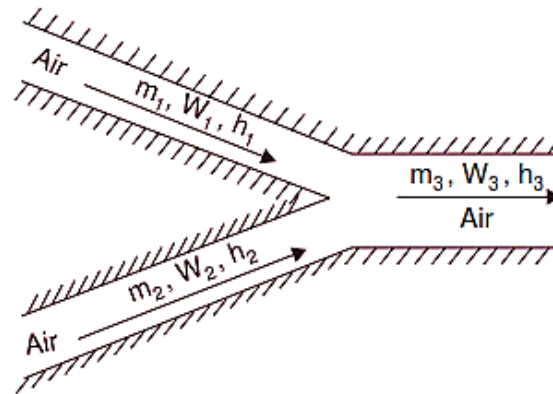


Fig. C. Mixing of air streams.

$$\begin{aligned} m_1 + m_2 &= m_3 \\ m_1 W_1 + m_2 W_2 &= m_3 W_3 \\ m_1 h_1 + m_2 h_2 &= m_3 h_3 \end{aligned}$$

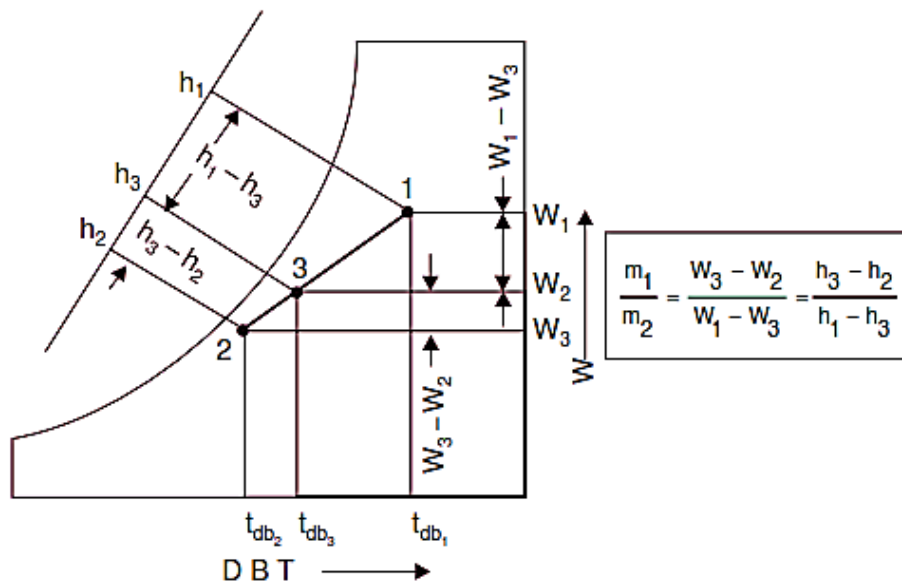


Fig D

Rearranging of last two equations gives the following:

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

Where,

m = Mass of dry air at particular state points

W= Specific humidity at particular state points

h = Enthalpy at particular state points

On the psychrometric chart, the specific humidity and enthalpy scales are linear, ignoring enthalpy deviations. Therefore, the final state 3 lies on a straight line connecting the initial states of the two streams before mixing, and the final state 3 divides this line into two parts that are in the same ratio as were the two masses of air before mixing. If the air quantities are known in volume instead of mass units, it is generally sufficiently accurate to units of m³ or m³/min. in the mixing equations. The inaccuracy introduced is due to the difference in specific volume at two initial states. This difference in densities is small for most of the comfort air conditioning problems.

Sensible Heating

When air passes over a dry surface which is at a temperature greater than its (air) dry bulb temperature, it undergoes sensible heating. Thus the heating can be achieved by passing the air over heating coil like electric resistance heating coils or steam coils. During such a process, the specific humidity remains constant but the dry bulb temperature rises and approaches that of the surface. The extent to which it approaches the mean effective surface temperature of the coil is conveniently expressed in terms of the equivalent **by-pass factor**.

The by-pass factor (BF) for the process is defined as the ratio of the difference between the mean surface temperature of the coil and leaving air temperature to the difference between the mean surface temperature and the entering air temperature.

Thus on Fig. E, air at temperature t_{db1} , passes over a heating coil with an average surface temperature t_{db3} and leaves at temperature t_{db2}

The by-pass factor is expressed as follows :

$$BF = \frac{t_{db3} - t_{db2}}{t_{db3} - t_{db1}}$$

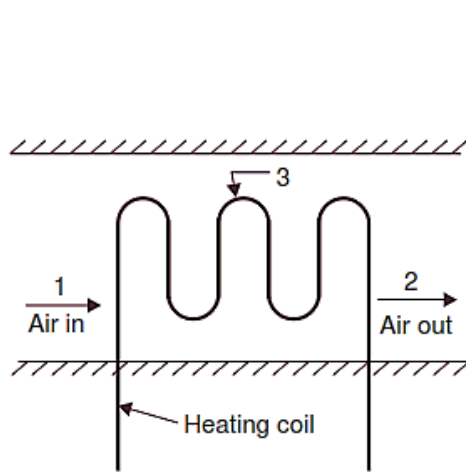


Fig.E. Sensible heating.

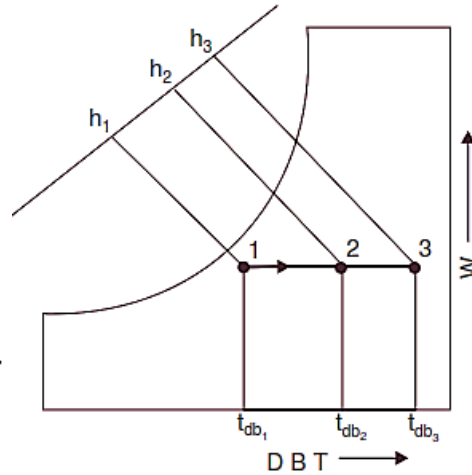


Fig.F

Or in terms of lengths on the chart (Fig. F) it is $\frac{\text{length } 2-3}{\text{length } 1-3}$

The value of the by-pass factor is a function of coil design and velocity. The heat added to the air can be obtained directly from the entering and leaving enthalpies ($h_2 - h_1$) or it can be obtained from the humid specific heat multiplied by the temperature difference ($t_{db2} - t_{db1}$).

In a complete air conditioning system the preheating and reheating of air are among the familiar examples of sensible heating.

Note. 'By-pass factor' can be considered to represent the fraction of air which does not come into contact with coil surface.

Sensible Cooling

Refer Fig. G. Air undergoes *sensible cooling* whenever it passes over a surface that is at a temperature less than the *dry bulb* temperature of the air but *greater than the dew point temperature*. Thus sensible cooling can be achieved by passing the air over cooling coil like *evaporating coil of the refrigeration cycle* or *secondary brine coil*. During the process, the *specific humidity remains constant* and *dry bulb temperature decreases*, approaching the mean

effective surface temperature. On a psychrometric chart the process will appear as a horizontal line 1–2 (Fig. H), where point 3 represents the effective surface temperature. For this process:

$$\text{By-pass factor BF} = \frac{t_{db_2} - t_{db_3}}{t_{db_1} - t_{db_3}}$$

The heat removed from air can be obtained from the enthalpy difference ($h_1 - h_2$) or from humid specific heat multiplied by the temperature difference ($t_{db_1} - t_{db_2}$).

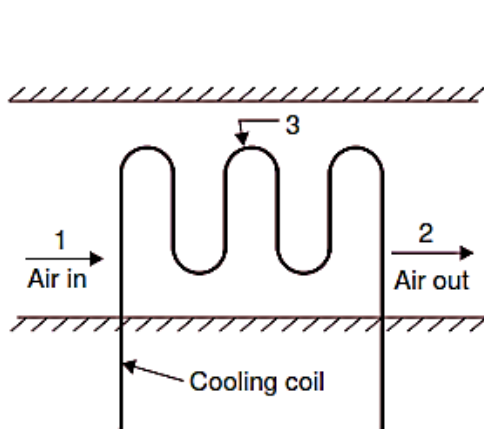


Fig.G. Sensible cooling.

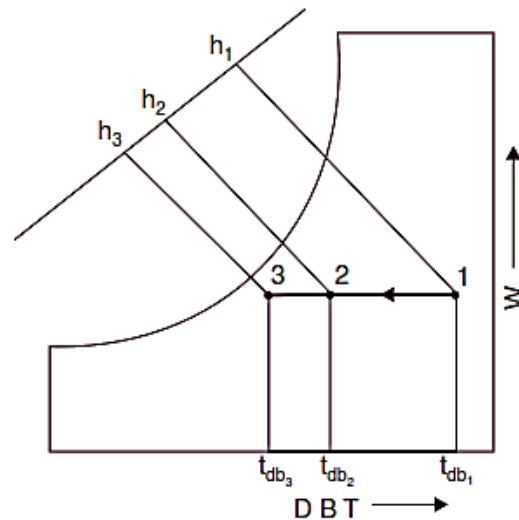


Fig.H

Cooling and Dehumidification

Refer Fig. I. Whenever air is made to pass over a surface or through a spray of water that is at a temperature less than the dew point temperature of the air, condensation of some of the water vapour in air will occur simultaneously with the sensible cooling process. Any air that comes into sufficient contact with the cooling surface will be reduced in temperature to the mean surface temperature along a path such as 1-2-3 in Fig. I, with condensation and therefore dehumidification occurring between points 2 and 3. The air that does not contact the surface will be finally cooled by mixing with the portion that did, and the final state point will be somewhere on the straight line connecting points 1 and 3. The actual path of air during the process will not be a straight line shown but will be something similarly to the curved dashed line 1-4. It will result from a continuous mixing of air which is contacting a particular part of the coil and air which is bypassing it. It is convenient, however, to analyse the problem with the

straight line shown, and to assume that the final air state results from the mixing of air that has completely by passed the coil with air that has been cooled to the mean effective surface temperature. If there is enough contact between air and surface for all the air to come to the mean surface temperature, the process is one of zero by pass. In any practical system, complete saturation is not obtained and final state will be a point such as 4 in Fig. I with an equivalent by pass factor equal to $\frac{\text{length } 3-4}{\text{length } 3-1}$ or processes involving condensation, the effective surface temperature, e.g. t_{db3} in Fig. I is called '**apparatus dew point**' (ADP). The final state point of air passing through a cooling and dehumidifying apparatus is in effect a mixture condition that results from mixing the fraction of the air, which is equal to the equivalent by-pass factor (BF) and is at initial state point and the remaining fraction which is equal to one minus by pass factor ($1-BF$) and is saturated at the apparatus dew point (ADP).

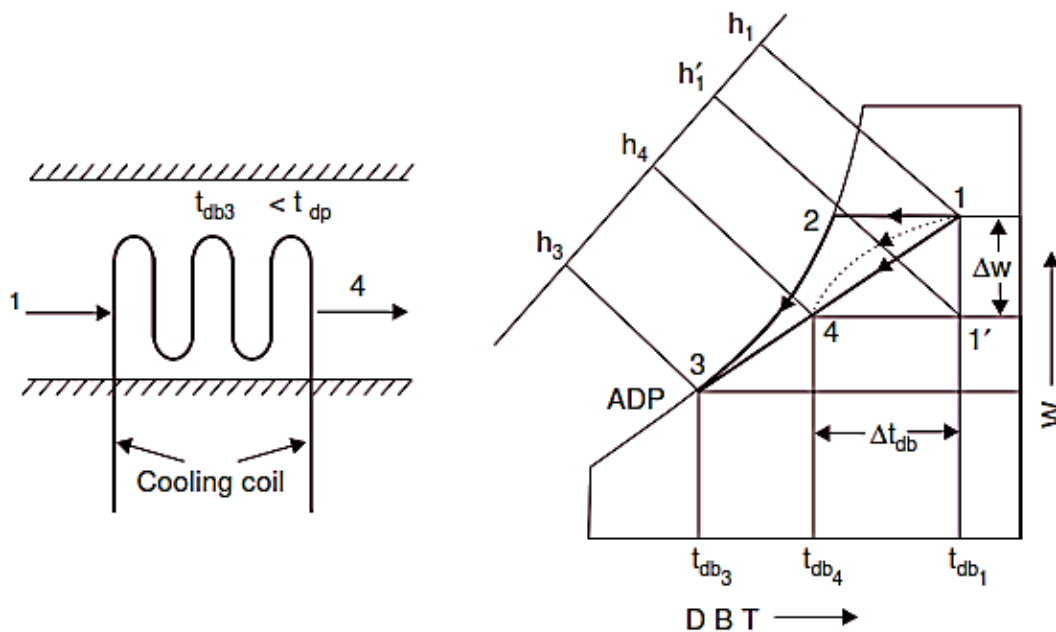


Fig. I Cooling and dehumidification

Total heat removed from the air is given by

$$Q_t = h_1 - h_4 = (h_1 - h_1') + (h_1' - h_4) \\ = Q_L + Q_S$$

where, W_L = Latent heat removed $(h_1 - h_1')$, and
 Q_S = Sensible heat removed $(h_1' - h_4)$

The ratio $\frac{Q_S}{Q_L}$ is called sensible heat factor (SHF) Or
sensible heat ratio (SHR)

$$\therefore \text{SHF} = \frac{Q_S}{Q_L + Q_S}$$

The ratio fixes the slope of the line 1—4 on the psychrometric chart. Sensible heat factorslope lines are given on the psychrometric chart. If the initial condition and SHF are known for the given process, then the process line can be drawn through the given initial condition at a slope given by SHF on the psychrometric chart.

The capacity of the cooling coil in tonnes of refrigeration is given by,

$$\text{Capacity in TR} = \frac{m_a(h_1 - h_4) \times 60}{14000},$$

where m_a = mass of air, kg/min and h = enthalpy in kJ/kg of air.

Cooling and Humidification

If unsaturated air is passed through a spray of continuously recirculated water, the specific humidity will increase while the dry bulb temperature decreases. This is the process of **adiabatic saturation or evaporative cooling**. This process is one of constant adiabatic-saturation temperature and for all practical purposes, one of constant wet bulb temperature. The process is illustrated as path 1-2 on Fig. J, with wet bulb temperature of air being that of point 3, which is also equilibrium temperature of the recirculated water if there is sufficient contact between air and spray, the air will leave at a condition very close to that of point 3. The concept of equivalent by pass can be applied to this process but another term is more used to describe the performance of a humidifying apparatus. It is the ‘**saturating**’ or ‘**humidifying efficiency**’ which is defined as the ratio of dry-bulb temperature decrease to the entering wet bulb depression usually expressed as percentage. Thus, from Fig. J, the saturating efficiency is :

$$\% \eta_{sat} = \left(\frac{t_{db1} - t_{db2}}{t_{db1} - t_{db3}} \right) \times 100$$

As a fraction, it is equal to one minus the by pass factor for the process. This adiabatic process, for all practical purposes, is line of constant enthalpy. The moisture added can be obtained from the increase in specific humidity.

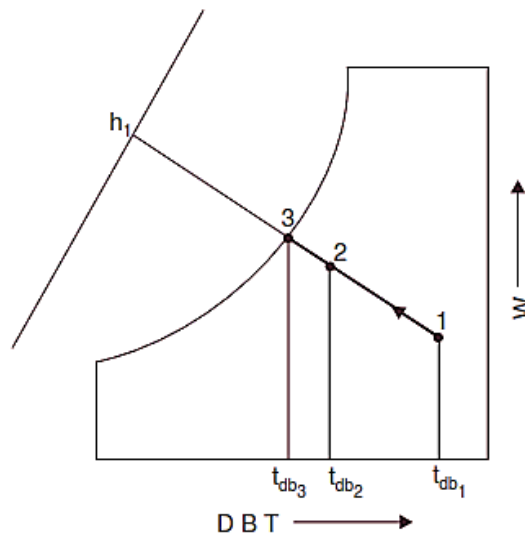


Fig J Cooling and humidification.

Heating and Dehumidification

If air is passed over a solid absorbent surface or through a liquid absorbent spray simultaneous heating and dehumidification is accompanied. In either case the dehumidification results from adsorbent or absorbent having a lower water vapour pressure than air. Moisture is condensed out of the air, and consequently the latent heat of condensation is liberated, causing sensible heating of air. If these were the only energies involved, the process would be the inverse of the adiabatic saturation process. There is, however, an additional energy absorbed or liberated by the active material, termed the heat of adsorption or absorption. For the solid adsorbents used commercially, such as silica gel or activated alumina, and for the more common liquid absorbents, such as solutions of organic salts or inorganic compounds like ethylene glycol, heat is involved and results in additional sensible heating. Thus the path lies above a constant wet bulb line on the psychrometric chart such as path 1-2 in Fig. K

Heating and Humidification

If air is passed through a humidifier which has heated water sprays instead of simply recirculated spray, the air is humidified and may be heated, cooled or unchanged in temperature. In such a process the air increases in specific humidity and the enthalpy, and the dry bulb temperature will increase or decrease according to the initial temperature of the air and that of the spray. If sufficient water is supplied relative to the mass flow of air, the air will approach saturation at water temperature. Examples of such processes are shown on Fig. L

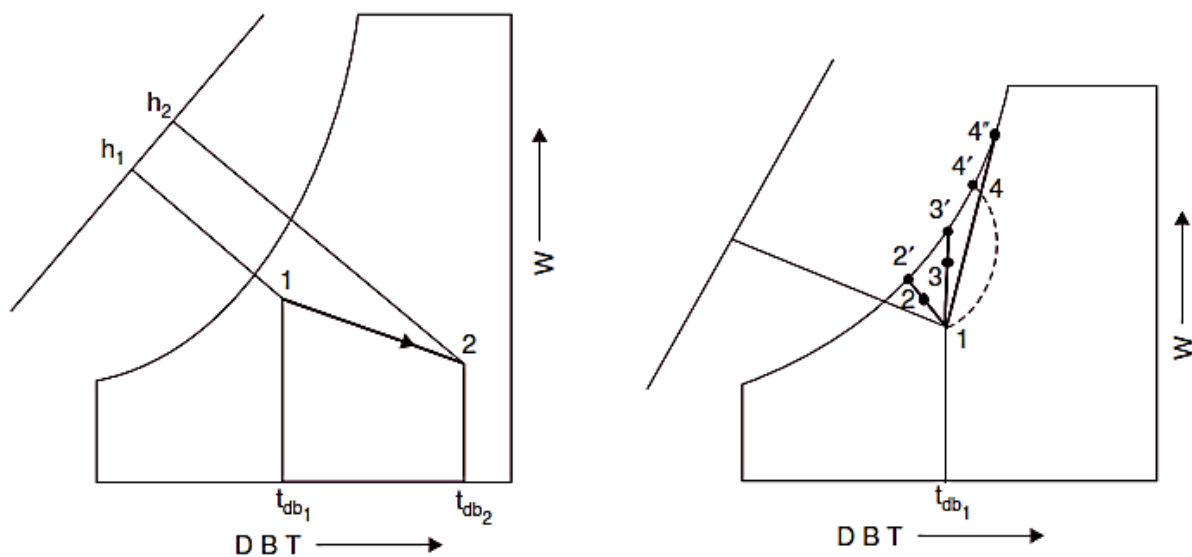


Fig. K. Heating and dehumidification. Fig. L. Heating and humidification.

Process 1-2 : It denotes the cases in which the temperature of the heated spray water is less than the air DBT

Process 1-3 : It denotes the cases in which the temperature is equal to the air DBT.

Process 1-4 : It denotes the cases in which a spray temperature is greater than air DBT.

As in the case of adiabatic saturation, the degree to which the process approaches saturation can be expressed in terms of the by-pass factor or a saturating efficiency.

If the water rate relative to the air quantity is smaller, the water temperature will drop significantly during the process. The resultant process will be a curved line such as the dashed 1-4 where 4 represents the leaving water temperature.

Note. It is possible to accomplish heating and humidification by evaporation from an open pan of heated water, or by direct injection of heated water or steam. The latter is more common. The process line for it is of little value because the process is essentially an instantaneous mixing of steam and the air. The final state point of the air can be found, however by making a humidity and enthalpy balance for the process. The solution of such a problem usually involves cut-and-try procedure.

1. A 5 tonne refrigerator plant uses RR as refrigerant. It enters the compressor at -5°C as saturated vapour. Condensation 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. [Nov 2002]

T($^{\circ}\text{C}$)	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

$$T_1 = -5^{\circ}\text{C} = 268\text{K}$$

$$h_{g1} = 249.3\text{kJ/Kg}, S_{g1} = 1.557\text{KJ/Kgk}$$

At point 2

$$T_2 = 32^{\circ}\text{C} = 305\text{K}, h_{f2} = 130.5\text{KJ/Kg}, h_{g2} = 264.5, S_{g2} = 1.542\text{KJ/KgK}$$

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

$$\text{At } -5^{\circ}\text{C}, \text{ i.e at } 268\text{K}$$

$$h_{g1} = 249.3\text{KJ/Kg} = h_g$$

$$h_1 = 249.3\text{KJ/Kg}$$

$$\text{At } 32^{\circ}\text{C}, \text{ i.e at } 305\text{K}$$

$$h_{g2}=264.5\text{KJ/Kg}=h_2'$$

$$h_2'=264.5\text{KJ/Kg}$$

Entropy is constant during the compression process so,

$$S_1=S_2$$

From T- S diagram

At point (1) dry,

$$S_1=S_g \text{ at } -5^\circ\text{C}$$

$$S_1=S_{g1}=1.557\text{KJ/Kgk}$$

$$S_1=S_2=1.557\text{KJ/Kgk}$$

At point (2) (super heated)

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

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

$$S_2'=S_g \text{ at } 32^\circ\text{C}.$$

$$S_{g2}=1.542=S_2$$

$$S_2'=1.542\text{KJ/Kg k}$$

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

$$T_2=307.44\text{k}$$

For super heated vapour the enthalpy is

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

$$h_2= 264.5 + 1.884 (307.44-305)$$

$$h_2= 269.1 \text{ KJ/Kg}$$

From P-h diagram, we know that,

$$h_3= h_4$$

$$h_3=h_f \text{ at } 32^\circ\text{C}$$

$$h_{f2}=130.5= h_f$$

We Know that,

$$\text{COP} = \text{Refrigeration effect} / \text{Work done} = (h_1 - h_4) / (h_2 - h_1)$$

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

$$\text{Refrigeration effect} = m \times (h_1 - h_4)$$

$$m = (2 \times 210) / (249.3 - 130.5)$$

$$m = 8.84 \text{ Kg/min}$$

$$\text{Work done} = \text{Refrigeration effect} / \text{cop}$$

$$= (2 \times 210) / 6 = 175 \text{ KJ/min}$$

$$\text{Power} = 2.92 \text{ kw.}$$

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 12140 KJ/hr. The properties of refrigerant are given

T($^\circ\text{C}$)	sensible Heat (h_f)	Latent heat (h_{fg}) KJ/Kgk	Entropy of liquid (KJ/Kgk)	Entropy of vapour S_g (KJ/Kgk)
-7	-29.3	1297.9	-0.109	4.748
27	1117.23	1172.3	0.427	4.333

in table. [May 2003]

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)

$$T_1 = -7^\circ\text{C} = 266\text{k}, \quad h_{fg1} = 1297.9 \text{ KJ/Kg}, \quad S_{f1} = -0.109 \text{ KJ/KgK}$$

$$h_{f1} = -29.3 \text{ KJ/Kg}, \quad S_{fg1} = 4.478 \text{ KJ/KgK}$$

At point (2)

$$T_2 = 27^\circ\text{C} = 300\text{k}, \quad h_{fg2} = 1172.3 \text{ KJ/Kg}, \quad S_{f2} = 0.427 \text{ KJ/KgK}$$

$$h_{f2}=117.23 \text{ KJ/Kg}, \quad S_{fg2}=4.333\text{KJ/KgK}$$

$$\text{We point } S_1=S_2$$

At point (1) (wet)

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

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

$$S_1= -0.109+x_1(4.857)$$

At point (2) (dry)

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

$$S_2=4.33\text{KJ/KgK}$$

$$S_1=S_2 \quad \text{So, } 4.33= -0.109+x_1(4.857)$$

Dryness fraction

$$x_1=0.913$$

At point (1) (wet)

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

$$h_1=-29.3+0.913 \times 1297.3$$

$$h_1=1156.3 \text{ KJ/Kg}$$

At point (2) (dry)

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

$$h_2= 117.23+1172.3$$

$$h_2= 1289.53 \text{ KJ/Kg}$$

From P-h diagram

$$h_3=h_4$$

$$h_3=h_{f2}$$

$$h_3=1172.3 \text{ KJ/Kg}$$

$$h_4=117.23 \text{ KJ/Kg}$$

$$\text{COP} = (h_1-h_4) / (h_2-h_1) = (1156.3-117.23) / (1289.53-1156.3) = 7.7$$

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 $\eta_{\text{Isen}}=78\%$ the load is 3 tons and find COP, power, mass flow rate.

[May 2005]

Given data:

$$P_1 = 100\text{kpa}, \quad T_3 = 328\text{k}$$

$$T_1 = 277\text{k}, \quad P_4 = 100\text{kpa}$$

$$P_2 = 300\text{kpa}, \quad \eta_T = 78\%$$

$$\eta_c = 72\%$$

Solution:

process 1-2 Isentropic compression

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

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

$$T_2 = 379.14\text{k}$$

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

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

$$T_2 = 418.86\text{k}$$

Process 3-4 isentropic compression

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

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

$$T_4 = 239.64\text{k}$$

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

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

$$T_4' = 259.08 \text{ K}$$

$$\text{COP} = (T_1 - T_4') / (T_2' - T_1)$$

$$\text{COP} = (277 - 259.08) / (418.86 - 277) = 0.17$$

$$1 \text{ tonne} = 3.5 \text{ kW of heat}$$

$$3 \text{ tonne} = 3 \times 3.5 = 10.5 \text{ kW}$$

Energy balance.

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

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

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

$$\text{Mass of air, } m_a = 0.583 \text{ Kg/sec.}$$

$$\text{Power, } P = m_a \times C_{p_a} \times (T_2' - T_1)$$

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

$$= 83.12 \text{ kW}$$

1. An ammonia refrigerator process 20 tons 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. $\text{COP} = 70\%$ of theoretical COP. Determine (i) Rate of NH_3 circulation (ii) size of compressor, $N = 240 \text{ rpm}$, $L = D$, $\eta_{\text{vol}} = 80\%$. Take Latent heat of $I_{cc} = 335 \text{ kJ/Kg}$, $C_p = 2.8 \text{ kJ/Kg}$,

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

Sat. Temp($^\circ\text{C}$)	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 Ice per day at °C

$$T_3 = 20^\circ\text{C}$$

$$T_1 = -20^\circ\text{C}$$

$$T_3 = 50^\circ\text{C}$$

COP = 70% of theoretical cop

$$N = 240\text{rpm}$$

$$L = D$$

$$h_v = 80\%$$

Latent heat of Ice = 335 KJ/Kg

$$C_p = 2.8\text{KJ/KgK}$$

$$V_{s1} = 0.624\text{m}^3/\text{kg}$$

Solution:-

The refrigeration effect = $20 \times 3.5 = 77.55\text{kW}$

$$h_1 = 1419.05\text{KJ/Kg}$$

$$h_{g2} = 1461.58\text{KJ/Kg at } 20^\circ\text{C}$$

$$h_{f3} = 274.98\text{ KJ/Kg}$$

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

$$h_2 = 1461.58 + 2.8(50 - 20)$$

$$h_2 = 1545.58\text{ KJ/Kg}$$

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

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

$$= 9.04.$$